A COMBINED MULTI-PRESSURE SWITCHED INERTANCE HYDRAULIC CONTROL CONCEPT OF A DIFFERENTIAL CYLINDER

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Control of hydraulic drives using more than two pressure lines is studied as a means for improving energy efficiency, e.g. for excavators or for hydraulic power take off from sea wave power plant systems. All combinations of switching n different pressure levels to a differential cylinder provide in general 2.n force levels. In order to facilitate forces between these levels for a proper motion control, this digital concept was combined with throttling control. This, of course, causes energetic losses.

In this paper a combination of multi-pressure and switched inertance control is presented and studied theoretically by mathematical modelling and simulation. The integration of the switched inertance part requires only a pipeline as an inertance element and a so called “Mikota resonator” flow and pressure ripple suppression.

Simulation results show a good motion control performance and an efficiency gain over intermediate force adjustment by throttling in load lifting.

KEYWORDS

Multi-pressure line hydraulics, switched inertance control, MikotA resonator

1 INTRODUCTION

Numerous concepts to avoid intrinsic losses in hydraulic drives have been proposed and investigated in the last decades: Primary and secondary control by variable displacement pumps and motors or by speed variable electric motors, load sensing, independent metering control, multi-chamber cylinders, multi-pressure line drives, hydro-transformers, and switching hydraulics. From these principles so far only primary and secondary control found significant acceptance by machine and vehicle manufacturers and their customers.

Among the various digital concepts, the multi-chamber cylinder and multi-pressure line systems gained high attention primarily in mobile hydraulics. A good introduction in both concepts is given in [Donkov 2020]. [Linjama 2009] analyses the multi-chamber cylinder; an impressive demonstration of its high efficiency and recuperation rate in a log loader application provides the video [Norrhydro 2011]. A multi-pressure line control of an excavator was studied by [Vukovic 2013]. The combination of multi-chamber cylinder and multi-pressure line control for ocean wave power take off are presented in [Hansen 2016].

Both concepts have very low losses, if operated in a strictly digital way, i.e. with switching valves only. Then, however, only a discrete set of hydraulic forces (digital forces) can be realized which does not allow a precise motion control particularly for slow motion and small load inertia. For force smoothing, some throttling is added. To this end, additional valves are integrated. Throttling causes losses, which depend on the difference between the actual and the next digital force.

An additional problem causes the jump from one to another discrete force level. It provokes oscillations in the hydraulic and the attached mechanical systems.

In this paper a different approach to smoothen the hydraulic output force of a multi-pressure drive is discussed by combining it with switched inertance control. The latter is a sub-class hydraulic switching control which was intensively studied in the last three decades, see [Scheidl 2013] and [Chenggang 2020] for an overview. The basic objective of switching control is mimicking the effect of continuous valves by a high switching frequency on the one hand and by avoiding some of their typical downsides on the other hand. These problems are: energetic losses by throttling, limited accuracy by hysteresis, metering edge overlap and susceptibility to oil contamination. The combination of multi-pressure line and switching control is redundant, since switching control can provide a quasi-continuous output force by itself. The advantage of the combination is that the pressure amplitudes of switching are smaller, since an intermediate pressure level is available which can reduce pulsation and compression losses due to switching.

Another novelty of this paper is the attenuation of pressure pulsation caused by switching by a so called Mikota resonator [Mikota 2001], which in essence is a modified version of the well-known Helmholtz resonator.

This is a first theoretical investigation of the concept to explore its basic feasibility, assess its performance characteristics, and find main challenges for a technical realization.

