# COMPARISON OF STRATEGIES FOR UNNOTICEABLE MODE SHIFTING IN MOBILE INDEPENDENT METERING SYSTEMS

Jan Lübbert[1]*, Jürgen Weber[1], Peter Bruck[2]

[1]*Institute of Mechatronic Engineering, Technische Universität Dresden, Helmholtzstrasse 7a, 01069 Dresden*
[2] *Hydac Fluidtechnik GmbH, Industriegebiet D-66280 Sulzbach/Saar*

* Corresponding author: Tel.: +49 351 463-39932; E-mail address: jan_frederik.luebbert@tu-dresden.de

## ABSTRACT

Independent Metering (IM) offers great potential to improve the energy efficiency of hydraulic systems. This is especially important in mobile applications due to the limited capacities of electric accumulators, which will probably become the primary energy source in many applications in future. One energy saving mechanism of IM are regenerative operation modes. In many applications, the load direction changes during an ongoing movement. In these cases, regenerative modes are feasible only if there is a way to shift unperceivably between the modes while moving. In this paper, four different mode-shifting techniques are described and compared on an excavator-arm test rig. These are continuous shifting with three active valves (CSA) or with two active valves and a passive path (CSP), and discrete shifting with a pressure compensator (DSMR) and without one (DS). It will be shown, that unnoticeable mode shifting is possible using serial production mobile hydraulic components – either continuously with a low-pressure regenerating valve layout comprising simple check-valves, or fast and discrete using a pressure compensator.

**Keywords:** mobile hydraulics, independent metering, mode shifting, operating behaviour

## 1. INTRODUCTION

Modern hydraulic systems for operator-controlled mobile machines have undergone a long developing history. The first fully-hydraulic excavators appeared on the market in the 1950s [1]. Developing efforts were focused on optimizing the control behaviour for a long time. The outcome of decades of system development and optimization are Open- and Closed-Centre-valve-controlled systems, with their respective specific operating behaviour. With increasing energy costs since the 1970s (first oil crisis) energy efficiency gained importance and Negative-Flow-Control- and Load-Sensing-systems emerged, being wide spread until today.

The current demand for elimination of fossil energy sources and combustion engines forces OEMs to shift to battery-electric drives in many application cases. For example in some cities it is already required by law to operate machine on municipal construction sites with electric drives, to reduce air pollution and the municipal carbon footprint [2]. Under the circumstances of very high battery costs, limited battery capacities and long charging times energy efficiency is not only a question of operating costs any more, but also of acquisition costs, operating time and productivity. This lifts the importance of energy efficiency on a higher level, which most state of the art solutions cannot meet anymore.

In respect to costs and installation space Independent Metering valve technology (IM) is a promising option in the range of medium-sized machines, while in large machines (i.e. mining) multi-pump-systems are common [3].

## 1.1. Energy saving mechanisms of IM

IM offers two mechanisms to save energy: Reduction of unnecessary throttling losses at the outlet throttling edges applies to consumers with altering load direction like an excavator’s stick cylinder or swing drive. Conventional valves need small outlet openings to keep control of overrunning loads. These increase the required pressure for moving resistive loads. With IM the valve’s opening ratio can be adapted to the current load situation thus preventing the mentioned energy loss. The second saving mechanism are regenerative operating modes, which reduce the pump volume flow. The goal is to use the mode, which requires the least pump flow. This optimization can be performed either on axis level [4] or on system level. [5] describes such a solution for a stationary application.

In applications with heavily varying load forces during ongoing movements regenerative modes can only be used, if it is possible to shift between these and normal operation while moving. This task is one of the major challenges of IM to overcome [6], because multiple valves and the pump are involved in a shifting event. Flow paths and pressure drops as well as volume flows change and all involved components must perform so well coordinated that the operator does not take notice of this complex event.

## 1.2. Mode shifting approaches

There are different ways to deal with the mode-shifting problem:

*Prevent shifting* by predefining modes for each consumer and moving direction [7]. For specific applications with foreseeable loads, like crane booms, this is sufficient, but in case of changing loads, i.e. in earthwork machinery, the method will lead to inefficient operating situations when the machine cannot react on the load change by shifting into the proper mode.

The downside of *discrete shifting* is the jerk induced into the movement. This is unavoidable, because valves and pump must react on a step signal, which exceeds their limited dynamics. In [8] and [9] suggestions for minimizing the jerk by selecting an optimal shifting moment are given. Apart from the dynamic limitations, the shifting action must not interrupt the flow path. This demand excludes some valve topologies from mode shifting.

Early extensive research on *continuous shifting* can be found in [10]. Shenouda used the Incova valve system, having a Wheatstone-Bridge layout, applied to a loader backhoe boom. The idea has been adapted to stationary applications in [5] and a different valve layout with a short circuit path in [11].

*Speed-controlled regeneration* is established in certain advanced conventional systems, but can also be found in some IM valves [12], [13]. In this case, a spool valve opens a regeneration path at large spool deflections, mostly from rod to piston side. The classic motivation is to obtain higher speeds with limited pump flow. This strategy is rather simple to implement but limits the usage of regeneration to high speeds, thus loosing much of the energy saving potential.

## 1.3. Mode shifting in serial production IM systems

State of the art IM valve systems available on the market are mostly not capable of shifting in an ongoing movement:

The CMA by Eaton [14] is able to regenerate but not to shift into regeneration while moving, because the shift would interrupt the flow path. The PVX valve by Danfoss [12] incorporates only a high-speed-regenerating edge (speed-controlles regeneration). A completely different approach has been implemented by Husco in their Incova valve [15]. This is a module comprising four 2/2-poppet-valves arranged in a Wheatstone-Bridge. This allows for normal mode, as well as high- and low-pressure-regeneration and continuous shifting between these modes. The module is intended for decentralized mounting and connection with a P and T bus hose line. Probably this novel layout, which is incompatible to traditional machine architectures, and the anticipated high price led to its failure on the market. The latest public sources date back to 2008. To the knowledge of the authors, apart from Incova there is or has been no other IM system on the market, which allows for mode shifting at any consumer velocity.

Mode shifting enhances the application spectrum of regenerative modes and thus energy efficiency of IM systems a lot, making IM much more attractive. Apparently, there is no satisfying solution for the mode-shifting problem yet. This paper will show approaches to shift between modes without disturbing the ongoing movement in a noticeable way, using common mobile hydraulic components.

## 2. OPERATING MODES

The algorithms in this paper use distinct modes and continuously variable modes (CVM). In a distinct mode, the volume flows in the flow paths only depend on the work port flows. Dividing or merging of flows may occur, but with fixed ratios depending on the consumer's geometry or maximal valve openings. A CVM is characterized by a variable dividing or merging ratio at at least one junction.

