Secondary Control Based Hydraulic Architectures For Mobile Applications

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This paper proposes a new approach for hydraulic architectures in compact mobile machines based on a secondary control logic. With this approach, travel and implement functions can be supplied with a single open circuit pump and the travel drive itself can be realized without any directional valves. A small sized tractor shows the feasibility of the concept. Drive and implement hydraulics use cases were tested simultaneously. To achieve a well-balanced drive performance, the dynamic behavior of both pump and motor were analyzed, optimized, and evaluated. As a result, it is shown that the dynamic performance of the system approach meets the requirements for use in hydrostatic drive and implement hydraulics.

Keywords: Secondary control, electronic open circuit pump control, simplified hydraulic architectures, energy recovery, hydrostatic drives
Target audience: Mobile Hydraulics

1 Introduction

The concept of secondary control is widely established in stationary hydraulic applications such as in test benches, presses, or offshore applications [1]. For mobile applications, however, the use of secondary controlled concepts is not common as they require complex control algorithms, hydraulic accumulators, and expensive hydraulic and electronic components. Guo et. al. [2] describe a secondary controlled hydrostatic travel drive for a wheel loader using hydraulic accumulators. Busquets et. al. describe in [3] a robust control method for a secondary controlled swing drive of a compact excavator. The system contains a hydraulic accumulator with variable pressure to reduce losses due to a constant high-pressure situation. The primary unit supplies fluid solely to the swing drive. There is no information whether the pump/primary unit is used to supply additional implement or drive functions. Another study about secondary controlled drives were done by Li et. al [4]. In this paper a variable pressure control without hydraulic accumulators is proposed for the secondary controlled circuit in the loading test bench for driving heads of rotary drilling rigs. The objective of the variable pressure is to reduce volumetric losses and increase energy feedback efficiency. The improved efficiency comes at the expense of the pressure control bandwidth which leads to a significant deterioration of the torque control performance. In contrast to conventional hydrostatic drives in mobile machines, in secondary controlled drives a pressure coupling is required meaning that the motor displacement is proportional to the drive torque and not to the drive speed. The speed is controlled in a secondary manner by adjusting the motor displacement, and therefore the motor torque, such that a balance is found between load torque and drive torque. Advantages of secondary controlled architectures is the easy integration of multiple rotary consumers on a single pressure line without any directional valves, the possibility to recover energy in deceleration phases and the combination of drive and implement hydraulics into one hydraulic circuit.

2 System architecture

In contrast to the known approaches from [2], [3] or [5], the proposed architecture in this paper uses variable pressure and does not need hydraulic accumulators. A pressure adaptation is also foreseen in [3]. Due to the use of a hydraulic accumulator the pressure limits are not freely selectable and stays in the operation limits of the accumulator (e.g. between 100 – 350 bars). Further it is unclear whether this approach is extendable to additional

implement or drive functions. As mentioned in [3] and [4] hydraulic accumulators have a positive effect on system damping and control performance. The pressure control performance and stability of the primary unit significantly influences the suitability of the proposed architecture. The advantage of the proposed architecture is that the pressure can be adjusted according to efficiency aspects or the implement hydraulics pressure demand; an easier integration of implement hydraulics is possible with fewer components and reduced system losses. Both pump and motor must be able to over-center for 4-quadrant operation with only one high pressure side. This means that pump and motor work in pump and motor mode depending on the drive direction and torque request. In this paper the primary unit is called pump and the secondary unit motor. Additionally, a main control valve is supplied by the pump. This concept is shown in **Figure 1** and is known from the literature as *secondary controlled architecture with impressed pressure*. This concept can be extended with multiple motors on the same pressure line for additional rotary consumers.

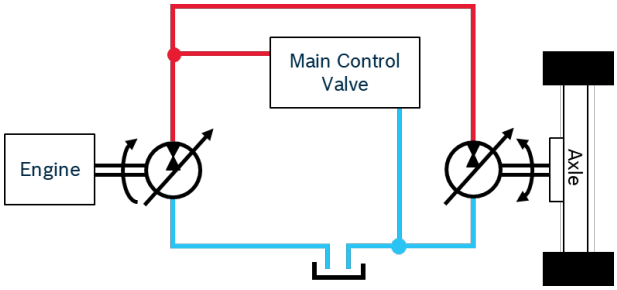


Figure 1: Secondary controlled architecture for drive and implement hydraulics

3 Requirements for mobile machines

A key element to utilizing secondary control in mobile applications is the availability of hydraulic and electronic components which fit to the cost, space, and technical requirements of mobile machines. This requires cost optimized hydraulic pumps and motors for mobile machines, a high degree of electronic integration with reduced space requirements, and dynamic control concepts of said pumps and motors.

