Pressure compensator control – a novel independent metering architecture

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Abstract

This contribution presents an operating strategy for a novel valve structure for mobile machines' working hydraulics which combines the flexibility and energetic benefits of individual metering with the functionality of common primary pressure compensation (IPC). The aim is to set up a system that uses a minimal amount of sensors and simple control algorithms. A control strategy theoretically described in /1/ is modified to facilitate the practical implementation on a mini excavator implement as a test rig. This test rig consists only of components that are currently available off-the-shelf to show that it is possible to develop an individual metering system under these economic restrictions. The novel is more energy efficient than common flow sharing systems but provides the same functionality. The control algorithm is experimentally evaluated in terms of functionality and energy consumption. Simulations show potential for further improvements.

KEYWORDS: independent metering, mobile working machines, electrohydraulic systems, control strategy

1. Introduction

Manufacturers of mobile machinery as well as suppliers find themselves persistently confronted with increasing requirements regarding energy efficiency, safety and operator-comfort. This demands for continuous development of control and system architectures. Control systems in mobile machinery provide hydraulic power to numerous parallelly operated actuators. For small and medium sized machines typically one single pump supplies several actuators. This inherently leads to throttling losses in the inlet paths of the lower loaded actuators. The mechanical coupling of inlet and outlet throttling edge causes further avoidable losses. Requirements on controllability of pulling loads and energy consumption lead to a design conflict regarding the valve spools. For systems with individual metering of the inlet and outlet this conflict is avoided. Furthermore individual metering opens up for enhanced

operation modes like high pressure regeneration. This reduces the energy losses at lower loaded consumers. To increase acceptance in industry the production costs must be kept low and the control algorithm as simple as possible. Therefore a control strategy using only one pressure sensor in the common supply pipe and spool stroke sensors at the IPC is developed and implemented on a test rig that consists only of commercially available components. Many other approaches to individual metering use two pressure sensors per cylinder /2–5/, which negatively effects system availability because the increased risk of failure. The pressure compensator's operation point (spool position or pressure drop) gives indication about a consumer's load situation using just one sensor /1; 6/. In this paper measurement of the IPC position is favoured.

The used valve structure is the outcome of preliminary works at the IFD and will be briefly described in section 2. A basic control strategy, that mirrors the IPC's function with the meter out throttle edge, has also been developed at the IFD. Theoretical investigations of the valve system's behaviour in section 3 show that this strategy needs to be modified to facilitate the practical implementation. A control algorithm is derived from the modified strategy and implemented on an ECU. In section 4 functionality and energy consumption are evaluated in simulation and experiment on a mini excavator implement test rig.

2. Design of hydraulic system and test rig setup

The valve structure shown in **Figure 1** is used to actuate the boom and stick cylinder of an excavator implement. Individual metering systems are multiple input-multiple output systems (MIMO). Usually these require complex multi-variable control strategies. Previous research at the IFD has shown that an individual pressure compensator (IPC) in the inlet flow path is advantageous to decouple piston load force and velocity. This enables single-variable control approaches /1; 7/.