2 DRIVE CONCEPT

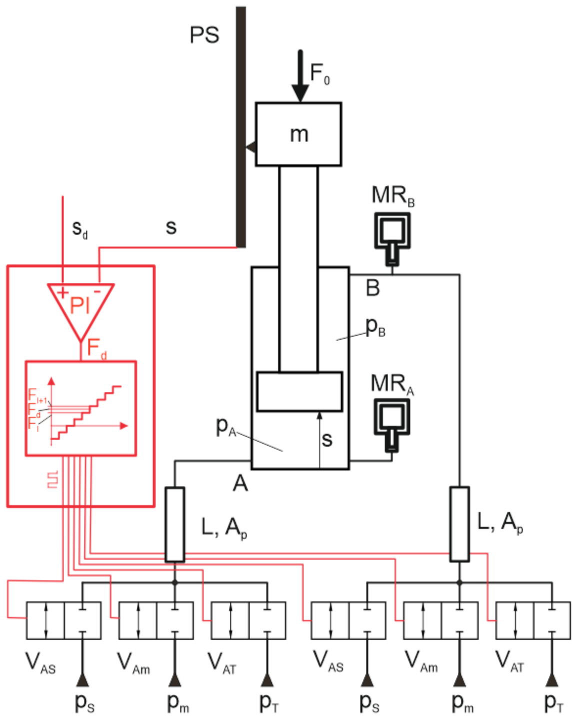

Figure 1. Schematic of the combined multi-pressure switched inertance hydraulic drive and Mikota resonators (MR) for pulsation attenuation.

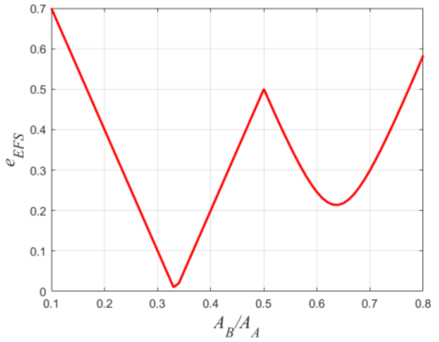

Figure 2. Related force stepping error as function of area ratio for the optimum mean pressure value $p_m = (p_T+p_S)/2$.

Fig. 1 shows the schematic of the drive concept. Three pressure lines provide a tank pressure p_T (possibly elevated), a medium pressure p_m, and a high pressure p_S. Each cylinder chamber (A, B) can be connected to any of these pressures by in total six 2/2 way on-off valves. Nine meaningful combinations of the valve states (only one valve per chamber is on) correspond to nine different (digital) cylinder forces. If $p_m = (p_T+p_S)/2$ and the rod sided area (A_B) is one third of the piston area (A_A) the forces are equally spaced between minimum and maximum forces $F_{min} = p_T A_A - p_S A_B$ and $F_{max} = p_S A_A - p_T A_B$. Other area ratios cause uneven force steps. Quantification by the following related force stepping error e_{EFS}

$$e_{EFS} = \left\| \Delta\left(\Delta \frac{\mathbf{F}}{F_{\max}} \right) \right\|_2 \; ; \quad \Delta \mathbf{u} = \left[u_{1+k} - u_k \right] \qquad (1)$$

$$k = 1..\,\text{dimension}(u) - 1$$

is shown in Fig. 2.

In order to realize forces between the digital forces F_i (i=1..9) the valves can be switched at high switching frequency (reasonably between 40 and 100 hertz). If two valves at one side, e.g. V_{AT} and V_{Am} at A-side, switch alternately with a certain duty cycle κ the time average pressure $\bar{p}_A$ is

$$\bar{p}_A = p_T + \kappa\left(p_m - p_T\right) \qquad (2)$$

The resulting average cylinder force

$$\bar{F}_Z = A_A \bar{p}_A - A_B \bar{p}_B \qquad (3)$$

can be achieved with different combinations of $\bar{p}_A$, $\bar{p}_B$, even if the system switches between the pressure levels which correspond to the neighbouring digital forces F_i, F_{i+1}. A simple rule is applied to obtain uniqueness for PWM control. If $F_Z < 0$, PWM is applied to B side, else to A side. Thus, switching takes place only between two valves of one side.

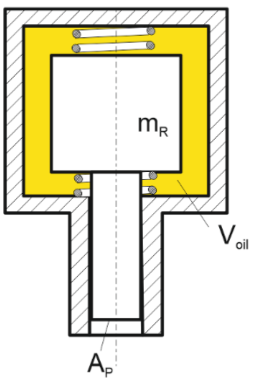

Figure 3. Mikota resonator.

