DEMONSTRATION OF EFFICIENT ENERGY RECOVERY SYSTEMS USING DIGITAL DISPLACEMENT® HYDRAULICS

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ABSTRACT

Heavy off-road vehicles using conventional hydraulic systems waste significant energy through the throttling of fluid to control the motion of their actuators. This paper demonstrates how Digital Displacement ® Pump Motors (DDPMs) can be used to enable efficient hydraulic energy recovery systems for these vehicles by controlling the motion of actuators directly without the need of throttling.

Experiments were carried out on a test rig consisting of a 10 tonne boom supported by a hydraulic ram designed to mimic the setup of a heavy off-road vehicle. In order to demonstrate the DDPM's potential for energy recovery systems the round-trip efficiency was measured by lifting and lowering the boom. The round-trip efficiency was taken to be the ratio of the mechanical energy output from the DDPM, when motoring to lower the boom, to the mechanical energy input to the DDPM, when pumping to raise the boom, over a known ram extension. The results showed measured round-trip efficiencies of between 63% and 87% over a range of pressures, shaft speeds and displacement fractions.

Measured data obtained during the test was used to simulate the test using different system architectures and components to determine the energy efficiency. Both load sense and displacement controlled systems were simulated using both swashplate and Digital Displacement pumps. Comparison showed that the Digital Displacement systems used between 1.1 and 10.8 times less energy than the equivalent swashplate based systems. This work forms the basis for further development of energy recovery system architectures using DDPMs. Future challenges include development of the actuator control valves and transformers required to implement such systems.

Keywords: Digital Displacement, hydraulic energy recovery.

1 INTRODUCTION

Hydraulic systems used in heavy off-highway machines are used to transfer power from the engine to the actuators. Typically, these systems suffer from poor energy efficiency, in the region of around 30% [1].

Poor system efficiency leads to high fuel consumption and carbon emissions – in the United States, excavators alone contribute to 15% of the CO2 emissions produced by construction equipment and machinery [2]. It is well understood that the efficiency of a hydraulic system can be increased by recovering otherwise wasted energy [3, 4, 5, 6], and that the boom service on a hydraulic excavator offers good potential for energy recovery [6]. The recovered energy can be utilised by the system thus unloading the prime mover and ultimately leading to fuel savings. Two primary methods of energy recovery have been proposed in the literature, throttling to an accumulator [7, 8] and transforming [9, 10, 11], however transforming is the focus of this paper.

When transforming, the crankshaft of the machine is used as a torque summing junction which transfers the fluid power from one service to another. The transformer can be used to transfer fluid power between two actuators for instantaneous energy recovery, or between an actuator and an accumulator for energy storage. Due to the nature of the torque summing junction, the hydraulic transformer enables energy to be transferred between services at different pressure levels, higher or lower, without the need to throttle the flow.

The pumping torque available to the second service is equal to an opposite to the motoring torque from the first service. Of course, there are losses associated with the machine so in reality the torque at service two is reduced, depending on the

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machine's efficiency. Clearly it is important that efficient hydraulic machines are used in order to maximise recovery efficiency. A recent study by Artemis Intelligent Power has shown that a hydraulic system architecture based around a multi service Digital Displacement Pump Motor (DDPM) and utilising energy recovery via transforming could enable fuel savings of >50% on a 16-tonne hydraulic excavator on recorded duty cycles [12].

A multi service DDPM consists of several independently controlled high-pressure outlets known as 'pumplets'. Each pumplet can either pump or motor, providing flow in both directions. This system uses the DDPM to transform energy stored in actuators to power other actuators (shown in FIGURE 1), be stored in a hydraulic accumulator (shown in FIGURE 2) or to defuel the prime mover when idling or powering other accessories on the machine.

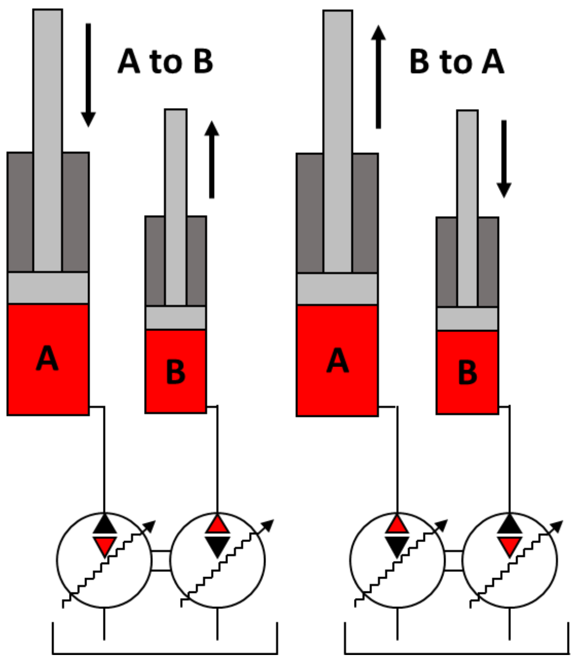

FIGURE 1: SIMPLIFIED SCHEMATIC OF TRANSFORMING BETWEEN ACTUATORS WITH A MULTI SERVICE DDPM.

