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Utilization of electric prime movers in hydraulic heavy-duty-mobile-machine implement systems

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ABSTRACT

Electrifying vehicles yields advantages such as reduced emissions, better performance and more flexibility. While electric machines can directly drive the wheels or tracks of heavy-duty mobile machines (HDMMs), the implements require a combination of electrics and hydraulics for robust, high-force linear actuation. This survey focuses on different of such electro-hydraulic implement systems that have been proposed by industry and academia over the last decades. For the hydraulic circuits, centralized valve-controlled architectures are identified as less progressive but easy to implement for a fast market penetration, while novel decentralized circuit concepts can be more efficient but also more challenging for HDMMs compared to stationary or aircraft applications. The electric machine (EM)-pump combinations are mostly formed out of standard components so far, while customized, integrated or even linear-pump concepts offer room for improvement. Different forms of non-stationary electric energy supplies were also found to be numerous, but many technologies require more development.

1. Introduction

Hydraulics is the state-of-the-art technology to transmit and control power on heavy-duty mobile machines (HDMMs), which are widely used in sectors such as construction, agriculture, forestry or mining. Conventional hydraulic systems for HDMMs are mainly valve-controlled and suffer, as a matter of principle, from high throttling losses. A study from 2012 showed that the average energy efficiency of mobile hydraulics in US industries is only as high as 21% [1]. Facing rising energy prices and tightening emission restrictions, the low efficiencies of conventional hydraulics have become a problem but also incentive for industry as well as academia to replace hydraulic systems or to improve them.

Two main functions of an HDMM need to be considered for this—-driving and fulfilling work tasks with their implements. If driving is seen as the pure propulsion without steering, it can be said, no matter if wheeled or tracked, that it involves only rotary actuation. Accordingly, hydraulics—with its simple, continuously variable transmission capability—is a well suited standard concept for driving but not irreplaceable, and fully electric or electro-mechanical alternatives are emerging due to higher efficiencies.

On the other hand, the implement functions typically require mostly linear actuation. In this case, hydraulics is demonstrating its unique qualities which make it indispensable for HDMM implements. Firstly, hydraulic cylinders offer a low-cost solution for linear actuation due to their simple design, and they can easily be actuated in parallel. Furthermore, hydraulics is known for high power and force density, which allows compact designs and handling of heavy loads typical for HDMMs. Natural damping, power-less load holding, robustness, reliability and easy overload protection via relief valves are additional properties that make hydraulics superior to other technologies [2,3].

However, nonhydraulic linear actuator concepts have also been tested for HDMM applications, but the field of feasible application has remained limited. In [4], Maurus et al. replaced the hydraulic actuator of a road-sweeper with a linear electric machine (EM). The actuator efficiency, controllability and dynamics could be improved; still, issues like significantly increased size, overheating and high material costs demonstrated that even for such low-power applications hydraulic cylinders remain superior to linear EMs. On the other hand, electro-mechanical actuators, which utilize rotational EMs in combination with linear screw mechanisms, can be a more efficient and easy-to-control alternative to hydraulic actuators according to Hagen et al. [5], but only if the continuous transmitted power stays below 2 kW, which is likely to be too low for many HDMMs. However, in [6], the prototype of a 6 tonne excavator with an electro-mechanical actuator for the boom movement was presented, but

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the actuator requires the support of a hydraulic spring system to reduce the peak forces and power. Furthermore, the electro-mechanical actuator did not achieve the same stroke range as the original hydraulic actuator due to less compactness. Next to more space requirement, a limited robustness and tolerance of impacts as well as the high price make electro-mechanical actuators unsuitable for most HDMM applications [5].

Nevertheless, even with hydraulics remaining essential as the final step in power transmission for HDMM implements, electrification can help to improve the efficiencies of implement power trains significantly by providing prime movers for hydraulic pumps in the form of EMs. Hereby, hydraulic systems with electric prime movers are defined as *electro-hydraulic* systems and will be surveyed. This definition does not include systems that use no EMs in the actual power transmission but only electronically controlled valves.

Only a few other surveys or reviews could be found that already partially cover areas of this field. In [7], for example, Ketelsen et al. have reviewed pump-controlled cylinder drives with a focus on electro-hydraulic actuators (EHAs), which represent one option for the electrification of hydraulic implements, but the review discusses them for general applications, without explicit references to the special requirements of HDMMs. Quan et al. surveyed technologies for the energy-efficiency improvement of HDMMs in general, which also included electro-hydraulic implements [8]; however, the survey is focused on earth moving machines only, and no in-depth review of aspects besides energy-efficiency was presented. Inderelst et al. have conducted an SWOT-Analysis (Srength, Weaknesses, Opportunities and Threats) on electro-hydraulic drives in construction machinery in [9], but the review part is of limited comprehensiveness since only pump-controlled concepts are studied and the focus is on the concrete application case of a crawler excavator. Similarly, Opgenoorth et al. have presented advantages and challenges of the utilization of EMs in mobile machinery, but conducted only a limited review of technology options and rather focus on the example of a specific electro-hydraulic system for an excavator [10].

In contrast to this, the authors aim at presenting a more comprehensive survey in this paper that covers component aspects as well as actuator subsystems and whole architectures of electro-hydraulic HDMM implement systems including also the way of supplying electric energy. This way, an assessment of the potential of certain electro-hydraulic solutions in the context of the special requirements of different typical HDMMs and the interaction with other system components is provided. For different perspectives, a large number of academic publications as well as major trends in industry are reviewed. To the authors' knowledge, this is the first time electrified hydraulics are reviewed on this comprehensive level with reference to typical HDMM implement applications.

The survey is thematically subdivided into sections and covers all in all the whole electro-hydraulic implement system of an HDMM, as can be seen in Fig. 1. Section 2 is dealing with hydraulic circuit architectures which can be electrified but do not depend on EMs as prime movers, while Section 3 is analyzing novel EHA circuit concepts that are based on the control capabilities of EMs. Furthermore, Section 4 discusses EM-pump combinations which form the interface between the hydraulic and electric domain. The latter is surveyed in Section 5 with its different alternatives of supplying electric energy for the EMs. After the actual survey, Section 6 concludes the findings.

2. Valve-controlled circuits

The simplest way of electrifying an HDMM is to use an existing hydraulic architecture which is based on an internal combustion engine (ICE) as the prime mover, and to replace the ICE with an EM, as Gottberg et al. did for example [11]. For hybrid solutions, the EM can also be added to the ICE shaft as it was shown by Sakamoto et al. [12]. However, both options do not allow to make use of the advantages of EMs in terms of controllability, as the EM only has to rotate at semi-constant or step-wise controlled speeds like an ICE does. On the positive side, well-established hydraulic circuits with known and reliable behavior can still be used.

2.1. Conventional systems

Most mature are the conventional valve-controlled systems that are nowadays used on HDMMs. For electrification, load sensing (LS) systems are favored among those, as they represent one of the most efficient forms of valve-controlled systems. With LS systems, multiple actuators can be supplied by the same pump. The pump senses the maximum load pressure of all actuators and adjusts the supply pressure to be only as high as necessary for moving even the actuator with the highest pressure requirement. This concept avoids energy losses due to over-pressurization, but the flow distribution and speed control of the actuators is achieved by means of valves. These lead to high metering losses in the form of throttling and do not allow to recuperate energy from load braking, which increases the energy demand and is—considering electrified HDMMs—especially critical in terms of battery capacities and limited operation time,

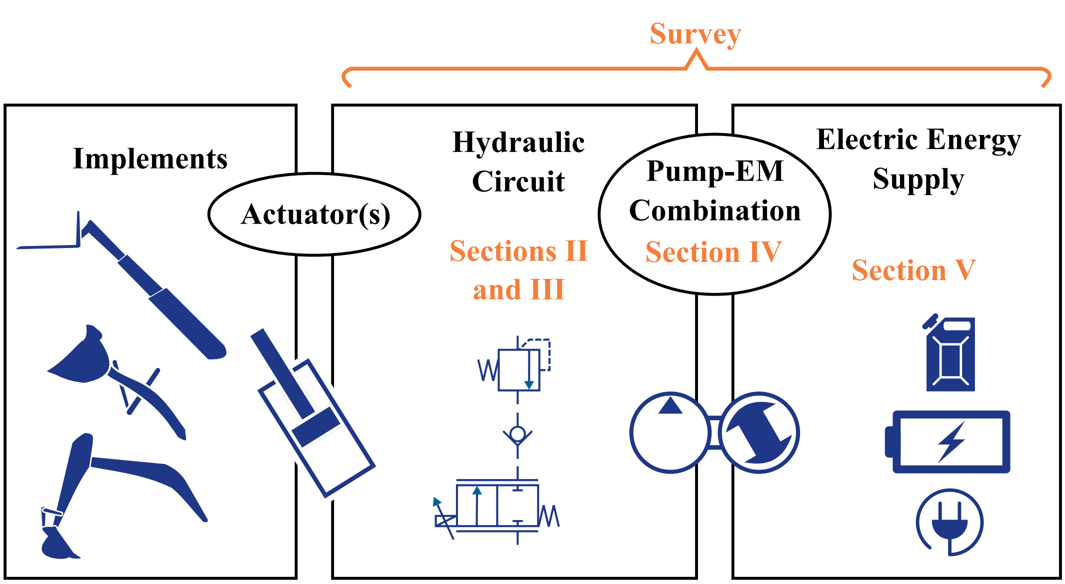


Fig. 1. The electro-hydraulic system of an HDMM, demonstrating the structure of the survey.

respectively. A detailed explanation of conventional LS systems is given by Padovani et al. in [13].

At present, machines that simply replaced the prime mover or rotational actuators with EMs are the only electrified HDMMs that have entered the market so far [14]. It seems that manufacturers are reluctant concerning the change of whole systems as it requires more design effort and new safety certifications. Still, even with conventional hydraulic systems, electrified HDMMs already offer lower noise levels during operation, zero local emission and more operator comfort due to less vibration and noise. Commercial examples of HDMMs with conventional hydraulics in combination with EMs can be found in the following:

- JCB launched its first fully electric mini excavator, the 19C-1E [15], which still utilizes one centralized pump driven by an EM that can run at three different speeds. Compared to the ICE version of the excavator, the EM offers the feature to quickly and automatically switch between operational and very low speed of the pump, which is used to reduce idling losses.
- Volvo Construction Equipment launched two fully electric construction machines. The wheel loader, L25 Electric [16], uses separate EMs for driving and implements, which enables a more independent and efficient power distribution, but the implement motion is still controlled by valves. Also the mini excavator, ECR25 Electric [17], utilizes a conventional LS system with variable-displacement pump that can be operated at three different EM speed steps.
- Due to the utilized cable connection to a stationary electric power supply, Liebherr can also offer fully electric machines with higher power. The 200 t hydraulic mining excavator, R 9200 E [18], utilizes a single EM, which leads to the conclusion that the conventional centralized valve-controlled system is still used. Moreover, the 27 t material handler, LH 26 Litronic [19], has only one EM as well and the same hydraulic system specifications as the diesel-powered version.
- Suncar, a company developing electrified versions of existing diesel-powered excavator designs, also states that they are currently only replacing the ICEs with EMs while keeping the original hydraulic systems [20].

Still, the dynamic control capability of EMs can technically also be utilized in conventional LS valve-controlled systems. If they are equipped with fixed-displacement pumps, as it is done in [21] or [22] for example and shown in Fig. 2b, the pump pressure p_{Pmp} can be controlled according to the sensed load pressure p_{LS} by adjusting the pump speed or torque. Conventionally, the LS control is accomplished by changing the pump displacement (shown in Fig. 2a), which requires more expensive and complex variable pumps, or by using an LS pressure valve, which leads to even higher throttling losses.

2.2. Novel, more efficient valve-controlled systems

The conventional valve-controlled systems mentioned in the previous section still belong to the group of systems that achieves only the already mentioned average efficiency of 21% [1]. Accordingly, novel systems have been designed to improve the efficiency of hydraulic systems powered by ICEs, but EMs could be used as well:

- *Independent metering* utilizes multiple simple valves to fulfill the function of a single complex multi-way valve. This enables more freedom in control and to reduce metering losses to a certain extend. Abuowda et al. have presented a detailed review in [23].
- *Digital hydraulics* uses components with discrete states (valves, pumps, motors or cylinders) to achieve semi-continuously variable behavior for the control of hydraulic actuators. This comprises two approaches: Multiple similar components can be combined to a super component of which different behaviours can be achieved by separately engaging or disengaging the subcomponents; the higher the

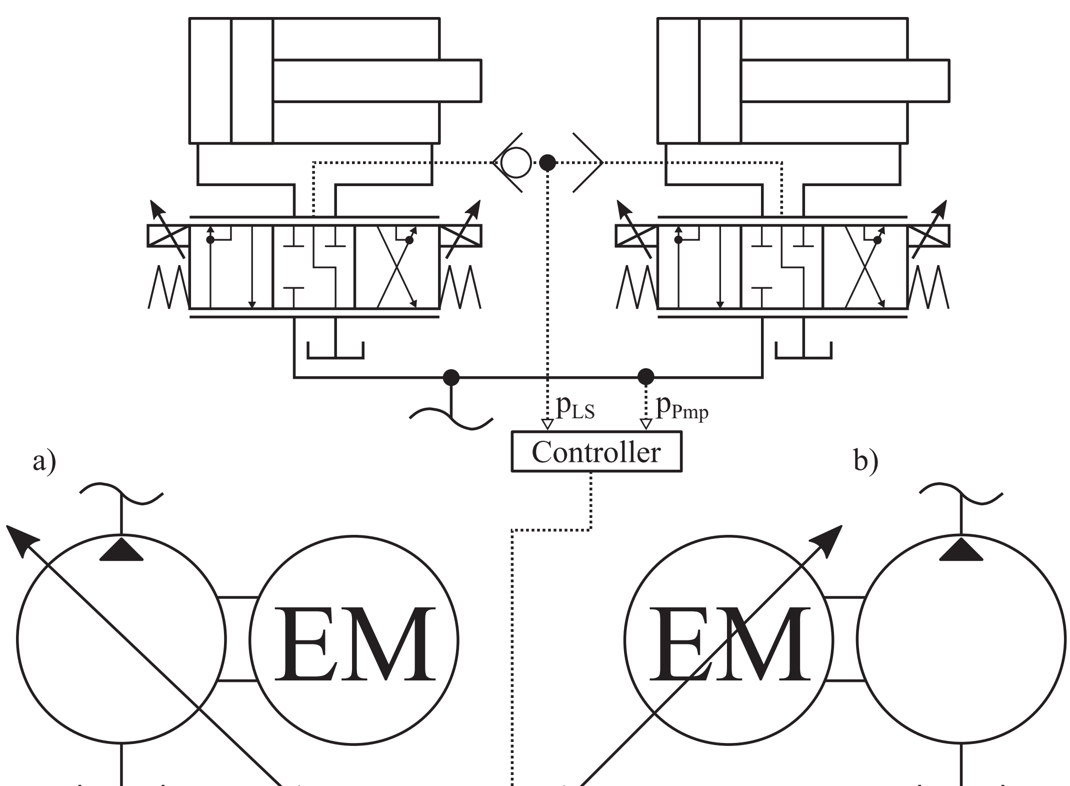


Fig. 2. Two exemplary versions of an electrified valve-control LS system a) conventional with fixed speed and variable pump b) novel with controlled EM and fixed-displacement pump.

number of subcomponents, the closer is the behavior to a continuously variable system. The other approach uses a single component and pulse-width modulation to activate and deactivate it at a high frequency, which results in a quasi-variable component. A review of digital hydraulics carried out by Linjama can be found in [24]. All forms of metering losses in conventional valve-controlled systems can be avoided by digital concepts, but compression losses and component losses can increase, and suitable valves for high power levels are still missing.
- The *hydraulic transformer* in its common form is not a valve-controlled concept, but listed here because it also aims at improving energy efficiency while not being dependent on a specific type of prime mover. The principle is similar to an electric transformer. Instead of using valves to throttle down excess supply pressure, hydraulic transformers can convert a high supply pressure down to the pressure which is required by the actuator without throttling and while increasing the flow or vice versa. Concepts for HDMMs were presented by Vael et al. in [25,26]. The remaining losses are line losses and parasitic component losses, which could be significantly reduced for the so-called floating cup transformer design by Innas [25]. For speed or position control of the actuators, additional speed sensors are required for feedback; otherwise the lack of commercially available, efficient transformers is the only major issue with this concept.
- Conventional systems throttle energy from braking instead of recuperating it while *hydraulic hybrid systems* include hydraulic accumulators to store and feed back braking energy. Hydraulic accumulators can be added to conventional systems, but additional valves or pumps are required [27] and the recuperation efficiencies depend on the operation point. For a forwarder, Linjama and Tammisto showed that the fuel consumption can generally be reduced by approximately 15% with different hybrid concepts combined with conventional systems [27]. Digital hydraulics and hydraulic transformers, on the other hand, usually allow a direct connection of hydraulic accumulators to a constant pressure supply, which leads to higher energy savings. However, in the latter case and if an EM is used, it would also be possible to recuperate the energy electrically.

