Energy efficient digital hydraulic valve control utilizing pressurized tank line

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Abstract

Digital hydraulics is a quickly developing alternative for applications requiring cylinder motion control. This paper focuses on digital hydraulic valve system consisting of parallel connected on/off-valves. The energy efficiency of the system relies on distributed valve configuration, electrically load sensing supply pressure and pressurized tank line. Topics of the paper include the analysis of the system dynamics through linear modelling, development of non-linear control mode selection algorithm, steady-state analysis of the energy efficiency and experimental testing of the control performance and the energy efficiency. Measurements show that energy losses can be reduced 53 - 71 % when compared to traditional load sensing proportional valve.

KEYWORDS: Digital hydraulics, Controller design, Pressurized tank line

1. Introduction

At the moment the research in the field of digital hydraulics include valve control /1/, /2/, /3/, switching transformers /4/, reciprocating transformers /5/, pump control /6/, /7/ and secondary control of a multi-chamber cylinder /8/. The wide variety of approaches listed aim at improving the energy efficiency of hydraulic systems. When modifying an existing system or developing a completely new one, there is always a compromise to be done between the functionality, efficiency and price of the machine. The research presented in this paper aims at developing a highly energy efficient digital hydraulic system, that will not require complete redesign of the hydraulic system (e.g. a mobile machine with a load sensing supply unit).

The studied system includes a digital valve, which controls the cylinder, a supply unit with electrical load sensing function and a pressurized tank line. The modifications needed to typical state of the art mobile machine are the replacement of the proportional control valves with their digital hydraulic counterparts and pressurization of the tank line with relief valve and an accumulator. The research presented builds on previous studies, which include the analysis of the energy efficiency of a digital hydraulic valve with typical zero-pressure tank line. The novelty of the paper is based on the newly developed control mode selection scheme that allows the utilization of the pressurized tank line and further improved energy efficiency.

The digital valve studied includes four separate control edges, each controlling the fluid flow between cylinder chambers and the two supply lines (the pump line and the pressurized tank line). Each control edge is realized using parallel connected on/off-valves (see **Figure 1**), which are optimally controlled using a model-based controller. As the cylinder chambers can be fed from either the load sensing supply line or the tank line, there is a strong potential for energy efficient operation through different control modes, which are utilized according to load force variations.





The work presented includes the analysis of the energy saving potential of the digital hydraulic valve concept with the pressurized tank line for a single actuator system. In addition, a steady-state analysis is performed for multi-actuator system to get a broader insight on the overall system efficiency. As the varying load forces and different velocity demands of the simultaneously driven actuators affect the energy efficiency of the overall system in highly non-linear way, the analysis is performed for steady-state situation by discretizing the different velocity demands and load forces. In order to evaluate the feasibility of the proposed control method in practise, the measurements are carried out on a single actuator test system, which mimics the properties of a mid-size mobile machine boom. The results of the measured load cycle are compared with the measurement results of a conventional load sensing proportional valve system.

2. Analysis of the energy efficiency

The energy efficiency of the digital hydraulic valve control is based on the distributed structure of the valve package and different control modes.

2.1. The control modes

Possible control modes are presented in **Figure 2**. Depending on the load force and the supply pressure, all the control modes are feasible for both moving directions.





Control modes PT and TP are typical inflow-outflow modes, when used for extending and retracting motion, respectively. However, the pressurized tank line allows the use of control mode PT also for retracting motion and TP for extending motion, when cylinder is subjected to overrunning loads. These control modes enable significant energy recuperation. Control modes PP and TT are differential connections via supply line and tank line.

2.2. Feasibility of the control modes

In order to analyze the efficiency of different control mode combinations, the feasibility of each mode has to be determined first. The feasibility of each control mode depends on the measured load force *F*, measured tank line pressure p_{T} , target pressure difference Δp , acceptable minimum pressure p_{min} and maximum pressure p_{max} as well as cylinder chamber areas A_A and A_B .

For example the extending control mode (PTe) is feasible (1) whenever the load force does not require exceeding of the maximum pressure on the A-side (restricting loadings) or on B-side (overrunning loadings). Similar conditions can be written for each control mode assuming that the supply pressure p_P can be chosen freely.

$$M_{PTe} = F \le (p_{max} - \Delta p)A_A - (p_T + \Delta p)A_B \text{ AND } F \ge p_{min}A_A - p_{max}A_B \tag{1}$$

The energy consumption of a control mode depends on the actuator velocity and the supply pressures. Whereas the tank line is pressurized to a constant pressure level, the pump line pressure is controlled by the model-based digital valve controller.

