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ENGINEERING TOMORROW

Motion Control of a Hydraulic Cylinder with a Digital Displacement Pump-Motor

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Displacement controlled recuperative systems have shown promise at realising fuel savings and thus reducing CO_2 emissions in mobile machines. Operators of these machines expect the boom to lower faster than it raises so the hydraulic system must accommodate this. Two methods of achieving this are: 1) by throttling and motoring from the cylinder in parallel and 2) by operating the cylinder in differential mode. These mode transitions are typically made using proportional valves however Digital Displacement technology offers the possibility to implement these modes using simple switching valves, reducing the system complexity. This paper describes how this can be achieved and evaluates the energy recovery efficiency of both modes.

Keywords: *Efficient systems, digitalization, mode switching, energy recovery, differential cylinder*

Target audience: *Mobile Hydraulics, Excavators, System Design*

Construction machines such as excavators suffer from very poor hydraulic system efficiencies, typically around 30% from the pump shaft to the useful work done at the actuator. This is mainly due to the large throttling losses between the pump and actuators as well as their inability to recuperate energy. Efficiency improvements can therefore provide a significant contribution to worldwide CO_2 emission reductions [1].

Previous publications have demonstrated how Digital Displacement hydraulic machines can be used to drastically cut fuel consumption with incremental improvements to the system architecture. System Architecture (SA) 1 was implemented on a 16-tonne hydraulic excavator. SA1 consists of a simple pump swap. The conventional swashplate pump is swapped for a Digital Displacement Pump (DDP). The DDP offers increased pump efficiency and improved engine torque utilisation decreasing measured fuel consumption by between 18% and 21% [2].

The same excavator was then upgraded to SA2. The SA2 DDP consisted of eight independently controllable pumplets. A valve block known as the "elastic pump block" was used to route each pumplet to one of the excavator's two main services. These were connected to the original directional valves for controlling the actuators. This configuration allowed the pump capacity to be more closely matched to the flow requirement of each service, allowing some of the throttling losses in the directional valves to be reduced. This system resulted in up to 36% fuel reduction compared to the baseline vehicle [3].

The most advanced architecture, SA3, has been demonstrated on one axis on a static test rig. It uses direct displacement control by a Digital Displacement Pump-Motor (DDPM), which is also used to recover energy from the load. Directional valves are used to route the fluid to and from the DDPM. No throttling is done by the directional valves to control the motion of the actuator. DDPMs can be used to facilitate highly efficient energy recovery, with round trip efficiencies of up to 87% measured [4]. Simulation models show that SA3 systems can reduce fuel consumption by 58% on a 16-tonne excavator [5].

Displacement controlled system architectures have been covered extensively in the literature and have been nicely summarised by Axin [6]. These systems are characterised by the allocation of a separate pump or pump-outlet to each actuator therefore restrictive throttling valves are not required to reduce the pump pressure to the individual load pressure as is the case with a typical multi-actuator system. The motion of the actuator is controlled directly by the fluid flow supplied or received by the pump-motor. This removes the need for wasteful throttling to control the motion of the load.

One of the main problems faced with a displacement controlled recuperative hydraulic architecture is that in off highway machines the operator often expects the lowering velocity of the boom to exceed the lifting velocity. We can see this in the data in Figure 1 and Figure 2, recorded on a 16-tonne excavator for a dig and dump cycle. The flowrate was calcualted using the actuator velocities derived from string pot measurements recorded throughout the cycle. The pumping (lifting) flowrate doesn't exceed 200 l/min however the motoring (lowering) flowrate exceeds 300 l/min for short periods.

A study by Danfoss showed that the DDPM capacity would need to be increased by 210% over the baseline case to implement a full displacement controlled hydraulic system on a 16-tonne excavator which meets all operator flow demand and maximises energy recovery [5]. This requirement for a larger DDPM can make the displacement-controlled systems unfeasibly expensive for OEMs.

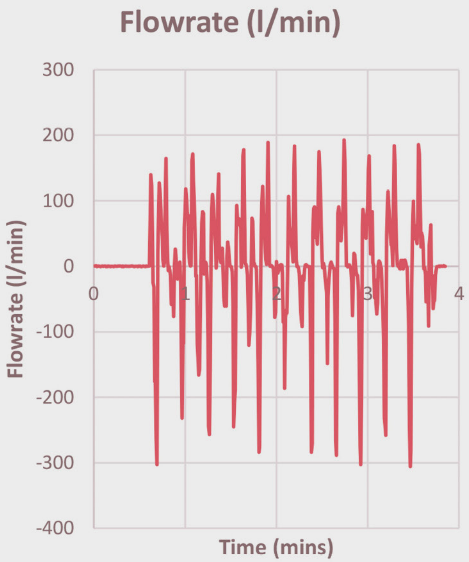

Figure 1: Flowrate duty cycle for boom cylinder on 16-tonne excavator.

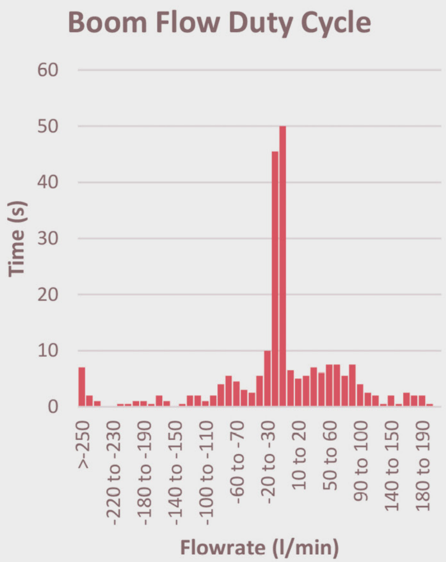

Figure 2: Histogram showing duration of time spent at different flowrates in duty cycle.