Fast switching provokes pulsation in the hydraulic system (pressure and flow rate) and oscillations of the mechanical motion. The usual mean to reduce these unwanted effects is a high enough hydraulic capacitance at the end of the inertance tube, analogue to the smoothing capacitance in electrical switching converters. The higher the capacitance the lower is the pulsation, but unfortunately also the softness of the hydraulic drive. Other principles to reduce oscillations are oscillating attenuation devices, such as the Helmholtz resonator or the lambda/4 resonator. They have a high, but small banded working range since they exploit a resonance effect. The "Mikota" resonator (Fig. 3) is a modified Helmholtz resonator and replaces inductance of a fluid tube by the inertance of a solid body (m_R). That body is embedded in the capacitance volume (oil volume V_{Oil}) and is connected to a piston (area A_P) which separates that volume from the entrance port. The advantage is the more compact design due to the higher inertia of solid materials like steel compared to fluids. If there is more than one frequency to attenuate, additional, properly tuned resonators can be employed. The resonance, hence optimum working frequency f of that drive is

$$\omega = f\,2\pi = A_P \sqrt{\frac{E}{V_{Oil} m_R}}\,. \qquad (4)$$

The two springs shown in Fig. 3 assure a centring of the mass in V_{Oil} and have negligible influence on f. A further dimensioning rule assures avoidance of cavitation [Mikota 2001, 2002]. This rule yields large size dampers for low average outside pressure. Therefore, that like other resonators is well not suited for low pressure systems.

The controller comprises two major units: An outer control loop, which derives the desired cylinder force F_d from the superior control task, and a discretization unit which determines the proper valve switching. In this paper, the outer control is a simple PID controller, which should make the actual position s(t) following the desired position $s_d(t)$. Its output is the desired cylinder force F_d which is adjusted in a feed forward fashion by the discretization unit. It operates in three steps: (i) Determination of the two neighbouring digital force steps F_i,F_{i+1}; (ii) determination of the best suited line (A or B) for PWM control and the corresponding duty cycle to adjust F_d in an average way; (iii) generation of the appropriate signals for the six on-off valves.

Outer controller and force control are clearly separated. Thus, the outer controller can be different from the one of this paper. It can, for instance, involve a human operator to generate the control input or to close the outer control loop by his observations, as is usually the case for many mobile machines.

3 MODELING AND SIMULATION

The assessment of this drive concept is done by a numerical model. It uses Simulink as simulation platform in combination with the hydraulics open component library hydrolib [Manhartsgruber 2012]. The switching valves have a finite response dynamics, modelled by a PT2 dynamics. The inductance tubes use the hydrolib pipe elements with laminar viscid wave propagation solved by the methods of characteristics [Manhartsgruber 2015]. The differential cylinder model considers pressure build-up dynamics in both chambers. One Mikota resonator is attached to each chamber and modelled by obvious equations of motion of the oscillating mass (m_R), pressure build-up in V_{Oil} and the flow rate and pressure coupling to the cylinder chambers. The linear equation of motion of the load mass m with the cylinder pressure forces and friction and the dead load F_0 are modelled in a straightforward way.

The controller blocks (PID and discretization) employ standard Simulink elements.

The exemplary system and controller data used for assessment are specified in Table 1. They do not refer to a specific application but generally rather to some mobile application than to heavy duty or high precision industrial manufacturing systems.

Fig. 4 shows the Simulink model with the used sub-systems, Fig. 5 and 6 the three sub-models for motion trajectory generation, PID control and for the "discretizer".

