Improving the Energy Efficiency of Single Actuators with High Energy Consumption: An Electro-Hydraulic Extension of Conventional Multi-Actuator Load-Sensing Systems

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Abstract

A load sensing (LS) supply in combination with control valves is one of the most common solutions for the actuation of implements on heavy-duty mobile machines (HDMMs). A major drawback of this approach is its relatively low energy efficiency due to metering losses—especially for multi-actuator operation and load braking. Several novel, more efficient concepts have already been proposed but involve high component costs for each actuator, which is not acceptable for HDMMs with many actuators that have a medium to low energy turnover. Therefore, this work proposes a novel system design which is based on a conventional LS system—for cheap operation of a high number of low-energy-consuming actuators—but allows to avoid metering losses for single high-energy-consuming actuators by replacing their metering valves with electric-generator-hydraulic-motor (EGHM) units that work similar to pump-controlled concepts. The benefits of the novel concept are explained in detail by looking at the three main throttling functions of an actuator in a typical valve-controlled LS systems, which the novel concept avoids by applying pressure in the actuator return lines and recuperating energy electrically instead of dissipating it by throttling. Furthermore, advantages and challenges for the novel concept are analyzed, and ways to address the latter are presented. Before the novel concept is simulated, the required control algorithms are presented. The simulation study in Amesim and Simulink focuses on a telehandler that utilizes the novel concept for the boom, extension and tilt actuators. Simulation results show that the novel system can decrease the required input energy for typical duty cycles by up to 34% compared to a conventional LS system. At the same time, simulations show that, from an economic perspective, it seems most reasonable to utilize the novel EGHM units only for the boom and extension actuators of the studied telehandler.

Keywords: Electro hydraulics, energy efficiency, cost efficiency, heavy-duty mobile machinery, load-sensing system

1 Introduction

Hydraulics is the state-of-the-art technology for actuating the implements of heavy-duty mobile machines (HDMMs), such as excavators, wheel loaders, telehandlers or cranes. The main advantages that make hydraulics indispensable in this field are its power and force density, robustness, ease of linear actuation, and the low effort for adding multiple valve-controlled actuators to a single hydraulic supply unit.

On the downside, conventional hydraulic implement systems have the common disadvantage of low energy efficiency. Regardless of the specific type, open-center, negative-flow-control or load sensing (LS), conventional hydraulic implement systems achieve motion control by means of valves which lead to metering losses. For multiple actuators that are supplied by the same pump, the losses are especially high if the loads vary between actuators and valves need to compensate the load pressure differences by throttling. Such a system still requires energy—under bad conditions even a lot—when a load needs to be braked and energy could actually be recuperated.

Even though the low energy efficiency of these systems leads to higher energy consumption, for a long time, it has been accepted by industry for the sake of low component costs, and more efficient alternatives could only be perceived in academia. However, recently, energy efficiency or emissions, respectively, are becoming more important for industry since climate change is leading to stricter emission legislation and more extensive political programs started. An example is the European Union's Green Deal, which aims at zero net greenhouse gas emissions by 2050 and lists a "Proposal for more stringent air pollutant emissions standards for combustion-engine vehicles" in its action plan for 2021 [1]. A first approach to meet those new requirements is to design HDMMs that are still diesel-powered but use hydraulic systems with increased efficiency. A step further, HDMMs can be electrified, which allows to avoid local emissions entirely and enables zero-emission construction sites that become more and more common in cities such as Oslo for example [2]. Nevertheless, also for electrified machines, energy efficiency is highly important since the capacity of modern batteries is still rather limited and high efficient systems are the only way to achieve sufficiently long operation times.

Accordingly, there is a major interest in the development of new concepts that are more efficient than conventional valve-controlled systems but still competitive from a cost and performance perspective. So far, a concept that satisfies all demands is missing. Independent metering, for example, reduces metering losses but does not eliminate them entirely, and currently available valves do not fully satisfy the needs of such systems [3]. Furthermore, several concepts exist that can eliminate metering losses entirely and even recuperate energy from braking loads, such as hydraulic transformers, digital hydraulics, displacement control or electro-hydraulic actuators (EHAs). However, hydraulic transformer concepts are, so far, not reasonable due to the low efficiency of transformers that could be formed out of commercially available components [4]. Similarly, several digital hydraulic concepts depend on high performance valves that are, as well, commercially not available yet [5]. On the other hand, digital concepts that are based on multi-chamber cylinders proved to be feasible [6], but due to the more complex design the costs for each cylinder rise. The component costs rise as well for displacement controlled concepts or EHAs since a separate pump or pump and motor, respectively, is necessary for each actuator. Furthermore, issues with cooling, filtering, mode-switching, or low stiffness are compromising pump-controlled systems [7].

All in all, a few novel concepts appear more efficient and feasible from a technical point of view, but certain issues remain and generally the component costs per actuator are increased significantly. This might be acceptable as long as the reduction in energy costs can compensate the increased capital costs over time– in other words, if the payback time is short enough. For main actuators with high energy turnover and amounts of braking loads, such as a boom lift actuators, this might be the case; however, it is very common for HDMMs to have also several auxiliary drives that have a low energy turnover due to being rarely actuated or having low loads for most of the time. For those actuators, the potential for saving energy is rather low and increased component costs cannot be justified—the payback time of novel, sophisticated concepts would be too long. Moreover, many HDMMs traditionally offer a hydraulic power-take-off for attachments, which is a simple connection to the centralized pump supply on most conventional machines. For the aforementioned novel concepts, establishing such a power-take-off would require additional effort. Accordingly, it is advisable to keep up a conventional hydraulic supply, such as an LS pump, on the HDMM—for auxiliary actuators and flexible power take of— and apply novel concepts only for the main actuators.

Instead of having two separate hydraulic systems on the same machine, this paper proposes a novel energy-efficient concept which can be integrated into an existing LS system. That way, conventional valve-controlled actuators are still compatible with the hydraulic system and, in contrast to typical EHA concepts for example, the novel concept can benefit from the connected LS supply in terms of actuator stiffness, oil maintenance, handling of large cylinder sizes and more. In the next section, the novel concept is explained in detail, followed by a section dealing with the control of actuators that apply the novel concept. Another section is dealing with the simulation model of a telehandler that was used to compare the energy consumption with the novel concept to the consumption of a purely conventional LS system. The simulation results are discussed and followed by a conclusion and an outlook section.

2 Novel System Concept

The main idea of the novel concept is to replace the conventional control valves of single actuators in an LS system by an electric-generator-hydraulic-motor (EGHM) unit which requires only an additional directional valve for switching the direction of movement and for load holding. The energy that is conventionally throttled and lost in the form of heat can be converted into electric energy and reused this way, which leads to a significantly increased energy efficiency. In Figure 1, the difference between actuators with the novel concept (left side) and a conventional actuator (right side) can be seen. Any number of novel and conventional actuators can be connected to the same LS pump, and operation is only limited by the maximum flow that can be provided by the pump.

Ideally, the energy that is recuperated by the EGHM units is directly used again by the LS pump. Therefore, a combination of a servo electric machine (EM) and a fixed displacement pump is ideal to form the LS supply, as it was already shown in [8] or [9] for example. However, conventional variable-displacement LS pumps driven by internal combustion engines can be used alternatively if an additional EM is attached to support the engine, as shown in Figure 1 on the bottom right. Accordingly, the novel concept does not rely on a full electrification of the HDMM, and diesel can remain the primary energy source if desired.

