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Universal and Highly Efficient Speed Control of the Hydraulic Motor in a Wire Saw Unit for a Hybrid Constant Flow and Constant Pressure System

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Abstract.

The control of hydraulic motors supplied via long hydraulic lines presents significant challenges, particularly for high-speed operation and loads with high rotational inertia. This study investigates a hybrid hydraulic wire saw attachment equipped with a fixed displacement motor and supplied by two alternative power sources: a constant-flow system driven by a variable-speed internal combustion engine and a constant-pressure system powered by an electric motor with a variable displacement pump. The proposed hybrid configuration combines operational flexibility and mobility with low-noise and low-emission electric operation, while allowing independent operation from either power source. The key contribution of this work is a dual-mode speed control strategy that switches from conventional valve-based flow control during transient phases to direct, valve-less speed control under steady-state conditions. This approach enables energy savings of approximately 13–42% during the steady cutting phase, which accounts for the majority of operating time. Maintaining stable motor speed under varying control modes remains the primary control challenge.

Keywords. Mobile Application, Hybrid machine, Direct Flow control, Construction and Mining Machinery

1. INTRODUCTION

The control of hydraulic motors in construction machinery is a well-established and widely studied topic. However, challenges arise when long hydraulic lines (>20 m) are involved, particularly in achieving precise control of hydraulic motors operating at high rotational speeds (>1000 RPM) and driving loads with substantial rotational inertia. The system analysed in this study is a hydraulic wire saw attachment with a fixed displacement motor implemented as a hybrid configuration, like shown in Figure 1. Hydraulic power is supplied via two distinct sources: a constant-flow system driven by a variable-speed internal combustion engine, and a constant-pressure system powered by an electric motor coupled with a variable displacement pump. This configuration enables the combination of the mobility and flexibility of an internal combustion engine-powered machine with the low-noise and low-CO_2 emission operation of an electric motor. The saw unit can be operated independently by either system.

Zhongyi Quan [1] and Tianliang Lin [2] have proposed various control strategies for hydraulic motors, including multiple combinations of variable pumps and motors tailored to specific application requirements, with the goal of maximizing energy efficiency and recovery. A particular emphasis is placed on different forms of secondary control strategies. Also Berg and Ivantysynova [3] examined secondary control strategies for hydraulic motors utilizing variable displacement.

In contrast, the hybrid concept examined in this work requires a configuration capable of operating effectively with both constant-flow and constant-pressure supply systems with a fixed displacement motor. In this specific application, energy recovery is not deemed advantageous.

Figure 1 Wire saw attachment cutting, mounted on carrier vehicle including internal combustion engine and electric powerpack at the back

The novelty of the proposed concept lies in the ability to switch between two different speed control modes during operation. For the short, undefined setup and cutting initiation phase, speed control is handled conventionally via a flow control valve. Once steady-state conditions are achieved, the system transitions to direct, valve-less speed control—either by adjusting the internal combustion engine speed or through electric motor control using the swivel angle of the variable displacement pump. This enables energy savings of approximately 13–42% in the steady cutting phase, which typically accounts for over 90% of total operation time. The switching mechanism can be realized using a bypass valve to circumvent the flow control valve, a load-sensing (LS) pressure valve for targeted

pressurization of the compensator, or a control spool with a non-linear characteristic.
The main challenge is maintaining a constant motor speed in accordance with process requirements, where either the combustion engine speed or the pump swivel angle serves as the control variable.

2. System Overview

Figure 2 Hydraulic setup of the saw motor

The hydraulic architecture of the system, shown in Figure 2, comprises a telehandler-mounted attachment equipped with a main valve block and a fixed displacement hydraulic motor, as well as two alternative power units: an internal combustion engine driving a gear pump and an electric motor coupled to a variable displacement axial piston pump. The attachment can be operated independently by either power unit, depending on the availability of an external electric power supply. The decision to combine a constant-pressure system for the electric drive with a constant-flow system for the internal combustion engine was made at an early stage of development. This dual-system approach was necessary because modifications to the existing hydraulic system of the telehandler were not permitted, while the electro-hydraulic power supply was required to be as compact and energy-efficient as possible—requirements that could not be met using a constant-flow system alone.

2.1. Hydraulic Plan

The main hydraulic block is composed of several standard sandwich-plate valves assembled to provide all functions required for operating the attachment. For the sake of clarity and reduced complexity, only the components relevant to motor control are shown in the schematic Figure **3**. The block incorporates an integrated pilot oil supply unit as well as a two-way safety valve.

