The Working Hydraulics of Valve-Controlled Mobile Machines: Classification and Review

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The Working Hydraulics of Valve-Controlled Mobile Machines: Classification and Review

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ABSTRACT

Productivity, reliability, controllability, flexibility and affordable costs represent key aspects in mobile machines. Additionally, due to the high fuel price and to the introduction of stringent emission regulations for diesel engines, the reduction of fuel consumption while persevering the existing performance is the current demand. In order to satisfy and maximize the above requirements, different hydraulic system architectures have been developed during the last decades. Both academia and industry have been investing considerable resources delivering numerous outcomes that require a classification. This review paper closes this gap by analyzing and classifying the working hydraulics of non-hybrid, valve-controlled mobile machines starting from the 1980s to the state-of-the-art. Hydraulic layouts are addressed and categorized by both discussing their fundamentals and evolutions, and by pointing out their pros and cons in a way to provide the readers with a comprehensive overview of the systems currently available on the market and at the research stage.

1. Introduction

Mobile hydraulics is used in many fields such as earth-moving, construction, agriculture, forestry, or aerospace; examples of popular applications are excavators, cranes, wheel loaders, telehandler, and harvesters. Due to the complex operations

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performed by these machines, their controllability is a key aspect along with productivity and reliability. Several hydraulic system architectures characterized by different levels of complexity were developed during the last decades to meet these requirements. Nevertheless, another crucial factor that drives the system design is the cost-effectiveness. This can mainly be achieved through the following measures: by adopting well-known and robust hydraulic circuits largely based on low-cost components, and/or by addressing the energy efficiency to decrease fuel costs over time. The latter characteristic became particularly relevant within the last years because diesel engines power most of the off-road machines. Diesel fuel has reached a high price during the last decade (U.S. Energy Information Administration [1]) therefore fuel consumption of mobile hydraulics has become an economic issue with significant impact on the operating costs. Furthermore, there is an increasing tendency of introducing stringent emission regulations for diesel engines all over the world (e.g., Stage V in Europe and EPA Tier 4 Final in the U.S.A.). Both aspects dictate a continuous strive toward system efficiency improvements. In order to strengthen this point, remarkable examples of low energy efficiency for state-of-the-art systems were presented for hydraulic cranes [2]; the ratio of the actuators' output energy over the pump's input energy ranges between 8.5% for a conventional constant-pressure system and up to 27.3% for a more advanced load-sensing layout. Another load-sensing solution [3] reports an overall efficiency value of about 27%.

Similar cases of poor performance are not unusual but rather frequently seen in many of today's mobile machines. The Oak Ridge National Laboratory quantified the consequences of these inefficiencies focusing on the mobile fluid power in the U.S.A. alone (industrial hydraulics and aerospace are excluded) [4]. At least 0.362 Quadrillion British Thermal Units (Quads) are consumed each year meaning that about 0.076 Quads of work are produced if the estimated average efficiency of 21% is assumed across all applications. The same amount of work could be generated by saving 0.08 Quads if the average efficiency were 27% (this represents the short-term goal for efficiency improvements) or by saving 0.15 Quads if the average efficiency were 37% (long-term goal). With the diesel's energy density equal to 138,700 Btu/gallon and the current cost of diesel fuel assumed as 3.00 US$/gallon [1], these efficiency improvements could save about US$1.74B or US$3.39B per year, respectively.

The substantial savings potential justifies important investments to support new investigations. Several years ago, both academia and industry started the effort to renew hydraulics delivering numerous outcomes that require a classification. Backé traced the development of fluid power [5] and summarized new approaches for efficient hydraulic systems [6] in outdated papers. Murrenhoff provided a limited state-of-the-art review focusing mostly on the propulsion systems of hydraulic vehicles [7]. In another paper, Murrenhoff et al. discussed energy saving architectures centering on concepts but without considering the implementation on real machinery [8]. Hence, an updated and more comprehensive overview on what has been developed is necessary.

According to a general categorization suitable for this literature survey, hydraulic systems pertain to the following three major classes:
1. *Valve-controlled systems*. Hydraulic valves represent the control elements (i.e., flow throttling takes place in the main transmission lines since it is necessary to achieve the control functions).
2. *Valve-less systems*. These throttle-less solutions do not include hydraulic valves in the main transmission lines. The control elements are, therefore, variable-displacement hydraulic units (i.e., displacement control is implemented), variable-speed prime

movers that typically drive fixed-displacement hydraulic units, or hydraulic transformers. Interested readers are referred to [9-10] for overarching discussions about valve-less systems that control single-rod, linear actuators.

3. *Hybrid systems*. These architectures comprise another energy source that could be electric, hydraulic, or mechanical in addition to the machine's combustion engine. Hybridization can be applied to both valve-controlled and valve-less layouts [11] and [12], respectively.

A specific hydraulic system can involve a combination of the technologies listed above even though this approach is not common practice in commercialized applications. In fact, many mobile machines available on the market fit into category #1. For this reason, this paper aims at reviewing and summarizing the most important investigations conducted to improve the performance of mobile fluid power systems dedicated to the working hydraulics of non-hybrid, valve-controlled mobile machines until today. While doing so, considerations concerning general system requirements as well as pros and cons of different design approaches are analyzed and summarized. Additionally, the authors provide a more comprehensive classification of the systems to give the reader a structured overview of the available solutions.

2. The main requirements of the working hydraulics

Mobile hydraulic machines are meant to be the extension of the operator's arms that operate under conditions extremely mutable, hard to predict, and usually involving simultaneous functions, which means that their controllability and their performance are key aspects to consider during the design of a system. Representative examples of multi-actuator machines are depicted in Fig. 1.
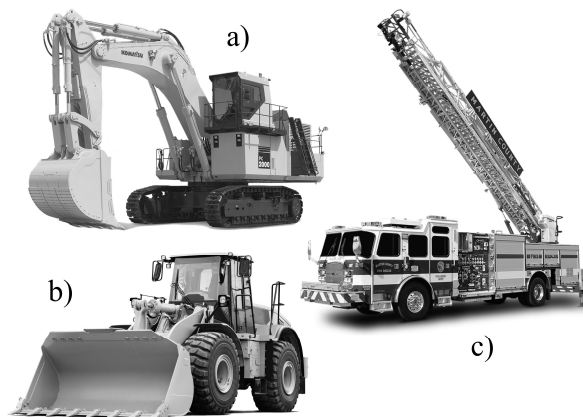

Fig. 1 Examples of multi-actuator machines: **a**) an excavator, **b**) a wheel loader and **c**) a turntable ladder (pictures not to scale).

The machine size and the specific application field set different requirements for the working hydraulics. Typically, the target of the hydraulic systems dedicated to compact machinery with a curb weight below 10 metric tons such as small excavators, backhoes, turntable ladders, harvesters, aerial work platforms, or telehandlers is to guarantee the independent velocity control of each actuator during contemporaneous movements. In medium/large size applications such as wheel-loaders or large excavators, perceiving the load is preferable instead of accurately controlling the actuator velocity. System damping

and reduced tendency to oscillate are common requirements, especially for machines used in the rescue field or to lift/lower significant loads; these characteristics define the operator's "feeling" or the "personality" of the machine [13].

Load-holding capability and overrunning load control are additional necessities that either the directives for machines [14] or the load itself dictate as a requirement in most of the applications; for further explanation see Appendix A. Managing the actuation priority is another necessity found in several machines, mainly concerning the steering sub-subsystem (e.g., in generic load-sensing equipment [15] or in a forestry log skidder [16]) but also regarding other sub-subsystems (e.g., it is advantageous to give priority to the swing motor of an excavator over the stick cylinder [17]). Avoiding the stall of the prime mover is also essential in every machine. This can be done both electronically by manipulating the system's control inputs, or in a mechanical-hydraulic fashion (i.e., power limiters restrict the maximum power absorbed by the pumps through appropriate restraint of the displacement) as discussed in Appendix B. Moreover, the system's overall energy efficiency is becoming a more significant parameter as described in the introduction. In a multi-actuator hydraulic layout characterized by a single pump that supplies actuators at different pressure levels, the throttling losses in the control valves represent a major contribution to the fuel consumption (e.g., a third of the energy supplied by the prime mover of state-of-the-art excavators is lost through throttling in the hydraulic system [18]). Therefore, some applications are conceived with multiple pumps dedicated to distinct parts of the hydraulic circuit in addition to the heavy use of electronic control strategies. This approach is, for instance, commonplace in refuse trucks where one or two pumps are used for the arm while a third one is used for the packer [19]. Furthermore, Skirde and Geerling discussed the trend for mobile hydraulics from the industry perspective showing the clear tendency toward electronic controls, higher automation, and more efficient circuits [20]. Other factors that steer the system design are cost-effectiveness, robustness, minimum downtime, and reduced control complexity.

The importance of all the above-mentioned requirements is application-related. Multiple system architectures were therefore developed in the past and this paper divides the non-hybrid valve-controlled systems into two major categories, namely the mechanical-hydraulic and the electro-hydraulic solutions (Fig. 2). Further classifications are then proposed depending on the different control concepts.