Moreover, Figure 1 shows that the load pressures of each actuator are sensed, transmitted and processed electronically instead of electro-mechanically, which was already described as electronic LS in [10] for example. As a result, the pump pressure is still controlled in order to reach the maximum load pressure plus an LS margin, but the margin is defined in the software and can be adjusted, which is necessary for the novel concept as will be explained in Section 3. Furthermore, it should be noted that the concept is applicable for rotary actuators, such as an excavator swing, as well, but only cylinders will be discussed in this paper.

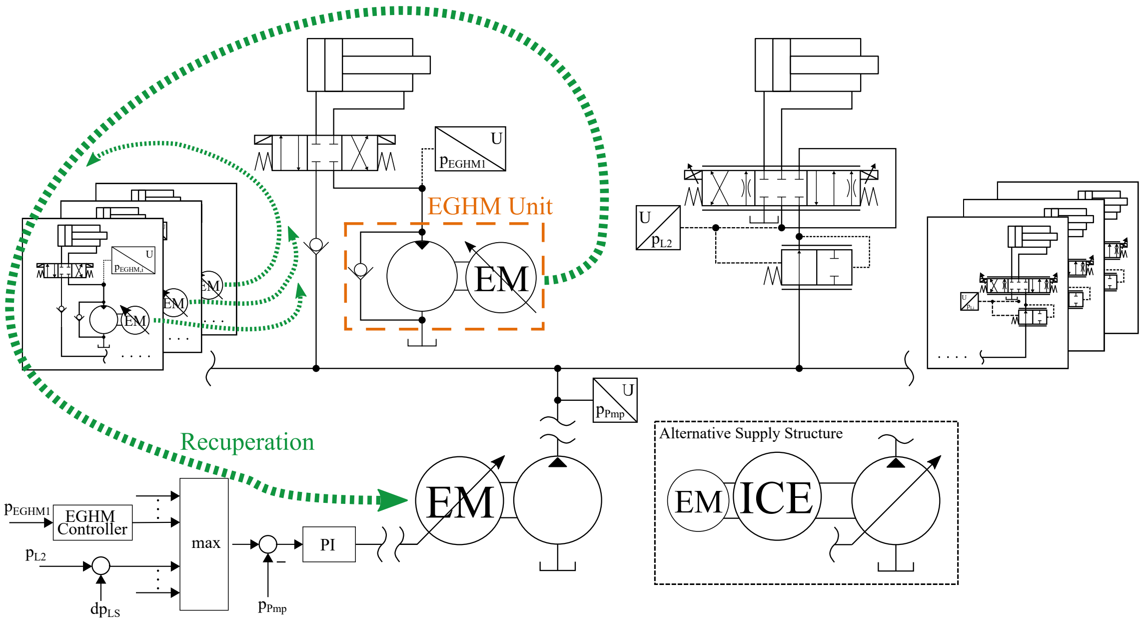

Figure 1: Simplified schematics of LS system including undefined number of actuators with novel concept on the left side and conventional valve-controlled actuators on the right side

2.1 Replacing Throttling by Electric Braking

In order to understand how the novel EGHM unit can fulfil the control functions of conventional valves and to analyse under which conditions the efficiency improvement of the novel system is most significant, the throttling mechanisms of conventional LS control valves are analysed. Three main mechanism can be identified, each related to one valve in the schematics that are shown in Figure 2a.

2.1.1 Throttling for Speed Control

Next to controlling the flow direction, the flow control valve V1 in Figure 2 controls the actuator speed by metering the supply flow of the actuator. This is achieved by varying the relative valve spool displacement y_{vlv}, knowing that the flow across the metering edge, in this case Q_{spl}, follows the equation

$$Q_{\mathrm{spl}} = y_{\mathrm{vlv}} \cdot c_{\mathrm{vlv}} \cdot \sqrt[2]{\Delta p_{\mathrm{ctr}}} \quad . \tag{1}$$

c_{vlv} depends on the valve geometry as well as fluid properties and can be assumed constant. In order to achieve a sufficient maximum flow with reasonably large valve sizes and maximum spool displacements, the pressure drop across the edge Δp_{ctr} must be high enough. Common are pressure drops around 20 bar, which are held constant by the pressure compensator (V2 in Figure 2) in order to obtain a proportional behavior between actuator speed and spool displacement. The resulting losses at the flow control valve equal the product of Q_{spl} and Δp_{ctr} and can be seen in Figure 3a.

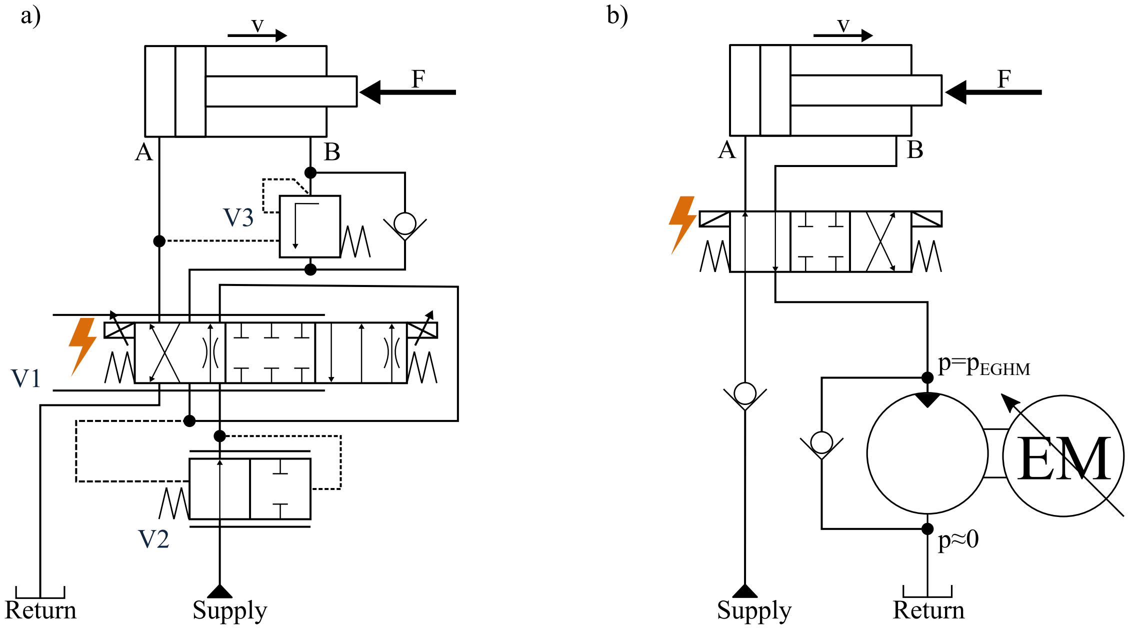

Figure 2: Schematics of two actuators during extension a) with conventional control valves b) with novel EGHM unit

Contrary to this, the novel concept controls the actuator speed by metering the return flow Q_{ret} according to the equation

$$Q_{\mathrm{ret}} = n_{\mathrm{EM}} \cdot V_{\mathrm{mot}} \cdot \frac{1}{\eta_{\mathrm{vol,mot}}} \quad , \tag{2}$$

which includes the speed of the electric machine that is working as a generator n_{EM}, and the constant displacement of the hydraulic motor V_{mot}. $\eta_{\mathrm{vol,mot}}$ is the volumetric efficiency of the hydraulic motor. It may seem that there is no need for a pressure drop across the EGHM unit because pressure is no part of Equation 2; still, it is advisable to apply a certain minimum pressure $p_{\mathrm{EGHM,min}}$ in order to achieve a minimum torque on the EM and improve the control behavior as will be discussed later. However, the pressure drop across the EGHM unit is transformed into electric energy which can be reused and is not dissipated like Δp_{ctr}. Accordingly, the novel concept can always increase the efficiency at least slightly due to the more efficient speed control mechanism.

