Modiciency - Efficient industrial hydraulic drives through independent metering using optimal operating modes

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Abstract

Independent metering poses a possibility to improve energy efficiency of throttle-controlled hydraulic single-rod cylinder drives. This paper deals with energetic potentials gained through variable circuitry that come along with independent metering. A method to assess energetic potentials is described, based on load specific, optimal operating modes. As a means of yielding maximum energy efficiency for a wide range of applications, a smooth mode switching algorithm that minimizes losses and allows good motion tracking is proposed. The mode switching algorithm is validated in simulation and on a test stand.

KEYWORDS: independent metering, mode switching, energy efficiency

1. Introduction

Valve-controlled hydraulic cylinder drives are a wide-spread technology whenever high loads are manipulated in linear motion along with challenging requirements in terms of precision and dynamics. However, by the nature of its physical effect, valve metering is always accompanied by dissipative pressure loss, and thus energy dissipation.

Metering valves eliminate the difference between the system pressure and the load pressure level which is demanded by the load. A common approach to improving efficiency with conventional directional valves is adapting the system pressure to the highest load pressure in the system. This increases system efficiency if the current highest load is significantly lower than the maximum nominal load, yet axes with low load and especially overrunning loads still cause substantial losses. **Figure 1** exemplarily shows the minimal losses with the use of conventional directional valves in a system with three cylinders driven by a single pump.

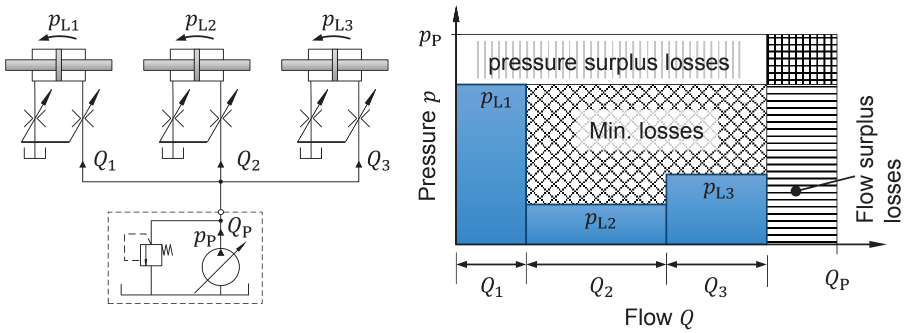

Figure 1: Losses in a throttle-controlled system with conventional directional valves

Primary controlled cylinder drives are a means of delivering load-specific pressure and load-specific flow to the displacement volumes /1/, /2/. Here, speed-variable or displacement-variable pumps are used to control the cylinder speed. The major drawback compared to valve control is a significant increase in component effort: each cylinder needs to be coupled with a high dynamic variable pump unit.

When using throttle control, energy efficiency can be significantly improved by applying independent metering (IM) edges. A conventional directional valve features metering-in and metering-out edges with mechanical coupling through the valve spool. In contrast, independent metering edges is a general term that comprises a class of valve architectures which allow the individual control of meter-in and meter-out edges. This concept allows individual fluid flow paths (modes), such as regeneration, and flexible overlap characteristics for specific control targets. Furthermore, by offering a greater degree of freedom in terms of command variables, multiple target-variables can be controlled independently, such as cylinder position and pressure level.

Extensive research has been carried out on the field of independent metering. Fundamental structural investigations for independent metering architectures with a focus on mobile applications were conducted by Sitte /3/, /4/. His work analyses the solution space for independent metering structures, assesses the solutions and outlines a control strategy for a selected subset of valve structures. Shenouda /5/ gives a comprehensive overview of energy saving potentials of independent metering approaches in distinctive load scenarios and proposes mode switching strategies based on analytical switching procedures in open-loop control, using four 2/2-directional valves. The findings are validated in exemplary cycles of construction machinery. Eriksson /6/ introduces a control strategy and system structure using four piloted poppet-type valves ("Valvistor") that allow load compensation and optimal mode detection with minimal need of sensors. Linjama /7/ systematically investigates feasible switching modes when using five 2/2-directional valves. Mode switching algorithms are proposed and implemented by means of digital-hydraulic system design. Further examples of independent metering approaches in the context of mobile applications and their specific requirements are given in /8/, /9/, /10/, /11/.