3. Mechanical-hydraulic systems

The systems belonging to this class entirely rely on pure mechanical-hydraulic regulations, which means electronics is not involved in the control of the system. Examples of the devices used to control these applications are manually-operated control valves of an agricultural tractor [21], or mechanical-hydraulic joysticks of an excavator [22]. Some of these primal hydraulic architectures developed throughout many decades are still commonplace since they typically guarantee robustness, reduced initial investment and simplified maintenance (often accessible to non-specialist workers). More in general, these systems are traditionally divided into open-center and closed-center type depending on the design of the proportional directional control valves (PDCVs), where the definition refers to the nature of the connection established between pump and reservoir when the valves are centered.
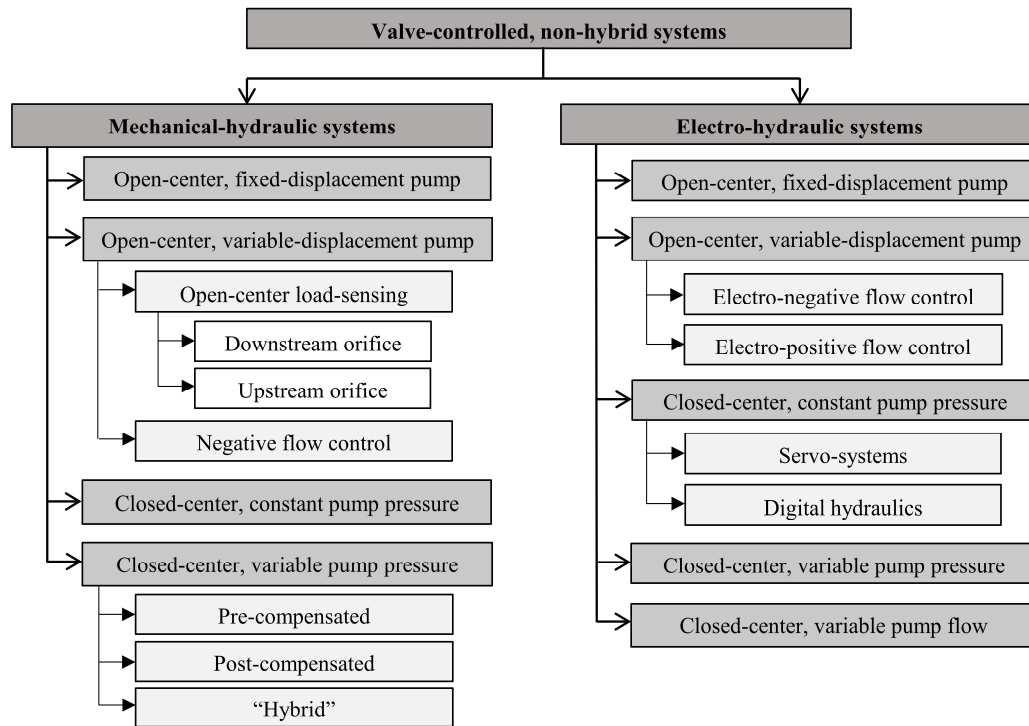
Fig. 2 Proposed classification of the working hydraulics for non-hybrid, valve-controlled mobile machines.

3.1 Open-center systems with fixed-displacement pumps

Starting from basic mechanical-hydraulic layouts, several systems make use of open-center architectures with control valves supplied by fixed-displacement pumps. This approach is popular in simple or dated mobile hydraulics (e.g., in wheel loaders [23], or in old-fashioned excavators [24]) due to the system cost-effectiveness and the reduced tuning effort. When the control valves are in neutral, they bleed-off the pump flow to the reservoir due to their open-center design. By shifting the valve spools, the open-center gallery closes progressively directing the pump flow to the actuator's inlet port and the return flow into the reservoir. Different priority can be provided to the actuators depending on the valve arrangement (i.e., parallel, tandem, or series [25]). An example of a parallel connection is shown in Fig. 3 where the two actuators can be driven independently or simultaneously, and all loads have the same priority.
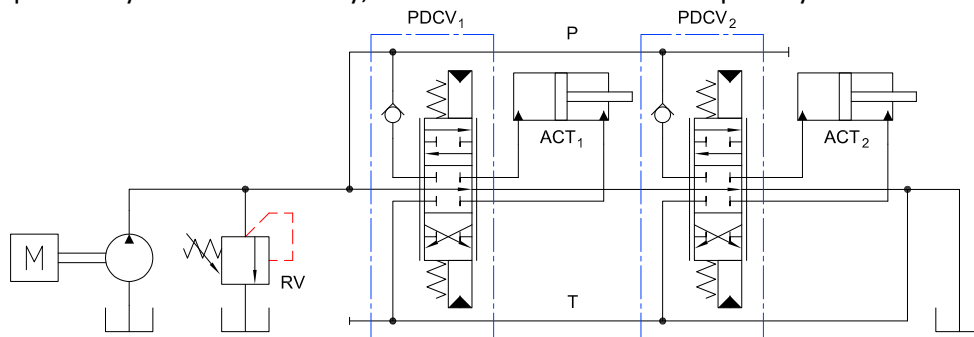

Fig. 3 An open-center system with two actuators connected in parallel (simplified schematic).

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Alternative configurations of the control valves are depicted in Fig. 4. The series arrangement only allows a concomitant actuation of the hydraulic cylinders. The tandem layout permits independent actuation or gives priority to ACT_1 if both control valves are energized.
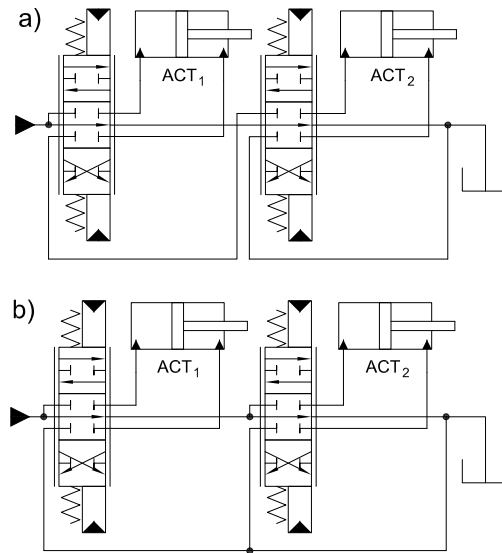

Fig. 4 Different methods to connect the actuators (simplified schematics with load-drop check valves and power supplies omitted): **a**) series; and **b**) tandem.

Open-center circuits are characterized by a very sensitive load dependency. The flow rate directed to the actuator is not only contingent to the control valve's spool position but also to the load pressure, leading to load interaction when multiple actuators are driven simultaneously. This aspect represents the main disadvantage of this system topology and makes precise velocity control of numerous actuators challenging (i.e., poor system controllability). However, the operator has a direct feel of the external load and can directly control the load acceleration by taking advantage of both the system's pressure sensitivity [13] and the high system's natural damping that facilitates the handling of large inertia loads (e.g., the swing function of a mobile crane [26]). These solutions are characterized by relevant energy losses when control valves are not actuated, or when the actuators require only a portion of the entire pump flow. Therefore, variable-displacement pumps can be used to mitigate this energy-related issue.

3.2 Open-center systems with variable-displacement pumps

Combining open-center control valves with variable-displacement pumps leads to hydraulic layouts that are more efficient in many operating conditions; the power losses emphasized in Section 3.1 can be reduced even though they are not completely canceled out. Additionally, the inherent system damping of open-center architectures that enables excellent control of high inertial loads is preserved [13]. Both the nomenclature and the classifications traced in the literature are not consistent, thus the following organization is proposed.

3.2.1 Open-center, load-sensing systems

The well-established "load-sensing" definition applied to these open-center architectures might be confusing because the load-induced pressure is not sensed at all. What is being sensed and controlled is the inlet pressure (or the outlet pressure) of the by-pass orifice, so that the pump displacement is adjusted to keep a constant by-pass flow. This orifice can be located upstream or downstream the PDCVs. Figure 5 depicts the latter option; the changes compared to the open-center layout presented in Fig. 3 concern the introduction of a pressure-compensated pump, the orifice (OR) in the by-pass gallery, and a control valves' spool design characterized by reduced flow areas in the by-pass line connecting the pump delivery to reservoir [13].
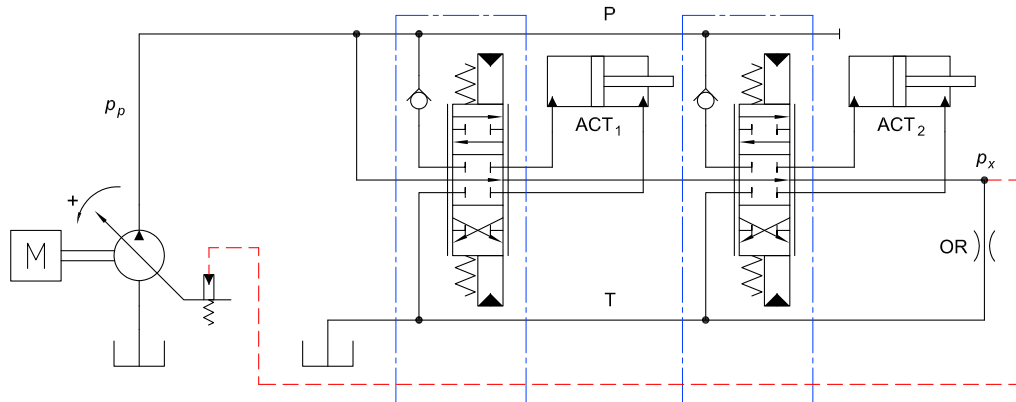

Fig. 5 Simplified schematic of an open-center, load-sensing system where the by-pass orifice is located downstream the proportional directional control valves.