3.1 Components

The employed axial piston pump with an electronic control for mobile applications was introduced in [6] and can be seen in **Figure 2**. The medium pressure axial piston pump A10VO eOC needs less space and is cost optimized compared to the electronically controlled industrial A10 VSO SYDFE pump. The basic A10VO pump is widely used in mobile machines.

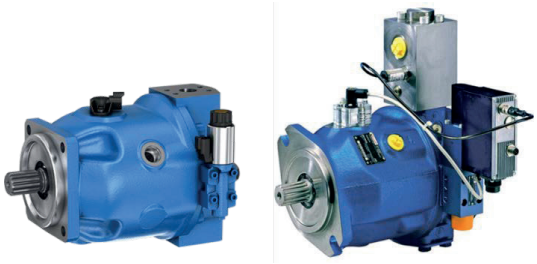


Figure 2: Electronically controlled axial piston pump for mobile (left, A10 VO eOC pump control) and industrial applications (right, A10 VSO SYDFE)

The “Electronically Controlled Open Circuit Pump” (eOC pump control) comprises an electronically controlled pump in combination with a control software. The basic eOC configuration contains an electro-proportional 3/2-way valve and an ECU running the eOC control software. Additionally, a swivel angle sensor and a pump pressure sensor are used. The control modes (pressure, torque and displacement control) are realized by software, can work in parallel, and can be combined (see **Figure 3**). An additional speed sensor is used on the motor to measure the motor speed. The control concept for the eOC units is based on a cascaded closed loop control architecture. Three

parallel-arranged outer control loops, one for angle control, one for pressure control and one for torque control, calculate the setpoints for the inner control loop. A selector function prioritizes the most relevant result of these three controllers and forwards it to the inner control loop. The output of the controller cascade is a duty cycle value that drives a power stage, generating the current for the pump solenoid. To have a high dynamic performance no dampening orifice is used in the pump control valve. Dampening is purely done by software controller gains. The controller gains are chosen work point dependent to have an optimal performance while ensuring stability in all use cases.

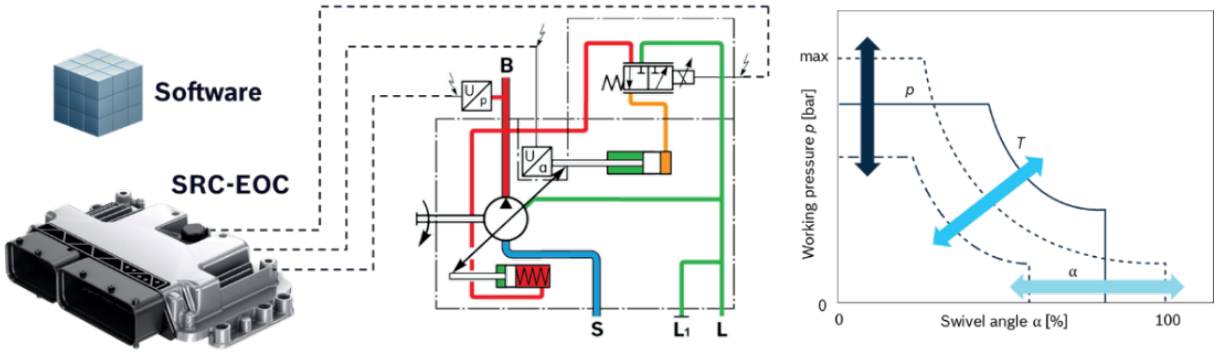


Figure 3: Hardware set-up and control modes of the eOC pump control

3.2 Pump and motor dynamics

In **Figure 4** the step response characteristics for swivelling out/in of a 45cc hydrostatic unit is shown. With short response delay and fast rise time of less than 60 ms for a step from 10% to 90% angle command, the pump is capable of realizing secondary control concepts for pump and motor.