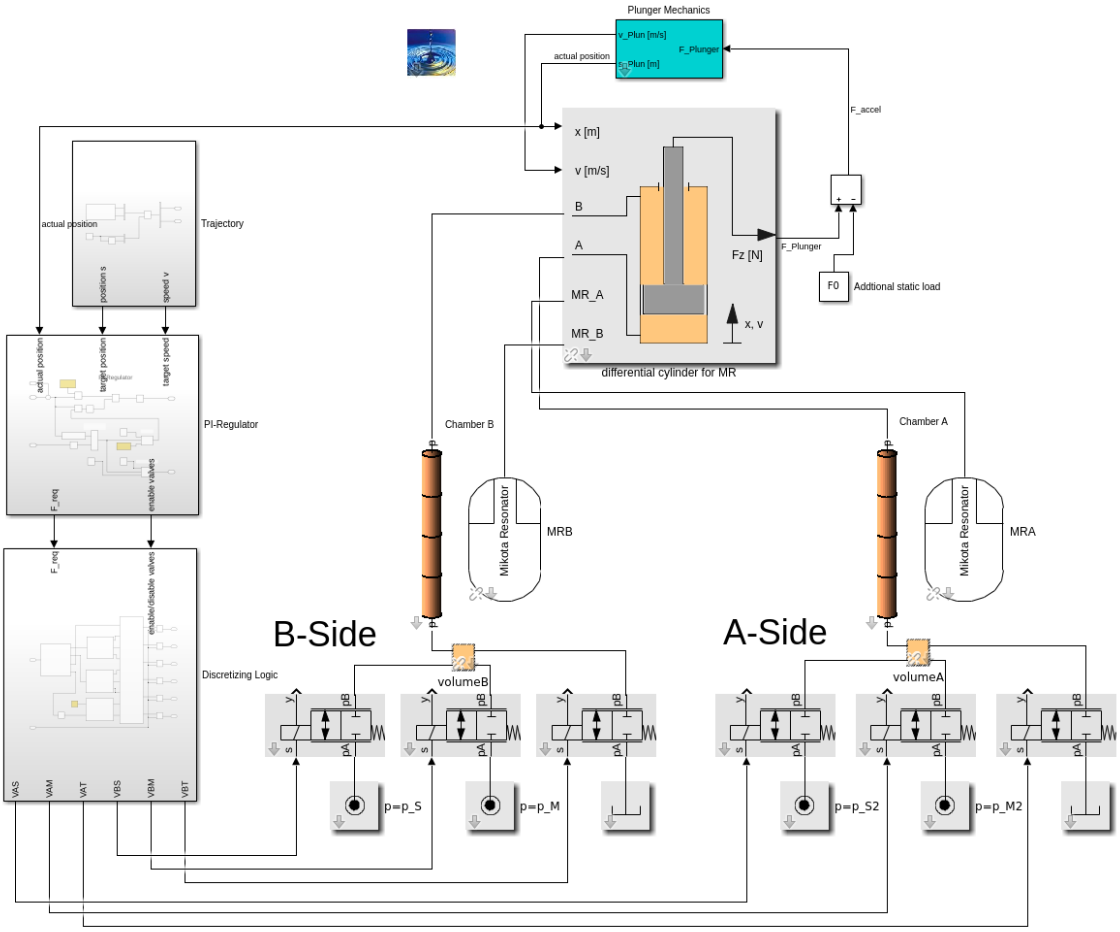

Figure 4. Simulink model of the drive – upper layer.

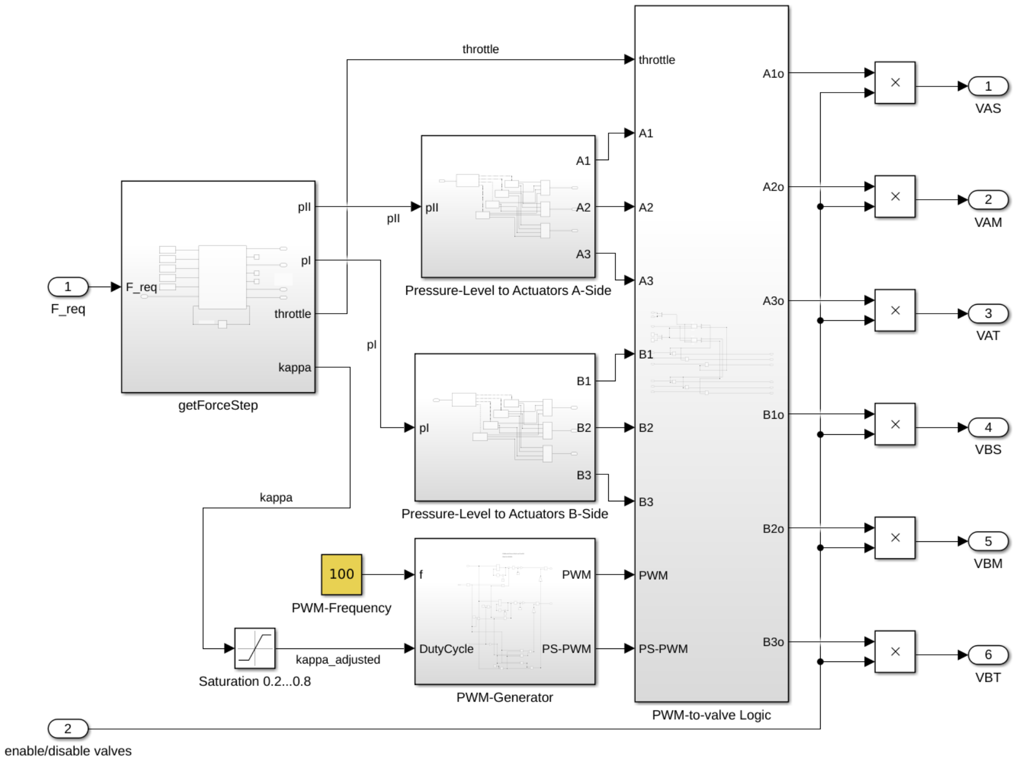

Figure 5. Simulink model of the discretization sub-system.