The improved energy efficiencies of those systems are beneficial for electrification—especially in terms of increased operation times of battery-powered vehicles. However, because EMs are no essential components for these concepts and their functionality is independent of the type of prime mover, a further survey of these concepts is beyond the scope of this paper. For sake of completeness, it still must be mentioned that they offer a potential alternative for effectively electrifying HDMMs, even though the EM would play a minor role and none of the concepts has been commercially utilized on HDMMs yet—neither with nor without EMs.

3. Electro-hydraulic actuator circuits

Another electro-hydraulic concept, which is based on the control capabilities of EMs, is utilizing one EM in combination with at least one pump for each actuator. This concept is commonly known under the name EHA. EHAs are already commercialized for aircraft and stationary applications, but not for HDMMs. The need for multiple EMs and pumps on a single implement system, novel hydraulic circuits and new safety standards [28] might be the reason why HDMM manufacturers stuck with more conventional systems so far. Still, this technology is further surveyed here, as it allows much higher energy efficiencies than conventional valve-controlled hydraulic concepts, which will be further discussed in Section 3.10. Therefore, the authors believe that EHAs will also enter the HDMM market sooner or later. Table 1 shows a list of HDMM EHA applications that were already presented by academia, except for [29] which is a patent assigned to Volvo Construction Equipment AB. This list provides proof for the feasibility of utilizing EHAs on HDMMs. The following subsections discuss specific aspects of EHAs, of which some might need to be reconsidered for the case of HDMM applications.

3.1. Control concept

The direct dedication of one EM to each actuator allows to control the actuator movement simply by controlling the torque or speed of the EM. For a better understanding of the control concept, the main equations are depicted: Due to the immediate connection, actuator and pump volume flows are equal ($Q_{\mathrm{Cyl}} = Q_{\mathrm{Pmp}}$), which can also be seen in the simple example depicted in Fig. 3. Accordingly, the basic Equations (1) and (2) result in (3), which shows the actuator speed as a function of the EM speed. In the equations, v is the cylinder-actuator speed and A is the cylinder area related to the flow, while n_{EM} represents the pump or EM speed and V_{Pmp} the pump displacement.

$$v = \frac{Q_{\mathrm{Cyl}}}{A} \tag{1}$$

$$Q_{\mathrm{Pmp}} = n_{\mathrm{EM}} \cdot V_{\mathrm{Pmp}} \tag{2}$$

$$v = \frac{n_{\mathrm{EM}} \cdot V_{\mathrm{Pmp}}}{A} \tag{3}$$

Equation (3) represents the fundamental equation for pump-control of cylinders. A detailed review of this topic carried out by Ketelsen et al. can be found in [7], while this paper only focuses on the characteristics that are important for HDMM implement applications. EHAs that utilize variable displacement in addition to variable speed are not considered here because of the increased costs that are not acceptable for HDMMs.

In case of rotary actuators, the control principle is similar, but the cylinder area is replaced by the actuator displacement in Equation (3) in order to achieve the rotational speed of the actuator. Same as the pump flow is proportional to the actuator speed, also pump pressure and actuator force (or torque) are directly linked and allow for force control, which is not further explained here because it is uncommon for HDMMs. However, it should be noticed that, as an effect of this actuator-force torque linkage, the torque of the prime mover in a pump-controlled system—in this case the EM—can also be negative, which means that EHAs can recuperate negative loads in form of electric energy.

3.2. Differential-flow handling

One of the main challenges of pump-controlled cylinder drives is the handling of differential cylinder flows. Double-rod cylinders, which are acceptable for aircraft and stationary applications, or rotational actuators show symmetric in- and outflows, which makes it easy to couple them directly to a pump, which has a quasi-symmetric flow behavior as well. However, HDMMs require compact single-rod cylinders due to limited space and high force requirements. Such a cylinder can be seen in Fig. 4 a.

Due to the differential cylinder areas on the rod side (A_R) and piston side (A_P), the volume flows at both ports, Q_R and Q_P, are not symmetrical

Table 1
List of publications that present EHA applications on HDMMs, containing the machine and actuator type if known.

Publication reference	HDMM type
[30]	scissor lift
[31]	forklift (5 m lifting height)
[32,33]	mini excavator; boom, stick and bucket
[34,35]	mini excavator, stick
[36]	9 t excavator
[37]	16 t excavator
[38]	Deere JD-48 backhoe
[29]	wheel loader boom and shovel tilt
[39,40]	crane (40 t payload, not mobile); knuckle boom (59 kW) and boom actuators

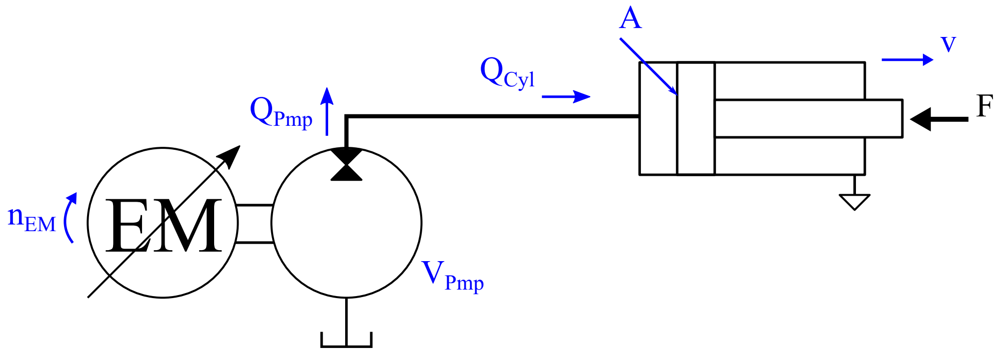


Fig. 3. Electro-hydraulic speed control of a cylinder in two quadrants corresponding to Equations (1), (2) and (3).

but differential, as can be seen in Equation (4). The ratio of the flows can be described by the cylinder area ratio Φ according to Equation (5).

$$Q_P = Q_R \cdot \Phi \tag{4}$$

$$\Phi = \frac{A_P}{A_R} \tag{5}$$

However, in a few special cases, even single-rod cylinders can show symmetrical flows:

- One case is the inversed connection of two cylinders that move with the same speed, which can be seen in Fig. 4b. This is convenient for articulated steering cylinders, which can be found on many HDMMs and were investigated in an EHA setup in [41].
- In [37,42], multi-chamber cylinders with one vented chamber (Fig. 4c) have been proposed, but this design reduces the maximum retraction force if the piston diameter is not increased. Furthermore, the cylinder would be more expensive due to the complex design.
- Instead of using a multi-chamber cylinder, alternatives with standard cylinders have been proposed in [37,43]. The concepts involve conventional single-rod cylinders that are actuated in parallel. Those can be found on most excavator booms for example. If $\Phi = 2$, both rod-side chambers can be connected while one piston-side chamber is vented, as can be seen in Fig. 4d. However, this solution reduces the maximum extension force as well, and the resulting unsymmetrical forces can lead to clamping.

All in all, the three proposed solutions for symmetric flow show only limited application areas. Thus, differential cylinder flows remain problematic in most cases where HDMM cylinder actuators are supposed to be coupled to a pump in order to form an EHA.

The solution is flow compensation mechanisms. Several concepts for those were presented by different researchers and show certain merits and demerits. Two subcategories can be distinguished: compensation by valves and compensation by using an asymmetric pump constellation. Both can be further divided into open- or closed-circuit versions. Table 2 shows exemplary circuits of those types and publications that make use of them. In the following descriptions of the principles, the same codes (A.I, A.II, B.I and B.II) are used.

3.2.1. *Valve-compensated differential flow handling*

The valve solution depicted under A.I is a very simple solution because the compensation works passively. Instead of two pilot-operated check valves, an inverse shuttle valve can be used as well. The downside is that A.I valves show undesired oscillation when being close to the pressure difference at which one valve closes and the other opens. Reviews and detailed analyzes of this problem can be found in [57–59]. In [44], a solution was proposed in which the oscillations could be avoided for a wide cylinder speed range while slightly increasing the energy consumption by using a special underlapped shuttle valve. In [60], Imam et al. used pilot-operated check valves alongside a throttling valve that stabilizes the mode-switching but also increased the energy consumption by 12% in the experiments. In [38], Costa and Sepehri successfully used an actively operated solenoid shuttle valve that fulfills the same main function as a passive valve in combination with a new, more detailed load-quadrant definition for the control. The quadrant definition included hydraulic throttling effects and resulted in oscillation-free steady states during experiments. In [57], Grønkær et al. further developed and improved this control approach for dynamic load conditions by adding artificial damping, signal filtering and more detailed physical modelling of the system. This led to stable operation over a wide range of also dynamic operation conditions, but was only tested in simulation so far. All these approaches show that mode-switching problems can be solved with some extra component and control effort.

The open-circuit solution shown for A.II is characterized by the fact that the returning oil passes the reservoir before it enters the pump again. This makes a directional valve necessary in order to achieve that the pump can always be connected to the high pressure side of the cylinder. Thus, similar to the compensation mechanism in A.I, mode switching is also problematic in this setup. Different than in A.I, a total loss of control, jerks of the cylinder during tilting loads and cavitation can appear if the valve is still switching and in the wrong position (pump connected to the

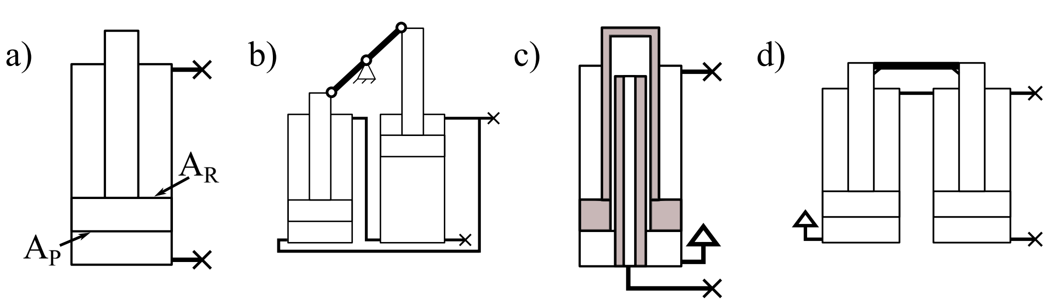
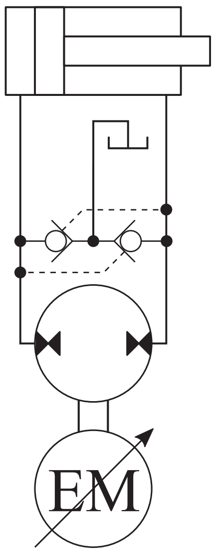
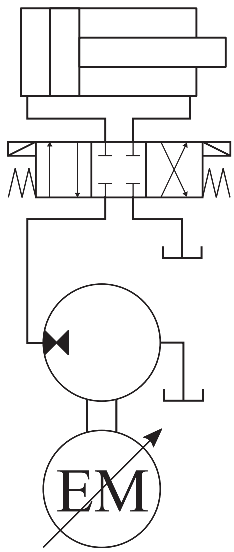
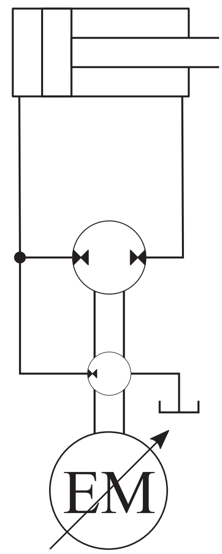
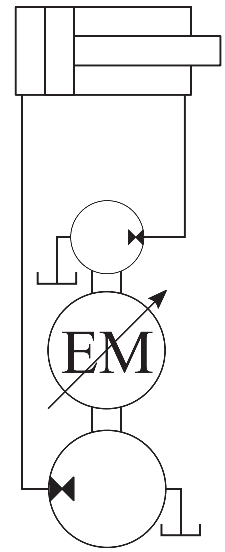


Fig. 4. One-rod cylinders: a) conventional and its areas with differential flows, b) combination of two counteracting cylinders for symmetrical flows [41], c) multi-chamber cylinder for symmetrical flows [37,42] and d) combination of two parallel cylinders for symmetrical flows [37,43]

Table 2
Different ways to compensate differential cylinder flows, with exemplary circuits and utilization examples.

Differential cylinder flow compensation			
By valves (A)		By asymmetric pump flows (B)	
Closed circuit (A.I)	Open circuit (A.II)	Closed circuit (B.I)	Open circuit (B.II)
 [44,38,29,45–48]	 [49,34,50,51]	 [52,53]	 [54,55,39,56,46,36]

low pressure side). The solutions proposed for mode switching issues in A.I [57,38] seem less promising here since the 4/3 valves handle higher flows, are thus larger and accordingly switching more slowly. Alternatively, in [50], Lin et al. placed a hydraulic-motor electric-generator combination in the return line in order to apply a load to the pump even during load braking. This avoids valve switching and the problematic oscillations for tilting loads. On the downside, higher control complexity, additional component costs and energy circulation losses appear. These are reasons why A.II architectures should generally be avoided for applications with frequently tilting loads. Still, the fact that the pump only operates in two quadrants, meaning the high- and low-pressure sides do not switch, is a major benefit of A.II. For two-quadrant pumps, the design of the ports and pressure grooves can be optimized for this condition, which leads to higher efficiencies compared to four-quadrant pumps as they are used in A.I [61].

For both, A.I and A.II, it should be further noticed that the circuits require pump displacements Φ-times larger than those of valve-controlled actuators in order to reach the same retraction speeds during load braking. This becomes more clear by looking at Fig. 5. To reach the velocity v_x, the pump in the valve-controlled system (Fig. 5b) only needs to supply the flow Q_x while the EHA pump (Fig. 5a) needs to supply Q_x for restive loads as well but receive $Q_x \cdot \Phi$ for braking because the high pressure side switches. The same phenomena makes it also necessary to consider the load direction in the EM-speed control, because the different

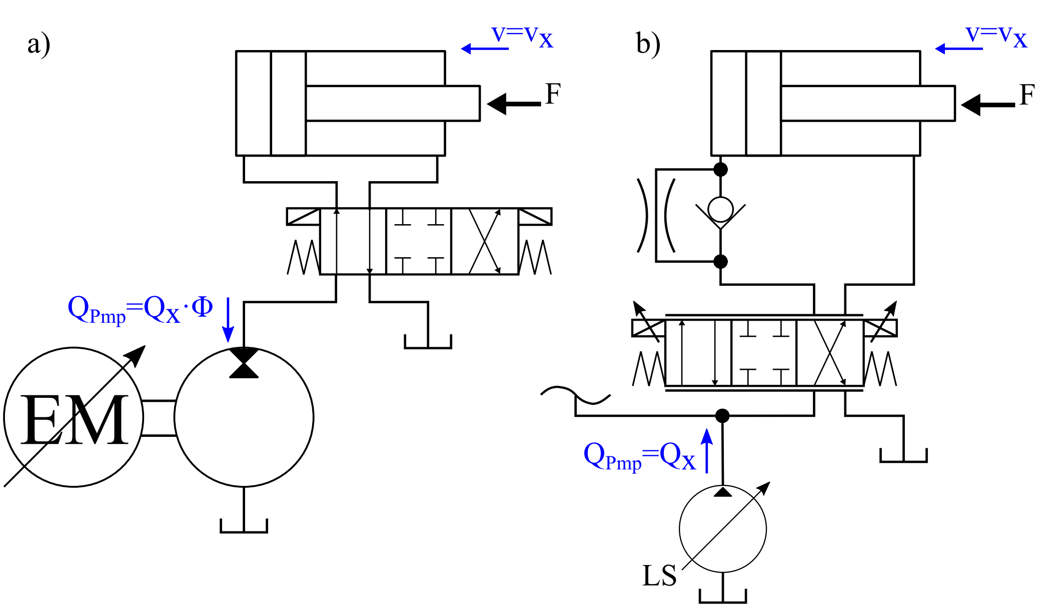


Fig. 5. Pump flow during load-braking retraction for a) single-pump EHA and b) valve-controlled LS system.

flows require different EM speeds to avoid control discontinuities during tilting loads.

