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Case Study on Independent Metering Hydraulics for Energy-Efficient Battery-Electric Telescopic Loaders

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Abstract.

This paper discusses the topic of energy-efficient hydraulics on battery-electric telescopic loaders. For this, two hydraulic systems are compared. The first one is a standard setup that can be found in classical machines that are driven by a combustion engine. The second one is a proposal using a decentralized independent metering approach. Through a comprehensive case study, the energy efficiency of both systems is evaluated. Thereby, it is shown that modern hydraulic setups can generate a great benefit for electrically driven machines by extending battery load cycles or reducing the required energy capacity to be installed, respectively.

Keywords. Mobile machines, energy efficiency, independent metering, battery-electric telescopic loaders, electrification.

1. INTRODUCTION

The transition from diesel-hydraulic to fully battery-electric mobile working machines is a process that is fundamentally reshaping the system-level energy architecture of such equipment. In this new paradigm, it is imperative that every kilowatt-hour drawn by the hydraulic circuit be meticulously balanced against a series of stringent constraints on battery capacity, including weight, installation space, upfront cost, and charging intervals. Consequently, the hydraulic subsystem, which was previously regarded as a passive energy consumer, has evolved into a pivotal mechanism for attaining an uninterrupted, eight-hour operating window within a designated energy budget [1].

Conventional hydraulic architectures – including open-center and load-sensing systems – are predicated on centralized valve blocks and extensive pressure lines. These designs introduce inherent inefficiencies, including high stand-by losses, throttling across metering orifices, and dynamic mismatches between pump output and actuator demand [2]. These inefficiencies are becoming increasingly unacceptable in battery-electric platforms, where energy is a scarce and costly resource. This comes from the fact that additional battery packs are adding both weight and cost to machines, which is undesirable from an economic point of view.

In order to address these limitations, this case study explores a decentralized independent-metering (IM) architecture, in which compact, seated valve cartridges are mounted directly onto each hydraulic actuator [3]. This physical reconfiguration of components eliminates the necessity for external load-holding and counterbalance valves. Additionally, it offers more direct control, thereby reducing pressure losses. Notably, this system facilitates native support for gravity-assisted lowering, a feature that is crucial for the system's functionality. When paired with a variable-speed electric motor-pump unit (MPU), the IM concept provides access to advanced supervisory functionalities, including electronic flow sharing, adaptive pressure control, and load-aware energy scheduling.

In this work, a full-scale experimental evaluation is conducted using battery-electric telescopic loader. The machine is tested in two configurations: (i) a baseline layout in an open-center system setting, and (ii) the reengineered IM-based hydraulic circuit. Both systems are assessed on the exact same machine using repeatable, standardized loading cycles that are representative of typical material handling tasks in logistics, construction, or agricultural environments.

In summary, the study establishes a foundation for the advancement of intelligent energy-efficient hydraulics by demonstrating its potential as a component-level enhancement and a fundamental enabler for the cost-efficient, space-conscious, and operationally viable electrification of next-generation mobile machinery. This work aims to demonstrate the transformative potential of rethinking hydraulic systems from a first-principles energy-efficiency perspective, tightly integrated with modern electric powertrain technologies.

1.1. Literature Review

Recent shifts toward higher energy efficiency in the mobile machinery market are driven by regulatory requirements and ongoing electrification. However, research on more energy-efficient hydraulic circuits has been active for many years. A brief literature review highlights some significant work in this area.

Opgenoorth et al. [4] present an advancement of the STEAM hydraulic system originally developed at IFAS, RWTH Aachen. Their work introduces a multi-pressure hydraulic architecture applied to an excavator platform, featuring three pressure lines that are dynamically regulated via electric motors. The system employs two separate electric drives – one powering a conventional pump and the other driving a pump-motor unit. These pressure sources, in combination with the tank pressure, are selectively routed to the actuators through a configurable valve network. A central focus of the study is the design of a supervisory controller responsible for optimal state selection, determining both the active pressure lines and their respective pressure levels based on actuator demand. In a simulated 90° dig-and-dump cycle, the proposed system demonstrates a 29% increase in efficiency of hydraulic System compared to a conventional reference model, underscoring the potential of multi-pressure architectures and reduction of throttle losses for energy-optimized mobile hydraulics.

A novel independent metering valve configuration for excavator boom control, integrating three electrohydraulic 2/2-way proportional poppet valves with a 4/3-way directional control valve is proposed by Nguyen et al. [5]. The control strategy combines pump speed modulation with valve actuation, wherein the inlet valve is fully opened to minimize throttling losses, while the outlet valve is regulated based on a predefined flow rate and position feedback. The architecture supports multiple operating modes, including regenerative functions such as low-side regeneration retraction, in which the boom is retracted solely by gravitational force without active pump contribution. This functionality is enabled by an additional short-circuit valve integrated into the system. Experimental evaluations demonstrate energy savings of up to 22% in power retraction mode and 6.5% in high-side regeneration mode compared to a conventional reference system. Moreover, the configuration achieves improved position accuracy while reducing overall system complexity.