The resulting valve arrangement consists of two proportional 2/2 way valves for throttling and four 2/2 way switching valves to set the flow paths. The individual pressure compensator and the throttling valves are equipped with displacement encoders. The structure depicted in Figure 1 allows individual throttling of both cylinder chambers and their connection either to high or to low pressure. The IPC always throttles the flow from the pump in order to regulate the flow through the inlet throttle edge into the inlet cylinder chamber. An ECU commonly used in mobile applications actuates the electrohydraulic components. The measurement signals are delivered to the ECU and captured by a data acquisition system. The user operates the excavator implement with two joysticks transmitting their data to the ECU via CAN.
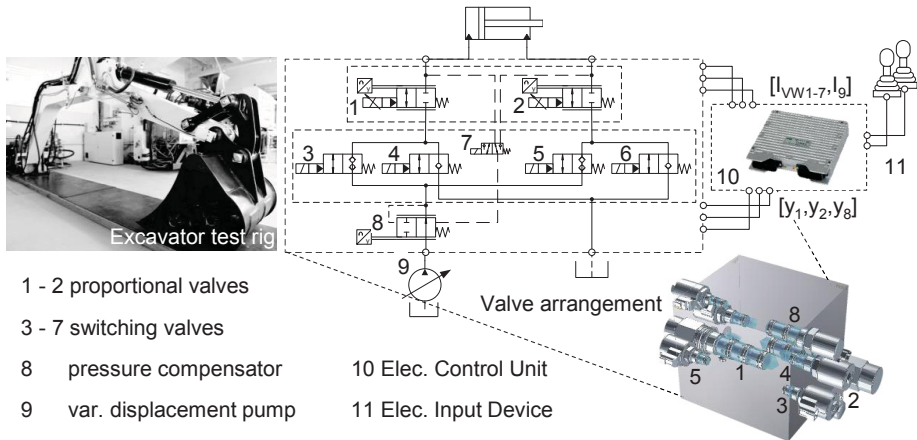
Figure 1: hydraulic circuit for one consumer

3. Theoretical analysis and control strategy

The first part of this section is dedicated to a theoretical analysis of the proposed valve arrangement's static behaviour. Afterwards the control strategy given in /1/ will be briefly explained and refined based on the system analysis given before. This leads to the development and implementation of the control algorithm.

3.1. Static behaviour of independent metering circuit with primary pressure compensator

The static behaviour of the controlled system - a double acting differential cylinder actuated with individual throttling edges and a primary IPC in the inlet path - is theoretically investigated. For this analysis the circuit can be simplified according to **Figure 2.**
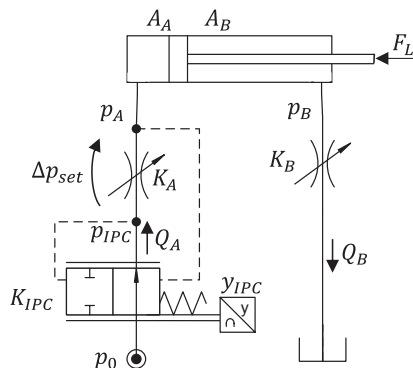

Figure 2: Simplified hydraulic circuit

To design a suitable controller it is necessary to know the relation

$$K_{IPC} = f(K_A, K_B, F_L) \tag{1}$$

between the openings of the three throttling edges involved in the hydraulic circuit.

K summarizes the valve flow constants according to the turbulent throttle equation for a proportional valve:

$$Q(y, \Delta p) = \alpha \cdot A(y) \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{\Delta p} = K(y) \cdot \sqrt{\Delta p} \tag{2}$$

The following equations depict the relevant behaviour of the interacting components. The pressure compensator is described with

$$Q_A = K_{IPC}(y_{IPC}) \cdot \sqrt{p_0 - p_{IPC}} \tag{3}$$

while the volume flow through the inlet throttling edge is given by

$$Q_A = K_A(y_A) \cdot \sqrt{p_{IPC} - p_A} \quad . \tag{4}$$

Assuming that the IPC operates within its control range the pressure drop over the inlet throttling edge K_A matches

$$p_{IPC} - p_A = \Delta p_{set} \tag{5}$$

which is set with the spring adjustment in the IPC valve. The flow equation for the outlet throttling edge is similar to the inlet edge:

$$Q_B = K_B(y_B) \cdot \sqrt{p_B} \tag{6}$$

The cylinder delivers the piston's force balance

$$0 = p_A \cdot A_A - p_B \cdot A_B - F_L \tag{7}$$

and the relation between inlet and outlet volume flow:

$$\frac{Q_A}{A_A} = \frac{Q_B}{A_B} \tag{8}$$

Putting equations (3) to (8) together results in:

$$K_{IPC}(K_A, K_B, F_L) = K_A \cdot \sqrt{\frac{\Delta p_{set}}{p_0 - \frac{1}{A_A} \cdot \left(F_L + \Delta p_{set} \left(\frac{K_A A_B}{K_B A_A}\right)^2 \cdot A_B\right) - \Delta p_{set}}}$$

(9)

3.2. Basic control strategy and refinement

This section is started with the description of a basic control strategy that was theoretically developed at the IFD in previous research. Obstacles to a practical implementation of this strategy will be pointed out and circumvented with the help of a refinement.