The following modes will be used in the experiments in this paper:

**Normal extension NM+** (Figure 1 left). In normal extension, the pump delivers the volume flow $Q_S$ to the cylinder's piston chamber P while the rod chamber R is drained to tank ($Q_R$). This is a distinct mode, meaning that there is no option to divide or merge volume flow in a definable ratio.

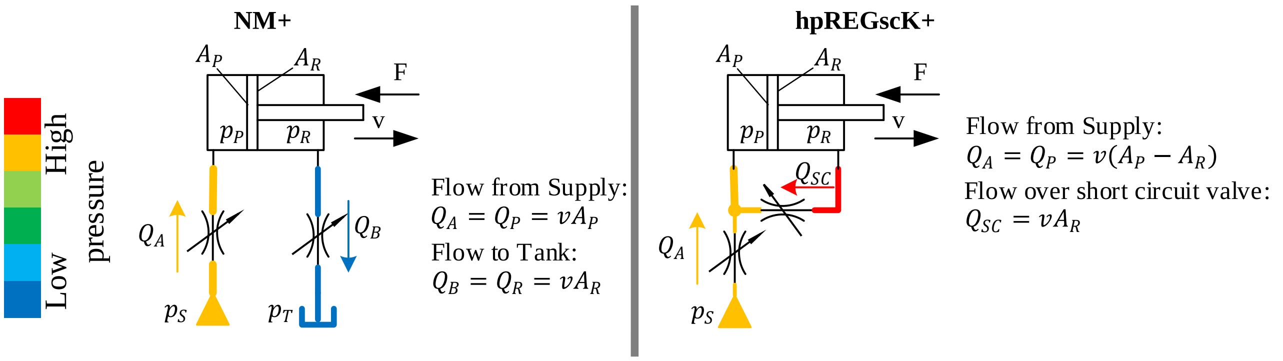


**Figure 1:** Normal extension NM+ (left) and high-pressure-regeneration hpREGscK+ (right). The thickness of the lines indicates the volume flow, the colour the pressure level.

**High-pressure regeneration extension with junction on piston side (hpREGscK+,** Figure 1 right**)**: In this distinct mode, a short-circuit-valve SC feeds the drain flow from the rod side back into the piston chamber. The regeneration flow merges with the pump flow on the piston side. The rod side pressure must be higher than the piston side pressure. The pump delivers only the differential flow. This mode is used for low loads, when the pump pressure is much higher than required for normal extension, and overrunning loads.

**Continuously variable mode (CVM) NMhpREGscK+** (Figure 2 left): This is a mixture between NM+ and hpREGscK+ [11]. The return flow from the rod side divides into a share $Q_B$ drained into the tank and a second share $Q_{SC}$ regenerated to the piston side. A shifting variable $\alpha = 0..1$ defines the dividing ratio. The edge case $\alpha = 0$ is equivalent to distinct normal extension NM+, while $\alpha = 1$ equals distinct high-pressure regeneration hpREGscK+. The mode is intended for shifting between NM+ and hpREGscK+ over a short time span.

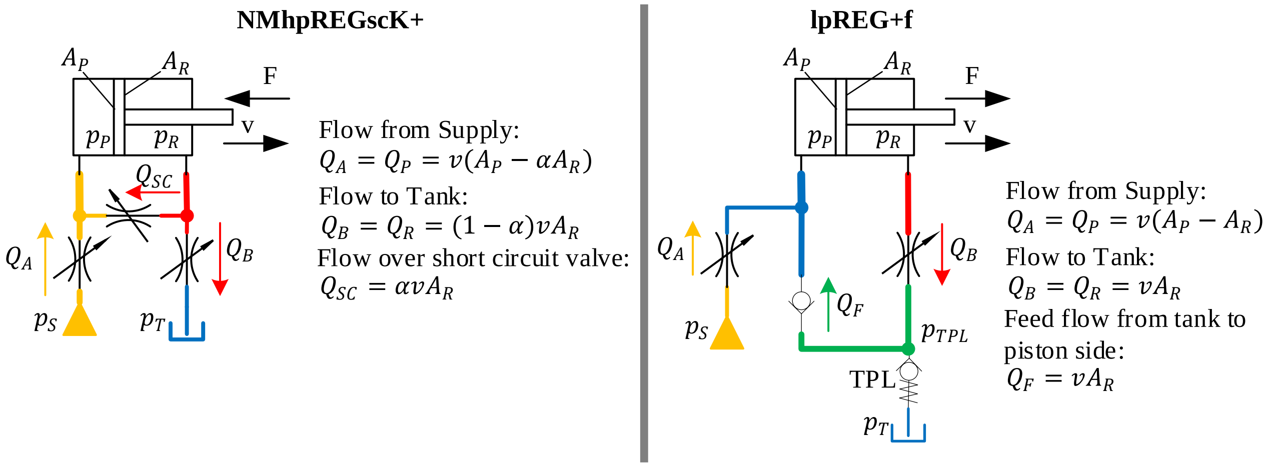


**Figure 2:** Extension in continuously variable mode normal extension to high-pressure short circuit regeneration NMhpREGscK+ (left) and in low-pressure regeneration lpREG+f (right), requiring active load and a pressurized tank line.

**Low-pressure regeneration extension lpREG+f** (Figure 2 right): In this mode, the rod side flow $Q_B$ is drained to tank while the piston side P is being fed partially from the tank line ($Q_F$) and partially from the pump ($Q_A$) [5]. The mode is feasible only for aiding loads. In this case, the load $F$ defines the rod side pressure $p_R$, while the piston side pressure $p_P$ is on a constant low level below the tank line pressure $p_{TPL}$. To enable regeneration from tank line to the cylinder, the tank line must be pressurized, i.e. by a spring-loaded check valve TPL (*t*ank *p*re*l*oad valve). The rod side flow regenerates to the piston side completely, making this mode a distinct mode. The rod side valve B controls the velocity while the piston side valve A delivers exactly the required differential flow.

**CVM NMlpREG+f (Figure 3):** This mode, being a mixture between NM+ and lpREG+f, enables continuous shifting between NM+ and lpREG+f. In contrast to lpREG+f, here the rod side flow $Q_B$ partially drains into tank instead of completely regenerating into the piston chamber. While valve B still controls the velocity, valve A now delivers more than the differential flow $v(A_P - A_R)$ and thus controls the merging ratio $\alpha$ between feed flow $Q_A$ from pump and regenerating flow $Q_F$ from the tank line, making this mode a CVM. $\alpha = 1$ is equivalent to the distinct mode lpREG+f, whereas $\alpha = 0$ equals NM+.

