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DEXTER - THE APPLICATION OF A DIGITAL DISPLACEMENT® PUMP TO A 16 TONNE EXCAVATOR

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ABSTRACT
Environmental and economic factors are driving the development of lower emission and more fuel efficient off-highway vehicles. While a great deal of this development is focused on hybrid technology and novel system architectures, the simple application of a Digital Displacement® Pump (DDP) in place of a conventional pump can deliver significant fuel savings and productivity benefits, whilst also acting as an enabler for more radical future development. This paper describes the 'DEXTER' project, in which a tandem 96cc/rev DDP was installed in a 16 tonne excavator. The energy losses in the unmodified excavator are calculated based on test data, confirming the scope for efficiency improvements. Next, the basic operating principle and efficiency of the DDP and its application to the excavator system are outlined, alongside simulation based fuel saving predictions. The model based design and 'operator in the loop' testing of the control system are then described. Side by side testing of the modified excavator and a standard excavator showed that when the modified excavator was operating in 'efficiency mode' a fuel saving of up to 21% and productivity improvement of 10% is possible. In 'productivity' mode, a 28% productivity improvement was recorded along with a 10% fuel saving. These results are validated with reference to the higher efficiency of the DDP and improved control system which allows the engine to run closer to its torque limit.

INTRODUCTION
With their combination of high power density and reliable, high force, low cost linear and rotary actuators, hydraulic systems are ideally suited to excavators and other off-highway construction machines. However, increased focus on exhaust emissions and fuel costs has highlighted the poor efficiency of the hydraulic systems in these machines where typically only 30% of the fluid power at the pump outlet is converted into useful work. Many solutions have been explored to address this issue [1], for example by changing the system topology to reduce valve throttling losses [2] [3], and recovering energy from decelerating functions [4]. Some of these techniques appear in commercially available 'hybrid' machines and have generated real-world fuel saving data [5]. Market penetration is still small however, due to increased capital cost and operator concerns over reliability and more complex servicing requirements. This paper presents a method by which similar fuel savings can be achieved with much lower complexity and technical risk, by simply exchanging the traditional variable displacement, axial piston, swashplate pump with a Digital Displacement Pump. Beyond this no changes were made to the valve block or hydraulic circuit. Combining the use of a DDP with an optimised hydraulic system including energy recovery would give fuel savings far greater than can currently be achieved.

Digital Displacement pumps are radial piston machines, where the total pump displacement is varied by controlling the displacement of each cylinder on a cycle by cycle basis. This is achieved by selectively closing a solenoid controlled inlet valve as the piston passes bottom dead-centre, causing fluid to leave the cylinder via a passive outlet check valve to the pump outlet. The efficiency benefits of Digital Displacement machines are well documented in for example [6] and [7]. Further details of the operation modes, control methods and operating efficiencies of a DDP can be found in [7], [8], [9] and [10].

The target machine, a JCB JS160 tracked excavator, was chosen because, as standard, it is fitted with an 80 cc/rev tandem pump that is close in displacement to Artemis Intelligent Power's 'E-dyn 96' cc/rev tandem machine. The excavator's negative flow control system (or 'Negacon') is also common across many excavators in this size range (see Figure 7). The specification of the excavator, the original pump and the E-dyn 96 pump are shown in Table 1 and Table 2.

Table 1: JCB JS160 excavator specification [11]

Operating weight [kg]	17774
Engine net power [kW]	93 (at 2200RPM)
Max. operating pressure [Bar]	343
Max. pump flow [L/min]	2 x 164

Table 2: Original pump and E-dyn 96 specifications.

	Original pump (Kawasaki K5V80) [12]	E-dyn 96
Displacement [cc/rev]	2 x 80 (tandem)	2 x 96 (tandem)
Rated operating pressure [bar]	343	420
Max. operating speed, self-priming [RPM]	2460	2700
Max. torque [Nm]	529	1000
Dry weight [kg]	81	111
Control	Hydro-mechanical	Electronic

EXCAVATOR ENERGY LOSSES

By instrumenting the excavator, the energy loss in each component of the hydraulic circuit can be studied. Figure 1 shows the basic machine layout, and the instrumentation installed during energy loss testing. Oil is delivered from the pumps, via a series of valves (mostly housed inside a complex valve block), to each function. Exhaust oil from these functions is returned to the tank via the same valve block. The delivery and return of oil is controlled by pilot operated spool valves, the pilot signal being derived from the movement of the operator controls.

A trenching cycle was used to generate baseline test data. Details of the test cycles referred to in this paper can be found in Annex A. The data was analysed using a backward-facing simulation, the basic layout of which is shown in Figure 2. The work done by each ram is calculated using ram velocity and force, based on the ram displacement measurement, known piston areas and ram pressures. Depending on the sign of the velocity and the direction of the resulting force, the work can be classified as useful work or available regenerative energy. The valve block losses, which include all losses between the pump outlet and the function inlets, are the difference between the actuator useful work and the pump output energy. The simulation includes a loss model of the pump and a model of the diesel engine to allow estimation of the pump losses, engine operating point and engine fuel consumption.