Basic strategy. The basic idea of the approach described in /1/ is to set the consumers velocity with the inlet throttling edge K_A in an open-loop manner while controlling the outlet edge K_B in a closed loop in such a way that the IPC is nearly fully open regardless of velocity and load force, thus shifting the inlet pressures of all consumers to the same level. This simple concept has numerous benefits:

- The strategy uses the IPC as a sensor for detecting the load situation and does not need any pressure sensors.
- With the IPC almost completely open the inlet chamber pressure is almost as high as the supply pressure regardless of the load situation. With a reasonably high supply pressure a pulling load can be moved securely at the desired velocity without causing cavitation in the inlet chamber. Energy inefficient counterbalance valves are not necessary.
- There is no need to detect the load force direction.

System behaviour and obstacles. The diagram at the top of **Figure 3** shows the IPCs opening K_{IPC} depending on the outlet throttling edge's opening K_B for a movement of the test rig's boom cylinder at 50 % of maximum speed against different load forces at a supply pressure level of 100 bar as a specific example, resulting from equation (9).
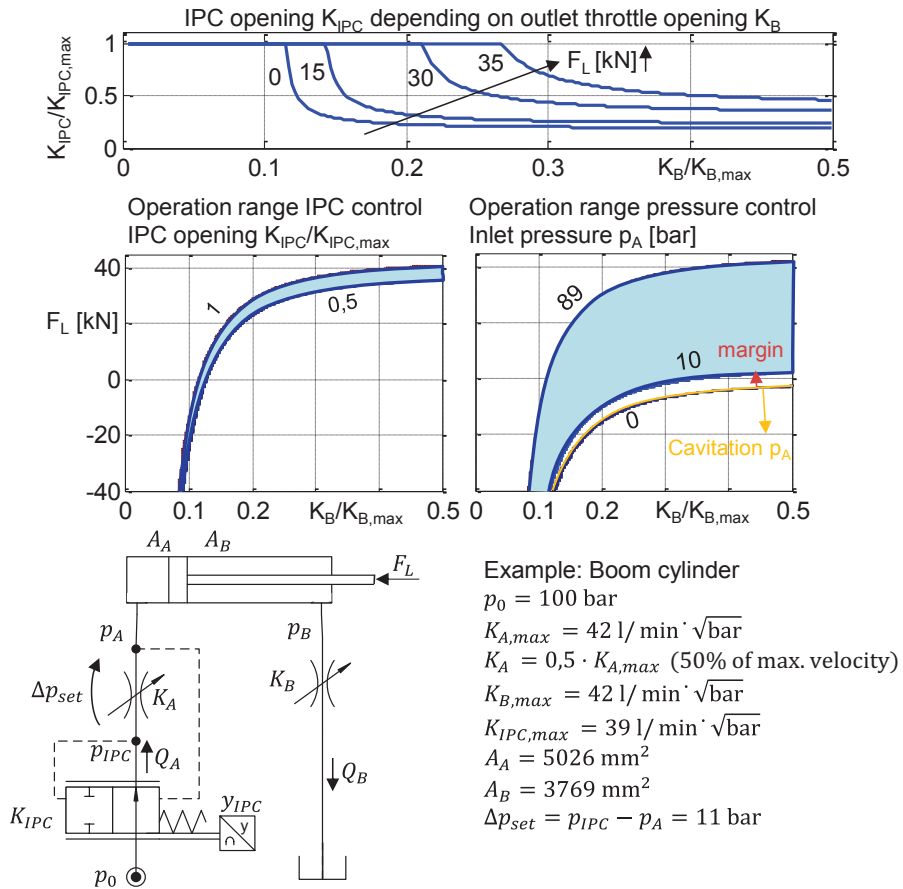
Figure 3: Operation ranges for IPC and inlet pressure control

