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The Hydraulic Power Generation and Transmission on Agricultural Tractors: feasible architectures to reduce dissipation and fuel consumption – Part I

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Abstract. This paper is aimed at investigating the benefits in terms of energy efficiency of new electro-hydraulic architectures for power distribution systems of a medium-size agricultural tractor, with a focus on the hydraulic high-pressure circuit. The work is part of a wider industrial research project called TASC (Smart and Clean Agricultural Tractors [1]). Traditional and alternative architectures have been modelled and energetically compared through simulation, using a lumped parameter approach. Experimental data previously acquired have been used to validate the models and to replicate real working conditions of the machine in the simulation environment. A typical on-field manoeuvre has been used as duty cycle, to perform an effective energetic analysis. The standard hydraulic circuit is a multi-users load sensing system that uses a single variable displacement pump to feed steering, trailer brake and auxiliary utilities in that order. The key idea of the proposed solutions is the separation of steering from the other implements, to optimize the entire energy management. In particular, the paper investigates new and flexible solutions for the auxiliary utilities, including an electro-hydraulic load sensing architecture with variable pump margin, an electronic flow matching and flow sharing architecture, and an electronic strategy for automatic pressure compensation. The simulation results show that good energy saving can be achieved with the alternative architectures, so that physical prototyping of the most promising solutions will be realized as next step of the project.

Introduction

In the field of off-road vehicles, the increasing need to reduce fuel consumption and pollutant emission is pushing the research towards solutions that allow reducing the overall energy consumption of the machines, without affecting the performance. A key role in energy demand is played by the hydraulic sub-systems of the machine, which absorb power from the

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engine to perform several operations, as steering, braking, load handling or implements managing, paying for the high versatility and power density with rather low efficiency. This work analyses the hydraulic high-pressure circuit of a medium-size agricultural tractor, chosen as the target vehicle of the research project, with a particular focus on the auxiliary utilities, which are those requiring most of the power [2].

In mobile machinery applications, the state of art for the hydraulic circuit architecture is represented by load sensing (LS) system, that offers a good compromise between costs and efficiency. LS systems are robust, reliable and more efficient if compared to standard open-centre (OC) systems equipped with fixed displacement pump. However, a LS system, being load independent, offers a lower damping contribution with respect to an OC one: to improve that, a combined LS with a virtual OC characteristic has been studied for example in [3]. Typically, because of weight and dimensions constraints, a single pump is used to feed several users connected in parallel. As a consequence, when multiple actuators work together at different loads, the pump pressure is adjusted according to the highest load, and pressure compensation is required to maintain the control of the lower loaded users, leading to significant energy dissipation. Hence, it is common today to investigate the possibility to separate the users, to improve the overall efficiency.

Other possibilities for energy saving rely on the use of electronic control strategies to control the power supply unit [4] and the regulating valves: in [5], a LS with independent metering (IM) valves with both meter-in and meter-out pressure compensation is analyzed. Borghi et. al [6], [7] investigated the benefits of an IM architecture, coupled with both a traditional LS and an electronic flow-controlled pump. The electronic control of the displacement of the pump, known as Flow Matching (FM), is rapidly spreading in mobile applications, since it can improve machine hydraulics' efficiency, stability, dynamic behaviour and flexibility with respect to LS systems.

Given the complexity of the systems, the combination of simulation tools and experimental activities represents the best approach to energetically analyse the behaviour of the hydraulic circuit and to investigate new possible solutions to reduce power consumption (see for instance the works related to agricultural machine hydraulic sub-systems reported in works from [8] to [13]). A very important issue is that often the amount of energy saving that can be achieved with the same architecture strongly depends on the operations and the required performance during the working cycle of the machine, hence it is important to consider the real operating conditions of the vehicle in the picture. In [14], [15], [16] for example, the combined experimental and simulation analysis have been developed for a middle-size excavator, and different hydraulic systems configurations have been compared, referring to the JCMAS standard cycle. Axin et al [17] used a short loading cycle to evaluate the benefits of FM versus LS architecture on a wheel loader vehicle. Unfortunately, unlike earthmoving machinery, for an agricultural tractor there are not standardized duty cycles for evaluating the performance of the hydraulic circuit, since several specific operations can be performed by changing the equipment (e.g. plough, seeder, loader, baler, harrow, etc.) When considering a tractor, experimental tests represent therefore a fundamental step in the definition of mission profiles, to identify a reference working cycle, as explained in [18]. For this reason, a duty cycle involving an on-field manoeuvre experimentally performed has been considered in this work, to perform an effective analysis.

The paper is organized as follows: firstly, the standard LS architecture is presented, together with the description of the mathematical modelling using Simcenter Amesim [23]; in section 2, the validation of the model on the basis of experimental data is reported, together with a power flow and dissipation analysis. Section 3 is dedicated to the new investigated architectures; results are finally shown, in terms of performance and power consumption, in section 4.