Hansen et al. [6] present a system design aimed at reducing energy consumption in mobile hydraulic applications. Their approach is implemented on a JCB telescopic handler and utilizes a separate-meter-in/separate-meter-out valve configuration embedded within a load-sensing (LS) hydraulic system. The conductive subsystem is realized using two 3/2-way proportional valves, each governed by distinct PI controllers. The study focuses on the development of a control strategy that enables precise modulation of flow paths and pressure levels to match actuator demands. Comparative analysis against a conventional pressure-compensated proportional valve setup reveals significant energy-saving potential. For the evaluated duty cycle, the proposed configuration achieves a reduction in energy consumption exceeding 30%, demonstrating the effectiveness of the control-oriented valve architecture in enhancing hydraulic efficiency.

1.2. Main Contribution and Outline

Although there is a wealth of literature on the advantages and disadvantages of modern IM systems, this work focuses on a standardized comparison with a conventional hydraulic circuit on a real-world telescopic loader. The main efforts are:

- The Definition of a standardized test procedure for both the conventional system and the IM system.
- The equipment of a battery electric telescopic loader with a conventional hydraulic circuit and a modern decentralized IM system.
- A comprehensive testing phase followed by a thorough analysis of the results.

The structure of the work is as follows: First, the telescopic loader and the hydraulic circuits are introduced. Next, the experimental setup and the type and number of experiments are presented. Next, the results are presented and interpreted. The work concludes with a summary and a recommendation for future research.

Figure 1: Image of the telescopic loader used in this work.

2. SYSTEM

In this section, the hydraulic system used in the telescopic loader is introduced. An image of the overall machine can be seen in Figure 1. We start with an explanation of the machine's conventional configuration, which serves as the basis of our study. Afterwards, an introduction of the adapted IM system is given.

Maximum load	3200 kg
Maximum lifting height	7 m
Battery capacity	63 kWh
Peak power overall	120 kW
Peak hydraulic power	60 kW
Volume flow	20 – 100 l/min
Maximum pressure	240 bar
Pump rotational speed	500 – 2200 rpm
Motor rotational speed	0 – 6600 rpm

Table 1: Technical data of the telescopic loader used in this work.

2.1. Technical Data

The telescopic loader that is used for this case study has a maximum load capacity of 3200 kg and a maximum lifting height of 7 m. Moreover, it is equipped with a fixed-displacement axial piston pump that delivers flows from 20 to 100 l/min with a rotational speed from 500 to 2200 rpm. The rotational speed of the motor that is driving the pump ranges from 0 to 6600 rpm, i.e. there is a gear with a ratio of 3 in between the two components. The maximum pressure is 240 bar.

The drivetrain of the machine is fully electrified, i.e. it contains no combustion engine. For this, eight battery packs with a total capacity of 63 kWh are installed. It is assumed that the machine lasts an average working day of 8 hours with only one loading cycle. The system can provide a peak power of 120 kW, from which the hydraulic system accounts for 60 kW. Table 1 provides an overview of those key points.

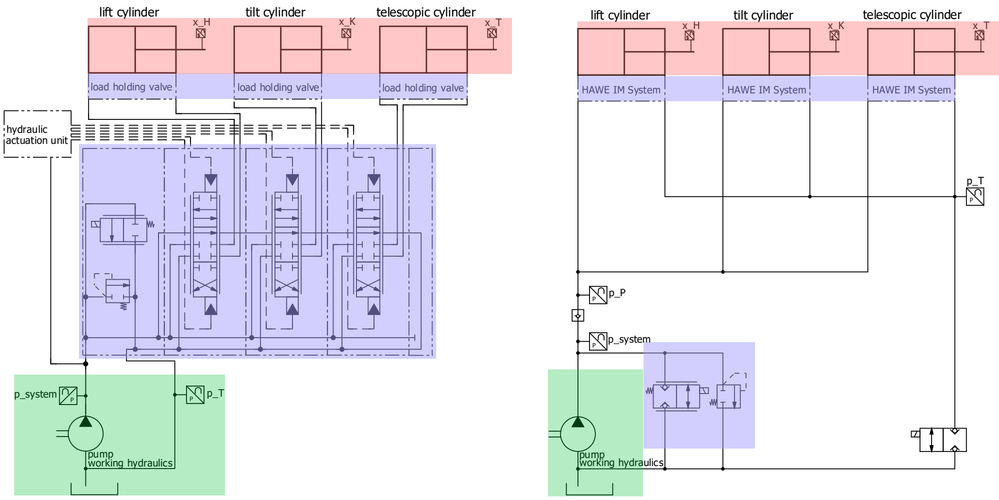

Figure 2: Excerpt of the hydraulic circuits used in this case study. On the left-hand side, the conventional configuration is shown. On the right-hand side, the IM system is shown. The colors indicate the different parts of the system, i.e. actuator, control valves, and pump.