The key feature which enables these methods of energy recovery is the DDPM's ability to transform fluid power. In order to develop an energy recovery system based around the DDPM it was necessary to understand how efficiently it could be used to transform energy. This paper details the testing undertaken to measure the transformation efficiency, or 'round-trip efficiency' as it is herein referred to, of a DDPM.

An analysis of the system losses during the test was undertaken in order to characterize them and better understand how different elements affect the round-trip efficiency.

Finally, to put the round-trip efficiency result in context, the energy consumption of the DDPM is compared to conventional swashplate machines in load sense and displacement controlled systems, allowing comparisons to be made between different pump technologies and system architectures.

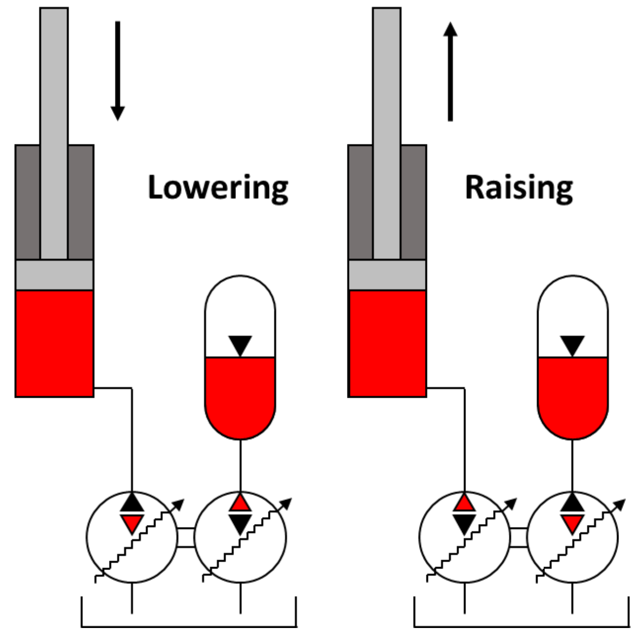

FIGURE 2: SIMPLIFIED SCHEMATIC OF TRANSFORMING TO AND FROM AN ACCUMULATOR WITH A MULTI SERVICE DDPM.

2 CALCULATING ROUND TRIP EFFICIENCY

The round-trip efficiency is defined as the ratio of the mechanical shaft power produced when motoring fluid to the mechanical shaft power required when pumping fluid for a net fluid displacement of zero. This is shown diagrammatically in FIGURE 3.

The round trip efficiency is calculated using equation (1), where p_1 and p_2 are the start and stop times during the pumping phase, m_1 and m_2 are the start and stop times during the motoring phase, T is shaft torque, ω is shaft speed and η is round trip efficiency.

The start and stop times correspond to an upper and lower extension limit for the ram for both the lifting and lowering cycles. For a given displacement command, the effective displacement of the DDPM is different whether the machine is pumping or motoring. The time taken to pump and motor the same volume of fluid will therefore be slightly different.

This difference is due to the design of the high-speed valves and the way in which the DDPM actuates a motoring stroke. The design of the high-speed valves is such that they will not open when the outlet pressure is higher than the cylinder pressure. In order to actuate a motoring stroke, a short part pump is required in order to raise the cylinder pressure and allow the valve to open. By considering the sum of flow into and out of the cylinder during one motoring stroke, it is clear that the net fluid displacement will be lower because of this part pump.

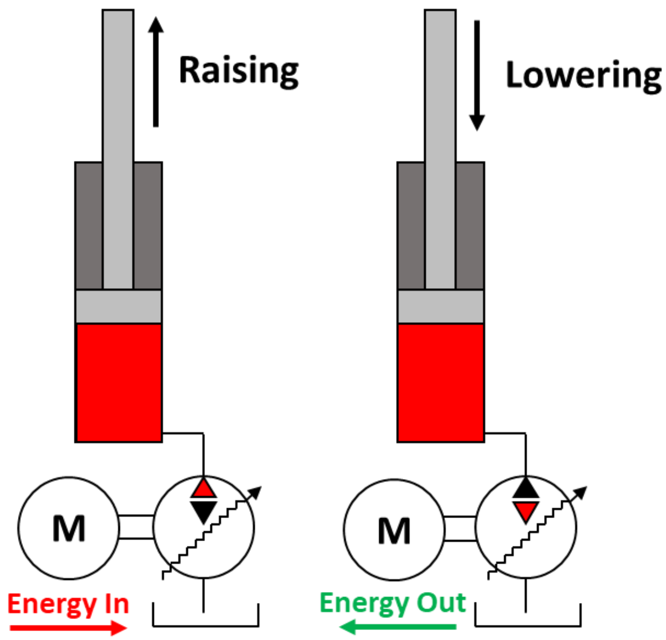

FIGURE 3: SIMPLIFIED HYDRAULIC SCHEMATIC AND ENERGY FLOWS IN ROUND TRIP EFFICIENCY TEST.