A smaller outlet edge opening K_B leads to a wider IPC opening K_{IPC}, since closing the outlet throttle raises the pressure levels p_A and p_B in the cylinder chambers. This decreases the pressure difference between inlet pressure p_A and supply pressure p_0, causing the IPC to open its throttle further. The relation between K_B and K_{IPC} is extremely nonlinear with the IPC being almost closed over a wide range of the outlet throttle K_B and opening rapidly in a very small band of K_B (i.e. 0,12-0,13 $K_{B,max}$ for F_L = 0), when the pressure p_{IPC} in front of the inlet throttle gets close to supply pressure level p_0. Furthermore the threshold at which the IPC fully opens heavily depends on the load force F_L. The nonlinear characteristic of $K_{IPC}(F_L)$ varies the controlled system's amplification $\partial K_{IPC}/\partial K_B$ over a large range depending on K_B and F_L. Without measurement of the load force F_L this amplification is unknown. Therefore its variation cannot be compensated by adapting the controller gain.

The left diagram in Figure 3 shows the IPCs opening K_{IPC} depending on outlet throttling edge opening K_B and load force F_L for the described example scenario. The isolines mark the operation points (K_B, F_L) at which the IPC is fully open and half open. Both isolines bound the narrow operation range within which the outlet throttle opening K_B must be set to open the IPC between half and full way ("IPC control"). This requires throttle valves with high resolution.

This demand and the varying system amplification are obstacles to a practical implementation of the proposed control strategy.

Refinement of the control strategy. To overcome the revealed problems the range within which IPC and outlet throttle can be set without compromising the control strategy's benefits shall be enlarged to reduce the requirements on the valves' resolution. Furthermore the controlled system will be linearized to obtain a constant amplification.

Instead of a specific IPC opening K_{IPC} the inlet chamber pressure p_A is now used as the reference variable for the control circuit which actuates the outlet throttling edge K_B. The IPC opening is now used to determine the pressure drop over the IPC in order to calculate the inlet pressure p_A without using an individual pressure sensor at the consumer. Knowing p_A and allowing values down to a certain margin against cavitation (i.e. 10 bar, see Figure 3 right diagram) smaller IPC openings are acceptable without compromising the control strategy's benefits mentioned before. The operation range ("pressure control") of the outlet edge is enlarged considerably compared to IPC control thus reducing the requirements on controller performance and proportional valves.

The controlled system is linearized by using the chamber pressures as the input and output variables instead of the valve spool positions **(Figure 4)**. The control circuit (highlighted) is constructed around the control variable p_A with its reference value $p_{A,d}$ and the outlet chamber pressure p_B as the manipulated variable (back pressure manipulation, /8/). In steady state these both values have a linear correlation according to equation (7) with the constant piston area ratio as the controlled system's amplification and the load force F_L as the disturbance variable. An ordinary linear PI controller is sufficient to fulfil this control task.

*: normal operation mode
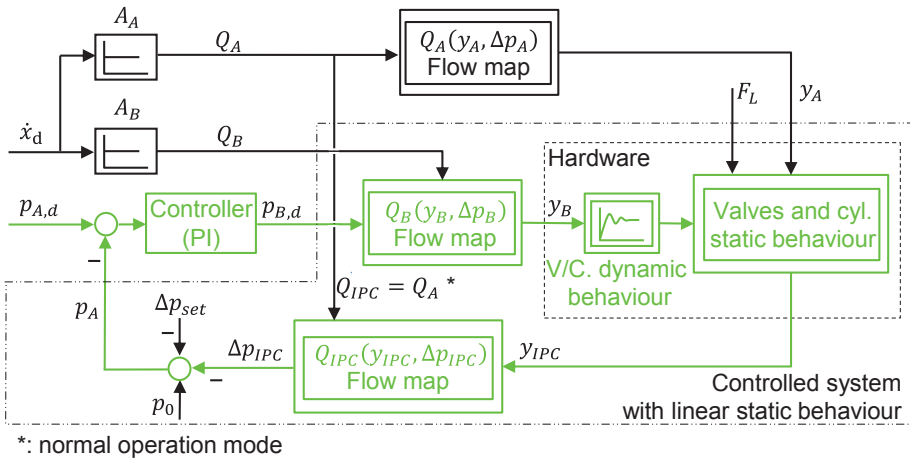
Figure 4: Linearized drive system and pressure control loop (highlighted)