As previously mentioned, the pump aims to maintain a constant control pressure (p_x). When a reduction of the pressure p_x is sensed, the pump displacement then increases to maintain the desired pressure setting. More specifically, the control valves' spool displacements dictate the pump delivery pressure (p_P) for a given load condition since they affect the overall pressure drop across the by-pass flow path. The resulting value of the pump pressure is comparable to the one of the open-center systems based on fixed-displacement units with equivalent external load [27], but the flow rate delivered by the pump matches the quantity requested by the actuators plus the relatively small by-pass term. This specific system configuration is also known as negative, open-center, load-sensing [27] because it exploits pressure-compensated, load-sensing pumps with an inverted control logic (i.e., the pump's displacement setting grows when the pressure p_x decreases).

An alternative arrangement [28-29] that guarantees similar system properties is possible by locating the by-pass orifice upstream the control valves (Fig. 6); this solution is also known as variable-flow, open-center concept [30]. The necessary modifications compared to Fig. 5 refer to open-center valves with properly designed spools and to the pump's displacement adjustment system that decreases the displacement setting when the control pressure p_x is reduced (i.e., conventional pressure-compensated, load-sensing pumps discussed in Section 3.4 are used).
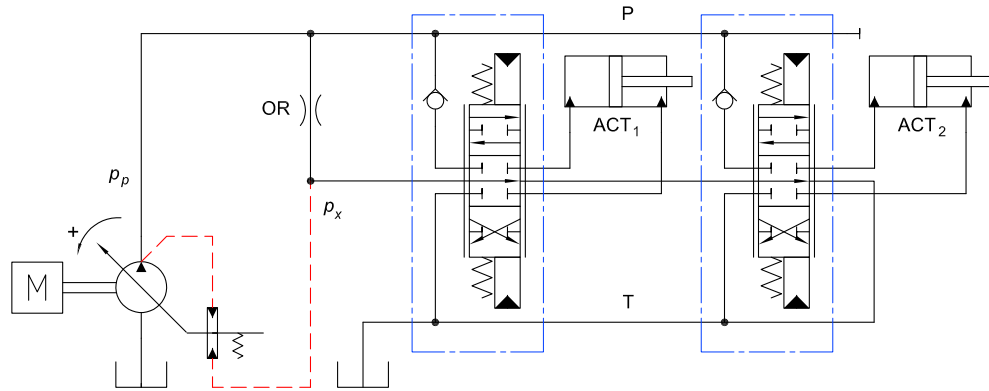
Fig. 6 Simplified schematic of an open-center, load-sensing system where the by-pass orifice is located upstream the proportional directional control valves.

3.2.2 Negative flow control

The solutions defined as negative flow control make use of a system layout almost identical to the one presented in Fig. 5, where the by-pass orifice (OR) is located downstream the PDCVs (e.g., in [31]). However, the pump's displacement adjustment system presents a peculiar design that involves a mechanical feedback [32-33]; the unit is not pressure-compensated since the actual displacement setting depends on the pilot pressure (p_x). In detail, the pump displacement increases as pressure p_x decreases, so that the dynamic response results particularly fast and the designation "negative" is coined. Figure 7 presents the qualitative characteristic of the pump flow as function of the pilot pressure p_x: with the PDCVs in neutral position, the by-pass orifice introduces a moderate pressure drop (e.g., about 40 bar) that destrokes the pump to minimum displacement. As a PDCV is shifted, the pressure drop across the orifice decreases proportionally to the valve command (e.g., down to about 5 bar) upstroking the pump displacement up to full value. In the proposed schematic, the pilot pressure (p_{ser}) used to control the pump displacement is generated by a dedicated charge pump (not shown in figure). However, solutions that take the pilot pressure from the pump outlet are also available.
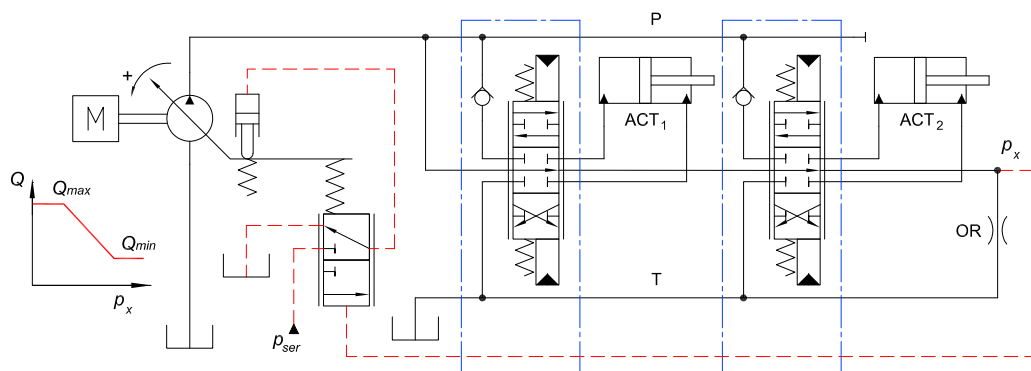

Fig. 7 Simplified hydraulic scheme of a negative flow control architecture with details of the qualitative relationship between the pump flow and its control pressure.

In such hydraulic layouts, when the control valves are energized, the bypass flow decreases with increased pump flow, resulting in an improved system's energy efficiency especially for relevant valve openings. This is a key difference in terms of functioning with respect to the open-center, load-sensing system depicted in Fig. 5. A drawback is instead the higher stand-by pressure and the requirement on the minimum pump displacement (about 15% of the full value [32]) in order to have a proper pilot signal. This makes the stand-by power consumption of the machine slightly higher compared to other system architectures.

Management of the maximum power requested to the engine is often included in this type of system by using additional components that adjust the pump displacement. However, those components do not alter the basic working principle. This architecture is also referred in literature as "open-center, load-sensing system" [33], even though it is conceptually different from either the open-center or the load-sensing systems discussed in section 3.2.1 for the reasons explained before. Negative flow control is a popular technology for mid-size excavators [34].

3.3 Closed-center systems with constant pump pressure

Figure 8 illustrates a closed-center system where the pump displacement is controlled by means of an absolute pressure limiter (also known as "pressure control") to enforce a constant pressure at the pump outlet. An example of such a solution is the working hydraulics of a forestry log skidder [16].
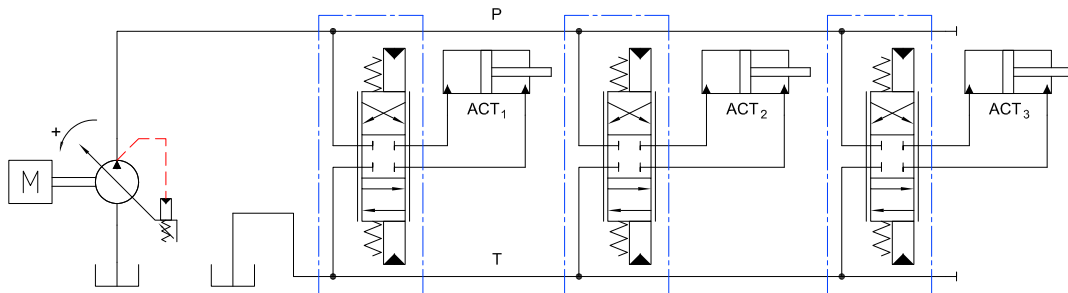
Fig. 8 Simplified schematic of a closed-center architecture for controlling actuators in parallel.

When the control valves are not energized, then the pump delivery port is closed. Therefore, these valves have a simpler design than the open-center counterparts (i.e., the load drop check valves are not strictly necessary, and less spool overlap is sufficient). Most importantly, the system controllability usually improves when benchmarked against open-center architectures for the following reasons [13]: the system damping increases (this is especially true when the valve flow rate is high), there is no load interaction between active functions (the pump delivery pressure is always constant unless flow saturation takes place), and the machine behavior is repeatable for equivalent loads (pressure drops across control valves do not depend on their spool displacements). Lastly, the overall energy efficiency depends on the pressures of the different loads which means that partial loads affect this parameter.

3.4 Closed-center systems with variable pump pressure (load-sensing)

Due to their structure, open-center configurations cause load interaction when more actuators are simultaneously driven. This is not the case for closed-center, load-sensing (LS) architectures that allow independent control of the loads. This feature is usually achieved by means of local pressure compensators (LCs) added to the proportional directional control valves. However, an approach where each actuator is supplied by a dedicated LS pump and the control valve does not have the LC is also possible (e.g., the mixing drum of a cement mixer [24] or the boom and lift cylinders of a reach stacker [35]).