2.3. Efficiency of a single actuator system

To investigate the efficiency of the proposed system, example parameters are used: cylinder size 80/40 mm, which results in $A_{\rm A}$ =5027 and $A_{\rm B}$ =3770 mm², feasible pressure

range p_{min} = 1 MPa and p_{max} = 20 MPa, desired pressure differential over the control edges Δp = 1 MPa and tank line pressure p_{T} = 3 MPa.

The power consumption of a proportional valve is also calculated for comparison. The geometry of the proportional valve is set to match the cylinder areas. Power consumption, optimal supply pressure and optimal control mode is presented in **Figure 3**.





The power consumption of the proportional valve is calculated by multiplying the supply pressure by the supply flow rate. In case of the digital system, the input power includes also the power taken or fed back to the tank line.

2.4. Efficiency of a multi-actuator system

Steady-state analysis of the efficiency of two-actuator system is carried out by discretizing the feasible load force range (resolution 2 kN) and velocity range of the actuators -200 ... 200 mm/s (resolution 20 mm/s) and calculating the hydraulic loss related to load sensing proportional valve and digital valve at each possible load case. Using the parameters of the single actuator study, the losses are reduced 55 %, when compared to load sensing proportional valve system. When analyzing a three-actuator system, the losses are reduced 51 %. Load force resolution is set to 10 kN to reduce computational load in the three-actuator case.

3. Online selection of the control mode

To be able to switch from control mode to another during motion, the control mode selection algorithm along with suitable chamber pressure references are developed.

3.1. Chamber pressure references

The chamber pressure references are defined so that control mode switching during motion does not induce a step wise change to the chamber pressures, which enables disturbance free velocity tracking also during control mode change. The control mode selection builds on the algorithm developed in /1/. To enable the use of pressurized tank line, the chamber pressure references are derived and presented graphically in **Figure 4**. In order to allow stepless mode changes, the control mode PP is exluded from the mode selection algorithm. Only the modes PT, TP and TT are enabled as they allow the use of identical chamber pressures throughout the overrunning (negative) load force range.



Figure 4: Pressure references for extending motion

3.2. Mode selection algorithm

As the supply pressure is controlled by a variable displacement pump, the mode switching instants are demanding from supply pressure control point of view. In case switching happens from control mode TPe to PTe, the pump displacement is required to turn rapidly from negative to positive value. However, by introducing a transition via the control mode TTe, the requirements for pump dynamics are decreased. Load force range of 10 kN is reserved for mode TTe between control modes TPe and PTe. Another important aspect is the control of tank line pressure. Sufficient tank line pressure is kept by enabling only control modes with positive tank line flow, whenever the tank line pressure has decreased below certain limit value.

The control mode selection is realized using a Stateflow state machine. Two separate control modes are used: The target control mode is the desired control mode with optimal energy efficiency. The supply pressure reference is calculated according to the target control mode. Resulting from the dynamics of the supply pressure control, the supply pressure will differ from the reference to a certain degree. Therefore the actual control mode is used as a basis for the valve control, and it is determined using the measured supply pressure.

4. Linear analysis of the control performance

Linear model can be utilized to analyse the control performance of the developed valve control scheme.

4.1. Modelling

Figure 5 presents an upper level view of the input/output characteristics of the different subsystems of the digital hydraulic cylinder control. Pump and pump pressure filter are modelled as first order systems with time constants of 15 ms. Controller and pump pressure controller are both zero-order systems implementing the linearized model of the model based digital hydraulic controller. Cylinder model uses flow rate as an input and force as an output implementing the typical lumped parameter model of the cylinder dynamics. Load model consists of the external load force as an input, first order piston velocity dynamics and friction force. The effect of the piston position to capacitance of the hydraulic cylinder chambers is neglected to reduce the complexity of the model.



Figure 5: Linear model of the controller-cylinder-load system

Modelling is straight forward for other subsystems excluding the model-based digital hydraulic controller. Equations 2-3 represent the command signal for DFCU PA and BT in inflow-outflow control mode:

$$u_{PA} = \frac{v_{ref} A_A}{K_{PA} \sqrt{p_{P_est} - \frac{F_{est} + (\Delta p + p_T) A_B}{A_A}}}$$
(2)

$$u_{BT} = \frac{v_{ref} A_B}{K_{BT} \sqrt{\Delta p}}$$
(3)

The controller is capable of controlling the pressure level and piston velocity simultaneously. The controller takes into account the effective flow capacity of the DFCU, cylinder chamber areas, measured pressures and the desired pressure difference. The linearization of the controller model leads to MIMO system with two outputs (u_{PA} and u_{BT}) and three inputs (v_{ref} , $p_{P_{est}}$ and F_{est}). Pressure of the tank line p_{T} and the desired pressure difference Δp are considered as constants in this study.

