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ENGINEERING TOMORROW

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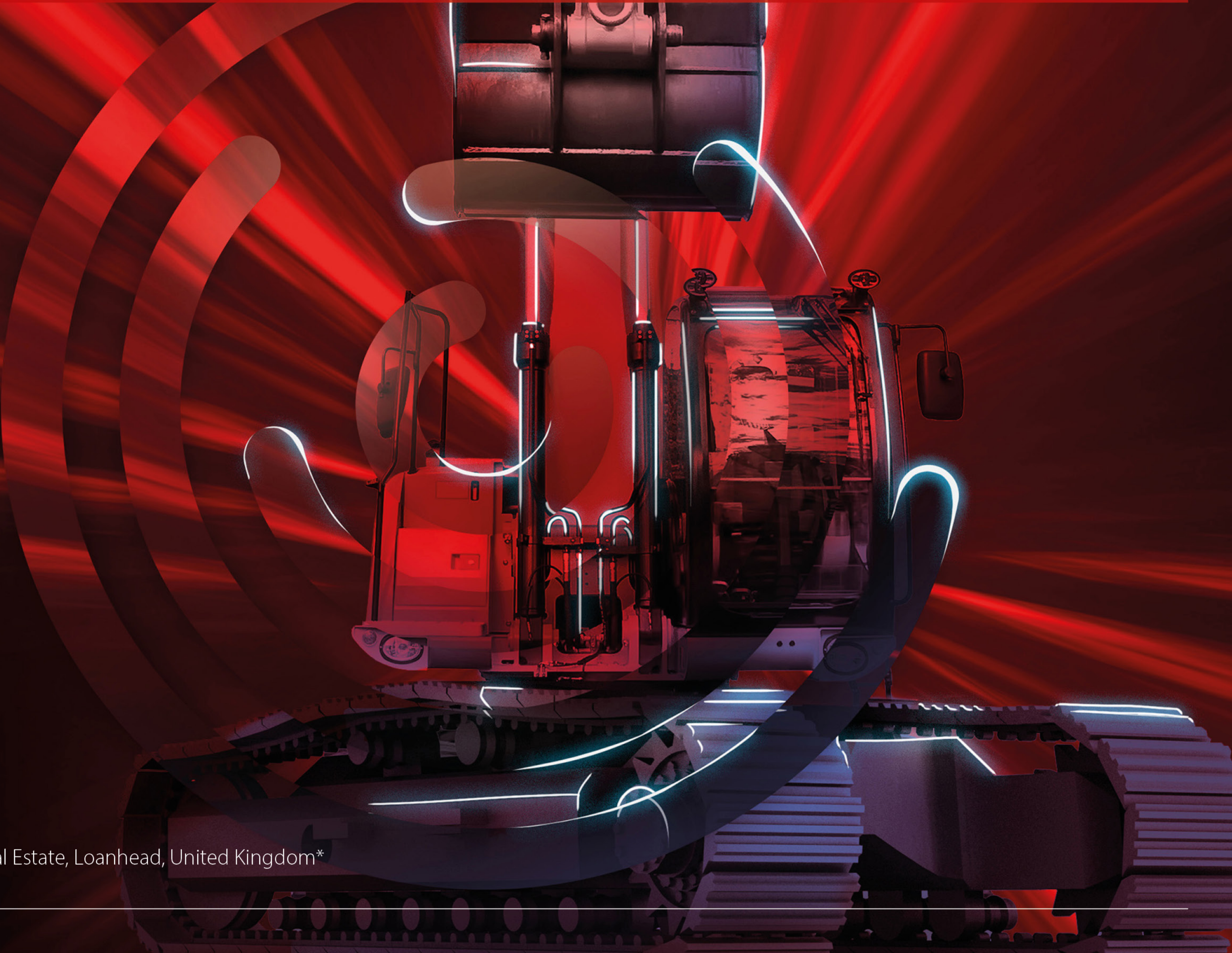
# Application of a Digital Displacement Combined Propel & Work Function Transmission to Off-Highway Machines | Case Study on a 4.5t Forklift



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**This paper presents the design, implementation and testing of a Digital Displacement combined hydraulic propel and work function system, on a 4.5 tonne forklift with a diesel engine. This uses a two service Digital Displacement® pump, operating the machine's propel and working functions with displacement allocation varying in real time depending on the needs of the operator. Through a combination of reduced pump idle losses, replacement of the torque converter drivetrain and more advanced engine and system control, the performance and efficiency of the forklift were improved. During a test cycle based on 'VDI60', fuel consumption was reduced by up to 41% compared to the baseline forklift.**

**Keywords:** *Digital Displacement, Hydraulics, Efficiency, Digital, Pumps*

**Target audience:** *Mobile Hydraulics, Material Handling, Digital Hydraulics*

Mobile hydraulic machines face increasing requirements for system efficiency and reduction of emissions. Small to medium sized machines such as forklifts, tele-handlers and small wheel-loaders are often equipped with separate systems to operate their Propel and Working Functions. Propulsion is usually powered through a torque converter and mechanical gearbox, hydrostatic transmission, or electric drive. The Work Functions (WF) are operated hydraulically, usually with a single hydraulic pump and a directional valve block where flow is split between the various actuators. As original equipment manufacturers (OEMs) strive for electrification, parasitic losses from these separate systems become increasingly problematic while separate prime movers add to the total system cost and may create packaging issues.

Wadsley [1] described how the unique features of Digital Displacement® pumps (DDP) provide an alternative option where a single hydraulic machine can operate both Propel and Work Functions independently, and with the addition of 'Dynamic Ganging', the displacement of the pump can be more effectively utilised. Digital Displacement and Dynamic Ganging have already shown significant benefits in terms of fuel saving and productivity in excavators [2] [3].