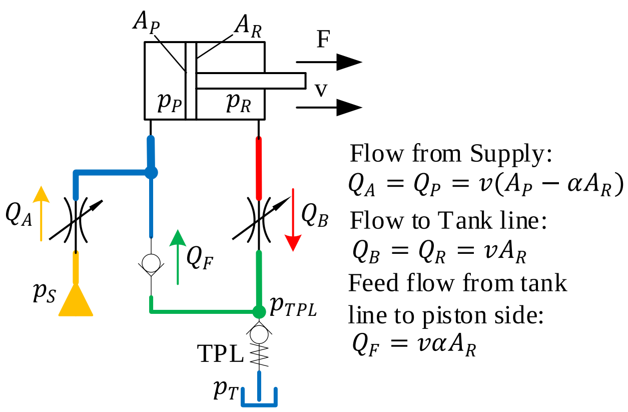


**Figure 3:** Extension in continuously variable mode normal mode to low pressure regeneration NMlpREG+f.

## 3. MODE SHIFTING METHODS

This section explains the different mode shifting methods, which will be evaluated in experiment. Independent from the mode shifting method itself, the shifting threshold defines the circumstances, under which a mode shift shall occur.

### 3.1. Definition of the mode shifting threshold

There are different approaches to define a mode-shifting threshold. Shenouda describes a method based on the power $P = F_L v_{Des}$, which results from the current load force $F_L$ and desired velocity $v_{Des}$ [10]. His aim is to use maximum acceleration at low speed and the additional speed capability of regenerative modes in low load conditions. In this paper the actuator shall never move faster than the limits of normal mode because otherwise a fast moving low loaded cylinder will shift down from regeneration to normal mode when the load force rises, causing a sudden slowdown. A manual operator would probably not accept this kind of behaviour.

Here, solely the load force $F_L$ defines the shifting threshold. Figure 4 shows the force-controlled shifting schedule at the example of the transition between normal extension NM+ and high-pressure regeneration hpREGscK+ via the CVM NMhpREGscK+.

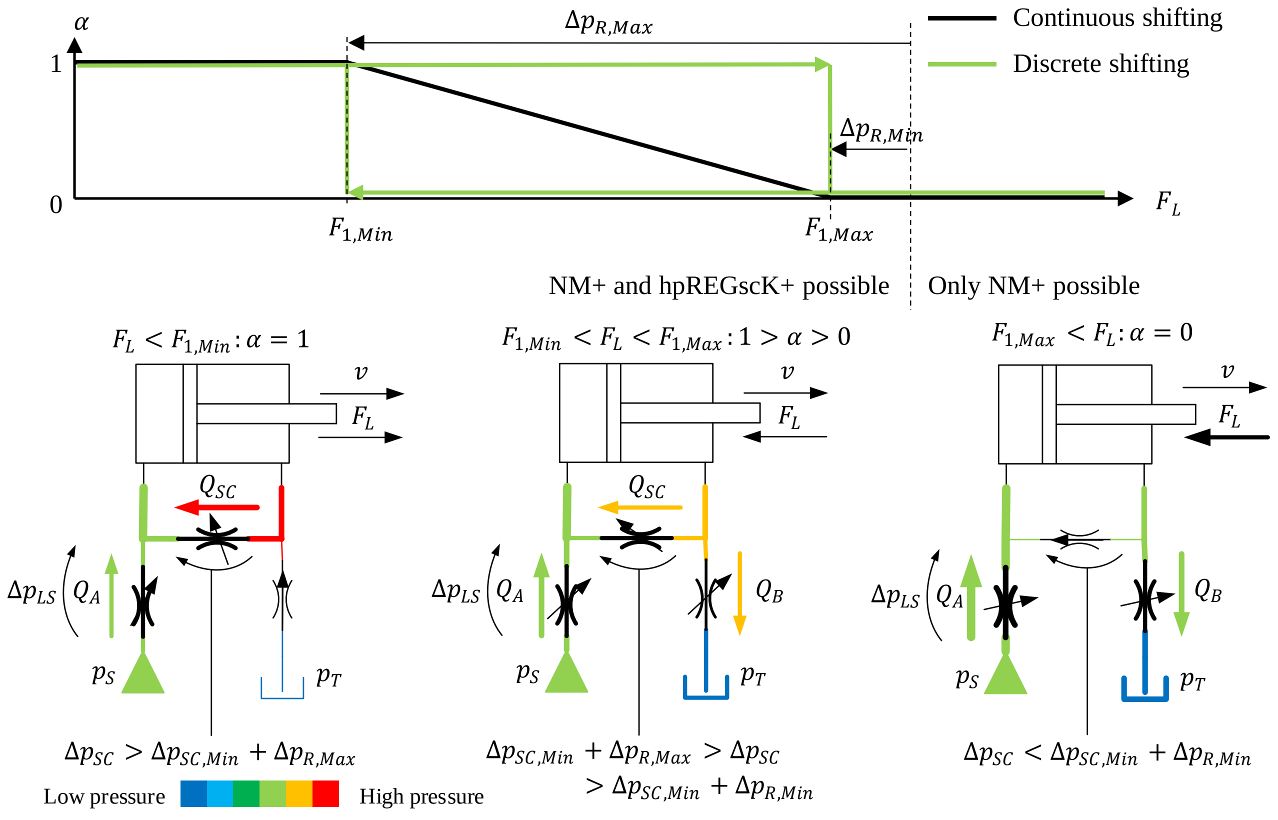


Figure 4: Definition of shifting threshold and CVM-variable $\alpha$ at the example of the transition between high-pressure-short-circuit-regeneration hpREGscK+ and normal extension NM+.

To make an operation mode feasible, certain pressure drops must exist over the corresponding flow paths. In the case of NMhpREGscK+ it is the pressure drop $\Delta p_{SC,Min}$ over the short-circuit-path (bottom part of the figure). If $\Delta p_{SC} > \Delta p_{SC,Min}$, NMhpREGscK+ is feasible, otherwise only NM+. An offset $\Delta p_R$ added to $\Delta p_{SC,Min}$ creates a margin: This way $\Delta p_{SC}$ does not fall below $\Delta p_{SC,Min}$ if $\Delta p_{SC}$ drops quickly due to a sudden load change and the controller triggers the backshift into NM+

when $\Delta p_{SC} < \Delta p_{SC,Min} + \Delta p_R$. Defining a margin interval $\Delta p_{R,Min}..\Delta p_{R,Max}$ creates an interval between two load force values:

$$F_{1,Min} = (p_S - \Delta p_{LS})(A_P - A_R) - \left(\Delta p_{SC,Min} + \Delta p_{R,Max}\right)A_R \tag{1}$$

$$F_{1,Max} = (p_S - \Delta p_{LS})(A_P - A_R) - \left(\Delta p_{SC,Min} + \Delta p_{R,Min}\right)A_R \tag{2}$$

The continuous shifting methods use a linear interpolation of the shifting variable $\alpha$ between the two force values while the discrete methods use a hysteresis loop.