2.1.2 Throttling for Pressure Compensation

If two or more actuators with different pressure requirements need to be supplied, the pressure difference between the actuators must be compensated by control elements in order to keep control of the system. Conventionally, this is done by a pressure compensator (V2 in Figure 2). The power that is dissipated at this valve equals the product of supply flow and load pressure difference and can be seen in Figure 3b.

For the novel concept, the EGHM unit fulfils the pressure-compensation function as well. Automatically, the EM applies a higher torque if the supply pressure is higher than necessary, and the pressure p_{EGHM} rises until the pressure forces at the cylinder are in balance with the load force. The higher torque leads to more electric energy which can be circulated back to the LS pump. Even though the circulation involves transformation losses, a significant amount of energy can be saved this way.

The efficiency improvement due to this mechanism is always present during simultaneous operation of an actuator that applies the novel system and another actuator with a higher pressure requirement. Furthermore, the improvement is most significant for large load-pressure differences and for high flows of the actuator(s) with lower load pressure, as can be seen by looking at the loss area in Figure 3b.

2.1.3 Throttling for Load Braking

Because the control valve V1 applies meter-in, not -out control, overrunning loads (negative load in Figure 2) bear the risk of cavitation and loss of control if no pressure can be build up in the return line to brake the load. To prevent this, counterbalance valves, such as shown in Figure 2a on the B side, can be used or the cross sections of the return edges of the directional valve could be reduced. As a result, the actuator can still be controlled by metering the supply flow, but the pump is still required to apply pressure even though the actuator power is negative.

With the novel concept, the whole load-braking power, which is lost in conventional systems through throttling at V3 and can be seen as an area in Figure 3c, can be recuperated by the EGHM unit instead. Again, this happens automatically and can be perceived in the form of a rising pressure p_{EGHM} and torque of the EM whenever the load turns negative. This benefit of the novel concept is especially significant for actuators that frequently need to brake loads at relatively high speeds. However, for market applications, it should be noticed that the EGHM units also have to fulfil the same safety functions as conventional braking valves, such as load holding/braking whenever the energy supply fails.

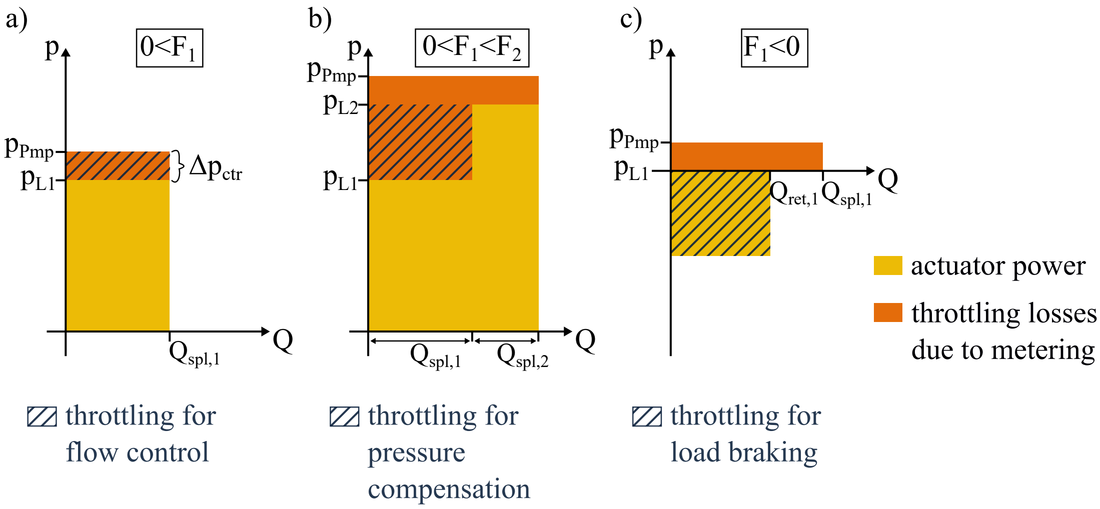

Figure 3: Pressure flow diagrams of one or two valve-controlled actuators (such as shown in Figure 2a) and shaded power-loss areas that can be reduced with the novel system a) during all operation, b) during simultaneous operation of multiple actuators, c) during load braking

2.2 Challenges for Novel Concept

The principle of metering the return flow with an EGHM unit might simplify some of the control mechanism of conventional LS systems, but it leads to new challenges as well, which are discussed here, alongside ways of addressing them.

2.2.1 Circulation Losses

As explained in the introduction, increased component costs can only be justified by an early payback time, which is also true for the novel EGHM concept. Ideally, the novel system avoids all metering losses that appear in the conventional system entirely, but in practice the recuperation and circulation of energy from electric power to hydraulic power and back to electric power involves losses as well. The hydraulic losses along lines and the directional valves cannot be neglected, and the efficiencies of hydraulic motor, EM and inverter might be high, but in sum they will lower the amount of energy that can actually be fed back significantly. Therefore, the novel concept should only be applied to actuators for which the throttling losses discussed in Section 2.1 would be especially high so that energy circulation still leads to sufficient savings. Section 4.6 will deal with the detection of actuators that are suitable for the novel system.

2.2.2 Amplified Pressures

In Section 2.1, it was already mentioned that the novel concept achieves pressure compensation as well as load braking by applying pressure in the cylinder return line. At the same time, it is possible that the full pump pressure is applied to the supply side of the cylinder. In this constellation, the hydraulic cylinder with its differential areas behaves like a pressure transformer, and high pump pressures can lead to much higher return line pressures. The transformation ratio is equal to the cylinder area ratio

$$\Phi = \frac{A_A}{A_B}, \tag{3}$$

where A_A is the piston-side area and A_B the ring area. In extreme cases, such as for the extension cylinder of the telehandler which will be investigated later, Φ can be as high as 1.9, and Figure 4a shows the pressure conditions that can appear at this cylinder. If the pump supplies a typical maximum pressure of 250 bar—because another actuator demands such a high pressure—and flow losses are neglected, the full 250 bar are applied to the piston-side of the cylinder. Without any load to counteract the pressure force, this results in a pressure of 475 bar in the return line. Even higher pressures would be obtained if a negative load is applied.