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1. The Standard High-Pressure Circuit and its Modelling

Actual high-pressure circuit of the tractor is a typical closed-centre load sensing (CCLS) multi-actuators system. It uses a single power supply unit, composed of a charge pump and a variable displacement axial piston pump, equipped with hydraulic flow rate and pressure compensators, as shown in Fig. 1. In normal operating conditions, the flow rate compensator controls the displacement of the pump to maintain a constant pressure margin between the pump and the highest load, so that flow rate in the circuit would be independent of load pressure. The pressure margin is set through the spring preload of the flow rate compensator and it is determined in order to overcome the pressure drop between the pump outlet and the load in the system, at maximum flow rate condition [4], [7]. The pressure compensator valve instead works limiting the pump displacement if the pressure in the delivery line reaches the maximum permitted value.

Downstream of the pumps, the circuit includes a priority valve (PRV), which has the task of ensuring and distributing the flow among the users of the tractor, according to the following priority order: steering (ST), trailer brake (TB) and auxiliary utilities (such as rear remotes and hitch). The PRV sets the limits of the operating pressure and flow rate of the users, ensuring functionality to the different subsystems. The PRV block consists of two valves in parallel, as schematically represented in Fig. 2: the first is dedicated to the steering, and works as a local pressure compensator; the second, called master spool, distributes the flow to the secondary utilities only once the steering line is fed with enough flow and the minimum pressure is ensured. The block also contains the check valves to select the highest load sensing pressure signal to be delivered to the pump as load pressure feedback.

The steering consists of a dynamic hydrostatic unit, similar to the one described in [19], and it is described and analyzed with more detail in the Part 2 of this paper. The TBV task is to manage the pressure signal generated by the brake pedal, also providing the parking brake function. However, the TBV does not involve high power consumption, so it has not been considered in this work.

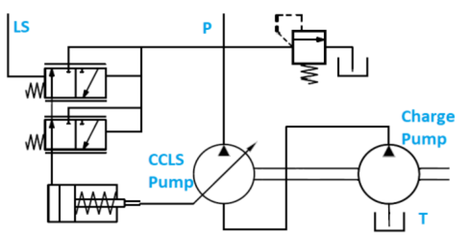

Fig. 1: Hydraulic scheme of power supply unit

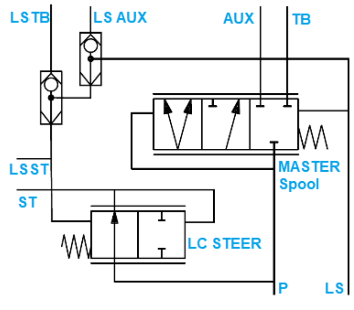

Fig. 2: Hydraulic scheme of priority valve block

At the rear of the tractor, five sections of electrohydraulic remotes (EHR) distributors are connected in block, in a modular architecture. They are designed to manage parallel actuations and can serve a wide range of utilities, depending on the equipment connected to the tractor. In Fig. 3, the hydraulic scheme of a single section is shown. The core of the block is represented by the pilot operated main proportional control valve; the pilot pressures are selected through two electro-valves and the metered flow rate supplied to the user corresponds to the degree of opening of the spool. Since more than one section can work simultaneously at different operating pressure, together with other utilities too, a local pressure compensator (LC) is placed upstream of the main proportional control valve to guarantee control. Two check pilot operated valves provide non-return function for the actuator, connected to the block through fast couplers. Finally, a shuttle valve selects the highest load-sensing pressure to be delivered back. Since EHR control valve represents the most dissipative element in the circuit, a detailed reverse engineering model representation of the block, based on the one developed in [7], has been used.

The rear hitch is made of two hydraulic cylinders in parallel that act the tree point hitch and the control valve section is directly connected to the rear remotes block. The scheme, see Fig. 4, comprises two electro-hydraulic valves, one for lowering control, which is gravity assisted and does not involve pump flow, and the other one for lifting operations, which is pre-compensated. The available high-fidelity model, presented in [20], has been re-adapted and used to simulate the rear hitch.

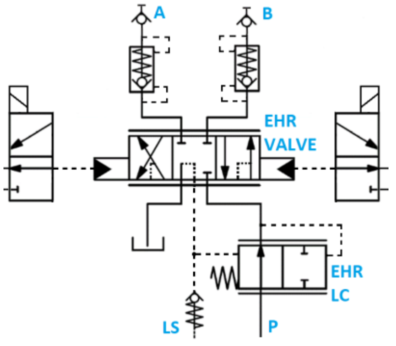

Fig. 3: Hydraulic scheme of EHR section

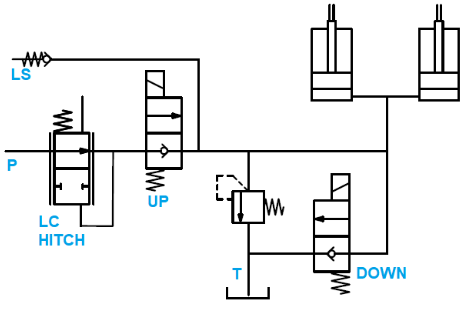

Fig. 4: Hydraulic scheme of rear hitch block valve

The overall Amesim model of the actual high-pressure circuit (that will be called Baseline hereafter) is presented in Fig. 5. The main pump has been modelled as a generic component, characterized by a map that describes the variation of the efficiency as function of the shaft

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speed, the pressure and the fractional displacement. The two compensators are modelled in detail using Amesim Hydraulic Component Design library, thus allowing to consider in the model the dynamic behaviour of the power supply unit. The model includes also other main accessory components such as the main filter and the heat exchanger, placed at pumps' suction. Each block encloses a mathematical representation of the parts described above, each with its own level of detail.