In the aforementioned literature, mobile applications in which human operators close the control loops are targeted. Few publications aim at the specific requirements and conditions given in industrial hydraulic applications in terms of tolerated sensor equipment and required accuracy. Closed-loop control approaches that take the opportunity of controlling multiple output variables by means of multiple-input-multiple-output (MIMO) control are summarized in the following. Meyer et al. /12/ implement a linear state-feedback controller in order to control both cylinder position and pressure level via two 4/3-directional valves and one continuous short-circuit valve. A linear-quadratic optimization algorithm is chosen for determination of the feedback matrix. No general method for online generation of the specific working-point feedback matrix is given, however. Mattila and Virvalo /13/ realize a pressure control through input/output feedback linearization for an experimental manipulator. The pressure-responses of a cylinder controlled by two 4/3-directional valves are linearized and decoupled by a nonlinear feedback law. Two independent pressure control paths are thus created and closed via PI-controllers. Bindel /14/ controls the same valve architecture with a flatness-based approach for large-scale manipulators. Both cylinder position and pressure level are controlled and stabilized along predefined target trajectories.

Despite the diversity of research on the field of independent metering, no methodology exists that offers both a universal assessment of the expected energy saving potential compared to conventional valves, and a control strategy to be able to attain the savings. This paper introduces the *Modiciency* method that comprises the assessment of minimal energy demand, an outer-layer control and mode switching strategy and a suitable inner-layer closed-loop controller, capable of smooth and efficient mode switching.

2. Assessing energy saving potentials of independent metering edges

By means of variable circuitry, independent metering structures in general allow for individual fluid flow paths, such as regeneration on low or high pressure side. That means, the conventional power flow path from high pressure supply to the expanding displacement volume, and from the contracting displacement volume to the low pressure return line can be suspended. Load-dependent circuitry modes can be applied in order to decrease the energy demand for a required cylinder movement at a given load. These potentials are summarized under the term volumetric potentials. General circuitry modes using independent metering structures with two pressure levels (supply and reservoir) are depicted in **Figure 2**.

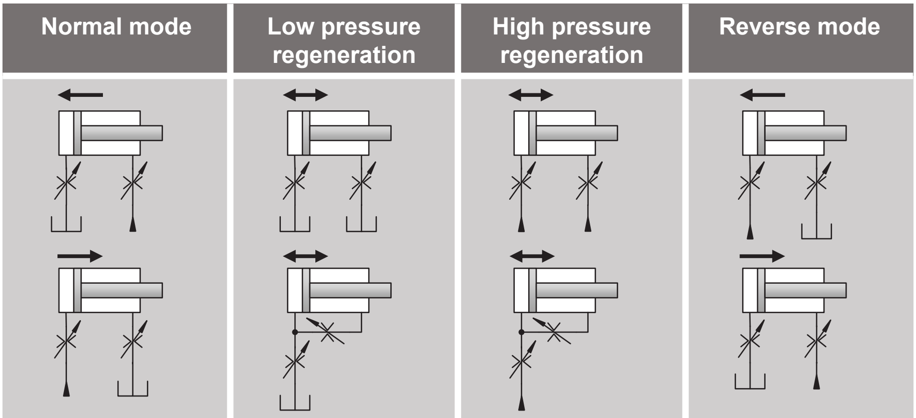

Figure 2: General circuitry modes with a valve controlled differential cylinder

In *normal mode* the expanding displacement volume receives oil from the supply pressure level and oil from the contracting volume is conveyed to the reservoir line, analogous to conventional 4/3-directional valves. As a standard mode with the widest operation range, normal mode does not provide volumetric potentials for energy saving.

In *low pressure regeneration mode*, filling and discharge of both displacement volumes takes place via the reservoir. It is only possible at the existence of an overrunning load, since the system is decoupled from the high pressure supply. Therefore, no external hydraulic power is needed to drive the actuator – it is driven by the load. Low pressure regeneration can be aided by a direct regeneration path, see the lower icon in Figure 2. With a direct regeneration path, cylinder filling of the rod side chamber during cylinder retraction takes place at a higher pressure than reservoir level, which helps avoid cavitation. In any other case, low pressure regeneration involves very low pressure in the expanding displacement volume, since oil is drawn from the reservoir.

As opposed to low pressure regeneration, *high pressure regeneration mode* accomplishes filling and discharging of displacement volumes via the supply pressure line. Again, a direct regeneration path can be used to expand the operation range. High pressure regeneration acts as a hydro-transformer of load pressure and volume flow with respect to the supply line. Flow is decreased and pressure is increased by the interaction of the differential piston areas. During cylinder extension, high pressure regeneration can be used to reduce the volumetric flow from the supply line if the load is low with respect to supply pressure. Cylinder retraction in this mode requires an over-running load. The increase of load pressure through the hydro-transformer principle helps exceed the supply pressure level. If the supply level is exceeded, high-pressure oil is conveyed to the supply line and thus energy is recuperated. This requires either a flow demand by another actuator in the system or the capability of recuperation by the supply system.