Another common issue of solutions A.I and A.II is that the reservoir must be pressurized in order to avoid cavitation when oil coming from the reservoir needs to pass valves before it reaches the cylinder or pump. Operation points critical for cavitation can be seen in Fig. 6a–c, where an accumulator—a common solution for small- to medium-sized pressurized reservoirs—is used. In these conditions, a minimum accumulator pressure $p_{min,Acc}$ must remain.

At the same time, the reservoir pressure must stay below the pressure limitation $p_{max,Acc}$ of the low-pressure-pump port or drain line, respectively. This is symbolized in Fig. 6d and e. This limitation is caused by the pump shaft sealing and is typically not higher than a few bars—for example 3 bar [62] or 5 bar [63]. Those two pressures, $p_{min,Acc}$ and $p_{max,Acc}$, define the permissible pressure range for the accumulator.

Remaining in the pressure range is especially challenging for large cylinders that require high volume flows and for cylinders with high rod volumes such as telescope cylinders—both are common for HDMMs. The higher the maximum flows—which cause the cavitation-critical pressure drops—are, the higher the lower pressure limit $p_{min,Acc}$ and the smaller the pressure range get. According to Fig. 6, the accumulator pressure will vary between minimum and maximum pressure when it charges and discharges the volume equal to the cylinder rod. The higher the rod volume is, the higher the nominal accumulator volume needs to be in order to stay within the permissible pressure band. Those accumulator sizes can easily get unacceptably high. In [64], Ketelsen et al. addressed this issue by placing a so-called bootstrap reservoir in between the hydraulic chamber and the gas chamber of an accumulator. The bootstrap reduces the necessary gas volume and increases the gas pressure. However, the bootstrap reservoir needs to be available in the right size, increases costs, and the volume reduction might be still insufficient for extreme cases. Also charge pumps, which are sometimes used in combination with accumulators for reservoir pressurization in pump control (e.g., [65]), would need to be very large and would result in high losses. In [66], the accumulator and low pressure port of the pump are decoupled in order to operate with higher accumulator pressures; however, this approach also has downsides in the form of a decreased maximum cylinder pressure difference and the need for a charge pump as well as an additional sealed, depressurized reservoir next to the accumulator. Furthermore, valves—especially rarely used ones like inverse shuttle valves—are not always available for high flows. In this case, for each function, two valves need to be used in parallel, which increases the costs significantly as shown by Schmidt et al. [67]. All in all, A.I and A.II can come to their limits with large HDMM cylinders, and alternative solutions should be considered for those.

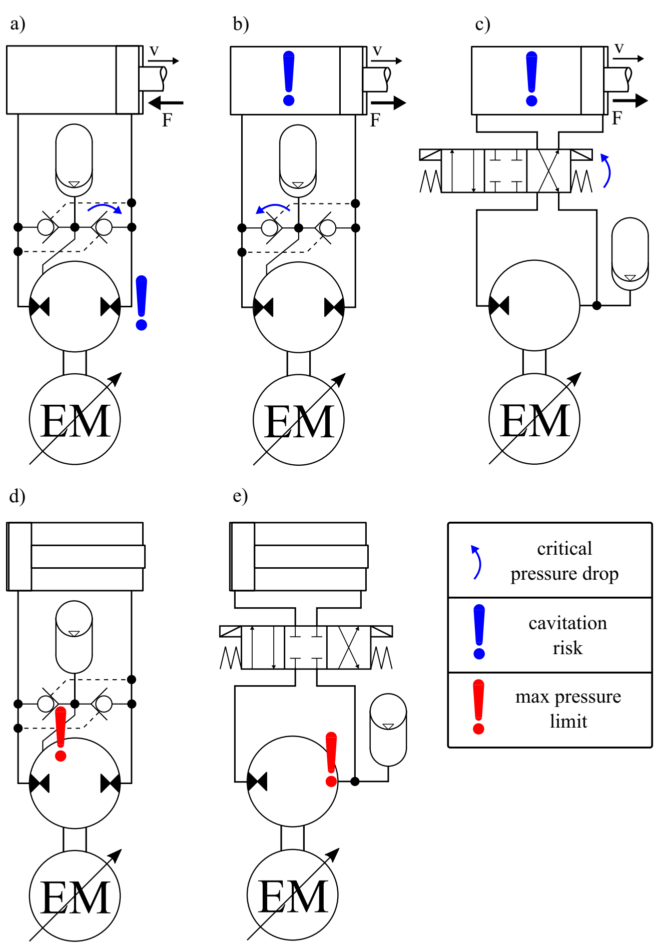


Fig. 6. Critical accumulator-reservoir pressures in valve-compensated EHAs a), b) and c) minimum pressure, d) and e) maximum pressure.

3.2.2. *Pump-compensated differential flow handling*

Principles B.I and B.II seem to have a more simple design compared to A.I and A.II because no compensating valves are necessary. The result is that less throttling appears along the flow paths, and no valves need to be controlled or can show mode switching issues. Also, suction lines are direct and short; thus, the flow resistance is low and the reservoir must not be pressurized. This makes large cylinder sizes less problematic. Furthermore, the relation between EM speed and cylinder speed is load-independent which also leads to simpler controls. On the down side, not one but two pumps are required, which, in case of B.II, also have a higher combined displacement than the pumps necessary for A.I and A.II. This increases costs, weight and mounting effort. Furthermore, an often reported [68,56,55] functional issue appears if the ratio between the pump displacements does not match the cylinder area ratio. The mismatch leads to two effects: In one cylinder direction, the flow supplied by pump one results in too much returning flow for pump two, and over-pressurization appears. In the other direction, the flow supplied by pump two is not enough to return a sufficient flow to pump one, and cavitation appears. In this scenario, pump one had to be smaller and/or pump two larger to match the cylinder ratio better.

To minimize these effects, in [46], ±5% is said to be the maximum permissible error between displacement and cylinder ratio, which seems to match with most other similar published setups. Finding a suitable pump match within that error range can prove difficult for all the many different cylinder area ratios available on the market. A pump-matching algorithm proposed by Ketelsen et al. [39] took 1000 different pump designs into account in order to find close ratio matches. In [68], the option to reduce the mismatch by adding a mechanical transmission stage to one of the pumps was proposed and successfully validated by simulations, but extra gear-box components can be costly. However, a perfect pump-cylinder ratio match can never be achieved as effects like non-linear pump leakage change the effective pump flow ratio over changing operation conditions. Thus, the mismatch problems will always remain to certain extents and should be addressed.

Cavitation can be relatively easily avoided by using anti-cavitation check valves as it was done in [56,55,46,36]. Handling over-pressurization at the same time is harder to achieve, and different approaches have been presented:

- The simplest approach utilizes relief valves, but this will result in increased pressure and high losses caused by the relief flow.
- In [56], proportional solenoid valves were used to minimize the over-pressurization.
- A step further was done in [55,69], where cavitation was avoided by adding a third, small pump to the shaft that compensated missing flow, but also increased the lower pressure of the cylinder and thus its stiffness, which is beneficial for closed-loop control; also here,

proportional solenoid valves were used to limit the over-pressurization and to control the cylinder stiffness.

- In [68,52], an accumulator was directly connected to one cylinder chamber, which prevents over-pressurization due to a ratio mismatch, but also resulted in a permanently low cylinder stiffness for one load direction—which can be problematic for four-quadrant HDMM applications. This issue could be solved by connecting the accumulator with pilot operated check valves similar to A.I.
- Another option is using separate EMs for each pump [70–72] or two variable-displacement pumps [71]. This decouples the flows of the two pumps and even allows to control the cylinder stiffness by applying defined back pressure. However, the costs and control complexity will increase significantly.

As all those solutions require extra components and come with certain demerits, multiple-pump solutions are less common for EHAs in stationary or aircraft applications. Still, for HDMMs, they attract interest as high-flow requirements and large rod volumes are less problematic compared to valve-compensated EHAs.

3.3. Load-holding capability

Load-holding is essential for HDMM applications since they often require actuators to be locked at a certain position—also under load and for long times. Since most stationary applications do not show these requirements, a majority of the published EHAs do not include specific load-holding mechanisms. In this case, with a direct cylinder-pump connection, the pump leakage needs to be compensated in order to keep the actuator at its position. This requires the EM to spin constantly at low speeds, which results in a constant power consumption—the higher the load, the higher the consumption—and position sensing or exact prediction of the pressure-dependent leakage is necessary to keep the cylinder at its exact position. To avoid this, a few EHA designs—mainly intended for HDMMs—have been proposed that use additional locking valves between pump and cylinder. Those are listed in Table 3.

In general, all those valves increase the flow losses compared to a direct hydraulic connection, even if the valves are fully opened. Still, those designs can be more efficient since the EM can be totally switched off during load holding and will not consume standby power anymore. Passive pilot-operated check valves are opened when high pressure is applied to the other cylinder side by the pump. This design is very simple, but does not allow for recuperation and should only be used in applications with low or no braking loads. The same is true for counterbalance valves.

On the other hand, active pilot-operated valves and other active valves open when pressure is applied by a solenoid valve. This allows recuperation, but requires extra control inputs as well. Furthermore, it should be noticed that two-position valves can be *poppet* valves, which have less leakage and result in a lower risk of creeping cylinders compared to the 4/3 directional *spool* valves in the open-circuit EHAs.

Table 3
List of published EHA designs with load-holding valves.

Publication reference	Load-holding valve type
[73,46,74,75]	passive pilot-operated check valves—on both cylinder sides
[76]	counterbalance valve—only on rod side of pulling cylinder
[3]	counterbalance valves—on both cylinder sides
[77–79]	active pilot-operated check valves—on both cylinder sides
[80,34,81]	open circuit with solenoid directional valve(s) that have a locking position
[37,82,31]	2/2 solenoid valve—only on the piston side of a lifting cylinder
[83,48,36,32]	2/2 solenoid valves—on both cylinder side
[84]	2/4 solenoid valve—for both cylinder sides together

Moreover, putting only one locking valve on one side of a cylinder can save costs, but is only feasible for applications in which the static-load direction does not change—such as forklift or dumping cylinders.

3.4. Oil filtering

Particle contamination of the hydraulic fluid can cause increased wear of components such as pumps and valves and is a broad field of research. Specific aspects of this field are reviewed here because oil contamination is likely in dirty HDMM work environments and EHAs have issues with oil filtering. In conventional systems, filters can be easily mounted in the common low-pressure return line. EHA designs, on the other hand, can be often found without any filters, because compactness is favored or no suitable location can be found for the filter. However, Michel and Weber conducted endurance tests with an EHA in an industrial environment and could show that, even with low initial oil contamination, oil should be filtered due to the critical self-contamination of the circuit [85]. Thus, filtering appears mandatory for HDMM applications.

The *ideal* filter location stays constantly at low pressure, since high-pressure filters are heavier, harder to maintain and more expensive than the more common low-pressure filters. Furthermore, it is ideal to filter the whole amount of circulating oil. For A.II-like open circuits, the return line offers these ideal conditions. For B.II designs, two filters, one for each pump arranged like in Fig. 7a, or additional valves forming a common return, which can be seen in [67], would be necessary. Thus, the filter or respectively the combined valves, which assure that flow is passing the filter only in one direction, increase the pressure drop for the pump suction flows. This means for high suction flows this case would require a pressurized reservoir to avoid pump cavitation. Alternatively, a separate filtering (and cooling) circuit could be used to filter the oil in the reservoir, but would increase the costs; furthermore, not the full flow would be filtered, leaving a risk for high contamination.

For closed circuits (A.I and B.I), filtering is even more problematic since the only low-pressure location is only passed by a part of the circulating flow—the differential flow between rod and piston side. To assure a defined, low contamination, filtering of the full flow is necessary and high-pressure filters are required. In [42], two of such are used, one for each line, in combination with four check valves each (see Fig. 7b left), which allows filtering in both flow directions. In [86], the same setup with only two check valves each (see Fig. 7b right) and thus filtering in only one direction is used. Both designs appear costly due to the high number of components with high nominal flow as well as high pressure requirements and should only be used if very low contamination is of high importance, e.g., for the pump. For an A.I like EHA, [48] shows a design that filters the differential flow on its way to the low-pressure source—an accumulator in this case—whenever the cylinder is retracting. In [78], the differential flow is only filtered during assistive retraction, as depicted in Fig. 7c. This leads to less filtering, but the design has one less check valve in the suction path for assistive extension and thus reduces the risk of cavitation or the required accumulator pressure, respectively (see Section 3.2.1 for explanation). An example for filtering with an additional charge pump can be seen in Fig. 7d. The charge-pump flow is filtered, and contaminated oil can leave the circuit through a low pressure relief valve. That way, the filtered flow is potentially even smaller than the differential flow, but a single filter can be used for multiple EHAs if they are utilizing the same charge pump as their low pressure supply.

3.5. Oil temperature management

Also temperature management can be challenging for EHAs, which is explained in the following. Temperature influences the properties of the hydraulic oil—mainly viscosity—significantly. At low temperatures, the viscosity is high and pressure drops accordingly high as well; this leads to lower efficiencies, which was investigated in [87] for an EHA in sub-zero conditions, but more importantly it can lead to cavitation. At

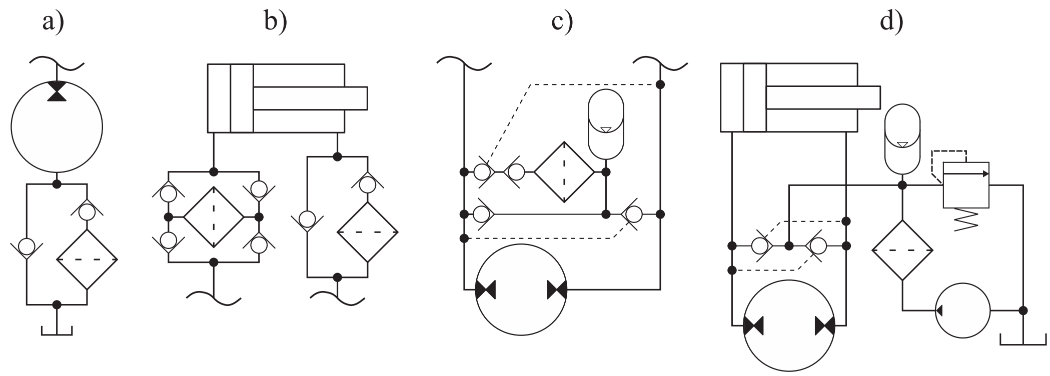


Fig. 7. Options for filtering in EHA circuits a) for pumps in B.II type b) high-pressure in-line filter for filtering in both directions (left) [42] and one direction (right) [86] c) return line filter for closed circuit according to [78] d) filtering with charge pump.

high temperatures, the low viscosity increases leakages and thus volumetric losses, but also lubrication can decrease dangerously. Furthermore, high temperatures are critical because the lifespan of conventional hydraulic oil decreases by half for every ten degrees that the maximum oil temperature increases, starting from 20 years in a system with 40°C maximum [88].

For low-power stationary applications, in which EHAs are already commercialized, the passive cooling capabilities of the actuators—also known as calm cooling—alone can be sufficient to maintain a sufficiently low oil temperature, also because the energy losses of EHAs are typically low. In [89], Michel and Weber describe and validate how the self-cooling capability can be calculated—at least for constant ambient temperature. However, in the experiments, even a small EHA reached temperatures of 76°C during continuous oscillations with peak power of 1.5 kW. In [90], Ketelsen et al. mention 5–10 kW as the approximate limit at which passive cooling is not sufficient anymore; furthermore, they present a simplified lumped parameter modelling approach specifically for the purpose of predicting the cooling capability of EHAs during the design stage, since cooling is a crucial issue for this type of actuators. A similar model was presented in [91] for an EHA with a maximum load of 50 kN for the prediction of cooling requirements in a mobile application.

For HDMMs, power levels—and thus energy losses—are potentially higher than for smaller stationary EHA applications while surface areas—and thus calm cooling—might only slightly increase for larger actuator sizes. Additionally, HDMM environment temperatures can vary from extremely low in arctic regions to very high in tropics or deserts. Thus, coolers or even heaters will be necessary for some HDMM applications. For a mini excavator, this was demonstrated by Busquets and Ivantysynova [92] with a displacement-control system, which shows similarly low losses as EHAs; the system required a charge pump with a cooler whose heat coefficient had to vary between 10 to 450 W/K in order to maintain an oil temperature below 65°C during a 45 min digging cycle. However, for the same excavator, it was also shown in [93] that the required cooling power can be reduced by half for pump-controlled actuators compared to valve-controlled actuators.