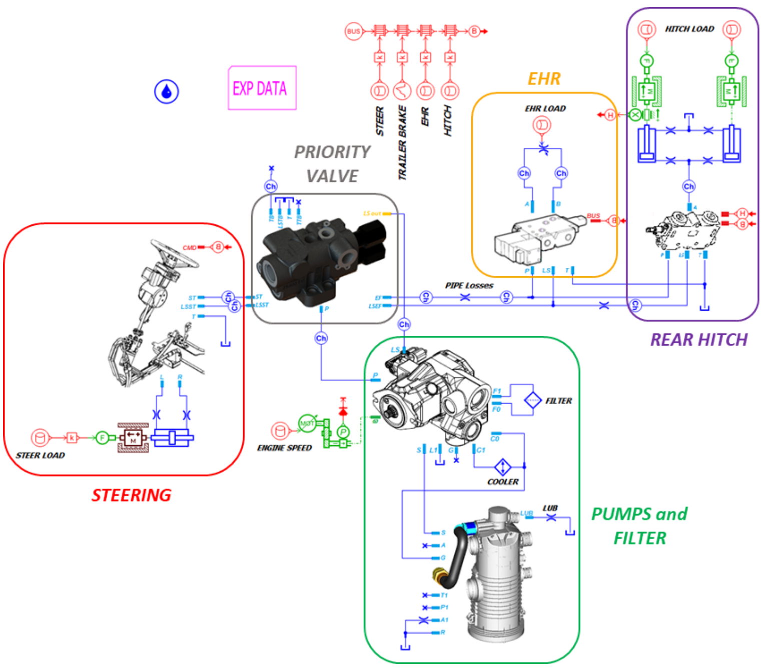

Fig. 5: AMESim model of the Baseline high-pressure circuit

2. Experimental Data Acquisition, Baseline Model Validation and Power Consumption Analysis

To characterize the standard circuit of the tractor in terms of energy consumption, experimental data measured by the manufacturer of the tractor in a previous research activity has been post-processed and analysed. Both stationary and dynamic tests, at different engine speed and load conditions were performed, involving either stand-alone or simultaneous steering and rear utilities operations. To monitor the hydraulic quantities at the different points of the circuit, the tractor was equipped with flow rate and pressure sensors. The data collected form the sensors, together with the ones coming from the electronic control unit (ECU) of the vehicle via CAN BUS, have been analysed initially to define the loads and signals set of inputs to be used in the virtual model to replicate the duty cycle. Then, the numerical results from the model and the measured data were compared to perform the validation of the model.

Fig. 6 reports the comparison between experimental data and numerical results, considering an experimental on-field test. The values of pressure and flow rates are normalized with respect to their maximum values in the system. The test reproduces a typical end-field manoeuvre, and has been performed at 6 [km/h], using a plough connected to the

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rear three points hitch of the tractor. The rear hitch is used to lower and lift the plough: once the hitch is reaching the upper maximum displacement, the plough is rotated using the hydraulic power provided by one rear remote section. During these operations, also the steering has been continuously actuated, performing a sinusoidal steering input of approximately 250 [degrees] of amplitude. The correlation between the numerical and simulated data is fine, so that the baseline model adequately describes the behaviour of the standard hydraulic circuit, both in terms of system and single components.

