A Pump Decoupled Architecture to Allow Increasing Energy Efficiency of Hydrostatic Transmission Solution based on Fixed Displacement Secondary Units

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Abstract

Despite being inherently throttle less system, Hydrostatic Transmission (HT) for off-highway applications suffers from low efficiency when the primary and secondary units do not operate at their optimal efficiency point. This paper proposes a novel HT architecture that decouples the primary unit's operating point from the load (i.e., external torque and rotational speed) allowing it to always operate at a more favorable point, while maintaining roughly the same energy efficiency for the secondary unit, resulting in an overall increase of the total HT efficiency. Essentially, the decoupling is achieved by integrating a hydraulic accumulator into the supply line and controlling the primary unit to regulate supply pressure within a narrow range rather than delivering a reference flow. The supply line is connected to a fixed displacement secondary unit, with its angular velocity controlled by another hydraulic rotary actuator connected to the same shaft. The paper details the architecture design, the power flow management and the architecture's simple and scalable implementation for a range of applications.

To assess the energy efficiency advantage of the proposed solution, the reference case of the basecutter system in a sugarcane harvester is considered. A lumped-parameter model for both the baseline commercial solution, validated based on available experimental data, and for the proposed system were developed. Results for a realistic drive-cycle show a 15% reduction in energy consumption compared to the baseline flow-supply series HT used in the reference machine, thus proving the effectiveness of the proposed system.

Keywords: Hydrostatic Transmission, Off-Road Machines, Pump-Load Decoupling, Power Management

1 Introduction

A hydrostatic transmission (HT) is a throttle-less hydraulic control system for driving rotary actuators. Unlike conventional mechanical transmissions, it provides a continuously variable transmission (CVT) ratio, making it ideal for high-load applications. HTs are widely used to power the drive and work functions of off-highway machinery in the earthmoving and agricultural industries. However, HTs are also characterized by relatively low efficiency compared to mechanical or electric transmissions predominantly used in on-road vehicles, leading to higher fuel consumption and emissions in machines that utilize them. Rising fuel costs and growing environmental concerns in recent years highlight the need for more efficient HT architectures for off-highway machines.

Conventionally, HTs are classified into two configurations: Flow Supply Primary Control (FSPC) and Pressure Supply Secondary Control (PSSC), as illustrated with simplified schematics in Figure 1. In the FSPC configuration, the fractional displacement of the pump (ϵ_P) determines the flow going to the fixed displacement motor, essentially controlling the motor' shaft speed (n_M). Also, the supply line pressure (p_H) is governed by the external torque (T_{Ext}) applied on the motor's shaft. The design simplicity of this configuration enables ease of implementation and high reliability, making it a common choice in off-road machinery. However, it exhibits slow dynamic response due to oscillations caused by the high compliance of the connecting hose and its expansion under pressure variations [1]. Additionally, the direct flow coupling between the pump and motor restricts its application to single-drive function.

In the PSSC configuration, the supply line pressure can is kept constant. Thus, fractional displacement of the motor (ϵ_M) determines the torque generated by motor (T_M) and is controlled to get a desired motor's shaft speed using a feedback loop. The fractional displacement of the pump (ϵ_P) is controlled to maintain the supply line at a fixed pressure using hydro-mechanical feedback. To stabilize the supply line pressure during transients, when pump response may be insufficient, an accumulator is often integrated into the supply line to increase its capacitance. This configuration enables fast dynamic response with minimal oscillations. Additionally, the absence of direct flow coupling between the pump and motor allows a single pump to drive multiple actuators connected to a common supply line. However, the increased system complexity due to additional components and advanced control requirements limits its application to niche domains demanding precise control, such as large winches, marine cranes, and offshore heave compensation systems [2].

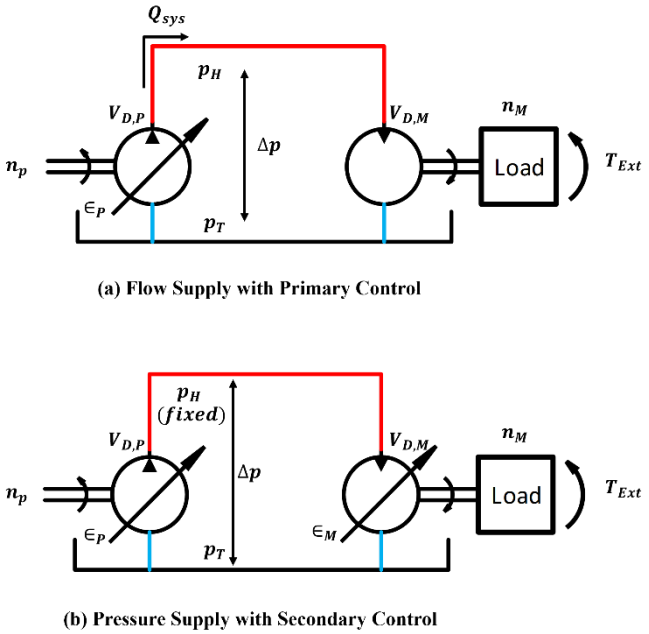

Figure 1: Conventional HT configurations.

The steady-state transmission efficiency of an HT is largely affected by the energy losses in the primary and secondary unit used [3]. These units attain peak efficiency within a narrow operating range, typically near full displacement of the hydraulic unit and rated pressure. However, in both configurations, the operating conditions of the pump and motor are dictated by the HT's operating condition, i.e., motor's shaft speed and external torque. As a result, the wide operating range typical of off-highway machines often forces these units to operate under suboptimal conditions, such as partial displacements and lower pressure differentials, which can significantly reduce HT transmission efficiency, in some cases to below 40% [4]. Therefore, HT efficiency is highly dependent on machine operating conditions. Additional factors impacting efficiency include pressure drop across directional control valve (in open-loop HTs), flushing and low-pressure charge systems (in closed-loop HTs) and the inability to regenerate and reuse overrunning energy.