2.2. Conventional Configuration

In conventional mobile machinery – particularly within the lower-cost segment – open-center valves in constant-flow hydraulic systems are commonly employed. This comes from the easy-to-use and cost-effective nature of such system. The reference machine used in this study is likewise equipped with such a baseline hydraulic architecture. This constant-flow system, characterized by a fixed volume flow, serves as the benchmark for evaluating alternative hydraulic system topologies.

On the energy generation side, the baseline system utilizes an axial piston pump with fixed displacement. This can be seen in the green marked sections of the hydraulic circuits depicted in Figure 2. In the electrified machine, which is used in this work, this pump is driven by an electric motor, which also powers additional consumers mounted on the same shaft. The electric motor can be operated at discrete rotational speeds, which can be set by the driver and are regulated by the machine's control unit.

On the actuator side, the telescopic loader features three individually controlled consumers: a lift cylinder, a telescopic cylinder, and a tilt cylinder as depicted in Figure 2. These are responsible for manipulating the kinematic chain.

So, this investigation focuses on quantifying the impact on the hydraulic systems' impact on the overall energy balance of the machine, using working cycles that include the three cylinders. On the left-hand side of Figure 2 the conventional hydraulic configuration for the cylinders, which are marked in red, is shown. Hereby, the conductive section, which is marked by blue color, is realized as an open-center valve system. The valves of each cylinder are implemented as 6/3-way proportional spool valves with hydraulic actuation. An additional component of the conductive path is formed by load-holding valves, which are positioned between the spool valves and the actuators. Those are necessary to prevent free falling loads in case of a hydraulic hose failure.

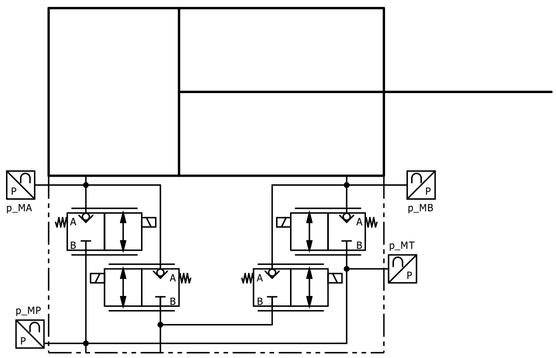

Figure 3: Detailed view on the used IM system. The valve and sensor configuration is the same for all the three equipped cylinders.

2.3. Independent Metering Configuration

As mentioned previously, the second configuration relies on a decentralized independent metering approach. In the selected system design, actuator control is achieved via four 2/2-way proportional seated valves, as shown in Figure 3. This configuration is applied to all three cylinders shown on the right-hand side of Figure 2.

Each valve of the IM system governs a single flow direction, necessitating the simultaneous and interdependent operation of multiple valves. To achieve this, all the seated valves are electronically actuated by a controller applying a PWM signal on the valves' coils. Direction and openness of the valves is controlled by the machine's operator via a joystick. To extend a cylinder, the inflow valve on the A-side and the outflow valve on the B-side are opened, while the retract direction requires to actuate the inflow valve on the B-side and the outflow valve on the A-side, respectively. While the design and tuning of the low-level control strategy is out of scope for this work, interested readers are advised to review e.g. [7, 8, 9].

To emulate a constant-flow system in conjunction with the IM valve architecture, an additional flow-controlled 2/2-way proportional seated valve is employed parallel to the pump. This configuration ensures that the desired flow is delivered from the pump to the actuated function at any time.

It is to note that there are no load-holding valves used in the IM configuration. This comes from the fact that the valve blocks are directly mounted to the cylinders. This decentralized approach enables the seated-valves to additionally fulfill the load-holding functionality. Moreover, it is to note that additional shock-valves are used to protect the cylinders, and no specific pressure compensator is used. To ensure readability, those are not drawn in the right circuit of Figure 2.

3. EXPERIMENTAL SETTING

To enable a consistent and comparable evaluation of the hydraulic systems implemented on the telescopic loader, a standardized test cycle is defined. This cycle is designed to facilitate a comprehensive assessment of both the machine's hydraulic subsystem and its overall energy consumption. The evaluation is conducted using the machine depicted in Figure 1 with the hydraulic circuits from Figure 2 installed. In the following section, the load cycle and evaluation metrics are introduced.

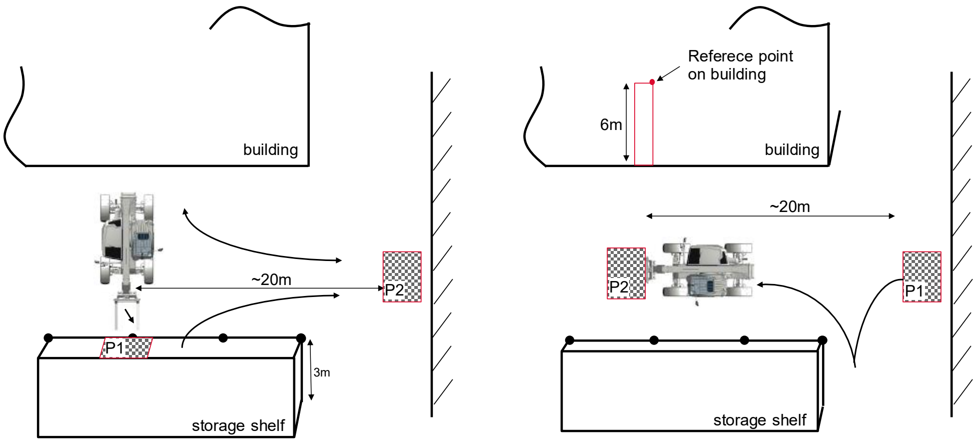

Figure 4: Illustration of the reference cycles used for the benchmark tests. The left image shows the "Pick and Go" task. The right image shows the lifting task.