### 3.2. Discrete Shifting (DS)

A discrete shift is an instantaneous transition from one distinct mode to another. There are different ways to perform a discrete shift:

The volume flow is being passed from one proportional valve to another one. This causes a sudden, step-like change of the desired *flow* in the participating valves, while the pressure drops over the valves remain constant. Thus, the valves must change their opening instantly.

If the valve layout consists of a series arrangement of switch/directional and metering valves, the directional valves perform the mode shift, causing the pressure drops over the metering valves to change. In this case, the flows through the metering valves remain constant but here, the metering valves must adapt their openings to the new *pressure drops* to maintain the desired constant flow rates.

In both cases with the mode shift, the required pump volume flow changes instantaneously. The limited dynamics of valves and pump induce disturbances and jerks into the movement. Some valve systems like Eaton’s CMA interrupt the flow path while shifting. The resulting pressure peaks intensify the disturbance.

### 3.3. Discrete Shifting with Pressure Compensator (DSMR)

If, during a discrete shifting event, only the pressure drops over the corresponding metering valves change, but not the desired volume flow (second case in the last section), it is possible to compensate for the sudden pressure drop change with an individual pressure compensator (IPC), eliminating the need for the metering valve to change its opening. An IPC reacts much faster than a mobile proportional valve with its relatively sluggish magnet.

Figure 5 shows a simplified example layout performing a compensated discrete switch from hpREG+ to NM+, induced by a changing load force. In the left part of the figure, the pump delivers flow through metering valve A into the piston chamber of the extending cylinder. Metering valve B drains the rod side, in series with the IPC, which controls the pressure drop over B. A switching valve SV compares the pressure behind B with the pump pressure. An aiding load raises the drain pressure to a high level, causing SV to close and divert the drain flow over the check valve CV into the pump line – high-pressure regeneration hpREG+. The pump delivers the required flow, so the unloading valve UV remains closed.

As the load force changes (central figure), the drain pressure decreases and the control device anticipates a shift into NM+. To prevent undersupply due to the relatively slow dynamics of the pump right after the quick shift, the controller increases the pump flow shortly in advance. The unloading valve UV opens and drains the excessive pump flow to tank, limiting the pump pressure to inlet pressure plus LS margin.

Between the central and right figure the mode shift takes place: A further drop of the rod-side chamber pressure causes SV to open. Now the pressure at the outlet of the IPC drops from pump to tank level. The IPC closes quickly and keeps pressure drop and volume flow over metering valve B constant. Since there is no volume flow regenerated any more, the pump flow matches the inlet flow through A again and the unloading valve UV closes.

Neither had one of the actively controlled components (A, B, pump) to perform a step nor have the volume flows at the cylinder ports been disturbed.

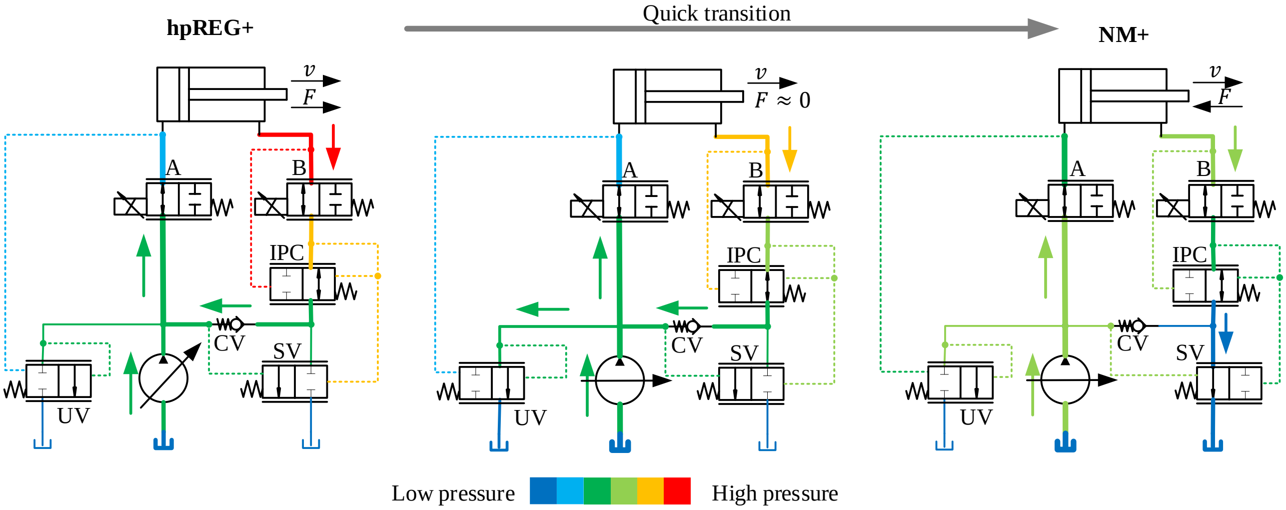


Figure 5: Discrete shifting DSMR with pressure compensator at a simplified example valve layout, in this case triggered by a load change. In this layout, also a changing pump pressure can trigger such a mode shift.

### 3.4. Continuous Shifting with three active Valves (CSA)

This strategy uses three active valves while shifting, similar to the approach in [10]. In at least one path, inlet or outlet, flow is divided or merged; see section 2 and Figure 4 bottom. The dividing/merging ratio is variable. In this condition the IM valve system as a multivariable system has three outputs: Velocity $v$, pressure level $p$ and mode shifting variable $\alpha$. Hence, always three valves are necessary to control these three variables independently.

Figure 4 bottom shows an example of a CSA shift: An extending movement of the cylinder may start in hpREGscK+ with an aiding load – bottom left. When the load force raises above the limit $F_{1,Min}$, the continuous shift begins. The shifting variable $\alpha$ drops below 1, and, according to the equations in Figure 2 right, a portion $Q_B$ of the outlet flow from the rod side is drained into tank. At the same time, the regenerating flow $Q_{SC}$ decreases and the pump flow $Q_A$ increases.

With increasing load force, $\alpha$ drops further and this portion increases until the regenerating flow $Q_{SC}$ becomes zero and the drive operates in normal operation NM+. If the shifting interval between $F_{1,Min}$ and $F_{1,Max}$ is defined large enough, the shifting process is sufficiently long for valves and pump to follow the changing set values with their limited dynamics.