This paper describes a system-level solution for smaller machines, the 'Digital Displacement Combined Propel & Work Function System', incorporating a multi-service DDP, hydraulic manifolds and a hydraulic motor, which can replace the conventional systems described above. The system is demonstrated on a torque converter forklift truck which was tested for fuel consumption and performance before and after conversion, with comparative results presented. Backwards-facing simulation models, based on recorded test data, were used to analyse energy use during the fuel consumption tests and recommendations are made for further potential improvements.

# 2. Baseline Forklift Architecture

**The forklift chosen for the Digital Displacement (DD) system implementation was a CAT DP45NB, with a rated load of 4.3 tonnes and powered by a 55 kW diesel engine. Figure 1 shows a simplified schematic of the DP45NB, including key components.**

## 2.1 Drivetrain

The forklift was driven through a torque converter on the engine output, a two-speed automatic gearbox, a differential and final gear reduction to the wheels. The torque converter input shaft could be decoupled from the engine by a clutch when the vehicle was in neutral or by the 'inching' pedal, which applied the brakes and clutch.

## 2.2 Work Function System

Two fixed displacement hydraulic gear pumps were connected to the engine power take-off (PTO). The first, with 52 cc/rev displacement, provided flow to the Work Function manifold. A priority valve in the manifold prioritised flow to a closed-centre steering unit connected to the steering cylinder on the rear wheels. When the work functions were not in use, any flow not used for steering drained through open-centre spool valves for each function (lift, tilt, and side-shift of the forks) and back to tank as shown in Figure 2. When the operator moved one of the WF finger control levers, the corresponding spool valve was actuated electronically, and flow was diverted to that function's hydraulic rams.

The second gear pump, with 4.5 cc/rev displacement, operated on a separate hydraulic circuit, controlling the wet disc service brakes and parking brake.

## 2.3 Control System

The baseline truck used a high-level system controller (VCM) to manage all the inputs from the operator and system and controlled the outputs for the transmission, work function manifold and safety systems. Two CAN buses were used for communication between the VCM and other controllers. The main human-machine interface (HMI) consisted of the pedals (accelerator, brake, and inching), the FNR (forward, neutral, reverse) switch, the WF finger controls, and a parking brake switch. The accelerator pedal was connected to the engine controller (ECM) which managed the diesel engine in speed control mode based on the accelerator pedal position. When the truck was in neutral, the torque converter was de-clutched to prevent internal energy losses. In forward or reverse, the torque converter allowed the engine and gear box input speeds to vary, the speed difference generating torque at the gear box input.

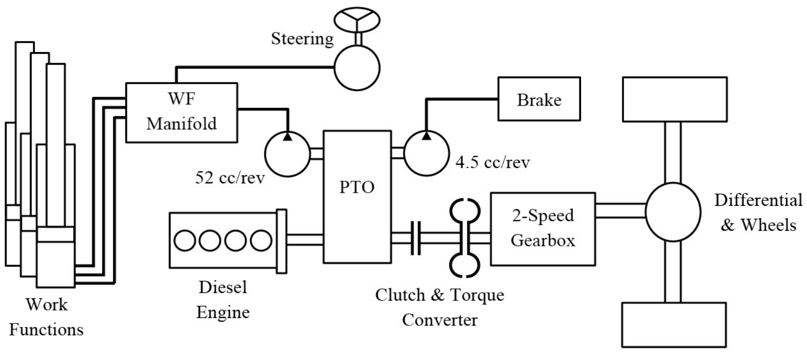


*Figure 1 - Simplified Schematic of CAT DP45NB Baseline Forklift*

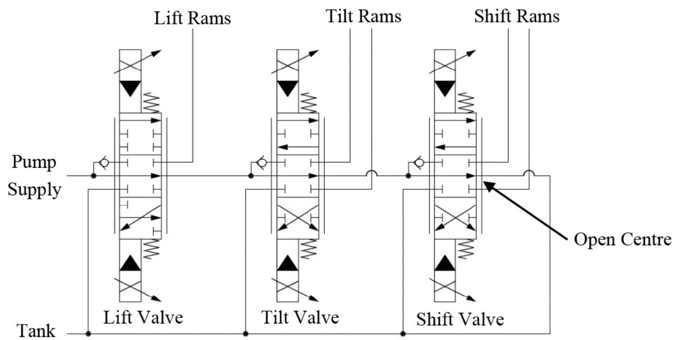


*Figure 2 – Simplified Schematic of Work Function Electro-Proportional Open-Centre Spool Valves*

Since the engine speed was controlled by the operator, they had to increase it with the accelerator pedal to operate the work functions at high speed. Once a WF spool had moved enough to close off the open centre, actuator speed was roughly proportional to engine speed due to the fixed displacement WF pump.

### 2.4 Baseline Forklift System Issues

The baseline forklift had several sources of inefficiency and issues with controllability. The fixed-displacement WF pump geared directly to the engine was a constant source of significant parasitic losses when not in use, particularly when the engine speed was raised. Using the method described in section 5.1, it was calculated that the pressure drop through the work function manifold, when not in use, created average losses of 2.21 kW during fuel consumption testing, with peaks of 8 kW at high engine speed. The linking of the WF pump speed and wheel speed (albeit through the torque converter) meant that simultaneous operation of propel and work functions was not optimised or efficient. Controlling both functions at the same time required a skilled operator, while energy was lost in the torque converter or through throttling in the work function manifold. With the mechanical transmission, the engine speed was linked to the wheel speed so the engine could not operate at the most efficient point for a given torque load. At 20 km/hr with no load on the forks, the engine had to run at 2200 revs/min while producing 37 kW of net power. From the Brake-Specific Fuel Consumption (BSFC) map, the same amount of power could have been produced at 1170 revs/min while using 13% less fuel. The DDP system was designed to combat these issues by decoupling the operation of the functions and reducing losses in the system.