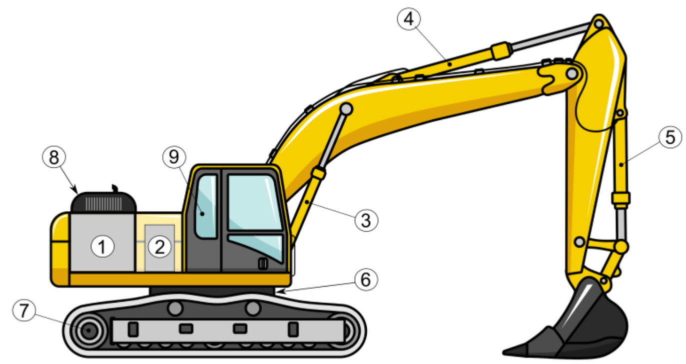

Figure 1: Basic excavator layout. Each outlet of the tandem pump (1) is fitted with a flowmeter and pressure transducer. The pilot pump pressure is also measured. The negative flow control pressure is measured using a pressure transducer at the valve block (2). The boom rams (3), dipper ram (4) and bucket ram (5) are fitted with string potentiometers and pressure transducers at each port. Pressure transducers are fitted to the swing motor ports (6) and both travel motors (7). An encoder measures swing angle. The engine (8) is fitted with a current and voltage transducer to measure alternator load. The ECU (9) reports engine speed, torque and fuel consumption among other parameters via the CAN network.

Pump loss model

To calculate the energy loss in the pump and therefore the required engine shaft torque, for a given flow, pressure and shaft speed, a parameterized model of the swashplate pump was used [13]. The model parameters were derived from the datasheet of a Mitsubishi MKV-11H pump [14], due to the lack of available data for the K5V80 pump. Two simulations were carried out with the swashplate model; the first with a pump displacement of 80cc/rev to match the original pump (case 1), the second with a displacement of 96cc/rev (case 2). A modified Dorey loss model, as described in [6], was used to model the losses of the E-dyn 96, with parameters determined by experiment. In the E-dyn 96 simulations (case 3 and 4) the maximum open centre flow used for negative flow control was reduced from 20 L/min to 5 L/min. This reduction was based on the assumption that the DDP could use a smaller pressure range to control displacement, and that any damping effect of the open centre flow could be replicated by a leakage function in the system controller. Figure 3 shows the efficiency of DDP at 1500RPM, 200 bar over the displacement range, based on experimental data produced by Artemis. In addition, curves are shown for three other swashplate pumps, based on experimental data produced by Artemis and manufacturer data.

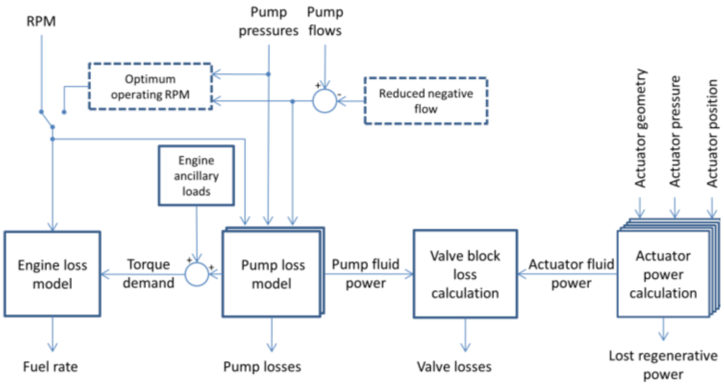

Figure 2: Backward-facing simulation layout

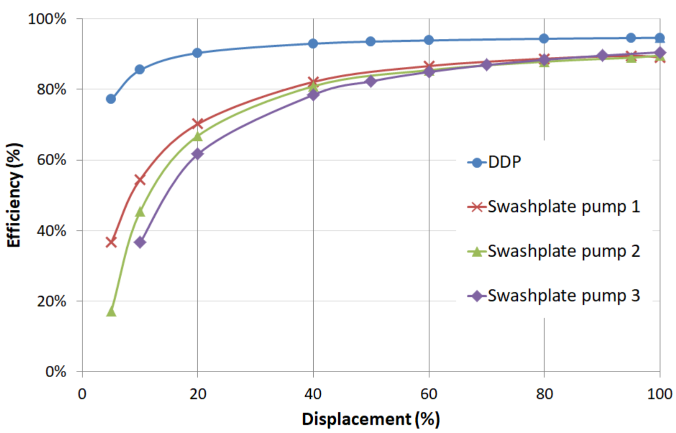

Figure 3: Pump efficiency and loss power of typical swashplate pumps (Pump 1 and 2 from experiment, pump 3 derived from datasheet [14]) and DDP (from experiment) as a function of displacement at 200 bar, 1500RPM

Engine model

The engine model uses a brake specific fuel consumption (BSFC) map to calculate the fuel rate needed to meet the required output torque at a certain shaft speed. The output torque in this case is the pump torque plus the ancillary load of the alternator and pilot pump. The shaft speed is the recorded shaft speed, except in the 'optimum RPM' cases (case 2 and 4), where the shaft speed is determined by calculating the approximate pump power demand and finding the optimum engine operating point to satisfy that demand while still satisfying the pump flow demand.