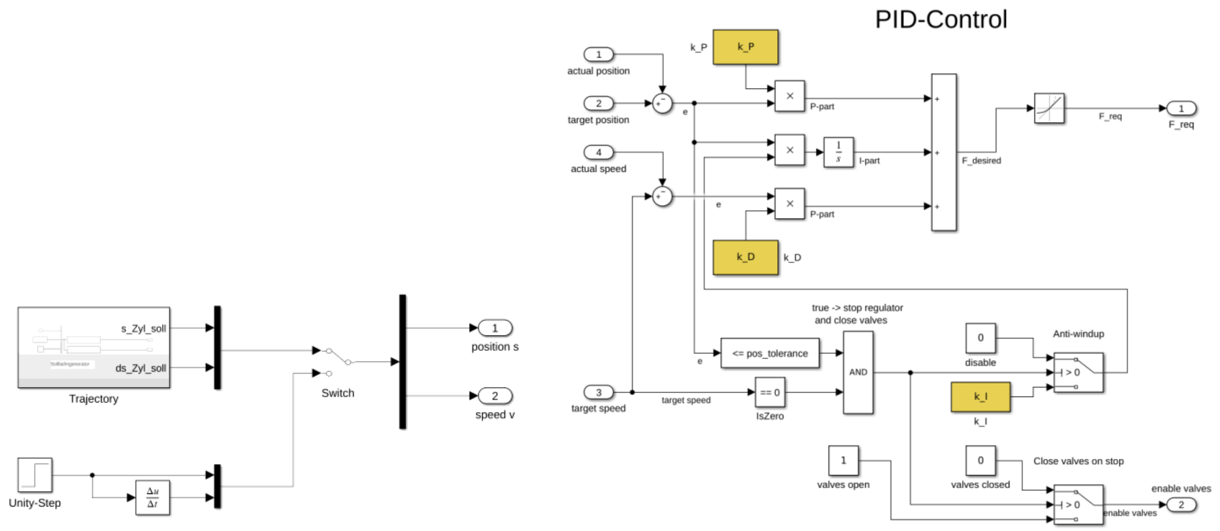

Figure 6. Trajectory specification and PID controller sub-systems.

Name	Value	Unit	Description
rho	8,60E+02	kg/m³	oil density
nu	4,60E+01	mm²/s	kinematic viscosity
p_S	2,00E+02	bar	high pressure
p_M	1,00E+02	bar	middle pressure
p_T	1,00E+01	bar	tank pressure
Plun_d	6,30E-02	m	plunger chamber diameter
Plun_A1	3,12E-03	m²	plunger chamber area
Plun_A2	1,04E-03	m²	rod-side plunger chamber area
Plun_d2	5,14E-02	m	rod diameter

Plun_m	2,00E+03	kg	plunger mass
Plun_l	1,50E+00	m	rod length
Plun_frict	1,00E+04	Ns/m	friction coefficient
pos_tolerance	1,50E-02	m	postion offset tolerated by controller
initialPos	7,50E-01	m	initial position of the plunger
capDeadVol	5,00E-01	l	cap side dead volume
rodDeadVol	5,00E-01	l	rod side dead volume
F0	1,00E+04	N	additional static load
p_A0	3,21E+01	bar	Initial pressure to compensate F0
F_Plun_max	6,23E+04	N	max. plunger force