Similar to filters, coolers are usually designed for low pressures, and high-pressure solutions are heavier as well as more expensive. An advantage over filters is that the cooler does not have to be in-line with the main circuit. Good cooling results can also be achieved with a separate cooling circuit connected to the reservoir, which is especially suitable for open circuits. For closed circuits, the cooler should be placed at the same position as a low-pressure filter, as it was done in [67,78,34] as well. Furthermore, it should be noticed that for cooling charge-pump solutions show the additional benefit of being able to drive a fan, which can be used for active cooling [94].

3.6. *Zonal self-contained actuators or centralized structure?*

For stationary or aircraft applications, EHAs can be found almost solely with self-contained designs. Most benefits of self-contained actuators that form so-called *hydraulic zones* also apply to HDMMs: shorter lines, which are cheaper, have lower flow resistance and are hydraulically more stiff; simple installation and interface; and compactness. However, compact self-contained solutions also bear certain challenges especially for HDMM applications, and centralized EHA designs that share a common reservoir or low-pressure supply can be more beneficial in some cases. This will be further discussed in the following.

3.6.1. *Common oil maintenance*

A major issue of zonal EHA concepts is the need for separate reservoirs as well as cooling and filtering for each actuator, while centralized solutions can combine the reservoir for multiple EHAs in one and also use a single charge pump as a low pressure supply for all actuators. Moreover, a common reservoir can also be used for combined cooling and filtering, which reduces the costs significantly. The cost reduction increases with the number of EHAs that are combined. A similar cost effect was described by Schmidt et al. for the comparison between a centralized LS system and zonal EHAs with a varying number of actuators [67]. Furthermore, reservoirs for self-contained EHAs need to be redesigned, as changing orientations of the EHA can make conventional vented tanks unsuitable. In [54], the use of an accumulator was proposed for this problem, but can prove difficult for large cylinders as it was already explained in Section 3.2.1.

3.6.2. *Pump and electric machine sharing*

In [95,96], Busquets and Ivantysynova proposed a pump-sharing approach for the reduction of the number of pumps in a displacement-controlled system. The same approach can be applied to an EHA system to reduce the number of pumps as well as EMs. This can be seen in Fig. 8: The same pump and EM can be used for two (or technically even more) actuators, by switching the directional valve—but only as long as they do not need to be actuated simultaneously. Many HDMMs comprise auxiliary actuators, such as side stabilizers, that are only actuated when the main actuators stand still, which is an ideal case for pump sharing. This approach reduces the component costs, but is only possible with a centralized, non-zonal structure.

3.6.3. *Increased volume and mass*

Another issue of zonal EHA concepts for HDMMs is the location of all necessary components right next to the cylinder since this requires more space and results in more mass compared to a hydraulic cylinder alone. The additional space might be already available, as can be seen for an excavator stick actuator in [66], but might as well require to redesign whole machines that currently offer no space next to the cylinders.

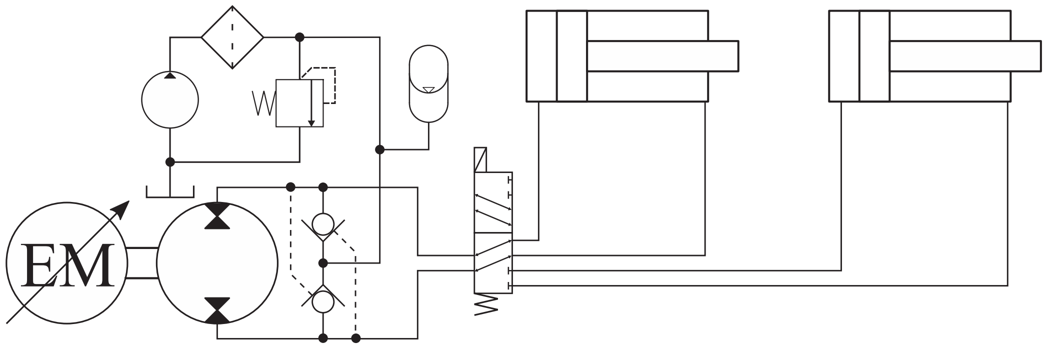


Fig. 8. A centralized system with two actuators with pump and EM sharing similar to [95,96].

The increased mass, on the other hand, can lower the payload of load carrying machines that have the EHA attached to their moving parts. In [40,97], this was investigated for different EHA designs and a crane application. The heaviest EHA was 50% heavier than a cylinder alone, but decreased the payload only by 2.1%, which seems acceptable. In [97], the effect of the increased mass on a micro excavator was investigated and 15% increased energy consumption could be observed for a typical working cycle. However, as many conventional HDMMs use the common hydraulic tank also as a counterweight for a boom, the tank mass would also have to be replaced if self-contained actuators without common tank are used.

3.6.4. *Changing implement attachments*

Many HDMMs, such as excavators or tractors, have the option to attach various additional tools. For novel HDMMs that utilize zonal EHAs, this would require that each attachment with actuators has its own EHA(s) or other electric actuator(s) included because the HDMM has no hydraulic power-take-off connected to a centralized supply anymore. Since this would not allow to continue using old attachments and new attachments would increase significantly in price, a centralized available EHA supply with power-take-off should be considered, at least for HDMMs that typically frequently switch their attachments.

3.6.5. *Actuation of parallel cylinders*

A last issue that should be mentioned here is the actuation of parallel cylinders. If they are close to each other, e.g., excavator boom cylinders, they can be easily coupled by short hydraulic lines and handled as one cylinder. On the other hand, if they are further apart, e.g., the lift cylinders on a skid-steer, right and left of the operator, two options remain: long hydraulic lines connecting both cylinders and the benefits of self-contained EHAs vanish; or two separate EHAs are used for each lift cylinder. The latter case would require a full set of components for each cylinder, which is costly. Moreover, a complex control would be necessary which assures that both cylinders move with the exact same speeds to prevent clamping forces. Approaches for that could be adopted from the aircraft industry, where often two redundant actuators are coupled for safety reasons. To prevent “force fighting”, one of those actuators can be speed- or position-controlled while the other has a force-tracking controller [98] that makes it follow and support the other actuator.

3.7. *Measures for reducing the installed power*

No matter if centralized or zonal, applying the EHA principle to all actuators of an HDMM will most likely go along with a significant increase in installed electric power and number of pumps compared to more convention valve-controlled concepts in which all actuators can share a common pump and EM instead of a set for each actuator. In terms of pumps, the additional component costs might be acceptable, but EMs and the required power electronics can become unacceptably expensive if the number of actuators und thus EMs is high. This might be the main reason why no market nor prototype solution of an HDMM with solely EHAs has been presented so far. To change this, measures for decreasing the installed power of EHA systems are required of which some have already been proposed.

In Section 3.6.2, the option but also challenges of pump sharing [95, 96] have been discussed. A further option is applying energy storage components for hybridization between the EM and the mechanical output of the actuator that allow downsizing of the EM due to lower peak loads. The hydraulic version of this will be discussed in Section 5.2.1. Another option is to install linear springs in parallel to cylinders that perform a lifting motion against gravity. Those springs are compressed during lowering and reduce the force that is required for lifting. In [6], such a spring has been realized by means of single-acting hydraulic cylinders that are directly connected to an hydraulic cylinder. In this study with an 6 t excavator boom, the resulting load on the actual actuator was low enough to utilize an electromechanical actuator with low power capability, but an EHA with a relatively small EM could have been used as well. A more straight-forward version of this idea—a simple gas spring in parallel to the hydraulic actuators—is already being used in material handling machines by Liebherr under the name *ECR* over several years; one of the related patents is [99]. However, the gas filling pressure can only be optimized for a specific handling weight. If weights vary or digging is also supposed to be performed, the choice of gas pressure will be a compromise

Finally, it remains questionable if each actuator on an HDMM should be transformed into an EHA. Many HDMM comprise actuator functions that are only rarely used or only for very short times, such as stabilizers, outriggers or adjustment cylinders of two-piece excavator booms. Accordingly, those cylinders show rather low energy turnovers, and the increased efficiency of EHA would only show minor consumption saving while still increasing the installed electric power. This was also explained in [100] and the justification for combining an EHA for the boom function of a backhoe with an conventional valve-controlled system for the remaining actuators. Similarly, in [101], an HDMM implement system is proposed with the option to choose between EHA-like structures for the main-consuming actuators and valve control for the low-consuming actuators.

3.8. *Partial adoption of the electro-hydraulic actuator concept*

Furthermore, electro-hydraulic circuits can be found that combine aspects of EHAs and conventional circuit solutions, which can be beneficial for HDMMs. In [102], a conventional valve-controlled circuit is combined with an A.II-like structure, which can be seen simplified in Fig. 9a. The conventional circuit is in charge of controlling speed and supplying the main power, but the pump-EM combination can still regenerate energy from load braking, store it in a battery and feed it back hydraulically if necessary. The control of this over-actuated system is unknown. Still, the benefits can be identified as the good and well-known controllability of valve-controlled systems with a high stiffness in addition to the recuperation capability.

In [103], actuators are supplied by one or multiple constant-pressure supplies, but a pump-EM unit is located in between actuator and supply, as depicted in Fig. 9b. The main power is supplied by the centralized constant pressure supply, while the EM can control the actuator speed just like it is done in an EHA. The advantage is that the EM only needs to slightly boost or reduce pressure which requires lower torques compared to conventional EHAs, which must supply the full power. Thus, the EM can be smaller and costs can be saved.

3.9. Control behavior and arising possibilities

The baseline for the control behavior of EHAs are conventional valve-controlled systems where the control bandwidth strongly depends on the valve-dynamics. For systems with variable-displacement pumps for variable flow, also the dynamics of the pump are important since the valves can only control an increase in flow if the pump can supply the additional flow fast enough. However, both components, valves and variable pumps, have rather low inertia in their mechanical adjustment mechanisms and thus fast dynamics. When it comes to EHAs, however, the actual control element is the EM, which has rotating components with a rather high inertia that needs to be accelerated or decelerated whenever the controller acts. For that reason, EHAs are generally known to have a lower control bandwidth than valve- or displacement-controlled systems, as it is also stated in [104] with the same explanation.

However, for traditional mobile applications, the dynamic differences due to different inertia are less significance due to several reasons: First of all, the standard way of operating an HDMM is still having a human operator on board. In this case, the electro-hydraulic system on the HDMM simply represents an open-loop control plant and the control loop—typically speed control—is closed by the human, who holds the joysticks. Due to human nature, the requirements in terms of control bandwidth of the electro-hydraulic system are much lower compared to the case of semi-autonomous applications where the control loop is closed by a software controller and stability issues arise if the plant bandwidth is not high enough. Moreover, the dynamics of valves and variable-displacement pumps depend not only on the inertia of the adjustment mechanisms but also the control bandwiths of the adjusting actuators [104]—solenoids and pilot valves. For mobile applications, such components are—due to cost reasons—typically less dynamic than for stationary or aircraft applications—which are the subject of [104]—where e.g., servo-valves are used instead of simple proportional valves. This shows again that the lower control bandwidth might be of lesser importance for mobile applications than it is for stationary or aircraft applications. In case that the bandwidth of an EHA should still be too low to allow the operator to perform e.g., fast shaking movements, there are also measures to increase the bandwidth or decrease the response time vice versa. To reach a faster acceleration with a high inertia, simply more torque is required from the EM. Accordingly the EM size and thus maximum torque could be increased or the short-time overload capability of EMs, which will be discussed in Section 4.1 later, can be utilized. Furthermore, if the maximum flow of the actuator is very high, which is typical for HDMMs that often require large cylinders for high forces, variable-displacement pumps could be used in combination with a variable-speed EM motor for an extra degree of freedom to increase the control bandwidth momentarily. This will also be elaborated further in Section 4.2.1.

3.9.1. Closed-loop control

However, the control bandwidth of EM-pump units but also of the connected hydraulic circuit become more important with the perceptible trend towards semi-autonomous [105] or even fully atomized [106] HDMMs, which require position control—typically with closed loops—to follow trajectories autonomously. Examples of closed-loop controlled EHAs can mainly be found so far for stationary applications, e.g., [46], but also HDMM applications such as a back hoe [107] have been investigated. Also with a mobile application background, Hagen et al. compared in [78] an EHA to an LS valve-controlled actuator in terms of position-control performance. Despite the concerns about higher inertia, the EHA control, which also included a pressure feedback for improved damping characteristics, achieved up to 66% less tracking error and 61% less overshoot with a 10 ms shorter rise time and 75% faster settling. Furthermore, it should be noticed that the bench-mark control valve in this study was a servo valve with integrated position control loop, which is more advanced than standard mobile valves for open-loop speed control. For EHAs, on the other hand, no extra hardware—except for the position sensor at the cylinder—is necessary to implement the closed-loop control.

Nevertheless, also challenges were reported for *closing the loop* of an EHA. In [108], increased losses due to idling at stationary conditions were reported for certain control approaches. Furthermore, several studies were solely focused on improving properties such as drive stiffness—which is usually low for EHAs due to constantly low pressures on one of the cylinder sides—damping or response time [108,72,69,109,110]. A promising approach for dealing with those properties was

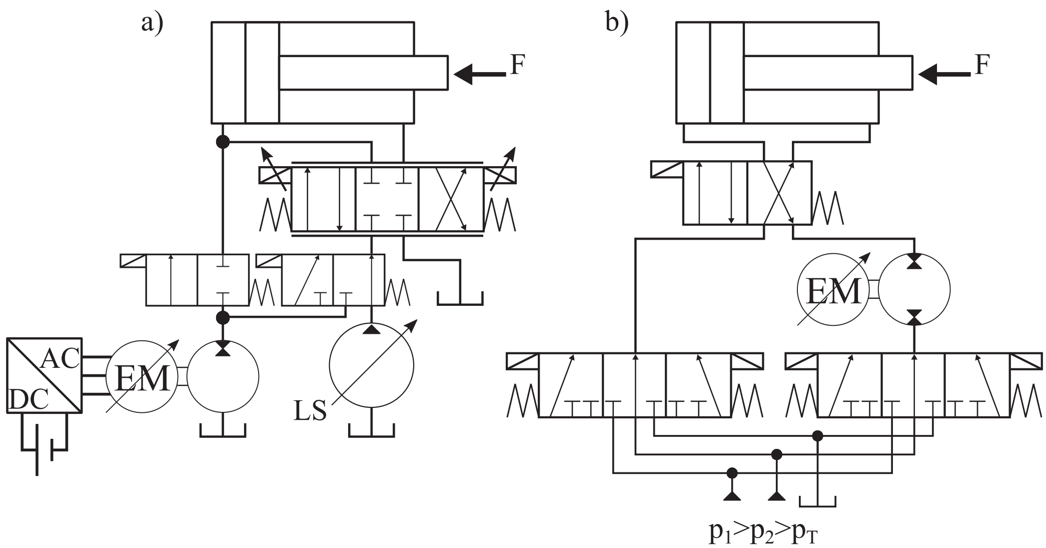


Fig. 9. Two hybrid circuits a) similar to [102] and b) similar to [103].

proposed by Schmidt et al., who proposed to use the EM not to control the pump flow but cylinder force instead [111]. This means opening the EM speed-control loop and using the high current-control bandwidth of the inverter (more than 400 Hz [111]) to directly control and compensate the whole electro-hydraulic power transmission from EM to linear cylinder force output. Position tracing can still be achieved by using the cylinder force to correct a measured position error. The approach showed very promising improvements in dynamics compared to traditional EHA-control approaches and might become the new standard for closed-loop control of EHAs.