To improve the transmission efficiency over the traditional HT, it is necessary to reduce the energy losses of the positive displacement units. Literature mentions three main strategies to achieve this, which are outlined in the following subsections: -

1.1 Digital Displacement Hydrostatic Transmission

A potential approach in this context is to implement HT with pumps and motors based on the digital displacement principle. The digital displacement pump (DDP) and digital displacement pump/motor (DDPM) are radial piston machines that regulate flow by individually connecting or disconnecting each piston to the high- and low-pressure sides using actively controlled high-speed solenoid valves [5]. This mechanism significantly enhances overall efficiency at partial displacements, compared to conventional variable displacement units. Swapping conventional pumps with digital displacement pumps (DDPs) in the 16-tonne DEXTER excavator resulted in 19% fuel savings during a typical loading cycle, with 8% attributed to reduced pump losses, without compromising productivity [6]. Implementing a similar approach for HTs, incorporating both DDP and/or DDPM, is expected to have similar fuel savings. Additionally, each piston in a DDP, known as a pumplet, can be accessed and controlled like an individual variable displacement unit. This capability enables the DDP to function as an "elastic pump" system [7], facilitating multi-actuator HT implementations with reduced total pump capacity. Despite the fact that DDP machines are not yet commercially available, there is an opportunity for solutions able to increase HT efficiency by utilizing cost-effective and consolidated hydraulic components.

1.2 Hydro-Mechanical Power-Split Transmission

A hydro-mechanical power-split approach for increasing HT's efficiency is to combine the HT with a direct mechanical transmission within a single transmission system, utilizing a planetary gear train (PGT) [8,9]. This allows routing majority of power through the highly efficient mechanical path while keeping the CVT feature of transmission intact. Macor and Rosetti achieved 84% mean overall efficiency [10] by optimizing a hydro-mechanical power-split transmission used for agricultural tractor propulsion application.

Nevertheless, the use of bulky PGT necessitates for a sperate PGT for every actuator, which can be impractical for implementation with multiple actuators. This is why this architecture is mostly limited to propulsion powertrains.

1.3 Design Modification and Optimal Sizing of HT

Another approach to enhance efficiency of HTs is to allow the hydraulic units to operate in more efficient regimes, i.e., higher displacements and pressure differential, thus reducing energy losses associated with them. Conventional HT configurations, which have a single control degree of freedom, lack the flexibility to optimize operating conditions of these components. Guo and Vacca proposed a split-pump design with a dual displacement motor as an alternative flow-supply HT configuration, along with genetic algorithm based optimal

sizing methods [4]. This design introduces additional control degrees of freedom to enhance operating conditions of the components at the most frequent operating condition, thereby improving transmission efficiency. Simulation results for a heavy-duty harvester demonstrated 14% improvement in efficiency and 20% reduction in power consumption, compared to a reference commercial system.

This approach does not require expensive components for implementation, making it a cost-effective solution suitable for off-highway machines. However, the operating conditions of the hydraulic units remain coupled to the HT's overall operation, limiting efficiency gains in drive cycles with multiple frequently occurring operating conditions. Moreover, adding control degrees of freedom often requires replacing fixed displacement secondary units with variable or dual displacement units, which is impractical for applications utilizing series hydrostatic drives.

While the above-mentioned approaches improve transmission efficiency over conventional HT configurations their adoption is constrained by inherent limitations and implementation challenges. This paper proposes an alternative HT implementation aimed at improving transmission efficiency while ensuring broad applicability in off-highway applications. The architecture decouples the operating conditions of the hydraulic units from that of the HT, allowing them to operate at optimal efficiency at all times and enhancing transmission efficiency across various drive cycles. Additionally, it utilizes fixed displacement secondary units and commercially available hydraulic components, facilitating simpler and more adaptable implementation across a wide range of applications.

The paper is structured as follows: Section 2 outlines the proposed Pump Decoupled Hydrostatic Transmission (PDHT) architecture and its underlying motivation. Section 3 presents a case study comparing its performance against a conventional flow-supply series HT system. Finally, Section 4 provides concluding remarks and scope for future work.

2 Pump-Decoupled Hydrostatic Transmission (PDHT) Architecture

2.1 Description of Proposed Architecture

The objective is to develop an HT architecture using a variable displacement primary unit and a fixed displacement secondary unit, where the primary unit consistently operates at maximum displacement ($\epsilon_{P,Max}$) during power delivery, and a constant pressure differential (Δp_c) across both units is maintained regardless of the HT's operating conditions (i.e., motor's shaft speed and external torque). This would enable appropriately selected units (i.e., those with Δp_c within optimal pressure range) to operate at peak efficiency, therefore minimizing energy losses and maximizing overall transmission efficiency.

Such a specific decoupling of the positive displacement units' operating conditions from those of the HT would classify the architecture as a pressure-supply HT with two key requirements:

1. The pump must regulate the supply line pressure within a specific (narrow) range while operating at maximum displacement ($\epsilon_{P,Max}$) or zero displacement.
2. The fixed displacement motor's shaft speed must be controlled in the pressure-supply HT.

To accomplish these requirements, the open-circuit PDHT architecture illustrated in Figure 2 is proposed. A variable displacement pump regulates the supply line pressure, which is governed by equations 1-3. To maintain the line pressure within a narrow optimal range $[p_{Min}, p_{Max}]$, the pump fractional displacement (ϵ_P) alternates between zero and maximum, following the pressure control logic shown in Figure 3. The accumulator integrated into the supply line to reduce its hydraulic stiffness is sized adequately to prevent rapid stroking and de-stroking of the pump, which could otherwise render the proposed architecture infeasible. A fixed displacement motor is connected to the supply line, with its shaft speed controlled by an independently controlled hydraulic rotary actuator connected on the same shaft. This rotary actuator is referred to as speed-control unit (SCU). The motor's shaft speed is governed by eq. 4-5.