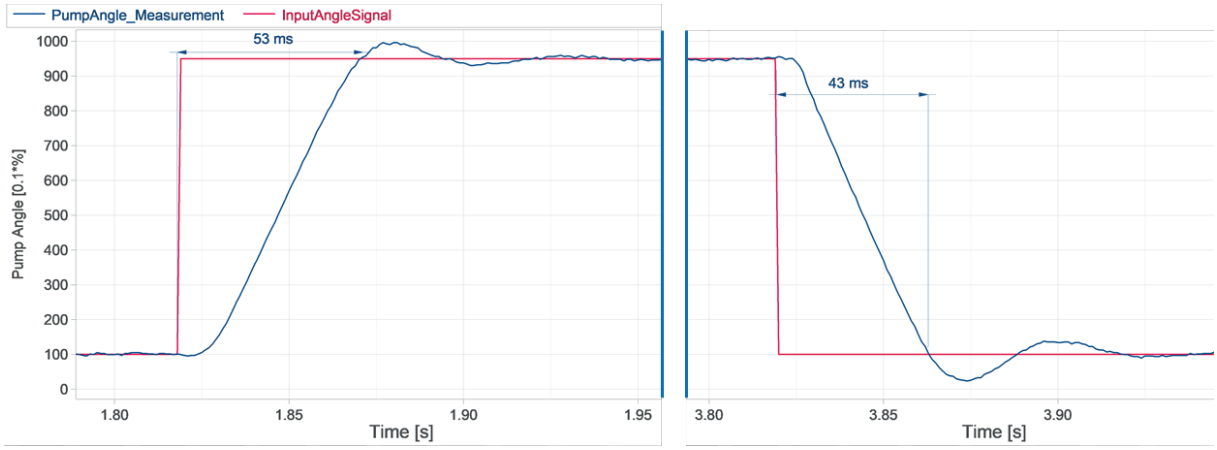


Figure 4: Measured angle control step response behavior with an A10 VO 45cc eOC-Pump

3.3 Torque based drive strategy

For the drive train, a torque-based approach similar to [7] is used. Depending on the drive pedal position and vehicle speed a torque demand is calculated. A sample torque map is shown in **Figure 5**. The drive behavior results in a load sensitive and automotive drive behavior. The operator adjusts the drive torque demand through the drive pedal. Depending on the driving resistance an equilibrium at a defined vehicle speed will be achieved. The falling characteristic of the torque map leads to a speed sensitivity of the drive train. As an example, if the machine is supposed to drive downhill the driving resistance would become smaller and the machine would accelerate downwards when keeping the drive torque constant. Due to the falling characteristic, the drive torque is automatically reduced as a result the acceleration in downhill direction is limited. For negative speeds an increasing positive counter torque is generated to prevent an acceleration downhill in the backward direction.

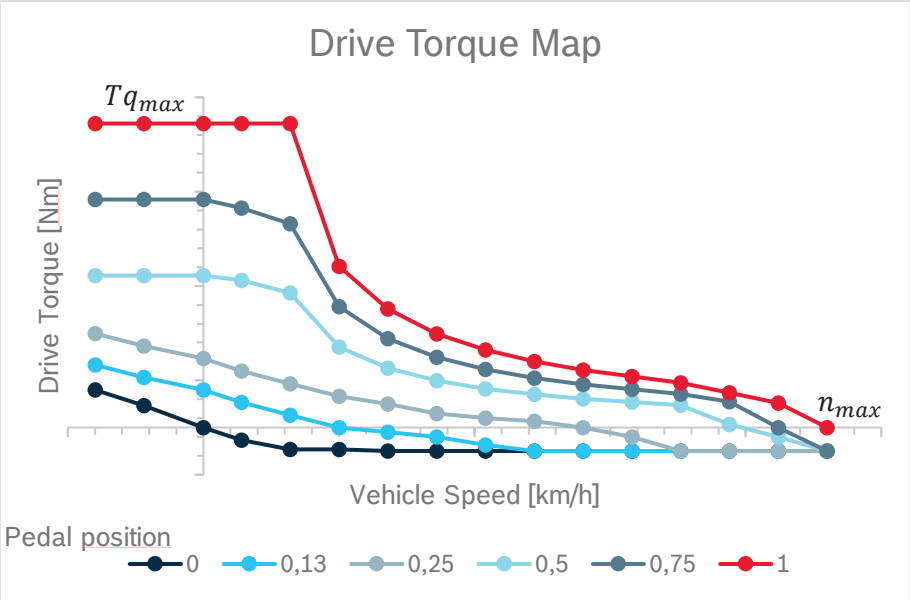


Figure 5: Drive Torque Map

3.4 Pump control

The primary unit – the pump is pressure controlled. According to Equation (1) the pressure target value of the pump $p_{\text{pmp,tgt}}$ is the maximum of the pressure demand from the implements p_{work} or the needed drive pressure p_{drv} to achieve the drive torque demand.

$$p_{\text{pmp,tgt}} = \max(p_{\text{drv}}, p_{\text{work}}) \tag{1}$$

The flow balance between pump and motor as well as efficiency aspects are considered for the target drive pressure calculation. The drive pressure has a lower limit $p_{\text{min,drv}}$ and an upper limit $p_{\text{max,drv}}$ as shown in Equation (2).

$$p_{\text{min,drv}} < p_{\text{drv}} < p_{\text{max,drv}} \tag{2}$$

The drive pressure can be chosen such that its minimized and the motor must be operated at maximum displacement to achieve the torque demand Tq_{drv} according the equation (3).

$$p_{\text{min,drv}} = \frac{Tq_{\text{drv}} * 2 * \pi}{\eta_{\text{mh}} * VgMot_{\text{flow,lim}}} \tag{3}$$

This pressure command calculation leads to optimally efficient operation. The flow balance is considered by a calculation of maximum possible motor displacement $VgMot_{\text{flow,lim}}$ in accordance with Equation (4) in order to not exceed the available pump flow considering the pump speed n_{pmp}, motor speed n_{mot}, pump displacement Vg_{pmp}, hydraulic efficiency η_{hyd}, and implement hydraulics flow Q_{work}.