Reverse mode is, in terms of flow circuitry, equal to normal mode, yet the cylinder movement occurs in the reverse direction. This mode is only possible at a high over-running load whose load pressure exceeds the supply load pressure. Oil is drawn from the reservoir and ejected into the supply line by the load. The magnitude of load required for reverse mode does not allow cylinder movement against the load direction. In this mode, energy is fed back to the supply system, involving the same requirements as described for retraction in high pressure regeneration mode.

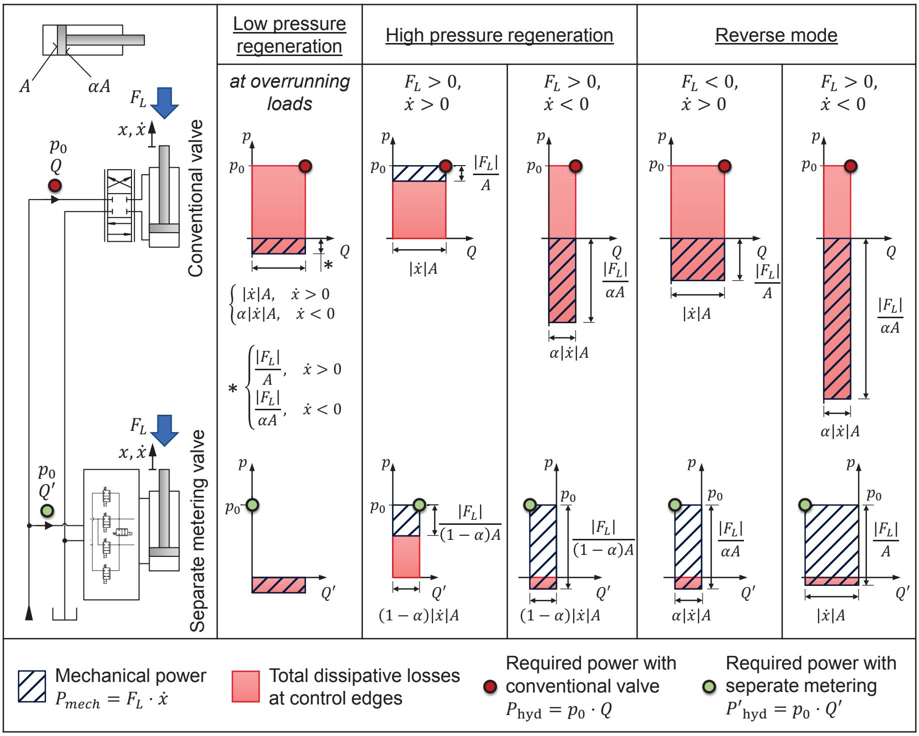

Figure 3: Energy saving mechanisms using circuitry modes

Energy saving potentials using the aforementioned circuitry modes are summarized in **Figure 3**, assuming constant supply pressure. The hydraulic working points in the p-Qplane at the system interfaces between supply line and the valve's supply port are compared for a conventional valve and independent metering valves in different modes and their characteristic load cases. Mechanical power transferred by the load and hydraulic losses are represented by areas. Overlapping regions indicate that hydraulic losses are covered by an overrunning load. Operating points in the first and third quadrant represent hydraulic power to be provided to the actuator-valve-system. Operating points in the second and fourth quadrant mean that energy can be regenerated.

Additional energetic potentials, that are not subject of this paper, exist for load sensing structures in mobile hydraulics, where pressure drop over the load-leading metering-in valves are held constant by the supply pump control. Further potentials, when facing high overrunning loads, lie in the ability to reduce the supply pressure level that is critical for cavitation, by opening the meter-in valve /4/. Beyond that, the ability to decrease the supply pressure level while maintaining the system accuracy is discussed in /13/.

In order to analyse the energetic potentials that emerge through independent metering by flexible circuitry, a general system structure featuring a maximum degree of freedom is chosen, see **Figure 4**, right. It features two variable orifices per cylinder displacement volume: one high-pressure valve and one low-pressure valve each. Additionally, a direct regeneration path is considered by means of a short-circuit valve between both cylinder work ports. In total, five 2/2-directional valves control the cylinder. Feasible regeneration modes can be differentiated by the availability of a short-circuit valve (suffix "sc"). In modes that require suction from the reservoir, cavitation can be avoided by additional high pressure feeding from the supply line (suffix "+f"). A total of nine control modes are defined for the extension of a differential cylinder. During retraction, low pressure regeneration using a short-circuit valve does not involve suction from the reservoir. Therefore, no high pressure feeding is taken into consideration in this case, resulting in one mode less for retraction than for extension.

