Characteristics of Energy Efficient Switched Hydraulic Systems


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ABSTRACT

Hydraulic drives are widely found in industrial manufacturing, energy power plants and other important application domains, due to their high power density and robustness. However, one major problem facing modern hydraulic drives is their considerably lower efficiency compared to other drives. Switched hydraulic systems present a possible solution to this challenge. By using a number of different operating modes, these systems can adapt themselves to current load conditions, thereby maximizing efficiency. Despite their great energy-saving potential, care must be taken when using such systems due to the possibility of unstable mode switching and non-smooth physical behavior not desired by operators. This paper reviews recent theoretical and practical developments in the category of switched hydraulics by specifically discussing an innovative design for mobile applications called STEAM.

KEY WORDS

Switched hydraulics, hybrid system, energy efficiency, mobile machinery

NOMENCLATURE

\( A \) Piston area
\( C_h \) Hydraulic capacitance
\( d \) Viscous friction coefficient
\( d_i \) Check valve \( i \) on/off state
\( F_L \) Load force
\( f \) Field vector
\( G \) Guard condition
\( H \) Hybrid automaton/ hybrid state space
\( K_{qp} \) Flow rate pressure coefficient
\( K_{qv} \) Flow rate valve opening coefficient

\( m \) Mass
\( p_{cp} \) Accumulator pre-charge pressure
\( p_r \) Gear transmission ratio
\( p_{hyd} \) Overall pressure on piston
\( p_a \) Supply pressure for piston chamber
\( p_h \) Piston chamber pressure
\( p_b \) Supply pressure for rod chamber
\( p_s \) Rod chamber pressure
\( \Delta p \) Pressure difference
\( Q \) Discrete state set
\( Q_A \) Flow into piston chamber
Fluid power systems are widely used for power conversion, transmission and motion control due to their competitive aspects such as simple linear motion, high power density, simple overload protection and robustness. However, growing concerns over their considerably lower efficiency compared to other drives, have lead to a large body of research concerned with energy-efficient alternatives to conventional hydraulic systems.

Ongoing trends towards energy-efficient hydraulic systems, such as displacement control, independent metering and secondary control, etc., have proven to be promising. What many people do not realize is that many of these advanced designs actually belong to the field of switched systems, since their system dynamics are constantly evolving both continuously and discretely. They are characterized by a discrete evolution of continuous states.

Before discussing these switched hydraulic systems, it is necessary to introduce some terminology and notation. To do so, consider an automatic gearbox with four gears and consequently four discrete operating modes:

$$q(t) \in \{1, 2, 3, 4\}.$$  

These affect the continuous dynamics by changing the transmission ratio \(p_r(q)\) as depicted in Fig. 1.

![Figure 1. Hybrid automaton of an automatic gearbox [1]](image)

Assume that the relation \(p_r(1) > p_r(2) > p_r(3) > p_r(4)\) holds. To maintain the vehicle speed \(v\) within a certain range, a gear change should occur if the angular velocity of the motor reaches the corresponding upper or lower limit \(\omega_{\text{high}}\) and \(\omega_{\text{low}}\), respectively. This hybrid automaton shown in Fig. 1 is a finite-state machine associating each discrete state with a continuous dynamic model. In such a system, each discrete mode has its own continuous dynamics, which can be modeled with a flow condition such as a differential equation, and each mode switch may cause a discrete change in the continuous state. Certain changes in the continuous state will invoke a mode switch, either autonomously or intentionally.

Care must be taken when dealing with such switched systems due to the possibility of unstable mode switching and non-smooth physical behavior, both of which are not desired. This paper aims to discuss the potential of using hybrid/switched systems in hydraulics. It begins by introducing some exemplary systems found in literature and then go on to discuss mathematical modelling approaches as well as the challenges facing the implementation of new systems. These techniques are used to analyse the STEAM system, which is a modern example of a switched mobile hydraulic system.

### EXAMPLES OF SWITCHED HYDRAULIC SYSTEM

In this section, examples of switched hydraulic system are presented. These systems can be divided into two classes: those that switch autonomously due to their inherent design (Type A), and those using a controller to switch in order to improve efficiency depending on the current operating point (Type B).

**Case 1: Displacement controlled differential cylinder (Type A)**

A displacement controlled single-rod hydraulic cylinder is also a switched hydraulic system. Williamson discussed the inherent pumping/motoring mode transitions in such a system in detail [2]. This is an example of a Type A system as it does not switch to improve efficiency but rather due to its inherent design. In a displacement controlled actuation circuit, the pump can operate either in pumping or motoring mode depending on the load condition.

