HYDRAULIC EFFICIENT AND ROBUST CONVERTER UNIT FOR LINEAR MOTION UNDER ENERGY SAVING

Helmut Kogler $^{\mathcal{L}}$, Rudolf Scheidl $^{\mathcal{J}}$, Martin Lauber $^{\mathcal{L}}$, Bernd Winkler $^{\mathcal{L}}$
$^{\mathcal{L}}$ Linz Center of Mechatronics
$^{\mathcal{J}}$ Johannes Kepler University
Altenberger Strasse 69, A-4040 Linz
E-mail: helmut.kogler@lcm.at
Phone:+43 732 2468 6059

ABSTRACT

Hydraulic cylinders are common and robust actuators for linear motion applications in harsh environments, like for instance on excavators. A hydraulic buck converter represents a digital switching concept transferred from power electronics to hydraulics, which is used for the control of hydraulic actuators at high energy efficiency. Basically, in a hydraulic buck converter the flow is controlled by the switching of digital hydraulic valves, which are operated in pulse width mode at a constant switching frequency. Actually, a single converter suffers from large pressure fluctuations due the digital valve switching. This drawback can be eliminated by a parallel arrangement of several smaller converters operated in a phase shifted mode. In this paper an efficient and robust converter axis is presented, where several hydraulic buck converters are unified with a hydraulic differential cylinder. Since the digital valves are directly integrated in the actuator, the resulting converter unit must be only supplied by one high pressure line and one tank line. Thus, in case of multiple hydraulic actuators, like on the arm of an excavator, one common rail supply is sufficient for all actuators and no interconnection piping is necessary. The energy performance of the presented converter axis is investigated by simulation experiments for selected load conditions, problems and limitations are discussed and an outlook for next steps in development are provided.

KEYWORDS: energy, efficient, digital, switching, hydraulics, smart, actuator

1. INTRODUCTION

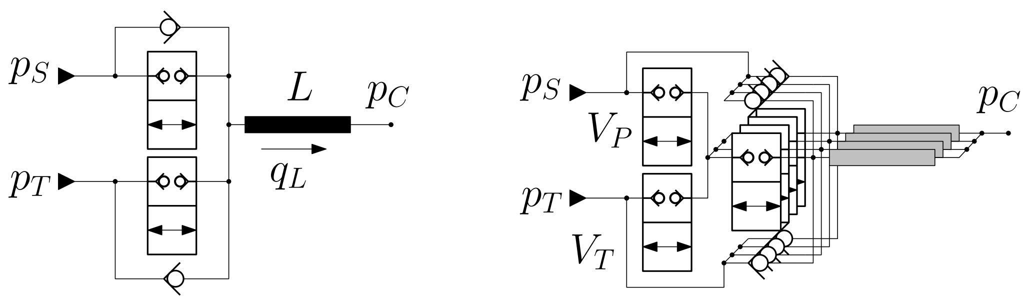
Reduction of losses in hydraulic drives is a major general R&D goal since long [1, 2, 3]. The necessity to limit global warming and the corresponding regulations gave the loss problem an even higher relevance. Loss reduction relates to components, control, and basic operation principles [4]. The suitability of a specific principle depends on several circumstances: the use cases of the drive or drives (if there is more than one on a machine) with their functional requirements; non-functional requirements, like, for instance, size, weight, and admissible noise emission; availability of adequate components to realize a certain concept with competitive cost; manageable engineering and maintenance complexity. Particularly the latter gains increasing importance since modern machine systems involve more and more disciplines which increases complexity. Therefore, machine builders ask for simple drive concepts with robust components and clear interfaces to the other sub-systems [5, 6].

If a machine has several drives and if significant energy recuperation potential exists a hydraulic constant pressure supply in combination with an accumulator and efficient hydraulic drives with recuperative capability are promising [7, 8]. Secondary control for rotary motions is such a concept, proven in practice since decades. Hydraulic transformers have been proposed several times to realize energy efficient linear motion control with variable displacement machines [9]. Multi chamber cylinders are a discrete approximation of the variable area hydraulic cylinder, the direct realization of secondary control for linear motion [10]. The modern versions of switched inertance control (Montgolfier's hydraulic ram was for water pumping and not for motion control) are attempts to transfer the standard control approach of most modern electric machines to hydraulics [11]. Practical realization poses great challenges: switching valve technology, suppression of hydraulic pulsation, mechanical vibration, and noise, avoidance of cavitation, and control which not only should optimize performance in terms of efficiency and motion control accuracy but also assist avoidance of the negative effects of switching. Switching valves which can be electrically controlled with adequate high feasible switching frequency and nominal flow rate are not available from the shelf [12]. A downside of switched inertance control is also the substantial length of the inertance tube, which for the manageable switching frequencies in the range of fifty to one hundred hertz, is in the range of one to two meters. The placement of these components might be a burden for mechanical system design. Another problem is if accumulators are used for pulsation attenuation, since they bring high softness to the system and potentially also lifetime problems of elastomer diaphragms or bladders of corresponding accumulator types.

One way of overcoming the valve availability, tube length, and accumulator problems is using several converter units with a phase shifted switching. More units need lower flow rates per unit and phase shifting means higher effective pulsation frequency. Lower flow rate allows smaller switching valves which are much more likely being available and creates less pressure pulsation in the cylinder chambers; in combination with phase shifting for N converter units the reduction factor is $^1/_{N^2}$. That was first presented and studied theoretically in [13], a small scale prototype and also experimental tests in [14]. A purely hydraulic actuation concept of such a multiple buck converter with an exemplary design of the piloted valve units and a valve block and a simulation study of its performance is published in [15]. This paper is about an integration of four buck converters in a single hydraulic cylinder with all its required components. It covers also all valves for full four quadrant control. An accumulator can be saved. A simulation study for the boom, bucket stick and bucket drives of an excavator in a trench digging cycle shows the motion control and energetic performances in comparison to other established concepts.

1.1. Hydraulic Buck Converter

A hydraulic buck converter (HBC) is a switching concept transferred from power electronics to hydraulics, where the inertance effect of the fluid in a pipe is used to bring the fluid from a lower to a higher energetic level. In Fig. 1a a single stage HBC is illustrated. The consumer pressure p_C is always lower than the supply pressure p_S. In forward flow direction ($q_L > 0$) the



(a) Single HBC (b) Multiple HBC

Figure 1: Types of hydraulic buck converters

supply sided switching valve is operated in PWM mode at a constant switching frequency. The resulting spillover of the kinetic energy in the inertance tube L causes a suction phase through the tank sided check valve during the off-time of the pulsed valve and, thus, fluid is lifted from tank pressure level p_T to consumer pressure level p_C. In the opposite flow direction, the fluid in the inertance tube is accelerated by the tank sided valve, which results in a pressure boost through the high pressure sided check valve to the supply system. This operating principle takes place at a switching frequency between 50 and 100 Hz. The repeated switching results in an inherent high flow pulsation in q_L, which causes high pressure fluctuations at the consumer port if no pressure attenuation device is applied at the output port of a single HBC. The pressure ripples are commonly reduced by additional pressure attenuation devices, like gas-loaded accumulators, which in turn result in a soft drive. In order to overcome this drawback a parallel arrangement of N smaller buck converters, like depicted in Fig. 1b, is operated in a phase shifted mode over one switching period, which lowers pulsations at the consumer by the factor $^1/_{N^2}$ and, thus, an additional pressure attenuation device is unnecessary. The pulse width modulation demands certain dynamic requirements on the switching valve. For this reason the PWM valves are more expensive compared to conventional on/off valves, however, with the directional valves V_P and V_T from Fig. 1b the number of expensive valves can be halved. The digital valves used for PWM switching in an HBC have a response time within a few milliseconds, which is in the same order of magnitude as with servo valves. But the digital valves do not require such a high oil cleanliness due to their simple and robust design.