$$\frac{dp_H}{dt} = \left(\frac{p_0^{\frac{1}{\gamma}} \cdot V_o}{\gamma \cdot p_H^{\left(\frac{\gamma+1}{\gamma}\right)}} + \frac{V_{line}}{k_{oil}} \right)^{-1} \cdot (Q_P - Q_M) \quad (1)$$

$$Q_P = n_P \cdot V_{D,P} \cdot \epsilon_P \cdot \eta_{v,P} \quad (2)$$

$$Q_M = \frac{n_M \cdot V_{D,M}}{\eta_{v,M}} \quad (3)$$

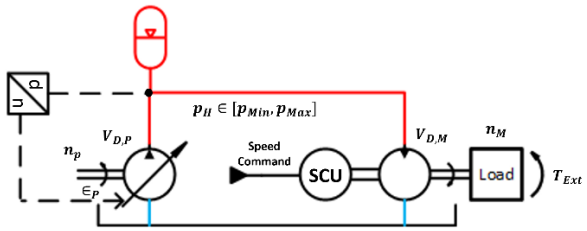

Figure 2. Generic layout of a PDHT architecture

$$T_{SCU} + T_M - T_{Ext} = J \frac{dn_M}{dt} \quad (4)$$

$$T_M = (p_H - p_T) \cdot V_{D,M} \cdot \eta_{HM,M} \quad (5)$$

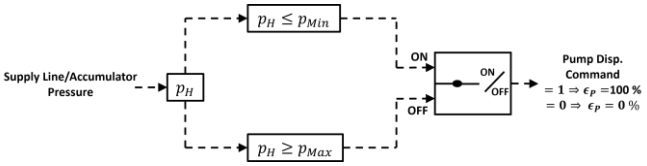

Figure 3. PDHT supply pressure control logic.

The strategy of alternating the pump displacement between zero and maximum, to regulate the supply line (or accumulator) pressure within an optimal range, as illustrated in Figure 4, is similar to that used in constant pressure systems [11-13]. This approach ensures that the pump operates at

maximum efficiency whenever supplying hydraulic power to the system, and only incurs parasitic losses when not delivering any hydraulic power. Alternatively, a fixed displacement unit with an accumulator unloading valve can be used. However, it presents certain limitations, such as throttling losses through the unloading valve during unloading and reduced control flexibility for operating in a flow-supply mode if required, which is demonstrated as beneficial in the case study presented later in the paper.

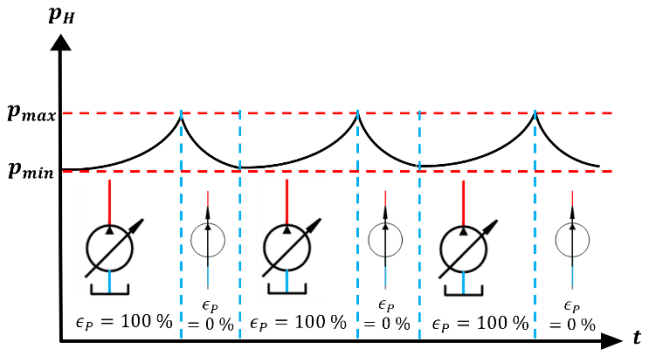

Figure 4. Pump supply pressure regulation by alternating between max and zero displacement.

The fixed displacement motor operates at its peak efficiency, converting hydraulic power from the supply line into mechanical power (P_{DHT}) by delivering torque (T_M) to the rotary function, as defined by eq. (6). As the pressure differential across the motor varies within a narrow range, the generated torque also remains within a narrow band, as defined by eq. (5). Additionally, the SCU provides the remaining torque (T_{SCU}) required for shaft speed control, contributing mechanical power (P_{SCU}), as given by eq. (7). Therefore, the PDHT architecture features two distinct power transmission paths from the input power source to the rotary function: a highly efficient decoupled HT path and a speed-control path, whose efficiency depends on the operating conditions, as shown in Figure 5. Consequently, the transmission efficiency of the PDHT architecture (η_{PDHT}) is described by eq. (8).

$$P_{DHT} = T_M \cdot n_M \tag{6}$$

$$P_{SCU} = T_{SCU} \cdot n_M \tag{7}$$

$$\eta_{PDHT} = \frac{Output\ Power}{Net\ Input\ Power} = \frac{P_{DHT} + P_{SCU}}{\left(\frac{P_{DHT}}{\eta_{DHT}}\right) + \left(\frac{P_{SCU}}{\eta_{SCU}}\right)} \tag{8}$$

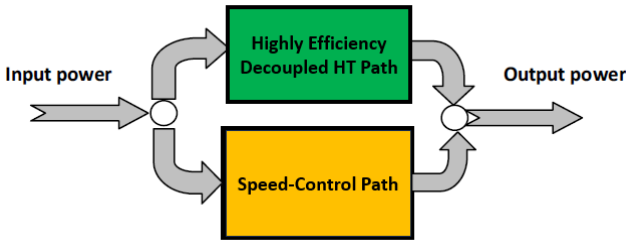

Figure 5.Two paths for power transmission in PDHT.

Eq. 6-8 indicate that high transmission efficiency of the PDHT architecture can be achieved if either of the following conditions is met:

1. Majority of power delivered to the rotary function should be transmitted through the decoupled HT path, i.e., $P_{DHT} \gg P_{SCU}$ (or $T_M \gg T_{SCU}$).

or

2. Efficiency of speed-control path (η_{SCU}) is high when power transmitted through speed-control path (P_{SCU}) is substantial, or T_{SCU} is substantial.

Therefore, a higher supply line pressure range $[p_{Min}, p_{Max}]$ is desirable for improved transmission efficiency, as it would increase the proportion of the total power transmitted through the highly efficient decoupled HT path.

However, the selection of the SCU in this architecture may impose a limit on the maximum allowable supply line pressure if the SCU is unable to regulate shaft speed under negative torque conditions. The steady-state relationship between the torque generated by the fixed displacement motor (T_M), the speed-control unit (T_{SCU}), and the external torque on the motor shaft (T_{Ext}) is defined by eq. (9):

$$T_{SCU} = T_{Ext} - T_M \tag{9}$$

If the chosen SCU can't handle negative torque, the maximum torque ($T_{M,Max}$) that can be supplied by the fixed displacement motor is constraint by eq. (10), also graphically illustrated in Figure 6. The corresponding maximum allowable supply line pressure ($p_{Max,Allowed}$) in the supply line is related to the maximum torque, as described by eq. (11).

$$T_{SCU} = T_{Ext,Min} - T_{M,Max} = 0 \tag{10}$$

$$T_{M,Max} = \left(p_{Max,Allowed} - p_T\right) \cdot V_{D,M} \tag{11}$$

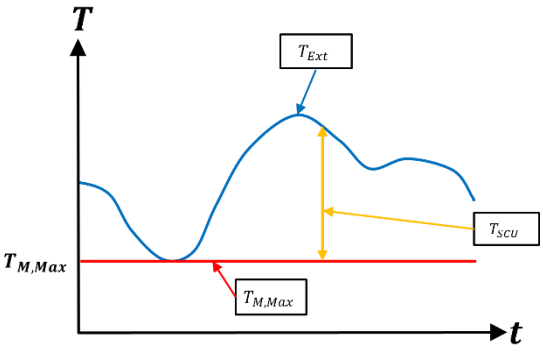

Figure 6. Max. allowable fixed disp. motor torque for cases where SCU can't handle negative torque.