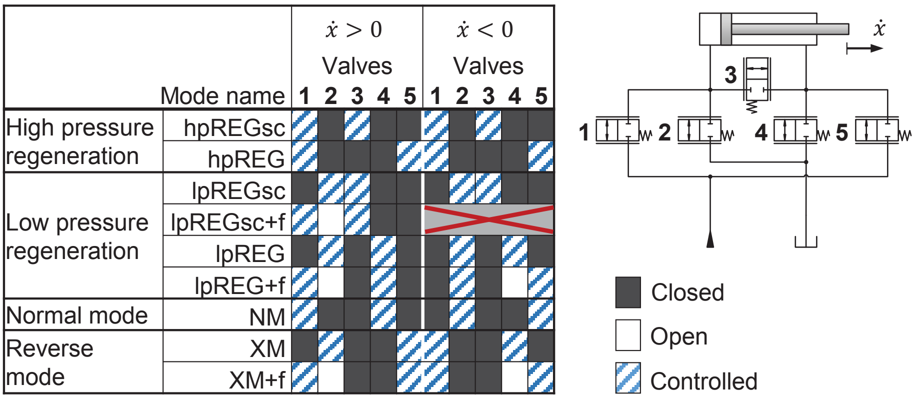

Figure 4: A general valve architecture and physically feasible control modes

The operation range of the control modes can be calculated analytically. Operation limits emerge either through cavitation, defined by a minimal pressure threshold, or by valve stroke limitation. Applied models for force equilibrium and orifice flow characteristics are

$$F_L = A \cdot (p_A - \alpha p_B) \tag{1}$$

$$Q = B' \cdot \frac{y}{y_{\max}} \cdot \operatorname{sgn}(\Delta p) \cdot \sqrt{|\Delta p|}. \tag{2}$$

Friction is included in the load force F_L. Based on these equations and the respective flow path, operation ranges for each control mode can be determined depending on the cylinder geometry, the nominal maximum valve flow rates, the supply pressure p_0, the reservoir pressure p_T and the minimal pressure threshold defining cavitation limits. Furthermore, the required hydraulic power at the supply port of the actuator-valve-system is determined as

$$P_{hyd} = p_0 \cdot Q'. \tag{3}$$

Figure 5 shows efficiency gains with IM for a differential cylinder with area ratio of $\alpha = 0.5$ both with and without a short-circuit valve over conventional valve technology. In every operating point that is feasible with a reference 4/3-directional valve, the most efficient circuitry mode is chosen and the efficiency ratio $\varepsilon = P'_{hyd}/P_{hyd}$ of required hydraulic power with separate metering over the reference valve is calculated. Coloured areas indicate the operation range of a directional valve with an edge ratio of 0.5, adapted to the differential area of the cylinder. The dotted line indicates the operation range using a symmetric reference valve. Operation ranges of control modes are indicated by black lines.

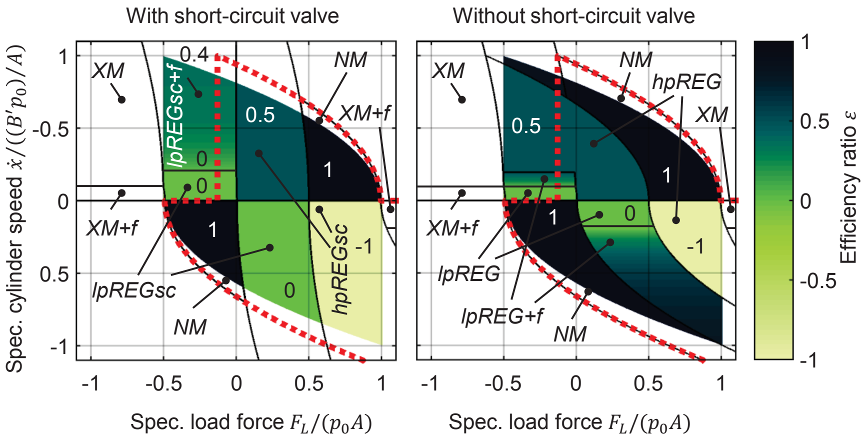

Figure 5: Maps of efficiency ratio $\boldsymbol{\varepsilon = P'_{hyd}/P_{hyd}}$ against force and cylinder speed for an asymmetric reference valve ($\boldsymbol{\alpha = 0.5}$). $\boldsymbol{p_0 = 210\ \mathrm{bar}}$, $\boldsymbol{p_T = 2\ \mathrm{bar}}$. For system layout, refer to Figure 3, left.