$$VgMot_{\text{flow,lim}} = \frac{n_{\text{pmp}} * Vg_{\text{pmp}} * \eta_{\text{hyd}} - Q_{\text{work}}}{n_{\text{mot}}};\ VgMot_{\text{flow,lim}} < VgMot_{\text{max}} \tag{4}$$

The upper limitation of the drive pressure $p_{\text{max,drv}}$ is calculated using the maximum available torque at given motor speed $Tq_{\text{max}}(n)$ (Equation (5)).

$$p_{\text{max,drv}} = \frac{Tq_{\text{max}}(n) * 2 * \pi}{\eta_{\text{mh}} * VgMot_{\text{flow,lim}}} \tag{5}$$

In this operating condition the motor displacement would be minimized, and the pressure maximized. An advantage of this operation is that the pump flow is also minimal and thus the pump has enough flow reserve to react to rapid implement hydraulics flow demands. The target pressure for implements $p_{\text{work}}$is derived from the

working load. In electronic load sensing architectures, it is calculated by the desired differential pressure Δp and measured load sensing pressure p_{LS} as shown in Equation (6).

$$p_{\mathrm{work}} = p_{\mathrm{LS}} + \Delta p \tag{6}$$

For the pump, a torque controller works in parallel to limit the engine torque in case of available engine power constraints. Further, the pump is operating in the pressure control mode in the negative quadrant. When changing the torque direction in braking phases, the motor works as a pump and the pump must work as a motor to take back the fluid and control the pressure to the desired value.

The capability of the hydrostat units to fulfill dynamically changing drive and work requirements with enough dampening in the system to master oscillations is crucial for the feasibility of the presented static concept.

3.5 Motor control

With the given target pump pressure $p_{\mathrm{pmp,tgt}}$, a target motor displacement $VgMot_{\mathrm{tgt}}$ according to Equation (7) can be calculated to fulfill the torque demand Tq_{drv}.

$$VgMot_{\mathrm{tgt}} = \frac{Tq_{\mathrm{drv}} * 2\,\pi}{p_{\mathrm{pmp,tgt}} * \eta_{\mathrm{mh,mot}}} \tag{7}$$

4 Validation on prototype machine

4.1 Machine set-up

The prototype machine for the performance validation is a small-sized tractor, shown in **Figure 6** equipped with a 45cc eOC-pump and a 45cc eOC-motor both working in two quadrants.



Figure 6: Prototype machine for the secondary controlled drive in open circuit

The secondary unit is optimized for motor operation and an additional speed sensor is integrated into the motor housing. A closed center main control valve equipped with a load pressure sensor is connected in parallel to the pressure line and behaves as flow and pressure disturbance for the drive train. The implement hydraulics load is simulated with a variable orifice. Further, it is possible to electronically control the flow through this main control valve. The axial piston units are electronically controlled with a single ECU controlling both the pump and the motor. A second electronic control unit is installed which manages the secondary controlled drive strategy. An electronic foot pedal, a drive direction switch, and an electronic joystick for the implement hydraulics are installed to receive the driver demands for the drive and implement functions. The system configuration is depicted in **Figure 7**.