u_min	1,00E-03	m	maximum contact-deformation at max. force
kc	6,23E+07	N/m	contact stiffness of plunger end stops
volA	5,00E-02	l	initial volume of connecting volume to A
volB	5,00E-02	l	initial volume of connecting volume to B
pVolA	3,21E+01	bar	initial pressure in volume connecting to A
pVolB	1,00E+01	bar	initial pressure in volume connecting to B
vot	2,00E-03	s	valve opening time
vofr	3,14E+03	rad/s	valve operating frequency
cofr	3,14E+03	rad/s	cut-off frequency
vdc	9,00E-01	1	valve dampening coefficient
vip	0,00E+00	1	initial valve position
nfr	1,00E+02	l/min	nominal valve flow rate
npd	5,00E+00	bar	nominal valve pressure drop
lfr	2,00E+02	l/min	laminar flow rate
v_max	3,00E-01	m/s	maximum plunger speed
Q_max	9,35E-04	m³/s	maximum volume flow
Re_max	2,00E+03	1	maximum tolerated Reynolds number
d_pipe	1,29E-02	m	Inner pipe diameter
L_pipe	2,00E+00	m	pipe length
k_P	1,00E+05	1	proportional gain of controller
k_I	2,00E+04	1	integral gain of controller
k_D	1,00E+04	1	differential gain of controller
rate_lim	1,40E+05	N/s	rate limit of controller output (slew rate)
cor	2,40E-01	1	cut off ration (to avoid overlapping of valve opening)
kappa_low	1,00E-01	1	lower limit for duty cycle
kappa_high	9,00E-01	1	upper limit for duty cycle
fx	1,00E+02	1/s	PWM-frequency
B	1,60E+09	Pa	bulk modulus of oil
m	2,00E+00	kg	piston mass of MR (Mikota Resonator)
V_HS	1,05E-03	m³	volume of MR
d_P	3,00E-02	m	working piston diameter of MR
c_CS	1,20E+04	N/m	stiffness of one centering spring in MR
l_Gap	1,00E-02	m	length of sealing gap in MR
s_Gap	1,20E-05	m	sealing gap in MR
eta	3,96E-02	kg/(m*s)	dynamic viscosity
A_P	7,07E-04	m²	working piston area of MR

Table 1. List of parameter values

4 SIMULATION RESULTS

As said, this is a first assessment of a new principle for applications with moderate motion precision requirements. Hence, only a very elementary load and motion scenario is investigated. The first is a ramp-up and down of the cylinder position s with a constant load F_0. The selected switching frequency of 100 hertz is rather an upper bound of currently feasible switching frequencies. It allows a compact design of the Mikota resonators.

The PID controller parameters were selected by usual adjustment processes based on step response observation.

The simulation results are visualized by time plots of several system states in Figs. 7 to 10.

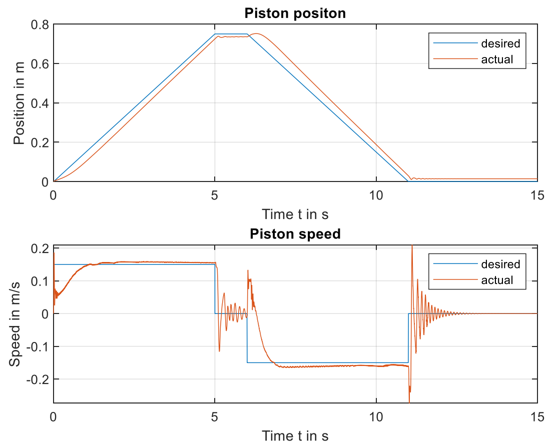

Figure 7. Desired and actual position (upper plot) and speed (lower plot) of a ramped position trajectory.

The actual piston position s(t) follows the desired trajectory $s_d(t)$ with some delay. This is mainly caused by the time needed for the integral part to find the value corresponding to the dead load F_0, and for the ramp down phase by the disturbance due to the speed jump at its beginning.

There is a substantial energy saving in load lifting of the switched system (8.6 kilojoule) compared to a speed control with throttling (15.6 kilojoule).

Energy recuperation of load lowering is not working properly. This stems from two effects: (i) Switching converters which operate by an alternate switching of the valves for the two pressure ports, may lose substantial energy by unfavourable wave propagation in the inertance tube. (ii) The flow rates over the valves are high and create a substantial pressure loss. Wave effect caused losses might be either reduced by a state dependent variation of the switching frequency [Pan 2017] or by employing fast check valves to the actual low and high pressure ports which work like freewheeling diodes in electric switching converters. The first approach requires a detailed analysis of the switched system dynamics and its varying switching frequency impairs the resonator’s attenuation performance. The second approach requires a more sophisticated circuitry since in the combination of multi-pressure and inertance control the p_m port may be the low and the high pressure port. Hence the orientation of the check valve must be changed, depending on the actual state. Quick enough check valves connecting volumes A and B with tank line can avoid cavitation, which is found in the simulation, because the models are not taking cavitation into account; see Figure 9. The role of these connecting volumes for efficiency and cavitation is studied in [Kogler 2015].