Based on the described model, a method has been developed that calculates the minimum energy demand of a system with separate metering edges. A discrete optimization algorithm assigns optimal modes to an arbitrary multi-axes system at a predefined supply pressure trajectory. This method has been applied to a typical duty cycle of a universal testing machine by simulation, see **Figure 6**.

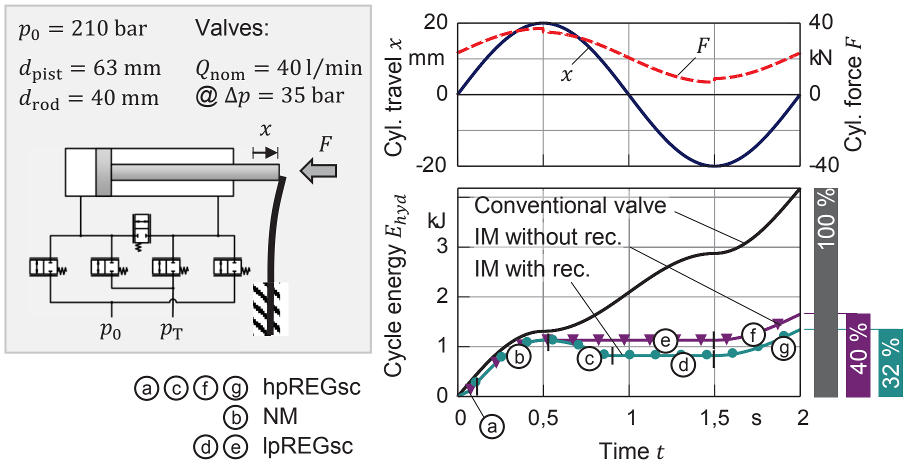

Figure 6: Minimum energy demand with IM for a duty cycle of testing machines

A pre-stressed specimen is subjected to a swelling, sinusoidal load cycle. The load on the cylinder is contracting throughout the entire cycle. At the beginning of the extension movement, the load is low enough to allow high pressure regeneration mode (a). Soon the mode limitation is reached and normal mode has to be engaged (b). On return, the load is high enough to allow contraction under high pressure regeneration (c). This can only be performed if the system allows recuperation or the return flow is needed for another consumer. Otherwise only low pressure regeneration is feasible (e), which will decouple the actuator from the supply system. As soon as the load falls below a certain threshold, high pressure regeneration is no longer possible on retraction and low pressure regeneration (d) will take its place. From the inner return point, extension can be performed by high pressure regeneration (f,g). If the supply system can recuperate or use the high pressure return flow, a total maximum energy saving of 68 % can be found. In case the system does not allow return flow, 60 % can be saved at maximum. These considerable energy savings can be found because the load is low with respect to the supply pressure over a large portion of the depicted duty cycle. Only in about one quarter of the cycle time, the load force exceeds 50 % of the static maximum force. However, precise position control requires a surplus of supply pressure, thus giving the shown example a realistic degree of utilized force capacity of a vast field of applications.

3. Smooth mode switching

Taking Figure 6 into consideration, it is clear that for general minimization of energy demand, switching of different modes during the cylinder movement is required. In /5/, Shenouda has proven that discrete switching between different control modes invokes substantial disturbances and pressure peaks in the cylinder. His continuous mode switching algorithm, however, induces losses that partly cancel the energetic gains. In the work at hand, a smooth mode switching strategy will be proposed that allows permanent control of cylinder speed and low energy demand by means of pressure controlled switching. This enables constant tracking of the desired position in the most efficient operational mode.

During mode switching, cylinder velocity and pressure level are constantly controlled by two valves in a closed loop using a MIMO-controller. The remaining three valves of the setup are either held at constant values or follow predefined trajectories to reach their target state. The benefit of pressure-controlled switching is that at all times

- cylinder speed can be maintained,
- no discontinuities in flow, pressure or valve spool positions occur and
- parasitic flow directly from the supply to the reservoir is prevented.