Hence, selecting an appropriate SCU is crucial for maximizing the transmission efficiency of the PDHT architecture, as it may constrain the power transmitted through the efficient decoupled HT path. Additionally, an SCU that results in low speed-control path efficiency when delivering substantial power would introduce significant losses, thereby adversely affecting PDHT architecture's efficiency. Therefore, to achieve maximum transmission efficiency across all operating conditions, the desired characteristics/technology for the SCU are:

1. It should be capable of regulating the motor's shaft speed under negative torque conditions.

2. It should maintain high efficiency of the speed-control path (η_{SCU}) while delivering substantial power (P_{SCU}) or torque (T_{SCU}).

2.2 PDHT Architecture using a Secondary-Controlled Motor as SCU

To maximize PDHT transmission efficiency across various operating conditions, different SCU alternatives can be considered based on the requirements outlined in the previous section. This paper proposes a PDHT architecture variant incorporating a secondary-controlled motor with over-center capability as the SCU, as shown in Figure 7. The inlet of the secondary controlled motor is connected to the supply line with the accumulator, where pressure is regulated within a narrow optimal range [p_{Min}, p_{Max}] by the decoupled pump, ensuring a constant pressure differential across the secondary controlled motor. Therefore, the torque generated by the SCU (or the secondary controlled motor) is described by eq. (12). The motor's shaft speed is controlled by varying its fractional displacement (ϵ_{SM}) using a feedback loop.

$$T_{SCU} = (p_H - p_T) \cdot \epsilon_{SM} \cdot V_{D,SM} \cdot \eta_{HM,SM} \quad (12)$$

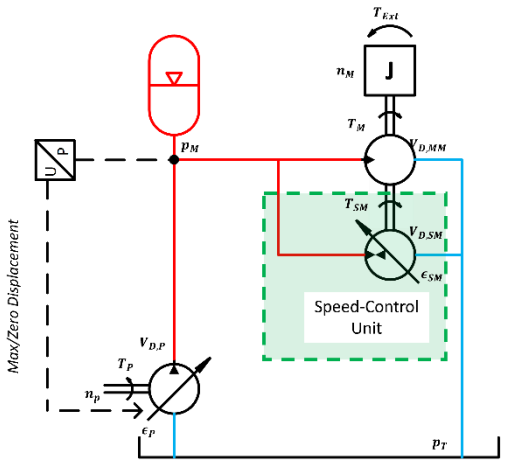

Figure 7. PDHT using a Secondary-Controlled Motor as an SCU

In this variant, power transmission through the speed-control path occurs in two stages: mechanical-to-hydraulic and hydraulic-to-mechanical conversion. The decoupled pump, operating at high efficiency, performs the mechanical-to-hydraulic conversion, while the secondary controlled motor, operating at varying frequency, handles the hydraulic-to-mechanical conversion. At high torque output, the secondary-controlled motor operates at high fractional displacement (ϵ_{SM}), ensuring high efficiency. Consequently, the speed-control path maintains high efficiency when the motor generates high torque.

A secondary-controlled motor with over-center capability can regulate shaft speed under negative torque conditions, allowing for a higher supply line pressure. This increases the proportion of power transmitted through the decoupled HT path. Additionally, a higher supply line pressure enables the use of a smaller displacement speed-control motor and, consequently, a smaller displacement decoupled pump while achieving the same combined maximum torque of the fixed displacement and secondary-controlled motors. During negative torque conditions, the secondary-controlled motor pumps flow into the accumulator, facilitating efficient regeneration of excess energy transmitted through the decoupled HT path, as illustrated in Figure 8.

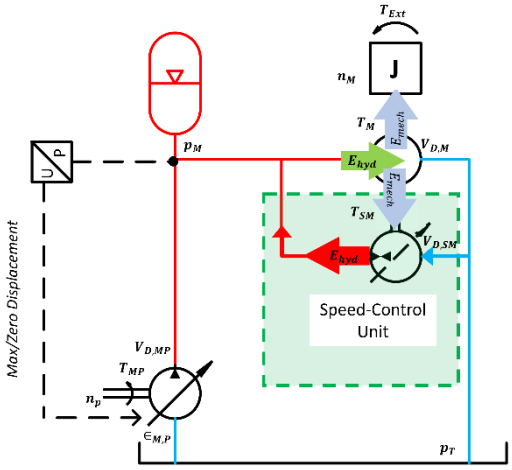

Figure 8. Regeneration of excess mechanical energy delivered by decoupled HT line.

2.3 Multi-Actuator Implementation with PDHT

The proposed variant of PDHT architecture incorporates an additional variable displacement motor per actuator compared to conventional HT. Furthermore, both the fixed displacement motor and the secondary-controlled motor receive power from a common supply line. This design enables straightforward implementation in systems with multiple actuators, as shown in Figure 9. Multiple actuators can be powered through a common pressure rail or a set of pressure rails, allowing them to share a set of pumps and accumulators. This centralized power supply reduces the number of pumps required, thereby lowering overall system costs.

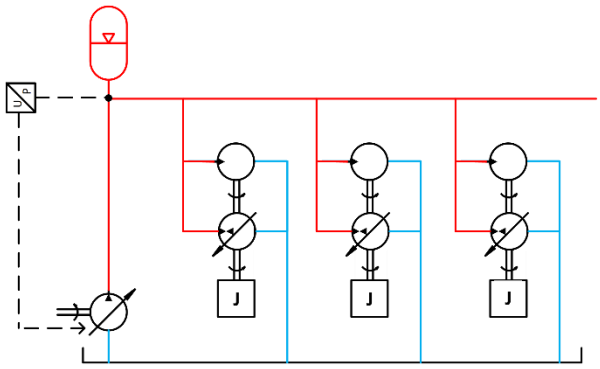

Figure 9. Multi-actuator implementation with PDHT.