The LCs most commonly used fall into the two following configurations even though alternative techniques discussed in the sequel exist:

1. Pre-compensated systems: the LCs are located upstream of the main spools and are normally open [36], as in the case of Fig. 9;
2. Post-compensated or "flow sharing" systems: the LCs are located downstream of the main spools and are normally closed, as shown in Fig. 10 [37-39].

With reference to Fig. 9 representative of a pre-compensated layout, the pressure at the inlet port of each actuator acts as the opening force on the local compensator with pressure setting $p_{c,pre}$. LCs keep a constant pressure drop across the metering edge of the directional control valves, making the flow rates, and consequently the actuator velocities, function of the operator's inputs only. The global LS pressure (p_{LSG}) is selected by shuttle valves in cascade and is used as input for the pump displacement control. The displacement is then adjusted to impose the pump delivery pressure equal to p_{LSG} increased of a constant margin (i.e., the setting p_s of the differential pressure limiter). The LCs are fed at pump pressure decremented by all possible pressure drops Δp_p in the supply line, typically due to the presence of a priority valve (VPRI) for the hydraulic steering unit. When the PDCVs are not actuated, the LS signal is connected to reservoir so that the pump displacement is set to minimum. A pressure-relief valve (RV) with setting p^*, along with the fixed orifice in the power supply, are used for limiting the pump delivery pressure at $p^* + p_s$.
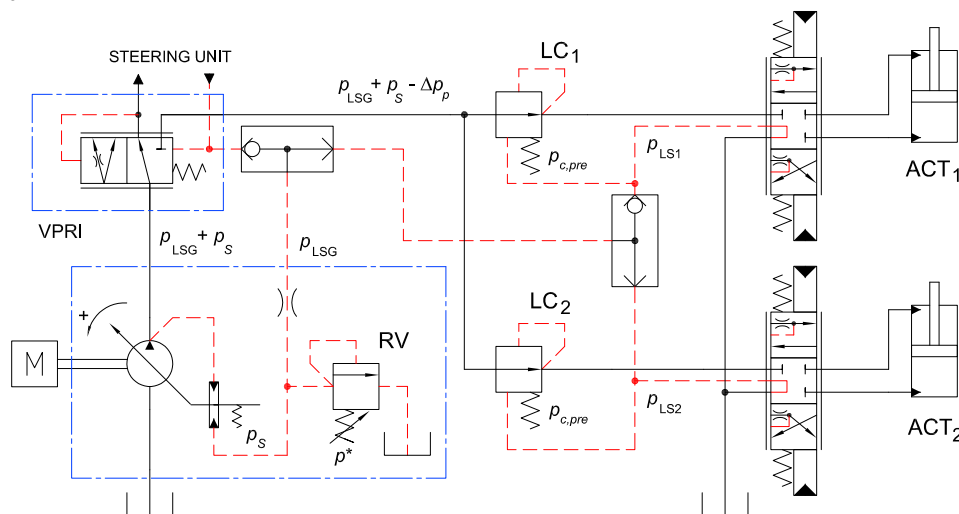

Fig. 9 Simplified schematic of a load-sensing circuit with pre-compensators for controlling two linear actuators and with a priority valve for the machine's steering unit.

With post-compensated technologies (Fig. 10), the same pressure signal p_{LSG} is used for generating a closing force on all LCs. The final aim remains the same since, also in this case, the regulation of the LCs maintains a constant pressure drop across the metering edge of the proportional valves. After passing through the LC, the flow rate is directed to the proper port of the actuator. Figure 10 shows a fixed-displacement pump equipped with a dedicated pressure compensator (PC) known as an "unloading valve". Conventionally, variable-displacement, load-sensing units as depicted in Fig. 9 are implemented due to the improved overall energy efficiency [25].

The reason of these diverse arrangements of the LCs lies in the problems related with flow saturation [40]. This phenomenon occurs if the flow demand is higher than the maximum flow rate of the pump. When it takes place, pre-compensated solutions cannot work properly unless electronic control of the main spool's position is implemented [41]. In fact, the flow rates directed to the actuators diminish in a sequential fashion starting from the highest load as function of the load pressure. On the contrary, post-compensated solutions maintain the same pressure drop across every metering area also during flow saturation. Even though these pressure drops are reduced, the actuator velocities still depend only on the operator's inputs. Therefore, uniquely prioritized circuits can be implemented by combining pre- and post-compensated valves together, where pre-compensated valves have priority. If the reduced actuator velocities are acceptable, then post-compensated architectures can also offer the possibility to decrease the pump size while maintaining adequate system performance [42]. It is also noteworthy that the setting of the LCs is substantially different between pre- and post-compensated solutions. In fact, in the former, the pressure drop across the metering edge of the main spools is equal to the setting $p_{c,pre}$ of the LC. In the latter, it results equal to the difference between the pump margin p_s, decremented by Δp_p, and by the setting $p_{c,post}$. At equal size of the valves, the pressure drop across the main spools must be the same to have equivalent flow rates which means $p_{c,pre} = p_s - \Delta p_p - p_{c,post}$. Representative values are $20 \leq p_s \leq 25$ bar, $7 \leq p_{c,pre} \leq 10$ bar, and $1 \leq p_{c,post} \leq 3$ bar.
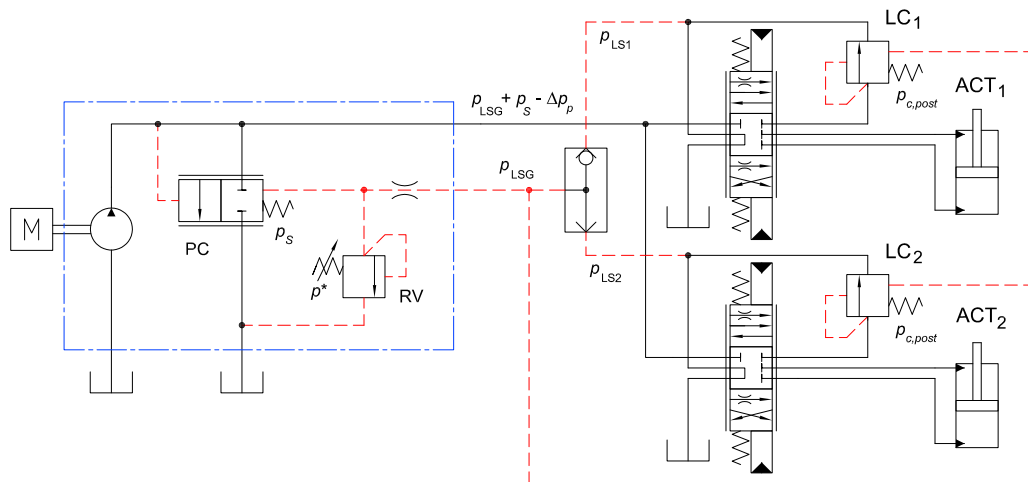

Fig. 10 Simplified schematic of the load-sensing circuit with post-compensators (solution with a fixed-displacement pump).

Variations to the more traditional techniques of implementing pressure compensation have been studied over the years. The most significant examples are the following: an approach that deals successfully with flow saturation [43], a local compensator that operates also as a pressure-relief valve [44], and alternative ways to implement both traditional and "flow-sharing" compensation [26]. Then, a well-known solution is the "hybrid" configuration of the LCs analyzed in [45]. It shares elements of both categories, namely LCs located upstream of the main spools but normally closed and guarantees independent control of the loads when flow saturation takes place [24], [46]. Moreover, a peculiar control valve has been designed based on separate meter-in and meter-out elements that actively use flow forces to accomplish compensation effects [47-48]. This component enables additional features such as meter-in and meter-out pressure control that improves the controllability of high inertia, overrunning loads [13].

In general, diverse LS configurations should not influence the behavior of properly designed systems during normal functioning [49]. However, some alterations arise since the system's "lag time" depends greatly on the LS creation mode. For instance, the "hybrid" LS configuration opts for generating the global LS signal by throttling a modest flow rate from the pump delivery [22] in order to come up with a faster system response. Compared to open-center architectures, systems using pressure-compensated control valves have reduced damping capabilities [13] that make them suitable only to control low inertia loads [27]. Lastly, the energy efficiency of LS circuits is generally good when compared to other valve-controlled systems but still remains problematic when simultaneous functions are being used and the load pressures present relevant differences. In order to avoid load interaction, the actuators at lower pressures require large dissipations across their LCs causing significant energy losses (e.g., a third of the total energy consumed by an excavator in a digging cycle [50]). A possible solution is to group the actuators in two separate sub-circuits based on their power demand, each fed by a LS pump.

4 Electro-hydraulic systems

Mainly due to the increased reliability and to the more affordable cost of electronic sensors and components, mobile hydraulics based on electro-hydraulic solutions started to gain ground in the last decades. In fact, advanced control strategies unfeasible in the past allow more flexible machine setups and/or increased energy efficiency. Some electro-hydraulic architectures are the developments of the mechanical-hydraulic counterparts so that electronics can be involved at different levels (e.g., electronic joysticks actuate the PDCVs to manipulate the control inputs accordingly, or a more massive usage of electro-hydraulic components can be explored). In parallel, new layouts based on different concepts were also studied.