The last aspect mentioned means that energy losses are minimized. For example, while continuously switching from normal mode (*NM*) to high pressure regeneration (*hpREG*) during extension, low pressure and high pressure valves on the cylinder rod side have to be open at the same time. To ensure that no flow from the supply to the rod-side chamber and from the rod-side chamber to the reservoir occur simultaneously, in this case the pressure in this chamber needs to be higher than the supply pressure.

Mode switching passes closed loop control to the valves which control the target mode. In each elementary mode transition, one valve, subsequently called *switching valve*, switches to fully open or closed state. If it formerly was a controlling valve, it passes control to another valve. Three valves are actively involved in each mode transition. Feasible mode transitions that satisfy the previously given requirements are shown in **Figure 7**. The selection of efficient modes is performed by an outer layer machine controller based on calculated load force and desired cylinder speed. Given the current load and cylinder speed, minimum and maximum feasible cylinder pressure levels at a given supply pressure can be calculated for all modes.

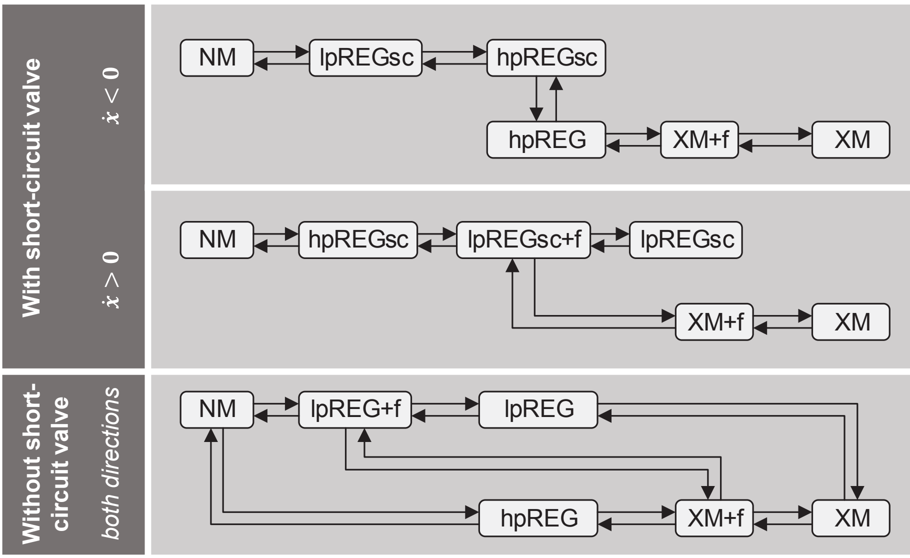

Figure 7: Mode transitions that allow continuous mode switching

The switching algorithm is described in **Figure 8** as an activity diagram. Mode switching basically performs three tasks: Calculating the pressure set-point on the cylinder side where switching is carried out, assigning of control valves and creating a command signal trajectory for the switching valve. The pressure set-point can be chosen in a range depending on the initial mode, the target mode and the current operating point of the cylinder drive. In the investigations carried out here, the pressure was set at 5 bar above the minimum overlap pressure of target and initial mode. During all these steps, the MIMO controller ensures tracking of the current pressure set-point and cylinder speed.