Assuming the flow direction is identical as shown in Fig. 2, where fluid supplied by the pump enters the cylinder piston chamber, in this case, the pump is operating in the right half-plane depicted in Fig. 3, check valve \(d_1\) is closed, and \(d_2\) is open. When the load pressure \(\Delta p\) drops to \((1 - \alpha)p_{\text{cp}}\), \(p_{\text{cp}}\) is the accumulator pre-charge pressure, a transition from the right to the left region will take place, and the velocity will suddenly increase by a factor of \(1/\alpha\). Autonomous switching of the check valves as load pressure evolves can be schematically described in Fig. 4.
This example provides an illustration of an autonomous event-driven switching mechanism as the transition is triggered by the load pressure reaching a certain value. Using a stability analysis, it can be shown that even though the continuous subsystems of both states are linear and stable, instability might be introduced by the discrete dynamics [2]. Such an unintentional mode selection may lead to uncontrolled switching, resulting in sliding behavior around the switching surface and output oscillation.

**Case 2: Hydrostatic transmission for wind turbines (Type B)**

In the field of wind power, hydrostatic transmissions offer distinct advantages compared to mechanical gearboxes. These include lower stiffness and an inherent continuously variable transmission ratio. With the aim of achieving high efficiency at rated power as well as at part load conditions, a switched hydrostatic transmission configuration was proposed by Schmitz [3]. As depicted in Fig. 5, two fixed displacement pumps convert the wind power into hydraulic power in the form of pressurized fluid. Two sets of motors are then used to drive two generators. Each component, except for the smallest pump, can be switched to idle mode, which allows different pump-motor combinations for different operating points. By switching between modes the overall efficiency of wind turbine can be improved.

When the wind speed is low, efficiency will also be relatively low if all the displacement units are switched on and forced to operate at low pressures and low displacements. To adapt to part load conditions, only one small pump and the smallest motor are switched on. As wind speed increases, the larger pump and more motors should be switched on to harvest more wind energy. The red-colored curve shown in Fig. 6 indicates how the overall efficiency is optimized by switching between different modes.

**Case 3: Independent metering**

Compared to conventional spool valve controlled systems, the main feature of the independent metering (IM) concept is that the meter-in and meter-out edges are mechanically decoupled from each other as depicted...
in Fig. 7. Unnecessary meter-out losses can be avoided by opening the valve completely, and a number of operating modes with energy saving potential are enabled, such as energy regeneration and recuperation [4, 5, 6]. One must be aware that operating state only refers to a discrete valve setting, and an operating mode is a state in certain load quadrant [8]. For a motion control system with independent metering configuration, switching logic can also be implemented for the purpose of energy-efficient power adaption under various load conditions.

By specifically discussing an innovative design called STEAM which is currently being developed at IFAS, Germany, independent metering and other switched hydraulic concepts are presented in this paper.

**STEAM**

As discussed in previous publications [6, 7, 8], the start point of this STEAM system design is to improve the overall efficiency of mobile machines by integrating the internal combustion engine (ICE) into the hydraulic design process. A constant pressure system combined with fixed displacement pump enables optimal point operation of the ICE with high efficiency. On the other hand, to decrease the hydraulic losses, the concepts of independent metering edges and intermediate pressure rail are implemented.

In addition to the multiple operating modes introduced by IM, a constant pressure system with an intermediate pressure line enables further switching states with different pressure potentials [9]. As shown in Fig. 8, each cylinder chamber can be connected to one of three pressures (HP, MP, TP). This results in nine possible operating states, with different pressure potentials [9]. See Fig. 9, note that Δp here is the pressure drop across the meter-out orifice, and \( p_{\text{hyd}} \) is the hydraulic pressure acting on the cylinder piston. Based on the current load condition, the cylinder can operate either in standard, regeneration, recuperation or float mode.

STEAM makes use of single edge meter-out control meaning that the cylinder motion is controlled solely using the meter-out orifice [8].

For example, in load Quadrants I and IV, the cylinder piston rod is extending either against a resistive load or with an assistive load, flow exiting the rod chamber should be regulated, as depicted in Fig. 10. If the cylinder rod is retracting, the flow will first be rectified by a 4/2 way switching valve, so that only one proportional unidirectional valve is needed. A further benefit is that it is also possible to integrate a pressure compensator, which normally cannot work in both directions.