$$\eta = \frac{\int_{p1}^{p2} T \cdot \omega \, dt}{\int_{m1}^{m2} T \cdot \omega \, dt} \tag{1}$$

3 TESTING ROUND TRIP EFFICIENCY

3.1 Test Apparatus and Procedure

The round-trip efficiency was tested using a 96cc/rev DDPM (Artemis designation type 'M96') with a single high-pressure outlet directly connected to a hydraulic cylinder. This is a radial piston machine, made up of two banks of six, 8cc, cylinders. The flow output of each cylinder is controlled by an active high- and low-pressure valve. Changing the timing of the valve actuations determines whether the machine pumps or motors. The hydraulic cylinder was mechanically connected to a ten tonne boom which it lifted and lowered as it was actuated. This test rig is known as the Motion Control Rig, or the MCR.

This test rig, seen in FIGURE 4, is designed to mimic the load pressures and range of motions seen in hydraulic excavator and wheel loader applications. Weights can be added and removed, and the position of the ram changed to alter the load pressure and system dynamics.

Pumping the M96 raised the boom and motoring lowered it. No additional valves between the pump and the cylinder were required for control of the ram position. By measuring the shaft speed and torque during pumping and motoring, the round-trip efficiency could be calculated.

An HBM T40B torque transducer was used to measure the shaft torque between the DDPM and the electric motor. This transducer has a nominal rated measurement output of between 500 Nm. The accuracy class of this device is 0.05 giving a measurement error of ±0.25 Nm. The torque offset was measured, and the transducer was recalibrated before testing at each pressure level. The sampling frequency was 2500 Hz.

The shaft speed was also measured using the HBM T40B transducer. With a nominal rated speed of 20,000 rpm and an accuracy class of 0.05 the measurement error in the shaft speed was ±10 rpm. The sampling frequency was 2000Hz

Flow to and from the pump was measured using a VSE VSI 4 flowmeter. This device has a measurement range from 1 to 250 l/min and a measurement accuracy of 0.3%. The sampling frequency was 2000Hz. The flowmeter was situated between the pump and the cylinder, with a switchable bypass. The bypass was used to allow the test to be run without the pressure drop associated with the flow meter, allowing the highest possible round-trip efficiency to be measured.

Pressures throughout the system were measured with HBM P3MB pressure transducers. These sensors have a measurement range from 0 to 500 bar and an accuracy class of 0.1. The sensors were calibrated prior to testing using a Druck portable calibrator. The sampling frequency for pressure sensors was 2000Hz.

As the efficiency of the M96 is dependent on its operating point, it was necessary to sweep test variables in order to characterise the pump over its range of normal operating conditions. The test variables swept were load pressure, shaft speed and displacement command.

The load pressure on the MCR was varied by changing the position of the load ram relative to the pivot point. The two extremes were chosen for this case; the positions nearest and furthest from the pivot point. The mass used was the maximum available and was not changed between tests. These two positions gave load pressures of 95 bar and 180 bar. These pressures are within the typical operating range of a hydraulic excavator and hence were deemed appropriate for these tests.

Displacement fractions of 0.25, 0.5,0.75 and 1 were tested to cover the full displacement range of the pump. Finally shaft speeds of 1000 rpm and 1500 rpm were tested, as these are typical operating speeds for diesel engines in excavator applications.

A one second dwell time was included in the displacement command between lifting and lowering to allow any pressure oscillations in the cylinder to stabilise before continuing. To ensure that each cycle was identical, a Danfoss Plus+1® controller was used to program the displacement command.

Dewesoft X DAQ software was used for data acquisition. The software calculated the energy consumption during the test by multiplying the speed and torque measurements then integrating with respect to time.

The elapsed time at which the upper and lower ram extension limits were reached were used as the integration limits for calculating the energy consumed during pumping and motoring. This ensured that the round-trip efficiency was calculated for a zero-net displacement of hydraulic oil. Each motion of lifting and lowering the boom was considered as one cycle and each operating point was tested for ten cycles to determine repeatability.

FIGURE 4: THE MOTION CONTROL RIG USED TO TEST THE ROUND-TRIP EFFICIENCY. THE 10-TONNE BOOM CAN BE SEEN IN YELLOW, AND THE AUTHOR IS SHOWN, FOR SCALE.

FIGURE 5: THE M96 DDPM USED IN THE TESTS.

3.2 Measured Round Trip Efficiency

The 95 bar tests showed round trip efficiencies ranging from 63% to 83%. The 180 bar tests showed round trip efficiencies ranging from 78% to 87%. Second order polynomial lines of best fit were found to fit the tested datapoints well and have been shown on the plots. The error associated with these calculations is ± 0.2% due to the accuracy associated with the torque and speed transducer.