3.9.2. Sensor-less position control

As already mentioned, position control capabilities might be required in the future for the automation of HDMM, but the need for additional, costly position sensors for closing the loop is a downside of this trend. Accordingly, studies on EHAs that focus on the idea of controlling the position of EHAs without additional sensors—also called sensor-less—seem promising. The concept uses the knowledge about the EM movement in combination with knowledge about the relatively simple and stiff hydraulic circuit to predict the cylinder movement:

According to Equation (3) and considering a volumetric loss term Q_d, the position $x_{\mathrm{Cyl}(t1)}$ of an EHA cylinder can be expressed as

$$x_{\mathrm{Cyl}(t1)} = x_{(t0)} + \int_{t0}^{t1} \frac{n_{\mathrm{EM}} \cdot V_{\mathrm{Pmp}} - Q_d}{A} \mathrm{dt}, \quad (6)$$

the integral of the cylinder speed. vice versa, this equation can be used to calculate the required revolutions of the EM to reach a certain position—to achieve a sensor-less position control. The EM speed and position—as well as the revolutions respectively—can be predicted by observers that utilize only the standard voltage and current sensors of the EM (back electro magnetic force method or flux linkage method [112]). Thus, the accuracy mainly depends on the knowledge of the initial position $x_{(t0)}$ and the disturbance flow, which combines the impact of pump leakage, cylinder leakage and oil compressibility [46].

One of the first published applications of this concept can be seen in [113,114] where the position of a forklift is controlled; in this study, Q_d is calculated considering a constant volumetric pump efficiency as well as the oil compressibility. The pressure change that causes the compression was also detected in a sensor-less way by observing the EM torque instead, which is proportional to the pressure at stationary conditions. A position error of 30 mm after a single position change could be achieved. In the paper [114], it is proposed to use a simple proximity sensor to achieve more exact positioning if necessary. However, proximity sensors can only be used for specific tasks as well as defined environments like a warehouse, and a 30 mm error might already be sufficiently low for most off-road applications anyway.

Similar studies have been done for a wheel loader [115] with 30 mm error (max stroke 850 mm) in simulation, an excavator [33] with 20 mm error (max stroke 290 mm and 410 mm) in simulation and a crane [116] with up to 10 mm error (max stroke 400 mm) in experiments (all errors detected after one or only a few cycles). All those studies reported one main problem of the sensor-less concept: The position error tends to increase with every work cycle. The reason is that Q_d depends on many parameters as well as influences like load, speed and temperature, which is systematically analyzed in [115]. Thus, an exact prediction of Q_d is impossible, and according to Equation (6), an increasing position error results. As a solution, only a few discrete positions along the cylinder stroke could be sensed by a proximity or similar sensor. This requires less effort compared to a continuous position sensor; still, $x_{(t0)}$ in the position estimation algorithm Equation (6) could be "reset" every time the discrete position sensor is passed, which limits the maximum position error.

3.9.3. Compensation of pump pulsation

Another benefit, arising from the special architectures of EHAs, was mentioned in [113]. It is the idea of using the fast torque and speed response capabilities of EMs to compensate pump pulsation. The mechanical designs of gear as well as piston pumps result in flow pulsation with a frequency proportional to the shaft speed and number of gear teeth or pistons. This flow pulsation can then also result in pressure pulsation and undesired noise. When an EM is dedicated to only one pump and the ripple profile is known, the EM can be controlled in a way that counteracts the pump pulsation by spinning faster during a low pulse and slower during a high pulse. This idea was already patented [117] and seems valuable as the ICEs as the previous main noise sources are starting to disappear and pump noises will become more noticeable on HDMMs.

3.10. Energy-efficiency benefit of electro-hydraulic actuators

Finally, the energy-efficiency advantage is discussed in this section as one of the main arguments for using EHAs. Ideally, due to the individual control of each actuator with a separate pump circuit, according to the principle explained in Section 3.1, there occur only losses of the pump, the connecting lines and the hydraulic cylinder—all in a relatively low region. Furthermore, the basic EHA principle allows recuperation for further energy savings. However, the previous sections showed that extra components might be necessary for a functional HDMM application, and those elements can compromise the energy efficiency. Valves that compensate differential flows lead to throttling losses, and load-holding valves are located in the main flow paths which increases the throttling losses as well, same as filters do. Casoli et al. further concluded from their simulation studies that the length of the supply lines, which is very short for zonal but long for centralized architectures, has a significant influence on the efficiency [36]. Moreover, certain types of load-holding valves can prevent recuperation as it is explained in Section 3.3. All in all, this explains why the efficiencies can vary quit significantly between different EHA types.

The efficiency of an EHA, or the EHA circuit respectively, is commonly defined as the ratio between the linear mechanical cylinder output power to the rotational mechanical pump input power. Moreover, the efficiency is not a constant value but can vary quit significantly across the operation range of an EHA. The line and directional-valve loss terms for example are flow and thus speed dependent but independent of the load; therefore, EHAs show higher efficiencies at high loads [61,87]. Accordingly, the ideal way of comparing different EHAs efficiency-wise is looking at their efficiency maps, such as it is done in [61]. However, most publications do not provide these maps—which can experimentally only be recorded with a suitable test rig at which defined loads can be applied—but only give specific values such as maxima or average values. For the hydraulic EHA efficiencies, up to 65% could be achieved in [118] as well as in [87], and in [61] up to 80% could be achieved. The latter value is even in the range of large electromechanical actuators which shows a high competitiveness of EHAs in terms of efficiency.

Nevertheless, those efficiency values only tell about specific operation points, and it is actually more valuable for HDMMs to look at the energy that is required for performing certain work cycles and to compare those values between different actuator types. Usually, this is done by taking the energy consumption of a conventional system—often a valve-controlled LS system—as a reference and calculating the reduction of energy consumption with the novel EHA systems. In [118], Hagen et al. achieved 62% energy savings for a simple crane lifting-lowering cycle with an EHA compared to a constant-pressure valve-controlled actuator in experiments. In [36], Casoli et al. demonstrated up to 60% reduced energy consumption for an excavator digging cycle compared to an LS system in their simulations. In [100], Kärnell et al. calculated consumption reductions of a backhoe with one EHA and two LS actuators compared to the conventional backhoe with LS only. Around 55% less energy was required for a digging cycle and even 65% less for a grading cycle.

Comparing those consumption reductions between the different EHA concepts or also other novel concepts that focus on energy efficiency would proof valuable but is not feasible since most studies are based on different assumptions, applications and work cycles. Some simulation studies neglect certain losses while others do not, system losses can strongly depend on the number of actuators and thus HDMM machine types, and even two digging cycles can be very different from each other in terms of maximum loads and speeds. In [100], as an example, the consumption reductions of the same system were 10% different between two different cycles, and in [101]—also without changes in the EHA-like system—the reductions were up to 20% different between two cycles. For the same reason, an evaluation algorithm was proposed in [119] that rates the benefits of an actuator concept—which also include energy efficiency—always with reference to the machine type and work task that it is intended for.

All in all, the discussion and given references show that EHAs can lead to a significant energetic benefit, but absolute efficiency or reduction values can only be compared to other concepts if they were analysed under the same conditions, which is rarely the case. This issue could be addressed by using standardized duty cycles for simulations or experiments across different research institutions, or by sharing simulation models of HDMMs between researchers that can serve as standardized platforms where each researcher can apply their own actuator concepts on.

4. Electric prime movers and matching pumps

The electric prime mover or EM is the key component that leads to advantages of electro-hydraulic systems such as high energy efficiency or flexibility. Nevertheless, the attached pump is important as well since pump and EM can restrict each other if they have different operation specifications. Thus, it is important to find a good match between EM and pump type in order to achieve an efficient interface between the electric and hydraulic domain.

4.1. Electric machines

Rotational EMs have been used and developed since the 19th century and can nowadays be found in almost any industry with standardized mounting interfaces. Therefore, a series of different and generally mature types of EMs is available to be used for the propulsion of pumps in electro-hydraulic systems. Table 4 provides an overview of EM types, namely permanent-magnet synchronous machine (PMSM), induction machine (IM), switched reluctance machine (SRM), direct current (DC) machine and brushless direct current machine (BLDCM), which are considered in this survey and were frequently reported in electro-hydraulic studies. It should be noticed though, that several of the academic studies cited in this survey were solely focused on the hydraulic domain and mention the type of EM only incidentally or not at all. Thus, giving a detailed overview of which EM is used more often than others is difficult. Furthermore, it cannot be said for sure if an EM type was chosen by a researcher for a specific reason or maybe because it was already available in the lab and the EM characteristics were not essential for the results of the specific study. However, the table shows the general EM characteristics that should be considered to reason out under which conditions specific EM types can be suitable for mobile hydraulic applications and why. Also, it should be noted that power electronics or inverter technologies are not separately discussed in the scope of this survey as the EMs themselves define the main application characteristics.

Table 4
Comparison of EM concepts according to [120,121] ("+" meaning good, "-" meaning bad) and publications in which they were used to propel a hydraulic pump

	PMSM	IM	SRM	DC machine	BLDCM
efficiency	+++	+	+	-	++
price	- -	++	++	+	-
controllability	+	+	-	+++	++
power density	++	-	++	–	+
utilized in	[31,122,15,17,16,123]	[124,47,76,18,19,122]	[125,126]		[112]

4.1.1. Permanent-magnet synchronous machines

Table 4 shows that PMSMs—next to IMs—are most frequently used in electro-hydraulic systems. Moreover, the references include most of the few already commercially available electrified HDMMs [15,17,16]. The dominance of PMSMs has two reasons: Firstly, the permanent magnets require no electric current to create the magnetic rotor field, which results in a high energy efficiency and no heat generation at the rotor of these EMs. Secondly, the magnetic field is very strong—even at a relatively small magnet size, which leads to a high power density and compact EM designs. This is beneficial for HDMMs, which need to be compact themselves and do not offer a lot of space for components. Moreover, especially the zonal actuator concepts discussed in Section 3.6 require compact EMs because they need to be mounted right next to the hydraulic cylinders and, thus, often utilize PMSMs.

Furthermore, PMSMs tend to operate more efficiently at high torques rather than high speeds. For constant-displacement pumps, the torque is proportional to the pump pressure; consequentially, PMSMs operate especially efficient if the HDMM frequently operates with high pressure. Even more optimal would be constant pressure systems, such as shown in [127], in which the pump can operate with constant high pressure/torque but rather low speeds.

Moreover, a distinction can be made between PMSMs that have *interior magnets* mounted in rotor slots and those that have *surface magnets* on the rotor. While the interior-magnet machines have lower material costs [128], the surface-magnet machines offer a significant overloading capability. The short time peak torques can be 4-6 times higher than the rated torque with surface magnets due to low d-axis inductances [129]. Furthermore, better surface cooling reduces the risk of demagnetization due to high temperatures [129]. For HDMMs, this means that surface-magnet machines with lower rated power could be used because many HDMM duty cycles include only short-time high peak loads, which could be sufficiently covered by a smaller EM with high overloading capability. The relatively higher component costs of surface-magnet machines compared to interior-magnet machines could then be compensated by the downsizing effect, and the EM would be more compact as well. However, this requires that hydraulic engineers consider the overloading capability when sizing the electro-hydraulic system, which currently cannot be observed in studies such as [130].

A major downside of PMSMs is the costs of the permanent magnets. For their production, costly rare earths are required. Especially large and high speed PMSMs tend to be uneconomical because of the high magnet costs [131]. Among the references in Table 4, 32 kW is the highest power for a PMSM. In [16] for example, on the same machine, a PMSM is used for the hydraulics but an IM for the driveline, which requires a higher power. At the same time, the price for rare earths tends to vary a lot, and if new deposits are found or research on rare-earth-free magnets succeeds [131], the current high price of PMSM could also decrease.

4.1.2. Induction machines

IMs show lower efficiencies than e.g., PMSMs, especially at low speeds and for small nominal powers. Moreover, their low power factor leads to higher required currents and more expensive inverters [129] that are required to achieve the same torque as other EMs. Still, they can be found in many hydraulic applications as can be seen in Table 4. The applications listed in the table have all generally high power demands in common. An extreme example is Liebherr's electric mining excavator with an 850 kW IM. At these power levels, the advantages of IMs over other EMs are most significant: Especially squirrel-cage IMs have very simple designs and do not require special materials. This leads to generally low prices of IMs also at high nominal power because the machines can easily be scaled up. The price of PMSMs increases more significantly with rising nominal power, and also the efficiency advantage of PMSMs over IMs decreases with rising nominal power.

The best efficiencies of IMs can be achieved at high speeds. Related to hydraulic applications, this makes IMs especially suitable for constant-speed variable-pump-displacement applications such as those discussed in Section 2.1. On the other hand, IMs are forced to work with lower efficiency when they are used for EHAs, which operate at variable and thus frequently low speeds. Also, the low power density makes IMs critical for EHAs, which need to be compact.

Moreover, IMs can also be overloaded. With optimized designs, similarly high overload ratios as for surface-PMSMs can be achieved [129]. Furthermore, for IMs, there is no risk of demagnetizing permanent magnets when overloading the machine. Thus, also IMs offer the potential for downsizing on HDMMs if the overloading capability matches the load profile of the work cycle.

4.1.3. Switched reluctance machines

Table 4 lists only two examples of SRMs used in hydraulic applications, and both are prototypes. Current challenges that make SRMs less suitable than e.g., PMSMs or IMs are complex control due to strong non-linearities, torque ripples and noise emission. At the same time, the efficiency and power density of SRMs are only in the range of IMs [121]. Nevertheless, SRMs are discussed in this survey because they promise unique advantages in combination with hydraulic pumps, which are discussed in the following text. Furthermore, SRMs in servo applications are not a mature technology yet, which leads to the potential for significant future improvements.

The design of SRM does not require expensive rare earths like PMSMs do and it is rather simple as well as robust, which are both reasons mentioned by Nakamura and Sato for choosing an SRM for their hydraulic circuit [125]. Furthermore, the simple design makes them very cost-effective [121] and thus competitive with IMs for high-power levels. Another feature of SRMs that favours high-power applications is that their price increases only slightly for higher nominal torques. This was demonstrated for 34 different commercially available SRMs in [120]. Increasing the EM torque is the best way to increase the power of an electro-hydraulic unit because pumps generally have limited maximum speeds (typically in the range between 2000 and 4000 rpm) but the torque can easily be increased by choosing a larger pump with higher displacement at the same pressure. However, the problems of SRMs mentioned earlier need to be solved first in order to make use of their benefits. These issues could be addressed by advanced control approaches [125,126].

4.1.4. DC machines

The category of DC machines comprises the two main types, *brushed* and *brush-less*. Brushed DC machines are the most mature concept discussed here, relatively cheap and the easiest to control, but they suffer from significant disadvantages as well: the efficiency is rather low and the brushes wear, which leads to frequent demand for maintenance. Therefore, brushed DC machines are not suitable for electro-hydraulic applications, which is in line with the fact that no publications of this type could be found by the authors.

The BLDCM, on the other hand, does not show these disadvantages because it actively commutes the electric field by means of power electronics instead of using brushes for passive commutation. In fact, the BLDCM design is very similar to the design of PMSMs. The design involves permanent magnets as well, and the performance is similar too. Still, the difference in the control of both machines makes BLDCMs slightly easier to control, but also causes more torque ripples and noise. As noise is less critical for HDMMs in their typically noisy work environments than for other machines, BLDCMs can be a suitable alternative to PMSMs on HDMM and even slightly cheaper [120]. Furthermore, it is easier to achieve high speeds with BLDCMs than with PMSM due to the simpler control, but currently maximum pump speeds are still rather low and this potential cannot be used yet. These might be reasons why so few references could be found for Table 4. However, because the EM type is often not mentioned in hydraulic studies, there might be already more hydraulic BLDCM applications than Table 4 implies.

4.2. Hydraulic pumps

The interface between the EM as a prime mover and the hydraulic part of the power train is the hydraulic pump. In the scope of this paper, the term *pump* is also used for displacement units that can cover all four power quadrants, meaning they can work as hydraulic motors as well. A variety of pump types can be found in electro-hydraulic systems, as they have different characteristics that can be of different importance for a specific application or that match certain EM types differently well. Table 5 provides a list of standard pumps that can be frequently found in electro-hydraulic systems. The comparison is based on their characteristics, which are described in the following.

4.2.1. Standard variable-displacement pumps

In case of circuits with fixed-displacement pumps, the hydraulic output of the motor pump unit can solely be controlled by changing the pump speed. Variable-displacement pumps, on the other hand, are more expensive due to additional required adjustment mechanisms, but offer, next to the control of the pump speed, an additional "degree of freedom" for the control of the hydraulic output. In case of variable-displacement pumps, uncontrolled EMs can be used, which is common for stationary applications where they can run at grid frequency but inconvenient for HDMMs, which typically do not have a grid connection. The only HDMM applications with a variable-displacement pump in an EHA configuration that could be found are a fork-lift [82], an excavator [50,71] and a backhoe test rig [47].