For the experimental setup, two flow meters and pressure sensors were integrated into the hydraulic circuit: one upstream of the main valve block to measure the pump supply flow to the attachment, and a second upstream of the hydraulic motor. This instrumentation allows the determination of flow and pressure losses across the main valve block by comparison of the measured values.

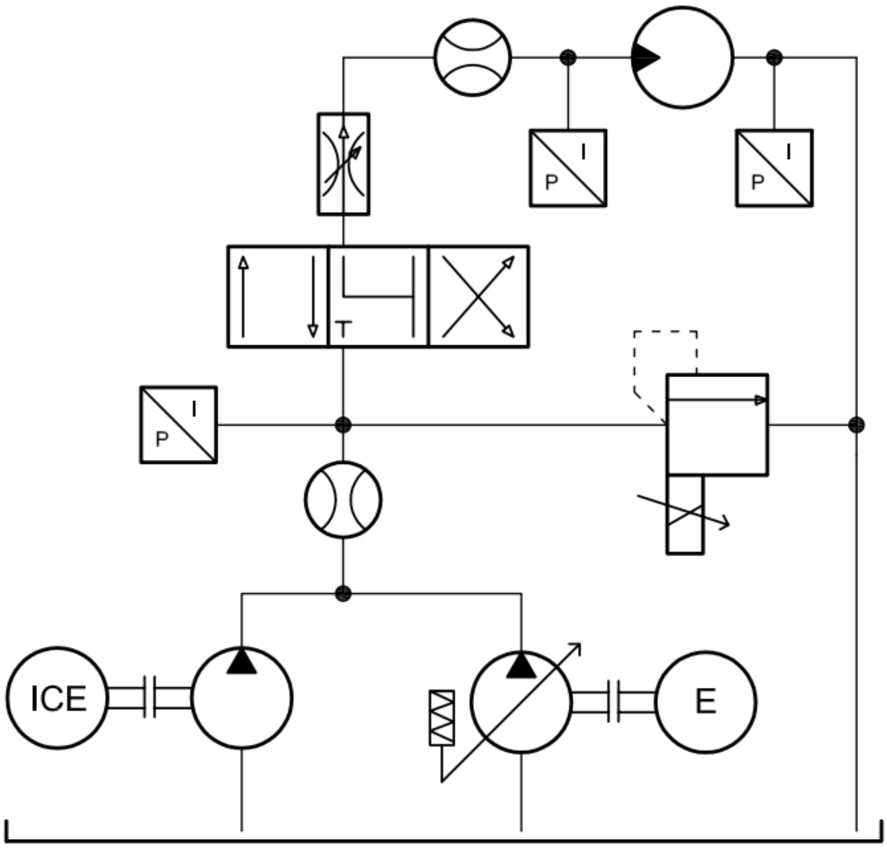

Figure 3 Hydraulic schematic (simplified)

2.2. Control Strategies

First, the hydraulic motor is controlled to reach the target rotational speed required to achieve the desired wire velocity. Second, the feed rate of the wire into the stone is adjusted to obtain the specified motor torque. These target values are predefined and are not the subject of this study. To ensure efficient operation, the hydraulic flow rate and pressure must be supplied in accordance with the required speed and torque demands, while avoiding excess energy provision.

2.2.1. Wire Speed

The wire speed is controlled using a 4/3-way valve equipped with a pressure compensator and a PID controller. This configuration exhibits stable behaviour, as the pressure compensator inherently adapts to pressure fluctuations. The PID controller primarily compensates for leakage and temperature-induced variations and contributes to the attenuation of oscillations. Oscillatory behaviour is a well-known challenge in the control of hydraulic motors driving large inertial loads in combination with long hydraulic hose lengths. During experimental testing with the proposed setup, oscillations were observed but without overshoot. Within the scope of these experiments, it was not possible to clearly distinguish whether the oscillations originated from the hydraulic circuit or from instabilities in the cutting process itself.

2.2.2. Hydraulic supply at constant pressure circuit

Hydraulic flow and pressure are supplied by a variable displacement axial piston pump driven by a three-phase electric motor. The pump is equipped with a pressure controller actuated by an electrically controlled valve, allowing the pump pressure level to be adjusted, while the flow rate is not actively regulated. As a result, the system operates as a pseudo load-sensing configuration. Due to the excessive length of the hydraulic hoses between the pump and the valve block, conventional transmission of a load-sensing signal via a hydraulic line is not feasible. However, under the relatively steady operating conditions encountered during cutting, load sensing is implemented electronically by adjusting the pump pressure based on measured pressures and predefined target values. The pump continuously supplies the flow required to maintain the desired pressure level at the attachment. The pressure difference between the pump outlet and the load pressure is regarded as a necessary and acceptable loss in this configuration.