Figure 4 compares the results of the simulations. All values are averaged over the complete test cycle. The effect of simply installing a larger capacity pump can be seen by comparing the two swashplate pump simulations – by increasing the displacement from 80cc/rev (case 1) to 96cc/rev (case 2) the fuel rate is reduced by 4.7%. The increased displacement allows the engine to be operated at a lower speed and better BSFC. It should be noted that the average pump efficiency is reduced. Comparing the 80cc/rev swashplate pump (case 1) and 96cc/rev DDP (case 3) results show the effect of improved pump efficiency, from an average of 82.1% to 91.5%. The average engine BSFC is in fact slightly worse than in the baseline case, because the engine is operating at the same speed but with lower load. The fuel rate is still reduced by 12.1%. Combining the higher pump efficiency of the DDP with operation at optimum engine speed gives a 15.8% reduction in fuel rate.

To further understand the excavator system, Figure 5 shows graphically the losses in the machine for the baseline case (case 1) and the predicted values for the DDP system (case 4). Using the DDP reduces the average pump losses from 10.4kW to 3.6kW. Reducing the open centre flow reduces valve block losses from 21.3kW to 19.3kW. The pump input power is therefore reduced from 60.6kW to 51.8kW. This reduction is compounded by operating at a lower engine BSFC (259g/kWh compared with 263g/kWh).