Furthermore, the PDHT architecture's use of a fixed displacement motor makes it compatible with series hydrostatic drives. A set of motors connected in series can be directly connected to the supply line, whose pressure is regulated by the decoupled pump, with the SCU regulating the speed of the first motor in the series. Since the first motor's speed is controlled by the SCU, the subsequent motors receive the required flow, operating as they would in a conventional series configuration. Additionally, this implementation

requires only an additional variable displacement motor and accumulator for the entire set of motors, as described in Figure 10.

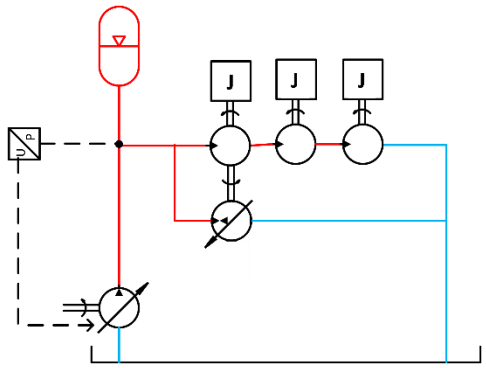

Figure 10. PDHT implementation for a series hydrostatic drive.

3 Case Study

To demonstrate the PDHT architecture's potential for improved efficiency with simpler implementation, it is compared to a conventional flow-supply primary control series HT system for a sugarcane harvester application. The comparison is based on simulations for a reference cycle, with measurements provided by an OEM.

3.1 Reference Machine/System/Cycle

The reference machine used in this study is a Case IH Austoft 9900 sugarcane harvester, with a focus on its basecutter system, a primary energy-consuming subsystem for the whole vehicle. The baseline version of this subsystem is an open-circuit flow-supply with primary control HT featuring multiple series of motors, as illustrated in simplified hydraulic schematic in Figure 11 . In this open-circuit layout, flow direction is controlled by the basecutter valve, which functions similarly to a directional control valve while maintaining roughly a constant pressure drop of 3 bar across a range of flow rates.

The system primarily operates in the forward mode, i.e., where the pump supplies flow to the basecutter motor. This flow is then directed through the motor, divided into subsequent branches, and delivered to the following motors. The speed of the motors is exclusively controlled in this mode. The system switches to reverse mode, i.e., reverse the direction of rotation of motors, by connecting the basecutter motor to tank using the DCV.

Measurement data for a 140-seconds drive cycle at a particular pump shaft speed (n_{Pump}) was provided, including pressures in each line, pump suction pressure, and basecutter motor shaft speed. To compare the baseline system with the PDHT architecture implementation, the external torque on the motor shaft was estimated from the measurement data and used as inputs for simulation model for both systems. The drive cycle defines the operation of the motors at a desired speed against the external torque on the motor shaft. As the system is a series hydrostatic drive with speed-synchronized motors, specifying the desired speed of one motor is sufficient. For the reference drive cycle, the desired basecutter motor shaft speed ($n_{Basecutter,desired}$) was provided.

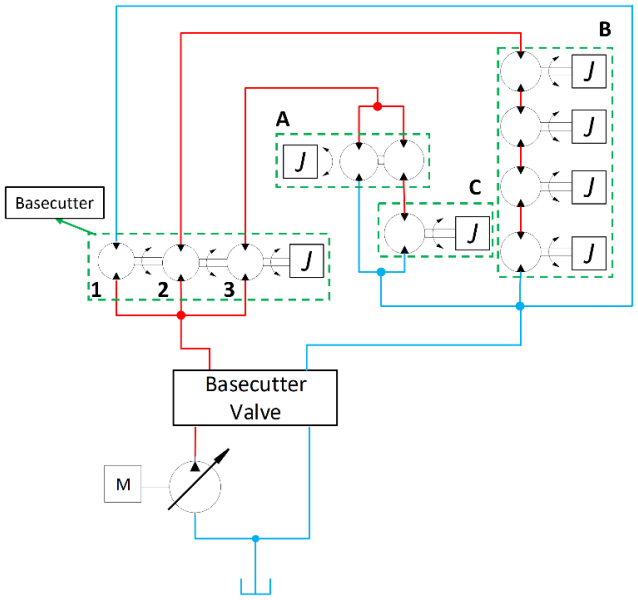

Figure 11. Simplified schematic of basecutter system.

3.2 PDHT Architecture Implementation for Basecutter System

The PDHT architecture is implemented on the basecutter system, as shown in Figure 12. To minimize modifications to the baseline system, the original layout is retained along with the components. The supply line connects to the basecutter motor inlet, while the SCU (i.e., secondary controlled motor) is coupled to the motor's shaft.

In the baseline system, the largest gear section (section-1) in the basecutter motor accounts for half of its total displacement and is incorporated solely to generate additional torque without increasing the motor's inlet pressure or pressure differential. In the PDHT architecture, however, the motor inlet pressure is regulated, and the SCU supplies the additional torque required to move against any load. Therefore, in the PDHT implementation, this gear section is replaced with an SCU of the same displacement. This modification preserves the total motor and pump displacement from the baseline system with minimal changes.

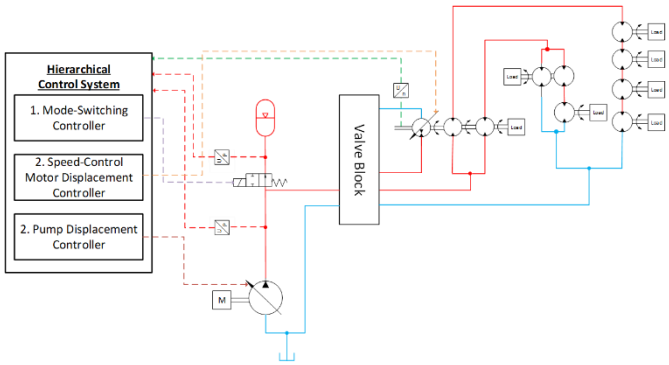

Figure 12. PDHT architecture implemented on basecutter system.