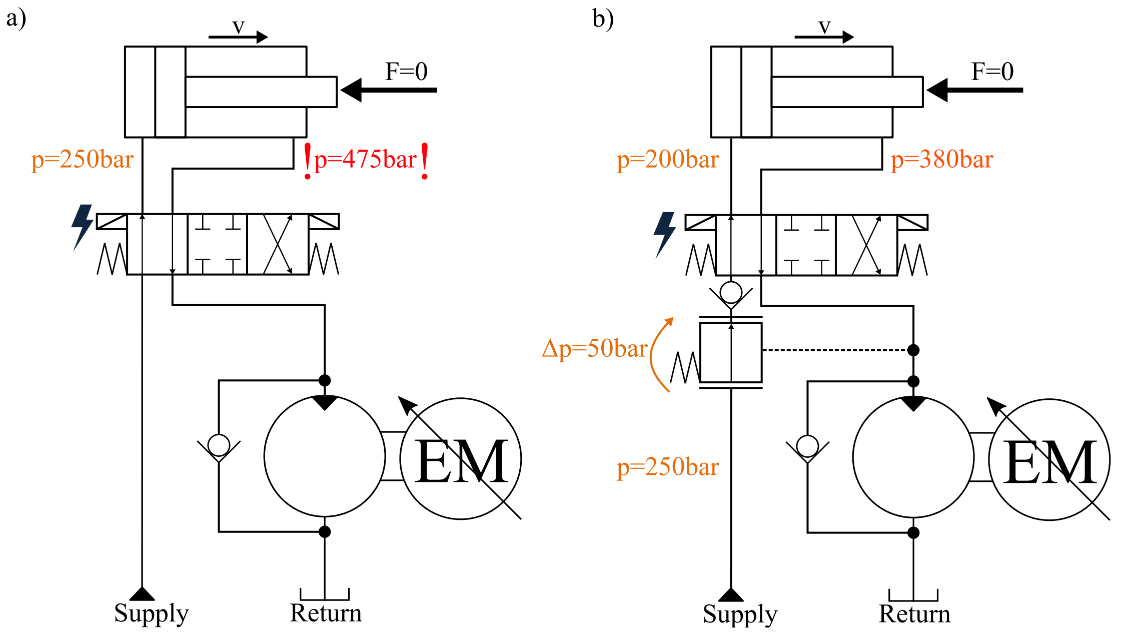

Figure 4: Pressure conditions in the novel concept at maximum pump supply pressure, no load and with a cylinder ratio of $\Phi = 1.9$; a) without pressure compensation and b) with pressure compensation. Other flow losses are neglected

Such high pressures are unacceptable as commercial mobile hydraulic components for this pressure level are not available. The highest standard mobile cylinder pressure rating that could be found by the authors was 380 bar [11], which should be the upper limit for the return line pressure. This can be achieved by adding a pressure compensating valve such as shown in Figure 4b. The valve prevents the recuperation of a part of the surplus pump power but keeps the pressure in an acceptable range at all times. However, the negative influence on the energy efficiency is expected to be less significant for cylinders with typical area ratios in the range of 1.3-1.6. Furthermore, the throttling only appears if the load pressure differences are close to the maximum and the cylinder is retracting.

2.2.3 Increased Flows

Next to high pressures, high cylinder area ratios also lead to significantly increased return line flows that need to be handled by the EGHM unit. During retraction, the return line flow is Φ-times larger than the supply flow, and, compared to the LS pump, the hydraulic motors need to be larger or spin faster in order to achieve the same retraction speeds as a convectional LS system does. However, this issue is not new and well known for other pump-controlled concepts. In [12], it was proposed to install a proportional bypass valve in order to reduce the amount of returning flow at the hydraulic unit of an EHA during retraction. Such a bypass valve could be installed in parallel to the hydraulic motor in the novel concept as well if the high return line flow cannot be handled otherwise. The downside would be a reduced efficiency during fast regenerative retraction.

2.2.4 Limited Compatibility with Conventional Valves

An important aspect of the novel concept is its compatibility with conventional LS control valves for low-cost actuation. It should be mentioned that the compatibility is limited to pre-compensated LS control valves, which can handle a varying LS margin as long as it is high enough. Post-compensated control valves would show an unstable control behavior if the margin is changing because the LS margin directly relates to the pressure drop across the metering edge Δp_{ctr}, which was discussed in Section 2.1.1. However, due to the electronic processing of the valve control signals, the behavior of post-compensated valves can also be achieved with pre-compensated valves, as will be shown in Section 3.1.

2.3 Advantages Towards Electro-Hydraulic-Actuator Concepts

After the elaborations in the previous sections it might be obvious that the novel EGHM concept can perform more efficiently than conventional valve-controlled concepts, but the last section also showed some challenges on the component side. Thus, it might seem more straight forward to choose a pump-controlled EHA concept instead of the novel concept since they usually require less hydraulic and electric machines per actuator and the issues with energy circulation and high pressures do not apply while efficiencies are high and regeneration is possible as well. However, the following part lists major benefits of novel EGHM-controlled actuators towards typical EHAs that prove the relevance of the novel concept:

- **A high minimum chamber pressure** can be achieved with the EGHM concept. EHAs typically suffer from low chamber pressures on one side or even cavitation [13]. The chamber pressures define the stiffness of the cylinder, and a low stiffness is problematic if closed-loop controls are to be applied to the cylinder, which becomes more relevant with a perceptible trend of HDMM automation [14]. Other researchers addressed this issue by adding proportional valves to the EHA [15], which introduces extra losses, or by using two electric motors per actuator [16]. Considering this, the EGHM concept with only one EM per actuator and a shared EM for the pump appears advantageous.
- **No accumulator** is needed for the novel system. Most EHAs include accumulators for differential flow compensation. The required accumulator sizes can be rather large, as discussed in [17], especially for large cylinders and high flows, which are typical for HDMMs. Thus, the EGHM concept can be used for cylinders that are too large for most EHA concepts.
- The novel concept is designed as an **open circuit**. Open circuit pumps show higher efficiencies than those for closed circuits [18], pump or motor leakages are easier to handle and oil cooling as well as filtering is more convenient, especially for a single centralized reservoir such as present in the novel concept. Open-circuit pump-controlled actuators exist as well, but issues with tilting loads and mode switching during movement were reported [13, 19], which does not apply for the EGHM concept.
- **High EM speeds** can be utilized in the novel system because the hydraulic unit, in contrast to EHAs, works solely as a motor. Medium-sized pumps usually have maximum speeds of 2,000-3,000 rpm, but hydraulic motors such as the Rexroth A2 motor [20], which is used for the simulation model in this paper, can operate at up to 5,000 rpm. This allows to select a relatively small hydraulic motor and an EM with high speed but low torque, which is more compact and potentially less expensive than an EM with higher torque and lower speed ratings.

3 Control of the Novel System

Even tough the novel concept is based on an LS system, the control is not as trivial as the control of conventional LS valves because many of the functions that were previously covered by valve mechanisms need to be covered by the control software instead.

3.1 Speed Control

The relations between cylinder flows and cylinder speeds v can be expressed for extension as

$$v_{\text{ext}} = Q_{\text{spl}} \cdot A_{\text{A}} = Q_{\text{ret}} \cdot A_{\text{B}}, \tag{4}$$

and for retraction as

$$v_{\text{ret}} = Q_{\text{spl}} \cdot A_{\text{B}} = Q_{\text{ret}} \cdot A_{\text{A}}. \tag{5}$$

Q_{spl} is the supply flow going to the cylinder, and Q_{ret} the returning flow. Furthermore, it is assumed that the speed control signal, such as a joystick signal, represents the ratio of desired supply flow for the cylinder to the maximum permissible supply flow, which is common for conventional HDMM LS systems. Moreover, the maximum supply flow for each actuator is equal to the maximum pump flow in this study. In this case, the control software must assure that the sum of required supply flows for multiple active actuators is not higher than the maximum pump flow. Otherwise, the EGHM could not control the actuator speed and cause cavitation in the return lines. Accordingly, the control applies the following rule to the operator's speed control signals y_i for each actuator i out of N actuators in order to obtain the saturated control signals $y_{\text{sat},i}$:

$$y_{\text{sat},i} = \begin{cases} y_i \cdot \frac{Q_{\text{pmp,max}}}{\sum_{i=1}^{N} y_i \cdot Q_{\text{act,max,i}}}, & \text{if } \sum_{i=1}^{N} y_i \cdot Q_{\text{act,max},i} > Q_{\text{pmp,max}} \\ y_i, & \text{otherwise} \end{cases} \tag{6}$$

The saturated control signals $y_{\text{sat},i}$ are used to control the valves of conventional actuators in the system and for calculating the required speeds of the EGHM units. Also, it should be noted that this control approach leads to the same behavior which is obtained with a conventional post-compensated system—even though pre-compensated valves are used. However, the rule could also be changed in order to prioritize certain actuator functions such as steering.