1.2. Smart Actuator

In Fig. 2 a Hydraulic Efficient and Robust Converter Unit for Linear motion under Energy Saving (HERCULES) is depicted. In this design study of a so called smart actuator the multiple hydraulic buck converters from Fig. 1b are integrated directly into a differential cylinder. All valves are located in the block at the rod side of the cylinder. The inertance tubes of the multiple HBCs are arranged helically around the cylinder in an annulus. Due to the parallel arrangement of smaller converters in combination with their phase shifted operation no additional components for pressure attenuation are necessary and, thus, a highly compact design of the converter unit is possible. All valves used in this design study are commercially available, which means that an industrial realization is basically possible.

The hydraulic scheme of the HERCULES axis is illustrated in Fig. 3a. The consumer ports of the multiple converters are connected to the piston sided chamber and the rod side chamber is connected to either supply or tank pressure by two directional poppet valves (FD) in order to adjust the direction of the resulting force of the drive. The switching strategy of the rod sided

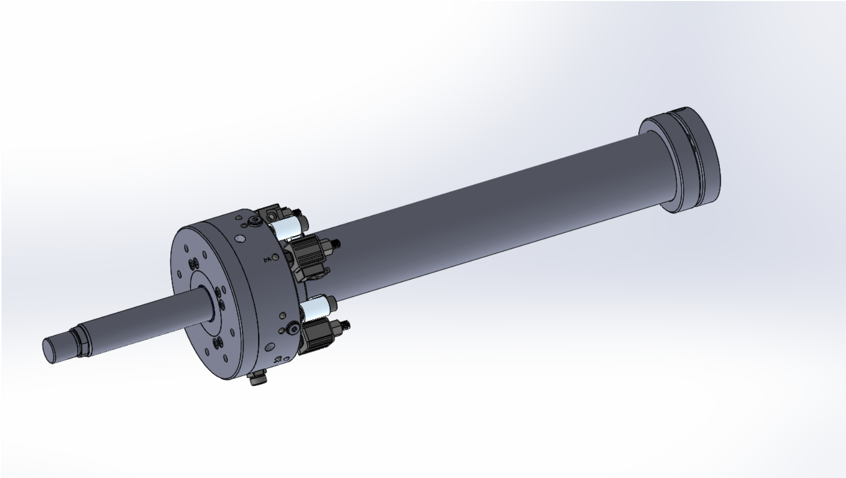
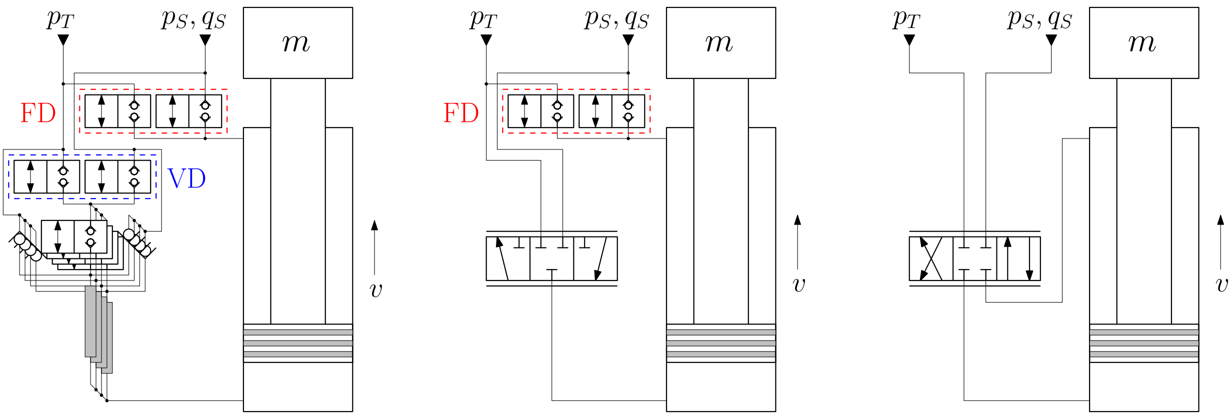


Figure 2: HERCULES axis



(a) mHBC (b) HDP (c) HPD

Figure 3: Investigated configurations

valves FD is designed to operate the multiple hydraulic buck converters (mHBC) at highest possible efficiency. In case of an extending movement the pressure difference between the piston sided chamber and tank must be as low as possible in order to draw maximum oil from the tank through the free-wheeling check valves. In the opposite moving direction the pressure difference between the piston sided chamber and supply pressure is minimized such that the largest possible amount of fluid can be recuperated into the supply system. Like in Fig. 1b also in Fig. 3a the velocity direction is realized by two directional poppet valves (VD). The valve groups FD and VD must be large enough in order to minimize additional throttling losses. Thus, the mHBC axis has full functional range and can operate in all four quadrants of power. Furthermore, since all valves are located directly at the actuator the drive must be only supplied by a high pressure and a low pressure line. In case of multiple actuators only one common rail supply circuit is sufficient for all drives, which may be relevant in mobile applications in order to reduce costs for the interconnection piping.

For evaluation of the power consumption two more drive configurations will be investigated. In Fig. 3b a hydraulic directional plunger (HDP) is illustrated, where the mHBC is replaced by a $3/3$-proportional valve and the force direction is again controlled by a valve group FD. In an extending movement at low process forces this configuration is able to operate in a so called regenerative mode, where the rod side chamber is connected to supply pressure, respectively, the

piston diameter	$d_P = 63\,\mathrm{mm}$
rod diameter	$d_R = 45\,\mathrm{mm}$
maximum stroke	$l_C = 0.5\,\mathrm{m}$
dead load	$m = 40\,\mathrm{kg}$
supply pressure	$p_S = 200\,\mathrm{bar}$
tank pressure	$p_T = 10\,\mathrm{bar}$

(a) Cylinder

number of HBC stages	$N_{\mathrm{HBC}} = 4$
length of inductance	$l_{\mathrm{HBC}} = 1.5\,\mathrm{m}$
diameter of inductance	$d_{\mathrm{HBC}} = 4.5\,\mathrm{mm}$
PWM valves (WS22)	$Q_N = 10\,{}^{\ell}/_{\mathrm{min}}@5\mathrm{bar}$
switching time of WS22	$t_S = 3\,\mathrm{ms}$
PWM frequency	$f_{\mathrm{PWM}} = 50\,\mathrm{Hz}$
check valves (RV)	$Q_N = 25\,{}^{\ell}/_{\mathrm{min}}@5\mathrm{bar}$
FD/VD valves (GS02)	$Q_N = 60\,{}^{\ell}/_{\mathrm{min}}@5\mathrm{bar}$
switching time of FD/VD	$t_S = 50\,\mathrm{ms}$