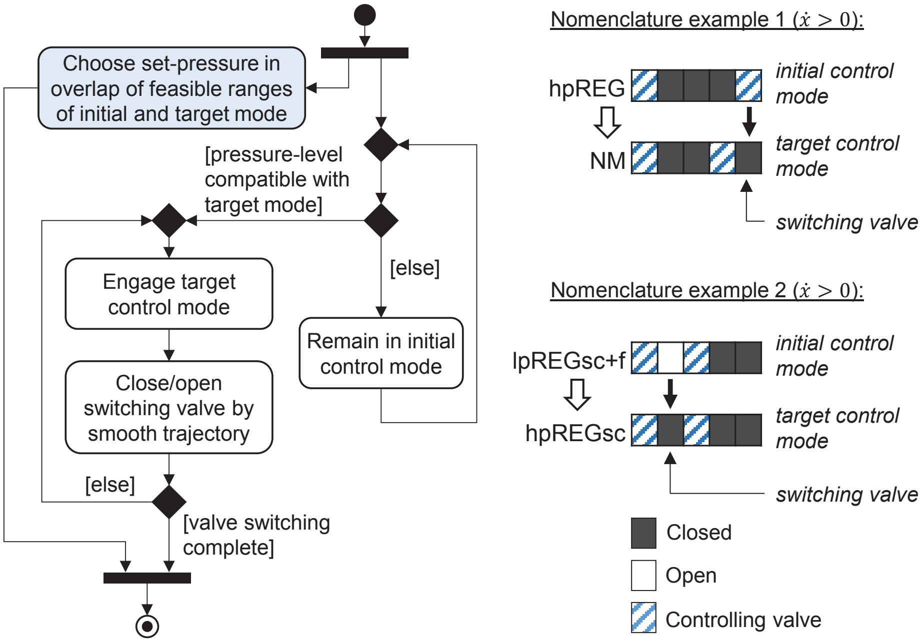

Figure 8: General switching algorithm for mode transitions given in Figure 7

4. Implementation using a flatness-based control approach

The described method of continuous mode switching requires an inner layer control strategy that allows the manipulation of cylinder speed and pressure level independently. Based on a literature and simulation study, a flatness-based control algorithm is chosen as a means of nonlinear MIMO closed-loop control. Feasibility and capability of this control approach has been proven in /14/ and, furthermore, it does not require extensive parametrisation effort to cover the whole operation range, as opposed to linear control approaches. Theory and application are comprehensively described in /15/, /16/.

Control input of the system are the command-signals of the two valves currently in control. As a flat system output, cylinder speed and pressure on one side of the cylinder is selected. Measured cylinder pressure levels and spool positions of all valves are fed to the controller as state variables. The process-model for derivation of control laws is described by second-order valve dynamics, nonlinear orifice flow characteristics, a constant oil compressibility and the force equilibrium using Newton's Second Law.

For validation of the switching strategy and control approach, a simulation model is set up. While the controller is implemented in MATLAB/Simulink®, the process model is created in Modelica-based ITI SimulationX® for convenient modelling of non-constant fluid compressibility $K(p)$ and density $\rho(p)$. The process model for validation features valves with an eigenfrequency of $150\ \mathrm{Hz}$ and a rate limiter that limits the valve spool velocity to $500\ y_{\max}/\mathrm{s}$.

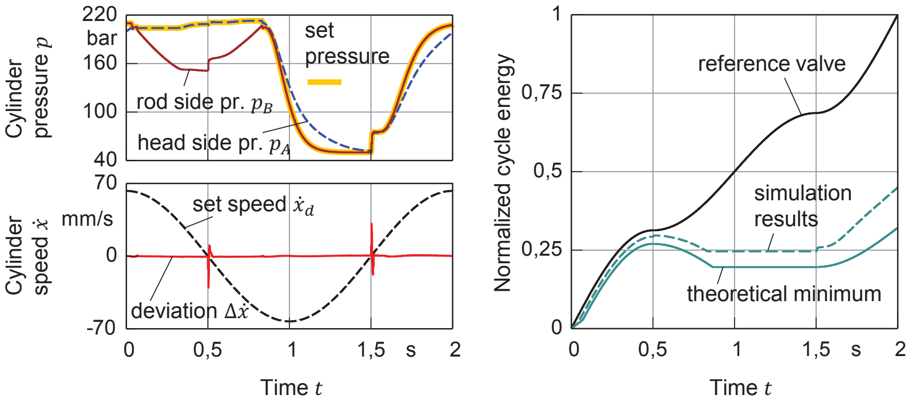

Figure 9: Simulation results of optimal modes using a complex plant model with a flatness based controller. For cycle information and modes, see Figure 6

Dynamic simulation results of the testing machine process described in Figure 6 are depicted in **Figure 9**, assuming a supply structure that allows recuperation flow. The pressures in both displacement volumes and the set pressure which feeds as an input to close-loop control are shown. It becomes clear that during mode switching, depending on the mode transition, pressure control is passed from one cylinder side to the other. Also shown are the set cylinder speed and the control error. Evidence is given that mode switching during cylinder movement can be performed without significant disturbances on the controlled cylinder speed. However, during standstill in reverse points, significant deviations in speed occur. This is due to a force step induced by Coulomb's friction and an instable control strategy in case of reversion of motion. In order to overcome this issue, improved position control at low cylinder speed will be implemented in future strategies. Otherwise, very good motion tracking can be observed. Finally, the resulting energy consumption is compared to the theoretical assessment which considers infinitely fast valves and infinite fluid stiffness. In total, a plus of energy consumption of 13 % in relation to the reference directional valve can be seen. This is a result of three main causes: First, the switching delay into more efficient modes ($t = 0.8\ \mathrm{s}$) or necessary premature switching into less efficient modes ($t = 0.1\ \mathrm{s}$) due to limited valve speed, second, the control error in standstill ($t = 0.5\ \mathrm{s}$ and $t =$

$1.5\ \mathrm{s}$), and third, compression effects of oil in case of pressure increase and low cylinder speed ($1.6\ \mathrm{s} < t < 1.8\ \mathrm{s}$).