# 3. DD Forklift Architecture

Figure 3 shows the forklift after the conversion to the DD Combined Propel and Work Function system and Figure 4 shows a simplified schematic of the installed system. The DDP and all other hydraulic components are electro-hydraulic, using solenoid valves for actuation and are managed by Danfoss PLUS+1® and DPC12 DDP controllers in conjunction with the CAT stock controllers.

### 3.1 Multi-Service DDP with Dynamic Ganging

DD system replaced the torque converter, gearbox, and gear pumps to provide power to both the Propel and Work functions and is based around a single DDP096 multi-service pump with a total displacement of 96 cc/rev. The DDP096 is a radial piston pump where variable displacement is achieved by real-time selective activation of the twelve cylinders using high-speed mechatronic valves by an embedded controller (DPC12). The twelve cylinders are arranged into four groups of three, known as 'pumplets', each with its own output flow gallery. Depending on how these galleries are connected in the pump endcap, the DDP096 can be configured with multiple services (groups of one or more pumplets) operable at independent pressures and flows.

The DDP configuration used for this forklift application, see Figure 4 & Figure 5, has 48 cc/rev of displacement dedicated to the Propel function, 24 cc/rev to Work Functions and 24 cc/rev to the 'Dynamic Ganging' service. Dynamic Ganging uses a manifold connected to the DDP endcap to reallocate the displacement of this service to either Propel or Work Function in real time, depending on the needs of the operator. In applications where the different functions do not tend to be used heavily at the same time, Dynamic Ganging allows a smaller total pump displacement to satisfy the requirements of the operator. Pellegri et al. [3] showed how a similar system could be successfully applied to excavators, providing further fuel savings compared to a simpler DD implementation [2].


*Figure 3 - Converted Digital Displacement Forklift*

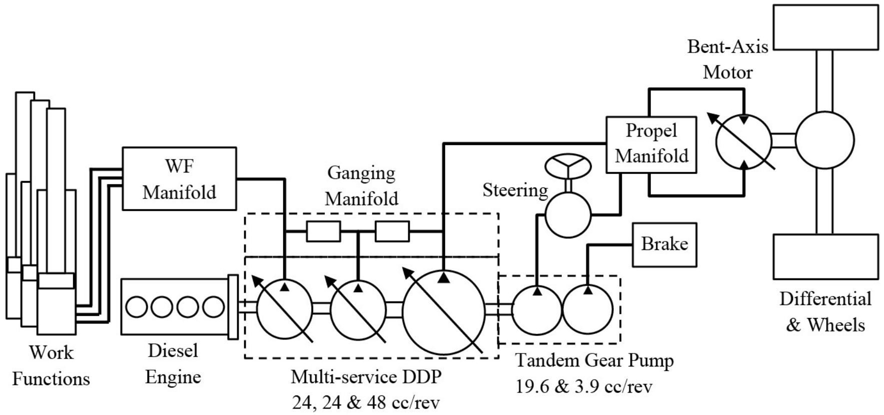


*Figure 4 - Simplified Schematic of DD Combined Propel & WF System*

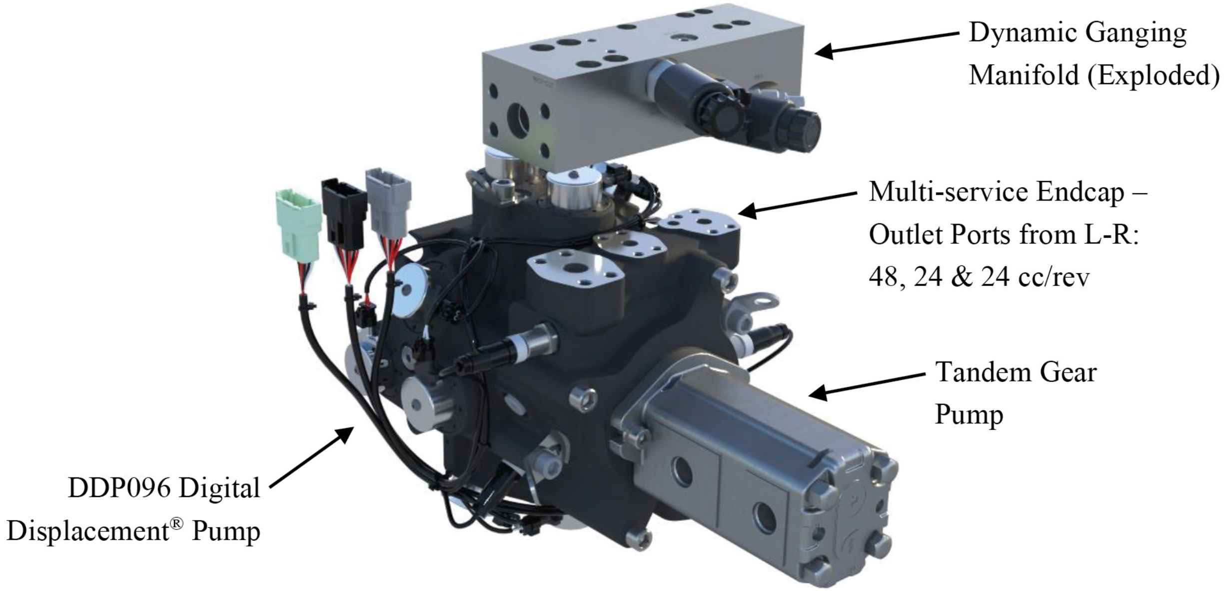


*Figure 5 - Multi-Service DDP096 with Dynamic Ganging Manifold and Gear Pumps*

**3.2 DD Work Function System**

The DDP WF service was connected to the input to the stock open-centre work function manifold with the DDP controlled to only pump to WF when commanded. The steering priority valve was removed, and a new, open-centre steering unit was fed from the 19.6 cc/rev gear pump mounted on the back of the DDP.