3.1. Load cycles

Telescopic handlers exhibit highly variable load profiles during operation, depending on their specific application domain in e.g. construction or agriculture. Related machines, such as wheel loaders, are typically evaluated using so-called "Pick and Go" tasks, which are represented by a standardized Y-cycle [10, 11].

Since the aim of this study is to test both hydraulic systems using a representative working scenario, two adapted Y-cycles are used to reflect the operational characteristics of the machine. This includes a simple "Pick and Go" task and a lifting task in which the telescopic functionality is used.

3.1.1. Pick and Go

The "Pick and Go" test cycle involves a defined load being moved in a stacking and transport operation. An illustration of this is shown in the left part of Figure 4. Starting from Position 1 (P1), located at a vertical height of 3 m above ground level, the telescopic loader picks up the load. In the second step, the load is transferred to Position 2 (P2), which is located at ground level, covering a horizontal distance of 20 m. This operation represents a downward transfer from elevation level 1 to level 2.

In the third phase of the cycle, the telescopic loader drives backward in an unloaded state. In steps 4 and 5, the load is picked up again at P2 and transported back to P1, followed by a return to the starting position.

3.1.2. Lifting

To take the far-reaching telescopic functionality into consideration, a second load cycle is introduced. Thereby, the load is transferred to a position that is elevated to 6 m above ground level. This extended reach scenario is meant to represent typical use cases in e.g. construction and logistics where elevated placement of materials is required

In the first step of the cycle, the telescopic handler picks up the load at P1, which lies at ground level. This is followed by a transport manoeuvre that includes a 180-degree rotation of the machine. In the third step, the load is elevated to the height level of the target position P2. The horizontal distance between both points P1 and P2 is 20 m. Upon reaching the target position, the load is lowered back to ground level in step 4. In step 5, the telescopic handler traverses the transport path back to P1. Finally, in step 6, the load is placed back at the initial location. An illustration of this can be seen in the right part of Figure 4.

	Conventional System	**IM System**
#1	800 kg | 23 l/min | Pick & Go	800 kg | 23 l/min | Pick & Go
#2	800 kg | 40 l/min | Pick & Go	800 kg | 40 l/min | Pick & Go
#3	800 kg | 100 l/min | Pick & Go	800 kg | 100 l/min | Pick & Go
#4	1500 kg | 23 l/min | Pick & Go	1500 kg | 23 l/min | Pick & Go
#5	1500 kg | 40 l/min | Pick & Go	1500 kg | 40 l/min | Pick & Go
#6	1500 kg | 100 l/min | Pick & Go	1500 kg | 100 l/min | Pick & Go
#7	800 kg | 23 l/min | Lifting	800 kg | 23 l/min | Lifting
#8	800 kg | 40 l/min | Lifting	800 kg | 40 l/min | Lifting
#9	800 kg | 100 l/min | Lifting	800 kg | 100 l/min | Lifting
#10	1500 kg | 23 l/min | Lifting	1500 kg | 23 l/min | Lifting
#11	1500 kg | 40 l/min | Lifting	1500 kg | 40 l/min | Lifting
#12	1500 kg | 100 l/min | Lifting	1500 kg | 100 l/min | Lifting

Table 2: List of all experiments conducted for this work.

3.2. Load and Pump Conditions

Both test cycles are conducted under two distinct load conditions. In the first scenario, a transport weight of 800 kg is selected. This corresponds to 25% of the machine's nominal lifting capacity, which would be 3200 kg. In the second scenario, a heavier load of 1500 kg is used to evaluate system performance under increased mechanical stress.

Due to the constant-flow hydraulic architecture, the test cycle is executed at discrete volumetric flow rates. In regular operation, the pump speed is adjusted according to the flow demand of each subtask. However, to ensure comparability across measurements, each test run is performed at fixed flow rates. Three discrete flow settings are selected: 23 l/min (motor speed of 1550 rpm and pump speed of 516 rpm), 40 l/min (2550 rpm | 850 rpm), and 100 l/min (6600 rpm | 2200 rpm). Each flow rate is tested in a separate experimental run.

Overall, this results in a total of 12 experiments. For the sake of clarity, an overview of all the experiments and their different settings can be found in Table 2. Each experiment listed in Table 2 contains of two runs, where one is conducted with the conventional system and the other one is conducted with the IM system.