During the shifting phase the outlet flow of the cylinder is the sum of both $Q_B$ and $Q_{SC}$, meaning that the errors of *both* valves contribute to the velocity error of the cylinder. So it is anticipated, that some velocity deviations may occur during shifting. Since the volume flow is controlled by different valves before and after the shift, a constant velocity change due to static valve errors is expected.

### 3.5. Continuous Shifting with two active Valves (CSP)

Drawbacks of CSA are the need to have and control three proportional valves and the summation of flow control errors during the shifting phase. Furthermore, in CSA in general and in some DS(MR) shifting methods, depending on the valve layout, before and after the switch, different valves control the consumer's velocity. This may lead to permanent velocity deviations due to the different flow accuracy errors of the valves. These effects might be noticeable for the operator.

To prevent the aforementioned issues, a novel method shall be developed, in which no summation errors occur and that uses the same valve to control the velocity before and after the shift.

Figure 6 shows a continuous shifting between extension in low-pressure-regeneration (lpREG+f) and normal mode (NM+).

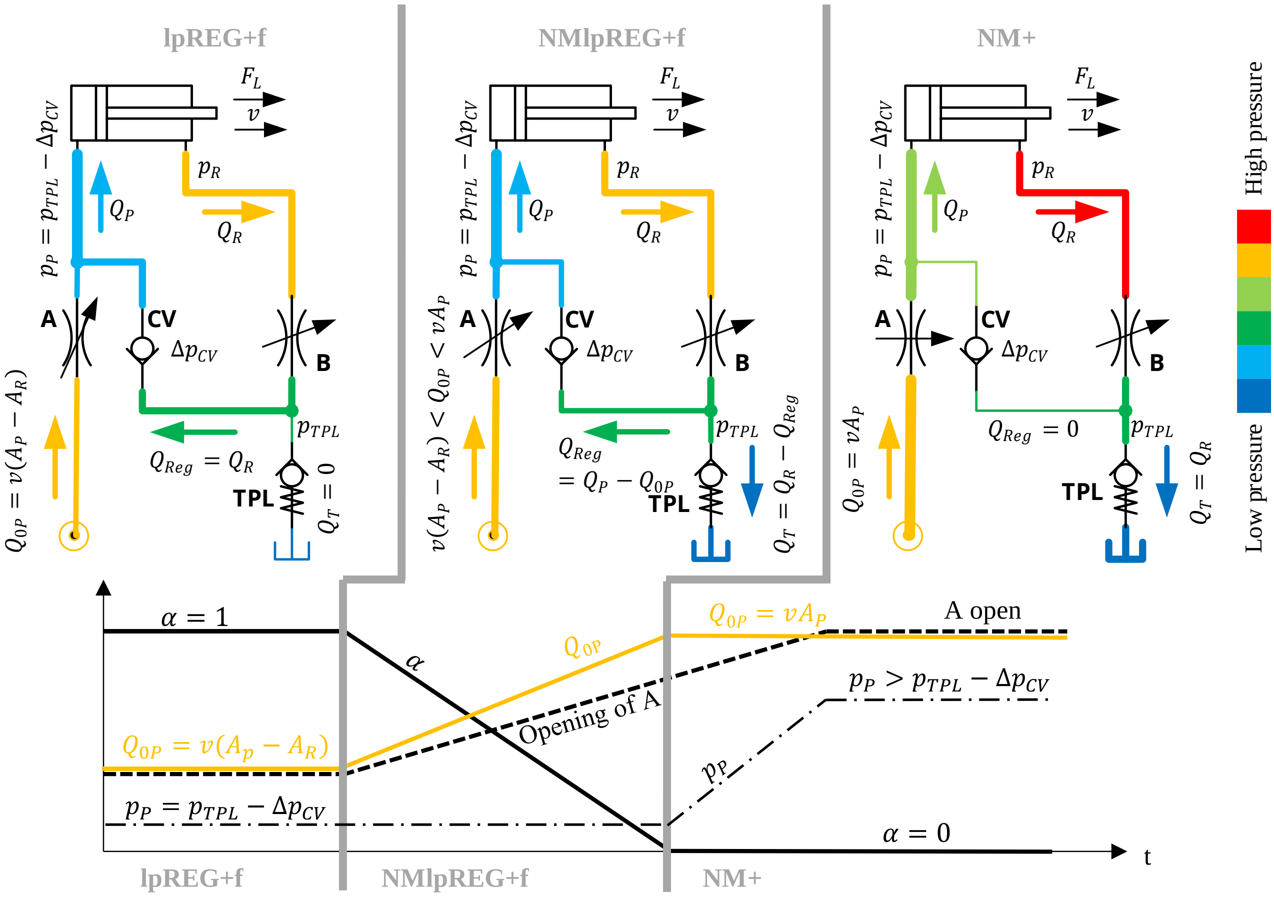


Figure 6: Shifting from low-pressure-regeneration lpREG+f to normal extension NM+ using CSP in a simplified example valve layout.

In one flow path, in this case the inlet on the piston side, there is a junction, at which and actively controlled flow is merged with a flow from a third path reacting passively on the actively controlled flow. Here the actively controlled flow is $Q_{0P}$ through valve A, and the passive flow is the regeneration flow $Q_{Reg}$ through the check valve CV in the centre of the setup. Together with the tank preload valve TPL in the bottom right the regeneration check valve ensures a minimal pressure $p_{TPL} - \Delta p_{CV}$ in the rod chamber.

In lpREG+f (Figure 6 left) valve B controls the velocity while valve A is in control of the operation mode. If A is partially opened so that $Q_{0P}$ exactly matches the differential flow $v(A_P - A_R)$, the consumers operates in distinct lpREG+f. The rod side outlet flow regenerates completely into the piston path while the pump delivers exactly the required flow to fill the piston chamber (see also

Figure 2 right). Once valve A opens further, the transition to NM+ begins by gradually shifting the rod side outlet flow from the regeneration path to the tank path with the further opening of A. In this situation, still B is in control of the velocity, while A controls the mode shift. Once $Q_{Reg}$ has become zero (Figure 6 right), normal mode NM+ is reached and the valves A and B control velocity and pressure together with the common cross-interactions between these two variables.

It should be noted, that in lpREG+f only one active valve is in control and during the shifting phase NMlpREG+f two active valves are used, meaning that in lpREG+f only the velocity is controllable and in NMlpREG+f additionally the shifting variable $\alpha$. In both cases, it is not possible to control the pressure level, which results from the valve layout and load situation instead. Since valve B alone determines the velocity over the whole shifting process, valve precision errors will not lead to velocity disturbances during shifting.