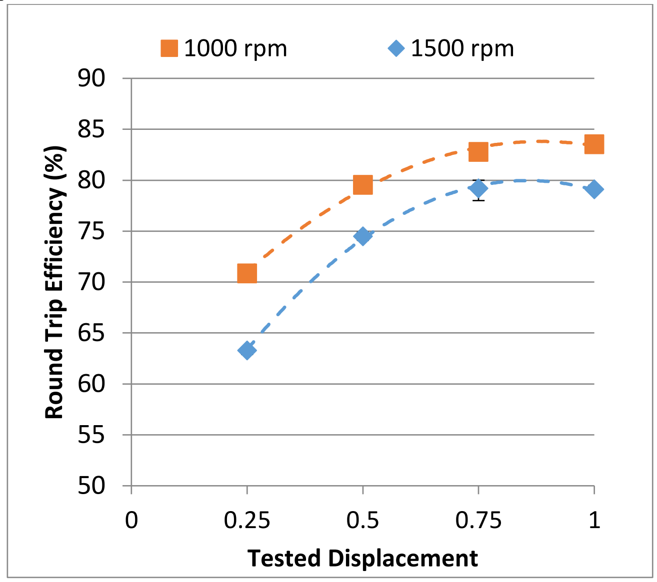

FIGURE 6: ROUND TRIP EFFICIENCY RESULTS AT 95 BAR.

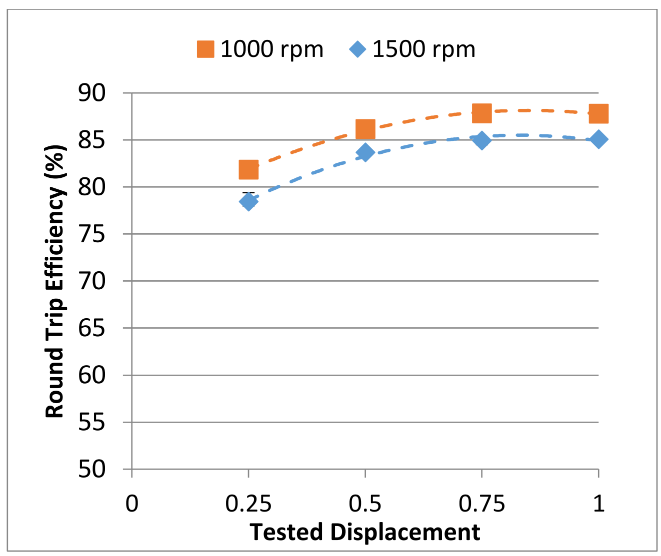

FIGURE 7: ROUND TRIP EFFICIENCY RESULTS AT 180 BAR.

3.3 Round Trip Efficiency vs. Shaft Speed

The losses of a DDPM can be simulated using a semi-empirical model based on experimental data gathered from a test rig dedicated to efficiency testing [13]. The losses for a given output pressure and displacement fraction are modelled by a third order polynomial with respect to shaft speed. The loss model interpolates between loss curves for a given pressure and displacement. The polynomial coefficients for the M96 have been gathered using experiments and the resulting loss model is used for comparison with test results from the MCR test rig. FIGURE 6 and FIGURE 7 show that the round-trip efficiency is greater at the lower shaft speed of 1000 rpm in comparison with 1500rpm. This is predicted by the loss model for the M96 machine. For a given displacement and pressure, at low shaft speeds, the leakage power is dominant. At high speeds internal flow losses and speed dependent friction losses become more significant as a proportion of the total output power. As a result the efficiency peaks at mid-range shaft speeds and for the current M96 machine design 1000RPM is a more efficient operating point for a given pressure and displacement than 1500 RPM.

3.4 Round Trip Efficiency vs. Pressure

The results showed that the round-trip efficiency is higher for a system pressure of 180 bar in comparison to the results at 95 bar for a given displacement and shaft speed. This result is also corroborated by the loss model which predicts an increase in efficiency with pressure for a given displacement and shaft speed. This is since pressure dependent losses such as leakage become less significant as a function of useful output power as the pressure level of the pump increases [13].

3.5 Round Trip Efficiency vs. Displacement

The results showed that, for a given pressure and shaft speed, the round-trip efficiency increases with displacement fraction. This result is predicted by the loss model. The idle losses are effectively a fixed loss for a given pressure and shaft speed regardless of displacement fraction. Some losses vary with displacement fraction. The fluid and mechanical losses for a given pumping or motoring cycle are smaller than the useful output power. Therefore, for a given pressure and shaft speed, the ratio of losses to useful fluid power output reduces as the displacement of the machine increases and thus the efficiency is higher) [13].