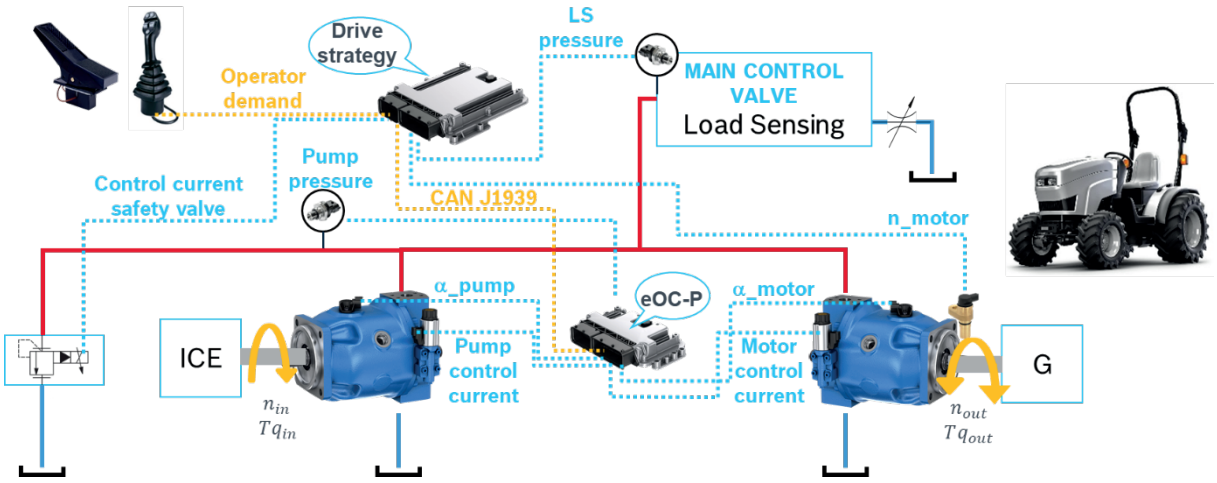


Figure 7: System overview of prototype vehicle

4.2 Test cases

Drive and implement function test cases were performed individually and in combination on the prototype machine. To achieve an acceptable drive behavior the pump and motor control parameters, as well as the drive strategy, meaning pressure and torque calculation, were optimized. The following test cases are analyzed in this paper.

1) **Reversing:** The machine is accelerated to a certain speed and the driver switches from forward direction to reverse direction. After decelerating to standstill, the machine is accelerating immediately in the reverse direction and the driver switches back from reverse direction to forward direction.

2) **Flow disturbance during constant driving**: The machine is driving at a defined speed and the operator actuates a joystick to demand a flow through the main control valve for an implements function. The operator then releases the joystick to stop the implement function.

3) **Flow and pressure disturbance while driving at torque limit of the pump**: This test case is similar to the second test case. In this test case the requested load pressure on the implement side is increased as well such that the torque limitation of the pump is reached.

4.3 Measurement results

The result for the reversing test case can be seen in **Figure 8**. The operator pushes the pedal up to ~70% position (light blue line first diagram) to generate a torque demand for the drive (dark green line first diagram). A target pump pressure (blue line second diagram) and a target motor displacement (blue line third diagram) are requested to achieve the target drive torque. Pump and motor swivel out to fulfill their requested values. The machine starts to accelerate up to a speed of 70% of v_{max}. At ~13s the drive direction (magenta line first diagram) is switched from forward to reverse. The target torque changes its sign and becomes positive to decelerate the machine while driving forward. To change the torque direction in the open circuit, the motor swivels to the second quadrant meaning from negative displacement -22% to positive displacement +12%. As the motor is working in pump mode, the pump must work in motor mode to keep the pressure at the desired level and thus it swivels from +55% to -45%. A pressure peak at ~60% of p_{max} can be observed when the motor changes its displacement from negative to positive because the pump is still in pump mode due to the dynamic reaction of the pressure controller. This pressure peak can be further reduced by optimization of the control parameters and drive strategy. A small speed ripple (red line in first diagram) can be observed too due to the pressure peak and thus due to the peak in the actual torque compared to the target torque. This speed ripple is not negatively recognized by the driver, rather it is desired to sense an immediate reaction of the machine to a changing drive direction request. The machine is then linearly decelerated to standstill and then accelerated in the reverse direction. The transition trough the standstill/

0 speed is fast and continuous. The drive behavior is generally very dynamic and without noticeable jerks common in closed-circuit hydrostatic drives.