The tilt spool solenoids were disconnected from the stock controller and driven directly from the DD system controller as the stock controller had a limit on the tilt solenoid current that was lower than the other functions and prevented the spool from fully closing off the open-centre drain, resulting in wasted flow while tilting. The impact of this modification is assessed in section 5.2.

**3.3 DD Open-Circuit Propel System**

The DD Propel system uses a conventional closed-circuit motor(s) to operate the Propel function. In this conversion, a variable displacement (0-80 cc/rev) Danfoss H1B bent-axis motor was mounted to the input shaft of the stock differential and final gear reduction, but a pair of low speed (e.g. cam-lobe) motors also could be used to drive the wheels directly.

The DDP096 is an open-circuit pump and uses a proprietary manifold, the 'Propel Manifold' in Figure 4, to operate the closed-circuit motor. The Propel Manifold uses a patented arrangement [4] of solenoid-controlled hydraulic valves to set the rotation direction of the motor when applying positive torque (accelerating in forwards or reverse), and the hydrostatic braking torque when decelerating. When positive torque is not required, the Propel Manifold circuit allows flow returning from the motor to recirculate back to it again without passing through the DDP, meaning that the DDP only needs to pump to propel when positive torque is required. Flow from the return port of the open-centre steering unit was supplied to the propel block to guarantee the minimum inlet pressure at the closed-circuit motor A and B ports.

**3.4 Brake System**

A 3.9 cc/rev gear pump supplies flow to the stock brake manifold to operate the parking brake, and the wet disc service brakes which were kept for emergency use, although hydrostatic braking made them redundant in normal operation.

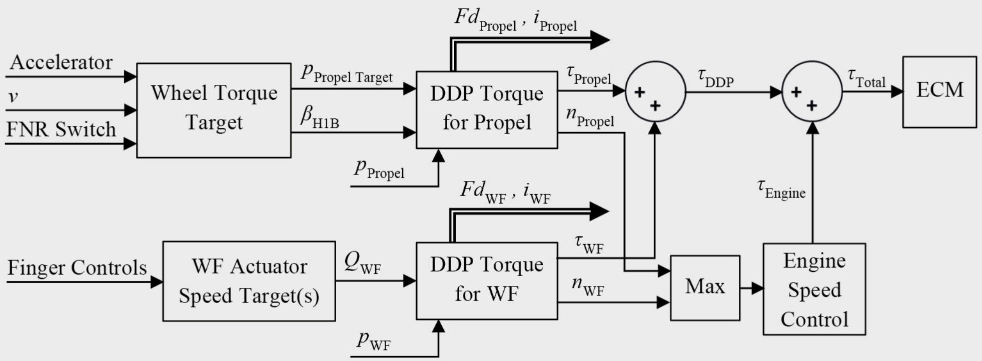


*Figure 6 – Simplified Schematic of DD System & Engine Control*

## 3.5 Control Strategy

A well-designed control system was key to maximising the benefits of the DD system hardware with the PLUS+1® system control software developed in Simulink using a model-based design approach. Due to the extremely fast response of the DDP, it was necessary to take over the closed-loop speed control of the engine in the system controller, including a feed-forward signal of the DDP torque load.

The basic operation of the control system is described below and shown in Figure 6:

1. Read the inputs from the operator and the current state of the machine and DD system components

2. Determine the operator's requests in terms of targets for torque at the wheel and WF actuator speeds

3. Calculate DDP speed and torque required to meet operator demands for Propel and WF

4. If necessary, calculate additional torque needed to increase the engine speed, sum all the torque loads, and feed the total demand to the ECM (now running in torque control mode)

5. Apply a delay to the DDP response, then send displacement commands for Propel and WF to the DPC12 and drive the other outputs of the DD system – ganging and propel manifold valves, H1B motor displacement control, etc.

The Propel system has two control modes based on behaviour of hydrostatic (HST) and torque converter (T/C) forklifts, allowing an operator to choose the drive feeling they are most familiar with. Each mode uses a look-up table of torque at the wheel depending on the current vehicle speed and accelerator position, which is used in step 3 of the control system described above.

In HST mode, with forward selected, and the parking brake released, the forklift will not move until the operator presses the accelerator and when the accelerator is lifted, hydrostatic braking will be applied by valves in the Propel Manifold. In T/C mode, with forward selected and the parking brake released, a small amount of torque will be applied at the wheel and the forklift will start to drive at low speed on the flat (i.e. creeping). Pressing the accelerator will increase the torque at the wheel and when the accelerator is released, the truck will coast gradually down to creeping speed. If the brake pedal is pressed, hydrostatic braking will be applied initially followed by the service brakes as the pedal is pressed harder. The electronic control of the DD system components enables this flexible behaviour and simplified the creation and tuning of the different drive feelings.

The Work Function service is flow controlled in the system controller. Based on how far the operator moves one or more finger controls, a flow demand is calculated and if it cannot be met with the currently allocated WF DDP displacement, the dynamic ganging service may be reallocated to WF and/or the engine speed will be increased automatically to compensate.

# 4. Testing

**To quantify and analyse any improvements following conversion to the DD Combined Propel & Work Function System, a range of tests were carried out on the baseline truck and then repeated on the converted machine, centred around fuel consumption and aspects of performance including maximum acceleration and lifting speed.**

### 4.1 Fuel Consumption Test Definition

Fuel consumption was measured using a test cycle based on that prescribed in VDI 2198 [5], commonly known as “VDI 60”. The forklift was loaded with the rated load of 4.3 tonnes on the forks and drove the cycle shown in Figure 7. Starting in Load Bay 1, with forks lowered and mast tilted back, the operator would reverse out and to their right, drive forwards and stop at Load Bay 2, tilt the mast to vertical, tilt the mast back to the end stop, lift the forks to 2 m, lower the forks to 20 cm and then repeat the pattern back to Load Bay 1. The cycle would be completed in 60 ± 2 seconds by controlling the driving and lifting. Ten cycles were completed for each test with the measured fuel volume consumed converted to a consumption rate in litres per hour (l/hr). Multiple 10-minute tests were carried out for both systems with the best result for each presented and used for further analysis. This test procedure differed from [5] in that the lifting height is 1.8 m rather than 2.0, that the tests were carried out for ten minutes rather than an hour and that the load bays were on opposite sides of the driveway to account for a small gradient across the test area. The same procedure was followed for the baseline and converted trucks.