3.3. Development and implementation of the control algorithm

A lumped parameter simulation model was used to develop and test the control algorithm using the software-in-the-loop method (SIL). The model provides an interface including all signals of actuators and sensors at the test rig. The control algorithm was primarily run on a virtual ECU which controlled the simulation model via the Open Platform Communication System (OPC). Afterwards the algorithm has been verified on the real test rig.

There are five subtasks the control algorithm has to fulfil in order to move the actuators boom and stick cylinder energy efficiently at the desired velocities. These are determination of the current chamber pressures, selection of the optimal operation mode, calculation of the common desired inlet chamber pressure, setting the valves and actuating the pump.

The inlet chamber pressure p_A is calculated with the supply pressure p_0 and the valve spool positions of IPC and inlet throttle:

$$p_A = p_0 - \Delta p_{set} - \Delta p_{IPC}(y_{IPC}, Q_A(y_A, \Delta p_{set})) \tag{10}$$

The current inlet volume flow Q_A is obtained from the inlet throttle's flow map $Q_A(y_A, \Delta p_A)$ using the inlet valve opening y_A and the assumed pressure drop Δp_{set} over the inlet throttle edge which is determined by the IPC. With the IPC's spool position y_{IPC} and the inlet flow Q_A the pressure drop Δp_{IPC} is calculated with the IPC's flow map $Q_{IPC}(y_{IPC}, \Delta p_{IPC})$, while the supply pressure p_0 is measured with one single sensor in

the common supply line. This method neglects further pressure losses that occur in the pipe between pump and valve block, the hoses between block and cylinders and throttling losses in the channels of the blocks and the switching valves. These simplifications lead to an overestimation of p_A, which will have the largest relative impact at high velocities and a widely opened IPC.

The outlet pressure p_B is estimated to match with its desired value $p_{B,d}$.

If the condition for high pressure regeneration

$$p_{out} > p_{in} + 2\Delta p_{set} \; and \; piston \; extension \tag{11}$$

is fulfilled the operation mode is set to high pressure regeneration, otherwise normal operation. In the former mode the outlet flow from the rod side chamber (R) is redirected to the piston side chamber (P) between IPC and meter in edge during piston extension thus reducing the required pump flow.

With both chamber pressures the load force F_L and the least required inlet pressure p_r to move the load are estimated for each actuator. The highest required pressure is the common desired inlet pressure $p_{in,d}$ for all actuators.

The PI pressure controller sets a desired outlet pressure $p_{B,d}$ in accordance to the control deviation between current inlet pressure p_A and desired pressure $p_{in,d}$. While the inlet throttle position y_A is set with the required volume flow corresponding to the desired velocity v_d and the constant pressure drop Δp_{set} controlled by the IPC the outlet throttle position y_B depends on v_d and the desired outlet chamber pressure $p_{B,d}$.

The pump is controlled in an open loop manner utilizing the flow matching algorithm as described in /9/ and suggested by /1/ to deliver the overall required volume flow. The proportional valves are actuated by a feed forward signal combined with a PI-based closed loop spool stroke control.

4. Measurement and simulation results

The described valve system and control algorithm are evaluated in terms of the proposed pressure control, dynamic handling performance and energy consumption at the mini excavator implement test rig and in simulation. As an example movement the levelling **(Figure 5)** has been chosen because it contains all relevant operation points to demonstrate the system's functionality. These operation points are resistive and overrunning loads F_L, both time-varying (Figure 5, centre and right), different required

pressure levels p_r of both consumers and the ability to regenerate at the lower load consumer.