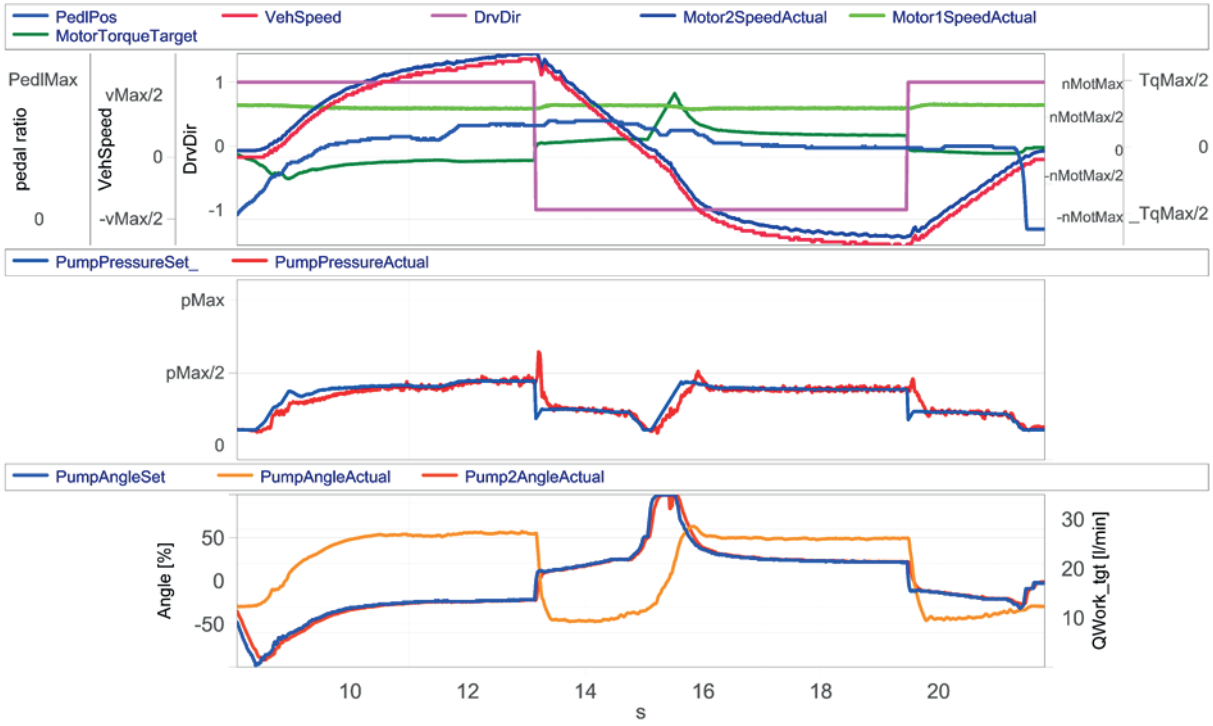


Figure 8: Test result for test case 1- Reversing

The validation result of test case 2 can be seen in **Figure 9**. In this use case, the operator requests ~50% drive pedal position (light blue line first diagram) and the vehicle is accelerating to a speed of ~50% max vehicle speed (red line first diagram). The actual pressure (red line second diagram) follows the pressure setpoint (blue line second diagram) whereby in parallel the motor angle setpoint (red line third diagram) and actual angle (blue line third diagram) are adjusted according to the torque demand (dark green line first diagram). The dotted black line marks the instance where a flow rate of 30 l/min is requested (black line third diagram) for the implements via joystick actuation. The pressure balance is disturbed for a short time. Within this time, the pump angle is increased from 58% to 82%, the motor angle is decreased from -38% to -24% and the pressure is increased from 35% of p_{max} to 54% of p_{max} such that the motor torque is kept constant. The motor speed changes by less than 10% of the current speed and is compensated such that this speed change is not recognizable by the operator.

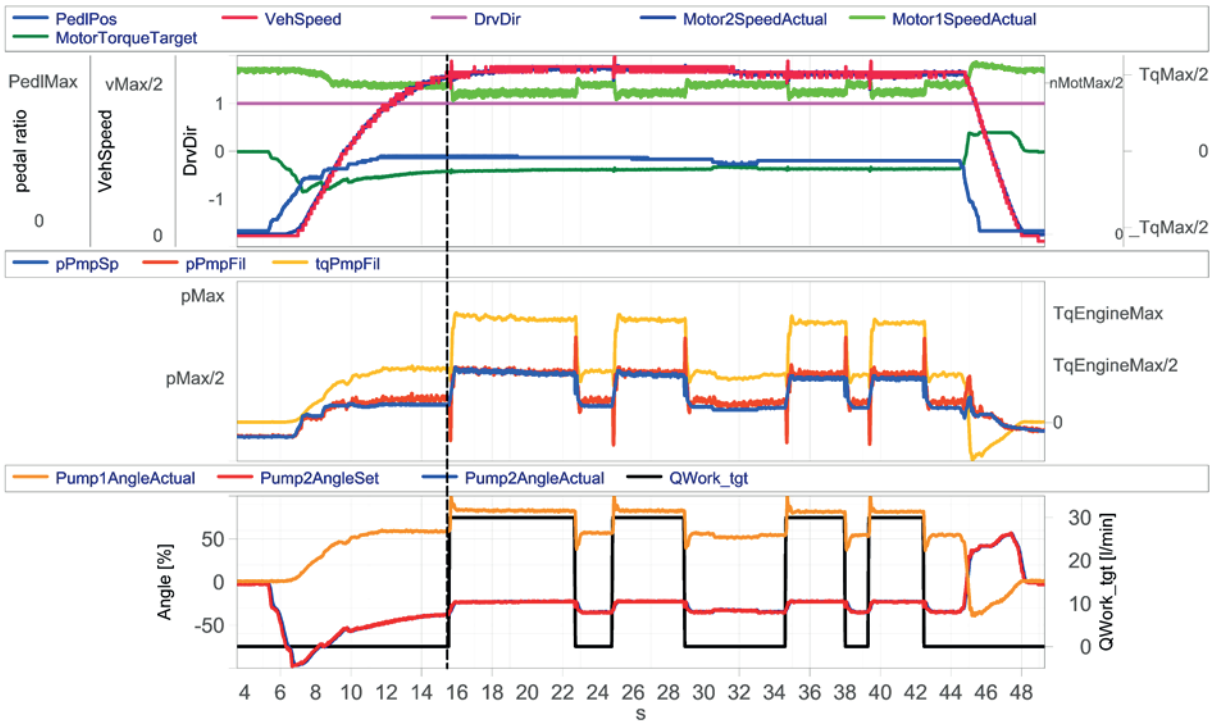


Figure 9: Test result for test case 2 - Constant driving speed and implement hydraulics request of 30 l/min