3.6 Pseudo Average Efficiency

When calculating the efficiency of a hydraulic machine from measurement data it is important to consider how filtering the measurement data may affect the calculated efficiency. The real efficiency of a hydraulic machine is calculated by averaging the product of instantaneous measurement results over a finite time period. In this case we are calculating the machine efficiency using measured shaft speed and torque. If measurement data is filtered before the efficiency calculation, we have what is called the 'pseudo average efficiency'. If fluctuations in the measurement data are large, then filtering the results prior to the efficiency calculation can result in large errors, as the averaging process results in lost accuracy in the efficiency calculation. This is discussed in detail by Caldwell [14]. To ensure that the efficiency calculation was accurate, no filtering was applied to the measured signals prior to the efficiency calculation.

4 BREAKDOWN OF SYSTEM LOSSES

It is interesting to consider the breakdown of the system losses present during the tests. This breakdown gives insights into the individual contributions of different elements of the system to the overall losses and allows areas for improvement to be identified. The pie chart in FIGURE 8 shows the typical energy breakdown for one test point. It shows that the main system losses are a result of pressure drop across the system, friction within the ram and the DDPM's pumping and motoring losses.

In order to calculate the individual system losses, a forward-facing simulation model of the rig was constructed in MATLAB Simulink. Measured data from the test rig was used to calibrate the model.

The measured pressure drop between the DDPM and the cylinder was used to define the pressure drop in the model and the power loss associated with this pressure drop was calculated using the flow obtained from the model. As the DDPM was directly connected to the cylinder with a short section of hose, with no valves in between, the pressure drop in the line was very small (less than 2 bar). This explains why the losses we see from the pressure drop are so low, compared to the other sources. In this case the pressure drop in the system accounts for 5% of the system losses.

The pumping and motoring losses were calculated in the model using a loss model for the DDPM. Results showed that the pumping and motoring losses are approximately the same accounting for around 40% of the system losses each. The motoring losses are slightly greater than the pumping losses, due to the requirement of a short part pump to open the high-speed valves before a motoring stroke. This results in a slightly reduced motoring stroke volume, resulting in slightly greater losses.

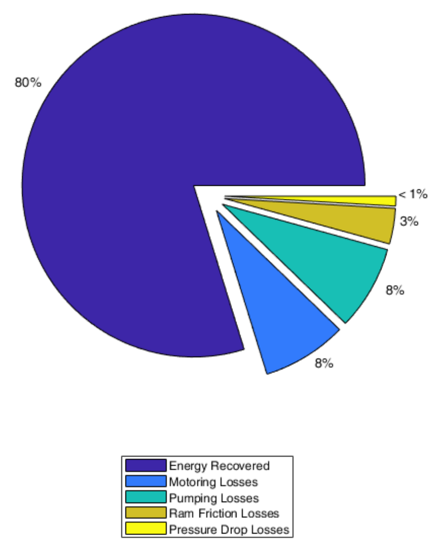

FIGURE 8: TYPICAL ENERGY BREAKDOWN FOR ONE CYCLE.

The losses associated with the ram friction was also calculated. Tests were carried out to determine the relationship between ram friction and shaft speed. The cylinder pressure required to raise and lower the boom for a range of different shaft speed was measured. By taking the difference between this pressure and the theoretical pressure required to lift the load (assuming no friction), the pressure drop associated with the ram friction was calculated. Using the piston dimensions the friction force was calculated. This data was used to calculate the friction force based on the calculated extension speed in the model. Results show that the ram friction accounted for approximately 15% of the system losses.

5 SIMULATED COMPARISON WITH CONVENTIONAL MACHINES AND SYSTEM ARCHITECTURES

5.1 Overview

In order to demonstrate the significance of these measured round-trip efficiencies, the data obtained from the round-trip efficiency tests was used to simulate the energy consumption simulations of different systems. Two system architectures were considered –load sense (LS) control, which is commonly used in mobile hydraulic machines and not capable of energy recovery, and displacement control (DC), which is capable of energy recovery.

The load sense system was modelled with both a typical loss model from a conventional medium-power swashplate pump (like the Rexroth A10) and an E-dyn 96 Digital Displacement Pump. Both machines are unable to go over center, and hence cannot recover energy. The displacement control system was modelled with both a conventional high-power swashplate pump loss model (like the Rexroth A11 or Kawasaki K3V) and the DDPM. Both machines were assumed to be capable of going over center and therefore able to recover energy. This study allows comparison of conventional machines to be made to Digital Displacement (DD) machines, as well as the energy consumption of different system architectures.

5.2 Load Sense System Model

The circuit configuration assumed for the load sense system is shown in FIGURE 9. The swashplate pump of equivalent geometric displacement, i.e. 96cc.was modelled to operate in load sensing mode with a 20-bar margin pressure. During lifting, the swashplate pump's output to the cylinder passes via a proportional directional control valve, and the pump operates at 20 bar over the cylinder load pressure due to the 20-bar load sensing margin. During lowering, the energy cannot be recovered and must be throttled to the tank. The pump must still operate with a 20-bar margin pressure, and due to internal leakage within the actuator control valves and the pump itself energy is still consumed, even when lowering.