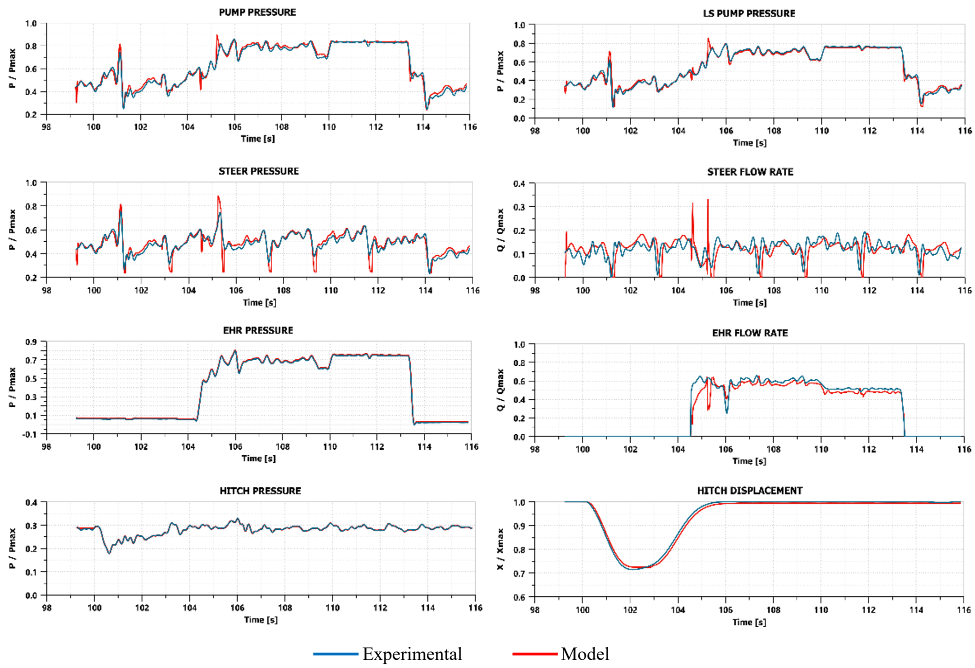

Fig. 6: Correlation between experimental and model results, Baseline circuit, End-Field test

The validated baseline model represents the benchmark for the evaluation of performance and consumption. A detailed power flow analysis has been performed with the model, to identify the most dissipative parts of the standard circuit at different operating conditions. Fig. 7 shows the average power consumptions of each part of the circuit expressed as percentage of the mechanical power required by the pumps (which represents the 100%). Fig. 7.a) refers to a rear remotes' operation, performed with tractor at rest and engine at maximum speed, involving two sections at 50% opening, with variable differential loads. The 25% of the total power is already dissipated on the pumps and on the main filter; for both EHR sections, overall block valve dissipations are half the load power usage: this is caused by the presence of hydraulic operated check valves and quick couplers, which waste approximately 5% of the power, and the local pressure compensator, which wastes approximately 6% of power to compensate for differential pressure between the two sections. The PRV's master spool doesn't affect efficiency significantly, since it is fully open to the auxiliaries' line, working as a fixed large orifice.

In Fig. 7.b), steering and one remote section are actuated simultaneously, while the tractor is moving at 10 [km/h]. The remote valve is fully open, with low pressure load. As per the previous case, 20% of losses are due to the pump and filter. In addition to the dissipations on check valves and quick couplings, which are almost 30% of the power, over 5% of losses occurs on EHR local compensator since load sensing pressure is given by the steering. Significant losses, almost 10%, occur on the master spool of the PRV: the reason is that pump has saturated, so that the spool closes the way toward other utilities, to feed steering first, causing additional losses. Finally, power flow for the previously presented end-field test is shown in Fig. 7.c): no losses occur on the PRV, while dissipations on the local compensators of the steering (approximately 5%) and the hitch (2,5%) are introduced, because in this case the EHR requires the highest pressure to rotate the plough. Nevertheless, also the EHR local compensator slightly affects the power consumption, introducing an undesired loss even when it is not compensating. In all the considered tests, about 3-5% of power returns to the pressurized tank.

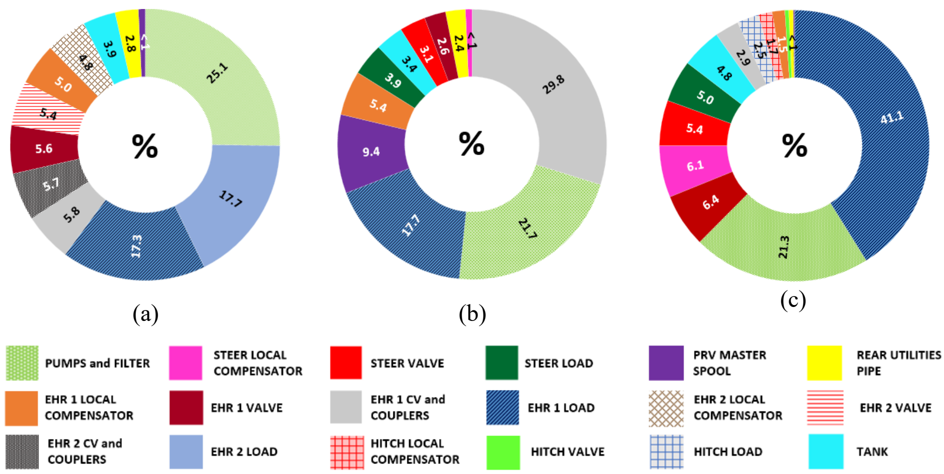

Fig. 7: Baseline model power flow: **(a)** Remotes test; **(b)** Steering + Remotes test; **(c)** End-Field test

3. New Architectures

As shown in the power flow analysis, in normal operating, the master spool of the PRV does not significantly affect power consumption. However, it sets limits of the operating pressure, since it always requires a certain pressure margin to open the connection towards auxiliaries' line. If the system pressure margin is reduced or falls below that value, as the pump gets saturated, the master spool closes the way toward auxiliary users to guarantee steering first, introducing a significant loss. In addition, during on field operations, which are the more interesting ones for that kind of tractor, often the steering operates at lower pressure level than auxiliaries, so that dissipation is introduced by the primary spool of the PRV.