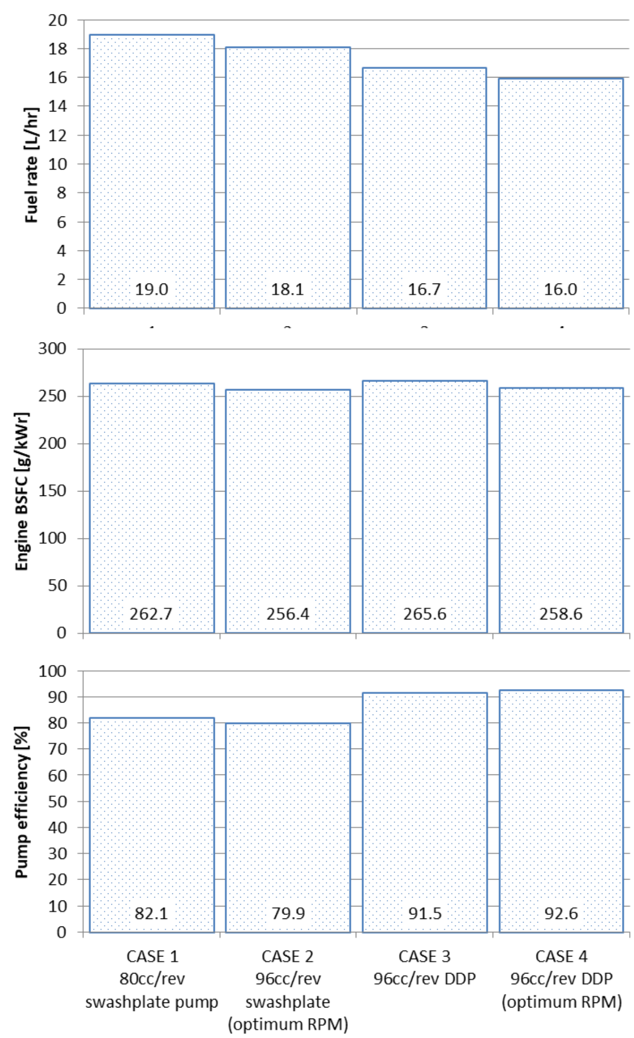

Figure 4: Backward facing simulation results

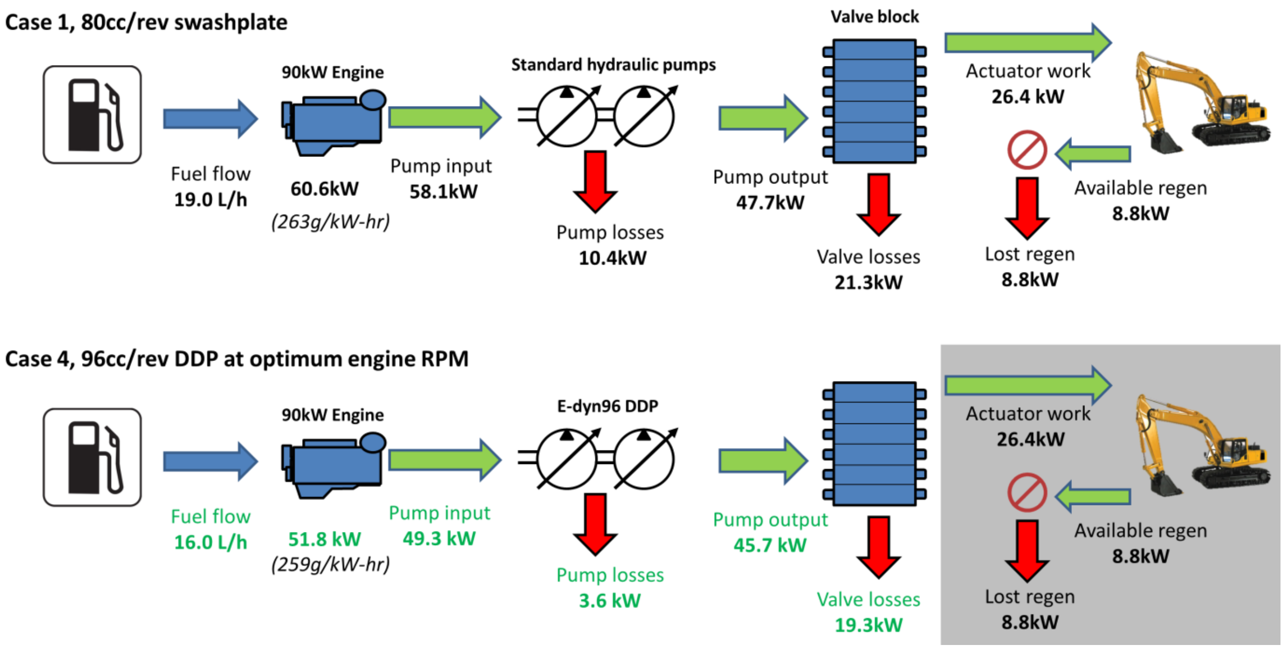

Figure 5: Losses in the excavator system. Values shown are averaged over the test cycle.

Table 3 presents the loss contribution of each component as a percentage of the total. The overall efficiency of the baseline hydraulic system is 30%. It is evident from the results that there are large efficiency improvements to be made by reducing valve block losses and incorporating an energy recovery system. The work presented in this paper is however limited to exchanging the pump, which accounts for 26% of the wasted energy.

Table 3. Simulated baseline and DDP trenching cycle results from backward facing model

	Calculation method	Simulated baseline result, Case 1 (%)	Simulated DDP result, Case 4 (%)
Pump mean efficiency	$\frac{P_{pump\ output}}{P_{pump\ input}}$	82	93
Hydraulic system mean efficiency	$\frac{P_{actuator\ work} - P_{lost\ regen}}{P_{pump\ inlet}}$	30	36
Pump loss as % of total losses	$\frac{P_{pump\ loss}}{P_{pump\ loss} + P_{valve\ loss} + P_{lost\ regen}}$	26	11
Valve block loss as % of total losses	$\frac{P_{valve\ loss}}{P_{pump\ loss} + P_{valve\ loss} + P_{lost\ regen}}$	53	60
Wasted regenerative power as % of total losses	$\frac{P_{lost\ regen}}{P_{pump\ loss} + P_{valve\ loss} + P_{lost\ regen}}$	22	28

DDP INSTALLATION

Beyond the instrumentation described in Figure 1, the hardware modifications to the excavator were limited to removing the K5V80 pump and installing the E-dyn 96 tandem pump (Figure 6). Two additional pressure transducers were installed to measure Negacon control pressure to provide an input to the control system. No other changes were made to the machine.

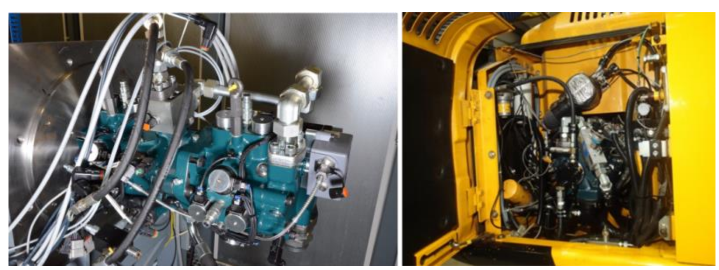

Figure 6: E-dyn 96 tandem pump during commissioning (left) and installed in excavator with through shaft pilot pump (right)

CONTROL DEVELOPMENT

The instantaneous displacement of the DDP is commanded by a system controller. The system control logic was developed using Simulink, allowing quick comparison with the original control system using the baseline data as a reference. Once the basic functionality was achieved (i.e. the correct displacement demand for a given negacon pressure and pump outlet pressure) additional features were added to improve efficiency and productivity. Successive revisions of the control software could be rapidly compiled for use in the system controller, allowing a