In the LS system, we would expect a pressure drop over the directional valve used to control the ram. In this analysis this loss has been omitted in order to keep the pump load the same for both the load sense system and the displacement-controlled system, drawing a fairer comparison.

A Dorey model of the swashplate pump had previously been created by Artemis Intelligent Power and was used for the simulation. The Dorey model is a mathematical model that can be used to account for the losses within a hydraulic pump. The model uses several coefficients that account for viscous friction, Coulomb friction, leakage losses and compressibility losses within the pump [15]. Using physical test data and an optimisation routine Dorey model parameters were calculated to minimise the error.

The energy consumption of the swashplate pump and the DDP was calculated using a backwards facing simulation model. A backwards facing model of a hydraulic pump takes the desired flow output, shaft speed and load pressure as inputs. Using the loss model of the machine, it calculates the required operating parameters of the device such as shaft torque and power consumption to match the input parameters. This type of model can be used to determine the energy consumption of a hydraulic pump for a known duty cycle. Measured flow and pressure data gathered from the DDPM during the round-trip efficiency tests were used as inputs to a backwards facing model of the swashplate pump, in order to calculate the energy consumption for the same cycle.

As the flowrate was only measured for certain operating points during the round-trip efficiency tests it was necessary to calculate the required flow for the backwards facing model. In order to calculate the flow a forward-facing model of the DDPM was used. This forward-facing model takes the operating speed, displacement fraction and pressure level as inputs, and using the loss model of the machine calculates the output flow. Measured shaft speed, displacement fraction and load pressure from the round-trip efficiency tests were used as inputs to the forwards model in order to calculate the flow in cases where it was not measured.

During lowering of the boom, the swashplate pump was assumed to idle at the 20-bar margin pressure, as no energy could be recovered with this machine.

5.3 Displacement Control System Model

The displacement control architecture was identical to the circuit set up for the round-trip efficiency tests. System losses such as pressure drop through hoses and ram friction were therefore identical to the tests. The forward-facing model of the test rig, described in Section 4, was used to compare the energy consumption of the displacement-controlled system. To model the system with the overcentre swashplate machine, the swashplate loss model was simply swapped into the forwards facing pump model. A Dorey model was also used to model the overcentre swashplate pump, parameters for this model were obtained through a series of tests by Danfoss Power Solutions. These tests were based on a machine that could not go over center, therefore the losses for pumping and motoring were assumed to be the same. Furthermore, these measurements did not include losses associated with the control leakage, therefore a simulated control leakage of three l/min was applied to the model to account for this. The same measured inputs from the test rig were used for both the DDPM and the swashplate case.

5.4 Simulated Results

The results are presented in terms of the normalised energy consumption of each machine. This term was determined to be the ratio of the energy consumed by the pump during one lifting and lowering cycles to the energy required to only lift the boom during on cycles, which was determined as the useful work. The useful work was defined to be the fluid power of the output flow from the M96 as it was directly connected to the hydraulic cylinder with minimal system losses. The fluid power was calculated from measured pressure data and simulated flowrates using the forwards facing model described previously.

The normalised energy consumption is a metric that allows the energy efficiency of different systems to be compared. A normalised consumption of one would indicate that the pump is 100% efficient but with no energy recovery. Values greater than one indicate that the pump requires more energy than the useful work required to raise the load, indicating that the pump has some degree of inefficiency with no energy recovery. Values of less than one are only possible when energy is recovered. A normalised energy consumption of zero would indicate that all the useful work is recovered again and would represent a system with no losses. A normalised energy consumption close to zero would be indicative of an efficient hydraulic system utilising energy recovery.

Figures 9 to 12 present the results for each of the main test cases. From the results at all test points we can see that the two load sense systems, one with the DDP and the other with the medium power swashplate pump, both have a normalised energy consumption of greater than one. This is as expected because neither system can recover energy. The system with the DDP has a slightly lower normalised energy consumption across the range due to its improved efficiency over the swashplate pump.

The displacement-controlled systems both exhibit a normalised energy consumption of less than one due to their ability to recovery energy. Similarly to the load sense case, we see that the DDPM has a slightly lower energy consumption at all tested points, due to the superior efficiency of the Digital Displacement machine.

By taking the ratio of the normalised energy consumption of each system relative to a benchmark case, the improvement in system efficiency can be determined. In this study the benchmark case is taken as the medium power load sense swashplate case, as this exhibits the lowest efficiency yet represents the benchmark in terms of commercially implemented hydraulic systems in mobile off highway machines. The percentage improvement for each case is seen in Table 1. We can see that the DDP load sense case is between 1.1 and 1.6 more efficient, the displacement-controlled swashplate is between 2.9 and 6.8 more efficient, and the DDPM system is between 5.6 and 10.8 more efficient.

