1. Introduction

Environmental and economic factors have impacted the advancements in fluid power systems over the last two decades. Gradual development has been made over the last years; however, no major advancement has led to a radical improvement on systems' efficiencies using the state-of-the-art technology. A number of modifications to load-sensing actuation have been proposed by researchers all over the world in order to minimize metering losses. The idea of adding a second pump to a traditional load-sensing system has been investigated to separate actuators according to their pressure and flow requirements, thus reducing losses\(^1\). Another approach is the decoupled or independent metering valve system. In such system, the inlet and outlet metering orifices are independently controlled through the use of additional valves per actuator. This type of system has been both researched\(^2\) \(^3\) and commercialized\(^4\). Further improvements have been demonstrated through the use of an intermediate pressure rail and a network of proportional valves\(^5\). It is important to note that the control for this particular system is complicated and that a large number of valves is required to realize every operating mode. All of the above mentioned technologies reduce metering losses in load-sensing systems but do not eliminate them.

Displacement-controlled (DC) actuation has been under investigation by the authors’ group since its conception in 1998 as a highly efficient alternative to its valve controlled counterpart. The major advantages of DC actuation include the complete elimination of losses due to resistive control and the recuperation of energy due to overriding loads. One obstacle for the introduction of DC actuation to the market is the increased machine production costs due to the one-pump-per-actuator requirement. To overcome this impediment, the authors’ research group propose the idea of pump switching. The idea consists on utilizing a distributing manifold comprising a set of on/off valves utilized to direct flow either from/to a hydraulic unit to/from a particular actuator. Then, the concept allows for the reduction of machine installed pump power for multi-actuator machines, thereby minimizing parasitic losses and production costs. In this paper, the challenges and implications, as well as the control strategies developed to realize this technology are outlined for a multi-actuator system. Furthermore, an extension of work previously proposed by the author’s research group is made by presenting a validation of the proposed control strategies on an excavator prototype. Measurement results show that the pump switching concept is attainable while maintaining the same basic DC concept and relatively simple actuator-level control algorithms.

**Keywords:** Displacement Control, Pump Switching, Efficient Hydraulics, Controls
shows a hydraulic circuit of a DC double acting hydraulic cylinder. A major obstacle to the introduction of this technology into the market is the increased production costs due to the one-pump-per-actuator requirement. To overcome this impediment, the authors’ research group investigated displacement-controlled systems with pump switching. Figure 2 shows a hydraulic circuit which illustrates such idea for two hydraulic actuators.

![Hydraulic Circuit](image)

Fig. 2 Displacement-controlled with pump switching

The main principle behind this approach involves the use of a set of on/off valves to direct flow from/to a hydraulic unit to/from the machine actuators in a sequential manner based on priority levels and commanded motion. As a way to illustrate the pump switching concept, the working hydraulics of a DC excavator with pump switching are studied. The proposed hydraulic architecture comprises 8 actuators and 4 variable displacement axial piston machines. The flow distributing manifold is discussed in detail for this application. Further, a study of the improvements on efficiency though the reduction of parasitic losses and operability by the implementation of advanced control strategies for this technology are subject of study.

The derived control strategies address the challenges posed by the proposed technology through knowledge of the system. For tracking purposes, a feedforward controller is derived based on the open-loop plant inverse. The first time implementation of the derived control strategies for displacement-controlled actuation with pump switching is presented for a test bench with two pumps and two actuators. The obtained results validate the developed algorithms achieving satisfactory tracking performance and allowing for seamless pump switching.

2. Proposed Excavator Hydraulics

A simplified hydraulic circuit of the proposed excavator architecture is shown in Fig. 3. The system comprises 8 actuators (swing, boom, arm, bucket, right and left track, offset and blade) and 4 swash plate-type axial piston machines. To realize DC pump switching, the flow distributing manifold encompasses 20 on/off valves. The proposed architecture has been designed with emphasis on maximizing machine operation while minimizing the number of installed hydraulic units and on/off valves. Due to the fact that the basic configuration of a DC circuit is maintained in all cases, typical DC actuation efficiency and operation is guaranteed.