Table 1 presents the system parameters used. Other parameters equal to the ones introduced in chapter 2.3.

Load mass = 40 000 kg	Hoses = Ø19 mm – 1 m
Linear friction coefficient = 500 N / (m/s)	Cylinder stroke = 200 mm
Flow coefficient = $4.7*10^{-7} \text{ m}^3 / (\text{sPa}^{1/2})$	Linearization position = 100 mm
Bulk modulus of the oil = 1200 Mpa	Linearization velocity = 30 mm/s
Bulk modulus of the hose = 400 MPa	

Table 1: Parameters of the linear study

4.2. Disturbance rejection

To enable accurate model-based velocity control the load force of the actuator has to be known. In /1/ a second order low pass filter is used to filter out high frequency content of the measured load force. Filtering is crucial from stability point of view as for example the closing and opening of the on/off-valves induce pressure variations, which would be erroneously interpreted as load force variations without the low pass filter. In /1/ cutoff frequency of 8 rad/s is used with a damping ratio of 0.7. One result of the linear analysis is presented in **Figure 6**, which shows that in this case it is beneficial to use a first order load force filter. The time constant is set to 170 ms, which matches the 0-100% rise time of the force estimator of /1/ to stepwise load force disturbance.



Figure 6: Force disturbance rejection

On the right hand side of the figure a stepwise load force disturbance of 10 kN is presented. The step response shows that the first order filter is capable of estimating the load force with significantly smaller oscillations. The benefit of the first order filter is also seen in the velocity response as the model based controller gets a well damped load force estimate, when calculating the valve control signals.

5. Experimental testing

To demonstrate the feasibility of the proposed system, a single actuator system is tested and also the energy consumption is measured. The same measurements have been previously done with a proportional valve /1/ and the results can be compared. The only significant change in the system configuration is the introduction of slightly smaller cylinder (in /1/ the chamber areas are 6234 and 4199 mm²).

5.1. Test setup

The test setup is presented in **Figure 7**. Pressurized tank line consists of a hydraulic accumulator with capacity of four litres and a pressure relief valve, which is set to 3 MPa. Loadings are the same as in /1/. The response time of the KSDEU valve (12 VDC coil) is roughly 10 ms with a special valve driver, which boosts the opening of the valve with a 8 ms long 48 V pulse and decreases the coil current to zero faster than two milliseconds when closing the valve.

The upper-level controller consists of low pass filtered (time constant: 170 ms) Pcontroller with gain 2.7 and velocity feedforward with gain 0.55. The time constant of the load force filter is 100 ms, supply pressure filter 15 ms, tank line pressure filter 200 ms.



Figure 7: Hydraulic diagram of the test system

5.2. Measurement results

Measurement results are shown in **Figures 8-10** for the three different loadings. Along with position and velocity measurements, the control mode used, load force with its estimate and supply pressure are presented. Also the output energy (thick grey), energy taken from the pump line (dashed) and sum of energy taken from both supply lines (continuous) are presented.



Figure 8: Almost balanced loading



Figure 10: Overrunning loading

The same trajectories have been measured with a proportional valve in /1/ and the measured energy losses were 8.1, 10.6 and 9.3 kJ for the loadings A, B and C respectively.

6. Discussion and conclusions

The method allows a significant reduction of energy consumption for both a single actuator system and a multi-actuator system. **Figure 11** gives a summary of measured energy losses of proportional valve and digital valve. Depending on the inclusion of the energy balance of the pressurized tank line, the average reduction of energy losses with digital valve is 53 - 71 % when compared to load sensing proportional valve.



Figure 11: Measured energy loss of proportional valve and digital valve in three load cases [kJ]

Finally, it can be concluded that even though the system with its model-based controller is strongly non-linear, properly linearized model can still be used for controller design and analysis.

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9. Nomenclature

F	load force	Ν
$p_{ op}$	tank line pressure	Pa
Δp	target pressure difference	Pa
p_{min}	minimum pressure	Pa
p_{max}	maximum pressure	Pa
A_{A}, A_{B}	area of cylinder chamber	m²
V _{ref}	velocity reference	m/s
K_{PA}, K_{BT}	flow coefficients	m ³ /s/Pa ^{1/2}
U _{PA,} U _{BT}	DFCU command signal	-
F _{est}	Estimated load force	Ν
$p_{P_{est}}$	Estimated supply pressure	Pa