Analysis of the duty cycle data shows that motoring flowrates that exceed the maximum required pumping flowrate (200 l/min) are only required 6% of the time. It is hard to justify the increased capital cost of a larger hydraulic machine to achieve higher lowering flowrates for only 6% of the duty cycle. Recuperative systems should therefore be designed to meet the peak lowering speed demand without requiring costly excess pump-motor displacement. Various solutions to this problem have been proposed in the literature.

The first and simplest solution is to add a throttling valve in parallel with the pump-motor, which provides an additional flow path from the cylinder, allowing the lowering speed to be increased. Heybroek investigated this method [7]. The joystick command was mapped to a flow command which was met as a combination of both motoring and throttling through a proportional valve. The disadvantage with this method is that some otherwise recoverable energy is lost to throttling.

Another method of increasing the lowering velocity is to switch the cylinder into a differential mode of operation. In this mode, the cylinder and the rod are both connected to the pump. When lowering in this configuration the flow from the cylinder side is split between the rod side and pump line. This results in a reduced motoring flow requirement for a given ram velocity, when compared to the normal operating mode. The caveat is that the operating pressure is intensified by the ratio of the head to the rod area and the load carrying capacity of the cylinder is reduced. A challenge associated with this mode is switching between differential and normal circuit configurations. The pressure intensification causes large changes in the loading on the cylinder, which can result in jerky motion if not handled correctly. It is important to ensure a smooth motion to ensure safe load handling and operator comfort. Both Hitachi [8] and Volvo [9] have filed patents relating to pressure control of the cylinder when switching between modes to ensure a smooth and stable motion.

This paper presents how both throttling and motoring and differential mode can be realised for a displacement-controlled architecture with a DDPM. These operating modes are implemented using simple non-proportional switching valves only. The idea is that by having precise control of the DDPM flow output we can remove the need for proportional valves to conduct the mode switch in a smooth and controllable manner. The parallel throttling and motoring control mode is referred to as discrete throttling and the differential control mode is referred to as differential lowering. Discrete throttling offers an option for increasing the lowering speed even at high load pressure. Differential lowering offers a more efficient solution for lighter loads.

Control principles for each mode are given along with analysis of test data from an application scale test rig. Particular attention is paid to the smoothness of mode switching and the energy efficiency of each operating mode. Section 2 describes discrete throttling. Section 3 describes differential mode. Section 4 compares the two operating modes and discusses how they may be used in the context of higher-level system control.

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2. Discrete Throttling

2.1 System Overview

Discrete throttling is a means of lowering a cylinder by both throttling to the tank and motoring simultaneously. This mode increases the maximum lowering velocity over what would be available with only motoring. This allows the lowering speed to be increased even when the motoring displacement has been saturated.

Instead of throttling with a proportional valve, with which the opening area can be adjusted to control the flow, discrete throttling opens an on / off valve connected in series with an orifice. This gives a fixed flowrate through the valve for a given pressure drop across it. Use of a fixed orifice removes the complexity associated with a proportional valve.

To ensure a smooth increase in flow from the cylinder when the valve is opened, the displacement of the DDPM is reduced simultaneously as the valve is opened, keeping the net flow out of the cylinder the same just before and after the switching event. The fast response time of the DDPM allows this to work smoothly. The graph shown in Figure 3 helps to explain how the displacement command changes in sync with the throttling valve opening to give a smooth increase in lowering velocity.

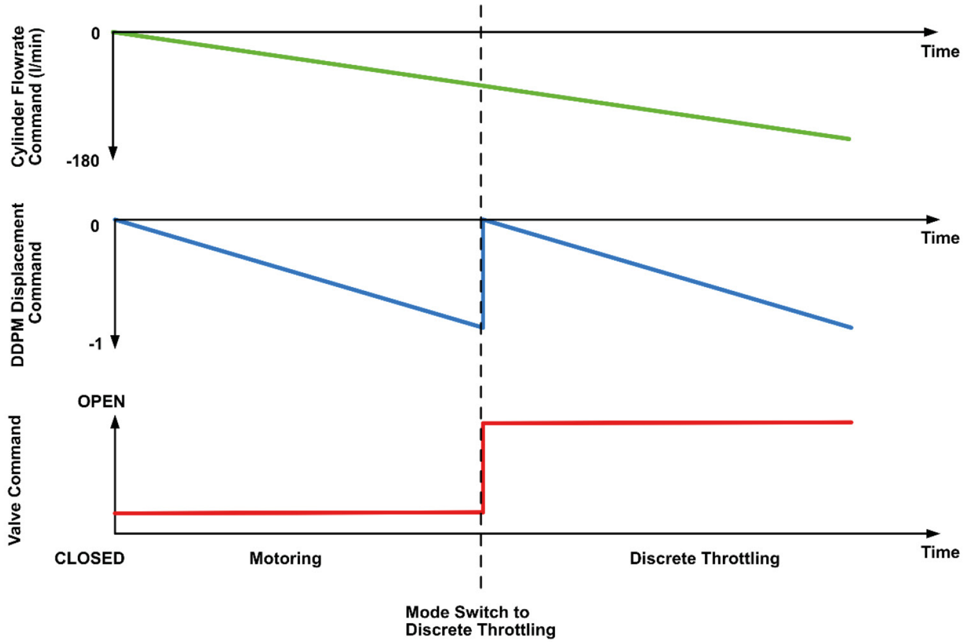

Figure 3: Discrete throttling scopes.