(b) mHBC

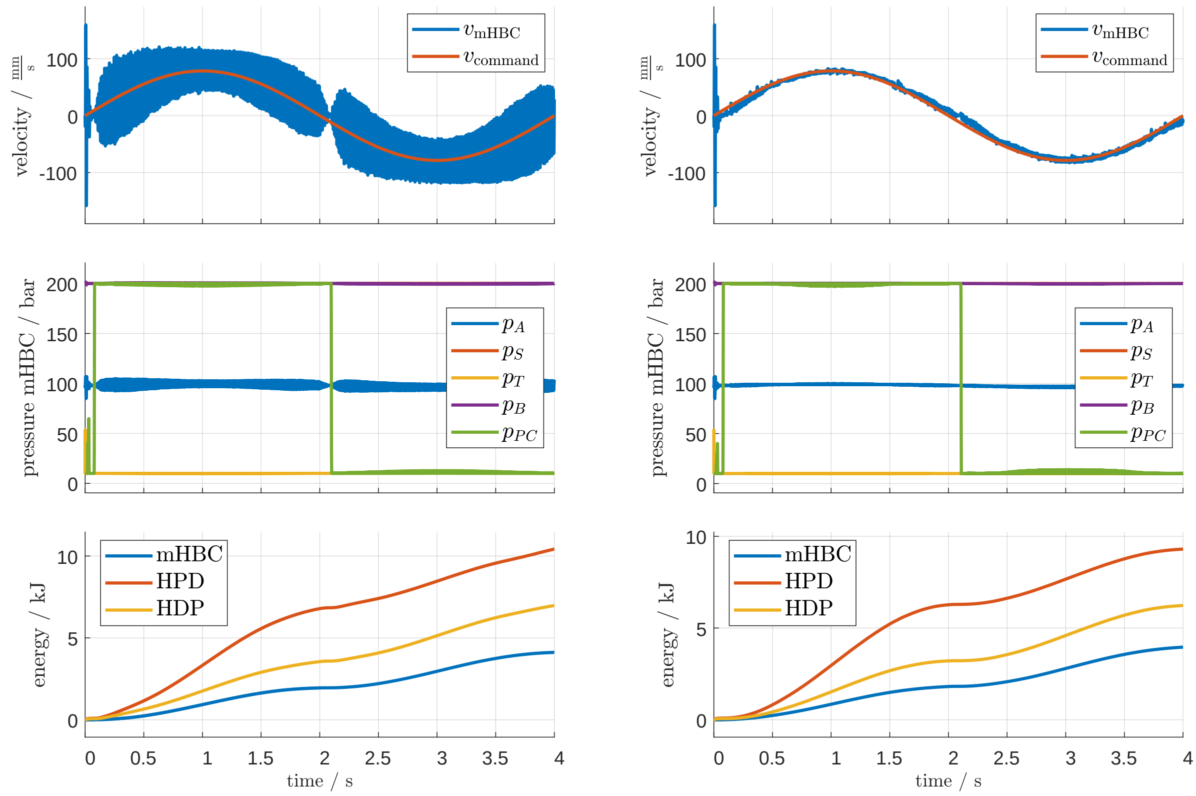
Table 1: Main parameters of the cylinder axis

supply sided valve of the FD valve group is active. In this operating mode the maximum force is limited to $p_S\,(A_P - A_R)$, however, in this case the necessary displacement volume and, thus, the power consumption are reduced significantly. The third configuration used for comparison is a conventional hydraulic proportional drive (HPD) using a $4/3$-proportional valve for motion control, as depicted in Fig. 3c.

The dimensions of the used cylinder depicted in Fig. 2 and load data are listed in Tab. 1a and result in a natural frequency of $100\,\mathrm{Hz}$ in the middle position of the cylinder's full stroke. The main parameters of the mHBCs are listed in Tab. 1b. The high speed digital valves are of type WS22 from *Bucher* with a sufficiently short response time for PWM switching. With these valves a reasonable switching frequency of $50\,\mathrm{Hz}$ can be realized. The free-wheeling check valves of the HBC are also available from the *Bucher* company. The valves for the direction of the velocity, respectively, the force are considered of type GS02 from *Parker* with a reasonable valve size and response characteristics. In the simulations the sizes of the proportional valves are not relevant, as long the valves are large enough and are not going to saturate. Furthermore, in resistance control the energy consumption is determined by the effective displacement of the load. Moreover, since in the considered simulation cases only slow varying trajectories are investigated the response dynamics of the proportional valves can be neglected.

2. BASIC SIMULATIONS

In the following the simulation responses of the mHBC axis for a slow sinusoidal trajectory of the piston with a constant pressure supply are presented. The period of the moving cycle is four seconds and the maximum velocity is approximately $100\,\frac{\mathrm{mm}}{\mathrm{s}}$. In Fig. 4 the simulation results for the unloaded axis with two different numbers of actuated HBCs are depicted. In Fig. 4a only 2 converters with a switching frequency of $50\,\mathrm{Hz}$ are used. The phase shifted operation of both HBCs results in an effective switching frequency of $100\,\mathrm{Hz}$, which is close to the natural frequency of the axis as already mentioned above. For this reason the actual velocity in the upper diagram shows large fluctuations due to the excitation of the natural frequency of the unloaded axis. The fluctuations can be also observed in the pressure signal p_A of the piston sided chamber in the middle diagram. The signal p_{PC} represents the pressure in the pilot cavity between the valve group VD for velocity direction and the high speed valves for PWM switching. Since in Fig. 4 only the results of the unloaded axis are presented, the pressure in



(a) Two HBCs

(b) Four HBCs

Figure 4: Unloaded axis with different number of HBCs

the rod sided chamber p_B remains constant at supply pressure level. In the lower diagram the energy consumption of the three configurations from Fig. 3 are illustrated. As expected the mHBC shows the best energy performance and the HPD the worst. Since the axis is unloaded in this case the HDP operates in regenerative mode, which means that the annulus chamber is connected permanently to supply pressure. Consequently, the fluid from the rod side chamber is directly used in the piston chamber and, thus, less fluid from the pressure supply is necessary for the intended motion.

In Fig. 4b the simulation results for four mHBCs operated in phase shift mode are presented, again in an unloaded situation. In this case the effective switching frequency is $4 \times 50\,\mathrm{Hz} = 200\,\mathrm{Hz}$, which seems to be sufficiently above the natural frequency of the axis, because the velocity signal indicates an acceptable smooth movement. Also the pressure signal p_A in the piston sided chamber shows no relevant fluctuation anymore. In the lower diagram the energy consumption of the HPD and HDP differ slightly from the previous case, because the results are calculated from the actual motion of the mHBC.

In Fig. 5 the results of two different load cases are illustrated. On the left hand side in Fig. 5a a compressive load of $40\,\mathrm{kN}$ is applied to the converter axis with a limited slope starting at a simulation time of 1 second. In the unloaded case the rod side chamber (p_B) is connected to supply pressure. When the pressure p_A exceeds a certain limit due to the load force, then the rod side chamber is switched to tank pressure by the valve group FD according to the strategy described above. This event results in corresponding spikes in the actual piston velocity. Since in the phase where the maximum load force is applied the pressure p_A is higher than in the unloaded case, the efficiency of the converter is reduced and, thus, the slope in the energy consumption of the mHBC is rising. On the other hand when the moving direction changes to the retracting