4.1 Open-center systems with fixed-displacement pumps

Some open-center systems supplied by fixed-displacement pumps are equipped with PDCVs controlled electronically; an example is given in [51] where a wheel-loader is addressed. This class of systems represents the improvement of the basic solutions discussed in Section 3.1 because the input commands can be manipulated suitably or feedback control can be implemented (e.g., the lift and tilt angles of a wheel-loader are

monitored). However, the energy efficiency is still the weak point due to the fixed-displacement pump that always delivers full flow.

4.2 Open-center systems with variable-displacement pumps

Introducing electronically-controlled, variable-displacement pumps and where appropriate, electro-hydraulic PDCVs improves both the utilization of the available energy as well as the machine behavior. Such a modification is achieved via the electro-negative flow control (i.e., the electro-hydraulic version of the approach detailed in Section 3.2.2), or by means of the electro-positive flow control. It appears that open-center, load-sensing systems (see Section 3.2.1) are not appropriate as electro-hydraulic solutions since a closed-center arrangement avoids the flow being discharged through the by-pass center.

4.2.1 Electro-negative flow control

Conventional electro-hydraulic versions of the negative flow control require minimal modification of the mechanical-hydraulic system architecture in Fig. 7. A first advantage is about modifying the flow characteristics of the PDCVs by varying the pump's flow rate [52]. Transducers sense the control pressure (p_x) upstream the by-pass orifice, or the Δp across this restriction, so that the pressure signal used to adjust the pump displacement is appropriately created via an electro-hydraulic, pressure-reducing valve. Similarly, improved metering characteristics can be achieved when driving heavy loads by sensing the pilot signals directed to the PDCVs and by controlling the pump displacement electronically [53]. When combining the sensing of the PDCVs' pilot signals and of the pressure p_x, velocity control of specific actuators in excavators improves [54].

Additionally, electro-hydraulic control increases the energy efficiency by forcing the pump to deliver the minimum flow rate when the PDCVs are in neutral (this is not the case when the engine speed falls below the rated value). Pressure sensors on the PDCVs' pilot signals detect the idling condition whilst an electro-valve introduces an increased control pressure that destrokes the pump; during normal operations, the system relays on purely mechanical-hydraulic controls [55].

4.2.2 Electro-positive flow control

Those variable-displacement pumps that involve a mechanical feedback to adjust the displacement setting (i.e., the ones addressed in Section 3.2.2 and Section 4.2.1) can also be controlled so that the displacement increases as the pilot pressure (p_x) increases; this approach is defined as positive flow control. The main motivation behind its introduction relies on the relevant amount of flow wasted in the by-pass gallery by systems based on negative flow control when the actuators do not require full pump flow [56]. Figure 11 depicts an open-center, multi-actuator system grounded on this concept.

The by-pass orifice located at the end of the open-center gallery in negative flow control (Fig. 7) is replaced by a restriction situated in the last control valve [56] or is simply removed [57]. This solution is possible since the control pressure used to adjust the pump displacement is electronically controlled via a proportional valve to increase the pump flow with increased control signal X_P: clearly, the control algorithm must be exactly matched to the specific application and electronic power management is also available. No flow is discharged in the by-pass gallery during operations if the last PDCVs is actuated. An initial movement of the spool closes off the flow directed to the reservoir prior to the

pump delivery port being connected to the actuator. This approach enabled 11.6% energy saving compared to the original layout grounded on negative flow control for a 22 tons excavator [57]. In some systems such as wheel-loaders, it is desirable to provide priority to selected functions and install a smaller and less expensive pump without the need for complicated control algorithms. Thus, the PDCVs are connected in series to create a tilt-priority system that is a standard design in construction equipment [51].
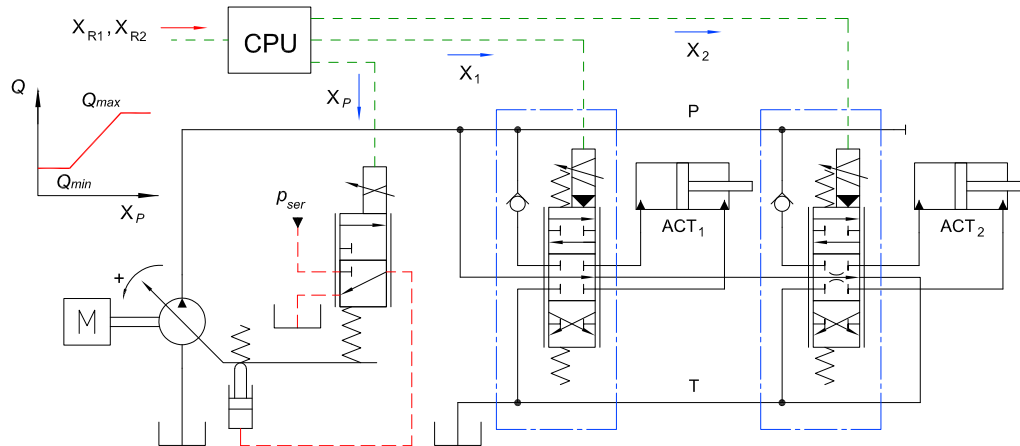

Fig. 11 Simplified schematic of an open-center, electro-positive flow control system.

"Positive" control of the pump displacement is also possible in a mechanical-hydraulic fashion [33]. However, references presenting machines based on purely mechanical-hydraulic systems were not found in literature, which suggests the implementation of positive flow control is typically realized by means of electro-hydraulic systems. The same conclusion is also derived by other researchers [58].

4.3 Closed-center systems with constant pump pressure

When introducing electro-hydraulic, pressure compensated PDCVs in systems with fixed pump pressure, developments of the solution presented in Fig. 8 can be implemented [59]. Furthermore, two classes of alternative systems deserve special mention: an option based on servo-valves (or high-response proportional valves) and an alternative built on digital valves.

4.3.1. Hydraulic servo-systems

In applications where high-frequency, closed-loop control of the actuator is needed, the standard PDCVs used in the architectures previously described must be replaced by servo-valves, or by high-response proportional valves. They are usually mounted directly on the cylinder, or on the hydraulic motor. The obtained assembly, called servo-actuator, can be used for controlling one of the following quantities by means of a proper transducer and an electronic circuit: linear or angular position, linear or rotary velocity, force or torque. The basic principle is illustrated in Fig. 12. The transducer senses the current value of the quantity to be controlled, the signal is compared with the desired value (set-point) and the error, properly elaborated by a controller, dictates the input signal for the valve.
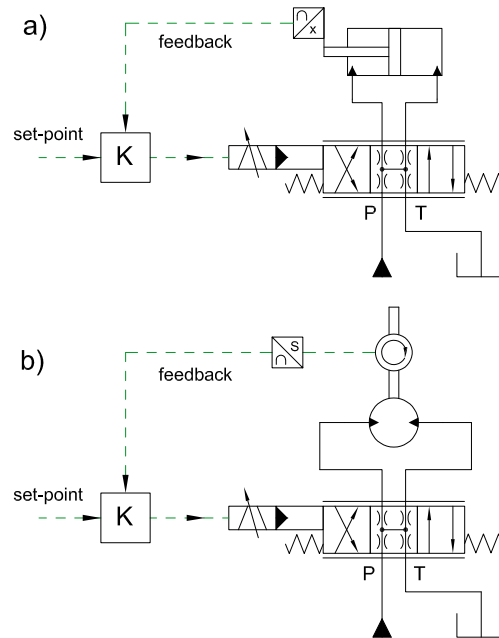
Fig. 12. Closed-loop controls by means of servo-valves: **a**) position control of a linear actuator; **b**) speed control of a hydraulic motor.

For avoiding controllability issues, the valves used in servo-actuators have two main characteristics: linearity error lower than 1% between the input command and the resulting flow rate (at constant pressure drop P-T across the valve) and substantially zero-lap (critical overlap). In the servo-valves, the deadband is lower than 3% of the spool stroke [60], even if the same characteristic is also available in the most recent high-response, proportional valves. The good linearity is obtained by adopting lands without notches and with maximum spool displacements between 0.5 and 2 mm, a quantity significantly smaller compared to the stroke of several millimeters in standard proportional valves (e.g. ± 6.4 mm [45]). The drawback is the small flow area that generates very high pressure drops across the metering edges and high-power dissipation. The advantage is the significant backpressure available in the outlet chamber of the actuator that allows high system's stiffness. In case of symmetric systems, the maximum efficiency (i.e., the ratio between the power delivered to the actuator and the total power received by the control valve) is achieved when the pressure drop across the actuator is two-thirds of the pressure drop between the ports P and T. For instance, considering for the servo-valves the typical supply pressure of 210 bar (3000 psi) means that 70 bar are lost across the metering edges. The symmetry or asymmetry of the actuator and of the control valve have also an influence on the generation of positive or negative pressure spikes in the cylinder chambers, as reported in [61]. Another disadvantage of the small flow area is the sensibility to the solid contamination that requires a very good filtration directly at the inlet of the valve. As far as the dynamic response is concerned, cut-off frequencies of the order of 200 Hz are not so uncommon, but in special designs it is possible to reach 500 Hz [62]. As a comparison, the bandwidth of standard PDCVs used in systems like the one depicted in Fig. 9 is about 5 Hz [63].