The upper limit of supply line pressure (p_{Max}) is set to the rated pressure of the basecutter motor ($p_{rated,Basecutter}$) to ensure that the maximum torque generated by the basecutter motor in the PDHT system matches that of the baseline system, as both systems have the same total motor displacement. Additionally, to keep the supply line pressure within a narrow range, the lower limit (p_{Min}) is set to $0.8 * p_{Max}$. Also, to keep the frequency of pump stroking/de-stroking low, the accumulator size (V_{Acc}) is set to be 20 liters.

The basecutter motor shaft may experience external torque spikes due to factors such as stuck cane, debris, or the basecutter blade being lowered excessively, causing it to hit the ground. In such an event, the torque demand on the motor shaft may momentarily exceed the motor's maximum capacity, potentially causing the motor to stall or rapidly decrease its shaft speed.

In a flow-supply HT system, such torque spikes are managed effectively without stalling the motors or causing a sharp speed drop. As the motor speed decreases, the inlet pressure rapidly increases above the rated motor pressure, resulting in the motor torque overcoming the torque spike with minimal speed reduction before restoring speed control. However, in a pressure-supply HT system, this response is not possible because the supply line pressure is regulated. Even if the system temporarily overrides the pressure control logic and switches to a flow-supply mode, the pressure increase remains limited due to the high capacitance of the supply line, primarily caused by the integrated accumulator.

To manage external torque spikes, the PDHT system for the basecutter implements a speed-control strategy. A solenoid-actuated, two-position ON/OFF valve, referred to as the switching valve, connects or disconnects the accumulator from the supply line. Under normal conditions, when the external torque on the motor shaft remains within the motor's maximum capacity, the system operates in pressure-supply mode with the accumulator connected and the system following PDHT logic as outlined in Section-2. However, during a torque spike, when the torque demand exceeds the motor's capacity, the system switches to flow-supply mode, disconnecting the accumulator to enable a rapid pressure rise. Additionally, the motor's shaft speed is regulated through primary control at the pump, while the secondary controlled motor (i.e., SCU) shifts to maximum displacement, functioning as a fixed displacement motor.

This strategy is implemented by the hierarchical control system, also shown in Figure 12, comprising of:

- Level 1: Mode-Switch Controller – for deciding which mode to operate on based on motor's shaft speed and pump outlet pressure feedback, as illustrated in Figure 13.
- Level 2: Pump/Motor Displacement Controller – to adjust the displacement of the pump and the speed-control motor to regulate the motor's shaft speed based on system mode and shaft speed feedback, as illustrated in Figure 14.

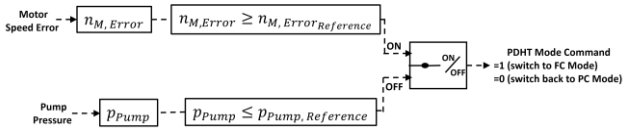

Figure 13. PDHT implementation for basecutter system: Mode-Switch Controller

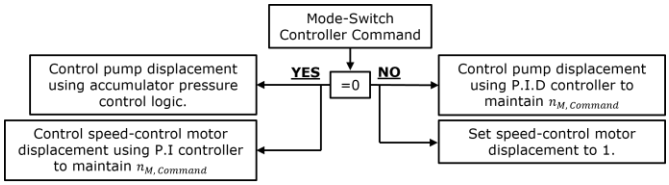

Figure 14. PDHT implementation for basecutter system: Pump/Motor Displacement Controller.

3.3 Simulation Model

The lumped-parameter simulation model for both the baseline and PDHT systems is developed in Simcenter Amesim. Key components include the basecutter pump, motors, rotating components, and hydraulic lines, with the accumulator and switching valve incorporated for the PDHT implementation. The accumulator is assumed to follow the ideal gas laws and operates under adiabatic conditions. Each component is modeled individually and integrated into the complete system model.

Both models use identical efficiency maps for common components to attribute efficiency improvements to architectural changes, not component differences. In the PDHT system model, the speed-control unit, a variable displacement motor with over-center capability, employs the efficiency map of a commercially available motor with similar over-center functionality. Moreover, it is assumed that the pump transient timescale is significantly shorter than that of stroking and de-stroking, justifying the use of steady-state efficiency maps.

However, the pump shaft input energy, derived from efficiency maps, does not account for parasitic losses at zero displacement. To incorporate these losses in the PDHT system's energy calculations, the input energy at zero displacement is assumed to be equal to that at 1% fractional displacement for the same pump outlet pressure and shaft speed. This approach provides a conservative estimate of parasitic losses, as energy consumption at zero displacement is generally lower than, or at most equal to, that at 1% fractional displacement under the same outlet pressure.

The relevant parameters assumed for PDHT simulation are: -

i. $p_{Max} = p_{rated,Basecutter}$
ii. $p_{Min} = 0.8 * p_{Max}$
iii. $\gamma = 1.4$
iv. pump stroking/de-stroking response time = 200 ms
v. $\Delta p_{switching\ valve}$= 10 bars at 325 L/min
vi. switching valve spool dynamics: 1^{st} order

vii. $\tau_{switching\ valve\ spool}$ = 100 ms

3.4 Simulation Results

The model takes the estimated external torque on each motor shaft as input and tracks the specified basecutter motor speed ($n_{Basecutter,desired}$) while maintaining the pump at the given shaft speed (n_{Pump}). The model outputs the pump's shaft torque (T_{Pump}), which is then used to compute the pump's shaft input power ($P_{shaft\ input,P}$) and input energy ($E_{shaft\ input,P}$), as defined in eq. 13-14 respectively. Additionally, for each motor, the model provides the total shaft torque (T_{M-i}) and shaft speed (n_{M-i}) for each motor, which are used to determine the motor's shaft output power ($P_{shaft\ output,M-i}$) and output energy ($E_{shaft\ output,M-i}$), as given in eq. 15-16 respectively.

$$P_{shaft\ input,P} = T_{Pump} \cdot n_{Pump} \tag{13}$$

$$E_{shaft\ input,P} = \int_0^t P_{shaft\ input} \cdot dt \tag{14}$$

$$P_{shaft\ output,M-i} = T_{M-i} \cdot n_{M-i} \tag{15}$$

$$E_{shaft\ output,M-i} = \int_0^t P_{shaft\ output,M-i} \cdot dt \tag{16}$$

The useful work for either of the two systems is defined by eq. (17). Both systems generate approximately 2400 kJ of total work over the 140-seconds drive cycle

$$\text{Usefull work} = \sum E_{shaft\ output,M-i} \tag{17}$$

The average system efficiency ($\eta_{sys-i_{Average}}$) for both the systems is defined by eq. (18). Also, the reduction in energy consumption of PDHT system over the baseline is computed using eq. (19).

$$\eta_{sys-i_{Average}} = \frac{Useful\ Work}{E_{shaft\ input,P}} \tag{18}$$

$$\Delta E_{PDHT} = \left(\frac{E_{shaft\ input,P_{Baseline}} - E_{shaft\ input,P_{PDHT}}}{E_{shaft\ input,P_{Baseline}}} \right) \tag{19}$$

Table 1 provides the summary of the comparison between the baseline and PDHT systems.