### 4.2 Performance Test Definitions

Acceleration tests were performed over approximately 100 m of ground with a gradual rise, so the test was carried out in both directions and average values of time to reach certain speeds were calculated. Speed was measured using the speed sensor on the hydraulic motor, converted to vehicle speed, considering the gear ratio from motor to wheel and the wheel radius. Lifting speed was measured with a pair of string potentiometers, attached to hydraulic rams on both stages of the two-stage mast assembly. For the maximum lifting speed tests, the operator commanded full lifting speed and values of maximum speed were obtained once the engine speed had increased and stabilised.

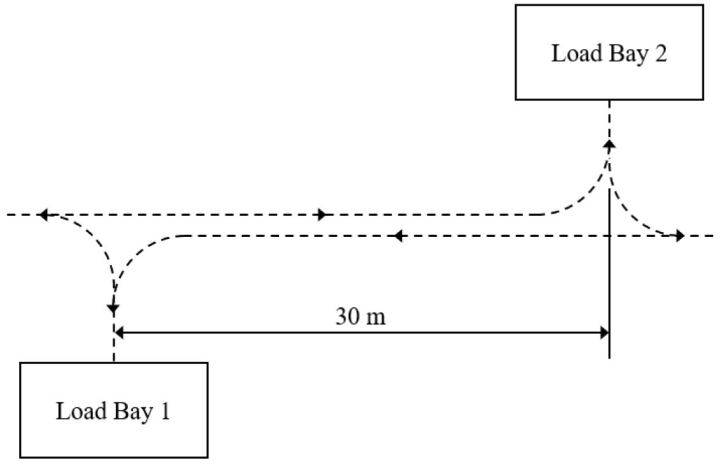


*Figure 7 - Fuel Consumption Driving Pattern*

### 4.3 Baseline Truck Testing

A DEWE 43 data acquisition module was installed to record test data. CAN messages from both buses, such as vehicle speed, engine speed, engine torque and HMI inputs were recorded. A diesel flow meter was added in the fuel system, which measured fuel flowing to and from the common rail system and output the difference to the DEWE 43. The string potentiometers signals were also recorded by the DEWE43.

| Test Description | Baseline Result | DD Result | Improvement (%) |
|---|---|---|---|
| Fuel Consumption – Rated Load (l/hr) | 7.97 | 4.69 | 41.2 |
| Acceleration – Time to 10 km/hr – Rated Load (s) | 4.44 | 2.98 | 32.9 |
| Acceleration – Time to 10 km/hr – No Load (s) | 3.22 | 2.03 | 37.0 |
| Acceleration – Time to 15 km/hr – Rated Load (s) | 9.24 | 5.95 | 35.6 |
| Acceleration – Time to 15 km/hr – No Load (s) | 5.62 | 3.98 | 29.2 |
| Acceleration – Time to 15 m – Rated Load (s) | 6.89 | 5.99 | 13.1 |
| Acceleration – Time to 15 m – No Load (s) | 6.01 | 5.06 | 15.8 |
| Maximum Lifting Speed – Rated Load (m/s) | 0.49 | 0.52 | 6.1 |
| Maximum Lifting Speed – No Load (m/s) | 0.53 | 0.53 | 0 |

*Table 1 - Test Result Summary*

### 4.4 DD Truck Testing

Following the conversion of the forklift to the DD system, the DEWE 43 was used again, with the inputs from the baseline testing plus additional pressure sensors added to the propel, work function, lift, steering and brake hydraulic circuits and to several points on the Propel Block. Data from the PLUS+1® and DPC12 controllers was recorded via CAN, including DDP service displacements, Ganging and Propel Manifold valve states and vehicle speed measured at the H1B motor.

For the DD truck fuel consumption tests, the maximum lifting speed was reduced in the controller so that full speed lifting could be commanded without completing the test cycle too quickly. If full speed lifting was not commanded, the spool valve would not close off the open centre and flow was lost to tank, similarly to the tilt flow loss in the baseline truck mentioned in section 3.2. Lift flow loss was prevented in the baseline truck fuel tests because the engine speed could be controlled separately (using the accelerator pedal) to the spool position. If the project had been in collaboration with an OEM, it would have been possible to optimise the control of the WF valves for DDP and to prevent flow losses. Since the OEM control could not be changed in this work, the use of modified software could be justified to allow similar lifting speeds during the baseline and DD tests.

### 4.5 Test Results

The results for fuel consumption and performance testing of the baseline and converted trucks are given in Table 1, with a calculation of percentage improvement included.

The most significant result of the DD System conversion project was the 41.2% fuel consumption reduction while carrying out the same useful work. Further analysis of the fuel consumption test data is included in section 5.

Despite having less pump displacement available for work function (48 cc/rev vs 52 cc/rev), the high volumetric efficiency of the DDP helped the DD forklift match or exceed the maximum lifting speed of the baseline truck. This was particularly noticeable when fully loaded where work function pressure was higher and therefore leakage in the gear pump would have been greater.