2.2 Control Strategy

Discrete throttling can be controlled via a displacement or flow command to the DDPM. The stick position controlled by the operator is mapped to the input command. In this test a maximum flow command of 250 l/min was specified. With a shaft speed of 1500 rpm and a maximum displacement of 96 cc/rev, the maximum lowering flowrate by motoring was about 134 l/min when accounting for compressibility, leakage, and loss of capacity due to the necessity to do a small part pump at the start of the DDPM's valve's motoring cycle. To reach flowrates greater than 134 l/min the throttling valve is required to provide an alternative flow path from the cylinder.

The lowering flowrate from the cylinder is the sum of the motoring flowrate and the throttling flowrate. Motoring can satisfy all flow demand up to the motoring capacity. When the motor displacement saturates it becomes necessary to open the throttling valve to tank, to increase the lowering speed of the cylinder. As the throttling valve is a fixed orifice the flowrate through the valve cannot be controlled and is load dependant. The motoring displacement must be reduced to prevent a step increase in lowering velocity when the throttling valve is opened. The system controller uses a lookup table of empirical data to determine the flowrate through the throttling valve based on the pressure drop across it, measured with pressure transducers on the rig. A DDPM loss model in the system controller is used to calculate the maximum flow at the current shaft speed and operating pressure. When the operator command exceeds what is possible with motoring only, based on the loss model calculation, the control logic will open the throttling valve and reduce the motoring displacement to ensure the flow is always matching the operator demand. The control logic also includes hysteresis to prevent rapid switching around the transition region. To account for the different transient dynamics between the throttling valve and the DDPM flow, the change in DDPM displacement command was delayed such that the throttling valve and DDPM reach their steady state values at the same time. This is discussed further in section 2.3.

2.3 Hydraulic System Architecture

The system schematic used for testing this mode is shown in Figure 4. The DDPM is connected to the cylinder side of a hydraulic ram with the throttling valve connected in series giving two flow paths for the oil to take out of the cylinder. The rod side was connected to a boost circuit running at 2.5 bar (this has been simplified to a tank symbol in Figure 4). In this test the throttling valve consisted of a directional control valve and a variable orifice with a screw adjustment. The directional valve was actuated by means of a hydraulic pilot system (simplified to a solenoid valve in Figure 4). The valves on the DDPM were controlled by a dedicated controller which was sent a reference displacement command from the PLUS+1 system controller over CANbus. The system controller used the operator stick command as an input and was also used to directly control the state of the throttling valve. Pressure sensors at the cylinder and tank were connected to the system controller for calculation of the pressure drop over the orifice. The pump pressure was also read by the system controller and used for the DDPM loss model calculation. This circuit was tested on the Motion Control Rig (MCR), which consists of a 10-tonne boom designed to mimic the power level and range of motion typically found on an off-highway machine. This test apparatus can be seen in Figure 5.

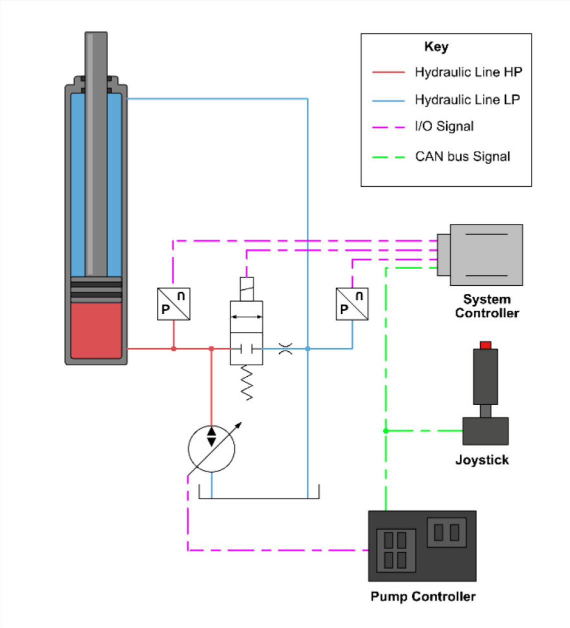

Figure 4: Hydraulic schematic for discrete throttling mode testing on MCR.

Figure 5: The Motion Control Rig in the Digital Displacement Lab.

2.4 Characterising the Pressure Drop Over the Orifice

The orifice size was set such that when the throttling valve was open, and cylinder was at maximum load pressure, the flow through the valve would not exceed the lowering velocity of the cylinder at full motoring displacement (the condition just before the throttling valve opens). This ensured that a smooth change in lowering speed could be achieved before and after the mode switch occurred. If the throttled flow exceeded the full motoring flowrate before the transition, the cylinder would speed up as soon as the throttle was opened, as even stopping the motoring flow completely could not compensate sufficiently for the net flow out of the cylinder.

The orifice pressure drop was characterised for various flow rates. The DDPM was used as the flow source for this test. Data obtained from these tests was used to create a lookup table used by the system controller to calculate the flow through the valve for a given pressure drop.

2.5 Optimising the Mode Switch Transition Timings

2.5.1 Theoretical Model of Delays Required

To determine the optimal control delays, the switching time of the DDPM and the directional valve were characterised. Measurements of the pilot pressure signal to the directional valve at the orifice were used to determine what the switching time of the valve was. It was possible to determine the time taken for the valve to start opening and to fully open from analysis of the pilot pressure trace. This gave a range of possible delays after which the flow path between the cylinder and the tank could be considered open. The switching time was found to take between 50 and 100ms.