In **Figure 10** the test result for test case 3 is shown. It is similar to the previous test. In this test case additionally, the load pressure is increased electronically to 85% of p_{max}. The actual load pressure is smaller (around 25% of p_{max} trough the orifice) but it is increased electronically to simulate a high load from the implement function. At 42.5s the pump swivels out to supply the implement function and to achieve the new pressure target. As the pump approaches the maximum displacement, the actual pressure drops to 19% of p_{max} and thus the motor speed drops by around 5% of n_{max}. A steady state pressure of 65% of p_{max} is achieved. The target pressure of 85% of p_{max} cannot be realized because the pump is in torque limitation and it cannot increase its displacement to achieve a higher pressure. In order not to stall the engine, the maximum available engine torque $Tq_{Engine,max}$ was requested as the eOC pump torque setpoint. The continuous engine speed drop (light green line first diagram) in torque limitation mode is around 8% at current engine speed. Due to the saturation of the pump, the desired drive torque is also not achieved. As a result, the vehicle speed/motor speed decreases from 55% to 50% of n_{max}. When stopping the implement function at 46.4s, the vehicle speed stays constant, and it is not disturbed because no pump limitation is reached. In this test case it is shown that the pump switches from pressure control to torque control mode. Further it is shown that the drive demand cannot be fully fulfilled when either torque or flow limit of the pump is reached and therefore the drive becomes slower similar to a conventional machine architecture with 2 circuits for drive and implement function under power limitation of the engine.

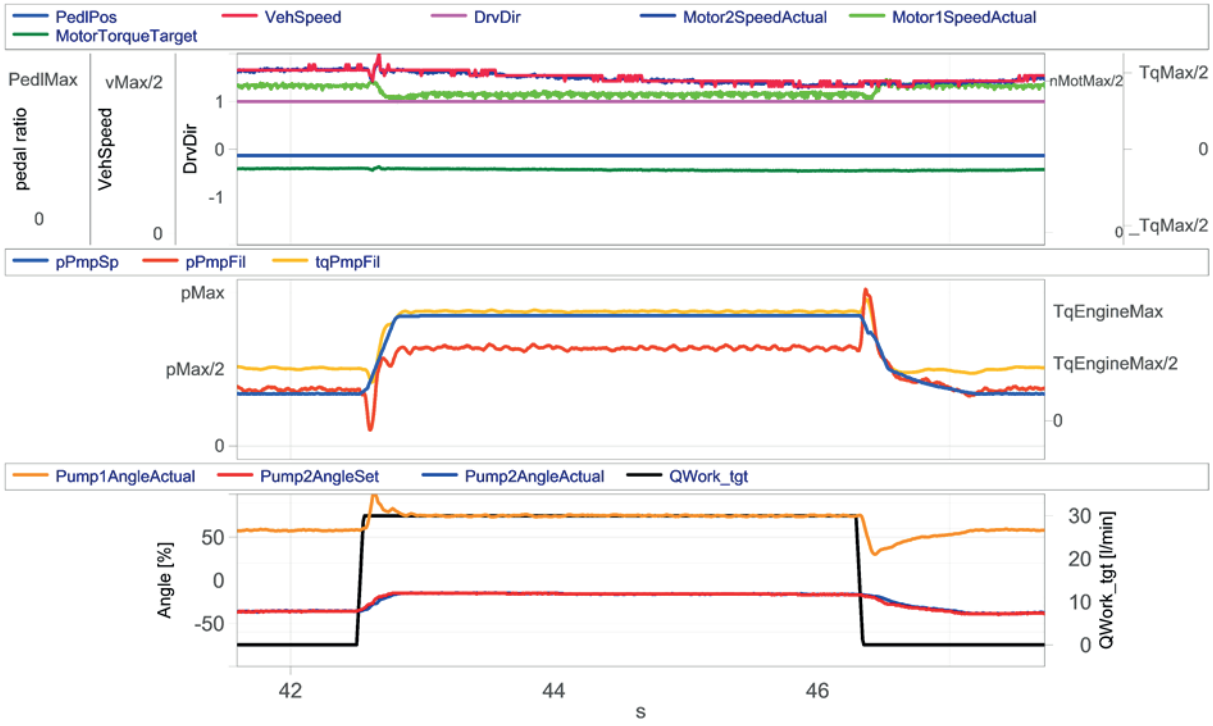


Figure 10: Test result for test case 3 - Constant driving speed and implement hydraulics request of 30 l/min and 200 bar at pump torque limitation

4.4 Evaluation of measurement results and comparison to conventional systems

In the measurements it was shown that the control performance for pressure control and torque limitation on the primary unit as well as the angle control performance on the secondary unit is good enough to achieve a similar drive behavior known from closed circuit applications. The system can react to rapid drive demand changes without delay (reversing) and disturbances due to additional implement functions without noticeably deteriorating the drive behavior (speed disturbance < 10%).