The levelling movement is driven in manual control with the bucket "in the air", which means that the implement is only loaded with inertial and gravitational forces, but not with digging forces. On a construction site this kind of movement will occur regularly when the operator transports material from the dug hole to a dump truck.
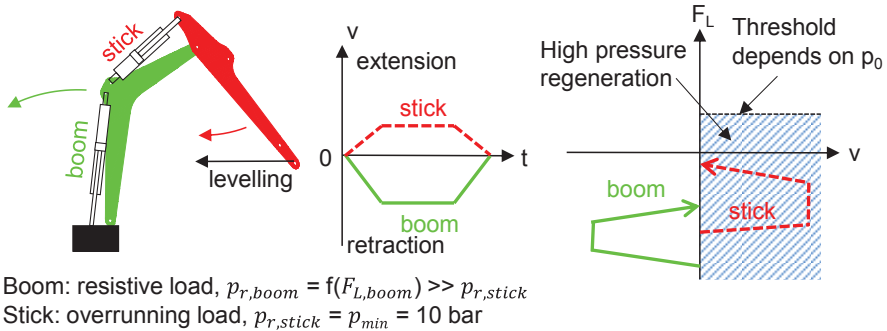

Boom: resistive load, $p_{r,boom}$ = f($F_{L,boom}$) >> $p_{r,stick}$
Stick: overrunning load, $p_{r,stick}$ = p_{min} = 10 bar

Figure 5: Operation points during a levelling movement

Pressure control. The measurement results in terms of the proposed pressure calculation and control are shown in **Figure 6**. The diagrams display the velocity commands for boom and stick cylinder, the measured chamber pressures ("meas.") as well as the reconstructed pressures ("rec.", see chapter 3.3) and the relative IPC spool positions.
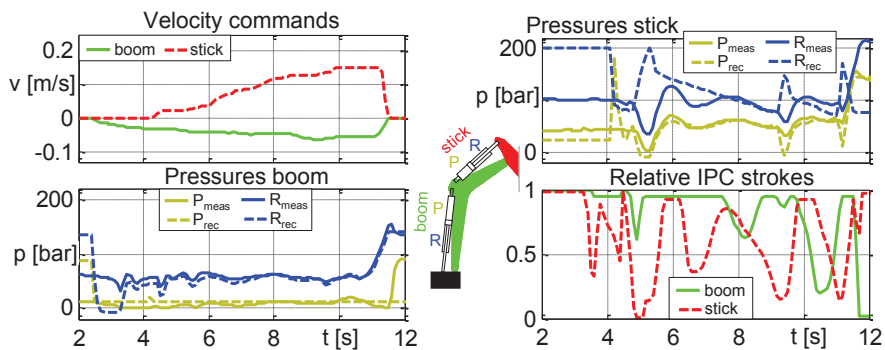

Figure 6: Measured pressures and IPC strokes for a levelling movement

The boom cylinder moves a resistive load in normal operation. The inlet chamber is the rod side (R). The minimal pressure p_{min} the controller shall maintain in each chamber