For the presented excavator, the limitation on the minimum number of hydraulic units is imposed by the main machine operation – digging, where the swing, boom, arm and bucket are used simultaneously. It may be noted that for operations such as trench-digging the use of the tracks is required. These however, are sequentially operated with the rest of the actuators. Nevertheless, due to the availability of two hydraulic units while operating the tracks and the design of the distributing manifold, any combination of 2 actuators including the boom, arm and bucket is available to the operator.

An additional advantage to the implementation of the proposed manifold design is the possibility to combine flow from two hydraulic units into a single actuator (also known as flow summing). In some cases, hydraulic units in DC actuation need to be oversized to meet high flow requirements. Through the combination of flows from two or more hydraulic units into a single actuator, the hydraulic units’ can be sized according to a typical working cycle rather than to meet maximum performance. Such case is represented by the boom in Fig. 3, where flow from up to three hydraulic units may be combined and directed to this actuator via valves 5 and 8, 10 and 13 or 17 and 18. Implementing this solution will lead to higher overall system efficiencies since the hydraulic units will in average operate at higher displacements. The drawback of this operating mode is that more than one hydraulic unit will be utilized for a single actuator, resulting on a maximum of two additional operational actuators.

3. Pump Switching Challenges

3.1 Pump Dynamics

One challenge of pump switching is posed by the
architecture itself and the nature of DC actuation. Since DC actuation relies on hydraulic units which may operate in 4 quadrants (as shown in Fig. 4), these will play an important role on achieving seamless pump switching. In reference to Fig. 3, assume the operator commands are such that the bucket and boom are sequentially operated using flow from unit 2 and that the operator commands fast transitions between actuators (i.e. no time delay exists in the transition between the boom and bucket motion). If for both actuators the hydraulic unit operates as a pump in quadrant I, the transition between actuators will require less control effort. On the other hand, if the boom actuator is lowered while holding an overrunning load that forces the hydraulic unit to operate in motoring mode and sequentially the bucket actuator is commanded to retract, forcing the hydraulic unit to operate as a pump, a greater control effort would be required. This phenomenon is observed due to the hydraulic unit having to go over-center. Problematic scenarios would be expected only in cases where the pump dynamics are very slow. A study of the pump dynamic behavior is conducted later in this paper to determine a low threshold on the pump dynamics. It is worth mentioning that the dynamic behavior of the on/off valves will have an influence on achieving seamless pump switching. However, this influence will only be significant if the on/off valve is slow, leading to actuator transients.

Nonetheless, off-the-shelf valves have typically fast dynamics (most of them with spool shifting times between 40 and 100ms). For that reason, in this paper on/off valves are considered as elements with fast enough dynamic behavior. Also, rather than attempting to utilize these elements to control the actuator transitions (as it is done in some applications such as digital hydraulics), the control strategy synthesized in this paper focuses on the use of the hydraulic unit displacement to achieve seamless transitions.

3.2 Pilot-operated Check Valves
Another challenge for the implementation of pump switching is the existence of pilot-operated check valves in the basic architecture of DC actuation (as shown in Fig. 1). Ideally, when a hydraulic unit is not in use, its displacement as well as the pressure differential across its working ports should be zero. However, in some cases, improperly calibrated swash plate sensors or improper control of the unit displacement may lead to small increments on either of the hydraulic unit’s working port pressures. With a differential pressure greater than zero, one of the pilot-operated check valves will be forced open (as shown in Fig. 4). The problem...
arises when these pilot-operated check valves have large hysteresis.

Take for instance unit 4 in Fig. 3, assume that the pilot-operated check valve corresponding to the boom piston side (left side pilot-operated check on Fig. 3) is open due to the pressure on port U4p2 being higher than that in port U4p1. Then, opening valves 17 and 18 would allow some flow through the pilot-operated check valve before the load pressure on the opposite working port (U4p1) is able to close it. This would cause a noticeable bump in the boom motion especially on cases where the actuator is under a large load.