To eliminate this pressure-related loss, the original pressure-controlled pump was replaced with a flow-controlled pump. The new pump is mechanically identical but is equipped with an electronic controller and a swashplate angle sensor, enabling the implementation of various control strategies purely through software. Load-sensing control can be realized using pressure feedback, while flow and power control are achieved by regulating the swashplate angle and motor speed. To minimize energy losses associated with Δp and to improve stability with respect to oscillations, motor speed is now controlled directly via the pump swashplate angle, while the 4/3-way valve is kept fully open. Consequently, pressure losses are limited to flow-dependent losses, with no additional losses introduced by control valves. Furthermore, the propensity for valve-induced oscillations is significantly reduced, as the valves operate in a fully open state.

2.2.3. Hydraulic supply at constant flow circuit

Hydraulic flow is supplied by a gear pump driven by the internal combustion engine of the telehandler. This pump provides hydraulic power for all machine functions, including boom actuation, stabilizers, and auxiliary attachments such as the wire saw. Consequently, the telehandler is equipped with a load-sensing hydraulic system in which a pressure compensator diverts excess flow to the tank at load pressure. In this configuration, only the pump flow rate and the engine speed can be adjusted. Owing to the operating principle of the circuit, auxiliary attachments must be controlled via the telehandler control valve to ensure proper functionality. When the flow rate is reduced, the pressure compensator increases the system pressure to its maximum, resulting in unnecessary pressure losses between the actual load pressure and the maximum system pressure.

To mitigate this issue, a proportional pressure relief valve was integrated into the circuit, and the relationship between engine speed and pump flow rate was characterized. By setting the relief valve to a pressure level equal to the load pressure plus a defined Δp, the valve effectively acts as a pseudo pressure compensator. In this operating mode, the telehandler's pressure compensator remains closed, and excess flow is discharged at a lower pressure level. With appropriate adjustment of the engine speed, the amount of excess flow can be minimized, resulting in improved overall efficiency.

To further reduce these losses, an alternative control approach was investigated. This concept, identified inadvertently during testing, involves fully opening the 4/3-way valve while closing the pressure relief valve. Under these conditions, the hydraulic motor rotates at a speed directly determined by the pump flow rate. Provided that the internal combustion engine speed can be adjusted rapidly and accurately, the motor speed can be effectively controlled through engine speed regulation.

2.2.4. Additional hydraulic actuators in the circuit

In addition to the hydraulic motor, the pilot oil supply of the valve block, the wire tension system, and the feed cylinders operate simultaneously. Wire tension is maintained at a constant level by a hydraulic cylinder and therefore requires only the leakage flow of its load-sensing valve. The feed cylinders operate at a comparatively low velocity relative to the hydraulic motor, resulting in negligible hydraulic flow demand.

3. DESCRIPTION OF A STANDARD CUT

Cutting stone with a diamond wire is a comparatively slow process relative to other cutting methods and materials. Feed rates depend on several parameters, including stone type, diamond bead characteristics, cutting width, and wire speed. In the application considered here, feed rates typically range from 1 to 20 cm/min, with an average of approximately 4–6 cm/min. Cutting stones with heights between 1 and 3 m and widths between 2 and 5 m generally requires between 30 min and 3 h, with an average cutting duration of about 1 h.

The cutting process can be divided into three distinct phases, shown in Figure 4. The initial phase corresponds to the start of the cut, during which the wire is not yet fully engaged in the stone due to the rough surface and the absence of a fully developed cutting arc. During this phase, the motor torque is low and unstable until full engagement is achieved and a stable cutting arc is formed, which typically takes approximately 5 min.

The second phase constitutes the main cutting stage and represents the longest portion of the process. During this phase, the wire is continuously fed through the stone under a relatively constant high motor torque with only minor variations. This phase accounts for approximately 90% of the total cutting time.

The final phase occurs toward the end of the cut, when the wire is no longer actively fed because the guide wheels approach the ground. The wire is then used primarily to lower the cutting arc as far as possible to ensure a clean edge before the stone is separated at the completion of the cut. During this phase, the motor torque decreases continuously to a low level, and the process concludes once only a minimal remaining section is left to be cut, typically lasting a few minutes.