is set to 10 bar. The outlet flow (P) is throttled slightly to obtain this pressure (bright, brown graphs). This and the load force yield to an inlet pressure of approx. 60 bar (dark, blue graphs), which is the common desired inlet pressure $p_{in,d}$ for both consumers.
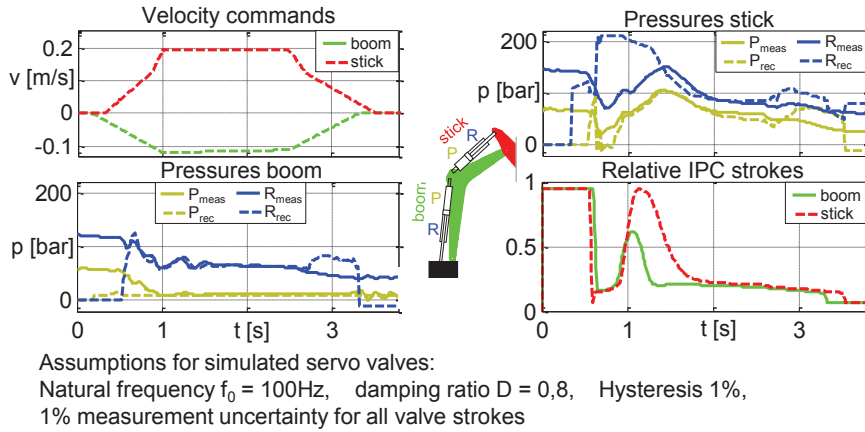
At the same time the stick cylinder lowers an overrunning load in regeneration mode. The load is balanced by the almost closed outlet throttle (chamber R), which shall also increase the inlet chamber pressure (P) to the level of the higher load consumer (boom, 60 bar). This leads to an outlet pressure level at the rod side of approx. 100 bar. The desired inlet pressure has settled at around t = 8 s.

For both cylinders the inlet chamber pressure is slightly underestimated during most of the time, which was not expected according to the simplifications made in equation (10). The reason is found in the undersupply condition, characterized by a very wide IPC opening. In this case the real pressure drop over the inlet throttle is lower than the estimated value Δp_{set} which is subtracted from the measured pump pressure p_0.

The deviations between measured and reconstructed chamber pressures are much higher at the stick cylinder which experiences the overrunning load. This is caused by the high sensitivity of the pressure drop Δp of a throttle edge to variations in the spool position y at small openings. Measurement errors in flow map and spool position are amplified much more than at wider openings, well observable at the stick cylinder's inlet pressure (P) at t = 9 s while the IPC is almost closed and its pressure drop Δp_{IPC} highly overestimated. The pressure deviations in the outlet chamber are due to the high controller activity which was necessary to raise the inlet pressure to the desired 60 bar. Since the cylinder drive has a hydraulic capacity the real outlet pressure p_{out} (R_meas) follows the desired value $p_{out,d}$ (R_rec) set by the pressure controller with a certain delay which becomes evident when $p_{out,d}$ changes.

Dynamic performance and potential for improvement. The shown levelling movement is very slow compared to common operation of an excavator at a construction site. Faster movements at the test rig lead to unstable behavior because the proportional valves act slower than the operator, due to hysteresis effects and a slow stroke controller tuning to prevent unacceptable overshoots. For practically satisfying and safe operation characteristics the valves need to be significantly faster than the operator. A simulation with fast and precise servo valves, shown in **Figure 7**, reveals potential for improvements. The faster valve dynamics allow a more dynamic pressure controller tuning which shortens the settling time for the inlet chamber

pressure to around 50 % (P, stick, phase between 0,5-2 s, compared to the time interval between 5-8 s in reality, Figure 6).

Assumptions for simulated servo valves:
Natural frequency f_0 = 100Hz,   damping ratio D = 0,8,   Hysteresis 1%,
1% measurement uncertainty for all valve strokes

Figure 7: Simulated fast levelling with servo valves

Energy consumption. Energetic aspects have also been investigated, **Figure 8**. The figure depicts the results for the separate metering strategy without regeneration (SPM) and with regeneration (SPMR). For reference purposes a conventional coupled metering strategy (CPM) has also been implemented. In this mode the inlet and outlet flow cross section area always stay in the same relation as the cylinders piston areas, analogue to a mechanical coupling of both throttle edges on one single valve spool. This comparison test has been performed with the simulation model using the desired velocities from the real levelling experiment depicted in Figure 6, top left.