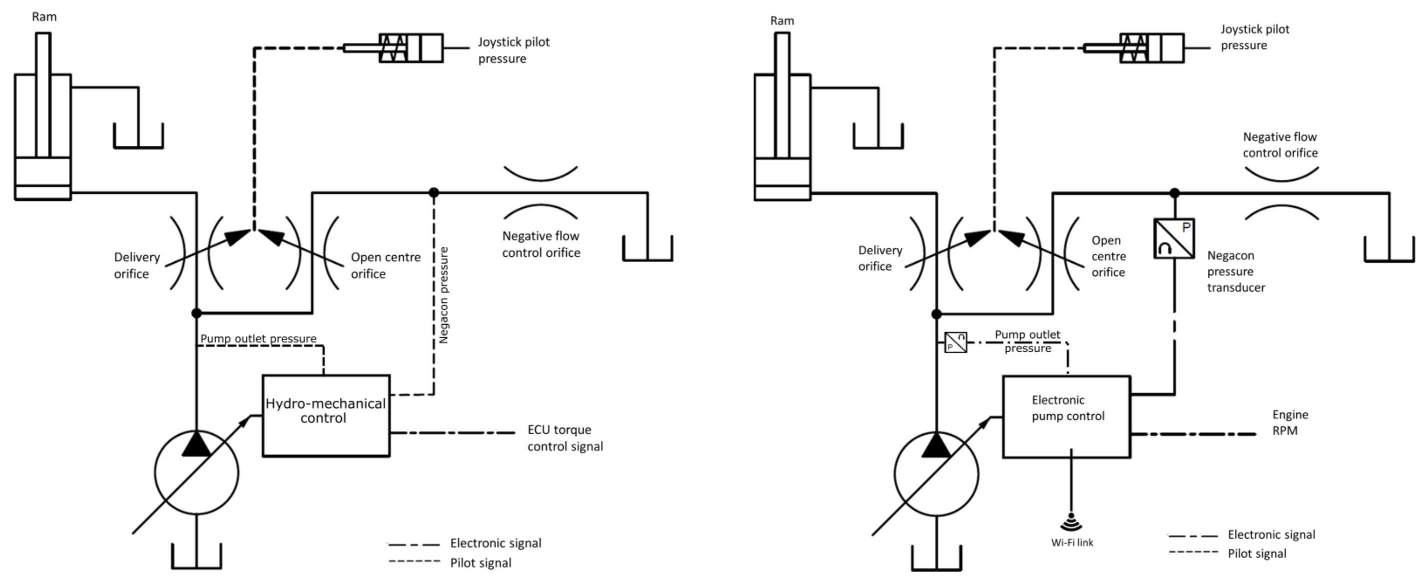

Figure 7: Simplified diagram of negative flow control system (left) and the DDP negative flow control system (right)

fast, iterative approach with extensive 'operator-in-the-loop' testing.

The original excavator control system is based on negative flow control. A simplified diagram showing the layout of this system is shown in Figure 7. Pilot pressure from the operator's joystick proportionally restricts the open centre orifice, and opens the delivery orifice of an open centre spool valve. As flow is diverted to the function, the open centre flow through the negative flow control orifice reduces. This reduces the negacon pressure, increasing the swashplate angle and therefore pump displacement. The maximum pump displacement is limited by feedback of the pump outlet pressure and a signal from the machine's ECU, commonly referred to as a 'horsepower control' signal. By experiment it was found that the effect of this is to limit pump torque to a level significantly below the available engine torque.

The DDP control system shown in Figure 7 uses the same negacon pressure, in this case measured by a pressure transducer, with an empirical lookup table to determine pump displacement. This pump displacement signal is then modified by the following, in the order that they are presented:

Swashplate dynamics low-pass filter
The DDP can go from zero to full displacement in half a shaft revolution. At the nominal excavator operating speed of 1450 RPM that is around 21ms. To match the performance of the swashplate pump, which is approximately an order of magnitude slower, a low-pass filter was applied to the pump displacement demand.

Pressure limiter (slew service)
To minimise the flow lost through the slew pressure relief valve a pressure limiter was implemented that acts only if the slew service is operating.

Torque rate limiter
To avoid excessive engine speed droop during a transient increase in pump torque demand a torque limiter was applied, with a varying rate according to engine operating speed.

Anti-droop
Where the original control system used the horsepower control signal to limit pump torque, the DDP controller imposes a torque limit based on the engine speed error. The pump controls engine speed droop according to a characteristic that can be tuned for minimum droop (the minimum being at the limit of stability, and also dependent on operator preference) or maximum power (by allowing energy to be taken from the engine flywheel during transient high power demand). The engine setpoint RPM is still under operator control, rather than being set by the controller to a calculated optimum operating point as used in simulation cases 2 and 4. Figure 8 shows the anti-droop control in action.

Torque Sharing
To ensure both pumps can be supplied with at least as much torque as was available to the originals, torque sharing logic was imposed. This guarantees each pump 50% of the available engine torque, but allows one pump to exceed 50% if the other pump does not require it.

Overall pressure limiter
An overall pressure limiter eliminates flow through the main pressure relief valve by reducing pump displacement as the pressure limit is reached.