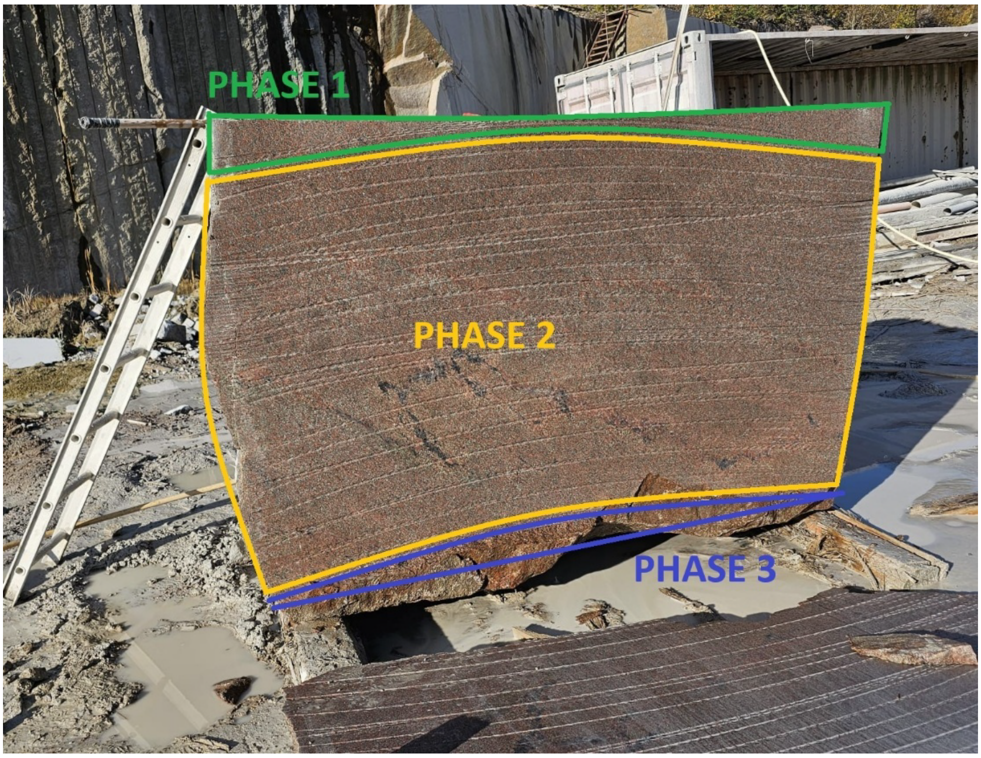

Figure 4 Stone block with the different cutting phases marked

4. Test and results

The experimental investigations were conducted using the same system configuration to ensure maximum comparability. All test cuts were performed on the same stone using the same diamond wire, and switching between the different control modes was carried out during the cutting process to minimize variations in operating conditions.

Due to the inherently irregular nature of the cutting process over time, a comparison based on instantaneous values is not meaningful. Instead, mean values over selected time intervals were used for the evaluation of the results. These intervals were chosen to be representative of typical cutting conditions and were kept relatively short to ensure comparable cutting states. Based on prior experience, these representative periods can be clearly identified within the measured data. The selected intervals are indicated by red and blue dotted rectangles in Figure 7 and Figure 8, and the corresponding mean values are summarized in Table 1. The blue rectangles denote measurements obtained using the proposed control mode, while the red rectangles represent measurements using the conventional control mode. For improved comparability, all measured quantities were normalized using system-specific reference values p_{ref} and Q_{ref} according to:

$$p = \frac{p_n}{p_{ref}} \quad , \qquad Q = \frac{Q_n}{Q_{ref}} \qquad (4.1)$$

Figure 5 presents the data for the constant-pressure system operating under both control modes. The test sequence began with the proposed control mode, in which motor speed is regulated via the pump swashplate angle. After 175 s, the system was switched to the conventional valve-based control mode. The variable Q_{supply} denotes the flow rate supplied by the pump to the system, while Q_{motor} represents the flow rate delivered to the hydraulic motor. The difference between these values corresponds to pilot oil consumption and leakage flows discharged directly to the tank. The pressure p_{supply} is the main supply pressure measured upstream of the main valve block, p_{tank} is the motor return pressure, and p_{motor} is the load pressure measured at the motor inlet. The motor torque-related pressure p_{torque} is calculated as the difference between the motor load pressure and the return pressure.