Although servo-systems are mainly used in aeronautics, robotics and test benches, some applications have been developed for mobile machines, as such hydraulically actuated manipulators for distributing concrete [64], railway tamping machines [65], or steering systems of vehicles [66].

4.3.2 Digital hydraulics

Digital hydraulics is an emerging technology conceptually different from other architectures presented in this paper. It refers to those hydraulic systems where the control elements are non-proportional valves, usually 2/2 components, commanded by an intelligent control algorithm. The main benefits of this technology when compared to traditional systems concern the use of simple, "unified" and reliable components, the potential for improved performance due to the fast dynamics of the control elements, the flexibility enhanced by the system nature (i.e., the control algorithm determines the system characteristics since functions often realized by complex mechanical components are performed by intelligently-controlled, on/off valves), and the opportunity for increased energy efficiency [67-68]. On the other hand, aspects such as pressure pulsation, lifetime of switching components, physical size, cost of architectures with multiple valves and the need for complex control techniques challenge the dissemination of digital hydraulics.

The essential idea to implement valve-controlled, digital hydraulics is to mimic proportional valves by means of solutions based on parallel connections or switching technologies. In the latter option (Fig. 13a), the average valve's flow area is adjusted via high-frequency modulation of a single on/off valve, or of a very few on/off valves. A comparison between different switching techniques is presented in [69]. Conversely, the equivalent valve's flow area is controlled by combining the flow areas of multiple on/off valves in the parallel connection (Fig. 13b), so that high-frequency switching is avoided.
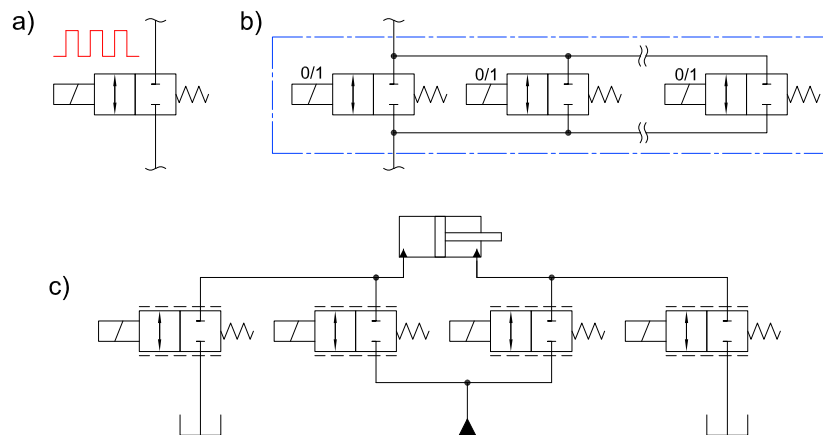

Fig. 13 Solutions used in digital hydraulics: **a**) detail of a high-frequency switching valve; **b**) detail of an equivalent valve grounded on parallel connection; **c**) a layout used to control a linear actuator.

Common layouts used to control linear actuators result inherently characterized by independent control of the metering edges (Fig. 13c), where each valve can be

implemented according to the solution a) or b). The introduction of a secondary-controlled, multi-chamber cylinder was also proposed to improve energy efficiency [70]: even if throttling losses in the on/off valves are not completely removed, a 60% reduction was shown compared to a traditional load sensing actuator. Then, digital pumps can also be implemented (i.e., several fixed-displacement pumps in parallel and a number of switching valves) to enable energy savings in working cycles with high amounts of partial load [71].

With regard to mobile equipment, commercial solutions are still not popular even though some researches have been conducted successfully (e.g., a 5-ton wheel loader [72], a small backhoe [73], a tilting system for railway carriages [74], or a mid-size excavator [75]). In fact, additional improvements are required to offer off-the-shelf components that can deal with the most demanding conditions: valve durability over one billion switching cycles and reduced electric consumption at frequencies above 100 Hz are crucial characteristics [76]. Concerning the proposed classification, it is relevant to highlight that digital hydraulics could work with a variable supply pressure (e.g., systems with multi-pressure levels [77]). Nevertheless, it was included into the "fixed pump pressure" category because the supply pressure is intended to stay approximately constant during a given operation even though different levels can be selected.

Finally, digital technologies can also benefit conventional (i.e., "proportional") valve-controlled systems. Those hydraulic circuits requiring variable flow are usually supplied by swash-plate type, variable-displacement pumps that can hardly be improved. Limits are almost reached concerning the control [78] and efficiency of these machines [79], where substantial power dissipations are present at partial displacement settings [80-81]. The introduction of the digital displacement concept to individually control the pistons within pumps or motors [82-84] improves the energy efficiency and offers control advantages. A recent publication investigated a 20 ton excavator equipped with such a digital displacement pump [85]: higher productivity (15% more meters of trench per hour) and higher efficiency (13% more meters of trench per liter of fuel) were achieved compared to the same machine using a traditional tandem pump. These improvements are not only due to the steady-state pump's efficiency since its faster dynamic response represents a major contribution to the overall system performance.

4.4 Closed-center systems with variable pump pressure

Electro-hydraulic solutions characterized by closed-center valves and variable pump pressure identify with electronic load-sensing (ELS) systems. In general, they exploit the output of transducers for adjusting the displacement of the LS pump electro-hydraulically with the aim of controlling the delivery pressure. Extending the definition, architectures that involve electro-hydraulic PDCVs but with mechanical-hydraulic pumps fall also into this category because the effects of the electronic control are tangible such as increased system damping and improved controllability. For example suitable inputs ramps can be transferred to the PDCVs when controlling high inertia loads to avoid jerky motion that, according to Andersson, is the single most important reason for using electronic load-sensing [13].

Several solutions with a progressively increased use of electro-hydraulic components and electronic controls have been developed over the years. In some simple systems, the LS pilot line is still present, but the pump margin can be dynamically reduced thanks to an

additional electro-hydraulic control that works in parallel with respect to the traditional differential pressure limiter [86]. It has been demonstrated that a dynamic management of the pump margin in the range of 15-25 bar can give significant power saving with respect to a constant setting of 25 bar [87]. The highest pressure margin can be used for fast operations, while the lowest setting represents the best compromise between energy saving and actuation velocity.

The real breakthough of the ELS system is the use of pressure transducers with the aim of removing the physical presence of the LS pilot line and the related problematics. In fact, the LS signal is transmitted by an electric wire and the pressure oscillations induced by the load can be electronically filtered. Moreover, the delay of the pressure increase at the pump delivery, is eliminated. Both double-stage and single-stage pump displacement controls have been developed: in the former, the delivery pressure is regulated by a proportional relief valve (first stage) that controls the pilot pressure of the second hydraulic stage [88], in the latter a directly actuated spool valve is used [89]. The introduction of a swash plate angular sensor can also lead to an electronic limitation of the power, which can be adjusted depending on the engine speed and on the operating modes.

Electronic control can be also applied to the proportional directional control valves allocating optimal flow to each function when flow saturation takes place; the flow directed to the active actuators can be reduced proportionally or according to a priority scheme. Most importantly, this "flow sharing" technique eliminates a potential source of system instability, since it manipulates the real-time control valves' input rather than reacting to flow saturation as with mechanical-hydraulic, post-compensated valves. However, it requires a sensor for measuring the current pump displacement. Additionally, depending on the flow demand, the speed of the engine can be adjusted in order to maximize its efficiency [41]. Furthermore, an electronically-controlled setting of the pressure compensators can be implemented in order to provide only the margin pressure that is required. This results in increased energy efficiency under most operating conditions, even though this approach is most well-suited to pre-compensated valves due to their faster response over post-compensated components, since a real-time control of the flow is required [42].

In the most advanced solutions, the local pressure compensators can even be removed. The independent load control is performed electronically by measuring the pressure drop across the main spools of the directional control valves. Figure 14 elucidates an example of an advanced electronic load-sensing system. The displacement of the pump is controlled in closed-loop by means of the electro-valve (EV) with the aim of keeping the pressure p_p equal to the most loaded actuator plus a margin. The measurement of the swash plate angle ϑ is used for improving the control and for implementing the power limitation.