The flow delay associated with the DDPM is made up of several components. First there is a 5ms delay associated with the transmission of the CAN message from the PLUS+1 system controller to the DDPM controller. Within the DDPM controller there is a 1ms iteration time. It takes half a shaft revolution for the flow output of the pump to fully change as previously committed cylinders must complete their current strokes. Therefore, the total DDPM delay time is given by:

$$DDPM_{Total} = CAN_{delay} + DDPM\_Controller_{delay} + \left(\frac{60}{\omega_{shaft}}\right)\left(\frac{1}{2}\right)$$

Therefore at 1500 RPM the total DDPM delay is 5ms (CAN) + 1ms (DDPM_Controller) + ((60/1500) * (1/2)) (response time of DDPM based on shaft speed) = 26ms. Therefore, the delay time between the throttle valve switching and DDPM switching should be: between Delaymin= 50ms – 26ms = 24ms and Delaymax = 100ms – 26 = 74ms. These delays have been visually presented in Figure 6. The optimal delay was expected to be somewhere between 26 and 74ms.

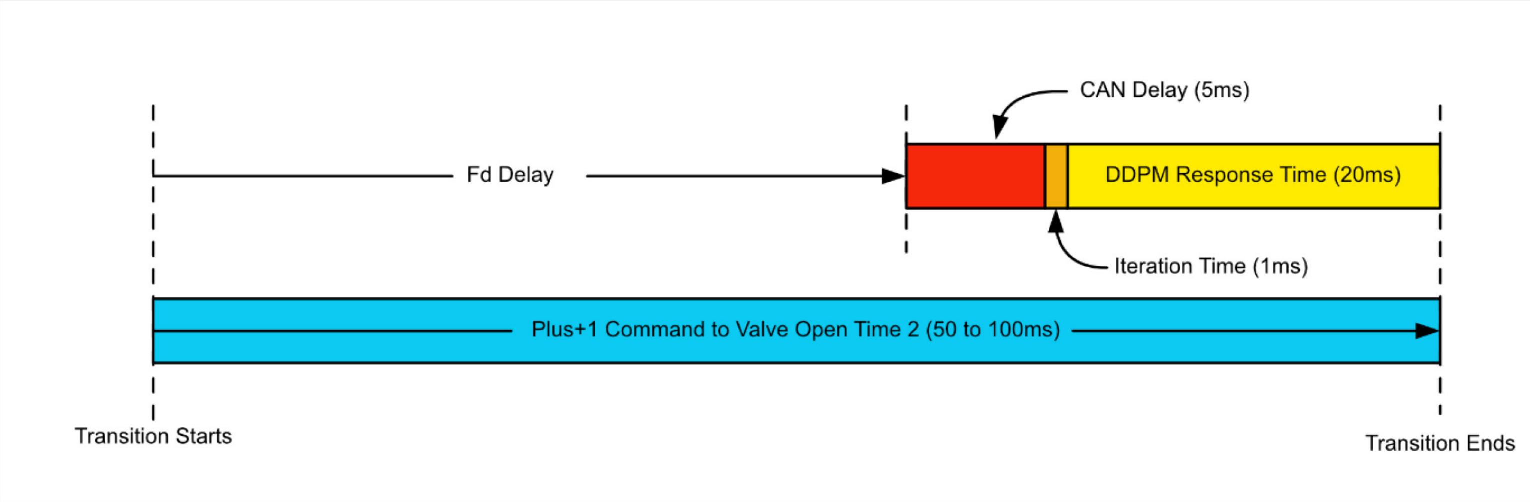

Figure 6: Diagram comparing the transition time for the valves with the transition time for the DDPM.

2.5.2 Assessing Transitions Smoothness

Two properties of the boom's motion were measured to assess the transition smoothness:

- **RMS Velocity**
- **RMS Acceleration**

ISO 20816-1 indicates the maximum allowable vibration for rotating and non-rotating machines and use the RMS velocity to describe the limit [10]. RMS velocity is used because low and high frequencies are equally weighted from an energy point of view and it is the vibration energy which should be minimised. Additionally ISO 2631 provides guidance on the allowable limits of human exposure to shock and vibration [11]. This standard specifies allowable vibration in terms of RMS acceleration. These standards specify the vibration limits for operators when they are at the control of the machine. The tests described in this paper were conducted in a lab environment with a specific control room and no possibility to gather realistic operator vibration data. The magnitudes of vibration were therefore not considered important – only the relative change in vibration which was used to understand how the control mode could be made smoother. A piezoelectric accelerometer was fitted to the MCR's boom. The signal from the accelerometer was integrated to give velocity and the RMS was calculated. The downward motion of the boom was controlled by the operator via a joystick. The lowering flowrate was gradually increased until about 180 l/min, well after the transition had occurred. The delay was implemented as a variable in a state flow chart in the system control logic. A delay of zero represented no delay when transitioning between states in this chart meaning that the pump displacement command and the valve switching command were issued at the same time. Measurements from the rig can be seen in Figure 7 to Figure 10.

2.5.3 From Discrete Throttling to Motoring Transition

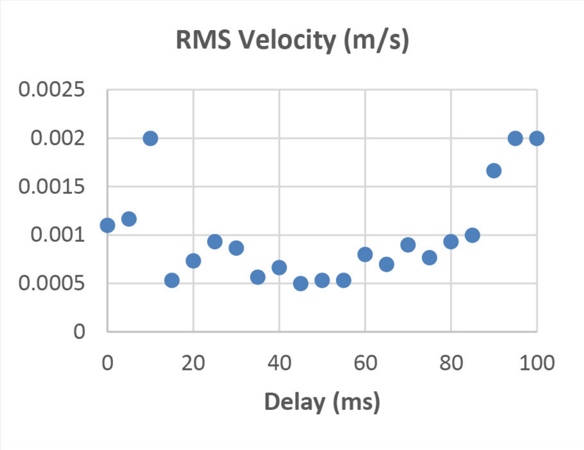

Figure 7: RMS velocity vs. displacement command delay for discrete throttling to motoring transition.