3.3. Evaluation Metrics

To evaluate the machine's energy consumption, measurement data is recorded during operation. Energy usage is captured across multiple reference boundaries (see Figure 5) in order to better understand the impact of different consumers on the overall energy balance.

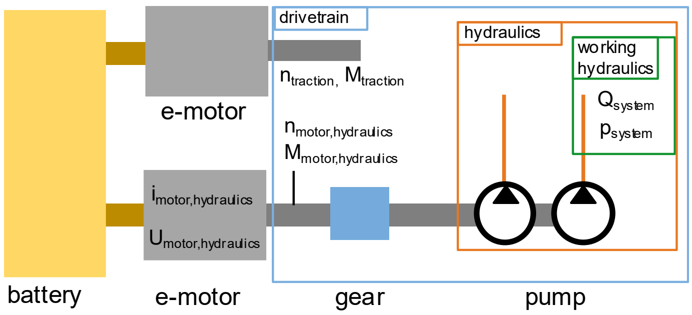

Figure 5: Schematic view of the energetic consumers of the telescopic loader including the reference boundaries.

The power input to the drivetrain encompasses the power for both the traction system and all hydraulic systems, the working hydraulics and auxiliary drives. So, the power of the whole drivetrain is calculated as:

$$P_{Drivetrain} = P_{Traction} + P_{Hydraulics}. \tag{3.1}$$

Hereby, $P_{Drivetrain}$, $P_{Traction}$, and $P_{Hydraulics}$ are powers in W. Moreover, the power of the traction system is:

$$P_{traction} = \omega_{traction} * M_{traction} = 2\pi * n_{traction} * M_{traction} \tag{3.2}$$

with $n_{traction}$ as a rotational speed in 1/s and $M_{traction}$ as a torque in Nm. Moreover, $P_{Hydraulics}$ is defined as:

$$P_{Hydraulics} = i_{Motor,Hydraulics} * u_{Motor,Hydraulics} \tag{3.3}$$

with $i_{Motor,Hydraulics}$ as a current in A and $u_{Motor,Hydraulics}$ as a voltage in V. It is to note that here, both the power consumption for the main and the secondary pump are included. However, this study focuses specifically on the working hydraulics.

Thus, the power output of the main pump is measured in addition to $P_{Hydraulics}$. For this, the system pressure, namely p_{System}, in the working circuit as well as the volumetric flow rate, Q_{System}, is measured. So, $P_{Hydraulics,Working}$ is calculated as:

$$P_{Hydraulics,Working} = p_{System} * Q_{System} \tag{3.4}$$

It is to note that losses introduced by the flow sensor are compensated analytically during post-processing to ensure accurate energy assessments.

Considering the different reference systems, two metrics are used to quantify the energy savings achieved using the IM system. The first one focuses just on the working hydraulics:

$$\delta_{Hydraulics,Working} = 1 - \frac{\int_{t_{start}}^{t_{end}} P_{Hydraulics,Working,IM} dt}{\int_{t_{start}}^{t_{end}} P_{Hydraulics,Working,Conventional} dt}. \tag{3.5}$$

Here, t_{start} and t_{end} is the total working cycle duration in s. The second one focuses on the whole drivetrain:

$$\delta_{Drivetrain} = 1 - \frac{\int_{t_{start}}^{t_{end}} P_{Drivetrain,IM} dt}{\int_{t_{start}}^{t_{end}} P_{Drivetrain,Conventional} dt}. \tag{3.6}$$

Beyond energy-related data, the position of the hydraulic cylinders and the chamber pressures are also recorded. Moreover, the operational state of the drive system is monitored via the rotational speed of the traction drive. Based on the collected measurement data, a comprehensive evaluation of the conductive hydraulic systems is performed with respect to energy consumption and overall system suitability.

4. Experimental Evaluation

The experimental evaluation of both hydraulic systems is conducted using the setting shown in the previous section. In the following, the results are introduced and thoroughly discussed. For this, the time series data of two exemplary working cycles is presented in detail. Afterwards, the energy consumptions of both hydraulic systems are shown and compared.

4.1. Working Cycles

Figure 6 shows an excerpt of the sensor signals of experiment #2 (see Table 2). Here, the volume flow is set to 40 l/min and the weight of the load is 800 kg. Moreover, the Pick & Go task is performed. The diagram illustrates two repetitions of two representative test runs:

the conventional baseline system is shown on the left in blue, while the proposed IM architecture is depicted on the right in orange.

In the first step of the work cycle, indicated by (I) in the graph, the telescopic handler needs to pick up the load at position P1 on the test track. During this phase, the boom arm is moved to an elevated position without load. Under identical boundary conditions, the IM valve system exhibits a system pressure approximately 10 bar lower than that of the conventional system.