The results of this study suggest that by using Digital Displacement Pumps can allow engineers to improve hydraulic system efficiency when compared to conventional swashplate technology in both load sense and displacement-controlled circuits.

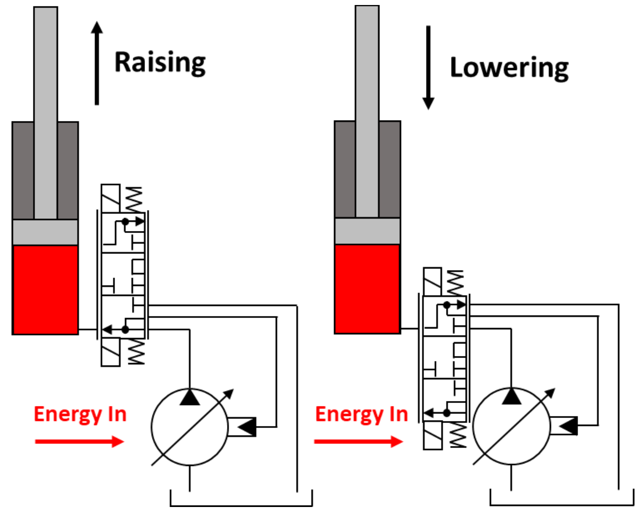

FIGURE 9: SIMPLIFIED CIRCUIT SCHEMATIC OF CONVENTIONAL LOAD SENSE PUMP RAISING AND LOWERING A BOOM.

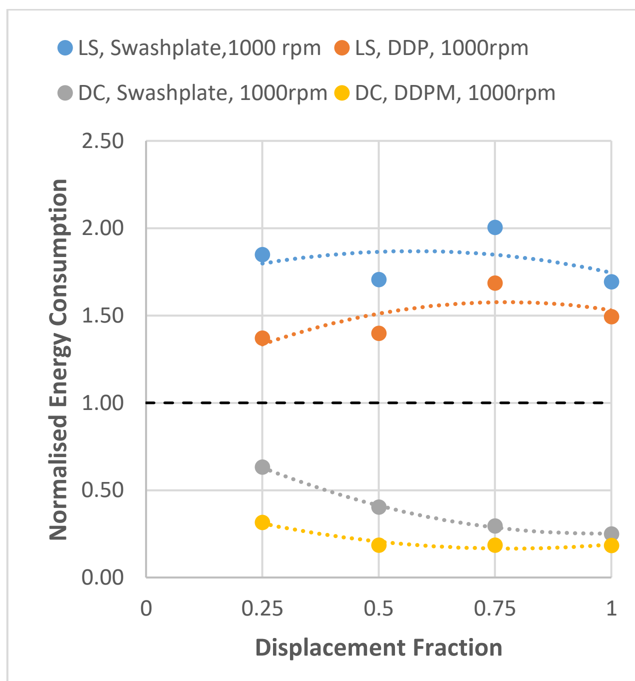

FIGURE 9: 95 BAR, 1000RPM SIMULATION RESULTS.

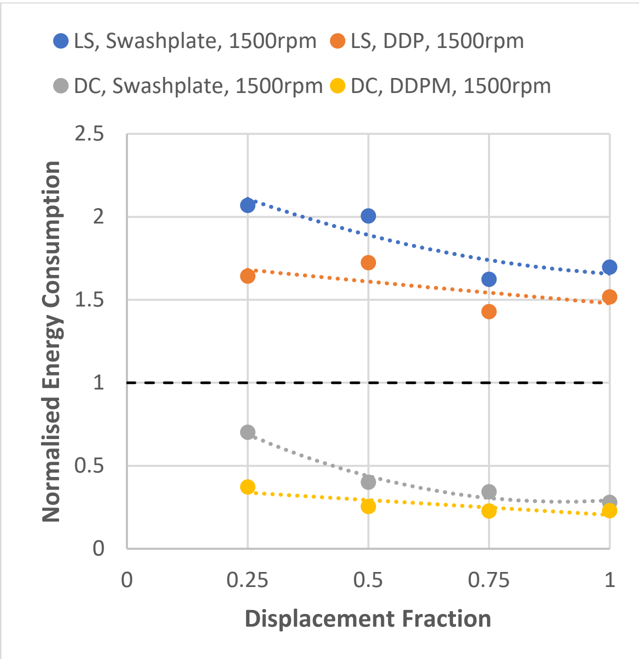

FIGURE 10: 95 BAR, 1500RPM SIMULATION RESULTS.