Finally, Equation 7 combines Equations 2, 3, 4 and 5 and shows how the control signal is transformed into a speed signal n_{des} for the EMs of each novel actuator:

$$n_{\text{des}} = \begin{cases} \frac{y_{\text{sat}} \cdot Q_{\text{act,max}} \cdot \eta_{\text{vol,mot}}}{V_{\text{mot}} \cdot \Phi}, & \text{if } y_{\text{sat}} > 0 \text{ (extension)} \\ \frac{y_{\text{sat}} \cdot Q_{\text{act,max}} \cdot \Phi \cdot \eta_{\text{vol,mot}}}{V_{\text{mot}}}, & \text{if } y_{\text{sat}} < 0 \text{ (retraction)} \end{cases} \tag{7}$$

In addition, the directional valves must be switched into extension or retraction position as soon as movement is required. Moreover, for the control of the electric generators, positive torques are permitted because otherwise they would cause cavitation when the cylinder reaches its end stops and no more flow can be received from the cylinder.

3.2 Pump Pressure Control

Because the novel concept is more complex than a conventional LS system, also the calculation of the required pump pressure has to be adjusted. Three objectives have to be considered:

- The pump pressure should only be as high as necessary in order to save energy.
- The conventional actuators should receive enough pressure to move the load and to apply the pressure drop Δp_{ctr} (Equation 1) at the control valve.
- The novel actuators should receive enough pressure to build up at least the required minimum pressure in front of the EGHM unit (without pressure, the EM cannot apply a torque and control the speed).

Figure 5 shows the signal flow diagram of the control for a system with one novel actuator (index 1) and one conventional actuator (index 2), which respects all three objectives that are mentioned above.

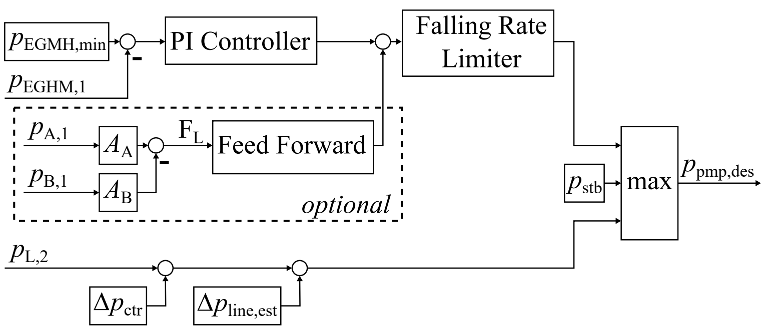

Figure 5: Signal flow for the calculation of the required pump pressure.

The pressure in front of the EGHM unit is detected by a pressure sensor and used as a feedback to control it in order to reach the minimum required pressure $p_{\text{EGMH,min}}$. The relation between the pump pressure, which is adjusted by the controller, and the pressure in front of the unit is highly nonlinear and depends on the direction of movement and amount of flow. Therefore, the parameter tuning of the controller can proof difficult, especially if the attached mechanical system has a low damping ratio. The performance can be improved by adding a force feedback signal, which is done in the following simulation part, but this would also require two additional sensors for a real application. Simulations also showed that oscillations of the mechanical system that cannot be compensated fast enough by the controller lead to pressure spikes close to zero in front of the unit. This may lead to cavitation and definitely to discontinuous movement since the EM cannot apply torque and control the speed. To prevent pressures close to zero, $p_{\mathrm{EGMH,min}}$ should be chosen high enough as a buffer, and limiting the falling rate of the pump pressure signal can prevent low pressure peaks caused by the controller as well. On the downside, the rate limiter leads to a higher average pump pressure and potentially to higher losses which will be shown in Section 4.6.

In contrary to this, for the conventional actuator, it is sufficient to sum up the load pressure, which is sensed by a sensor at the LS port of the valve; the pressure margin Δp_{ctr}, which is required at the control edge; and a margin $\Delta p_{\mathrm{line,est}}$ that is added to account for flow losses between the pump and the control valve.

Finally, the highest required pump pressure among all actuators defines which pressure is send to the pressure controller of the pump. This is legitimate since all actuators can handle higher but not lower pressures. However, it should be noted that those switching as well as saturation effects make anti-windup measures in the PI-controllers indispensable. For the case that no actuator is active, also a standby pressure p_{stb} is considered, which assures that all supply lines remain pressurized for fast response.

4 Simulation Study

After explaining the concept, its characteristics, and the way of controlling it, this section deals with the simulation study that was done in order to show how much energy-efficiency improvement is feasible with the novel concept compared to a conventional LS concept. Furthermore, it will be analysed which actuators are most suitable to be controlled by the novel concept and which should remain valve-controlled. This is done by looking at the specific example of a telehandler. The different parts of the simulation model will be explained before the results are presented.

4.1 Telehandler

A 9-tonne telehandler is chosen as the platform for the simulation study because it represents a typical HDMM and its implements comprise hydraulic cylinders of significantly different sizes and functions. For this study, only the linear actuators for boom, extension and tilt are considered, which are depicted in Figure 6. However, additional actuators such as steering cylinders, stabilizers or attachments could also be connected to the LS pump of the novel system, but it seems unlikely that controlling those with the novel EGHM unit could be economical. The telehandler is typically equipped with forks or a bucket for load handling. Furthermore, it should be noticed that the reference telehandler includes compensation cylinders which can be seen in Figure 6 as well. Whenever the boom cylinders are actuated, the compensation cylinders are moving too, and their hydraulic connection to the tilt cylinders results in an almost constant fork or bucket orientation during boom lifting and lowering, even though the tilt is not actively actuated.

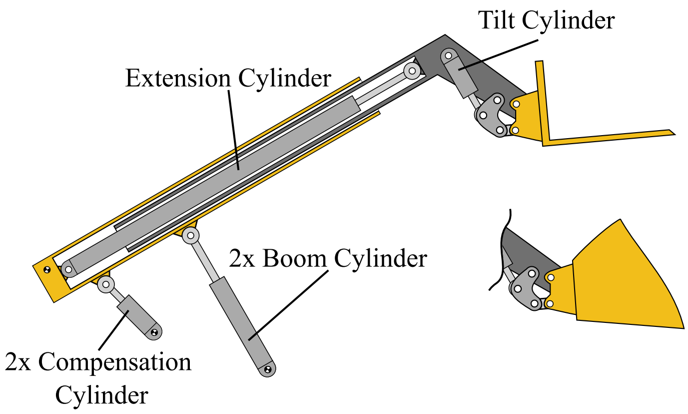

Figure 6: Mechanical parts and hydraulic cylinders of the modelled telehandler implements

The purpose of modelling the implements of the telehandler is to simulate a realistic load for the cylinders in the model of the novel concept. In order to do so, a planar mechanical model was created in the Simcenter Amesim environment, where it can be directly connected to the hydraulic model. Geometry data was obtained by measurements of the reference telehandler, and inertia data was retrieved through reverse engineering of the components with a CAD software. The exact piston and rod diameters of the cylinders are known. Gravity is included in the model, and friction was modelled for the cylinders with a viscous speed-proportional component and a constant coulomb component. The friction parameters were adopted from the model of a similar machine and slightly adjusted according to the different cylinder sizes. The fact that the model was not validated against measurements of a real version of the telehandler is assumed to be legitimate as only simulation results will be compared to each other that were obtained with the same model.