Similarly, take into account unit 3 in Fig. 3, and assume that the unit is utilized to retract the arm actuator through valves 11 and 12 thereby opening the pilot-operated check valve on the actuator piston side. Suddenly closing valves 11 and 12 and opening valves 17 and 18 would lead to the same scenario. Unfortunately, the only alternative to the pilot-operated check valves would be a shuttle or flushing valve; however, due to their application in DC hydraulics they will not solve this issue as the differential pressure would still force the valve spool to shift. This challenge is addressed on the controller development section later in this paper.

3.3 Flow Summing Transitions

A third challenge in the implementation of pump switching is the transition from flow summing to single unit actuation. Once again take into account the hydraulic circuit in Fig. 3. Assume that units 3 and 4 are providing flow to the boom actuator through valves 10, 13, 17 and 18. If the operator suddenly commands the arm to move, valves 10 and 13 must be closed to connect unit 3 to the arm actuator through valves 11 and 12. If the boom was in the middle of high velocity motion and unit 2 is not available to provide flow to the boom, the boom actuator will be left to operate with a single unit. This direct switch would then cause a severe step down on the boom actuator velocity. The remedy to the sudden change in speed can be handled through appropriate controls; however it is evident that a compromise on the actuator velocity must be made to maintain actuator availability regardless of the control strategy.

4. Pump Switching Test Bench

For the evaluation of the pump switching idea and control development, a test bench was built at the Maha fluid power research center. The simplified test bench hydraulic circuit is shown in Fig. 5.
retraction of the arm.

5. Plant Nonlinear Model

Each actuator dynamics may be considered individually. Focusing on the dynamics of the hydraulic motor, it can be shown that the dynamics of the inertia load can be described by Eq. (1).

\[
\dot{J}\phi = V_m/2\pi (p_1 - p_2) - b\phi - mg_l \sin \phi
\]  

where \( J \) is the load inertia, \( \phi \) and \( \dot{\phi} \) are the arm angular position and velocity respectively, \( V_m \) is the maximum actuator volumetric displacement, \( p_1 \) and \( p_2 \) are the pressures at each of the actuator working ports, \( b \) is the viscous friction coefficient, \( m \) is the mass of the load and \( l_{cg} \) is the arm center of gravity. The actuator dynamics can be expressed using the pressure buildup equation as

\[
\dot{p}_i = \frac{K}{V_i} \left( K_1 (p_2 - p_1) - K_2 \phi - \frac{V_m}{2\pi} \phi + nV_p \eta \beta + Q_{ak1} \right)
\]

\[
\dot{p}_2 = \frac{K}{V_2} \left( K_1 (p_1 - p_2) + K_2 \phi + \frac{V_m}{2\pi} \phi - nV_p \eta \beta + Q_{ak2} \right)
\]

(2)

where \( V_1 \) and \( V_2 \) are the fluid volumes in either one of the actuator’s working port, \( K_1 \) is the hydraulic motor coefficient of internal leakage due to pressure, \( K_2 \) is the hydraulic motor coefficient of internal leakage due to velocity, \( n \), \( V_p \) and \( \beta \) are the axial piston machine rotational speed, maximum volumetric displacement and normalized unit displacement and \( \eta \) is a constant unit efficiency independent of the machine speed, differential pressure and displacement was utilized. Finally, \( Q_{ak1} \) and \( Q_{ak2} \) are the pilot operated check valves flows given by

\[
Q_{ak1} = \alpha_0 \pi d_{po} \tanh \left( \frac{p_0 - p_1}{2p_l} \right) \sqrt{\frac{2|p_0 - p_1|}{\rho}} y_1
\]

\[
Q_{ak2} = \alpha_0 \pi d_{po} \tanh \left( \frac{p_0 - p_2}{2p_l} \right) \sqrt{\frac{2|p_0 - p_2|}{\rho}} y_2
\]

(3)

where \( \alpha_0 \) is the valve discharge coefficient, \( d_{po} \) is the pilot operated check valve spool diameter, \( p_0 \) is the low pressure system setting and \( y_1 \) and \( y_2 \) are the pilot operated check valve spool displacements which can be calculated from the spool force balance. For simplicity, define the expression in Eq. (4).