To make the following discussions and derivations simple, only the first case of scenario in Fig. 10 will be considered in this paper.
Switching logic

The decision making logic for mode selection is based on the pressure feedback of current load condition. There is a limitation on force capability for each state, which is termed as operating range shown in Fig. 9. A mode switching should occur when the mode capability is no longer sufficient or another mode is considered to be more efficient.

The finite-state machine depicted in Fig. 11 is designed in a way that only valves for one cylinder chamber are involved during a mode switch. To explain this, it is necessary to divide all states into three groups on the basis of common valve settings first. Any operating state denoted as \((p_1, p_2)\) falls in the union of three set states denoted as \(Q_T\), \(Q_M\) and \(Q_H\) respectively, given by

\[
q \in Q_T \cup Q_M \cup Q_H = \{q_{T1}, q_{T2}, q_{T3}\} \cup \{q_{M1}, q_{M2}, q_{M3}\} \cup \{q_{H1}, q_{H2}, q_{H3}\} \quad (1)
\]

or as shown in Tab. 1.

<table>
<thead>
<tr>
<th>Set</th>
<th>State</th>
<th>(p_1)</th>
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<tr>
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<td>(q_{T3})</td>
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<td>(q_{M3})</td>
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<tr>
<td>(q_{H3})</td>
<td>(p_H)</td>
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Table 1. Discrete states and pressure selections

For example, in set state \(Q_M\), there are three element states \(q_{M1}, q_{M2}\) and \(q_{M3}\) which are activated. Each of them corresponds to one pair of valve settings, in which the cylinder piston chamber is always connected to the intermediate pressure rail (MP). For the cylinder rod chamber, switching between available pressure rails (HP, MP, TP) is possible and numbered as 3, 2, and 1 respectively, likewise in the other two set states. When the switch between two set states is necessary due to the load variation, the valves for the piston chamber will be activated while the pressure rail selected for the other chamber remains the same.

Another reason for this arrangement is to avoid interferences between different element states from different set states, which are highly possible or even inevitable depending on the settings of pressure levels, if using a standard controller. If the load pressure changes around some interference region, frequent mode switches might happen within a short period, which in turn may have an adverse effect on the system behavior. Therefore, instead of disabling some operating states permanently, they can be activated in different phases as described above. In the next section, the hybrid automaton is modelled mathematically to describe the sequence of mode switching and conditions required.

SYSTEM MODEL: HYBRID AUTOMATON

A hybrid automaton can be defined as a tuple

\[
H = \{Q, X, f, G, R\} \quad (2)
\]

where \(Q\) is the finite set of discrete states, \(X\) is the finite set of continuous variables, \(f\) is the field vector which governs state trajectory, \(G\) is the guard condition, and \(R\) is the reset function. As depicted in

Figure 11. Hybrid automaton of STEAM.
Fig. 11, the STEAM system can be modeled in such a way to describe both continuous and discrete dynamics.

Discrete dynamics
As shown in Fig. 9, each state has an operating range corresponding to its own force capability, and a state shift would occur once the load pressure reaches the upper or lower limit of current state, this is defined as a guard condition

\[ G(q, q') = \{ x \in S_{q,q'} \} \quad (3) \]

where \( q \) is the current state, and \( q' \) is the successor state. A guard condition is fulfilled if certain switching set

\[ S_{q(k),q(k+1)} = \{(p_A, p_B); p_A - ap_B \geq p_{\text{max}}(q(k))\} \]
\[ S_{q(k+1),q(k)} = \{(p_A, p_B); p_A - ap_B < p_{\text{min}}(q(k + 1))\} \quad (4) \]

is triggered. \((p_A, p_B)\) are the pressure values of cylinder chambers which are denoted as chamber A and chamber B respectively. \( S_{q(k),q(k+1)} \) denotes a state transition from \( q(k) \) to \( q(k + 1) \), and \( S_{q(k+1),q(k)} \) denotes a state transition from \( q(k) \) to \( q(k) \), and \( p_{\text{max}}(q(k)) \) and \( p_{\text{min}}(q(k + 1)) \) which indicate the upper and lower threshold of force capacity for state \( q(k) \) and \( q(k + 1) \). As the continuous process evolves, further state transition might be invoked.