In addition to simulation, validation is carried out in an experimental environment. Five direct-actuated servo valves are used as proportional edges for motion control of a differential cylinder with a piston diameter $d_{pist} = 63\ \mathrm{mm}$ and rod diameter $d_{rod} = 40\ \mathrm{mm}$ at a supply pressure of $p_0 = 150\ \mathrm{bar}$. First validation is carried out without external load, thus mode switching is restricted to modes which do not require overrunning loads. In **Figure 10**, measurement data of mode switching between normal mode (NM) and high pressure regeneration (hpREG) during extension at constant cylinder speed and at no additional load are shown. At the beginning of the measurement, the cylinder is controlled by valves 1 and 4 (compare Figure 4) in closed-loop. Pressure on the rod side is controlled at a set-point of 180 bar. At the beginning of the switching process, the pressure is reduced to supply pressure plus an offset of 5 bar. As soon as this level is reached, control is passed from valve 4 to 5 and valve 4 is closed by a predefined trajectory. After the switching process, the pressure level is increased to the initial set value.

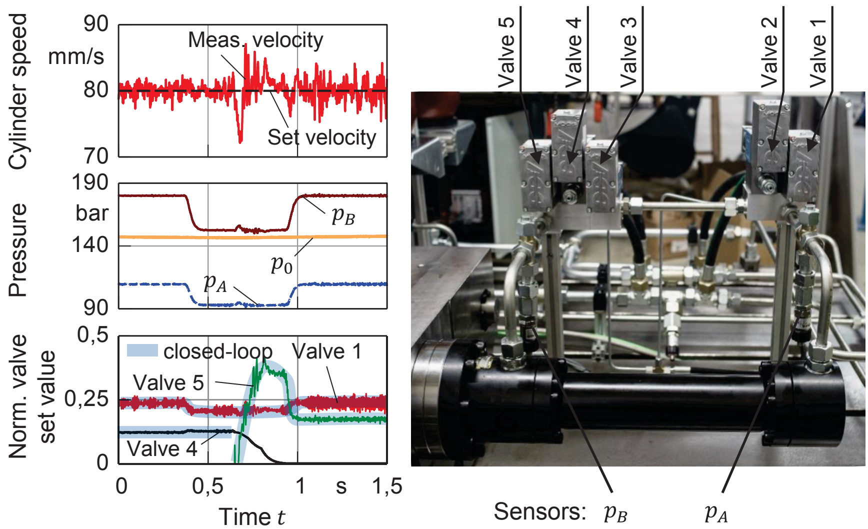

Figure 10: Experimental validation of mode switching NM → hpREG

The potential of the proposed control approach is proven by the fact, that despite a premature state of the control algorithm, good motion tracking can be obtained during mode switching. Yet at the beginning of the switching process, a short deviation from the set trajectory is recorded. This is due to measurement noise and its impact on state initialization while passing control between different valves. Possible optimization can be carried out by improving controller initialization strategy and manipulation of the Eigenvalues in the flatness-based controller setup, which have been chosen heuristically. Furthermore, timing of the switching process can be optimized.

5. Conclusion

This work provides a methodology for independent metering edges that assesses the technological minimum of energy demand for a given load cycle under the assumption of idealised valves and given supply pressure. A control approach is proposed which yields the minimum energy demand by a margin of 13 % in a demanding duty cycle with good motion tracking in a simulation environment. Measurement data show that the proposed control approach is capable of switching modes smoothly in experiment using high-precision control valves.

6. Acknowledgement

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by the German Bundestag

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8. Nomenclature

α	differential cylinder area ratio	$\mathrm{mm/s}$
ρ	oil density	$\mathrm{kg/m^3}$
A	cylinder piston area	$\mathrm{mm^2}$
B'	valve flow coefficient	$\mathrm{l/\left(min \cdot \sqrt{bar}\right)}$
d	diameter	mm
F_L	load force	N
P_{hyd}	hydraulic power	kW
p_0	supply pressure	bar
p_T	reservoir pressure	bar
p_A	cylinder head side pressure	bar
p_B	cylinder rod side pressure	bar
Q	volume flow	$\mathrm{l/min}$
Q'	volume flow with independent metering	$\mathrm{l/min}$
t	time	s
x	cylinder position	mm
y	valve spool position	mm