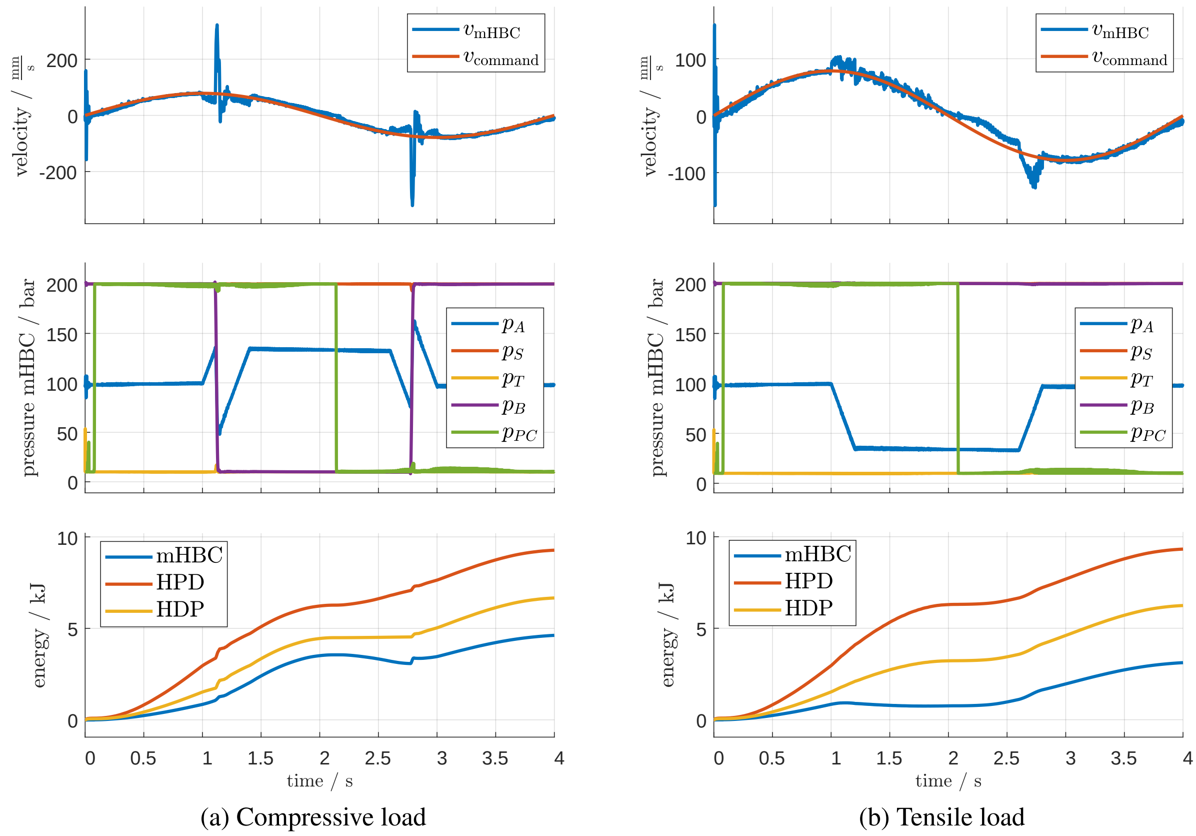


Figure 5: Different loads

direction under the high load force the consumed energy even decreases, because energy from the potential of the load force is recuperated by the mHBC. When the load force is released and the pressure p_A falls below a certain limit, the rod side chamber is connected to supply pressure again. Otherwise the pressure p_A would fall below tank pressure when the compressive load force vanishes. In such a case it would not be possible to stop the movement because fluid would flow through the tank sided free-wheeling valves. The energy consumption of the HPD is nearly the same as in the unloaded case, because in resistance control the consumed energy corresponds to the effective displacement of the piston. Compared to the unloaded case the energy consumption of the HDP is higher because due to the higher load force the regenerative mode is left by switching the rod sided chamber from supply pressure to tank pressure and, thus, the throttling losses increase.

The load case with a tensile force of $-20\,\mathrm{kN}$ is illustrated in Fig. 5b, where the rod side pressure is kept constantly at the supply pressure level over the complete working cycle. When the load force is applied the pressure p_A is further reduced and the efficiency of the converter is even increased in the extending direction of movement. This can be explained by a lower pressure difference between pressure p_A and tank pressure, which in turn reduces the deceleration process of the fluid in the inertance tube, more fluid is drawn through the tank sided check valves. Thus, the throttling losses decrease and, respectively, the efficiency increases.

3. SIMULATION STUDY FOR AN EXCAVATOR

According to the basic simulation results from the previous section, which represent a quite academic investigation, in the following a more realistic case is considered. Recently, the *Volvo*

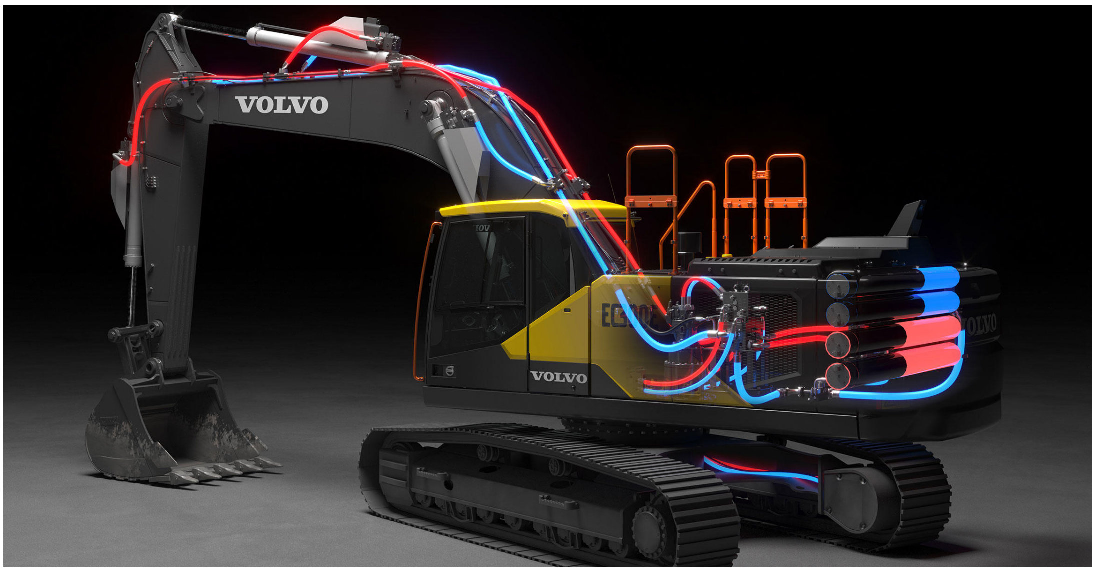


Figure 6: System with multiple chamber cylinders (source: www.volvogroup.com)

	boom (BM)	bucket stick (BS)	bucket (BU)
piston diameter	$d_P^{\mathrm{BM}} = 95\,\mathrm{mm}$	$d_P^{\mathrm{BS}} = 90\,\mathrm{mm}$	$d_P^{\mathrm{BU}} = 75\,\mathrm{mm}$
rod diameter	$d_R^{\mathrm{BM}} = 56\,\mathrm{mm}$	$d_R^{\mathrm{BS}} = 56\,\mathrm{mm}$	$d_R^{\mathrm{BU}} = 45\,\mathrm{mm}$

Table 2: Cylinder dimensions of the excavator

company published an excavator using multi-chamber cylinders as main actuators for the boom, bucket stick and bucket, which is depicted in Fig. 6. As the name promises, each cylinder has multiple chambers with different cross-section areas and, thus, its displacement volume can be adjusted with regard to the load force, which in turn reduces throttling losses. The motion is controlled by so called digital flow control units (see for instance [16, 17]) directly situated at each cylinder. Thus, one actuator has only two ports, one for high pressure and one for the tank pressure. As a consequence, only one supply pressure line and one tank line are necessary for all actuators, which reduces the effort for piping significantly.

Basically, like a multi-chamber cylinder also the HERCULES axis has also two ports for supply pressure and tank pressure. Inspired by the application of multi-chamber cylinders on an excavator, the HERCULES axis is investigated for the actuation of an excavator arm by simulation, as presented in the following.