The degree of freedom can also be used to avoid passing the low-speed region with the EM, at which low natural frequencies of chassis components can be excited and cause so-called structure-born noise. However, noise is commonly less critical for HDMM applications compared to other industries, such as the car industry. Furthermore, varying the displacement can be utilized in addition to the EM control for advanced systems with efficiency-optimized operation points [140] or for downsizing of the EM [138]. But the two references are examples for a stationary tool machine with high and constant power demand and for aerospace applications where weight is especially critical. For typical HDMMs that do not show the same conditions, the additional costs for a variable pump are less acceptable and mainly fixed-displacement pumps could be found in publications dealing with HDMM applications.

4.2.2. Standard fixed-displacement pumps

Looking at the different fixed-displacement pump types, external gear pumps can be identified as most commonly used on conventional

Table 5
Comparison of common pump types that are frequently used in combination with EMs with characteristics according to [46,132] ("+" meaning good and "-" bad) and where they were utilized.

	External gear	Internal gear	Axial piston fixed	Axial piston variable
efficiency	-	+	+	+(+)
price	++	+	-	- -
compactness	+	++	-	- -
speed range	- -	-	+	+(+)
maximum pressures [bar]	≤300	≤300	≤600	
achievable displacements [ccm]	50 to 1000	50 to 700	500 to 3000	
utilized in	[67,48, 51]	[22,133,46, 58,34,68, 113]	[134–136, 113,137,15, 78]	[138,139, 82,71,47]

HDMMs, which is mainly due to the low production costs, robustness and their use for low-power auxiliary functions where efficiency is less important. As electro-hydraulic drives are potentially used for high-power main functions as well, more expensive, but on the other hand, also more efficient internal gear pumps receive more interest and can be found in a large number of research studies, which can be seen in Table 5.

Moreover, Table 5 indicates that both gear-pump types have smaller maximum displacements than axial-piston pumps, which is also due to the typical use of gear pumps for auxiliary functions rather than for main supplies of large HDMMs. For EHAs, the maximum sizes of gear pumps might suffice since the maximum flow is equal to the maximum flow of one cylinder only; however, for centralized electro-hydraulic pump units that are supposed to supply multiple cylinders, it might be necessary to use axial-piston pumps with their higher maximum displacements. Another characteristic of axial-piston pumps that can be seen in Table 5 is their higher pressure capability. This might be beneficial for hydrostatic gearboxes in drive trains, but the hydraulic cylinders in implement systems usually tolerate not more than 250 to 300 bar. Therefore, the high pressure capabilities of axial-piston pumps appear less beneficial for electro-hydraulic implement systems and lowering the maximum pressures of those units represents a potential for cost reductions.

Furthermore, the compact design of gear pumps, especially the internal type, facilitates stacking and integrated designs for compact electro-hydraulic units. A downside of gear pumps in combination with speed-controlled EMs is their minimum speed limits that also limit the operation range of the EM. The reason for the limitation is that journal bearings are most commonly used, which achieve sufficient, wear-free lubrication only above at certain minimum speed of typically 200 to 600 rpm [141]. To overcome this issue, less common designs with roller-bearings could be used and lubrication interfaces should be improved.

Another solution that is shown in [81,48,142] and depicted in Fig. 10 uses a proportional bypass valve in parallel to the pump. Without the bypass valve, the pump must spin below the recommended speed limit n_{lim} in order to obtain the low cylinder speed v_{low} (Fig. 10a). If the bypass valve is used and opened to add pump "leakage" (Fig. 10b), v_{low} is achieved while the pump can spin above n_{lim}. In [141], Grønkær et al. used a similar setup and considered also the pressure influence on the minimum speed limit in the control. At the same time, these techniques increase the energy losses [61,141], but gear pumps tend to work with very low efficiencies at low speeds anyway. However, in other studies that utilize gear pumps, the minimum speed limit is simply ignored, but no long term tests for those setups have been shown, and pump manufacturers explicitly warn about going below the limit in speed-controlled operations as this damages the pump [143]. Fixed displacement axial piston pumps, as an alternative to gear pumps, solve the problem of minimum speed limits since roller bearings are usually utilized. Also, the efficiency is generally higher. On the downside, those pumps are significantly more expensive due to the high number of moving parts.

4.2.3. Integrated pumps

Next to the option of flanging standard pumps to standard EMs, integrated designs that combine EM and pump in one housing can also be seen in patents [144,134,145] and research [146,136]. Advantages of the integration are compactness, a lighter housing, a common bearing of pump and EM, less installation effort, as well as precise alignment and balance of rotating inertia. However, the integration requires additional design and manufacturing effort compared to using standard components. This can only be economic if a design is manufactured on a large scale. Since HDMMs require actuators at various power levels, the quantity of EM-pump units that would be required for each power level, especially the higher ones, is rather low, and it is unlikely that integrated EM-pump units will be economic for a wide range of HDMMs.

Another integration option that can be achieved even with standard components is combined cooling of the EM and the hydraulic circuit. A setup in which the EM is cooled by oil coming from the hydraulic circuit was analyzed by Ponomarev et al. in simulations as well as experiments [136]. That way, electro-hydraulic systems can be realized with only a single cooler instead of two separate ones. Still, in this matter, several questions arise:

- Is hydraulic oil as a coolant as suitable for cooling as conventional coolants?
- Is there a risk for some of the advanced additives in hydraulic fluids to degrade when they pass the gap between stator and rotor, which shows strong magnetic fields and high temperature?
- Are there problems with metallic particles in the hydraulic oil—rising iron particle contamination was reported by Michel and Weber for a closed-circuit application without filter at endurance tests [85]—that attach to permanent magnets inside the EM?

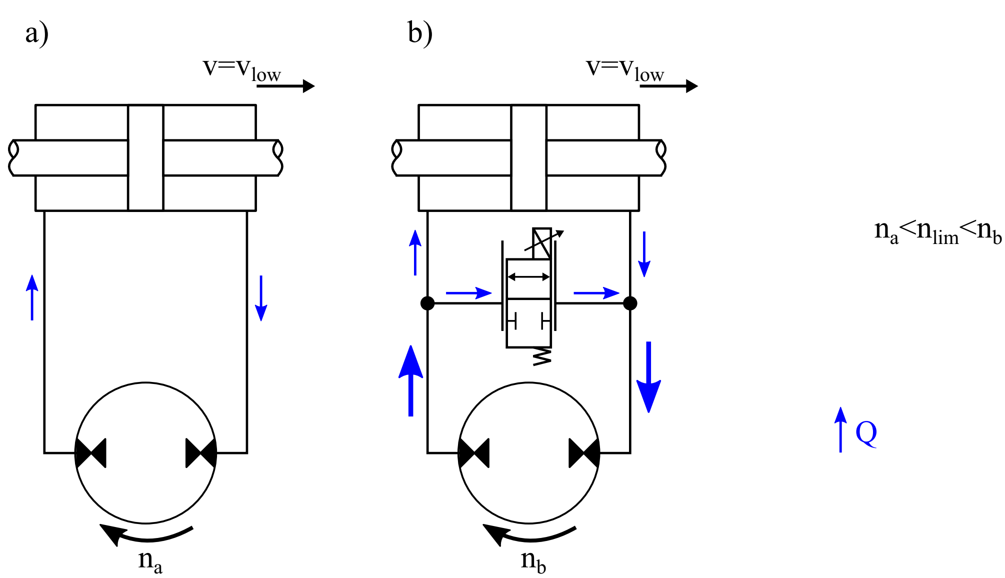


Fig. 10. An EHA a) without and b) with bypass valve, which allows to achieve low cylinder speeds at high pump speeds.

As no long term tests or commercial applications of this common cooling are known to the authors, these questions cannot be answered yet.

4.2.4. Pump designs optimized for electric prime movers

There are several approaches to optimize pump designs for electro-hydraulic applications or even to develop completely new designs. Most of them are not commercialized yet; still, they are discussed here since some of them might represent a key to a successful penetration of electro-hydraulic systems into the HDMM market.

In [35], Pietrzyk et al. show the design of an internal gear pump that can work at up to 10,000 rpm, which is more than double the maximum speed of most conventional pumps. Such high speeds make it possible to use smaller EMs for the same power output, because the size of most EMs mainly depends on the nominal torque. This leads to more compact designs and cheaper EMs. On the other hand, the pump proposed in [35] cannot switch its high- and low-pressure sides, which limits the potential applications.

Not only the maximum speed limit, but also the efficiencies at low speeds are increased by the Innas floating-cup pump design. By means of improved sealing interfaces, next to other innovative design features, high lubrication and high efficiencies can be achieved at very low speeds [147], which is especially beneficial for the speed-variable EHAs. Other conventional pump designs are still optimized for ICE applications that show speeds around 2000 rpm for most of the time. If more manufacturers focus on low-speed as well as very-high-speed conditions, electro-hydraulic systems will benefit significantly.

Another feature that offers potential for improvement is the maximum pressure limit of the low-pressure or drain ports of a pump. The limit is defined by the pump shaft sealing and should be relatively easy to increase. An increased low-pressure limit leads to more freedom in the sizing process of closed-circuit EHAs as it was discussed in Section 3.2.1.

Solely designed for EHA applications are three-port pumps [148–150]. A single three-port pump can be used in the same way as the two pumps in the B.I configuration of Section 3.2. The advantage is that generally even a more complex single pump is cheaper to manufacture and more efficient than two pumps. Still, the flow ratio of a three-port pump can only match a single cylinder area ratio, and to cover a wide range of cylinders with different area ratios and power levels, a high number of different three-port pumps would be necessary. Manufacturing that many different pumps is much less economic. By chance, the design might still be successful for a few specific power levels and area ratios that are most common.

In contrast to using hydrostatic pumps, Lee et al. proposed a hydrodynamic pump especially designed for the use with EMs [151]. The pump was simulated at up to 10000 rpm which leads again to the possibility of using smaller EMs. A further advantage over hydrostatic pumps is that the pump produces almost no pressure ripples. The special design, which is similar to those of journal bearings, leads to higher maximum output pressures compared to conventional hydrodynamic pumps and makes it more competitive besides hydrostatic pumps. On the other hand, the speed-flow relation of hydro-dynamic pumps is pressure dependent, which would complicate the control for most applications.

4.3. Mechanical transmission elements between electric machine and pump

In the previous sections, only the direct mechanical connection between pumps and EMs have been considered. However, the option of adding a mechanical speed reduction stage should not been neglected since this offers the potential for choosing smaller EMs that can operate at higher speeds but lower torques. Even if pumps are optimized for high speed operation, the maximum pump speed will always remain significantly lower than the achievable speeds of EMs due to cavitation issues at high pump speeds. This was demonstrated in [35] where the high-speed pump had a maximum speed of 10,000 rpm and a planetary gear box was used to operate the EM with up to 30,000 rpm for the sake of compactness. On the downside, the authors mention the high costs for such gearboxes which might be the reason why mechanical speed reduction can rarely be seen in electro-hydraulic publications. However, if a power-split gearbox is required anyways for connecting multiple pumps to one EM, as it is typical for conventional ICE-based systems of larger HDMMs, adding a speed reduction stage to that existing gear box should proof less expensive. An example is presented in [116], where a T-gearbox is required for distributing the EM power between the two pumps of the EHA; this gearbox also performs a speed reduction with the factor 1.5.

4.4. Linear electro-hydraulic pump units

The concepts discussed in the previous sections are based on transferring linear electromagnetic forces into rotational mechanical torque inside the EM, which is then used to drive the pump. At least in the case of piston pumps, the torque is then transferred back into a linear force, which pressurizes the fluid in the pressure chamber. Contrary to this, the idea behind linear pump concepts is to skip the rotational stage and directly use a linear electromagnetic force to drive a pump piston and pressurize the fluid. This concept is not commercially utilized yet, but still discussed here because of its potential: Due to a more direct transformation of electric power to hydraulic power at the pump, linear electro-hydraulic pump units promise to be much more compact than rotational concepts [152] and more efficient. Both are features that are desirable for HDMM applications. The arising challenge is to achieve an efficient and continuous operation with an oscillating linear pump and linear electric prime mover as well as sufficiently high power levels.

Similarly to rotational EMs, linear electromagnetic prime movers have further subtypes such as permanent-magnet, induction and switched reluctance. Compared to concepts that utilize piezoelectric, magnetostrictive or electrostrictive effects [153], linear electromagnetic prime movers offer the potential for larger displacements and thus power. Nevertheless, most existing concepts of this type are focused on low-power applications such as compact hand-held devices [152]. Still, as this technology is not mature yet, the authors see the potential for increasing power levels, which would allow to actuate at least low-power auxiliary actuators on HDMMs in a very compact and efficient way.

The general design of linear pump units makes use of springs to center the pump piston, which is also the mover of the electromagnetic prime mover. This can be seen in Fig. 11 for both units. The electromagnetic prime mover is then actuated in order to let the piston oscillate. In [152], for example, Leati et al. chose a frequency of 300 Hz. Next, the alternating output flow of such a single-piston pump needs to be rectified in order to achieve a one-directional output flow out of the alternating pump flow.

4.4.1. Rectification of alternating hydraulic flow

The first rectification option is depicted in Fig. 11a and uses passively actuated check valves, which are arranged similar to diodes in an electric AC-DC-converter. Vael and Achten show how the implements of a forklift truck can be supplied by a pump with a similar valve arrangement, which is driven by combustion instead of a linear electromagnetic prime mover [26]. If bi-directional flow is required, actively actuated valves can be added, which would relate to replacing the check valves in Fig. 11a with active on/off-valves. A third option was presented in [154, 84] alongside a prototype and can be seen in Fig. 11b. A valve spool is directly connected to the pump piston and directs the flow of a second linear pump-motor unit of the same design. The spool valves rectify the alternating flows, and the resulting flow behavior can be expressed as

$$Q_{\mathrm{Pmp}} = 8\mathrm{f}A_{\mathrm{Pmp}}S_m\sin(\phi), \tag{7}$$

where Q_{Pmp} is the pump flow, f the frequency (preferably the eigenfrequency), A_{Pmp} the pump piston area, S_m the motor stroke and ϕ the phase difference between the coupled pump-motor units, according to Li et al.

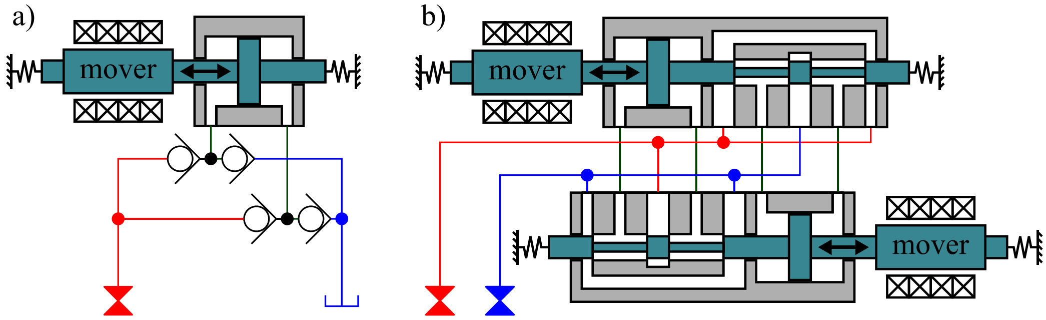


Fig. 11. Linear electro-hydraulic pumps with flow rectification by a) passive valves according to [152], b) a tandem unit according to [84].

[84]. Consequently, the amount of flow can be controlled by changing the stroke amplitude, and the direction can be changed by controlling a phase difference switch from $\pi/2$ to $-\pi/2$ or vise versa. This theoretically allows to use such pumps even in closed-circuit EHAs.

4.4.2. Feasibility of achieving high power levels

The prototype of a recent linear electro-hydraulic pump design presented by Leati et al. [152] showed total efficiencies up to 80%, high power densities and operating pressures up to 250 bar, but the maximum flow was limited to 1.3 l/min. To be a feasible solution for HDMM, the maximum flow needs to increase.

One option is to increase the oscillation frequency, but this would lead to higher compression losses during the flow rectification especially if the dead volumes increase with larger piston sizes. The company Artemis managed to decrease compression losses in their digital displacement pumps by smart control of the active valves that are connected to each piston in their pump [155]. The active valves in such digital pumps are very similar to rectifying valves of linear pumps, and the same approaches for reducing compression losses could be adopted.

The other options to increase the flow is increasing the piston stroke or choosing larger piston diameters. Both options lead to higher required forces for the acceleration or for withstanding a higher pressure force. For larger pistons, also the higher oscillating mass can be critical, which was reported in [156,152]. Moreover, multiple units could be combined to achieve higher flows, but it is hard to predict if this, or the linear electro-hydraulic pump concepts in general, can be economic. All in all, there might be several challenges but also different options to achieve an increase in flow and thus in the power of these units. However, more research interest is required in order to achieve significant improvements and obtain suitable concepts for HDMMs.