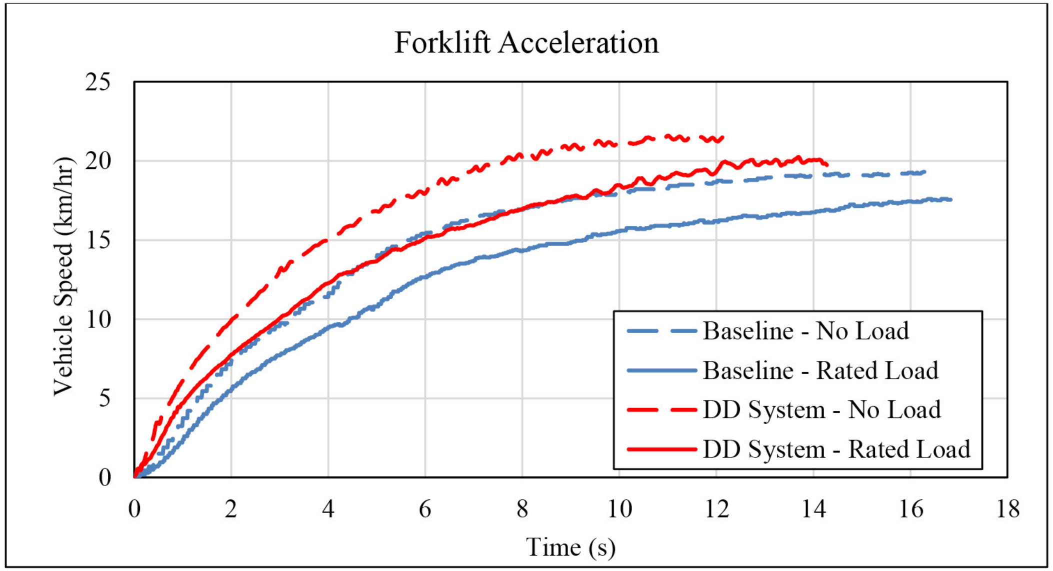


*Figure 8 - Acceleration Comparison of DD and Baseline Trucks*

Figure 8 shows timeseries of vehicle speed for the baseline and DD System forklifts, both with no load and the rated load (4.3 tonnes) on the forks.

The improvement in acceleration performance from the DD System is clearly apparent with greater acceleration throughout the speed range of the vehicle and a higher maximum speed. The DD forklift, fully loaded, can accelerate similarly to the baseline forklift with no load on the forks. Throughout all the tests, the engine was running close to, or at, its torque limit so these results show that the DD system is more efficient than the baseline while driving, as more of the torque could reach the wheels and accelerate the truck.

**The increase in acceleration from more effective use of the engine power leads to higher productivity**

# 5 Results Analysis

**To further investigate the fuel consumption results and study energy usage in each forklift, backwards-facing simulation models were created in MATLAB & Simulink. The models were fed timeseries data recorded during the fuel consumption tests whose results were reported in Table 1. The data were used to calculate torque loads on the engine and fuel consumption using BSFC data for the diesel engine. The calculated results could then be compared with measured fuel from the fuel meter and torque reported by the engine over CAN.**

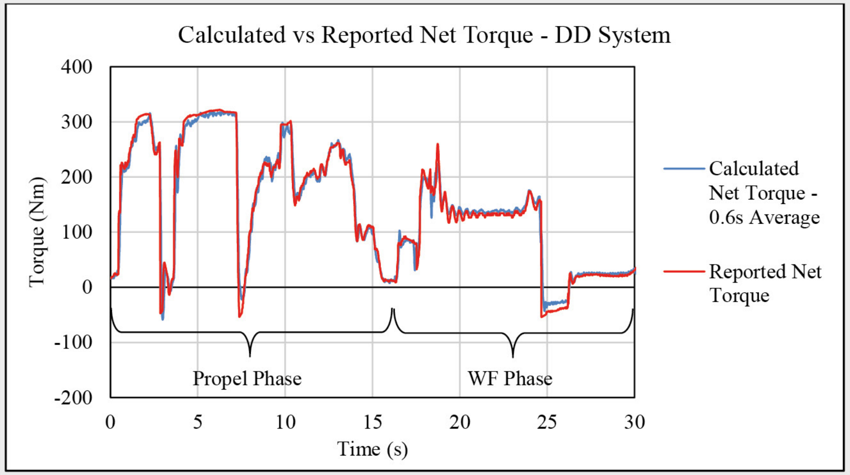


*Figure 9 - Calculated and Reported Net Engine Torque during a Half Cycle of the Fuel Consumption Test*

### 5.1 Energy Usage Simulation

For the baseline truck, the data captured was insufficient to simulate all the engine loads, such as the hydraulic pumps and torque converter. For the DD truck, a more comprehensive model could be produced, including calculations of the torque required by the DDP, gear pumps, fan, and to accelerate the flywheel. The DDP and gear pump torques were calculated from shaft speed, displacement, and pressure data, using methods based on [6] & [7] respectively.

The total calculated torque compared well with the net torque reported by the engine over CAN, as shown in Figure 9. The torque signals were integrated over the test and the two values differed by 1.2%. Since the theoretical and reported torques matched well, using reported torque from the baseline truck for analysis could be justified. The simulation data was passed through the BSFC map, with a 3.9% error between the calculated fuel and the volume measured by the fuel flow meter.

Figure 10 shows the average power reported by the ECM during the fuel consumption test, split into power consumed inside the engine (including the fan) and the net power (all loads outside the engine) used during the Propel and Work Function phases of the test cycle. The results show that the total energy used by the DD System was 41% less than the baseline, reflecting the fuel consumption results, and that the DD system reduced system losses while driving and while using the work functions.

The testing and simulation carried out showed that conversion to Digital Displacement improved the efficiency of the baseline machine with its gearbox and torque converter drive train and fixed displacement hydraulic pump, with benefits to fuel consumption and performance.