Table 1. Energy Comparison Summary Results

System #	$E_{shaft\ input,P}$ (Norm.) [-]	$\eta_{sys,Average}$ [%]	ΔE_{PDHT} [%]
Baseline	100	53 %	--
PDHT	85	63 %	15 %

Figure 15-16 illustrate the energy flow and losses in the baseline and PDHT systems. The diagrams show that the PDHT system significantly reduces pump losses without substantially increasing motor or other losses, resulting in 15% lower overall energy consumption compared to baseline system. Energy analysis of the PDHT system further indicates that approximately 2% of pump's shaft input energy is lost due to excess energy recirculation during the SCU's negative torque condition, highlighting the potential for additional efficiency gains by mitigating these instances. Moreover, selecting pumps and motors with optimal pressure ranges aligned with the supply line pressure range could further improve system efficiency and reduce overall energy consumption.

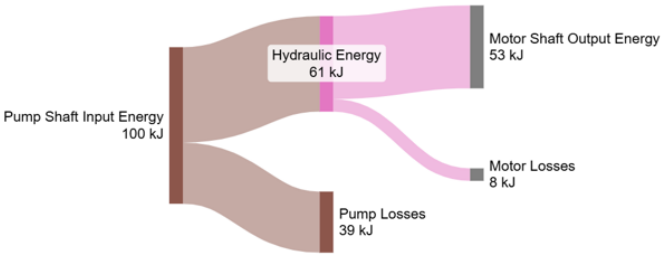

Figure 15. Normalized energy flow diagram of baseline system.

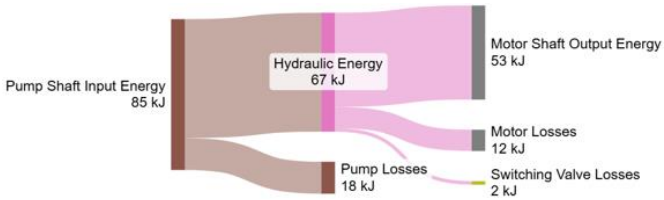

Figure 16. Normalized energy flow diagram of PDHT system.

The stroking/de-stroking frequency of the PDHT system for the given drive cycle is approximately 0.3 Hz, as indicated by Figure 17.

Due to the lower hydraulic stiffness of the accumulator, the PDHT system exhibits significantly fewer speed oscillations compared to the conventional flow-supply primary control HT. This is evident in Figure 18, which compares the basecutter motor's shaft speed tracking of PDHT system against baseline system.

4 Conclusion and Future Work

A novel hydrostatic transmission architecture for off-highway machines is presented, which decouples the operating conditions of positive displacement units from the HT, thereby enhancing transmission efficiency and improving performance across a range of drive cycles. The architecture enables straightforward and economical implementation for diverse applications. A comparative case study demonstrates that the proposed architecture reduces overall energy consumption by 15% compared to flow-supply primary control series HT, while minimizing modifications and additional components, for a sugarcane harvester application.

The study of the PDHT architecture is in its early stages. Current research focuses on: (1) assessing the accuracy of using steady-state efficiency maps in energy calculations for the PDHT architecture, (2) reducing the frequency of pump stroking/de-stroking or increasing its duty ratio without

widening the supply line/accumulator pressure range, (3) minimizing negative torque conditions in the SCU to reduce conversion losses during energy regeneration, and (4) optimizing the architecture to lower fuel consumption by enabling the engine to operate in a more efficient range.

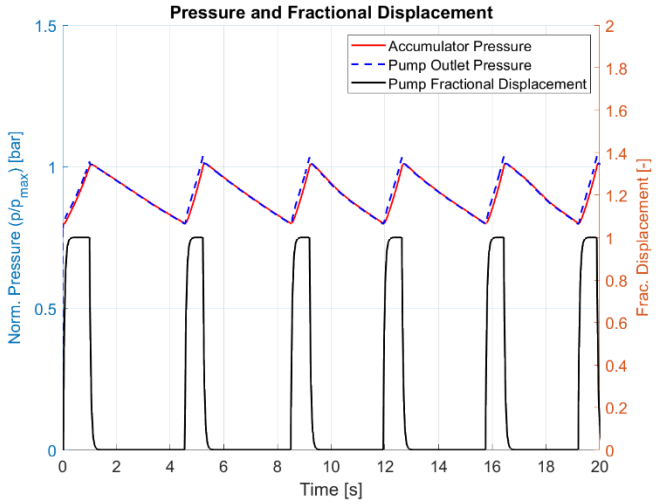

Figure 17. Pump stroking/de-stroking along with supply line/accumulator pressure variation of PDHT system.

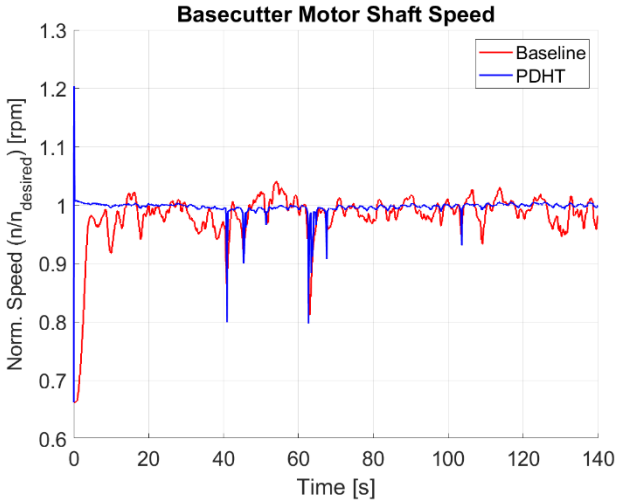

Figure 18. Motor's shaft speed oscillations comparison of PDHT against baseline system.