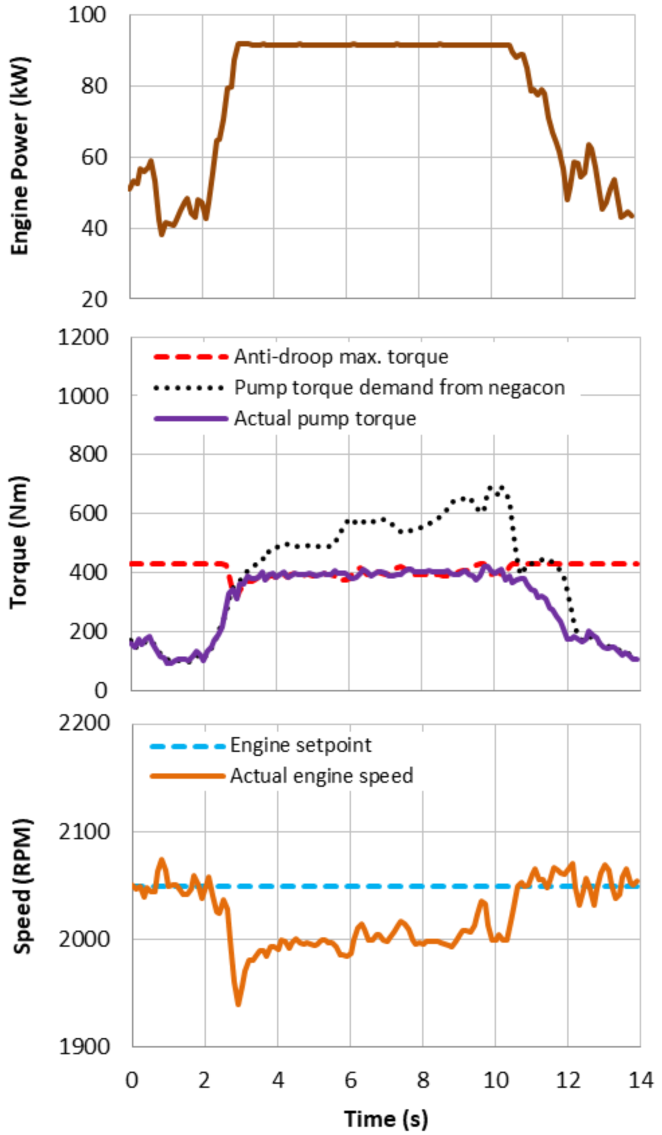

Figure 8: Test data showing the Anti-droop control reducing pump torque demand as the engine speed drops below the setpoint. This allows the pump to use high engine power without the risk of stalling.

Error states

Any parameters measured to be out of range cause the pump controller to enter an error state. This guards against negacon pressure transducer failure causing unwanted pump displacement for example.

The ‘operator feel’ of an excavator, or how the functions respond to the operator’s joystick commands, is crucial to ensure productive, fuel efficient operation. It also affects the operator’s comfort and perception of the machine’s quality. Qualitative evaluation of the modified excavator was carried out by an expert operator, who graded the machine on controllability, productivity, power, operator comfort and noise. This testing confirmed that the operator feel was satisfactory. To aid in this process an emulation mode was implemented in the system controller, allowing the DDP to operate with the same characteristics as the original pump and with the same torque limits. It was also decided that because reducing the open centre flow (as in the DDP simulation cases 3 and 4) would have a significant effect on the operator feel it was beyond the scope of this project.

TESTING

Testing of excavators to evaluate fuel consumption is a significant challenge. The range of machines and the diversity of tasks that each machine carries out means that there is no defined drive cycle over which all machines are benchmarked. A number of basic cycles are commonly used but the exact definition of these cycles is also not uniformly defined. Annex A depicts the cycles used during testing. To obtain repeatable results these must be performed multiple times with the same environmental conditions and the same operator. To eliminate environmental effects the modified excavator was tested alongside an unmodified excavator of the same model. To eliminate operator variability the cycles were repeated multiple times, and any test data where a large deviation from the mean fuel rate, cycle rate or volume of material moved was discarded.

The fuel consumption was measured using a fuel meter inserted into the fuel supply line. The productivity was measured by recording the rate at which the cycles were completed and where possible measuring the volume of material moved.

During testing, the DDP control logic was tuned to try to match the productivity of the two excavators, so that the fuel consumption comparison could be carried out independently of any productivity difference. This required that the engine speed in the modified machine was reduced by 600 RPM. As shown in Table 4 the productivity was still higher than the baseline machine in some test cycles by up to 10%. The fuel saving per cycle ranged from 16 to 21% on the working cycles, and 27% when idling. To compare the maximum productivity of the two machines the digging cycle was also carried out with both machines set to maximum engine speed (2050 RPM). The result of this was a 28% increase in cycle rate, with a fuel per cycle saving of 10% (Table 5).

Table 4: Comparative testing results, DDP engine speed reduced in attempt to match productivity

Cycle	Baseline RPM	DDP RPM	Fuel saving per cycle	Cycle rate increase
Trenching	2050	1450	21.2%	10.4%
Bulk dig	2050	1450	21.2%	10.6%
Lorry load 90º	2050	1450	18.4%	-0.4%
Lorry load 180º	2050	1450	16.1%	1.9%
Tracking	2050	1650	16.1%	-
Idling	950	950	27.1%	-

Table 5: Comparative testing results, maximum productivity

Cycle	Baseline RPM	DDP RPM	Fuel saving per cycle	Cycle rate increase
Digging	2050	2050	10.0%	28.0%

ANALYSIS OF RESULTS

Because the lorry loading 90º cycle productivity was very similar for both the modified and original machines, it was chosen for further analysis of the fuel saving result using the backward-facing simulation. The results of this analysis are shown in Table 6. The BSFC is improved by operating at lower speed and higher torque. This is made possible by the higher DDP displacement and smaller torque headroom requirement due to the high bandwidth torque control (see Figure 9). The engine power demand is also reduced due to the higher pump efficiency. Lowering the engine speed also has benefits in terms of reducing the fan load.

Table 6: Analysis of 90º lorry loading fuel saving using backwards facing simulation.