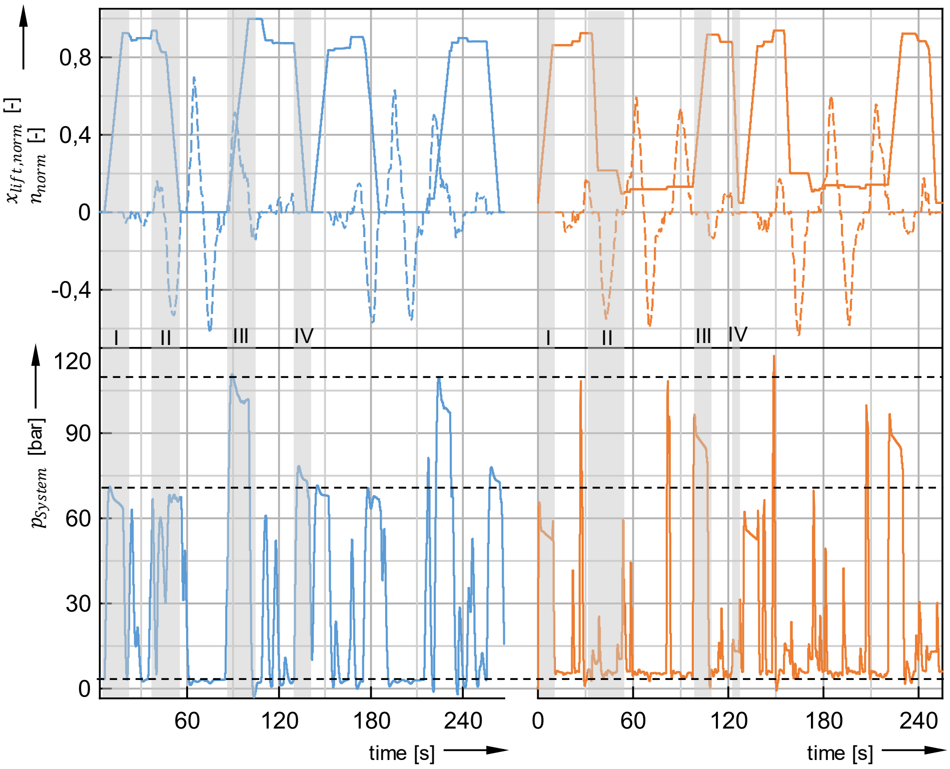

Figure 6: Excerpt of sensor signals from experiment #2 (800 kg | 40 l/min | Pick & Go).

In the second step, indicated by (II), the load is transferred to position P2. This involves lowering the boom arm under load from P1 to ground level. In the conventional system, a system pressure of 65 bar is observed, primarily required to actuate the load-holding valve. In contrast, the IM system operates at a significantly reduced pressure level of approximately 10 bar, which is the system's standby pressure. This comes from the gravity lowering functionality that the IM system offers since no load holding valves are required because of the decentralized mounting.

In the third step, (III), the load is lifted from position P2 back to P1, representing actuator movement under load. Similar to the behavior in (I), the IM system demonstrates a lower system pressure compared to the conventional design. In step four, (IV), the boom is lowered without load. The pressure behavior mirrors that of (II), with the IM system enabling gravity-assisted lowering without the need for additional pressure.

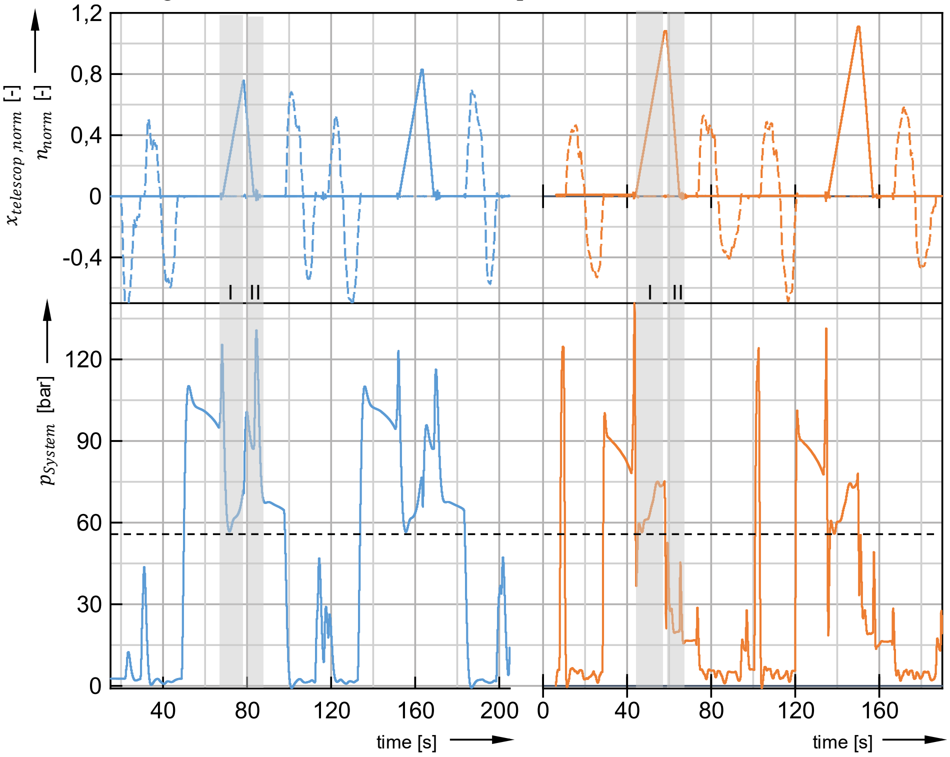

Figure 7: Excerpt of sensor signals from experiment #8 (800 kg | 40 l/min | Lifting).