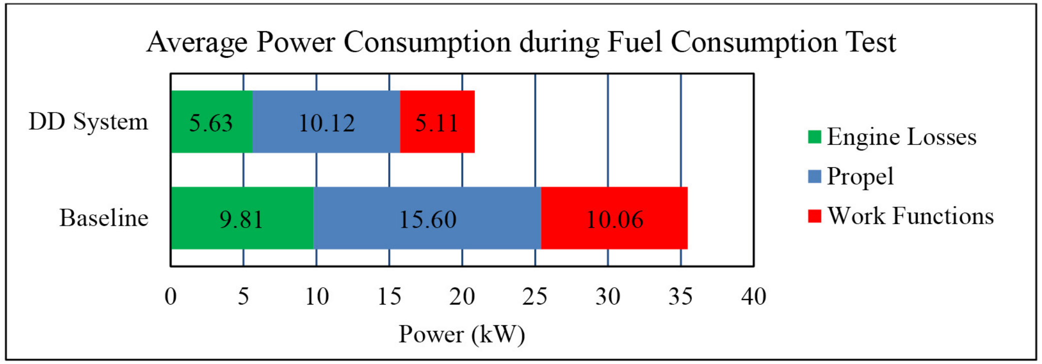


*Figure 10 – Calculated Average Power Consumption by Function during Fuel Consumption Test*

The main alternative to this type of truck is the hydrostatic forklift, where propel and working functions are typically powered by separate pumps, on the same shaft. A multi-service DDP also provides a distinct advantage over this kind of system due to the very low idle losses of the DDP compared to swash-plate pumps, where larger displacement pumps could be installed to improve hydrostatic transmission efficiency at high speed, or reduce engine speed, without a significant penalty of parasitic losses. With Dynamic Ganging the pump displacement can be optimised, and the total installed displacement may be reducible, further reducing parasitic losses.

## 5.2 Tilt Flow Loss Impact Simulation

As mentioned in section 3.2, the tilt solenoid control was modified as the part of the DD conversion work, providing the same maximum solenoid current to all three functions, allowing the tilt spool to close off the open centre – see Figure 2. This meant that the DD truck lost significantly less energy when tilting with heavy loads, compared to the baseline where most of the high-pressure flow from the gear pump throttled to tank through the partially closed open centre, rather than performing useful work in the rams. In the baseline forklift, the operator controlled the engine speed, and the tilting speed could not be limited by the VCM, so throttling was used instead. Since the DD system controls both engine speed and displacement for the Work Functions, maximum tilting speed could be set in the control software, based on the performance of the baseline machine, and the valve hardware used more efficiently.

The backwards-facing simulation was expanded to investigate the potential impact of having changed the tilt control in the baseline truck, assuming this would result in the same tilting speed as the DD truck. A parallel model was created which used mean values of the DDP Work Function flow and pressure, during tilting forwards and backwards, from the DD truck data for comparison. The engine speed for the theoretical baseline truck was calculated by matching the DDP and gear pump model flows, for tilting in either direction. Models of the gear pumps, fan, and flywheel were used to calculate the engine power and fuel consumption. The results of this simulation are shown in Table 2. Ignoring the error between measured and simulated fuel consumption, the DD system would still have used 36% less fuel if the same tilt valve control had been used on the baseline truck.

| WF Gear Pump Simulation Results | | Simulation - Using Real Engine Data | Simulation - No Tilt Flow Loss | Difference | % of Test Spent Tilting |
|---|---|---|---|---|---|
| Average Pump Mechanical Power During Tilting (kW) | Forwards | 9.39 | 5.83 | 3.56 (37.9%) | 3.6 |
| | Backwards | 36.22 | 19.49 | 16.73 (46.2%) | 5.8 |
| Fuel Consumption (l/hr) | | 7.74 | 7.32 | 0.42 (5.4%) | - |

*Table 2 - Results of the Tilt Flow Loss Simulation*

# 6. Engine Downsizing

**One way to take further advantage of the DD system efficiency could be to use a downsized internal combustion (IC) engine. As shown in Figure 10, engine losses account for more than a quarter of the energy consumed in both the DD and baseline trucks and if the size of the engine were decreased, these losses would also be reduced. A smaller engine is also likely to be cheaper, reducing the overall machine cost, to the benefit of the manufacturer. To investigate the impact on performance of a downsized engine, a control software mode was created which limited the torque demand to the engine to match the torque curve of a 36 kW engine. To compensate for a larger flywheel inertia, higher torque was allowed during engine acceleration.**