\[
Q_{po1} = \alpha_0 \pi d_{po} \tanh \left( \frac{p_0 - p_1}{2p_l} \right) \sqrt{\frac{2|p_0 - p_1|}{\rho}} y_1
\]

\[
Q_{po2} = \alpha_0 \pi d_{po} \tanh \left( \frac{p_0 - p_2}{2p_l} \right) \sqrt{\frac{2|p_0 - p_2|}{\rho}} y_2
\]

In this study, the pilot operated check valves spool dynamics are neglected due to their fast nature. Spool stroke limits are set to prevent oscillations in the low pressure system and avoid cavitation. Different from the pilot-operated valves, the dynamics of the axial piston machine swash plate may be modelled as a first order system. Then, the system state space is expressed in Eq. (5). Where \( \tau_p \) is the swash plate time constant, \( K_\beta \) is the swash plate dynamics gain and \( u \) represents the normalized unit displacement command. A similar analysis has been conducted for the system including the linear actuator.

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{1}{T} \left( V_m (x_3 - x_4) - b x_2 - mg_l \sin x_3 \right) \\
\dot{x}_3 &= \frac{K}{V_1} \left( K_1 (x_4 - x_3) - K_2 x_3 - \frac{V_m}{2\pi} x_2 + nV_p \eta \beta + y_1 Q_{po1} \right) \\
\dot{x}_4 &= \frac{K}{V_2} \left( K_1 (x_3 - x_4) + K_2 x_3 + \frac{V_m}{2\pi} x_2 - nV_p \eta \beta + y_2 Q_{po2} \right) \\
\dot{x}_5 &= -\frac{1}{\tau_p} x_5 + \frac{K}{\tau_p} u
\end{align*}
\]  

(5)

6. Reduced Order Model

The synthesis of the linear controller in this study is based on a reduced order single-input and single-output (SISO) model of the system derived from Eq. (5). A simple linearization of the load term is not possible since the trigonometric term \( \sin(x_1) \) is utilized for almost the entire range of motion. In order to proceed with the linear control design, this term must be neglected. Then, the load term is treated as a disturbance. Also for simplification purposes, the hydraulic unit swash plate dynamics have been neglected. Even though this last equation has been neglected, the input to the new model, \( u \), is still the normalized hydraulic unit displacement. In addition, the terms \( K/V_1 \) and \( K/V_2 \) are replaced by \( 1/C_{11} \) where \( C_{11} \) is a constant hydraulic capacitance. The four quadrant operation shown in Fig. 4 was taken into account to derive a linear model for each quadrant. Modeling the pressure differential across the actuator rather than each port’s pressure and considering only the actuator velocity reduces the model to a second order system. Finally, introducing the pressure scale factor \( S_c \) to minimize numerical errors and defining the new state
variables $x = [z_1, z_2] = [\phi, \Delta p]$ yields Eq. (6) through Eq. (9).

**Quadrant I ($\Delta p > 0$)**

$$\dot{z}_1 = \frac{1}{J} \left( \frac{V_m}{2\pi} S \Delta p - b z_1 \right)$$

$$\dot{z}_2 = \frac{1}{S C_H} \left( n V_p \eta u - \frac{V_m}{2\pi} z_1 - K_L S z_2 - K_p z_1 \right)$$

**Quadrant II ($\Delta p < 0$)**

$$\dot{z}_1 = \frac{1}{J} \left( \frac{V_m}{2\pi} S \Delta p - b z_1 \right)$$

$$\dot{z}_2 = \frac{1}{S C_H} \left( -n V_p \eta u - \frac{V_m}{2\pi} z_1 - K_L S z_2 - K_p z_1 \right)$$

**Quadrant III ($\Delta p < 0$)**

$$\dot{z}_1 = \frac{1}{J} \left( \frac{V_m}{2\pi} S \Delta p - b z_1 \right)$$

$$\dot{z}_2 = \frac{1}{S C_H} \left( n V_p \eta u - \frac{V_m}{2\pi} z_1 - K_L S z_2 - K_p z_1 \right)$$