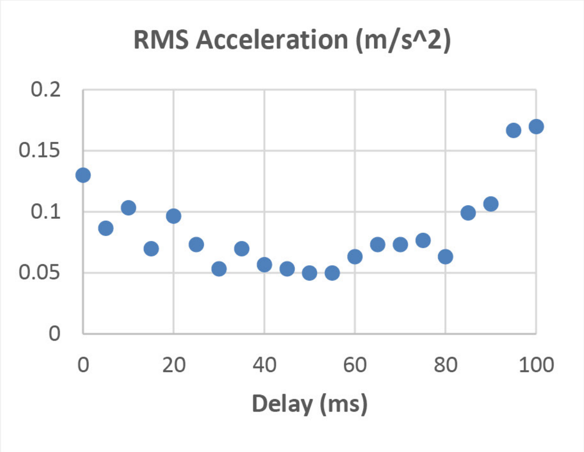

Figure 8: RMS acceleration vs. displacement command delay for discrete throttling to motoring transition.

This data shows a minimum in the vibration amplitude at a delay of 50ms. This is within the expected range of 26ms to 74ms, validating the timing calculation described in section 2.5.1.

2.5.4 From Motoring to Discrete Throttling Transition

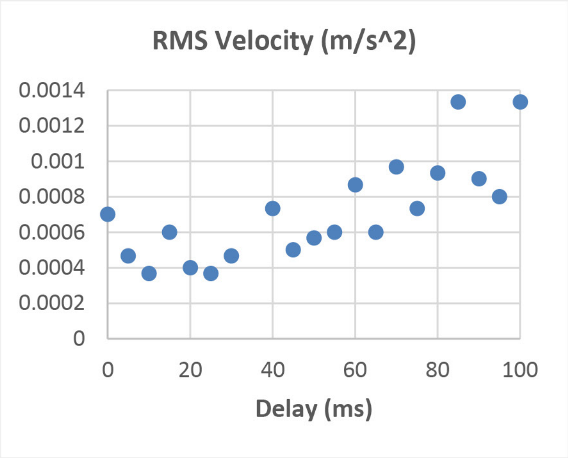

Figure 9: RMS velocity vs. displacement command delay for motoring to discrete throttling transition.

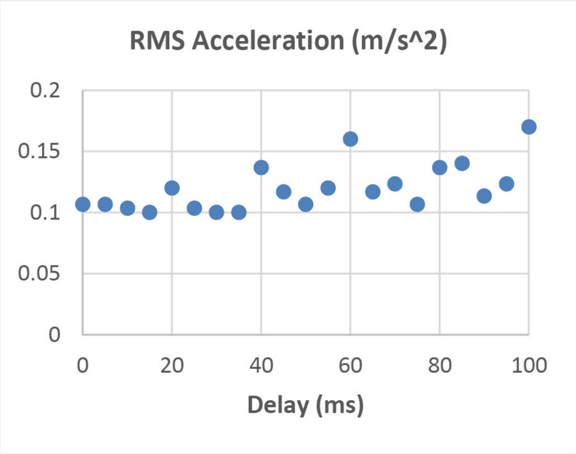

Figure 10: RMS acceleration vs. displacement command delay for motoring to discrete throttling transition.

The data for this transition is less clear than that presented for the previous transition. The general trend is that smaller delays yield a smoother transition, however the data doesn't give much more insight than this. A delay of 25ms was considered to correspond to the lowest RMS velocity and RMS acceleration.

2.6 Flow Accuracy

Comparisons were made between the flow command and the measured flowrate to establish how accurate the flow control is with this operating mode. The graph shows that the flow error is less than 5% across the range even when the operating mode changed to discrete throttling. Possible sources of this error are the parameters used in the DDPM loss model running in the system controller, the throttling valve characterisation data or the pressure sensor calibration. Furthermore, temperature change of the oil will affect the pressure drop characteristics and this could be investigated in the future.

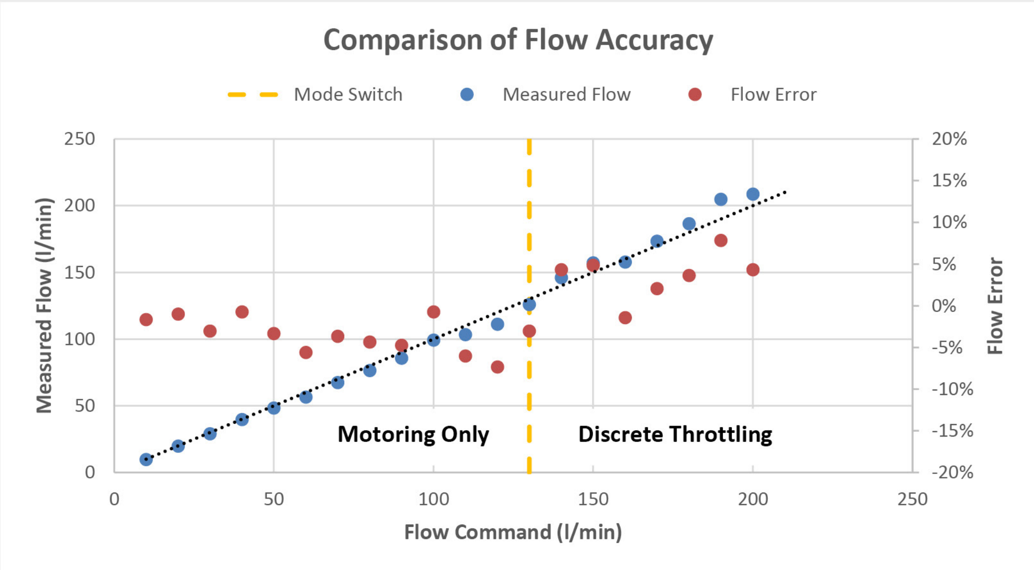

Figure 11: The flow error across the range of tested flowrates in discrete throttling mode.