For these reasons, a new system layout is proposed and investigated in this work: the priority valve is removed, and a new smaller pump is specifically added for steering, being separated from the rest of the circuit. In this layout, PRV losses are avoided and the risk for saturating the main pump is reduced, although another pump has to be installed. This represents the starting point to optimize the design and control of the whole hydraulic high-pressure circuit, increasing flexibility and degrees of freedom in the development of new control strategies. Different solutions for auxiliary utility users are presented in this section; both the unified and the new separated layout have been considered, and combined solutions have been tested too to evaluate the potential for energy saving of each of them. In the Part 2 of this paper, new possible steering architectures are considered.

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3.1 Variable Pump Margin (VPM)

In the standard LS architecture, the pump operates with a fixed pressure margin over the highest load, regardless of the flow rate delivered. However, the value of the pressure margin is set to overcome losses that occur at maximum flow rate; this means that, when the pump is working at reduced displacement, that pressure margin is not necessary and dissipations are introduced by the local pressure compensators, in order to maintain the desired pressure drop across the directional control valves.

One possible solution to reduce or even avoid unnecessary losses and increase the energy saving is to dynamically reduce this pressure margin, according to flow rate user's request. This would require to use an electro-hydraulic flow compensator, controlled by a proper electronic strategy to adjust the spring preload. This strategy is called Variable Pump Margin (VPM); Fig. 8 shows AMESim model of the new electro-hydraulic compensator, in which the solenoid is used to vary the force of the spring and so the pump margin setting.

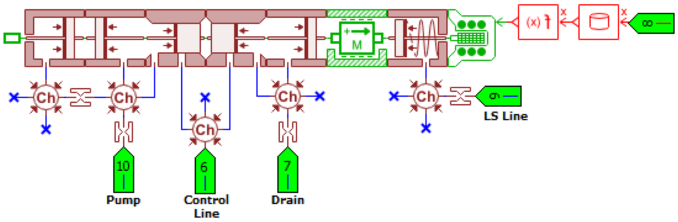

Fig. 8: AMESim model of electro-hydraulic flow compensator of the pump

The pump margin must be high enough to guarantee the correct flow rate in any operating condition: if not, the flow rate delivered by the pump does not match users' request and some of the utilities may slow down or even stop. It is hence fundamental to correctly map the variable pump margin as a function of the requested flow rate: the lower the flow rate, the lower the pump margin that the system requires and so the consumption. However, it is a matter of power rather than of pressure: the greatest saving with VPM solution occurs at intermediate flow rates [21]: no power saving is achieved at maximum flow rate condition, since all the pump margin is required, while for low flow rates, even if the pressure margin is reduced significantly, the power saving is quite small, since the hydraulic absolute power is small too.

In the following, the VPM Strategy has been analysed either for the standard and the separated layout, as single modification of the system and, afterward, as part of a more complex modification of the system architecture.

Two components in the circuit play a fundamental role in the determination of the pump margin: priority valve and rear utilities' local pressure compensator. The first always requires a minimum pressure margin to keep the way from the pump to the auxiliaries open. Local pressure compensator instead also provides a non-return functionality, so that a certain difference between feeding and local load sensing pressure is always needed to keep the compensator open against its spring preload. This implies again a minimum pump margin value similar to the one requested by PRV. This is the reason why in architectures involving either the PRV or the LC, a significant minimum pump margin is set if at least one of the auxiliary utilities is actuated, even with minimum flow rate request. Starting from this value, a quadratic characteristic Pump Margin vs Flow Rate has been considered, reaching maximum pump margin at maximum flow rate (see the red curve in Fig. 9). When no auxiliaries are actuated, the ECU maintains a lower fixed pump margin, that allows for standby or steering operations.

The blue curve in Fig. 9 refers to a control map suited to a separated architecture in which also the local pressure compensators are removed. In this architecture, the metering valve is used to compensate the differential loads at the remote utilities with an electronic control strategy, presented in the next subsection. In this case, a stand-by lower limit pressure margin is not needed, and the pump margin value only accounts for the pressure drops occurring in the pipe plus a fixed pressure, which fulfils the EHR valve requirement. Pump margin regulation in this case is noticeably lower, so that a higher power saving is expected.

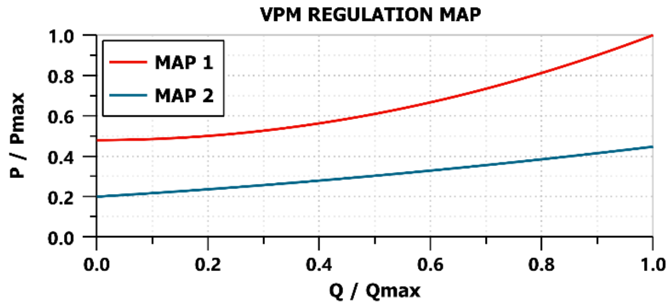

Fig. 9: Variable Pump Margin regulation maps

3.2 Remotes Electronically Compensated (REC)

Rear utilities' local pressure compensator plays a fundamental role in affecting the energy consumption, since it imposes a lower threshold for the pump margin, and also because it introduces losses even when not compensating for differential load pressure, especially at high flow rates.