## 4. TEST RIG SETUP

The test rig comprises multiple IM valve systems, two different excavator arms and an electronic control unit.

A TT Control TTC580, programmed in Codesys V3, serves as ECU. The control algorithms are sampled with 20 ms. A PC serves for programming, administration and measurement acquisition via Ethernet. CAN-Joysticks allow for manual control of the three available axes boom, stick and bucket. To obtain reproducible results, it is possible to perform automated movements, which is used for the measurements in this paper.

The "big implement" is a 2:1-model of a medium-sized excavator arm with two conventional supporting boom cylinders and an additional mass of 220 kg mounted at the bucket. This implement will be used for the shifting methods DS, CSA and CSP. An alternative "small implement" from a mini-excavator will be used for the method DSMR. This implement uses a top-mounted boom cylinder, which extends while lowering instead of retracting.

An electro-proportional variable-displacement-pump supplies the test rig using electronic Load-Sensing with flow feed-forward, according to [1].

Discrete shifting DS is tested with an IM valve system "MvS", which consists of one piloted proportional 2/2-spool valve at each work port and a set of four 2/2-poppet-switch-valves to connect each proportional valve either to pressure or to tank, thus enabling high and low pressure regeneration besides normal mode. An upstream pressure compensator decouples the inlet flow from the pump pressure. For details see [16].

A second IM valve system "MvSSC" performs the CSA and DSMR shifting methods. This system comprises an electro-proportional three-way pressure control valve at each work port to connect the port either to supply or to tank; the latter in series with a 2/2-proportional poppet valve and an outlet pressure compensator. A pressure-controlled 2/2-switch valve and a check valve enable high-pressure regeneration, compare to Figure 5. Additionally the systems contains a 2/2-proportional poppet valve in the short circuit path. Details are described in [11].

A third IM valve system "MvSND" serves for the CSP shifting method test (Figure 7). Two pressure-compensated 2/2-proportional spool valves (SP and SR) meter the pump inlet flow, while the outlet flow runs through one 2/2-proportional poppet valve at each work port (PT and RT). A tank preload check valve TPL in the tank drain line and a feed check valve at each work port (FP and FR) enable low-pressure regeneration.

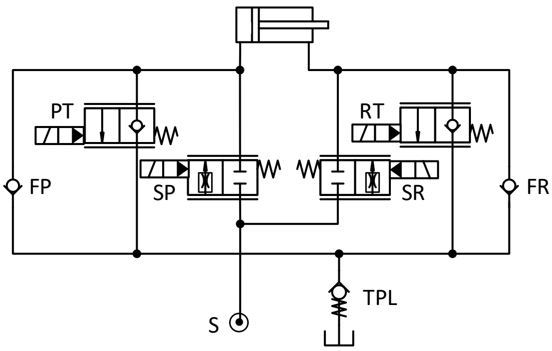


**Figure 7:** IM valve layout “MvSND” for test of switching strategy CSP.

## 5. COMPARISON IN EXPERIMENT

An excavator’s stick cylinder has the highest demand for smooth mode shifting: During operation of the stick in air, without ground contact, a continuous transition from aiding to resistive load occurs, when the stick approaches its lowest position and starts lifting from that point on. The operator expects a very smooth movement. In contrast, the bucket cylinder often operates for digging, where harsh and unpredictable load forces disturb the movement, but are expected and accepted by the operator. Besides, due to the mainly passive load, the bucket drive has almost no chance to use regenerative modes. The boom drive mostly operates against/with a force in constant direction and therefore does not shift between modes while moving. Hence, the shifting methods are tested at the stick cylinder.

The experiments to evaluate the methods DS, CSA and CSP are performed at the “big implement” with 220kg additional mass at its bucket. A constant pressure supply instead of Load Sensing excludes any influence of the pump control algorithm on the mode shifting performance. The boom rests in a relatively level position while the stick cylinder is retracted completely, meaning the implement is stretched horizontally (see faint sketch in Figure 8 central bottom). From this starting position, the stick cylinder extends with a constant set velocity of 100 mm/s. The load force during the resulting movement is aiding at the beginning, slowly increasing to a resistive value. Thus, the valve system can begin the movement in regeneration but needs to shift to normal mode later on.

The central graph in Figure 8 shows the desired stick cylinder velocity and its measured velocities for the different shifting methods. As a reference, the result of a conventional LUDV valve is displayed. The upper diagram shows the measured cylinder chamber pressures (P = piston side, R = rod side), while the bottom frame displays the shifting variable $\alpha$ for the IM systems. At the beginning the rod side pressure is high while the piston side pressure is low, indicating the aiding load. As the load changes, the pressures approach each other, triggering the mode shift.

The LUDV system, supplied by electronic LS, delivers a smooth movement, as can be seen in velocity and pressures. It is constantly too slow due to some inaccuracy in the used valve opening look up table.

The continuous shift of the valve system “MvSSC” from hpREG+ to NM+ via CSA begins at 3.5 s, where $\alpha$ starts to decrease to 0 smoothly. During the shifting process, some minor pressure fluctuations and a small velocity deviation occur, but these are not subjectively perceivable.

At around 3.4 s the shifting process CSP begins. In this case, there is no low-pass-filter for the shifting variable $\alpha$, since it has shown to be not necessary for smooth and oscillation-free performance of the axis. For this reason, $\alpha$ shows some noise in the transition phase. Velocities and pressures are almost constant, making the shift unperceivable.