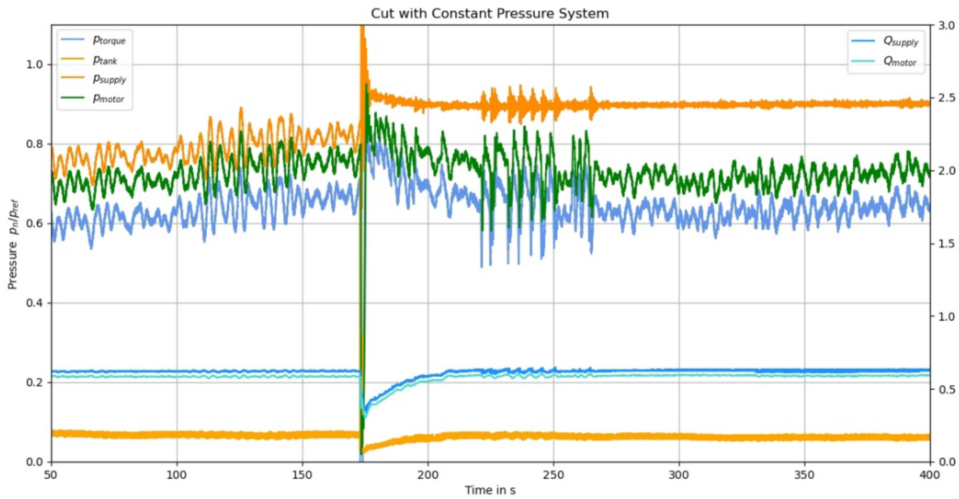

Figure 5 Test with constant pressure system

Figure 6 presents the data for the constant-flow system operating under both control modes. The test sequence began with the proposed control mode, in which motor speed is regulated via the internal combustion engines speed. After 260 s, the system was switched to the conventional valve-based control mode.

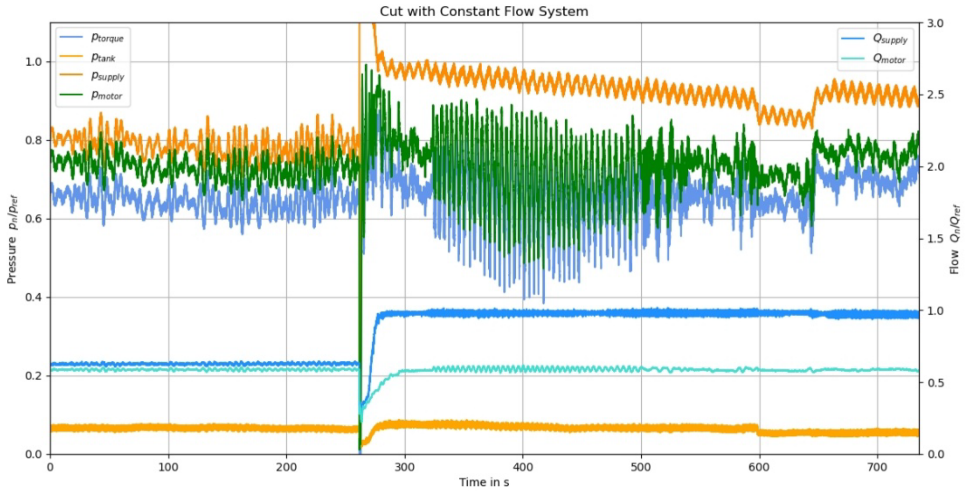

Figure 6 Test with constant flow system

4.1. Test with the constant pressure system

In the following figures Figure 7 and Figure 8, the green shaded areas represent the torque-related values, or equivalently the hydraulic power required for the cutting process. The yellow shaded areas indicate the pressure losses occurring in the valve block and the hydraulic hoses upstream of the motor. It is clearly evident that, after the mode switch, a significantly larger amount of hydraulic energy is dissipated across the pressure compensator. As expected, no significant difference in the flow rates can be observed, since the pump supplies only the flow required by the system, both before and after the change in control mode. The blue shaded areas represent the flow passing through the hydraulic motor.