Figure 7 shows an excerpt of sensor signals from two repetitions of experiment #8 (see Table 2). Again, the volume flow is set to 40 l/min and the weight of the load is 800 kg and the lifting task is performed.

As shown in Figure 7, the segments involving active drive mode without engagement of the working hydraulics exhibit recirculation behavior identical to that observed in experiment #2. Moreover, the lifting arm actuation remains as in the Pick & Go task. Thus, only the segments in which the telescopic cylinder is actuated, are marked in Figure 7.

Here, segment (I) of the cycle involves lifting the load to the target elevation. In this phase, the pressure profiles of both the conventional and IM systems show similar characteristics. In the subsequent lowering operation in segment (II), the conventional system requires a system pressure of 100 bar, primarily driven by the actuation of the load-holding valve. In contrast, the IM system operates at a reduced pressure level of 25 bar, which is sufficient to enable a stable and controlled movement. This pressure requirement is attributed to the static friction within the linear guide mechanism.

The data from both experiments demonstrate that the IM system requires significantly lower operating pressures within the working hydraulics. This reduction is primarily attributed to minimized flow losses resulting from direct piping to the valves and the elimination of hydraulic components, i.e. the load holding valves.

Furthermore, the IM system exhibits substantially lower working pressures during cylinder retraction phases. In these scenarios, the observed pressure corresponds directly to the mechanical requirements of the task, rather than being influenced by auxiliary valve actuation. Due to the system's design, energy savings of up to 98% are achievable in lowering operations where no active hydraulic energy input is required. It is to note that the recirculating flow, inherent to the constant-flow architecture, is included in the overall energy balance and contributes to the system's efficiency profile.

Both working cycles, Pick & Go as well as the lifting task, show a clear tendency when comparing the pressure signatures from the conventional and the IM system. Since the first working cycle corresponds to a more realistic Pick & Go task, based on the unloading of the load on both positions, the following evaluation of the energy usage is going to focus on this particular working cycle with different load conditions as well as flow settings.

4.2. Energy Consumption

Figure 8 illustrates the energy balance of the working hydraulics, highlighting the energy savings achieved by the IM system relative to the conventional architecture as shown in equation (3.5). It is to note that the results are averaged over five runs with each system to ensure repeatability. Moreover, it ensured that the time taken to complete the working cycle remains constant over each run. This analysis, focused exclusively on the working hydraulic subsystem, reveals a clear dependency between load magnitude, volumetric flow rate, and achievable energy savings.

For the Pick and Go task, the IM system demonstrates energy savings ranging from 25% to 40%, depending on the operating point. The observed decline in energy savings with increasing flow rate is attributed to the diminishing influence of gravity-assisted lowering, as flow-induced losses become more dominant within the system.

The load dependency is primarily governed by the behavior of the load holding valves. Their actuation is based on a force equilibrium across the valve ports, with the working port contributing to the balance. Under high load conditions, the resulting cylinder pressure supports the process of valve opening, thereby reducing the pressure differential between the conventional and IM systems.

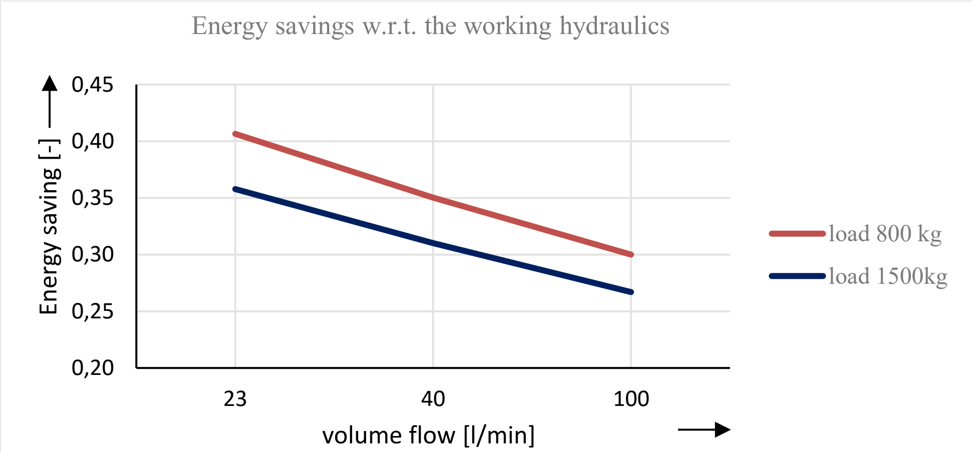

Figure 8: Energy savings $\delta_{Hydraulics,Working}$ of the IM system in the Pick & Go task w.r.t. the working hydraulics.