CPM = coupled metering, SPM = separate metering, SPMR = SPM with regeneration
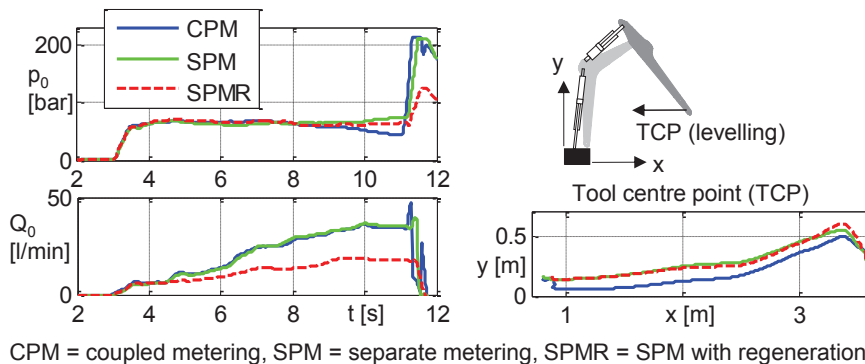
Figure 8: pump pressure and volume flow for different operation strategies (Sim.)

The consumed hydraulic power P_{hyd} is the product of supply pressure p_0 and volume flow Q_0. In comparison to coupled metering (CPM) the separate metering strategy without regeneration (SPM) shows no energy saving potential in the investigated scenario, since pressure level p_0 and volume flow Q_0 are roughly the same. For the stick cylinder as the lower load consumer energy can be saved with regeneration (SPMR). In this mode the volume flow to the stick cylinder is reduced by 66 % according to the piston area ratio. This leads to a considerable overall energy saving of 43 % between SPM and SPMR (see Table 1).

	CPM [%]	SPM [%]	SPMR [%]
Simulation	100	110	57
Measurement	100	90	45

Table 1: hydraulic energy consumption for a levelling movement

This comparison has also been performed on the test rig, where the hydraulic energy consumption can be estimated with the supply pressure and pump angle, neglecting the volumetric losses of the pump. Since the energy consumption heavily depends on the operator even the displayed average values only give a rough indication about the energetic relation between the discussed operation modes. Nevertheless, the tendency found with the simulations can also be seen at the test rig. The deviations between simulation and measurement are probably primarily caused by the fine tuning between pump and consumer at the test rig, which is negatively influenced by the proportional valves' slow dynamics. These have not been modelled completely for the simulation.

5. Summary and outlook

The developed system, using only one common supply pressure sensor and the positions of the IPCs and for valve control purposes the proportional valves' spool positions, is capable of actuating a mini excavator implement with load compensation up to certain low dynamics. The high pressure regeneration enables energy savings up to 48 % in case of a levelling movement without digging forces. More energy saving potential can be exploited by fine tuning minimal chamber pressure level and pump actuation, which requires a faster and more precise throttle valve response and possibly a closed loop pump control.

Currently the desired relation between inlet and outlet flow cross section area is lost due to insufficient valve dynamics during dynamic movements, which has a great impact on the consumers' pressure level. This results in unintended pressure peaks or

cavitation. This problem cannot occur with common mechanically coupled metering where the flow cross section area relation is set by the valve spool geometry. For independent metering the need for a precise tuning between inlet and outlet throttling edge leads to much higher requirements on the valves' controllability compared to mechanically coupled metering.

Simulation results with metering edges featuring the characteristics of high performance servo valves show that the dynamic stability and handling characteristics of the proposed valve structure and control strategy can be improved significantly by using suitable components. Continuing research will address the handling performance by refining the control strategy for the used proportional valves. First experiments show that their dynamic performance greatly improves by applying a suitable dither signal to overcome the hysteresis. Special attention should be paid to mode switching events during ongoing movements. Furthermore the strategy will be adapted to altered sensor setups (i.e. pressure behind IPC instead of IPC spool position) to reduce investment costs and possibly improve handling performance.

6. Acknowledgements

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8. Nomenclature

A	Piston area	m^2
F	Force	N
K	Valve opening factor	$l/(min \cdot \sqrt{bar})$
Q	Volume flow	l/min
p	Pressure	bar
P	Power	W
v	Velocity	m/s
y	Valve stroke	mm
α	Throttle coefficient	-
ρ	Density	kg/m^3