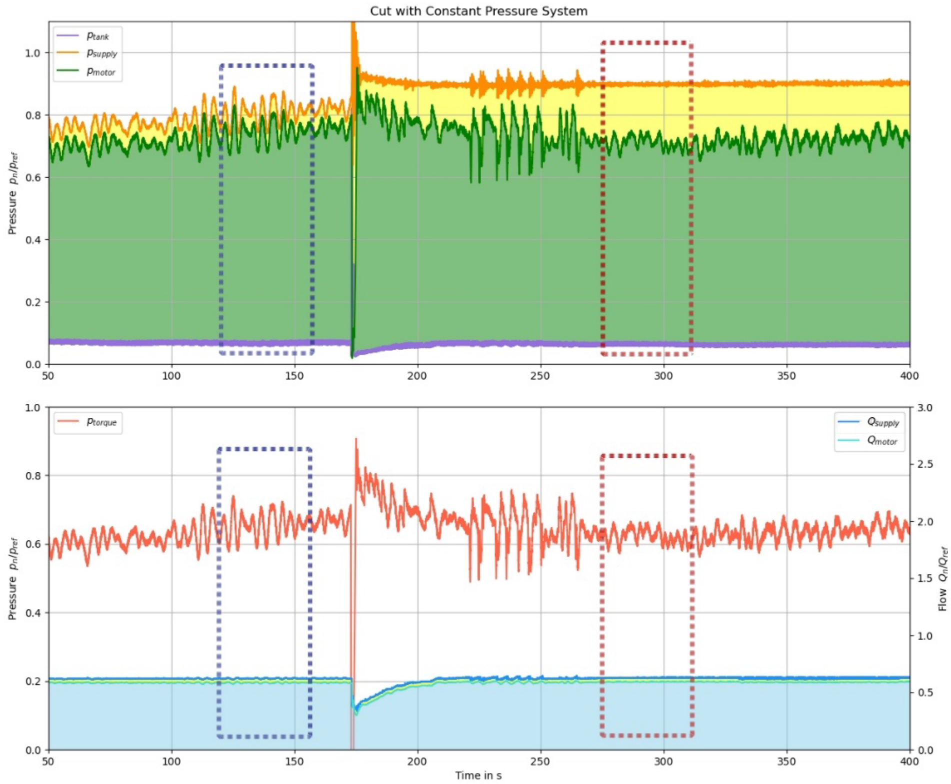

Figure 7 Result of the constant pressure system

4.2. Tests with the constant flow system

For the constant-flow system, the results are generally comparable, with the exception of the flow behaviour. The difference in flow is illustrated in Figure 8, where the yellow shaded area between the flow curves highlights the excess flow. This excess flow results in not only increased losses due to the oil discharged through the pressure relief valve, but also a substantially higher magnitude of loss than anticipated. During the experiments, it was observed that approximately 60% more flow than required was discharged, rather than the expected surplus of around 10%. This discrepancy was attributed to inaccuracies in the calibration of the engine-speed–to–flow characteristic, which therefore requires further refinement.

Because correct calibration could not be achieved at the time of testing, an additional efficiency calculation was performed within this study using the intended target values. This supplementary analysis illustrates the expected system performance under properly calibrated operating conditions.

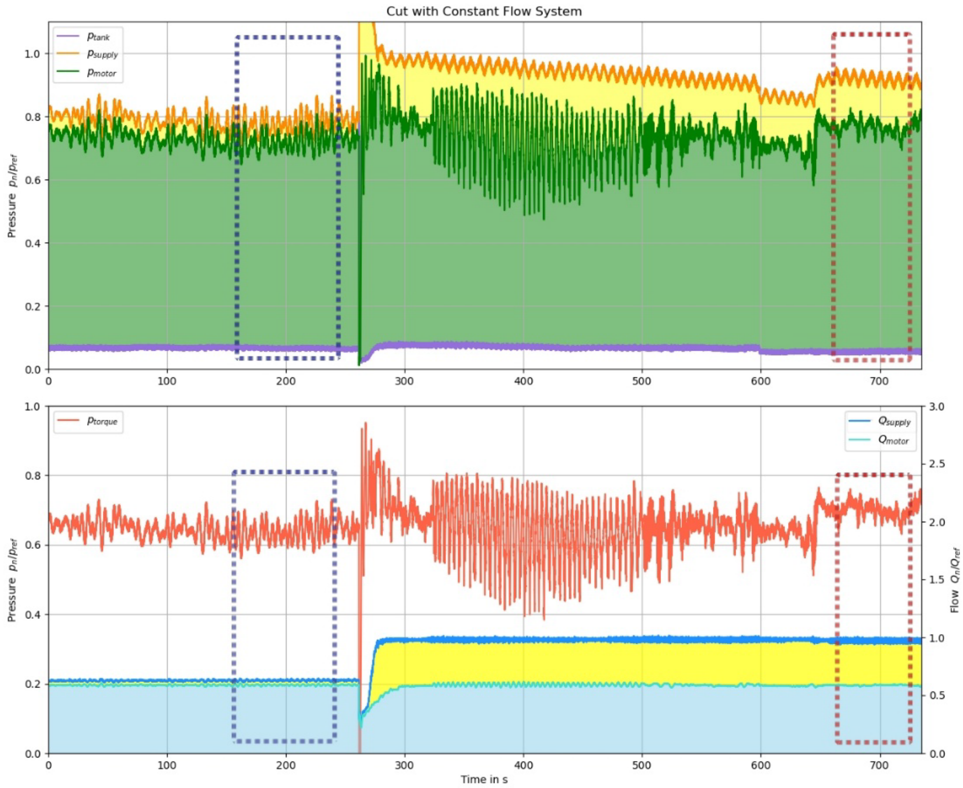

Figure 8 Result of the constant flow system

4.3. *Results*

To enable a meaningful comparison of the measured results, the converted hydraulic power was calculated based on pressure and flow rate measurements. This approach also allows a direct determination of potential energy savings. Furthermore, it facilitates comparison between different experiments, as all tests were conducted at a single operating point of the system.