The alternative system here presented, called Remotes Electronically Compensated (REC), is similar to the one investigated in [21]. It consists in removing the local pressure compensator of rear remotes and hitch, and replacing this function by controlling the main spool of the directional valve, using an opportune strategy to meet the requested flow rate, see Fig. 10.

To perform the compensation electronically, additional pressure sensors will be placed on the control valve to measure the pressure drop during each working operation across the meter-in section of the valve. The correct position of the spool is chosen on a 2D Metering Map (Pressure/Flow-Area) of the valve, implemented on the ECU, as function of the desired flow rate and measured pressure drop value. The ECU controls the EHR spool and hitch's raising valve position ensuring the desired performance. In this way, the metering flow area of the lower loaded users is automatically reduced, guaranteeing the correct flow rate distribution. Since the EHR is a traditional single spool valve, restrictions in the meter-in flow area results in reduction in the meter-out section too. Accordingly, an undesired backpressure may occur, causing a lower efficiency of the system. To avoid this backpressure in the REC architecture, the design of the directional control valve has been modified, enlarging discharge passages through the valve to the tank.

Conceptually, this solution does not significantly reduce power consumption itself: in the case of differential loading conditions, the pressure drops introduced by the local compensators of the lower loaded users are not eliminated, but replaced on the meter-in section of the main valve. In contrast, a significant energetic advantage occurs when the REC solution is combined with a VPM strategy, since removing the local compensator leads to higher flexibility and degree of freedom in the dynamic regulation of the pump margin of the system, which can be reduced further.

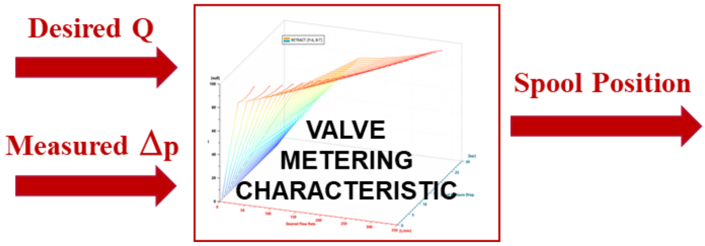

Fig. 10: REC control strategy

3.3 Electronic Flow Matching (EFM)

Instead of controlling the pump by pressure in closed loop, to maintain a certain margin, another possibility is to move to a flow controlled system, in which the displacement of the pump is directly electronically controlled in open loop, in order to match the users' flow rate request. In this way, the pump pressure is no more predetermined according to a certain margin, either fixed or variable, but it only depends on the resistances on the flow rate path from the delivery to the users. The working pressure is therefore always the minimum possible value required by the system to work over the highest load, so that a certain power saving is obtained.

The basic idea of this solution, named here as Electronic Flow Matching (EFM), but also known as Flow Demand, is to exploit users' joystick signal to simultaneously control the swivel-angle of the pump and the position of the valves. In case of multiple actuations, the pump displacement is adjusted according to the sum of flow rate requests, while, if no function is actuated, the pump can be fully de-stroked, since there is no more load sensing pressure input. To do this in the simulation environment, a PID controller has been used to control the displacement of the pump, with a first order filter to reproduce the dynamic response delay. However, when two or more actuators work together, also a flow-controlled system will be affected by load interaction, requiring pressure compensation to correctly distribute the flow rate among all the utilities. This can be performed either hydraulically, through pressure compensators or electronically, according to different control strategies. Two flow matching solutions for auxiliary utilities have been investigated in this work. The first one combines a flow-controlled pump with the REC control strategy (EFM REC Architecture); the second involves the use of flow sharing local pressure compensators placed upstream of the control valves (EFM FS Architecture). As explained in [22], also a traditional pre or post-compensated valve might be used, but the integration with a flow controlled pump leads to an over-determined flow rate condition, since both the pump and the valve will control the absolute flow rate value: in a standard compensated valve, in fact, a certain opening corresponds to an absolute flow rate request, since pressure drop across the metering orifice is fixed by the compensator's spring preload. If the pump flow rate does not perfectly match what expected from the valve, for example in case of saturation, the functionality of the system may be lost. The working principle of a flow sharing compensator, presented in Fig. 11, is obtained removing the spring and "sending" to the compensator the highest load pressure (p_{Lmax}), the inlet pressure (p_p), the pressures across the metering orifice (p_r and p_L), and using two opportunely designed active pilot areas (A_1 and A_2). From the equilibrium of forces on the compensator, the pressure drop Δp_s across the control valve is obtained, using the equations (3.1) and (3.2). The flow rate through the valve is expressed in equation (3.3), where C_d is the discharge coefficient, ρ the fluid density and A_s the valve opening area.