3.1. Investigated System

The simulations are focused on the load case of a trench digging cycle based on measurements with a real $5\,\mathrm{t}$ excavator. In Fig. 7 the arm of an excavator equipped with three HERCULES axes for the motion control of the boom (BM), the bucket stick (BS) and the bucket (BU) is depicted. According to their function the actuators have different dimensions, which are listed in Tab. 2. The three axes of the excavator are simulated individually with input signals for velocity and load force derived from the measurements. For comparison the simulations are carried out with three HERCULES axes (mHBC) and additionally with a conventional proportional valve control (HPD) for the corresponding actuators, like illustrated in Figures 3a and 3c.

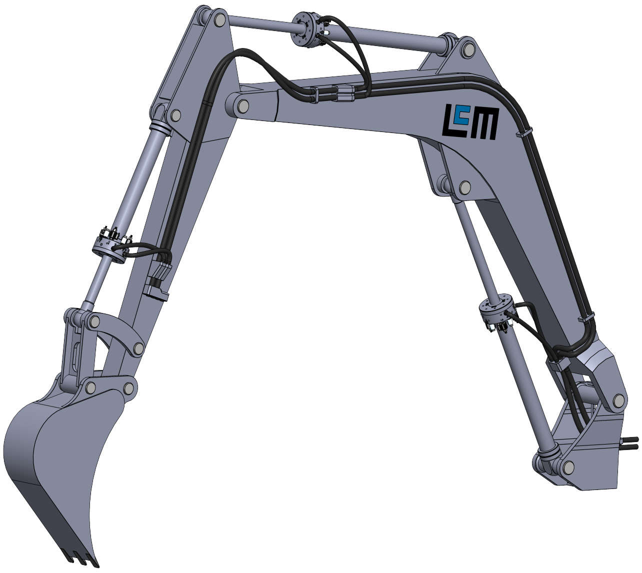


Figure 7: Arm of an excavator equipped with HERCULES axes

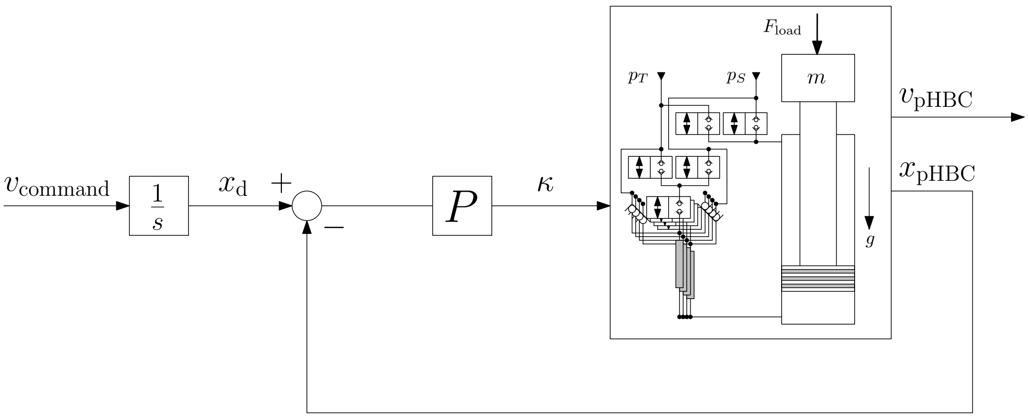


Figure 8: Control scheme of the mHBC axis

3.2. Controller Scheme

In the simulations the velocity and the load force taken from measurements are used as inputs to the model. The velocity is the commanded input of the operator and the load force represents a disturbance for the simulation model. For a reasonable comparison of the energy consumption it is important that the piston displacements of the individual axes are quite similar, which cannot be guaranteed by a simple velocity control. For this reason the control scheme of Fig. 8 is used, where the commanded velocity is integrated w.r.t. time to a position signal, which in turn is controlled by a simple P-controller. This assumption holds as long as the desired trajectory is slow compared to the system dynamics, which is the case for the considered excavator application and the investigated load cases. The resulting velocity signal from the simulation model v_{mHBC} is used for the evaluation of the tracking performance.

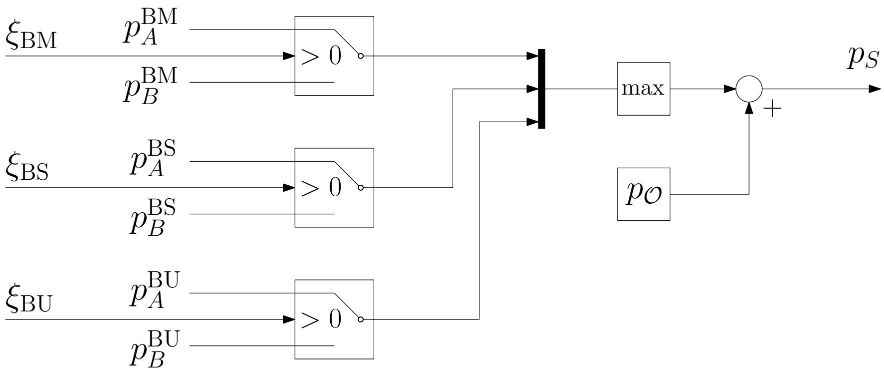


Figure 9: Load sensing for the HPD system

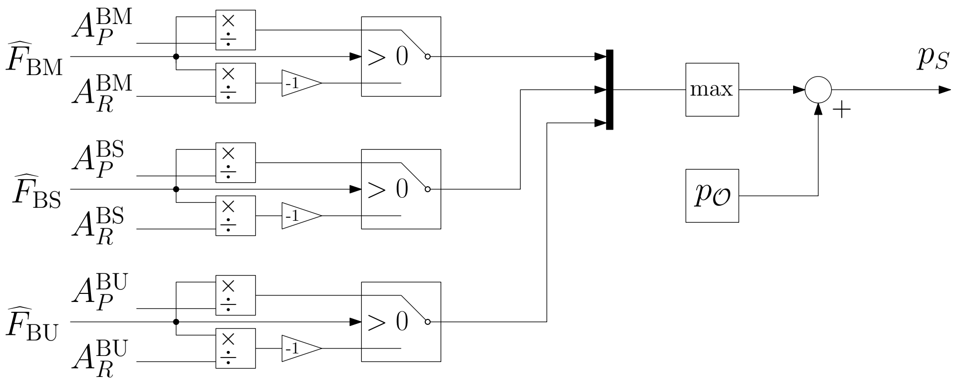
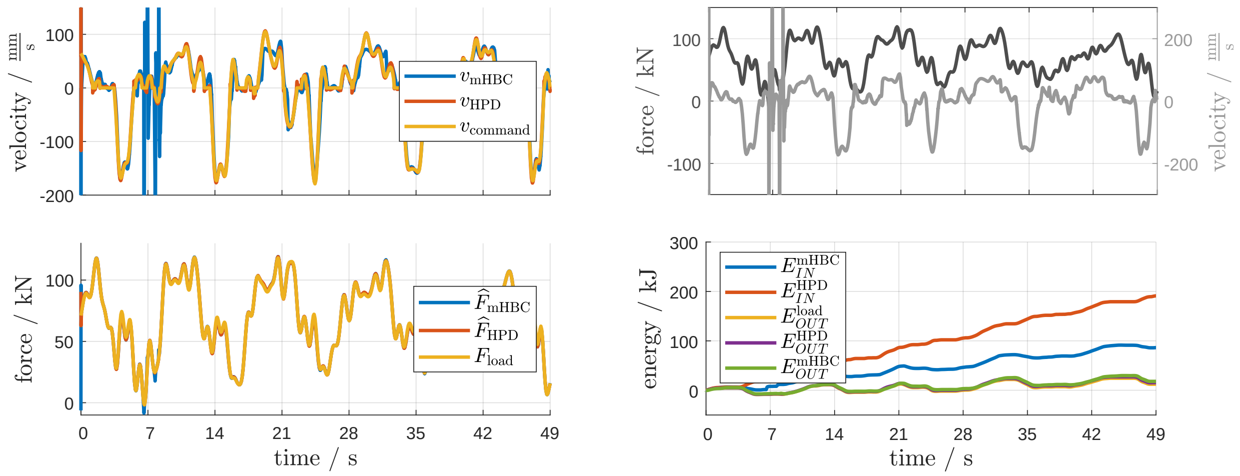