A quantitative analysis of the efficiency difference between the proposed architecture and a conventional drive system was not performed. Fur such an analysis the component efficiencies of primary and secondary units, system losses (standby losses of implement circuit and drive circuit, flushing circuits) and regeneration/recuperation potentials must be considered. Compared to the original system with a closed-circuit hydrostatic drive, this architecture uses medium pressure open circuit components. That's why the maximum pressure and maximum allowed speeds are smaller which leads to smaller transmission ratios of the drive train. However, this is not an architectural problem rather it is a matter of component availability with similar operating limits. With a correct sizing of primary and secondary unit displacements and adjusting the transmission ratio of the gearbox, the same Tq_{max} and n_{max} demands could have been achieved on this machine.

5 Summary and outlook

In this paper an approach of secondary controlled motors in a hydraulic open circuit is introduced for mobile machines with a single pump to supply the drive motor attached in parallel to an implement valve. The applicability was proven on a prototype machine. Secondary controlled architectures could be applied to many mobile machines to utilize advantages like saving separate hydraulic circuits, using energy recovery in deceleration phases, or improving control performance of rotary consumers. Three examples are shown in **Figure 11** where those advantages can be utilized. The figure shows a slew control in secondary control to recover energy in a mobile excavator. Secondary controlled track drives in compact excavators could improve the drive performance. The number of circuits and components can be drastically reduced in a road roller when using secondary control.

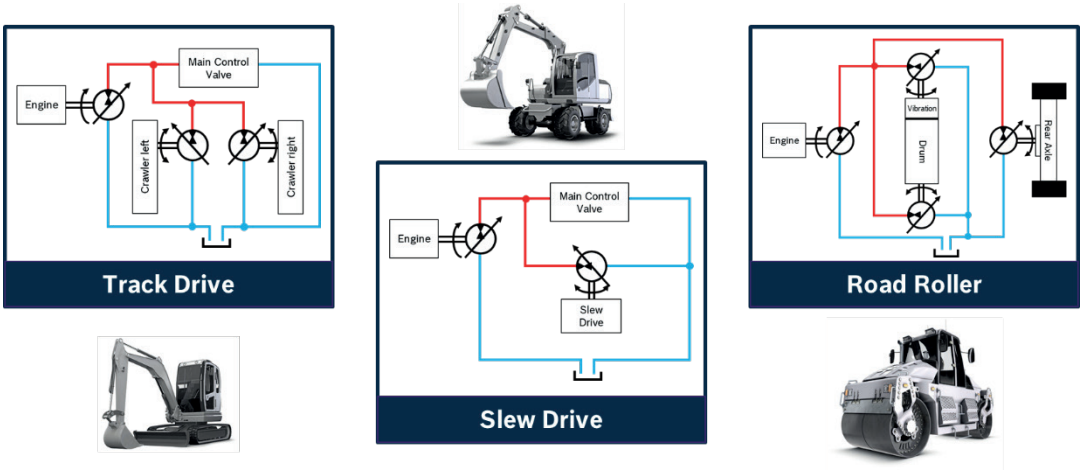


Figure 11: Further examples and architectures for secondary control in mobile machines

The proposed approach and respective electronically controlled components meet the requirements in mobile machines for parallel use in the open circuit as a hydrostatic drive and implement hydraulics by using secondary control on the drive motor side. Secondary controlled architectures can fundamentally change mobile hydraulic architectures. Hydraulic circuits can be dramatically simplified and on machine level, thus cost savings will be possible. In contrast to the state of the art, secondary control is realized without hydraulic accumulators and is realized with dynamically optimized electronically controlled axial piston units for mobile applications.

Nomenclature

Variable	*Description*	*Unit*
$p_{\mathrm{pmp,tgt}}$	Target pump pressure	[bar]
p_{drv}	Target drive pressure	[bar]
p_{work}	Implements hydraulic target pressure	[bar]
$p_{\mathrm{min,drv}}$	Minimum driving pressure	[bar]
$p_{\mathrm{max,drv}}$	Maximum driving pressure	[bar]
p_{LS}	Load sensing pressure on implements	[bar]
Δp	Desired differential pressure for load sensing control	[bar]
Tq_{drv}	Target drive torque	[Nm]
$\eta_{\mathrm{mh,mot}}$	Mechanical efficiency of motor	[-]
η_{hyd}	Volumetric efficiency	[-]
Q_{work}	Implement hydraulics actual flow	[l/min]
$VgMot_{\mathrm{flow,lim}}$	Available motor displacement in pump saturation	[ccm]
n_{pmp}	Pump speed	[rpm]
n_{mot}	Motor speed	[rpm]
$VgMot_{\mathrm{max}}$	Maximum motor displacement	[ccm]

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