2.7 Determining the Recovery Efficiency

The recovery efficiency was calculated using the ratio of the mechanical power at the DDPM's crankshaft to the fluid power of the oil leaving the cylinder for a constant lowering speed. Shaft torque and speed were measured using a torque transducer and speed encoder to determine the mechanical power. The fluid power was calculated by taking the product of the cylinder pressure and flow. The steady state velocity of the cylinder was calculated based on string pot measurements. The cylinder pressure was measured with a high precision pressure transducer.

To characterise the recovery efficiency at the chosen load pressure, the boom was lowered at a range of flowrates (from 10 l/min to 180 l/min) and the results can be seen in Figure 12. The dashed line on the graph shows the point at which the discrete throttling valve opens.

3. Differential Lowering

3.1 System Overview

Differential lowering is a well-known control strategy in which the hydraulic cylinder is switched into a differential mode of operation to reduce the motoring displacement required for a given lowering velocity. Directional valves are used to connect the cylinder and rod side of the actuator together. As the directional valves are switched the motoring displacement of the DDPM must be reduced to obtain a smooth increase in lowering velocity. Due to the pressure intensification that occurs in differential operation this mode is only suitable for lighter loads as the maximum system pressure is fixed by pressure relief valves.

3.2 Control Strategy

The operator controls the motion of the actuator via a joystick command. The joystick enables proportional control of the velocity of the actuator. The requested lowering velocity, and therefore flowrate, is calculated in the system controller based on the joystick position. The controller will calculate the motoring displacement needed by the DDPM to satisfy the operator command. In doing this it may decide that differential mode should and could be used. When switching into differential mode the motoring displacement is reduced by the ratio of the head side piston area to the rod side cross sectional area. If the load pressure is under a certain value and the flow demand exceeds the pump capacity assigned to the actuator, differential mode is used. This might be done to either: 1) increase lowering speed of the actuator or 2) to free up motoring capacity for another actuator.

3.3 Hydraulic System Architecture

Differential lowering requires a hydraulic circuit in which the DDPM can be connected either to the head side of the hydraulic ram, or both the head and the rod side simultaneously. The rod side of the cylinder can be connected to the tank, or to both the pump and the cylinder. This can be achieved with the circuit shown in Figure 13. As before this schematic has been simplified. The system tested had the low-pressure side boosted to 2.5 bar, and the valves used were hydraulically piloted with electro proportional control of the pilot pressures.

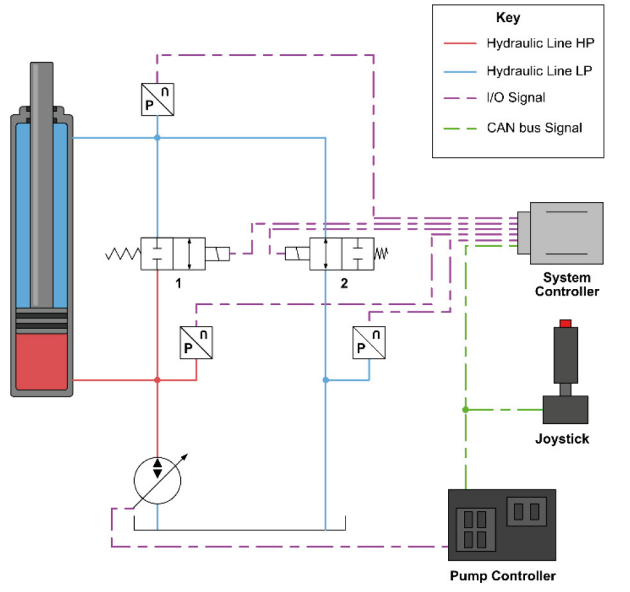

Figure 13: Hydraulic system required for differential lowering mode.

3.4 Differential Lowering Mode Transitions

3.4.1 Normal to Differential Lowering

One problem identified during preliminary testing of differential lowering was that when the rod - cylinder valve was opened, it took around a second for the rod to pressurise, due to the large volume. During this pressurisation period the boom dropped rapidly which was unsatisfactory for operability. The pressurisation time was reduced by pumping the DDPM for a short period of time during the transition until the rod and cylinder pressures equalised. The flow error was measured for a range of different transition pumping displacement fractions (Fd). Figure 14 shows a fairly linear relationship between the pumping displacement used during the transition and the flow error. Pumping at full displacement was found to give the smallest flow error. Such performance is only thought to be possible with a DDPM due to its ability to change from full displacement motoring to pumping in a very short space of time. i.e. 26ms at 1500 rpm. This result is obvious – we need to pump as much as possible in the shortest period available. After the transition the motoring displacement changes to a reduced value (relative to before the transition) to account for the new flow path to the rod side of the cylinder.