Figure 10: Load sensing for the mHBC axes

3.3. Load Sensing

For efficiency reasons in real excavator applications no constant pressure supply is used, rather often a load sensing (LS) system is applied in order to reduce throttling losses. Thus, in the simulations at least a simplified LS system is considered as well. In the investigations all actuators of the excavator arm are supplied with the same supply pressure. But the LS strategy is different between the mHBC and HPD configurations. In Fig. 9 the functional scheme of the used LS strategy for the proportional valve control is illustrated. All chamber pressures are measured, thus, the maximum pressure of all chambers with regard to the desired direction of motion according to the valve opening $\xi_i,\ i \in \{\text{BM}, \text{BS}, \text{BU}\}$ determines the desired supply pressure. As a pressure reserve for controlling the load a pressure offset $p_O = 20\,\text{bar}$ is added. Of course, in reality the desired supply pressure must be realized by a sort of pressure control, however, in the simulations such a pressure control is assumed to be ideal for simplicity reasons.

In the case of the HERCULES axes a different LS strategy must be considered. Due to the freewheeling check valves of the individual HBC stages the supply pressure must be high enough to maintain the position at any arbitrary load force even when the drive is not moving. For this reason the observed load forces $\widehat{F}_i = p_A^i A_P^i - p_B^i A_R^i,\ i \in \{\text{BM}, \text{BS}, \text{BU}\}$ of the actuators determine the necessary pressure for holding the load, like depicted in Fig. 10. As in the LS strategy for the HPD an additional pressure offset $p_O = 20\,\text{bar}$ is used as a reserve for motion control.



(a) Trajectory tracking

(b) Energy performance

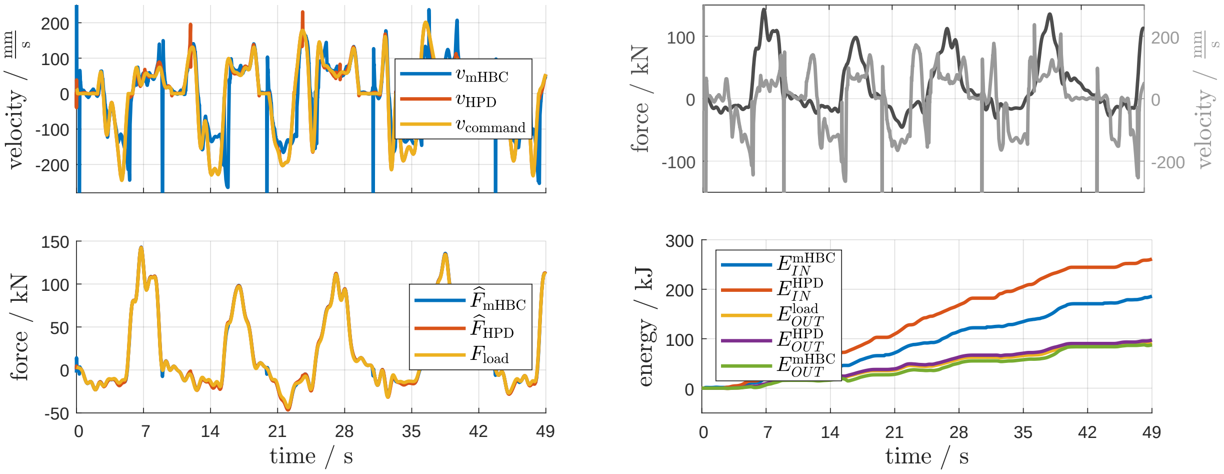
Figure 11: Boom

3.4. Simulation Results

In the following simulation results for five consecutive working cycles of a trench digging process according to the measurements with the already mentioned $5\,\text{t}$ excavator are presented. In Fig. 11 the simulation results for the boom of the excavator are presented. In the upper diagram on the left hand side the input signal of the desired velocity v_{command} is compared to the simulation results according to the two configurations v_{HPD} and v_{mHBC}. Basically, both configuration (mHBC and HPD) achieve a quite acceptable trajectory tracking. At a simulation time of approximately $7\,\text{s}$ two spikes in the velocity of the mHBC axis occur due to the switching of the force directional valves FD as similarly can be observed in the results of the basic simulations in Fig. 5a. In the lower diagram of Fig. 11a the simulated hydraulic forces in the boom according to the load force F_{load} are illustrated. Since a good accordance between the simulation and the load input is achieved the assumption of a slow trajectory compared to the system dynamics is confirmed.

In the upper diagram of Fig. 11b the simulation results of the force $\widehat{F}_{\text{mHBC}}$ and the velocity v_{mHBC} are opposed, where the actual power quadrants can be easily examined. When force and velocity have the same sign, the configuration acts as a motor, i.e. power is necessary for driving. In a generatoric power quadrant the sign of force and velocity are different, thus, the drive does not need any power from the supply and probably energy can be even recuperated. Particularly, this relation can be observed at the actuator for the boom very clearly, as depicted in the lower diagram of Fig. 11b. The signal $E_{IN}^{\text{mHBC}} = \int_0^t p_S^{\text{mHBC}} q_S^{\text{mHBC}} \text{d}t$ shows the energy consumption of the actuator used for the boom. In sections where force and velocity show a different sign, the gradient of the energy signal is negative, which is an indication for energy recuperation. Thus, over the whole simulation time the mHBC axis needs less than half of the energy $E_{IN}^{\text{HPD}} = \int_0^t p_S^{\text{HPD}} q_S^{\text{HPD}} \text{d}t$ that is consumed by the HPD configuration for the same trajectory output. The comparison of the energy consumption is supported by the output energies $E_{OUT}^{\text{load}} = \int_0^t F_{\text{load}} v_{\text{command}} \text{d}t$, $E_{OUT}^{\text{mHBC}} = \int_0^t \widehat{F}_{\text{mHBC}} v_{\text{mHBC}} \text{d}t$ and $E_{\text{OUT}}^{HPD} = \int_0^t \widehat{F}_{\text{HPD}} v_{\text{HPD}} \text{d}t$, which correspond quite well.