5. Supply of electric energy

Even the most sophisticated electro-hydraulic circuit solution can only be applied if a supply of electric energy for its EM(s) is established. Therefore, this section analyses ways of supplying and distributing electric energy on HDMMs that are already established or, in the authors' opinion, promising for the future. This analysis is performed considering the requirements coming from the previously discussed electro-hydraulic circuits for implements.

To start with, Table 6 lists the available technologies for the primary supply of electric energy and their main properties. Also, the properties of an ideal energy source for electro-hydraulic HDMMs are listed here as a reference for the assessment of the available options:

- a high capacity for long operation times,
- a high maximum power output for productivity,
- a low weight and volume for compact HDMM designs,
- low emissions, allowing to work in cities, indoor areas and nearby animals,
- recuperation capability to make use of load braking,
- robustness,
- an electric output compatible with the EM inverters,
- a low price,
- sustainability concerning resources and recycling.

From Table 6, it can be seen that each of the primary energy sources has at least one downside compared to the ideal energy source. It depends on the application which of these downsides are more acceptable than others, or if additional secondary energy storage components in combination with the primary energy source are necessary to form a suitable energy source. In this context a secondary—in contrast to a primary—energy storage is defined as a storage that is used for power management and typically not for fueling the vehicle. This is further explained and analyzed by looking at the energy requirements and methods for energy management.

5.1. Degree of electrification and terminology

Before the combinations of supply solutions and their management is discussed in the next section, the common terminology that is used in this context is explained. This terminology is adapted from the automotive field and can be studied in more detail in [162]. A first distinction is made between the nature of the primary energy source; if an ICE is used, the term *hybrid electric vehicle* is used, while the other primary energy sources in Table 6 are used for simply *electric vehicles*. Hybrid electric vehicles can further be divided into *parallel* or *series* hybrids. As can be seen in Fig. 12, parallel hybrids offer a direct connection between mechanical drive output and EM as well as ICE. This type of hybrid can work with smaller ICEs and EMs as they support each other, but it is less interesting in the scope of this paper and thus not further considered since the EM must operate in the same speed range and manner as the ICE, and the hydraulic systems that are attached cannot profit from the advanced control capabilities of EMs. In serial hybrids, on the other

Table 6
Options of primary electric energy supplies.

	Electro-chemical batteries	ICE + electric generator	Fuel cell	Electric cable
energy density	medium to low	high	high	∞
power density	low to medium	high	low	high
local emissions	non	critical, mainly CO_2	only water	non
can recuperate	yes	no	no	(yes)
price	medium to high	low	high	depending on infrastructure
temperature sensitive	yes	no	no	no
HDMM examples	[11,15,17,16]	[157,158]	[159, 160]	[18,19,161]

hand, the ICE is only connected to an electric generator and the EM is transmitting all the power to the mechanical drive output.

A further term that is used in connection to hybrid vehicles is the *degree of electrification* or *degree of hybridization* respectively. It describes the ratio between the power of the EM to the combined power of EM and ICE together [162]. With increasing degree of electrification, also different names are used: *Micro-hybrids* use recuperative braking but only utilize the electric energy for auxiliary functions; *mild-hybrids* use the EM to support the ICE, and *full-hybrids* have enough EM power to drive the system without the mechanical support of the ICE. If a full hybrid has a secondary energy storage it can operate for some time without the ICE running, and if that secondary storage can be charged by cable it is also called *plug-in hybrid*. Accordingly, for electro-hydraulic systems, which should utilize series structures, only the terms full and plug-in hybrid can apply. A detailed review of all hybridization aspects of construction machinery—a representative type of HDMMs—can be found in [163]; the following text will only add to this by putting a focus on the implications that having two types of energy storage solutions have on electrified hydraulic implements.

5.2. Energy management and recuperation

When selecting and sizing an electric supply, it is important to consider that the required energy supply rate—meaning power—typically varies significantly between different HDMM types. This is caused by the cyclic nature of most HDMM work tasks and other operation factors. For the loading cycle of an excavator as an example, the power requirement is high during lifting, but no power might be required at all to lower the boom. Thus, sizing the primary energy source with the goal to meet the peak power requirement can lead to immense sizes especially for technologies with low power density such as most electro-chemical batteries.

Instead, adding a power-dense secondary energy storage to the system enables power management strategies that are otherwise only possible if the primary energy supply is able to recuperate and has a sufficient power density. Thus, most supplies can be made more efficient and compact by means of adding a secondary energy storage, which is the basic principle of hybrid vehicles—but can also be applied to fully electric systems—and is a broad field of research. As a general result, power peaks in the duty cycle can be covered by the secondary energy storage, and during low power phases, it can be recharged, which results in lower and more constant power demands for the primary energy source that can be downsized or work more efficiently. This effect is visualized in Fig. 13a and 13b by the orange power arrows. Furthermore, a certain amount of braking energy can now be stored in the secondary storage also for systems with generator or fuel-cell as primary energy source. This effect is depicted in Fig. 13b by the green power arrows. For cable powered machines, it can also be beneficial to avoid using power electronics for feeding braking energy back into the grid.

For the secondary energy storage a high power density is most important while energy density is less relevant because of typically short cycle times and thus small amounts of braking energy per cycle. Thus, electro-chemical batteries are less suitable as secondary energy storage and should rather be considered as primary sources only. Hydraulic accumulators and electric supercapacitors, on the other hand, are preferable due to their high power densities. Table 7 shows a qualitative comparison of the two options next to brake resistors, which fulfill a similar function, and electro-chemical batteries. The background information is given in the following sections.

5.2.1. Hydraulic accumulators

Hydraulic accumulators have the advantage of easy integration into the already existing hydraulic circuit. Thus, their energy management does not involve the EM in the electro-hydraulic system. Furthermore, less conversion losses appear, and the EM as well as the inverter see less power peaks and can be downsized—for certain applications, up to 70% as shown by Padovani et al. [130]. On the other hand, the power management requires extra hydraulic components or special architectures in the hydraulic circuit because the accumulator charge/discharge pressure is not flexible but depends on the charge state. In [130] for example, an additional pump and valves are required, and in [103], constant pressure rails need to be present to couple the hydraulic accumulator. Moreover, the accumulator can only be used to manage the power of actuators in the same circuit. For zonal systems with multiple small hydraulic circuits, a high number of accumulators would be required, which can be costly.

5.2.2. Supercapacitors

Facing the just mentioned disadvantages of hydraulic accumulators, supercapacitors seem to be the preferable solution, especially because electro-hydraulic systems already require an electric interface. In the case of a constant-voltage DC-grid, the connection requires minimal effort. Furthermore, only one supercapacitor bank is necessary for the whole machine, because even zonal electro-hydraulic circuits are all

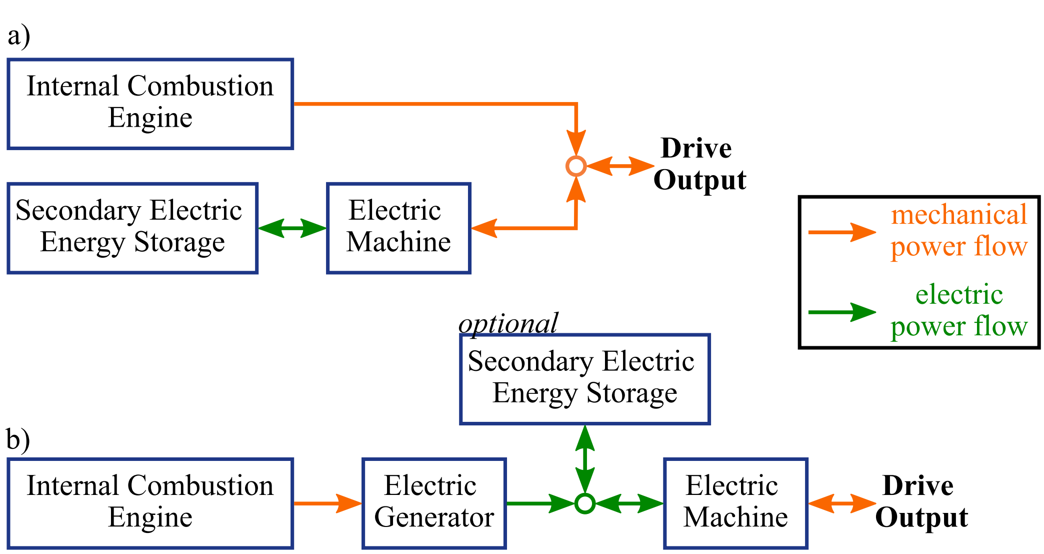


Fig. 12. Both types of diesel-electric hybrids, a) parallel hybrids and b) series hybrid; the drive output could be the drive train of a vehicle or in the scope of this paper one or more pump(s) (power flow back to the ICE due to friction and inertia is neglected).

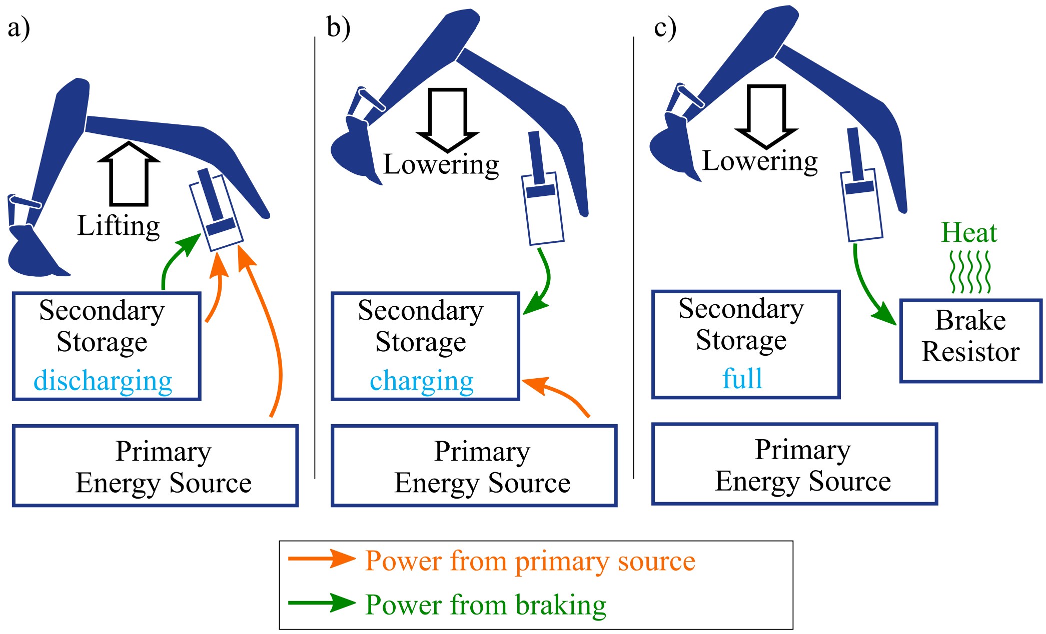


Fig. 13. Power flows in an excavator system with secondary energy storage during a) lifting b) lowering and c) lowering with full storage.

Table 7
Comparison of components that can be used for power management ("+" meaning good and "-" bad).

	Hydraulic accumulators	Super-capacitors	Brake resistors	Electro-chemical batteries
power density	++	+	+++	-
power costs	++	+	+++	-
energy density	- -	-	not applying	+
energy costs	- -	-	not applying	+
ease of integration	- -	+	+	+++

linked by the electric board grid. Leon-Quiroga et al. conducted a detailed comparison study between hydraulic accumulators and supercapacitors by means of experiments under same charge and discharge conditions [164]. Hydraulic accumulators were found to be more power dense and power cost efficient than supercapacitors, but the supercapacitors showed a better energy density as well as costs. The fact that multiple accumulators and additional components might be necessary for a zonal system was not considered in [164]. Thus, the study is also in line with the aforementioned arguments.

In [165], Kwon et al. utilize a supercapacitor bank for an elec tro-hydraulic excavator system in different typologies and investigate the power management strategy. Simulation results in this study demonstrated more than 50% fuel reduction compared to the conventional excavator and a reasonable payback time with the topology that involved fully electrified implements. Moreover, Komatsu offers hybrid excavators with supercapacitors [157] for more than a decade already, but only the excavator swing is electrified.

5.2.3. Brake resistors

Moreover, it is worth noticing that if electric braking is utilized for recuperation, also brake resistors must be included in the system for safety. If the energy storage is fully charged or the charge control fails, the brake resistor can dissipate the energy instead, which is depicted in Fig. 13c.

5.3. Energy supply solutions

With the knowledge about energy requirements and management options, the following part reviews already implemented electric energy supply solutions for HDMMs with reference to specific applications.

5.3.1. Battery-based supplies

The field of chemical batteries as mobile sources of electric energy is very wide. Because there is no difference in the principles of batteries for HDMMs and those for passenger vehicles, this paper only reviews battery application aspects that are HDMM-specific. For general information on batteries, the authors refer to reviews of battery-powered passenger vehicles such as [166]. Same as for cars, the by far most suitable commercially available battery type for HDMMs is the lithium-ion battery. The reason is its high energy density, which is likely to increase even more in the future as it is still a rather new technology.

The more mature lead acid batteries cannot reach the same energy density but have a lower price per capacity. Therefore, lead-acid batteries are sometimes used as well if the power demand is very low or the operation time very short such as for an academic excavator demonstrator [11]. However, even with lithium-ion batteries the achievable battery capacity and thus HDMM operation time are limited by three main factors: battery weight, volume and price.

Contrary to cars, weight is only a minor issue for HDMMs in terms of increasing energy consumption because moving work loads is far more energy-consuming. However, weight can be a problem for agricultural or forest machines as the soil compaction will increase. One the other hand, certain machines might even benefit from an increased weight as it can be used to replace necessary counterweights. The patent [167] describes this for an excavator and a conventional starting battery, but the principle can be applied to larger deep-cycle batteries as well. Nevertheless, high battery volumes remain problematic since HDMMs require compactness in order to be mobile. Furthermore, lithium-ion batteries are costly due to expensive resources, complicated recycling and short battery lives (limited number of charge-discharge cycles) that can make a battery replacement necessary before the life time of the

HDMM ends. The resulting increased capital costs are critical in the highly competitive, often short-sighted HDMM market.

Accordingly, the few battery-powered HDMMs that entered the market so far are of small sizes with low energy demands, and operation times are still rather limited. From Table 8 it can be seen that only the wheel loader allows continuous operation for the duration of a typical 8 hour shift. However, the operation time can be even shorter for specific work tasks that require more energy than average tasks, and non-stop two-shift operation or over-hours are impossible. On the other hand, charging during (lunch) breaks can slightly increase the operation time for the rest of the work day.

New approaches are necessary as long as batteries with more capacity are not feasible. Volvo's autonomous electric load carrier HX2 [168] is an example for the replacement of one large machine, in this case a mining truck, with a fleet of smaller machines. This allows to use multiple smaller, already available batteries instead of one large. The enabler of this approach is autonomous machines that do not require additional operator costs. Next to improved productivity, the independent operation of multiple small HDMMs would also allow to charge the HDMMs one-at-a-time in a row while the other HDMMs keep the work process running.

Battery charging itself represents another important issue especially for HDMMs. Next to general limitations such as temperature increase, battery chemistry, and state-of-health management, the charging of HDMMs is limited by the lack of a fast-charger infrastructure or even a standard grid connection in many cases. Without fast-chargers, smaller on-board chargers need to convert the conventional 110 or 220 V AC voltage, which leads to the long charging times shown in Table 8. Even though the battery capacities in Table 8 are still in the range of nowadays' car batteries that can have up to 100kWh, in the future, the charging of larger batteries for larger HDMMs might require higher charging power that cannot be provided by existing fast-charger solutions for cars anymore [169], and customized solutions would be required. However, the price for a fast charger will only be acceptable if many HDMMs can make use of the same charger, which is only possible for very large, long-time construction sites or mines. At the same time, it will be likely that many machines have to charge at the same time, during a break or after a shift, which would make using the same charger impossible—unless machines are autonomous and can plan their brakes more independently. Furthermore, the load on the grid can be critical when many large batteries are charged at the same time.