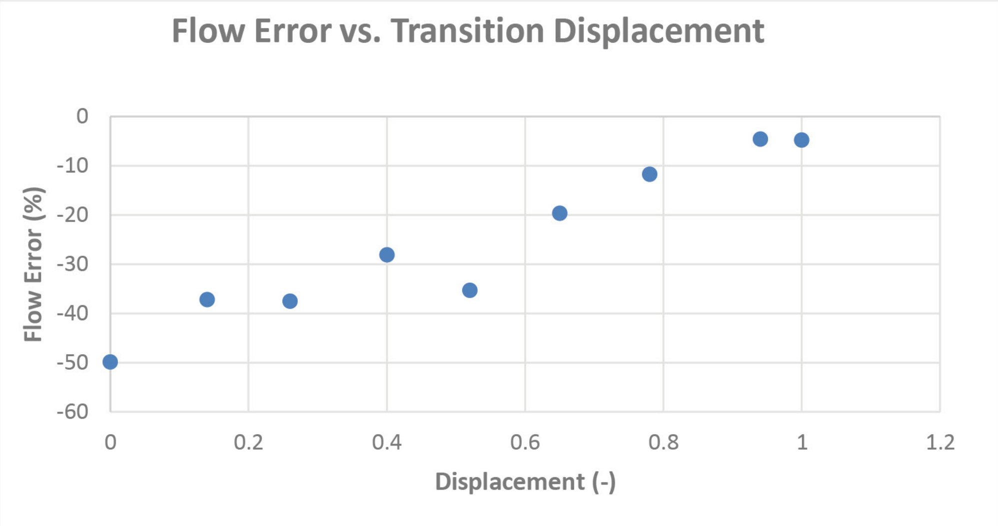

Figure 14: Relationship between DDPM displacement fraction used in transition and the flow error through the transition (over 500ms window).

3.4.2 Differential Lowering to Normal Mode

To transition from differential lowering to normal mode, the rod-cylinder connection was closed during the lowering motion. As the cylinder continued to lower after the valve closed, the rod volume increased without a flow source, dropping the rod and consequently the cylinder pressures. When the rod and tank pressures equalised, the rod - tank valve was opened, completing the transition to normal mode. This control logic gave smooth transitions back to normal mode. One issue identified was that if the lowering motion was stopped too soon, the cylinder could be stopped with the rod in a not yet depressurised. Opening the rod-tank connection in this state caused sudden depressurisation of the rod resulting in a jerky motion. More work must be done to determine a reliable way to depressurise the rod to avoid this situation.

3.5 Differential Lowering Recovery Efficiency

As with discrete throttling, the recovery efficiency was calculated by taking the ratio of the mechanical power measured at the DDPM's crankshaft to the fluid power of the oil leaving the cylinder for a constant lowering speed. To characterise the recovery efficiency at the chosen load pressure, the lowering flowrate was increased in steps of 10 l/min, up to 200 l/min, as seen in Figure 15. The dashed line on the graph shows the switching point between motoring only and differential lowering operating modes. This graph shows a slight reduction in recovery efficiency when operating in differential mode. This reduction can be explained by the pressure losses in the flow path between the cylinder and the rod. It should be noted that as the flow increases throughout differential mode, the pump displacement increases and the machine's efficiency improves. Furthermore, the pump is more efficient at the higher pressures it experiences in differential mode. This counteracts some of the losses associated with higher system pressure drops and explains the relatively flat efficiency curve of differential mode.

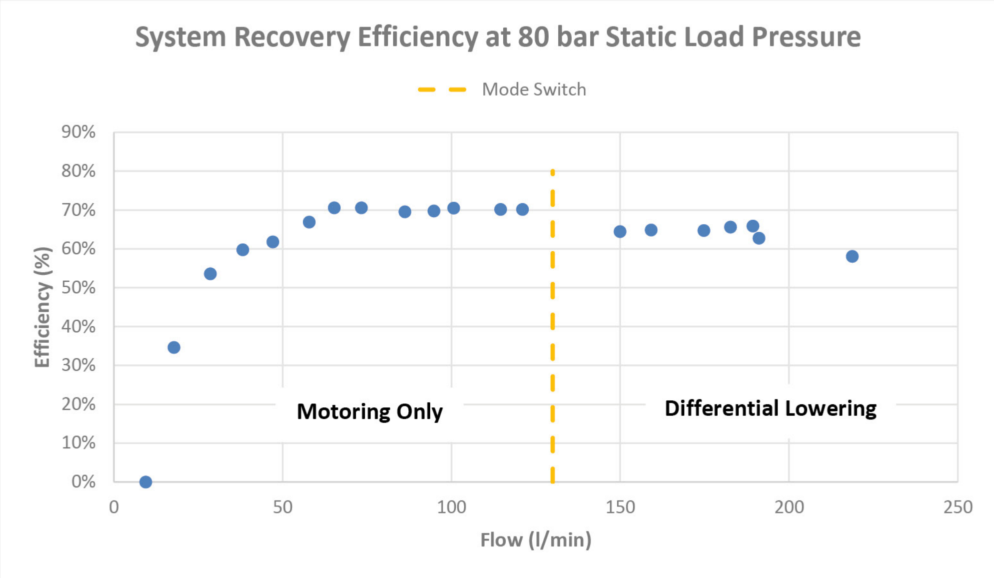

Figure 15: Recovery efficiency of differential lowering mode.