In Fig. 12 the simulation results for the bucket stick are presented in the same manner as for the boom. Particularly, in Fig. 12a some more spikes due to the switching of the force directional valves FD can be observed in the signal v_{mHBC}. This is an indication that the sign of the force is changing more often than with the boom actuator, which can be seen in the lower



(a) Trajectory tracking

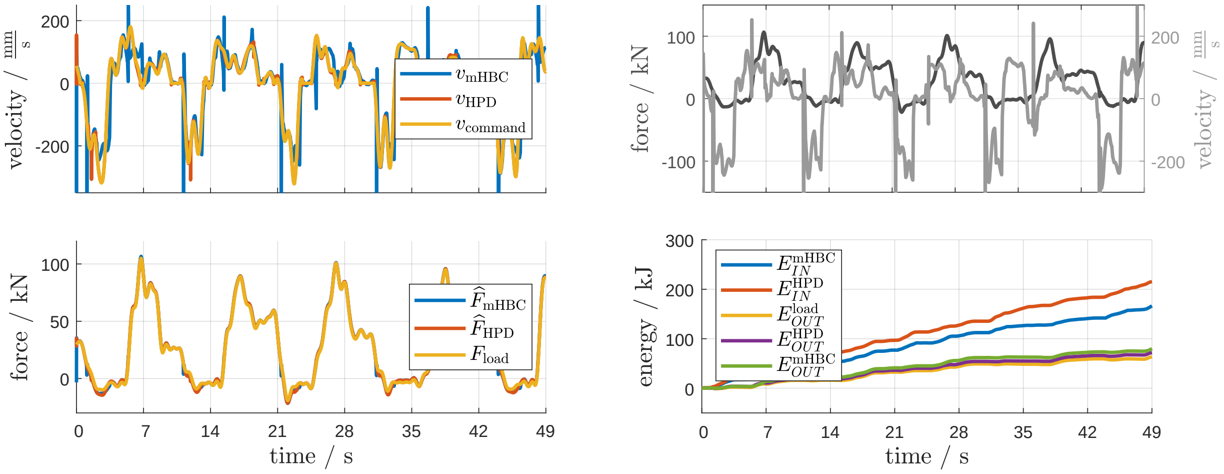
(b) Energy performance

Figure 12: Bucket stick

diagram. However, the velocity tracking over the complete simulation time is acceptable, although in some sections the actual velocity v_{mHBC} differs from the desired velocity v_{command} like, for instance, in the area around the simulation time $t \approx 14\,\mathrm{s}$. In this region the controller went in saturation, which in turn means that the nominal size of the mHBC is too small for a more precise trajectory tracking. In the particular case this could be avoided, for instance, by spending an additional HBC stage, i.e. using five instead of four parallel stages of hydraulic buck converters for the bucket stick. In order to quantify the saturation problem the output energy E_{OUT}^{mHBC} in the lower diagram of Fig. 12b is investigated. It shows a slightly reduced output energy than the load and the HPD configuration, at least in the range below $t \approx 40\,\mathrm{s}$. However, the deviation of the energy output over the whole investigated time span is only marginal and, thus, the velocity tracking is still acceptable. Moreover, the energy consumption of the mHBC axis shows an improvement of approximately 28% compared to the HPD configuration.

The simulation results for the bucket are presented in Fig. 13, which are quite similar compared to the previously discussed results of the bucket stick. In fact, no saturation effects occur with the actuator for the bucket, but slightly more spikes in the velocity signals are present due to the switching of the force directional valves FD at the rod sided port of the axis. The higher number of velocity spikes result in a slightly higher output energy depicted in the lower diagram of Fig. 13b, which can be explained by higher compression losses of the fluid in the chambers according to the higher number of switching cycles in the rod side chamber. Nonetheless, the energy consumption according to the signal E_{IN}^{mHBC} is approximately 22% less than the energy E_{IN}^{HPD} consumed by the HPD configuration.

Finally, the total energy consumption, respectively, the sum of the consumed energy of all three actuators is depicted in the lower diagram of Fig. 14. The excavator arm equipped with the parallel mHBC axes needs approximately 33% less energy compared to the HPD, both with load sensing. Furthermore, in the upper diagram the supply pressure signals according to the different load sensing strategies from Figures 9 and 10 are illustrated. The dashed lines indicate the averaged values of the actual ones resulting from the different LS strategies according to the investigated load cycles. It is remarkable that the supply pressure of the HPD system is approximately $15\,\mathrm{bar}$ higher than with the mHBC axes. On the one hand this is clear, since in the mHBC configuration the throttling losses of one chamber are completely eliminated due to the large force directing valves FD (see Fig. 3a). But on the other hand with the LS strategy



(a) Trajectory tracking

(b) Energy performance

Figure 13: Bucket

for the mHBCs a higher supply pressure would be expected for keeping the load in its position due to the free-wheeling check valves, even when the drive is not moving. One explanation could be that the additional throttling losses in the expelling chamber of the conventional system with $4/3$-proportional valves result in a higher supply pressure required for tracking than the mHBC system. However, the average supply pressure of the mHBC configuration is only approximately 10% lower than from the HPD system but the total mHBC energy consumption is approximately even one third less than with the HPD. This means that a major part of the improved energy performance results from the inertance effect due to the PWM switching of the mHBC axes, at least in the considered load scenarios.

4. DISCUSSION

In all investigated cases the energy performance of the HERCULES axis is significantly better compared to conventional proportional drives. Furthermore, due to the parallel arrangement of multiple digital valves the axis is robust against oil contamination. Moreover, in a case of a non opening faulty valve the operation of the axis can be maintained at least under a reduced performance. Another advantage of the phase shifted approach of multiple mHBCs are the minor remaining pressure fluctuations due to PWM switching of the valves and, thus, additional components for pressure attenuation are unnecessary. The HERCULES axis represents a so called smart actuator, which means that the control valves are situated directly at the cylinder. In the design study also the tight integration of the inertance tubes of the HBC stages in the drive by placing them in a separate jacket on the cylinder could be demonstrated. Thus, the axis only must be connected to a high pressure and a low pressure line and, thus, no interconnection piping is necessary. This saves piping costs. The dynamic performance required for PWM operation makes the digital valves more expensive than conventional on/off valves. Basically, the required on/off times of the PWM valves are in the range of a few milliseconds. Thus, the presented digital approach is basically qualified for high dynamic applications for which conventionally sensitive servo valves are used today. Another benefit compared to proportional, respectively, servo valves is the use of digital poppet valves without any leakage in the closed position. On the one hand this is advantageous for load holding and on the other hand an accuracy as high as with servo valves can be achieved. The very precise actuation is possible due to an operation in

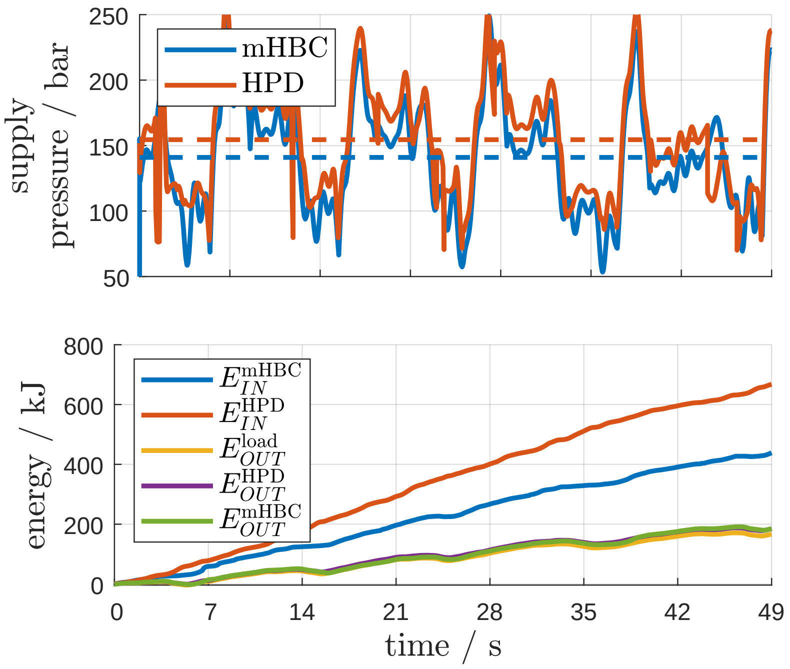


Figure 14: Total energy performance

the so called ballistic mode [18], i.e. at very low PWM duty ratios below the response dynamics of the digital valves and where the poppets do not fully open anymore. Furthermore, the digital poppet valves do not suffer from any valve overlap compared to conventional proportional or even servo valves.