The concept of battery swapping, where the whole battery is exchanged instead of charged while remaining in the vehicle, seems promising for solving many of the aforementioned problems. For cars, the concept was recently adopted on a large scale in eastern parts of China [170], but prices up to 500,000 US dollars [170] were mentioned for the large autonomous swapping stations. For HDMMs that operate in restricted areas and do not necessarily require fully automated swapping stations, the technology can be more simple and costs are lower. Swapping systems are already available for forklift trucks, but the same systems would fail in outdoor environments and for less accessible HDMMs. Furthermore, all machines need to be designed for battery swapping—in the best case with standardized interfaces—from the beginning. Mines are sites with many and similar machines working at the same place for a long time, which are beneficial conditions for battery swapping. Accordingly, first battery swapping patents were assigned and are currently applied to mining applications: Epiroc disclosed a battery pack and the method to change it with an external mechanism [171], while Artisan, nowadays a business unit of Sandvik, proposed an automated on-board battery swapping mechanism [172]. Because no external swapping facilities are required, the latter case might also be a viable solution for non-mining HDMMs that frequently change their work site, work very remotely or operate without many other machines around, which could share the swapping facilities. On the other hand, using an on-board swapping mechanism increases the costs of each machine significantly.

Table 8
Lithium-ion-battery specifications of available HDMMs.

	Capacity [kWh]	Operation time [h]	Charging time [h]
excavator 1.9 t [15]	20	5	8 (220V), 12 (110V)
excavator 2.7 t [17]	20	4	no information given
wheel loader 5 t [16]	39	8	12 (220V 16A), 20 (220V 8A), 2 (fast charging)

A further issue is that lithium-ion batteries, but also many other battery types, show significantly decreased performance at low temperatures. Especially for cold starts of a vehicle, this is problematic. Because HDMMs can potentially be used in very cold regions but often not parked inside, this can make the utilization of conventional batteries in these cases impossible. Another risk in harsh HDMM environments is damaging the battery. In case of conventional lithium-ion batteries, even slight damage can lead to fire or explosion [166].

Despite all this, even bigger battery-powered HDMMs can be a successful solution if the lithium-ion technology improves further or new battery types are developed and business models for the right infrastructure are created. A couple of promising, improved battery concepts already exist:

- Flow batteries can work as conventional batteries, but the two reacting chemical components are both liquid, which allows to recharge the batteries rapidly by draining the battery tanks and refilling them with charged fluid. Currently, the energy density of flow batteries is still very low and the energy management challenging [173], which is why no commercial mobile applications exist so far.
- Lithium-air batteries work with oxygen from ambient air to oxidize lithium which leads to up to 30 times higher power densities compared to lithium-ion batteries [174], which is almost as high as the power density of gasoline. Research on this battery type started more than a decade ago and targets, among other objectives, to increase the power density and to prevent other air components than oxygen to enter the process, which would decrease the life time of the battery [175].
- A more application-ready solution combines mature sodium-nickel chloride (ZEBRA) batteries with supercapacitors. The battery alone achieves energy densities similar to those of lithium-ion batteries at a much lower price but does not reach the same power density [176]. Regardless, the supercapacitor bank connected to the battery with a DC/DC converter can be used as a power buffer to make up for the low power density of the battery. Furthermore, the supercapacitor bank reduces the stress on the battery and thus increases its life time as it is described in [177].
- Similarly, the company Geyser Batteries presented a battery that is, from a chemical perspective, a hybrid of battery and supercapacitor with high energy and power density [178]. Furthermore, the battery does not require harmful or rare chemicals and is especially intended for heavy-duty applications. According to the company, sample batteries were already delivered to customers.

5.3.2. Generator-based supplies

ICE-generator-based supplies—relating to what is also called full hybrid and is depicted in Fig. 12,—are a mobile and mature solution for supplying high amounts of energy with convenient refuel capability. Thus it is the only option for the electrification of large machines with high mobility requirements so far. However, zero local emissions, a common advantage of other electrification approaches, is not given in this case. Still, the increased flexibility in control, which leads to higher efficiencies, and in the power train structure are arguments that led to the application especially on very large machines: In marine vessels, such as cruise ships, ferries or cargo ships, turbines or diesel engines are used to produce electric energy, and the absence of mechanical shaft lines leads to layout flexibility, increased space and lower costs—arguments that can be applied to very large hydraulic HDMMs as well, considering that long hydraulic lines can be replaced by electric cables. Another example are diesel electric mining trucks such as Liebherr's T 236 with 895 kW power [179] or Caterpillar's 795F AC with 2500 kW power [158], that is 10% more fuel-efficient than a conventional version of the truck. The electric energy is only generated for the driving function on these trucks, but the advantages in productivity and efficiency arise from the decoupling between ICE and actuators, which would also be applicable to implement functions.

The aforementioned machines include no electric energy storage components because the electrified functions require rather constant and positive power only. For large machines that work cyclic with varying loads and energy recuperation potential, hybrid concepts with storage can improve the efficiency further and allow to downsize the engine. As this paper deals with hydraulic implements that have only electric prime movers, serial hybrid structures [180] can be considered only.

Excavators can be found with ICEs in a serial-hybrid topology with supercapacitors: in [181], this was investigated, and in [157], it was commercialized but with electrified non-hydraulic swing only; in [165, 182], it was also investigated for hydraulic implements. Supercapacitors are used in these systems due to their high power density and because only small amounts of energy need to be stored from cycle to cycle. As excavators are representative for a high number of other HDMMs, this concept can potentially be applied to those as well.

5.3.3. Fuel-cell-based supplies

Because fuel-cells are not a widely adapted technology yet, component costs are currently rather high, and no large-scale applications in HDMMs could be found by the authors. Still, certain industries are starting to change: Forklift trucks, for example, often work indoors where emissions of an ICE are not acceptable, and long charging times as well as low energy capacities also make batteries problematic. Therefore, Linde Material Handling is developing fuel-cell forklift trucks [160], which can be refilled with hydrogen in only 3 minutes. The fuel-cells are combined with lithium-ion batteries, which act as the previously described secondary energy storage to cover power peaks and recuperate braking energy. It is likely that lithium-ion-batteries instead of hydraulic accumulators are used because their higher energy density allows to store more energy and let a smaller fuel-cell operate with a more constant output. Whether the fork lift implements remain hydraulic or change to fully electric is not mentioned by Linde. Another example of a fuel-cell HDMM is the yard tractor launched by Terberg this year [159]. The tractor includes electrified hydraulic auxiliary functions as well.

Also applications of larger size are under investigation. The engineering firm Williams Edvanced Engineering announced their cooperation with Anglo America with the goal to power a mining truck with a combination of fuel cell and lithium-ion battery that is capable of supplying 1,000 kWh of electric energy. This project could prove the suitability of fuel cells for large HDMMs, which could previously only be powered by diesel, and pave the way for electro-hydraulic implement solutions on these machines. Another advantage of fuel cells, which should be considered concerning HDMM applications, is their cold weather resistance. In cold environments where batteries might fail, electrified HDMMs could still operate with fuel-cell supplies. All in all, this technology is very promising and could revolutionize the HDMM industry as well as others, but more development is necessary to make the technology affordable, and the necessary infrastructure for supply of hydrogen or other fuel in a large scale is missing as well.

5.3.4. Cable-based supplies

The cable connection represents a direct supply of electric energy without the need for refills and with high transmission efficiencies but makes every HDMM less-mobile. Cable reel systems can improve the mobility with cable length of 300 [18] to 1000 m [161], but require space and advanced handling. The connection is still feasible if a machine moves only slightly and works in environments that have connection to the main grid or their own micro grid. A few already implemented examples of fully or partly hydraulic HDMMs exist:

- Liebherr's cable-powered material handler [19] with 90 kW power is a machine that often works in very limited closed environments such as disposal sites with grid connection.
- Liebherr's electric mining excavators [18] work in limited areas as well and receive up to 850 kW power by cable, which makes even slight efficiency improvements very valuable.
- Takeuchi's mini excavator TB216 [183] is an example for a machine that can operate with an ICE only for high mobility and switch to a cable-powered fully electric mode whenever a grid connection is available.
- John Deere's GridCon project [161] deals with a cable-powered autonomous tractor and proves that also machines with high amounts of driving time can be cable-powered-only, but only in combination with autonomous driving for more precise cable handling. The agriculture environment also offers energy supplies in the form of environmentally-friendly smart grid solutions that involve local biogas plants or wind turbines for example.

The option of feeding back braking energy to the grid is not mentioned in the previously listed examples. Thus, it is not known if the benefit of energy savings can outweigh the additional cost for bi-directional converters, which would be required for feeding back energy into the grid but can be up to two-times more expensive [184].

5.4. Electric power distribution and voltage levels

Transmitting the electric power from the energy supply to the EM inverters is not a trivial task, especially for distributed zonal concepts that can have long distances between power supply and EMs, which need to be bridged by electric cables.

A first decision to make is choosing alternating current (AC) or DC for the board grid of the HDMM. Arguments for AC are its maturity and related to that the existence of many safety standards, as well as the ease of stepping between voltage levels by means of transformers. Nevertheless, other modern applications such as airplanes utilize DC, and also industries such as the mining industry that traditionally used AC grids on a mine and machine level switch to DC grids [184]. Reasons are higher stability, the compatibility with many DC components such as batteries or supercapacitors, no reactive power and thus less losses. In [184], it is demonstrated that with the same cable 43% more power can be transmitted with DC compared to AC without increasing the cable losses. Even for HDMMs with an ICE and generator, which naturally outputs AC, converting the energy to DC for the distribution is beneficial because the EMs on such a machine can use DC/AC inverters in this case. Those inverters require less conversion steps and thus less components than AC/AC inverters and are accordingly cheaper as well as more efficient.

Another design parameter for the power distribution is the voltage level. Considering DC grids, choosing a constant grid voltage should be preferred because it allows to directly connect batteries or other DC components that work at semi-constant voltage. Variable grid voltage, on the other hand, requires DC/DC converters for those components, which increase the costs and losses. An exception are applications in which the DC link voltage is varied in order to improve the switching performance of inverters, which is possible in special cases and was proposed by Liu et al. [185].

When selecting the level of the voltage, two counteracting negative effects have to be considered: On one hand, higher voltages are more dangerous and require extra safety measures. On the other hand, for lower voltages, the cables but also the EMs and other components need to be larger [28]. For the cable size, this can be reconstructed by looking at Equation (8), which expresses the cable losses P_{Loss} as a function of cable current I and cable resistance R_{Cable}. Next, for two cable types, type 1 with DC voltage $V_{\text{DC},1}$ and type 2 with DC voltage $V_{\text{DC},2}$, it is assumed that the line length L, the resistivity ρ and the transmitted power P_{DC} are equal. Furthermore, the goal is to achieve the same cable losses with both types ($P_{\text{Loss},1} = P_{\text{Loss},2}$). According to the assumption and Equation (8), the ratio between the diameter of cable type 1 D_1 and the diameter of cable type 2 D_2 needs to fulfill Equation (9) in order to reach the goal.

$$P_{\text{Loss}} = I^2 \cdot R_{\text{Cable}} = \left(\frac{P_{\text{DC}}}{V_{\text{DC}}}\right)^2 \cdot \frac{4\rho L}{\pi D^2} \tag{8}$$

$$\frac{D_1}{D_2} = \frac{V_{\text{DC},2}}{V_{\text{DC},1}} \tag{9}$$

This means a 48 V DC cable, for example, needs to have a more than 14-times larger diameter than a 700 V DC cable, which is a significant difference. At the same time, it has to be considered that high voltage lines require a thicker isolation [28]. Furthermore, the cable length is also important. For compact machines, the cables are short and larger diameters less problematic, but for large machines and zonal architectures the cables are long and thick cables accordingly more critical. However, if the power is extremely high, the voltage must be chosen very high as well in order to achieve cables and components that can still be handled. Liebherr's 850 kW excavator, for example, is operated with 6000 V [18].

Standards such as EN ISO 16230-1, which is intended for agricultural machines, require neither enclosures, strong isolation or other measures to protect humans from high voltage, which is defined as the range from 75 V to 1500 V DC in this case. Limiting the accessibility can be challenging for zonal concepts like EHAs that require electric components all over the HDMM. Furthermore, maintenance personal requires special training to work with high voltage systems, which is another disadvantage of high voltage levels [9].

Therefore, it is comprehensible that several commercialized electric HDMMs (such as [15,17,16]) have utilized the rather low voltage level of 48 V so far. In [66], it is explicitly mentioned that 48 V have been chosen for safety reasons. Another HDMM example [11] uses 72 V DC for an excavator, which is higher than 48 V but still below 75 V, the limit for high voltage applications in EN ISO 16230-1. Most likely, future HDMMs will stay below that limit as long as the power level is low enough to allow the use of such low voltages.

Another functional safety issue related to the electric power distribution is electromagnetic compatibility [28]. Conventional HDMMs include only few low-power electric components and are not designed to operate close to strong electromagnetic noise. By introducing electro-hydraulic systems, the related power electronics will create this noise on HDMMs. For the future, this has to be considered for the whole machine design. For example, it might be necessary to switch from analog sensors to digital sensors which are more tolerant towards electromagnetic noise.

5.5. Diesel vs. hydrogen vs. electricity as energy medium

In the previous sections only the energy transmission within the vehicle has been discussed but not the way of supplying the energy to the vehicle. In related terms, the tank-to-wheel efficiencies have been discussed but not the well-to-tank processes. A detailed analysis of energy infrastructures is beyond the scope of this paper. Still, it should be noticed that these processes vary a lot between the different energy media—diesel, hydrogen and electricity—that are required by the different energy supply solutions discussed in Section 5.3. Even within the field of electric energy the processes can vary a lot depending where the energy is produced, where it is required and which primary source (fossil fuels, wind, sun, water-power etc.) is used. Those differences will influence the costs, safety and emissions of a system and need to be considered to judge if one solution is more economic, safe or environmentally friendly than another.

6. Conclusion and outlook

This paper reviewed major parts of the research done in the field of electro-hydraulic implement systems for HDMMs from an academic as well as industrial perspective. All required system components were surveyed by taking separate looks on the hydraulic domain with its circuit architectures, the electric domain with its mobile energy supply options and the alternatives of EM and pump combinations, which form the interface between hydraulics and electrics. Most of the findings are at the same time key points that need to be addressed in order to accelerate the market penetration of electro-hydraulic implement systems:

- Simply replacing ICEs with EMs while keeping conventional hydraulic systems is the first, obvious step in electrifying hydraulic implement systems and can already be observed on the market. Selling arguments are zero-emission and low noise.
- The next step, which only happened in academia so far, is utilizing novel EHA concepts that make more use of the strengths of EMs. In addition to zero-emission and low noise, these systems promise significantly reduced energy consumption and improved controllability.
- Challenges for commercial applications of EHAs on HDMMs comprise the handling of large actuator sizes, cooling, filtering and high component costs. Part solutions for these issues were already proposed, but so far no solution could address all issues sufficiently and effectively. However, double-pump EHAs appear more suitable for HDMMs than single-pump EHAs.
- The choice of the EM-pump combination has a significant influence on the efficiency and costs of the whole electro-hydraulic system.
- While PMSMs are mainly used for small and medium power applications, IMs rule at high power levels.
- In most published and commercialized electro-hydraulic systems, standard pumps are used—gear pumps for low costs or axial piston pumps for high efficiencies.
- The key to more efficient and compact systems is redesigning pumps specifically for electro-hydraulic applications. The main goals are integration and achieving higher speed limits as well as wear-free and efficient operation at low speeds.
- Limits of current mobile electric energy supplies are a major obstacle for electro-hydraulic systems on their way of entering the HDMMs market. Today's batteries are only sufficient for low- to medium-power machines and short operation intervals, and ICE-generator supplies lead to slightly more efficient systems, but do not offer zero-emission.
- Battery improvements are urgently required, but different promising approaches can already be reported, including lithium-air batteries and different hybrid technologies.
- Fuel cells are most promising as supplies with high energy density and easy refill capability, but component costs need to decrease and hydrogen infrastructures are required.
- Cable connections should be used wherever the application allows it.
- The combination of different energy supply technologies allows to compensate weaknesses of a single supply technology alone.
- Concerning the power distribution, for the few already commercially available compact machines, mostly 48 V DC board grids can be observed. Still, higher voltages that lead to more safety requirements but also smaller cables are expected and can already be found on HDMMs with higher power requirements.

The authors are convinced that competitive, efficient and cost-effective electro-hydraulic HDMM implement systems can and will be introduced in the future as long as these key aspects and the essence of the numerous research results presented in this paper are considered.

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Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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