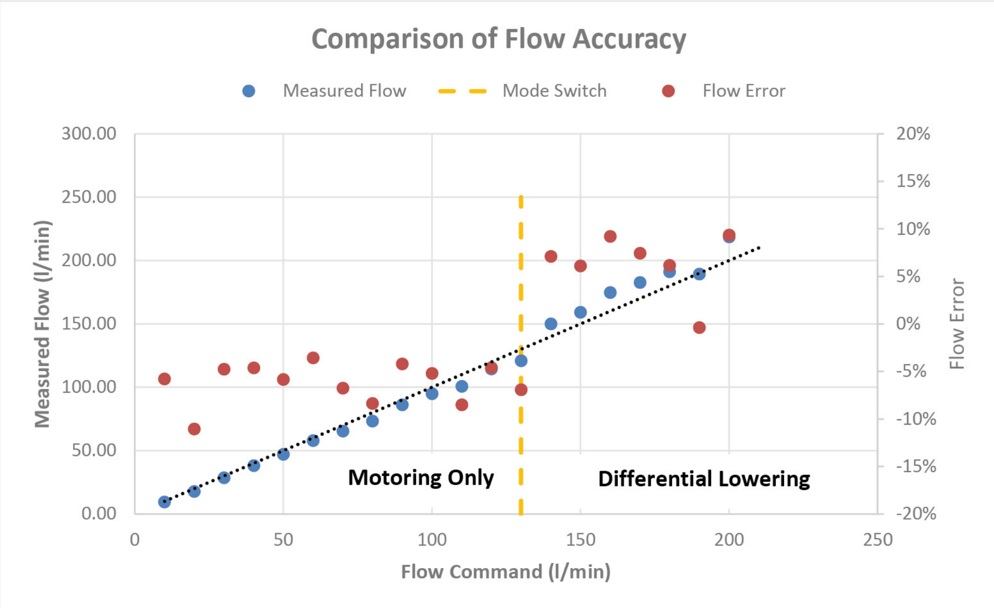

Figure 16: Flow accuracy of differential lowering.

3.6 Differential Lowering Flow Accuracy

Comparisons were made between the flow command and the measured flowrate to establish the flow control accuracy within this operating mode. The graph in Figure 16 shows that the flow error is less than +/-10% across the range of both normal motoring and differential lowering modes. The error could be further reduced by improving the loss model parameters in the flow calculation of the system controller or through better calibration of the pressure sensors giving feedback to the system controller.

4. Summary and Conclusion

Mobile machines need recuperative systems that can handle a larger lowering flowrate than lifting flowrate. One solution is just to install a large pump-motor however this may not be cost effective. By designing systems that are capable of throttling and motoring or of operating in differential mode, high motoring flowrates can be achieved without increasing the size of the pump-motor.

This paper has shown that DDPMs can be used to build effective mode switching systems which can extend the lowering flowrate of a boom cylinder without increasing the pump size. Such control modes can be implemented using simple non-proportional switching valves in the hydraulic system due to the DDPM's ability to change displacement fraction quickly and accurately. This characteristic is used in discrete throttling to ensure the mode transition is as smooth as possible. Furthermore, the ability to rapidly change from full motoring displacement to full pumping displacement was used to reduce the transition flow error when switching into differential mode.

Finally, this paper has highlighted the difference in recovery efficiency of the different operating modes on the same machine. A direct comparison of the recovery efficiencies can be seen in Figure 17. Differential mode is significantly more efficient than discrete throttling, however, is only suitable for lighter loads whereas discrete throttling can be used with any load pressure. Further work will test and tune these control modes on a real excavator. Additionally, development of a high-level system controller that can automatically select between the modes is required which will offer the possibility to optimise the machine performance for either productivity or fuel efficiency.

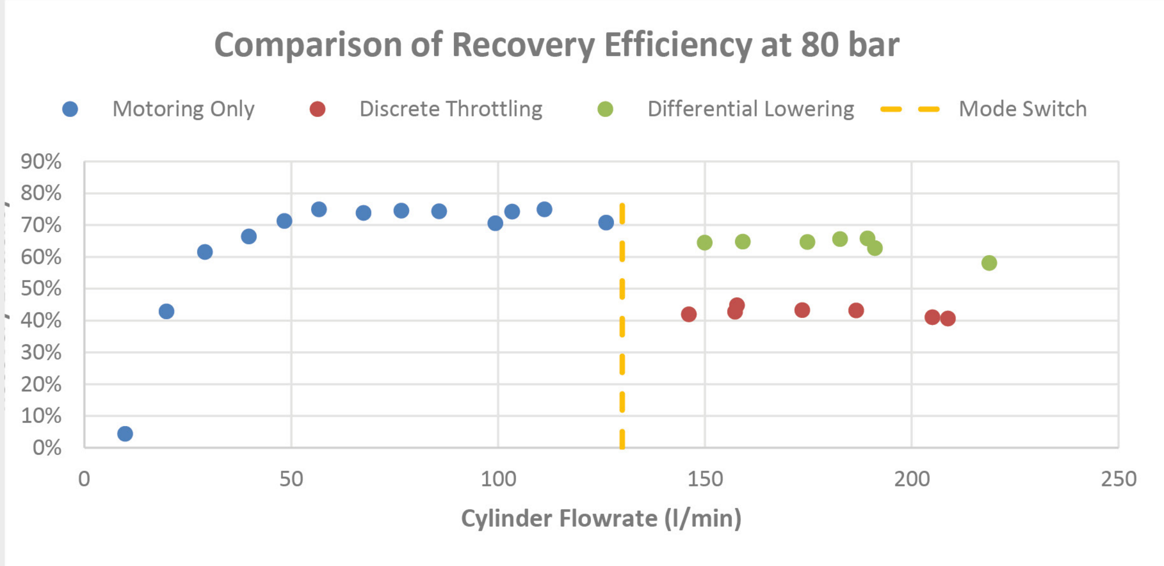

Figure 17: Comparison of recovery efficiency during lowering for the operating modes tested.

Nomenclature

Variable	*Description*	*Unit*
$DDPM_{Total}$	Response time of DDPM to change displacement	[s]
CAN_{delay}	Delay associated with transimission of messages on CANbus	[s]
$DDPM\_Controller\_delay$	Delay associated with the service layer in the pump controller	[s]
ω_{shaft}	Angular velocity of DDPM shaft	[rpm]

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