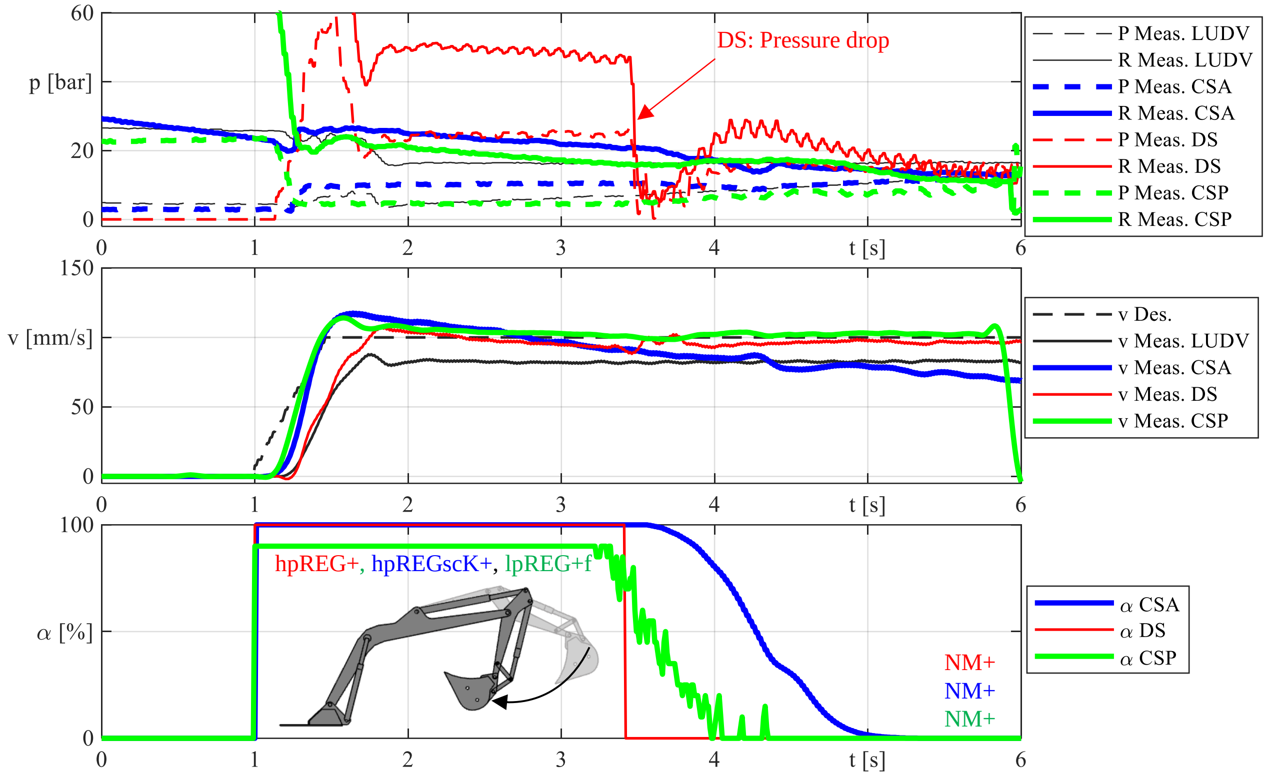


**Figure 8:** Comparison of shifting strategies CSA, CSP and DS.

At 3.5 s in the valve system “MvS” the discrete shift DS from hpREG+ to NM+ is triggered. This causes an abrupt pressure drop in the cylinder due to the sudden pressure drop from supply to tank level at the outlet of the rod-side metering valve. To compensate for this, the valve needs to decrease its opening in a very fast step. The limited valve dynamics cause an unintended peak in the outlet flow and velocity during the shifting phase leading to a well perceivable pressure drop and jerk.

One should notice that the shifting threshold slightly varies between the valve systems due to the different valve layouts and pressure drops across their flow paths.

The DSMR shifting method is tested with the valve system “MvSSC” with its outlet pressure compensators. The required switch valve SV and check valve CV (see Figure 5) are only available in the boom valve system. To test the DSMR shifting method between hpREG+ and NM+ with this valve setup, an aiding extension movement is required. Lowering the boom of the “small implement” with 180 kg additional mass at the bucket provides this required movement. This load results in a rod side pressure of approx. 80 bar, see the upper diagram in Figure 9. The movement is manually controlled with a constant maximal cylinder velocity of 75 mm/s. The pump pressure remains constant at 50 bar. This is achieved by delivering excess pump flow and bleeding it off over an electro-proportional pressure relief valve. With the pump pressure being lower than the rod pressure, the regeneration switch valve SV (see Figure 5) is closed and the rod-side outlet flow regenerates into the pump line, hpREG+.

At 4.8 s a manual signal triggers a step of the pump pressure to 130 bar, exceeding the rod pressure of 80 bar. This causes the valve system to shift into normal mode NM+ instantly by opening SV, causing the pressure behind the outlet pressure compensator to drop from 50 bar to tank level. The pressure compensator quickly regulates the pressure drop over the rod-side outlet valve, so that only a very short and little pressure drop is visible in the measurement, which is neither measurable in the velocity nor subjectively perceivable.

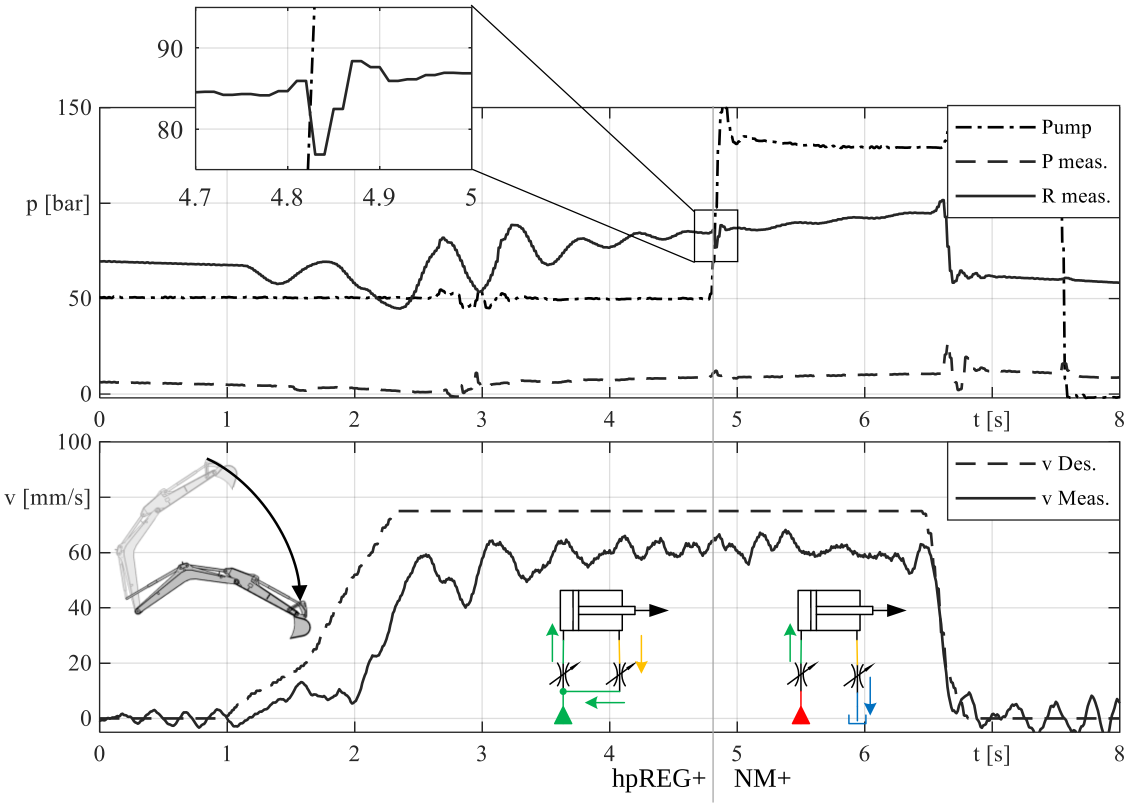


**Figure 9:** Result for shifting strategy DSMR.

## 6. SUMMARY AND OUTLOOK

Both continuous shifting with three active valves (CSA) and continuous shifting with a passive path (CSP) perform smooth, unnoticeable mode shifts. Compared to CSA, CSP is easier to implement, because one valve less needs to be controlled. Furthermore, low-pressure regeneration, as used in this paper, is very easy to implement by re-dimensioning the secondary-relief/anti-cavitation-valves, which must be integrated into most valve layouts for safety reasons anyway. A drawback of both continuous shifting methods is the relatively long shifting time, shortening the timespan in which regeneration is actually used.