**Quadrant IV ($\Delta p > 0$)**

$$\dot{z}_1 = \frac{1}{J} \left( \frac{V_m}{2\pi} S \Delta p - b z_1 \right)$$

$$\dot{z}_2 = \frac{1}{S C_H} \left( -n V_p \eta u - \frac{V_m}{2\pi} z_1 - K_L S z_2 - K_p z_1 \right)$$

7. Controller Design

The feedforward controller formulated in this study is based on the plant’s inverse. For illustration purposes, the computation of the open-loop transfer function for quadrant I leads to

$$G_{p,qI} = \frac{V_m n V_p \eta / 2\pi J C_H}{s^2 + K_L J + C_H b s + 4\pi^2 b K_i + V_m^2}$$

(10)

It is then evident that the plant inverse is improper. To obtain a proper or bi-proper feedforward controller, additional terms, which will only have an effect at frequencies higher than $\omega$, may be added to the denominator as

$$G_{FF,qI} = \frac{s^3 + K_L J + C_H b s + 4\pi^2 b K_i + V_m^2}{C_H s^2 J / V_m n V_p \eta / 2\pi J C_H (s/\omega + 1)^2}$$

(11)

To address the challenges posed by the pilot-operated check valves, a small displacement offset was added to the operator commands. This offset allows for the pre-pressurization of the proper working port, forcing the desired pilot-operated check valve open and preventing unwanted actuator motion.

To address the challenge posed by the transition between flow summing and single unit actuation, the following strategy was utilized: if the feedforward actuator velocity command exceeds 50% of the maximum possible actuator velocity, a second unit is gradually integrated to accelerate the actuator. Then, the worst case scenario is presented when the actuator command is 100% (two units are utilized at full displacement) and one unit is required by another actuator. Then, the only solution to this challenge is to slowly ramp down one of the units’ displacement command before switching it to a different actuator. The difficulty is that this slow down must be fast enough to prevent a delay on the other actuator motion but slow enough to smooth the actuator velocity transition.

8. Measurement Results on Test Bench

The first set of measurements presented show the influence of the hydraulic unit dynamics on the actuator dynamics on an especially challenging scenario: when a pump switch requires the unit to move over-center. It is
important to note that the hydraulic unit control valve will dominate the hydraulic unit dynamics. The hydraulic units on the test bench have been equipped with two Parker D1FH proportional directional valves with a maximum flow rate of 20 l/min and frequency response of 100 Hz at 5% spool displacement. For these measurements, different rate limits were utilized to slow down the unit response.

The synthesized feedforward control was deactivated and a rapid switch between the hydraulic motor and the hydraulic cylinder was commanded. The commanded motion corresponds to a positive unit displacement command when the motor is moving and a negative unit displacement command when the cylinder is moving. In addition, the secondary hydraulic unit was utilized to hold the hydraulic motor position while the primary hydraulic unit was commanded to actuate the hydraulic cylinder. The aforementioned operation corresponds to the worst case scenario where not only a large control effort is required, but two pumps are interchanged to provide flow to a single actuator. Three swash plate rates were analyzed. The first measurements were performed with an unlimited swash plate response, the second set with an imposed rate limit of 100%/s and the last set limited at 50%/s. To observe the effects of the hydraulic units’ dynamics on the actuator dynamics, the pressures at the switching instant of both the hydraulic motor and the hydraulic cylinder are shown in Fig. 7.