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Nomenclature

Designation	Denotation	Unit
k_{oil}	Equivalent line bulk modulus	Pa
p_H	Supply line pressure	Pa
p_T	Return line pressure	Pa
p_{Pump}	Pump outlet pressure	Pa
p_0	Accumulator pre-charge pressure	Pa
$p_{Max,Allowed}$	Max allowable supply line pressure in PDHT	Pa
p_{Max}	Max pressure of supply line in PDHT	Pa
p_{Min}	Min pressure of supply line in PDHT	Pa
$p_{rated,Basecutter}$	Basecutter motor rated pressure	Pa
$\Delta p_{switching\ valve}$	Pressure drop across switching valve	Pa
V_0	Total volume of accumulator	Liter
V_{Acc}	Accumulator size	Liter
ϵ_P	Fractional displacement of pump	--
ϵ_M	Fractional displacement of motor	--
ϵ_{SM}	Fractional displacement of secondary-controlled motor	--
$V_{D,P}$	Max volumetric displacement of pump	cc/rev
$V_{D,M}$	Max volumetric displacement of motor	cc/rev
$V_{D,SM}$	Max volumetric displacement of secondary-controlled motor	cc/rev
T_P	Pump's shaft torque	Nm
T_M	Motor's shaft torque	
T_{SCU}	Torque generated by SCU	Nm
$T_{M,Max}$	Max. torque generated by fixed displacement motor in SCU	Nm
T_{Ext}	External torque on motor's shaft	Nm
n_P	Pump's shaft speed	rad/s
n_M	Motor's shaft speed	rad/s
$n_{M,Error}$	Motor's shaft speed error in PDHT	rad/s
$E_{shaft\ input,P}$	Pump's shaft input energy	J
$E_{shaft\ output,M}$	Motor's shaft input energy	J

ΔE	Reduction in energy consumption	J
$P_{shaft\ input,P}$	Pump's shaft input power	W
$P_{shaft\ output,M}$	Motor's shaft input power	W
P_{DHT}	Power transmitted through decoupled HT path	W
P_{SCU}	Power transmitted through speed-control path	W
η_{DHT}	Efficiency of decoupled HT path	--
η_{SCU}	Efficiency of speed-control path	--
η_{PDHT}	Transmission efficiency of PDHT	--
$\eta_{sys,Average}$	Average system efficiency	--
γ	Polytropic coefficient	--
$\tau_{switching\ valve\ spool}$	Time constant for switching valve dynamics	s

References

[1] Burrows, C. R., 1972, Fluid Power Servomechanisms, Van Nostrand Reinhold, New York.

[2] Vacca, A., Franzoni, G. Hydraulic Fluid Power: Fundamentals, Applications, and Circuit Design; Wiley: Hoboken, NJ, USA, 2021.

[3] Manring, N. D. (January 12, 2016). "Mapping the Efficiency for a Hydrostatic Transmission." ASME. *J. Dyn. Sys., Meas., Control*. March 2016; 138(3): 031004.

[4] Guo, X., & Vacca, A. (2021). "Advanced Design and Optimal Sizing of Hydrostatic Transmission Systems." *Actuators*, *10*(9), 243.

[5] Caldwell, N. "Review of Early Work on Digital Displacement® Hydrostatic Transmission Systems." *Proceedings of the BATH/ASME 2018 Symposium on Fluid Power and Motion Control*. *BATH/ASME 2018 Symposium on Fluid Power and Motion Control*. Bath, UK. September 12–14, 2018. V001T01A061.

[6] Green, M, Macpherson, J, Caldwell, N, & Rampen, WHS. "DEXTER: The Application of a Digital Displacement® Pump to a 16 Tonne Excavator." *Proceedings of the BATH/ASME 2018 Symposium on Fluid Power and Motion Control*. *BATH/ASME 2018 Symposium on Fluid Power and Motion Control*. Bath, UK. September 12–14, 2018. V001T01A047.

[7] Matteo Pellegri, Matthew Green, Jill Macpherson, Callan McKay, and Niall Caldwell. "Applying a Multi-Service Digital Displacement ® Pump to an Excavator to Reduce Valve Losses." In *12th International Fluid Power Conference (IFK'20).* Dresden, Germany, 2020.

[8] Kress, J. H. (1968). "Hydrostatic Power-Splitting Transmissions for Wheeled Vehicles — Classification and Theory of Operation." *SAE Transactions*, *77*, 2282–2306.

[9] Mistry, SI, & Sparks, GE. "Infinitely Variable Transmission (IVT) of John Deere 7000 TEN Series Tractors." *Proceedings of the ASME 2002 International Mechanical Engineering Congress and Exposition*. *Fluid Power Systems and Technology*. New Orleans, Louisiana, USA. November 17–22, 2002. pp. 123-131. ASME.

[10] Macor, A., & Rossetti, A. "Optimization of Hydro-mechanical Power Split Transmissions." *Mech. Mach. Theory* 2011, *46*, 1901–1919.

[11] Xu, Z., Bertolin, M., & Vacca, A., and Nilsson, J., "A Hydraulic Architecture Based on Multi-Common Pressure Rail Principle Using Multi-Chamber Cylinders for Excavators", Proceedings of the Global Fluid Power Society PhD Symposium, Hudiksvall, Sweden, Jun. 2024.

[12] Li, PY, Siefert, J, & Bigelow, D. "A Hybrid Hydraulic-Electric Architecture (HHEA) for High Power Off-Road Mobile Machines." *Proceedings of the ASME/BATH 2019 Symposium on Fluid Power and Motion Control*. *ASME/BATH 2019 Symposium on Fluid Power and Motion Control*. Longboat Key, Florida, USA. October 7–9, 2019. V001T01A011.

[13] Vukovic, M, Sgro, S, & Murrenhoff, H. "STEAM: A Mobile Hydraulic System With Engine Integration." *Proceedings of the ASME/BATH 2013 Symposium on Fluid Power and Motion Control*. *ASME/BATH 2013 Symposium on Fluid Power and Motion Control*. Sarasota, Florida, USA. October 6–9, 2013. V001T01A005.