When extending the energy balance to the entire drivetrain, where the hydraulic consumers as well as the traction drive is included, the Pick & Go task reveals overall energy savings ranging from 15% to 30%. This behavior can be seen in Figure 9. A declining trend with increasing volumetric flow rate is again observed, driven by the same phenomenon of increasingly dominant flow losses.

In contrast to the behavior shown in Figure 8, the drivetrain-level analysis shows an increasing energy savings trend with rising load weight. As the load increases, the proportion of power demand attributed to the hydraulic subsystem grows relative to the electric drive. Consequently, the efficiency gains achieved within the hydraulic domain have a more pronounced impact on the overall energy balance.

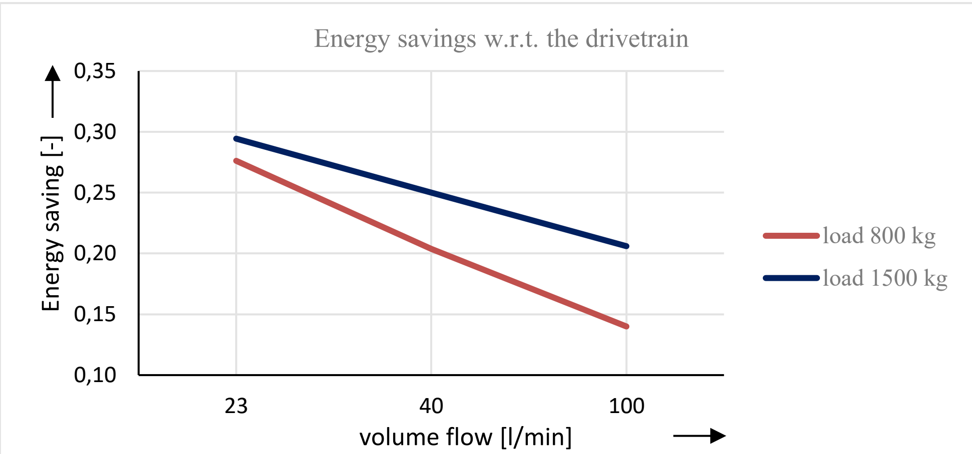

Figure 9: Energy savings $\delta_{Drivetrain}$ of the IM system in the Pick & Go task w.r.t. the drivetrain.

4.3. Practical Considerations

To emphasize the impact of the calculated energy savings, the potential impact of the telescopic loader's overall design on practical considerations is discussed. As listed in Table 1, the battery capacity of the machine used in this work is 63 kWh, which is just enough to use the machine for a working day of 8 hours with one loading cycle.

As stated before, the IM approach saves between 15% and 30% of energy, which can be seen as a range extension between 1.2 hours and 2.4 hours. This allows for more flexible machine usage, especially in areas where flexible shift models are required. Moreover, recharging is faster with a higher remaining battery capacity.

From a financial perspective, reducing the overall battery capacity up to 30% can significantly reduce the cost of the machine. Considering a retail price for off-highway applications of at least 300 € per kWh, the potential cost reduction lies in an area from 2835€ to 5670€ [12]. Additionally, the lower power usage might enable a redesign of the motor-pump unit and inverters, which can lower the price even more.

Overall, it is to mention that the energy gains of the IM approach can trigger further innovation and redesign of the whole machine. This underlines also the importance of taking a holistic approach on the design of modern mobile machinery, since hydraulic system, drive system, and the battery system significantly influence each other.

5. CONCLUSION

This work presents a comprehensive case study, highlighting the energy savings that a modern decentralized independent metering (IM) approach can be realized in comparison to a conventional hydraulic system design. To ensure practical relevancy, the accompanying experiments are conducted on an electrified telescopic loader, a machine which is heavily used in fields as e.g. agriculture and construction. Moreover, test cycles with a clear reference to real-life scenarios are used in order to come to a representative conclusion.

The case study clearly indicates that the decentralized IM approach can significantly reduce the power consumption of the machine's hydraulic circuit up to 40%. Related to the whole drivetrain, there is still a gain of up to 30% possible. Especially the gravity lowering functionality, which is possible because load holding valves are spared, makes nearly energy-free load lowering possible and accounts for a great amount of the savings. However, the pressure levels needed for cylinder extraction tasks are significantly lower using the IM approach.

This shows that redesigning conventional hydraulic circuits can be a great factor in making future machines more efficient. This is a benefit especially for electrified machines, since energy savings directly reduce the battery capacity needed. This reduces the space requirements of the electrical system, the weight of the machine, and the overall invest cost as well as the end users total cost of ownership. However, also mobile machinery using internal combustion engines can benefit from the presented IM approach, since a significant reduction in energy consumption reduces both fuel costs and emissions.

Despite the promising results, there is much room for future work on decentralized as well as centralized IM systems. So, stable control of IM systems is an ongoing challenge and requires modern control techniques. Moreover, the usage of the proposed architecture offers possibilities like regeneration and recuperation, which can be further explored and might even further improve machine efficiency. Overall, it is to state that modern hydraulic architectures such as IM offer a great possibility to design future-proof efficient and sustainable mobile machines.

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