Continuous dynamics
Assume that the pressure drop across a switching valve is very small thus can be neglected and the supply pressure of each rail stays constant, therefore, between two consecutive mode switches, pressure of the chamber that is connected directly to one rail is also constant as the rail pressure, however, pressure of the other chamber changes continuously due to the resistance of meter-in orifice and flow compressibility. In each operating state, the continuous system dynamics can be modeled with linearized functions as

\[ x(t) = f_{q(k)}(x(t), u_c(t)) \quad (5) \]

where \( x = (\dot{x} \quad p_A \quad p_B)^T \in X \) is the vector of state variables, input of continuous system is the valve opening signal, as \( u_c(t) = y_c \), the indexed vector field \( f_{q(k)} \) is associated with discrete state \( q(k) \in Q \).

After each transition, the continuous state variables \( x = (\dot{x} \quad p_A \quad p_B)^T \) will change to the valuation of \( x' = (\dot{x}' \quad p_A' \quad p_B')^T \), thus, a reset mapping function is thereby defined as

\[ R(q', x') = g(x_q, x'_q) \quad (6) \]

Such a hybrid system may exhibit strong discontinuity in continuous dynamics, which is caused by the discrete state transitions. Evolution of continuous dynamics are separated into different sections by discrete states with different pressure rails being selected at the supply side according to Tab. 1.

Mode switching
The main challenge here is to design a controller which can deal with the dilemma between mode switching and smooth velocity control. In widely used excavator hydraulic systems, such as Load-sening, the valve spool is directly driven by hydraulic joystick signals given by the operator, which allows the operator to control the cylinder motion intuitively.

In the STEAM system, open-loop control is not capable of dealing with the discrete changes in continuous state variables introduced by mode switching, therefore, a flow controller must be implemented. The control task is not only to make sure the cylinder motion will follow the joystick signal faithfully, it is also necessary for the proportional valve to counteract the effect of mode switching. Practice implications or the effect of such a switch on an operator are different and not well documented in literature, so the analysis here is focused on sudden velocity and pressure changes.

Assume that the load variation is relatively slow, and the piston stays at the same position through such a small time interval, thereby \( \Delta F_1 \approx 0, \Delta x \approx 0 \), and the hydraulic capacitance also remains constant for each cylinder chamber.

To analyze the transient response of cylinder motion, fluid compressibility must be considered. The cylinder dynamics can be modelled by the following differential equations

\[ \Delta \dot{p}_A = \frac{1}{C_{HA}}(\Delta Q_A - A \dot{x}) \quad (7) \]
\[ \Delta \dot{p}_B = \frac{1}{C_{HB}}(\alpha A \Delta \dot{x} - \Delta Q_B) \quad (8) \]

where, \( \Delta Q_A \) and \( \Delta Q_B \) are the incremental changes in flow entering or leaving the cylinder, which are given by linearized orifices equations below

\[ \Delta Q_A = K_{QP,A}(\Delta p_1 - \Delta p_A) \quad (9) \]
\[ \Delta Q_B = K_{QP,B}(\Delta p_2 - \Delta p_B) \quad (10) \]

where \( K_{QP,A} = \frac{\partial q_{a1}}{\partial p_{1\text{op}}} \), \( K_{QP,B} = \frac{\partial q_{a2}}{\partial p_{1\text{op}}} \), \( K_QV = \frac{\partial q_{v}}{\partial x_{1\text{op}}} \).

Abrupt changes of cylinder acceleration and velocity may result from the pressure changes in cylinder chambers, since the hydraulic force and load force are not balanced when mode switching occurs. Applying Newton’s law of motion leads to this transient dynamics

\[ m \Delta \ddot{x} = (\Delta p_A - \alpha \Delta p_B) A - \Delta F_L - d \Delta \dot{x} \quad (11) \]
Figure 12. Linearized model of system transient response during mode switching

Define the system state variables as

$$\mathbf{x} = (\Delta \dot{x} \; \Delta p_1 \; \Delta p_2 \; \Delta x_v \; \alpha A \; \Delta x_\ldots \; \Delta x_\ldots \; \frac{d}{m})^T$$

(12)

And this model is characterized as a multiple inputs single output system, with

$$\mathbf{u} = (\Delta p_1 \; \Delta p_2 \; \Delta x_v)^T$$

(13)

$$\mathbf{y} = \Delta \dot{x} = \Delta v$$

(14)

in which, pressure changes at the supply can be regarded as step signals due to sudden mode switching. By performing Laplace transform of equations (7) to (11), the linearized system model depicted in Fig. 12 can be obtained, and mathematically the multiple inputs and single output relations can be put in state space form as

$$s\mathbf{X}(s) = \mathbf{AX}(s) + \mathbf{BU}(s)$$

(15)

$$\Delta \mathbf{V}(s) = \frac{1}{N(s)} \left[ \frac{\Delta K_{\text{QP,B}}}{C_{H,A}} (s - \frac{K_{\text{QP,B}}}{C_{H,B}}) \mathbf{G}_c(s) + \frac{\alpha AK_{\text{QP,B}}}{C_{H,B}} (s + \frac{K_{\text{QP,A}}}{C_{H,B}}) \mathbf{G}_c(s) \right] \Delta \mathbf{P}(s)$$