Total fuel saving (measured)	Fuel saving attributed to pump efficiency (simulated)	Fuel saving attributed to engine BSFC change (simulated)	Other fuel saving (fan speed/power, ancillary load)
18.4%	7.5%	4%	6.9%

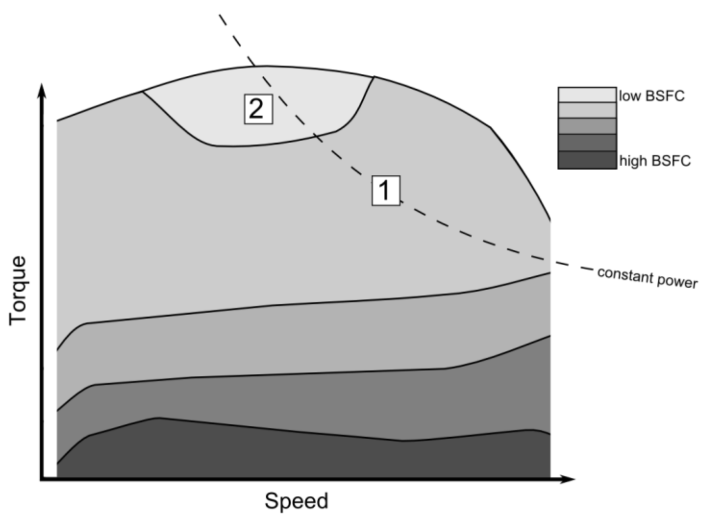

Figure 9: Abstracted BSFC map, with representative original operating point (1) and DDP operating point (2). The larger displacement and high bandwidth control of the DDP allows operation at lower speed and closer to the torque limit, where the BSFC is lower. Higher pump efficiency and lower fan speed reduce engine power demand, which is why the DDP operating point is below the line of constant power.

The same result could not be achieved by simply installing a larger capacity swashplate machine, for example the K3V112 (112cc/rev). While the pump flow demand could be satisfied at lower engine speed (with the associated fan power reduction) the pump efficiency would be lower [7], so average engine power demand would be higher. In addition the ability to operate at high torque and therefore improved BSFC would be affected by the slow speed and poor accuracy of the hydro-mechanical torque control. To illustrate this, Figure 10 shows the response of the excavator system to a step change in pump pressure. The actual response ('DDP') is fast enough that the engine speed only droops by 66RPM. By adding an additional low pass filter to the pump displacement demand the effect of a slower response machine can be investigated. The amount of speed droop that is considered to be acceptable is a matter of operator preference, but it can be assumed that it should be as small as possible.

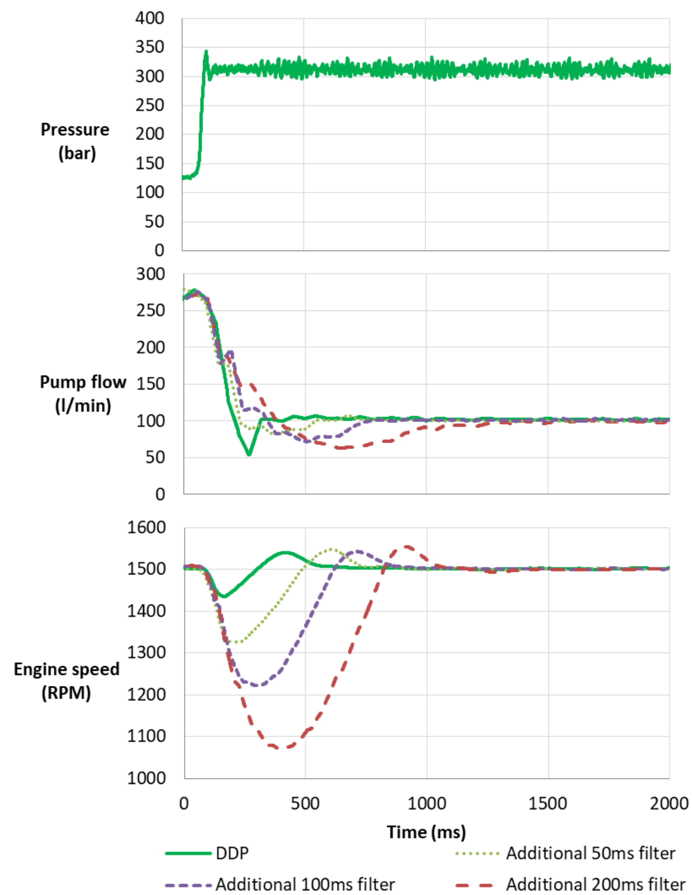

Figure 10: Experimental data showing the response of the excavator system to a step change in pump pressure. Adding an additional LPF shows the effect on engine speed droop of a slower response pump.

Figure 11 shows how the DDP can run the engine at a higher average load than is possible with the original pump. The data is from a trenching cycle with the original pump and with the DDP. In this data the average engine load is 19% higher in the DDP case because the DDP control system is acting as the engine speed governor, allowing the DDP torque to saturate the engine without risk of stalling. Combining the increased torque available to the pump and the increased efficiency of the pump under these operating conditions (around 9%, Figure 4) confirms that the productivity improvement of 28% measured during the digging test (Table 5) is possible.