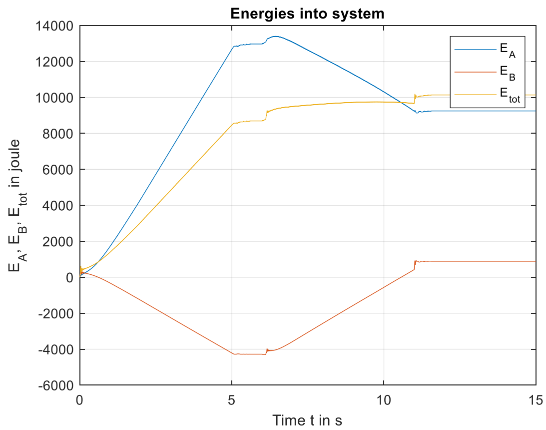

Figure 8. Accumulated energy flow into the system at A-(E_A) and at B-(E_b) side and their sum (E_{tot}).

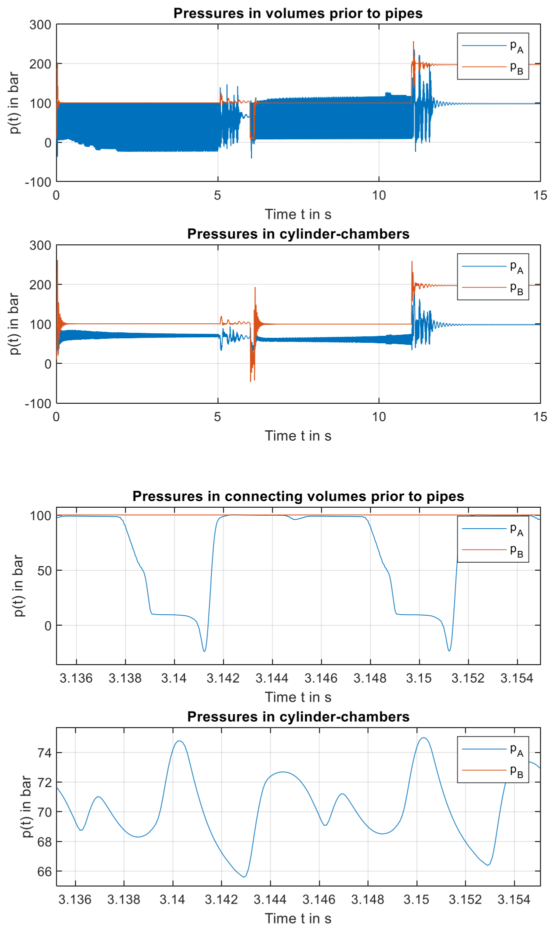

Figure 9. Pressures before inertance pipers and in cylinder chambers of the full motion cycle (upper plot) and a small time section (lower plot).

The effect of switching on motion control can be seen from the motion results for pure multi-pressure control, given in Figs. 11 and 12. The PID controller parameters were changed (k_P= 20e3, k_I= 20e3, k_D= 0) to reduce the number of switching events. It has to be mentioned that the amount of position error depends strongly on load inertia m. Loss in accuracy and higher speed oscillations is compensated by a substantial energy saving, particularly due to a slightly better recuperation. This would be even better, if the number of switching events would be lower. Compression losses when cylinder chambers are pressurized after a phase of low pressure take up nearly the whole energy gained from load lowering.

The controller commands inertance control related switching of the piston (A-) side. The rod side is switched only four times (including motion start) between p_T and p_m to accomplish force stepping. Mainly because of the attenuation effect of the Mikota resonator, inertance type switching causes smaller pressure and speed oscillation than switching for force stepping. The remaining pressure oscillations in chamber A (p_A) are dominated by the higher order harmonics of the switching process, since the base frequency is compensated by the resonator.

Switching for force stepping are isolated events. They can provoke substantial oscillations which are dominated by the mass spring oscillator formed by load inertia m and the compliance of the hydraulic cylinder due to fluid compressibility. Speed signal in Fig. 7 and cylinder pressure signals in Fig. 10 shows this clearly. Suppression of these oscillations cannot be done with resonators.