Beside the mentioned advantages the fluctuations, respectively, spikes in the velocity due to the switching of the force directional valves FD at the rod side chamber represent a limitation for certain applications, in particular, in high precision drive control. This drawback can be eliminated by an additional mHBC connected to the rod sided chamber instead of the FD on/off valves, like depicted in Fig. 15. With this configuration a smooth operation in all operating conditions can be achieved. Furthermore, different control strategies can be easily implemented, like for instance a power control, which only can be realized with independent metering.

5. CONCLUSION AND OUTLOOK

In this paper a simulation study of a linear cylinder axis incorporating multiple parallel hydraulic buck converters (mHBC) operated in a phase shifted PWM mode was presented. Since only digital valves are used the axis is insensitive against oil contamination. Furthermore, the robustness is supported by a parallel arrangement of multiple valves, which maintains the operation in case of a faulty valve at least at reduced performance. The simulations were focused on the application of an excavator arm with individual actuators for boom, bucket stick and bucket. Since the smart actuator axis has only one high pressure and one low pressure port, multiple actuators can be easily supplied by one single common rail circuit, which offers a cost reduction for piping on such an exemplary excavator application. The tracking performance and the energy consumption of the mHBC was compared to a conventional drive system using 4/3-proportional valves (HPD). For a more realistic consideration also simplified load sensing concepts were taken into account. The simulations showed an acceptable tracking performance under a significant reduction of the energy consumption in the order of magnitude of $^{1}/_{3}$ of the mHBC axes compared to the HPD drives. The investigated mHBC is connected only to the

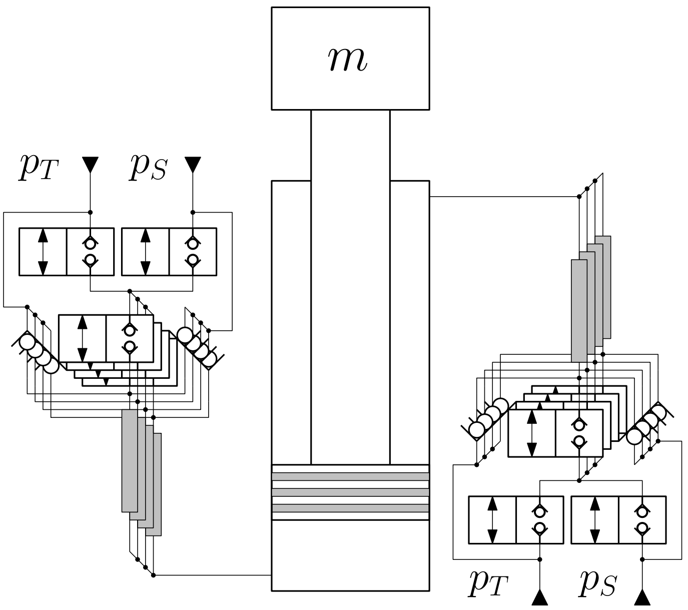


Figure 15: Advanced HERCULES axis

piston sided chamber of the cylinder, while the rod side chamber can be alternatively switched to supply pressure or tank pressure, which results in some unwanted fluctuations in the velocity. This drawback can be eliminated by a cylinder axis with two individual mHBCs connected to both cylinder chambers, which is a next step in development as well as an optimization of the mechanical design.

ACKNOWLEDGEMENT

This work has been supported by the Austrian COMET-K2 programme of the Linz Center of Mechatronics (LCM), and was funded by the Austrian federal government and the federal state of Upper Austria.

The authors want to thank Richard Emeder for the design study of the HERCULES axis presented in Fig. 2 during his internship at the LCM.

REFERENCES

[1] W. Backé and H. Ulrich. Design of electrohydraulic speed control for variable displacement motors on different pressure networks. In *SAE Technical Paper Series*, pages 394–403. SAE International, 1990.

[2] W. Backé. The present and future of fluid power. 207(4):193–212, 1993.

[3] R.C. Burrows. Fluid power-progress in a key technology. 37(4):691–701, 1994.

[4] Anil C Mahato and Sanjoy K Ghoshal. Energy-saving strategies on power hydraulic system: An overview. 235(2):147–169, 2020.

[5] Heino Försterling, Edgar Stamm, and P. Roth. Tailored solutions limit complexity. In *Proceedings of the* 4^{th} *Workshop on Digital Fluid Power*, pages 51–58, 2011.

[6] Reinhard Brandstetter, Till Deubel, Rudolf Scheidl, Bernd Winkler, and Klaus Zeman. Digital hydraulics and “Industrie 4.0”. 231(2):82–93, 2016.

[7] Georges E. M. Vael, Peter A. J. Achten, and Zhao Fu. The innas hydraulic transformer the key to the hydrostatic common pressure rail. In *SAE Technical Paper Series*. SAE International, 2000.

[8] Karl-Erik Rydberg. Hydraulic accumulators as key components in energy efficient mobile systems. In *Proceedings of the* 6^{th} *International Conference on Fluid Power and Motion Control 2005*, 2005.

[9] R. Kordak. *Hydrostatic Drives with Control of the Secondary Unit*, volume 6. 1996.

[10] Matti Linjama, H-P. Vihtanen, A. Sipola, and M. Vilenius. Secondary controlled multi-chamber hydraulic cylinder. In *Proceedings of the* 11^{th} *Scandinavian International Conference on Fluid Power SICFP'09, Linköping*, 2009.

[11] Chenggang Yuan, Min Pan, and Andrew Plummer. A Review of Switched Inertance Hydraulic Converter Technology. 142(5), 2020.

[12] Bernd Winkler, Andreas Plöckinger, and Rudolf Scheidl. State of the art in digital valve technology. In *Proceedings of the* 7^{th} *Workshop on Digital Fluid Power*, 2015.

[13] Helmut Kogler and Rudolf Scheidl. The Hydraulic Buck Converter exploiting the Load Capacitance. In 8^{th} *IFK - "Fluid Power Drives",* $26^{th} - 28^{th}$ *March, Dresden, Germany*, volume 2(3), pages 297–309, 2012.

[14] Harald Kloihofer. Entwicklung, Inbetriebnahme und experimentelle Untersuchung eines hydraulischen Mehrfach-Buck-Konverters. Master's thesis, 2015.

[15] Rudolf Scheidl, Helmut Kogler, and Philipp Zagar. A hydraulically controlled multiple buck converter system. In *ASME/Bath Symposium on Fluid Power and Motion Control FPMC2022*, 2022.

[16] Matti Linjama, Mikko Huova, Pontus Boström, Arto Laamanen, Lauri Siivonen, Loinel Morel, Marina Walden, and Matti Vilenius. Design and implementation of energy saving digital hydraulic system. In *Proceedings of the Tenth Scandinavian International Conference on Fluid Power, SICFP'07*.

[17] Alessandro Dell'Amico, Marcus Carlsson, Erik Norlin, and Magnus Sethson. Investigation of a digital hydraulic actuation system on an excavator arm. In *Linköping Electronic Conference Proceedings*. Linköping University Electronic Press, 2013.

[18] Ingo Schepers, David Schmitz, Daniel Weiler, Olaf Cochoy, and Uwe Neumann. A Novel Model for Optimized Development and Application of Switching Valves in Closed Loop Control. *International Journal of Fluid Power*, 12(3):31–40, jan 2011.