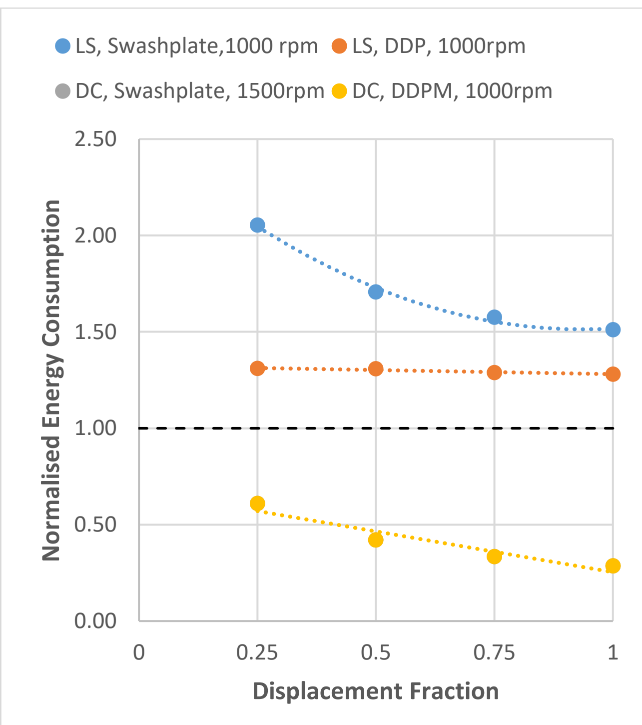

FIGURE 11: 180 BAR, 1000RPM SIMULATION RESULTS.

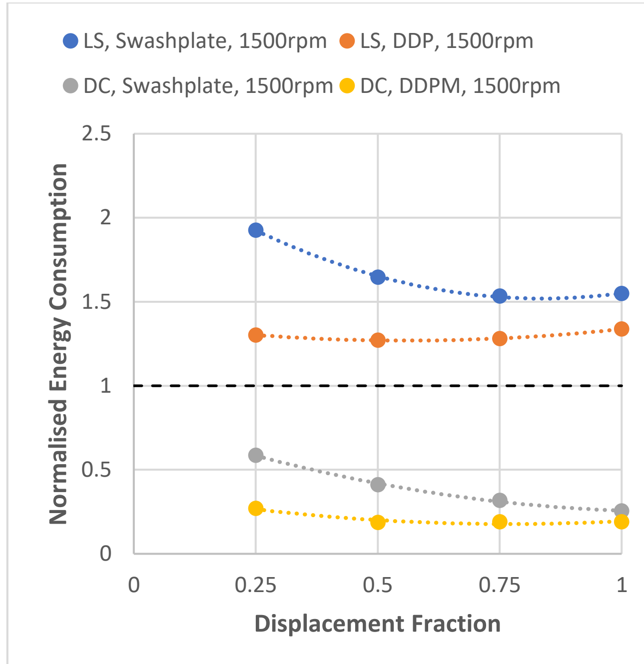

FIGURE 12: 180 BAR, 1500RPM SIMULATION RESULTS.

TABLE 1: ENERGY CONSUMPTION RATIO FOR EACH SYSTEM

Pressure (bar)	Speed (rpm)	Displacement Fraction	Energy Consumption Ratio			
			LS, Swashplate	LS, DDP	DC, Swashplate	DC, DDPM
95	1000	0.25	1.0	1.3	2.9	5.8
		0.5	1.0	1.2	4.2	9.2
		0.75	1.0	1.2	6.8	10.8
		1	1.0	1.1	6.8	9.2
	1500	0.25	1.0	1.3	3.0	5.6
		0.5	1.0	1.2	5.0	7.9
		0.75	1.0	1.1	4.7	7.1
		1	1.0	1.1	6.1	7.4
180	1000	0.25	1.0	1.6	3.4	8.6
		0.5	1.0	1.3	4.1	9.2
		0.75	1.0	1.2	4.7	9.2
		1	1.0	1.2	5.3	8.8
	1500	0.25	1.0	1.5	3.3	7.1
		0.5	1.0	1.3	4.0	8.8
		0.75	1.0	1.2	4.8	8.0
		1	1.0	1.2	6.1	8.1

6 FURTHER WORK

With a thorough understanding of the round-trip efficiency future work will focus on the development of control strategies for utilising a DDPM for energy recovery. Research must be done to determine which recovery methods are best suited to different machine operating states. A control strategy must be developed to ensure that recovery methods are prioritised appropriately based on operator inputs and the current machine state and that they do not adversely impact operator feel and machine productivity.

Secondly, work is underway to develop the valves required for displacement control of actuators with energy recovery. There are certain functional (e.g. four quadrant control, regeneration) and safety (e.g. load holding) requirements which must be met if the system is to be implemented in off-highway machines. Future work will test different valve configurations and control strategies and assess their suitability to real applications.

7 CONCLUSIONS

The M96 showed a round trip regenerative efficiency of between 63% and 87% for a range of displacement fractions, shaft speeds and output pressures.

The simulated system comparison showed that use of a DDP in a load sense system can improve energy efficiency by up to 1.6 times over the benchmark case. Using a DDPM in a displacement-controlled circuit improves energy efficiency by 10.8 times over the benchmark load sense case, or by 2.6 times when compared to the equivalent displacement-controlled system using a high-powered swashplate pump.

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