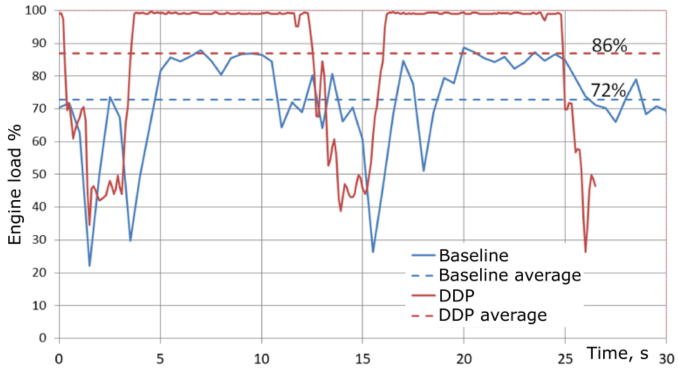

Figure 11: The DDP can operate at almost 100% available engine load, increasing the average power during the cycle and therefore improving productivity.

CONCLUSIONS

By replacing the tandem axial piston swashplate pump in an excavator with a tandem DDP, a fuel saving of between 16 and 21% was achieved. When operated at full speed, the modified excavator is 28% more productive than the un-modified machine while still maintaining a 10% fuel saving. These test results were obtained during side by side testing with a standard machine of the same type, and they are validated using a backward-facing simulation that confirms that the higher pump efficiency, higher pump displacement and improved control combine to deliver the fuel saving. The productivity improvement is attributed to the ability of the DDP to operate closer to the engine torque limit, combined with its higher efficiency.

The modifications to the excavator were limited to a pump swap only. No other changes were made to the hydraulic system or engine. The open centre flow was not reduced. Further fuel savings and productivity improvements would be possible by reducing delivery losses and open centre flow, and combining DDP with energy recovery techniques.

ACKNOWLEDGMENTS

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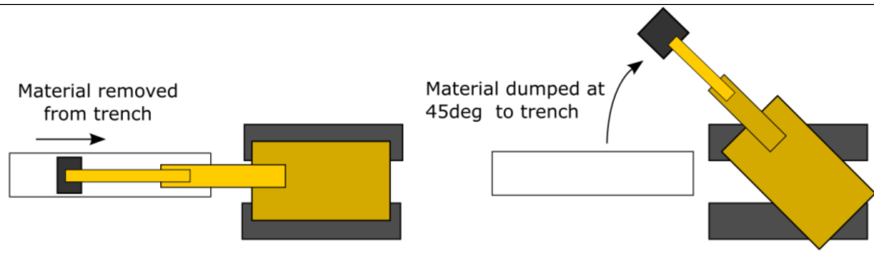
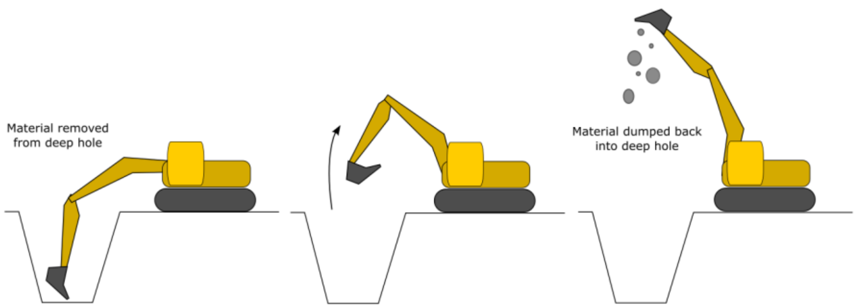
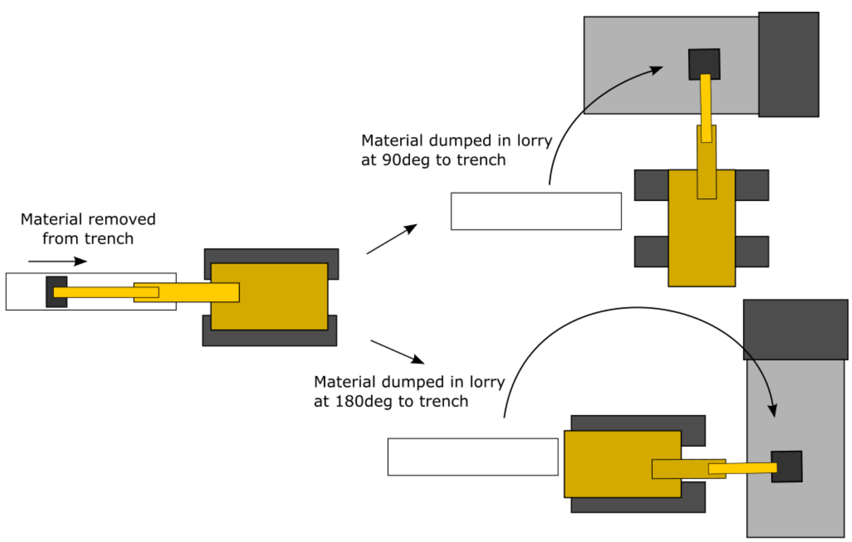
ANNEX A

EXCAVATOR TEST CYCLES

Cycle	Description
Trenching	
Digging	
Lorry loading 90º and 180º	
Tracking	Tracking on flat ground, uphill and downhill.