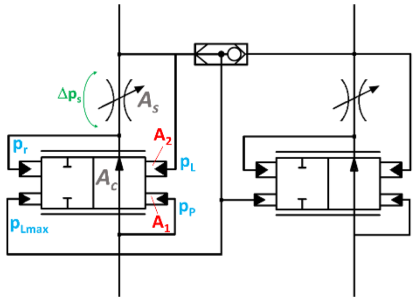

Fig. 11: Flow sharing compensator working principle

$$p_p A_1 + p_L A_2 = p_{Lmax} A_1 + p_r A_2 \tag{3.1}$$

$$\Delta p_s = (p_r - p_L) = \frac{A_1}{A_2}(p_p - p_{Lmax}) = \frac{A_1}{A_2}\Delta p_p \tag{3.2}$$

$$Q_s = C_d A_s \sqrt{\frac{2}{\rho}\Delta p_s} = C_d A_s \sqrt{\frac{2}{\rho}\frac{A_1}{A_2}\Delta p_p} \tag{3.3}$$

Neglecting the losses on the line, $\Delta p_p = p_p - p_{Lmax}$ represents the pump pressure margin over the highest load, that, as said, automatically builds up in the system. This means that the metering pressure drop Δp_s across the main directional valve, that is the same for all the control valves, will be automatically the one needed in the specific operating condition, avoiding the need to set a constant spring preload. Accordingly, no matching problems occur since the flow rate is univocally controlled by the pump. The valves just work as flow dividers: the pump flow rate is shared among the users in the same proportion of the opening areas. It is hence possible to fully open the valve of the actuator working at the highest flow, and increasing the valve areas of the other actuators in proportion with the flow rate requests. This will minimize the pressure drop across the valves, and thus save energy, without any change in the performance. To do this, a control strategy for remotes and hitch valves has been developed.

The same compensator has been used for remotes and hitch valves and it has been designed according to [17]. The functionality of the system is based on the equation (3.1): according to the equilibrium of forces, in the highest loaded section ($p_L = p_{Lmax}$) the pressure drop over the compensator Δp_c (with the compensator fully open) is expressed as in equation (3.4). Since the flow rate through the compensator equals that through the directional valve (see equation 3.5), a proportional relationship between the flow area of the compensator at its maximum opening A_c and the area of the valve A_s can be derived, as in equation (3.6). A_c must be large enough to fulfil the relationship, even for $A_s = A_{s,max}$.

$$\Delta p_c = (p_p - p_r) = \left(1 - \frac{A_1}{A_2}\right)(p_p - p_{Lmax}) = \left(1 - \frac{A_1}{A_2}\right)\Delta p_p \tag{3.4}$$

$$Q_c = Q_s \;\Leftrightarrow\; C_d A_c \sqrt{\left(1 - \frac{A_1}{A_2}\right)\Delta p_p} = C_d A_s \sqrt{\frac{2}{\rho}\frac{A_1}{A_2}\Delta p_p} \tag{3.5}$$

$$A_c = A_s \sqrt{\frac{A_1}{A_2 - A_1}} \tag{3.6}$$

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In Fig. 12, the ratio between the flow areas A_c/A_s and pressure drop $\Delta p_c/\Delta p_s$ of compensator and valve are reported versus the pilot area ratio of the compensator, A_1/A_2, which is a design parameter. A value of 0.9 has been chosen for this ratio, as to minimize pressure drops on the compensator when it is fully open. This results in an A_c/A_s ratio equal to 3. Considering the flow area of EHR control valve at maximum opening, the required area of the compensator has been obtained from equation (3.6). The Amesim model of the designed valve is shown in Fig. 13.

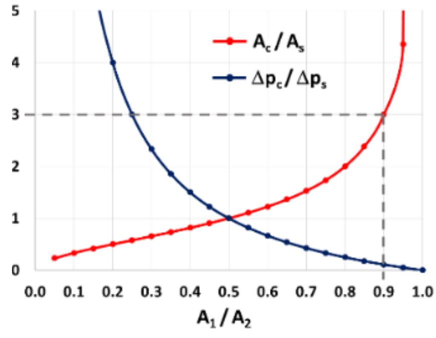

Fig. 12: Flow areas and pressure drops versus pilot area ratio

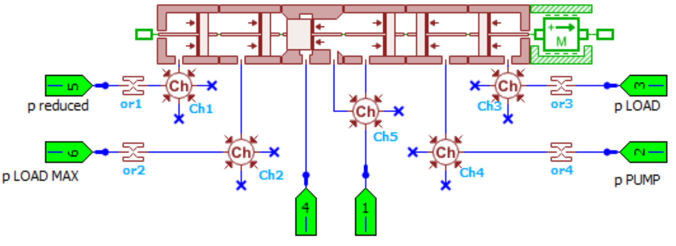

Fig. 13: AMESim model of the flow sharing compensator

4. Results

To perform an effective comparison, it is important to consider the real operating conditions of the vehicle. For this reason, the investigated architectures, summarized in Table 1, have been simulated and compared considering the experimental end-field manoeuvre. This cycle has been chosen since it reproduces rather faithfully a typical ploughing on-field operation, involving the entire hydraulic system of the tractor.