In reference to Fig. 7, it can be observed that the hydraulic units’ swash plate responses do not have a big influence on the actuator dynamics. Instead, proper management of the on/off valves is what allows for reduced undesired transients. For the case of the unlimited swash plate measurements, Fig. 7 a), it can be seen that the rapid displacement decrease leads to a rapid switch on the actuator motion. It may also be observed that an increase on the hydraulic motor pressure is caused by the secondary pump immediately holding the motor position after the switch. Even though both hydraulic units have equal swash plate responses, the limitation in this particular case is imposed by how aggressive the position control can be. In several instances, making the position command more aggressive would lead to undesired actuator behavior such as limit cycles. Also in reference to Fig. 7, it can be seen that b) and c) show a more desirable behavior in terms of actuator dynamics. However, c) displays a very large switching delay, which may be noticeable. To compare the results, the points at which the switch takes place may be taken into account. The interval between points a and b on Fig. 7 a) show a switching delay of 125ms. The interval between points c and d on Fig. 7 b) shows a switching delay of 250ms. Finally, the interval from e to f on Fig. 7 c) shows a switching delay of 500ms. It is evident that the 500ms delay is not acceptable. It can be concluded that for this particular case, a slower swash plate response may lead to a smoother transition between actuators. Nonetheless, the swash plate response must be fast enough to prevent noticeable interruptions on actuator motion. Regardless of the hydraulic units’ dynamics, this phenomenon will not be present if rather than holding the actuator position with the secondary hydraulic unit the actuator is held by the on/off valves in the distributing manifold. Finally, it may be observed that the pressure transients on the hydraulic cylinder are not affected by the swash plate response, having the similar magnitudes for all a), b) and c) cases.

The second set of measurements shows the ability of the outlined control strategy to overcome the challenges posed by incorrect pilot-operated check valve opening. In reference to Fig. 8, it may be observed that seamless pump switching is achieved repeatedly. Furthermore, Fig. 8 a) and c) show that no undesired transients are observed on the actuator position and/or pressure when switching. Finally, Fig. 8 b) shows the measured hydraulic unit displacement. In this case, the displacement offset which allows for smooth switching can be observed at the beginning of each switch. The magnitude of the offset for this particular test was 11% on top of the commanded displacement.
a) Measured actuator pressures

b) Measured actuator position

c) Measured pump displacement

d) Measured actuator angular position

Fig. 8 Measurement validation of the actuator switching under a load

9. Measurement Results on Excavator Prototype

The aforementioned control concepts were implemented and studied for the hydraulic architecture in Fig. 3. Measurements of pump switching for the boom actuator are presented under two different loads (a moderate load of 2000 N and a larger load of 4000 N) to show the ability of the control algorithms to prevent undesired actuator pressure and position transients as well as to show the effects of inappropriate control.

The intention of the presented measurements is to present the most challenging operation which is to allow the valve to open while the operator commands very small increments in motion. In all measurements presented in this section, an exaggerated miss-calibration of the swash plate was imposed to force the wrong pilot-operated check valve to open.

Fig. 10 Pump switching demonstration without control using the boom actuator under a moderate load

In reference to Fig. 10, it can be observed that the lack of proper control algorithms results in large pressure oscillations as well as transients in the actuator position. These two events are also evident through the audible noise coming from the structure as the pressure transients occur. Fig. 11 on the other hand shows how these undesired transients can be avoided using the control strategies outlined in this paper. When comparing Fig. 10 and Fig. 11, it can be seen that the key difference is the modified...
displacement command shown in the first plot of each figure. The additional displacement allows the correct pilot-operated check valve to open a short instance before the on/off valve is opened. This in turn results in more stable operation.

![Fig. 11 Pump switching demonstration with control using the boom actuator under a moderate load](image1)

![Fig. 12 Pump switching demonstration without control using the boom actuator under a large load](image2)

Similar to the measurements shown in Fig. 10 and Fig. 11, measurements were conducted under the same guidelines with a load twice as large. It can be observed that with the same control algorithms the elimination of undesired transients is possible.

![Fig. 13 Pump switching demonstration with control using the boom actuator under a large load](image3)

### 10. Conclusions

A new hydraulic architecture for multi-actuator systems utilizing displacement-controlled actuation and pump switching was proposed. The combination of displacement control and the pump switching architecture allows to reduce the total number of pumps required for those multi-actuator machines such as excavators. The presented pump switching control strategies demonstrate that the idea of pump switching in displacement-controlled systems represents a valuable alternative for highly efficient and cost-effective hydraulic systems. The concept was demonstrated both on a laboratory test bench and in an excavator prototype. The derived control solutions are simple, non-reliant on feedback thereby more reliable, and easy to implement. In addition, the work presented in this paper opens the door for future research in areas such as machine reliability due to control redundancy and increased energy efficiency through the reduction of parasitic losses.

### References

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