4.2 Electro Hydraulics

The whole hydraulic system is modelled in Amesim as well, using the basic libraries. The EMs for the pump and EGHM units are modelled as perfect torque sources with PI-controllers for pressure or speed control, respectively. The controllers include maximum torque and speed limitations, anti-windup functions and feedforward terms of the required flow for the pressure controller or feedforward terms of the load pressure in case of the speed controller. Parameters were obtained from the data sheets of suitable permanent magnet synchronous motors. A further description of the controllers is beyond the scope of this paper.

Simulations are done in 5 different setups that are characterized by the choice of actuators that are modelled with the novel concept or remain valve-controlled:

- Setup 0: All actuators conventional (benchmark)
- Setup 1: EGHM units for all actuators
- Setup 2: EGHM unit for boom only
- Setup 3: EGHM units for boom and tilt
- Setup 4: EGHM units for boom and extension

Comparing duty-cycle efficiencies between the different setups will allow to assess which actuator has a high potential for efficiency improvement with the novel system and which one does not.

The modelled schematics of the novel concept correspond to Figure 4b and the schematics of the conventional actuators to Figure 2a, with the exception that the counterbalance valves are installed on both cylinder sides for the boom cylinders and on the piston side only for tilt and extension. Relief valves for the pump and all cylinder chambers were added as well; otherwise, the overall system design corresponds to Figure 1, but energy recuperation is only considered in the post processing.

The sizing of the hydraulics was done in order to meet the same performance as the original reference telehandler with a maximum pump flow capability of $150\,\frac{\mathrm{l}}{\mathrm{min}}$. Each actuator is able to receive the entire maximum pump flow. The chosen displacements and related maximum speed limits of the hydraulic pump [21] and motors [20] can be seen in Table 1 and 2. The motors in the novel concept for boom and extension need to be larger than the tilt motor due to the higher Φ, as explained in Section 2.2.3. At the same time, this results in a lower maximum pressure (set by the compensating valve) for the EGHM units of boom and extension because the torque on the EM would be too high otherwise. The mechanical and hydraulic efficiencies of all hydraulic machines are modelled by implementing lookup tables that include reference data from measurements.

Because dynamic performance is of less importance in this study, the mechanical modelling of the valves has been omitted and mathematical equations without higher order are used to control the opening of single hydraulic edges, of which one or multiple form the pressure compensating or the directional valves. For the conventional valve-controlled actuator, the pressure drop characteristics from the data sheet of a Rexroth SB34 valve block [22] are imitated. The characteristics of the non-metering edges in the same LS valve block have been adopted for the maximum opening of the directional valve and the pressure compensating valve in the novel concept as well.

4.3 Control

The control of the electro-hydraulic system was implemented in Matlab Simulink according to the equations and signal flow diagram in Section 3. The key control parameters can be seen in Table 2. The control blocks for each actuator are chosen for each setup depending on the concept—novel or valve-controlled. The conversation between the hardware model in Amesim and the control in Simulink is achieved through discrete co-simulation with a sample time of 1 ms.

4.4 Duty Cycles

The efficiency performance of the 5 different setups should be compared for a realistic duty cycle in order to achieve meaningful results. Furthermore, simulating different cycles with varying times of actuation for each

Table 1: Key Model Parameters

Parameter	Value	Unit
$p_{\text{EGHM,max,boom/exte}}$	320	bar
$p_{\text{EGHM,max,tilt}}$	380	bar
Δp_{ctr}	20	bar
$V_{\text{mot,boom/exte}}$	56.6	cm^3
$V_{\text{mot,tilt}}$	44.9	cm^3
V_{pmp}	63	cm^3
Φ_{boom}	1.6	-
Φ_{tilt}	1.3	-
Φ_{exte}	1.9	-

Table 2: Key Control Parameters

Parameter	Value	Unit
$n_{\text{pmp,max}}$	2600	rpm
$n_{\text{mot,tilt,max}}$	5000	rpm
$n_{\text{mot,boom/ext,max}}$	5000	rpm
$p_{\text{EGHM,min}}$	20	bar
$p_{\text{pmp,max}}$	250	bar
$\Delta p_{\text{line,est}}$	10	bar

actuator and times of simultaneous operation as well as varying amounts of load braking will tell if the novel concept can perform better or worse under specific conditions.

The desired cycle movements are traced by a state-flow controller in Simulink, which separates each cycle into steps. For each step, movement commands are given to specific actuators until a desired position is reached and the next step starts. At the same time, the controller can increase or decrease the load mass in the telehandler model. All setups receive the same duty cycle tracing commands and can be compared to each other. The following cycles are implemented:

4.4.1 Loading with Bucket (Cycle A)

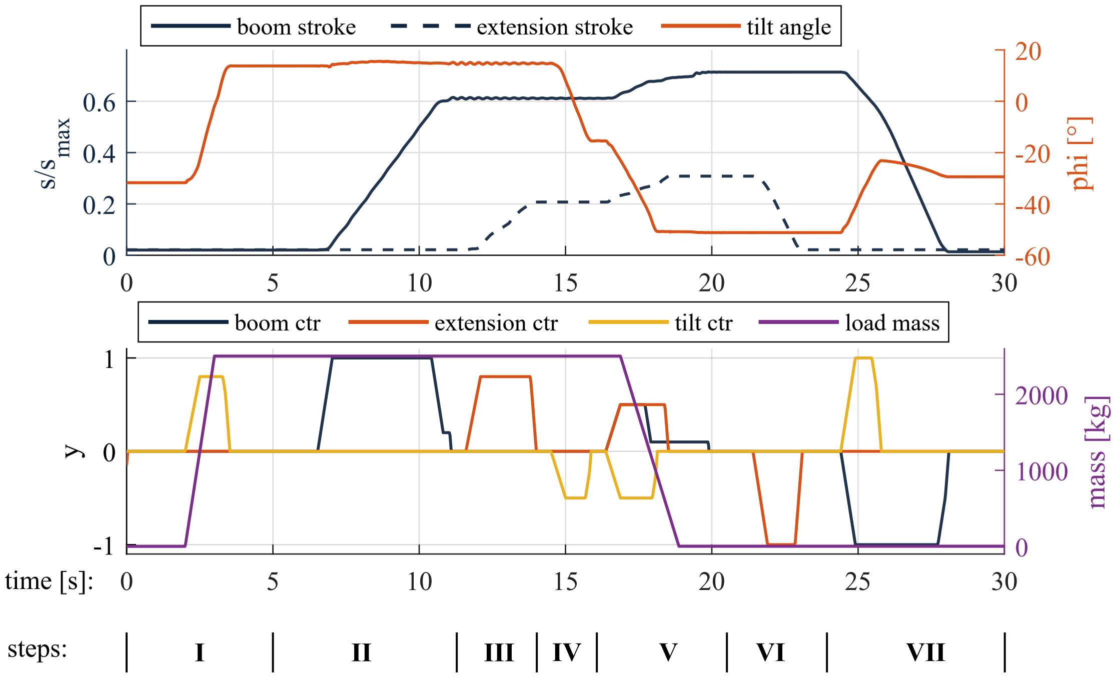

Figure 7: Load mass, control signals and resulting movement for duty cycle A

A typical telehandler task is loading a truck or a container with a relatively high opening by dumping material into the truck or container from above. Buckets with capacities of up to 4,000 l are used to handle loads such as earth or grain that are picked up from a pile. For the simulated cycle, a full bucket load of 2,500 kg is assumed, which can still be handled at low to medium extension length without the risk of tilting the whole telehandler. Figure 7 shows the command signals and resulting movements from the simulation of Setup 1. The smooth curves with only little oscillation proof the functionality of the novel concept, but further analyses of the dynamics are beyond the scope of this investigation. The single steps can be described as following:

I: Tilting the bucked up after the telehandler drove into the pile (load is applied) (→driving to the truck/container) →II: Lifting boom →III: Extending →IV: Starting to tilt bucket →V: Dumping load by tilting, extending and lifting simultaneously (load is removed) →VI: Retracting →VII: Return to load, by lowering boom and tilting bucket back to initial position

4.4.2 High Stacking of Pallets (Cycle B)

For another cycle, positioning a pallet at a high position, such as in a warehouse, is chosen as a task with more actuation time for the extension cylinder. Two versions of the task are simulated: Cycle B1 simulates the lifting of a pallet and empty lowering of the boom back to ground, and cycle B2 simulates empty lifting and lowering of the pallet, which involves more load braking. For the load mass, 1,500 kg, the maximum weight of a EUR-pallet, is chosen. Again, control signals and modelled movement of Setup 1 are depicted in Figure 8. The steps can be described as following:

I: Lift boom until fork is at eye level →II: Tilt forks until horizontally →III: Lift boom and extend simultaneously until pallet is above drop position →IV: Lower boom and drop pallet (weight decreases) →V: Retract while slightly lifting boom to get forks out of pallet →VI: Get back to drive position by lowering boom, retracting and tilting forks back

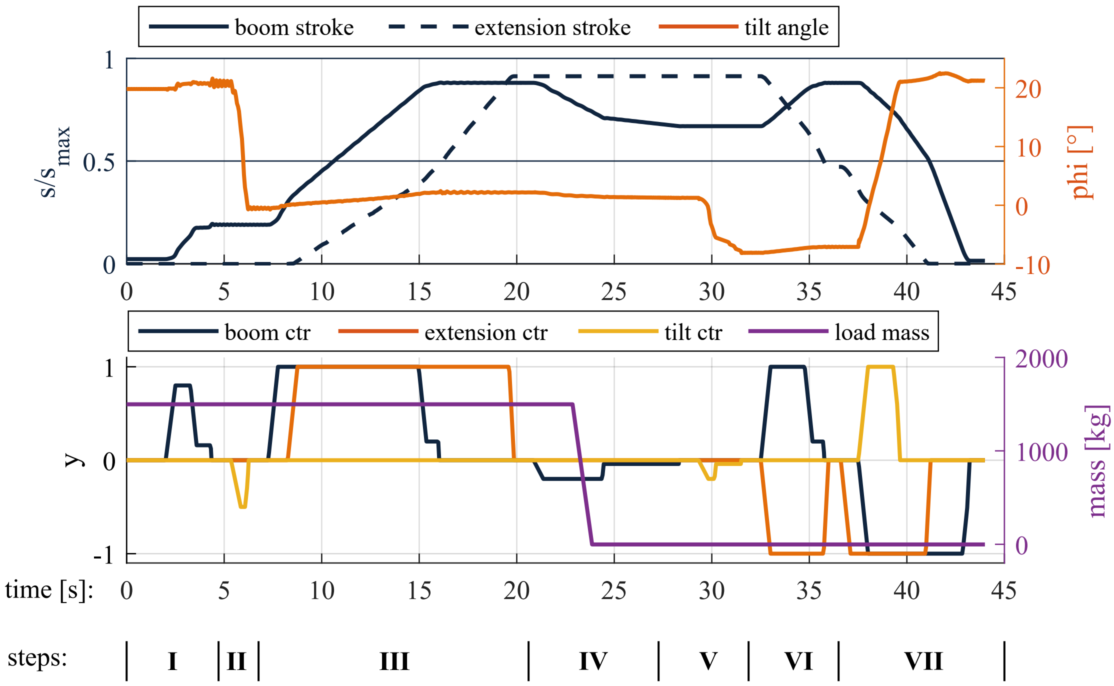

Figure 8: Load mass, control signals and resulting movement for duty cycle B1

Cycle B2 differs only slightly from the reverse cycle B1 in the way the pallet is picked up or dropped. Therefore, cycle B2 is not further described here.

4.5 Energy Analysis

In order to analyse the overall efficiency of the 5 different setups, the mechanical speed and torque of the pump and the the hydraulic motors in the novel concept are recorded and transformed into electric power input and output in the post processing. The transformation involves an assumed inverter efficiency of $\eta_{\text{inv}} = 98\%$. The value is assumed constant and adopted from [23], even though the inverter in this publication would be too small, but no specific parameters of larger inverters could be found. The efficiencies of the EMs (η_{EM}) are considered by lookup tables that were obtained from lumped parameter simulations of an permanent magnet synchronous machine. The parameters for the simulation were obtained from data sheets of EMs with matching torque and speed capability. Accordingly, the electric input power can be calculated as

$$P_{\text{el,in}} = 2\pi \cdot n_{\text{pmp}} \cdot T_{\text{pmp}} \cdot \frac{1}{\eta_{\text{EM}(n,T)} \cdot \eta_{\text{inv}}}, \tag{8}$$

and the electric output power of all N EGHM units as

$$P_{\text{el,out}} = \sum_{i=1}^{N} 2\pi \cdot n_{\text{mot},i} \cdot T_{\text{mot,i}} \cdot \eta_{\text{EM}(n,T)} \cdot \eta_{\text{inv}}. \tag{9}$$

Finally, the required electric energy for the whole cycle can be calculated in the following way:

$$E_{\text{el,in}} = \int_0^{t_{\text{end}}} P_{\text{el,in}} - P_{\text{el,out}}\, dt \tag{10}$$

The fact that it might be necessary to store some parts of the recuperated energy temporarily in an electric storage whenever $P_{\text{el,out}}$ is higher than $P_{\text{el,in}}$ is neglected here. This is justified by the relatively high charge and discharge efficiencies of modern batteries and supercapacitors.

4.6 Results

The results of the simulations and post-processing are the amounts of electric input energy $E_{\text{el,in}}$ that are required for each setup to complete the different duty cycles—in other words, the energy consumption. Those values can be seen in Figure 9 alongside a visualization of the amounts of energy that are circulated and recuperated with the novel EGHM units. The energy amounts are normalized with reference to the required energy for Setup 0 in each cycle, which was 473.6 kJ for Cycle A, 744.7 kJ for Cycle B1 and 533.2 kJ for Cycle B2.