In circuits where the hydraulic steering unit is fed by the same pump of the ELS system, an additional feature is the removal of the priority valve, allowing the reduction of the pressure drop between the pump and the directional control valves. The priority of the steering unit is obtained by measuring the steering wheel's angular velocity in order to calculate the required flow rate: in case of flow saturation, the flow rate requested by the PDCVs is decreased for assisting the steering maneuver [89].
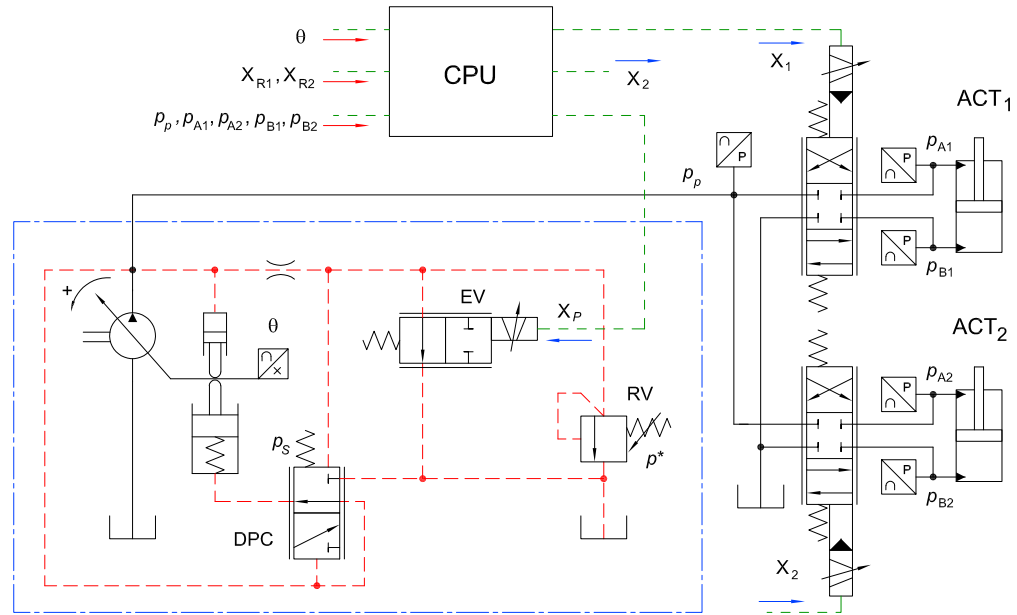
Fig. 14 Hydraulic schematic of an advanced electronic load-sensing system with two actuators.

Additionally, closed-center systems with variable pump pressure can be coupled to independent metering configurations [90] (independent metering is also possible by maintaining a constant pump pressure but this approach undermines energy efficiency). The goal is enabling autonomous control of both metering edges related to each actuator (i.e., the edge located in the supply line and the one situated in the return line) as depicted in Fig. 15.

Fig. 15 Example of an architecture based on independent metering technology where the pump is electronically pressure-compensated (simplified schematic with only one actuator shown).

Reductions in terms of energy consumption were demonstrated when using pressure-compensated pumps, even though some energy losses related to the metering edges still exist; for instance, Eriksson et al. applied this concept to a wheel-loader finding 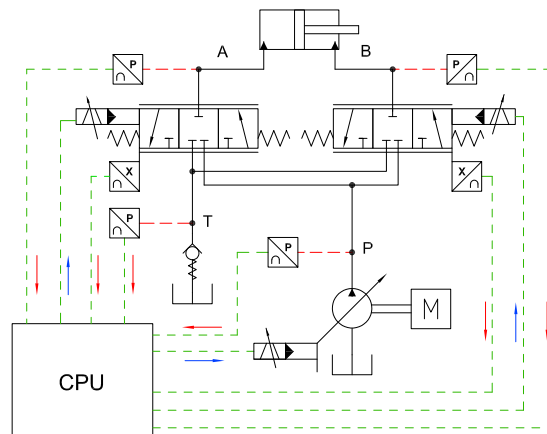energy savings of 25% [91] while improvements in the hydraulic system's efficiency of a tractor from 58.8% up to 69.6% were achieved [92]. A commercial solution of a suitable control valve was realized, namely a twin spool component, and studied for energy-saving purposes [93]. For more details about independent metering, an overview on these architectures and suitable control techniques is available in [94].

4.5 Closed-center systems with variable pump flow

This class of closed-center systems makes use of the principle known as Electronic Flow Matching (EFM) to control the displacement of the pump in order to generate the exact flow rate required by the actuators [95]. Hence, the main difference with respect to the ELS is that the pump control imposes the flow rate instead of the pump delivery pressure. Since the system is intrinsically open-loop controlled, it features better dynamic behavior [26]. Different control strategies have been developed over the last years. The simplest layout is characterized by electronic joysticks for controlling the PDCVs. The input signals can be used for estimating the required flow rate and, consequently, to set the pump displacement in order to satisfy the flow demand (the angular speed of the prime mover as well as the pump's swash plate angle must be measured).

In case of hydraulically- or mechanically-operated PDCVs, linear transducers for detecting the spool positions are necessary. Due to the volumetric efficiency and other uncertainties (e.g., the different characteristics of the PDCVs), the flow rate of the pump will not match perfectly the sum of the actuators' requirements. If the directional control valves are provided with post-LCs (Fig. 16a), the flow rate of the pump is divided proportionally to the flow areas of the main spools, therefore the directional control valves behave simply as flow divider. Moreover, lower power dissipation is achieved if the electronic control unit fully opens the valve with the highest flow and, accordingly, increases proportionally the flow areas of the remaining valves in order to maintain the same ratio of flow rates but with less throttling losses [96-97]. The correction of the input commands can also be performed for managing the situation when an actuator reaches the end-stop [98].

In architecture with pre-LCs (Fig. 16b), the main spools decide the value of flow rate and a pressure compensator (PC) with position sensor can be used for discharging the excess flow to the reservoir. This increases the robustness of the system at small flow rates compared to EFM post-compensated solutions [99]. Moreover, the pressure compensator, due to the higher dynamics with respect to the pump control, can also be used during the transient for compensating a sudden increment, or reduction, of the flow demand [100]. Alternatively, the spool positions of all LCs can be detected; in this case the pump displacement is controlled in order to keep the position of one of the LCs within a proper range. In case of correct value of flow rate, the LC connected to the highest load will work with a quite large flow area with respect to the others, anyway not fully open (it would indicate an insufficient flow rate). Solutions have been studied also for handling the flow rate required by non-electrically controlled auxiliary functions [101].
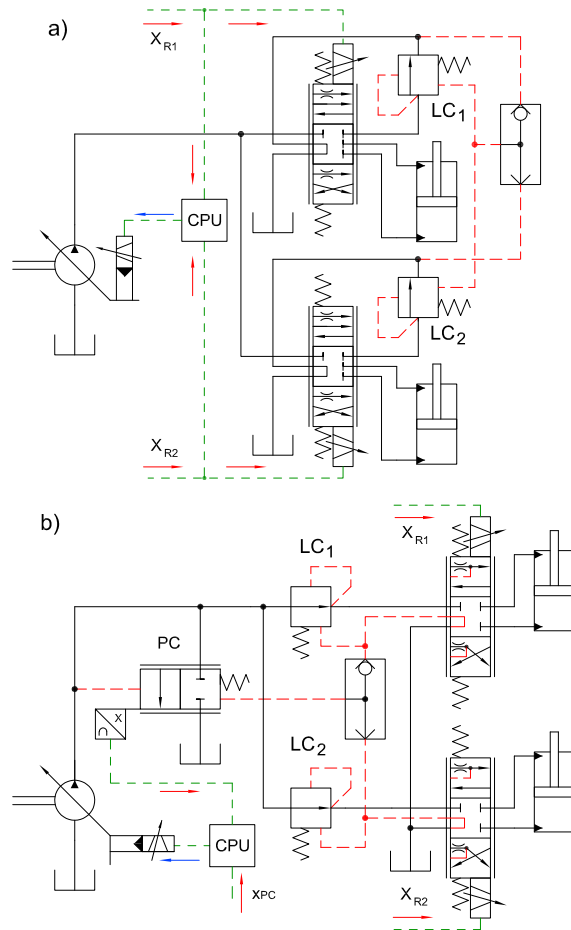
Fig. 16 Hydraulic scheme of EFM systems with electronic joysticks: **a**) with local post-compensators; **b**) with local pre-compensators and pump compensator.

For a better matching between the flow rate of the pump and of the actuators, a look-up table of the volumetric efficiency can be used; in this case the delivery pressure and the oil temperature information are also needed. A closed-loop control of the delivered flow rate can be performed by adding a flowmeter at the outlet of the pump; as an alternative, the current velocities of the actuators must be directly measured. Further improvements of the energy saving can be achieved by opening at the maximum value the directional control valve connected to the actuator with the highest load [99] or by optimizing the PDCVs' flow area [102]. In more complex solutions the EFM principle is extended to multi-circuit systems in order to reduce the throttle dissipations with respect to the case with a single pump feeding all actuators [103].

In general, since the pump displacement is electro-hydraulically controlled, it is possible to merge different control strategies. Systems have been studied where the hydraulic LS principle is active for low pump displacements, while the EFM takes control at high flow demand [86]. Independent metering of the PDCVs' control edges can also be combined with EFM (e.g., in [104-105]). Finally, ELS, EFM, and open-center architectures can be implemented in the same machine and properly modulated by the operator based on the type of activity to be executed [106]. Similarly, the proprietary solution IFC (Intelligent Flow Control) developed for a wheel loader [107] and for a refuse truck [19]

controls respectively the displacement of two and three pumps in parallel in order to deliver the desired flow rate to each actuator; in this case, the corresponding PDCV is maintained fully open. However, since each pump is also provided with a pressure transducer, the amount of flow rate delivered to the actuator can be also controlled through the PDCV.

5 Conclusion

This paper reviewed, summarized, and classified the most important system architectures used to control the working hydraulics of non-hybrid, valve-controlled mobile machines.

It emerged that relevant efforts were devoted to design and improve, over several decades, mechanical-hydraulic solutions suitable for different machines. The well-established, cost-effective open-center systems with fixed-displacement pumps characterized by very good inherent damping and fast response time are affected by poor energy efficiency. New implementations dominated by variable-displacement pumps were proposed to reduce the energy losses while maintaining good system damping. Significant improvements in terms of energy efficiency were then achieved by combining closed-center PDCVs with variable-displacement pumps at the expense of system damping, response time, and cost. The main features of the predominant mechanical-hydraulic solutions are summarized in Table 1.