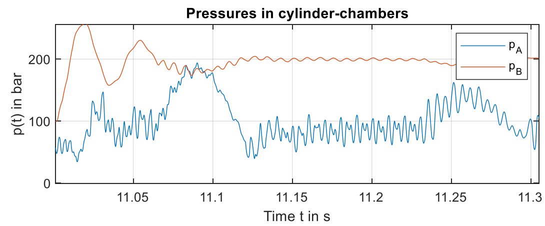

Figure 10. Section of pressure plot of Fig. 9 showing the pressure oscillation due to force stepping.

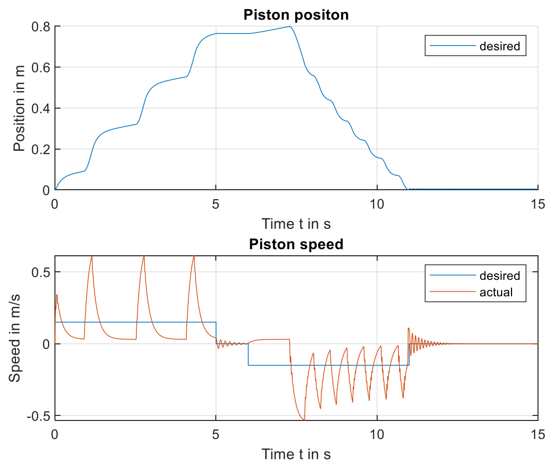

Figure 11. Desired and actual position (upper plot) and speed (lower plot) of a ramped position trajectory if the system is operated without PWM valve switching and with different controller settings.

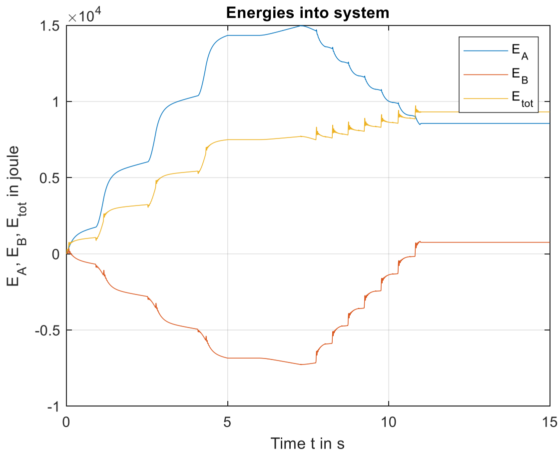

Figure 12. Accumulated energy flow into the system at A-(E_A) and at B-(E_b) side and their sum (E_{tor}) if drive is operated without PWM switching..

4 CONCLUSION AND OUTLOOK

In principle, switched inertance control can close the gap between the discrete force levels of multi-pressure control of a hydraulic linear drive. A smooth operation concerning output motion (speed and position) can be accomplished if the fast switching pulsation is compensated by a resonator. With additional resonators for higher order harmonics also pressure pulsation can be strongly eliminated. Force level switching, however, causes noticeable motion oscillation.

For the studied load case, a substantial efficiency advantage of switched inertance control over resistance control showed up. Energy recuperation, however, did not work convincingly.

Not all questions concerning the feasibility of this type of hydraulic drive are answered in this paper.

1. There is a substantial cost of fast and high flow rate switching valves which can realize permanent switching at frequencies in the range 50 to 100 hertz and, furthermore, such valves are currently not available on the market.
2. Fast and repeated switching may cause disturbing noise, which is not easy to eliminate.
3. Chance of cavitation is a general downside of switching converters if they operate in a step down mode and requires elevated tank pressure and fast switching valves.
4. There is room for improvement of efficiency of switched inertance control, particularly in recuperation mode, by the use of check valves of optimized switching frequencies. Generally, the achievable energy savings depend strongly on the operation scenarios. A general assessment cannot be given by the investigation of a simple load case but must be done for a representative collection of use operation scenarios for a specific application domain.
5. Force level switching caused oscillation can be reduced by modified switching process or by circuit modifications. Modified switching could be (i) sufficiently slow switching, to avoid excitation of the oscillations, or (ii) a few properly timed switching pulses for a mutual annihilation of the generated oscillations [Kogler 2022].

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