Discrete shifting with pressure compensator DSMR overcomes this downside. It offers a very fast and unperceivable shift. The price is the pressure compensator as an additional component and a more complex valve layout. Apart from that, the pressure compensator reduces the drive's damping ratio compared to a valve layout without load compensation [17]. A suitable dynamic strategy for operating an electronically controlled pump together with an unloading valve may eliminate possible dynamic under- or oversupply situations, which result from the sudden flow demand changes the discrete shifts induce.

Discrete shifting without additional measures (DS) has been tested for reference purposes in this study as well, although it is evident, that this strategy is not recommendable because of the unavoidable jerks.

To the authors CSP appears as the preferred solution because of its good operating behaviour and low complexity. To improve energy efficiency, one might add a central switch valve to the control block to bypass the tank preload required for low-pressure regeneration, when none of the active consumers regenerates.

## ACKNOWLEDGEMENT

The presented research activities are part of the project "LiftControl" (Ref. No. KK5023208RF1), which was funded by the German Federal Ministry for Economic Affairs and Climate Action within the ZIM programme.

Supported by:

Federal Ministry
for Economic Affairs
and Climate Action

on the basis of a decision
by the German Bundestag


## NOMENCLATURE

| | | | |
|---|---|---|---|
| $A$ | Area | *F* | Feed |
| $\alpha$ | Shifting variable for continuously variable modes | *IM* | Independent Metering |
| $\Delta p$ | Pressure drop | *IPC* | Individual pressure compensator |
| $F$ | Force | *L* | Load |
| $m$ | Mass | *LUDV* | "Lastunabhängige Durchflussverteilung" = conventional flow sharing valve system |
| $Q$ | Volume flow | *P* | Piston side |
| $v$ | Velocity | *R* | Rod side |
| | | *Reg* | Regeneration |
| *CSA* | Continuous shifting with 3 active valves | *S* | Supply |
| *CSP* | Continuous shifting with a passive valve | *SC* | Short Circuit |
| *CV* | Check Valve | *T* | Tank |
| *CVM* | Continuously variable mode | *UV* | Unloading valve |
| *DS* | Discrete Shifting | *TPL* | Tank pre-load (valve) |
| *DSMR* | Discrete Shifting with mechanic regulator (pressure compensator) | | |

## REFERENCES


1. Djurovic M (2007) Energiesparende Antriebssysteme für die Arbeitshydraulik mobiler Arbeitsmaschinen – „Elektrohydraulisches Flow Matching“. Dissertation, TU Dresden

2. Lee C (2021) "Grüne Baustelle" dank Batterie und Elektro-Bagger? https://www.dw.com/de/bauen-baustelle-nachhaltig-emissionsarm-elektro-bagger-baumaschinen-bausektor-zement-co2-klimaschutz/a-57250263. Accessed 2 Dec 2022

3. Weber J, Beck B, Fischer E, et al (2016) Novel System Architectures by Individual Drives. In: 10th International Fluid Power Conference, IFK Conference proceedings. Dresden, Germany

4. Eriksson B (2010) Mobile Fluid Power Systems Design. PhD thesis, University of Linköping, Sweden

5. Kolks G, Weber J (2016) Modiciency - Efficient industrial hydraulic drives through independent metering using optimal operating modes. In: Proceedings of the 10. International Fluid Power Conference. Dresden

6. Karem Abuowda, Ivan Okhotnikov, Siamak Noroozi, et al (2019) A review of electrohydraulic independent metering technology. ISA Transactions

7. Lee J-C, Jin K-C, Kwon Y-M, et al (2016) Development of the Independent Metering Valve Control System and Analysis of its Performance for an Excavator. In: Proceedings of the BATH/ASME 2016 Symposium on Fluid Power and Motion Control. Bath, UK

8. Ding R, Xu B, Zhang J, Cheng M (2016) Bumpless mode switch of independent metering fluid power system for mobile machinery. Automation in Construction 68:52–64

9. Xu B, Ding R, Zhang J, et al (2015) Pump/valves coordinate control of the independent metering system for mobile machinery. Automation in Construction 57:98–111

10. Shenouda A (2006) Quasi-Static Hydraulic Control Systems and Energy Savings Potential Using Independent Metering Four-Valve Assembly Configuration. Dissertation, Woodruff School of Mechanical Engineering, Georgia Institute of Technology

11. Lübbert J, Stauch C, Bruck P, Weber J (2020) Modular Independent Metering System for mobile Applications providing smooth Mode Transition. In: Proceedings of the 12th International Fluid Power Conference. Dresden

12. (2016) Proportional Valve PVX (product data sheet). https://assets.danfoss.com/documents/DOC152986483080/DOC152986483080.pdf. Accessed 23 Mar 2023

13. (2016) K220LS EcoFormance. https://www.parker.com/literature/Mobile%20Controls%20-%20Europe/Literature%20Files%20MCDE/HY17-8571-B1-UK_K220LS_Ecoformance.pdf. Accessed 7 Dec 2023

14. EATON (2016) CMA200 Advanced Independent-Metering Mobile Valve. Product data sheet by EATON Company. https://www.eaton.com/ecm/groups/public/@pub/@eaton/@hyd/documents/content/pct_1451917.pdf. Accessed 11 Feb 2019

15. Husco (2007) INCOVA® Hydraulic Control System for Excavators (Product data sheet by Husco). http://hydrodream.ru/upload/iblock/e56/e5619a3a7f4f1aed0b1be538c1b0e743.pdf. Accessed 3 May 2019

16. Lübbert J, Sitte A, Beck B, Weber J (2016) Load-Force adaptive Outlet Throttling - An easily commissionable Independent Metering Control Strategy. In: Proceedings of the 2016 Symposium on Fluid Power and Motion Control. Bath (UK)

17. Andersson R (1997) Valves Contribution to System Damping. In: 5th Scandinavian International Conference on Fluid Power. Linköping (S)