Table 1. Synthesis of the mechanical-hydraulic systems addressed in this paper based on the revised material (ratings: poor, acceptable, and good).

Criteria	Open-center fixed pump	Open-center variable pump	Closed-center fixed pressure	Closed-center variable pressure
Inherent damping	Good	Good	Good	Poor
Response time	Good	Good	Good	Acceptable
Cost-effectiveness	Good	Acceptable	Acceptable	Poor
Energy efficiency	Poor	Acceptable	Poor	Good

Mainly for the afore-mentioned limitations (i.e., system damping, response time, and energy efficiency), a clear tendency toward electro-hydraulic solutions is in place. Multiple layouts and many control algorithms have been designed and tested while some solutions are also commercialized. Among numerous benefits enabled by electronic control, it is worth to recall the following mentioned features. Electro-hydraulic pump displacement control is preferred because the pump does not generate excess flow, reducing therefore the lost power. The removal of the mechanical-hydraulic pilot pressure generation adds more flexibility in terms of system control and improves the response time (long pilot lines are no present anymore). System damping can be adjusted, especially when driving large inertial loads and feedback control can also be implemented so that many processes can be automated.

Concerning future innovations, an even more intensive use of electro-hydraulic solutions is expected. Widespread requirements about improved controllability and productivity are leading to more advanced control algorithms (e.g., smart utilization of the

prime mover). Then, necessities for higher reliability and less downtime are requiring the use of components with embedded sensors and intelligence to perform health monitoring. Finally, energy efficiency is playing more and more an important role due to both the oil price and the new emissions regulations, which could raise the number of hybrid systems.

Appendix A: Management of overrunning loads and load-holding capability

In all lifting systems, such as cranes, winches or telehandlers, the hydraulic circuit must be able to control the actuators' speed also in case of overrunning loads (i.e., during the lowering phase of the telehandler boom [108]). Moreover, the capability to keep the actuator blocked under load must be guaranteed. The typical layout, which can be applied to almost all architectures described in the present paper, involves the use of overcenter valves (OVC). Figure 17a reports the hydraulic scheme for controlling a linear actuator with overrunning loads active in both directions.
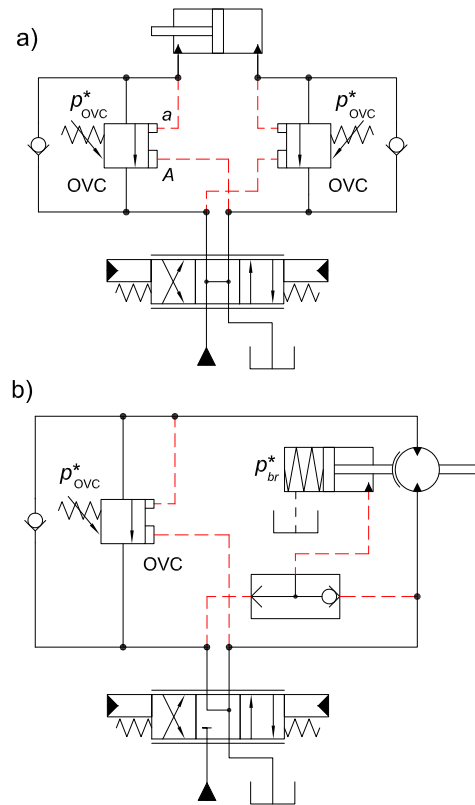
Fig. 17 Typical circuits for lifting systems; **a**) layout for controlling overrunning loads in both directions on a linear actuator; **b**) circuit for a hydraulic motor with integral holding brake and overrunning load acting in one direction.

For safety reasons, the OVCs are usually mounted directly on the actuator. In the unactuated position of the directional control valve, the pilot lines are connected to the reservoir and both OVCs are closed. Thanks to their leak proof design, the actuator is able to hold the load, whose maximum value is function of the pressure setting p^*_{OVC}. In case of overrunning loads during actuation, the OVC located at the actuator's outlet port

generates a back-pressure that allows balancing the external driving force. On the contrary, when the load is resistant, the pump delivery pressure can keep the OVC at the outlet port fully open (an unnecessary back-pressure is not introduced).

A drawback of the overcenter valves is that a quite high pump pressurization is necessary, even with overrunning loads, to keep the valve in regulation [109]. The delivery pressure can be reduced by adopting a higher ratio A/a between the two active surfaces of the valves (pilot ratio), but the risk is the instability of the system.

For non-critical load-holding applications, an alternative solution is the use of directional control valves with integral pilot-operated non-return valves on both working ports connected to the actuator.

In case of rotary actuators, such as for winches or for the rotation of the machine's cabin, the load-holding capability cannot be achieved through leak proof valves, due to the inevitable leakages inside the hydraulic motor; in this case, a mechanical holding brake must be used. In Fig. 17b, an example of a circuit that drives a winch is shown; the overrunning loads act only in one direction, hence only one OVC is necessary. The brake is automatically released by the pump delivery pressure, selected by a shuttle valve, once the pressure setting p^*_{br} is achieved. The braking torque at rest is ensured by the connection to reservoir of the pilot line through the floating (or by-pass) center of the directional control valve. The floating (or by-pass) center position of the directional control valves used in common applications is characterized by restricted flow areas so that, in case the OVC fails, the cylinder does not fall down abruptly. For the case of rotary actuators, the brake overcomes this failure.

Appendix B: Management of the power required to the prime mover

Regardless of the type of architecture, an additional feature that can be implemented in the pump displacement control is the function for limiting the maximum absorbed power. It is straightforward that, in case of constant angular speed of the prime mover, the regulation of this device dictates a constant torque.

For a mechanical-hydraulic system, the three-port valve DPC (differential pressure compensator) in Fig. 18a modulates the pressure in the displacement actuator A in order to impose the pump delivery pressure at the value $p_{LSG} + p_s$. However, the pressure-relief valve TL limits the pressure in the pilot line X to a value dependent on the current displacement of the pump. In particular, the compression of the springs is incremented linearly as the displacement decreases. Since one of the two springs is assembled with an initial gap, it generates a force only for low pump displacements and this leads to a relationship between the generated flow rate and the delivery pressure characterized by two different slopes, as reported in Fig. 18b (thick trace).

The two thick segments well approximate a hyperbola that represents a constant power curve. In synthesis, three operating conditions can be obtained depending on the pressure induced by the highest load:
1. Constant flow rate Q_0 imposed by the sum of the flow rates decided by the proportional valves;
2. Constant power due to the intervention of the torque limiter;
3. Constant pressure, when the pressure-relief valve RV regulates.
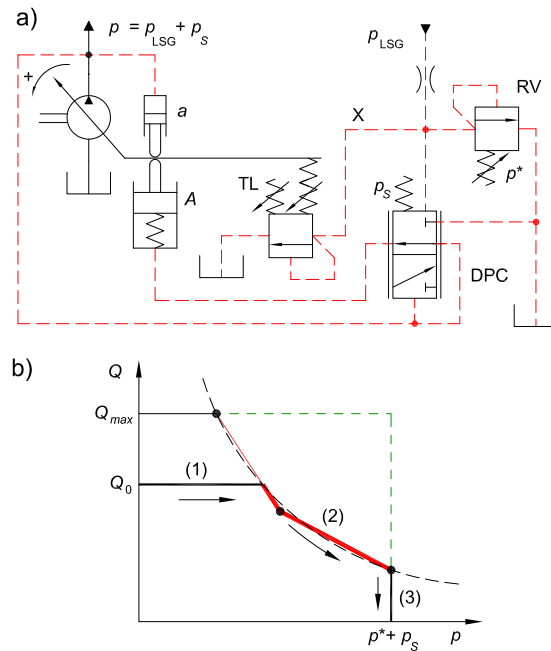
Fig. 18 a) Hydraulic scheme of a load-sensing displacement control with power limiter; **b**) the resulting flow-pressure characteristic of the pump.

Acknowledgment

Nomenclature

Abbreviations

A, B	Actuator ports
ACT	Actuator
DPC	Differential pressure compensator
EFM	Electronic flow matching
ELS	Electronic load-sensing
EV	Electro-valve
LC	Local pressure compensator
LS	Load-sensing
OR	By-pass orifice
OVC	Overcenter valve
P	Pump
PC	Pump's pressure compensator
PDCV	Proportional directional control valve
RV	Pressure-relief valve
T	Reservoir
VPRI	Priority valve

Symbols

A, a	Areas

p	Pressure
p^*	Equivalent pressure setting
$p_{c,post}$	Equivalent pressure setting of local post-compensators
$p_{c,pre}$	Equivalent pressure setting of local pre-compensators
p_{LS}	Local load-sensing pressure
p_{LSG}	Global load-sensing pressure
p_x	Control pressure or pilot pressure
Q	Flow rate
p_s	Equivalent pressure setting of the pump compensator
X	Control signal
Δp	Pressure difference
ϑ	Pump's swash plate angle

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