4.3.1. *Calculation of the results*

To determine the total input power P_{in} the hydraulic flow rate and pressure measured upstream of the valve block are used:

$$P_{in} = Q_{supply} * p_{supply} \qquad (4.2)$$

The total cutting power at the motor P_{cut}, including mechanical losses, is calculated using the flow rate upstream of the motor and the pressure difference across the motor P_{torque}:

$$p_{torque} = p_{motor} - p_{tank} \qquad (4.3)$$

$$P_{cut} = Q_{motor} * p_{torque} \qquad (4.4)$$

The overall leakage flow of the valve block Q_{delta}, including the flow through the load-sensing pressure relief valve, is determined as the difference between the flow rates upstream and downstream of the valve block:

$$Q_{delta} = Q_{supply} - Q_{motor} \tag{4.5}$$

Based on this, the power losses of the valve block $P_{valveblock}$ are calculated as:

$$P_{valveblock} = Q_{motor} * \left(p_{supply} - p_{motor}\right) + Q_{delta} * p_{supply} \tag{4.6}$$

This formulation allows the influence of the compensator state (active or fully open) and the presence of flow through the pressure relief valve to be evaluated. For comparison between control modes, the difference in valve block losses is defined as:

$$P_{valveDiff} = P_{valveblock\ NewMode} - P_{valveblock\ Standard} \tag{4.7}$$

The remaining losses, which include the return line and return filter losses, are calculated as

$$P_{lossOther} = P_{In} - P_{cut} - P_{valveblock} \tag{4.8}$$

The efficiency improvement is quantified by the saving factor, defined as the ratio of the reduced valve block losses to the input power:

$$Saving = \frac{P_{valveDiff}}{P_{In}} \tag{4.9}$$

Table 1 summarizes the calculated results for the comparison of control modes for both the constant-flow and constant-pressure systems. The third column presents the constant-flow system assuming an ideal flow rate under standard control conditions. The green-highlighted cells indicate improvements between control modes in terms of flow- and pressure-related losses, while the grey-highlighted cell denotes an ideally selected reference value.

	Constant Flow		Constant Pressure		Const. Flow improved	
	Standard	NewMode	Standard	NewMode	Standard	NewMode
p_{supply}	0,919	0,783	0,898	0,805	0,919	0,783
Q_{supply}	0,973	0,629	0,627	0,621	0,703	0,629
P_{In}	0,894	0,492	0,563	0,500	0,646	0,492
p_{torque}	0,700	0,642	0,625	0,651	0,700	0,642
Q_{motor}	0,586	0,588	0,592	0,584	0,586	0,588
P_{cut}	0,410	0,377	0,370	0,381	0,410	0,377
p_{motor}	0,767	0,725	0,712	0,742	0,767	0,725
Q_{delta}	0,387	0,041	0,035	0,037	0,117	0,041
$P_{valveblock}$	0,444	0,066	0,142	0,066	0,197	0,066
$P_{lossOther}$	0,039	0,049	0,051	0,053	0,039	0,049

$P_{valvediff}$	-0,378	$P_{valvediff}$	-0,076	$P_{valvediff}$	-0,131
$Saving$	-42,32%	$Saving$	-13,47%	$Saving$	-20,21%

Table 1 Test results

The results indicate that, for the investigated application, between 13% and 42% of the input energy can be saved by employing the proposed control mode. Even with an improved motor-speed–to–flow characteristic, the constant-flow system shows a potential energy saving of approximately 20% of the actual input power when operated with the new control strategy.

The Sankey diagrams provide an additional means of comparing energy losses. In these diagrams, all power flows are normalized with respect to the cutting power P_{cut}, which is used as the reference value. Consequently, the diagrams illustrate the differences in required input power for achieving the same cutting power.

Figure 9 illustrates that, in the constant-flow system under conventional control, approximately 108% of the cutting power is dissipated as losses to achieve 100% cutting power. When the new control mode is applied, these losses are reduced to approximately 17%, indicating a substantial reduction in dissipated energy and, correspondingly, in heat generation within the system.