**1.39 kW.**

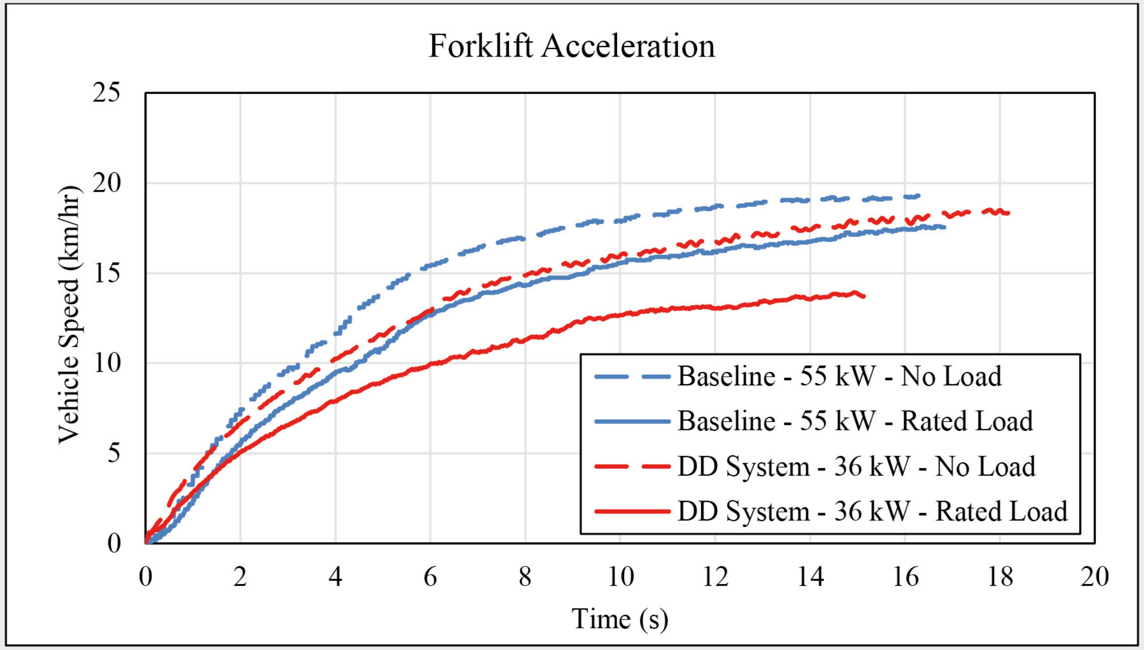


*Figure 11 - Acceleration Comparison of DD System with Simulated 36 kW Engine*

$$m = \frac{P_{\text{lim}} - P_0}{u \cdot g}$$

Figure 11 shows acceleration results of the DD forklift with the simulated 36 kW engine compared to the 55 kW engine baseline system. With the power limit, the acceleration of the DD truck was reduced, particularly at higher speeds. However, the disengaged wet disc brakes in the forklift axle may be adding significant losses to the transmission, as suggested by Morris et al. [8], who presented a model to estimate the viscous losses between the plates of a disc pack. Using this method, the brake oil temperature and disc spacing were found to be critical to the power loss. Since the wet disc brakes were made mostly redundant by the DD system, the oil would not be heated as much as in the baseline truck where the brakes were being used so it is expected that the disc losses in the DD truck would be worse than in the baseline truck. A full investigation of the brakes was beyond the scope of this work but using parameters suggested in [8], the viscous losses were estimated to be on the order of 0.75 kW at 15 km/hr with 25 °C oil compared to 10 kW of rolling resistance at the same speed. Removing the brakes would be a good avenue for future work with the possibility to remove further losses if the hydraulic circuit could be simplified and the brake pump removed. Simulation showed that, at maximum engine speed, the brake circuit consumed 1.39 kW.

Maximum lifting speed with rated load was slightly reduced (0.47 m/s compared to 0.52 m/s with 55 kW) as the power limit was reached but with no load the speed was unchanged. Test data showed that 16.5 kW (P0) was required to lift at 0.52 m/s with no load on the forks. Using Equation 1, with the power limit (Plim) of 36 kW, the maximum load (m) that could be lifted at a speed (u) of 0.52 m/s was found to be 3.82 tonnes, showing that engine downsizing to 36 kW had little impact on work function performance.

# 7. Conclusions and Further Work

This paper describes the conversion of a diesel engine forklift from a mechanical transmission and hydraulic gear pumps to the Digital Displacement Combined propel and Work Function System. By reducing the losses in both functions and utilising more advanced control systems, the DD truck consumed 41% less energy and fuel than the baseline truck on a test cycle based on "VDI 60", while surpassing it in performance tests. This substantial improvement in efficiency would significantly reduce the running costs to a machine owner as well as their carbon dioxide emissions for an IC engine machine. The DD system also enabled greater flexibility in driving behaviour as the propel response is dictated by control software rather than the hardware of conventional machines.

The system could be particularly interesting to enable electrification of vehicles where direct electric propel drive is not practical. Packaging limitations, or overall system cost with two separate high-power drives for propel and WF might be prohibitive. The high efficiency of the DD system would mean that energy consumption for propel could be comparable with a direct electric drive system.

While the results of the DD System tests presented above are very encouraging, the authors believe that further improvements in fuel consumption and/or performance are possible, by improving the integration of the DD system into the machine in collaboration with an OEM.

Firstly, the stock brake system is redundant in HST mode and is partially replaced by hydrostatic braking in T/C mode too. Removing the wet disc service brakes and simplifying the hydraulic circuit for the parking brake would remove more parasitic losses from system.

Secondly, the control of the work function spool valves is done by the stock VCM and was not optimised for DDP. Working with an OEM should allow optimised, combined control of the DDP and spool valves to reduce throttling when the operator requests low speed actuator movements. A WF system capable of working at higher pressure would also be advantageous as engine speed could be further reduced.

Thirdly, dynamic ganging of more pumplets, like the 'Elastic Pump' in [3], would provide more flow to a function operating on its own, meaning the engine speed could be further reduced during high-speed driving or lifting, which would reduce losses and save more fuel. Danfoss has developed prototype multi-pumplet dynamic ganging hardware to do this.

Finally, by implementing some of the system improvements suggested above it would be possible to significantly downsize the engine while matching the performance of the baseline torque converter truck. The preliminary test, carried out by simulating a 36kW engine as presented in section 6, show that downsizing is possible with some impact on performance. Fitting a real downsized IC engine and optimising the system as recommended could unlock further fuel savings, with no impact on performance, while reducing the overall system cost.

Given the proven benefits to vehicle manufacturer and operator, and the potential for future improvement, the Digital Displacement Combined Propel and Work Function System represents a significant step forward in tackling the challenges facing modern off-highway equipment.

# Nomenclature

| *Variable* | *Description* | *Unit* |
|---|---|---|
| $\beta$ | Displacement Fraction for Conventional Hydraulic Machines | [-] |
| $\tau$ | Torque | [Nm] |
| $Fd$ | Displacement Fraction for DDP Services | [-] |
| $g$ | Gravitational Acceleration | [m/s$^2$] |
| $i$ | Control Signals including Solenoid Current and CAN Messages | [-] |
| $m$ | Load Mass | [tonnes] |
| $n$ | Shaft Rotational Speed | [rev/min] |
| $p$ | Hydrostatic Pressure | [bar] |
| $P$ | Power | [kW] |
| $Q$ | Volumetric Flow Rate | [l/min] |
| $u$ | Mast Lifting Speed | [m/s] |
| $v$ | Vehicle Speed | [km/hr] |

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