The same experimental loads and boundary conditions have been imposed into the models. In Fig.14, the displacement of steering and hitch cylinders and the flow rate at the remote actuated sections are reported, for all the investigated architectures. It's worth observing that the new solutions here proposed achieve performance equivalent to that of the baseline architecture.

Results are presented in Fig. 15, in terms of mean power consumption of the system: for each architecture, the average mechanical power required at the input shaft of the pumps to perform the duty cycle is shown. Energy saving is expressed as percentage of the baseline architecture power consumption. The VPM strategy is the simplest to be integrated since it only requires the replacement of the pump load sensing compensator. The energy saving is

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about 6%, without any change in the circuit layout; the VPM REC solution without separating the utilities does not allow further saving, due to the presence of the priority valve. By removing it and separating the actuators, 13% of power saving has been obtained with the VPM architecture, mainly because the steering can work at lower pressure. The more efficient pump margin regulation map of the VPM REC separated architecture allows to reduce the consumption by approximately 17%. Flow matching solutions are, as expected, the most efficient ones, and a maximum power saving of 22.5% is obtained with the flow sharing architecture.

TABLE 1. Investigated architectures summary

Architecture	Description
BASELINE	Load Sensing, Fixed Pump Margin, Unified Layout
VPM	Load Sensing, Variable Pump Margin (Map 1), Unified Layout
VPM REC	Load Sensing, Variable Pump Margin (Map 1), Electronic Compensation, Unified Layout
VPM (Separated)	Load Sensing, Variable Pump Margin (Map 1), Separated Layout
VPM REC (Separated)	Load Sensing, Variable Pump Margin (Map 2), Electronic Compensation, Separated Layout
EFM REC (Separated)	Flow Matching, Electronic Compensation, Separated Layout
EFM FS (Separated)	Flow Matching, Flow Sharing Compensators, Separated Layout

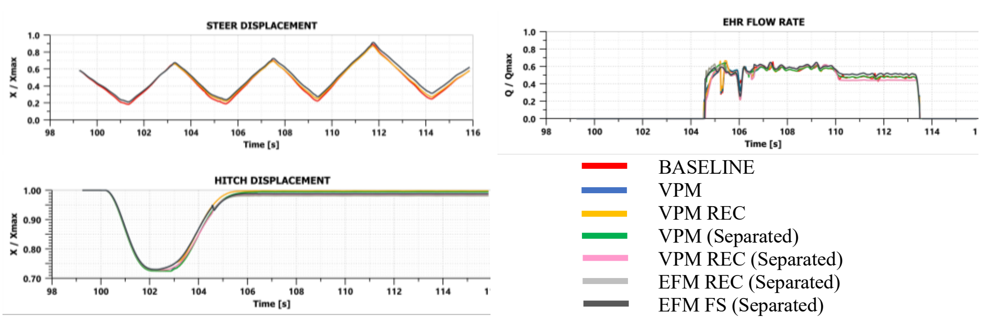

Fig. 14: Results: performances comparison, End-Field test

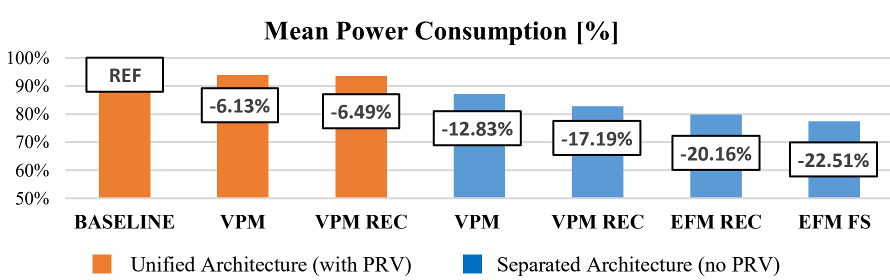

Fig. 15: Results: power consumption comparison, End-Field test

Conclusions and Future Works

This work investigated new possible energy saving architectures for the hydraulic remote auxiliary utilities of an agricultural tractor. Alternative solutions to the standard load sensing multi-actuators system have been studied and energetically compared through modelling and simulation. Experimental data have been used to validate the baseline model and to define a duty cycle representative of a typical on-field working condition of the machine. A power flow and dissipation analysis of the standard architecture have shown that priority valve and local pressure compensators play a significant role in affecting the energy consumption. An alternative circuit layout, in which the priority valve has been removed, and steering has been separated from the rest of the circuit, has been proposed.
Three main solutions have been investigated: i) an electro-hydraulic load sensing system, in which the pump margin is dynamically regulated according to flow request; ii) an electronic strategy for pressure compensation; iii) an electronic flow matching architecture, with flow sharing functionality. The same performance, in terms of user's displacement and flow rate has been obtained with the standard and the new architectures. The energetic comparison has been made considering the average mechanical power at the engine shaft. Results demonstrate that a good power saving can be achieved, ranging from 6% of the VPM solution, without splitting the actuators, to 22.5% of the separated layout with the EFM and flow sharing architecture. Since simulation results are encouraging, physical prototyping of the most interesting solution will be investigated.

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