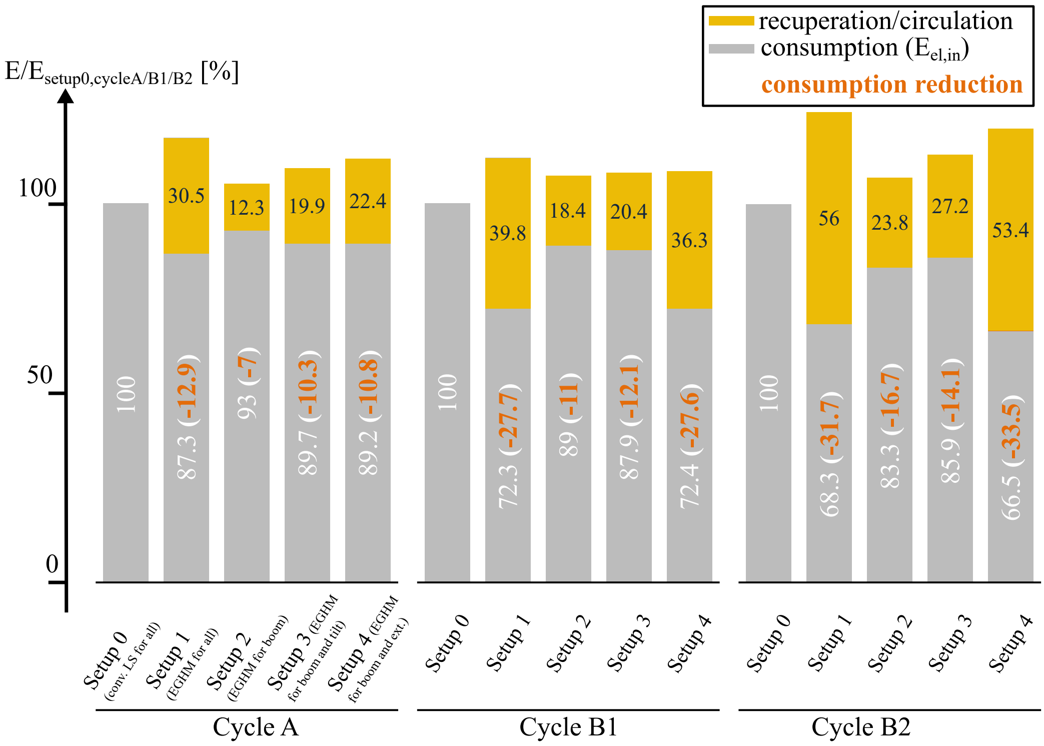

Figure 9: Electric energy consumption for each setup and cycle as well as energy that is circulated and recuperated by the novel EGHM units (the reference energy for each cycle is the energy required by the conventional Setup 0)

Figure 9 shows that applying the novel EGHM units to all three actuators can lead to significant reductions of the required energy. For cycle A, 12.9% are rather small, while 27.7% for B1 are remarkable and 31.7% for B2 even higher. The differences are most likely caused by the fact that the cycles B1 and B2 involve higher lifting and thus more potential for load energy recuperation than Cycle A. Because B2 involves lowering of the loaded forks, the potential of this cycle and thus the energy reduction is the highest. The influence of simultaneous operation, which was discussed in Section 2.1.2, can hardly be judged here since all cycles contain similar amounts of serial or simultaneous operation of the different actuators.

Furthermore, by comparing the energy bars of the different setups within one duty cycle, more statements can be made. Utilizing the novel EGHM units for all actuators (Setup 1) seems to yield the highest improvements, except for Cycle B2 which favors Setup 4. For this cycle, controlling the tilt cylinder with conventional valves is actually more efficient than using the EGHM unit. The reason might be the increased pump pressure caused by the EGHM pressure controller which is further discussed at the end of this section. Furthermore, the tilt is only moving a little in the cycles B 1 and B 2; thus, there is not much potential for load energy recuperation, which is different for Cycle A. For this cycle, the improvement due to the novel boom actuator is most significant (see Setup 2) but a novel extension actuator (difference between Setup 4 and Setup 2) as well as a novel tilt actuator (difference between Setup 3 and Setup 2) contribute a decrease of energy of around 3% as well. For cycles B 1 and B 2, on the other hand, the contributions of a novel boom actuator and a novel extension actuator are almost the same and both between 11-16.8%. Thus, it seems reasonable to apply the novel EGHM units in any case for the boom cylinders and also for the extension cylinder if B 1/B 2-like duty cycles are common. However, the tilt actuator should remain valve-controlled. This reduces component costs but still yields a significantly improved energy efficiency.

Moreover it can be noticed, that the stacks of the grey and yellow bars in Figure 9 are higher for the Setups 1-4 than for Setup 0. The stack of the grey $E_{\mathrm{el,in}}$ bar and the yellow recuperation/circulation bar corresponds to the energy which is transmitted by the EM of the pump during the cycle. It is increased by the novel EGHM units because their pressure controllers (depicted in Figure 5) include a pressure falling rate limiter, which increases the average pressure and thus power as well as energy of the pump. Since the increased pressure leads to more circulation losses for the EGHM units and more throttling losses for conventional valves in the system, this effect is assumed to have a significant impact on the overall efficiency of the novel systems. If the pressure controllers for the EGHM units could be improved and lead to lower pump pressures, this would earn the novel concept a further advantage towards conventional systems.

5 Conclusion and Outlook

This paper proposed a novel more efficient concept for the control of actuators in LS systems by means of units that combine an electric generator with a hydraulic motor. An analyses of the throttling losses of conventional control valves could show how the novel concept can avoid these and under which conditions. Using this knowledge, it can be estimated how significant the potential of efficiency improvement for a certain actuator on an HDMM is. Furthermore, simulations of three different characteristic implement work cycles of a telehandler could quantify the amount of energy that can be saved with the novel concept compared to conventional purely valve-controlled concepts.

The highest reduction of 34% could be achieved with a combination of valve-controlled tilt actuator and EGHM units for boom and extension. This shows how a significant efficiency improvement can be achieved by applying electro-hydraulic elements to single actuators with high energy turnover while the system remains compatible for conventional valve-controlled actuators.

In next steps, the concept could be applied to other common HDMMs and simulated as well in order to see if the results differ from the observations of the telehandler study in this paper. Moreover, the pressure control for the EGHM unit could be further improved and impacts on the efficiency as well as machine dynamic and control stability could be analysed. A step further, experiments with real HDMMs could validate the simulation results and confirm the relevance of the novel concept. Simultaneously, the additional component costs of a system with EGHMs compared to a conventional LS system could be estimated as well as the approximate amortisation time.

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Acronyms

EGHM	electric-generator-hydraulic-motor
EHA	electro-hydraulic actuator
EM	electric machine
HDMM	heavy-duty mobile machine
LS	load sensing

Nomenclature

Designation	Denotation	Unit
E	energy	Ws
F	force	N
n	rotational speed	rpm
N	number of actuators in system	-
p	pressure	bar
P	power	W
Q	flow	m^3/s
s	cylinder displacement	m
T	torque	Nm
v	linear speed	m/s
V	displacement of hydraulic machine	ccm
y	control ratio	-
Φ	cylinder area ratio	-
η	efficiency ratio	-

Index	Denotation
A	at piston side of cylinder
act	for an actuator
B	at rod side of cylinder
boom	for boom actuator
ctr	for control
des	desired value
EGMH	for EGHM unit
el	electric
EM	for electric machine
est	estimated value
ext	for extension
exte	for extension actuator
i	for counting of actuators
inv	for electric inverter
L	related to load
line	for hydraulic line
max	maximum value
min	minimum value
mot	for hydraulic motor
pmp	for hydraulic pump
req	required value
ret	at return line
rtr	for retraction
sat	satturated value
spl	at supply line
stb	standby
tilt	for tilt actuator
vlv	valve specific
vol	volumetric