(16)

where

$$N(s) = \left(s + \frac{d}{m}\right)G_{c_1}(s)G_{c_2}(s) + \frac{\alpha^2}{mc_{H,A}}G_{c_2}(s) + \frac{\alpha^2}{mc_{H,B}}G_{c_1}(s),$$

with $G_{c_1}(s) = s + \frac{K_{\text{QP,A}}}{C_{H,A}}, \; G_{c_2}(s) = s - \frac{K_{\text{QP,B}}}{C_{H,B}}.$

Transfer functions for each input and output are as follows

$$G_1(s) = G_{\Delta x_\ldots \Delta p_1}(s) = \frac{1}{N(s)} \cdot \frac{K_{\text{QP,B}}}{C_{H,B}} G_{c_2}(s)$$

(17)

$$G_2(s) = G_{\Delta x_\ldots \Delta p_2}(s) = \frac{1}{N(s)} \cdot \frac{\alpha AK_{\text{QP,B}}}{C_{H,B}} G_{c_1}(s)$$

(18)

$$G_3(s) = G_{\Delta x_v \Delta p_1}(s) = \frac{1}{N(s)} \cdot \frac{\alpha AK_{\text{QP,B}}}{C_{H,B}} G_{c_1}(s)$$

(19)

It is reasonable to cancel the constant term in polynomial $N(s)$, as it is considerably smaller than other terms with higher orders in the high frequency range present during a switch, thus these transfer functions can be simplified into

$$G_i(s) = \frac{K_{\text{QP,B}}}{s(\omega_i + 1/T_i) + \omega_i^2}, \; i = 1, 2, 3$$

(20)

where,
\[ \omega_0 = \sqrt{\frac{K_{QP,A} K_{QP,B}}{c_{HA} c_{HB}} + \frac{d K_{QP,A} + d^2}{m c_{HA}} + \frac{d K_{QP,B} + d^2}{m c_{HB}}} \]
\[ D = \frac{d c_{HA} c_{HB} + m (K_{QP,A} c_{HB} - K_{QP,B} c_{HA})}{c_{HA} c_{HB}} \]
\[ K_1 = \frac{d K_{QP,A}}{\omega_0^2 c_{HA}}, \quad T_1 = -\frac{c_{HB}}{K_{QP,B}}, \quad K_2 = \frac{d^2 K_{QP,B}}{\omega_0^2 c_{HB}}, \quad T_2 = \frac{c_{HA}}{K_{QP,A}} \]
\[ K_3 = \frac{c_{HA}}{K_{QP,A}} \]

These transfer functions indicate that the cylinder motion is sensitive to pressure changes as well as valve opening. Cutoff frequency \(1/T_1\) of each PI element in these transfer functions is estimated to be smaller than the eigen frequency \(\omega_0\) in value. Within the frequency band between \(1/T_1\) and \(\omega_0\), the magnitude of transfer function \(G(s)\) is determined by \(K_1\). Since the gain of \(G_{\Delta,\Delta x}(s)\) is estimated to be larger than the other two, it is possible to compensate for the velocity changes through dynamic adjustment of the valve opening. To ensure a smooth transition and allow the operator to control the cylinder motion intuitively, a flow controller must be implemented to compensate for the pressure changes resulting from mode switching.

**SUMMARY AND OUTLOOK**

A switched hydraulic system may involve either autonomous switching due to its inherent design (Type A), or controlled switching implemented to save energy (Type B). This paper has reviewed some examples from recent developments in this field. Despite the great potential of type B systems in improving efficiency, system characteristics such as undesired transient performance or instability caused by mode switching, are a challenge. The STEAM system is currently being developed as a possible solution for energy efficient mobile applications. With the single edge meter-out control configuration, a specific switching logic is designed as a tradeoff between system performance and efficiency improvement. Additionally, flow compensation is also necessary to smooth actuator motion during discrete mode switching. Further research will be conducted to implement a feasible hybrid control strategy based on the aforementioned characteristic analysis.

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