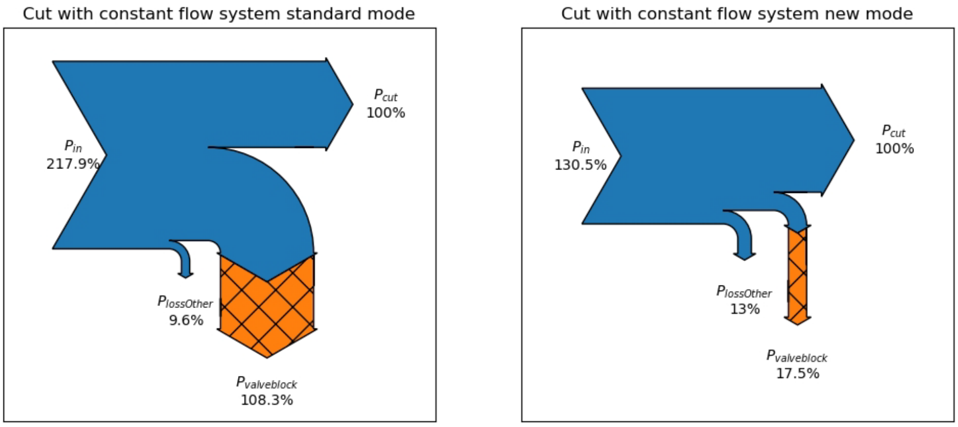

Figure 9 Sankey diagrams for constant flow system

Figure 10 further demonstrates that, for the constant-pressure system, a difference of approximately 20% in input power is required to achieve the same cutting power under conventional control. Both systems exhibit identical results when operated in the new control mode, which is expected, as the valve configurations and functions are identical in this case. In the proposed control mode, hydraulic flow is governed solely by the pumps, while system pressure adjusts according to the motor load.

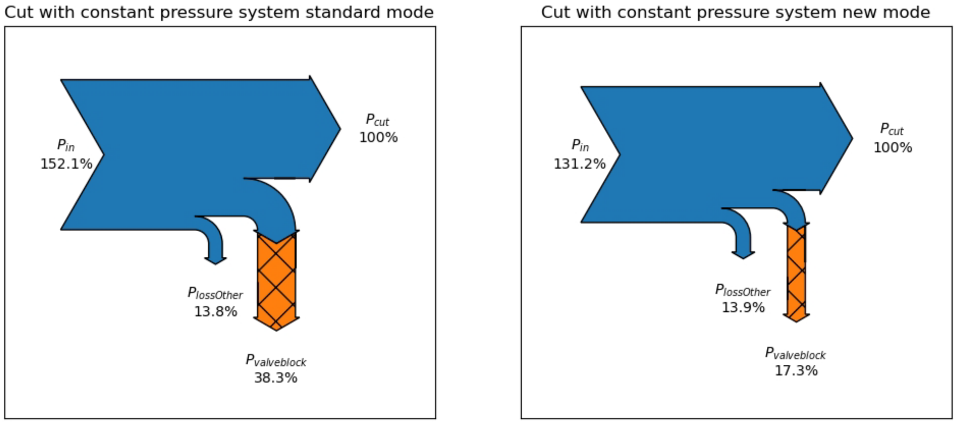

Figure 10 Sankey diagrams for constant pressure system

5. DISCUSSION AND CONCLUSION

The proportion of wasted energy observed in the constant-flow system is relatively high; however, the experiments were conducted on an entirely original system without additional modifications, apart from the integration of the new pump and measurement sensors. Further optimization of this configuration is therefore expected in future work. Nevertheless, even with improved adjustment, the results indicate a substantial potential for energy savings.

Additional improvement measures are conceivable for the investigated system. For example, the electrically driven piston pump could be replaced by a gear pump, and the main valve block could be equipped with a central pressure compensator, similar to an open-loop hydraulic system. Another potential approach would be the use of a variable displacement hydraulic motor capable of directly controlling its own rotational speed. However, the modifications presented in this study were deliberately kept to a minimum and can be implemented without major changes to existing machines. To further enhance system efficiency, other components such as hydraulic hoses and filters could also be reconsidered, as the experiments consistently indicated losses of approximately 13% attributable to these elements.

Further investigations at additional operating points would be beneficial, and the influence of oil temperature should be taken into account. Although the experiments were performed under nearly identical oil temperature conditions, explicitly incorporating temperature effects into the calculations would improve the accuracy of the results.

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Biographies

Daniel Kriegl received a Master's degree in Mechanical Engineering from Graz University of Technology (TU Graz) in 2018. Since 2016, he has been employed at the Institute of Manufacturing Technology at TU Graz as well as at Schwing GmbH in St. Stefan, Carinthia. In 2018, he began doctoral studies with a research focus on hydraulic drives and control systems.