Efficiency Improvement of a Constant Pressure System using an Intermediate Pressure Line

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Abstract

The paper introduces a new hydraulic system based on a constant pressure system with the aim to increase the efficiency of actuation of hydraulic cylinders in mobile machines. Using a third pressure level located between high pressure and tank pressure called intermediate pressure the system enables additional pressure potentials from high pressure to intermediate pressure and from intermediate pressure to tank pressure. This reduces throttle losses at hydraulic cylinders when driven at low or medium loads. An accumulator connected to the intermediate pressure line is being charged or discharged in function of which pressure potential is currently used. Using the example of a typical duty cycle of a wheel loader the paper describes how the accumulator can be applied in order to reach best efficiency results for the new system and it reveals a theoretical efficiency improvement of 20% compared to a conventional Load Sensing system. Furthermore it proposes an online control strategy using Model Predictive Control with an optimization algorithm created with the Dynamic Programming method.

KEYWORDS: hybrid, intermediate pressure, control strategy, accumulator

1. Introduction

For the actuation of hydraulic cylinders in mobile applications three mainly used systems can be identified: constant flow systems, constant pressure systems and Load Sensing systems (LS systems). Constant flow systems generate high power losses when cylinders with low flow demand are actuated. The flow which is not used is conducted directly to the tank and the pressure created by the pump is lost.

LS systems are considered today as one of the most efficient drive systems for linear actuators as the pump generates only the flow which is really needed. The pressure is adapted to the highest load pressure and throttle losses are generally kept low. Disadvantageous of LS systems are high losses at parallel drive of two actuators at different load pressures. As the system pressure is always kept at a constant pressure difference above the maximum load pressure, this pressure difference to the actuator with the lower load pressure must be throttled. Further inconvenient of this system is its incapacity to recover potential or braking energy.

Constant pressure (CP) systems are one of the simplest hydraulic systems and not very efficient as high pressure losses occur when lower loads are lifted. Advanced systems are equipped with a pressure controlled pump and a hydraulic accumulator connected to the main pressure line. In these “Advanced Constant Pressure Systems” /1/ secondary controlled hydraulic rotary drives can be operated without system-related losses /2/. These systems already exist at the market /3/, even though not in mobile applications yet. One inconvenient of a large scale application of this system is its incompatibility with linear actuators as the high pressure still needed to be throttled to adapt it to the actual load pressure. Concepts of secondary controlled linear actuators are presently in the focus of research /4/, /5/ but there is no market ready solution yet.

Another approach for the efficient integration of linear actuators in a CP system may be the introduction of a supplementary pressure line with a pressure level located between high pressure (HP) and tank pressure (TP) and therefore called “intermediate pressure (IP) line” /6/ (**Figure 1**). A hydraulic accumulator is connected to the IP line allowing to recover potential energy. A second accumulator is connected to the HP line to buffer pressure oscillations. Two switching valves for each actuator can change the pressure on the inlet and the outlet port of the proportional valve between high, intermediate- and tank pressure. Losses can be reduced by adapting the applied pressure difference at the piston of the hydraulic cylinder to the actual load.

Due to energetic reasons the IP line can be connected to the suction side of the pump which results in lower energy demand of the system when using the pump.

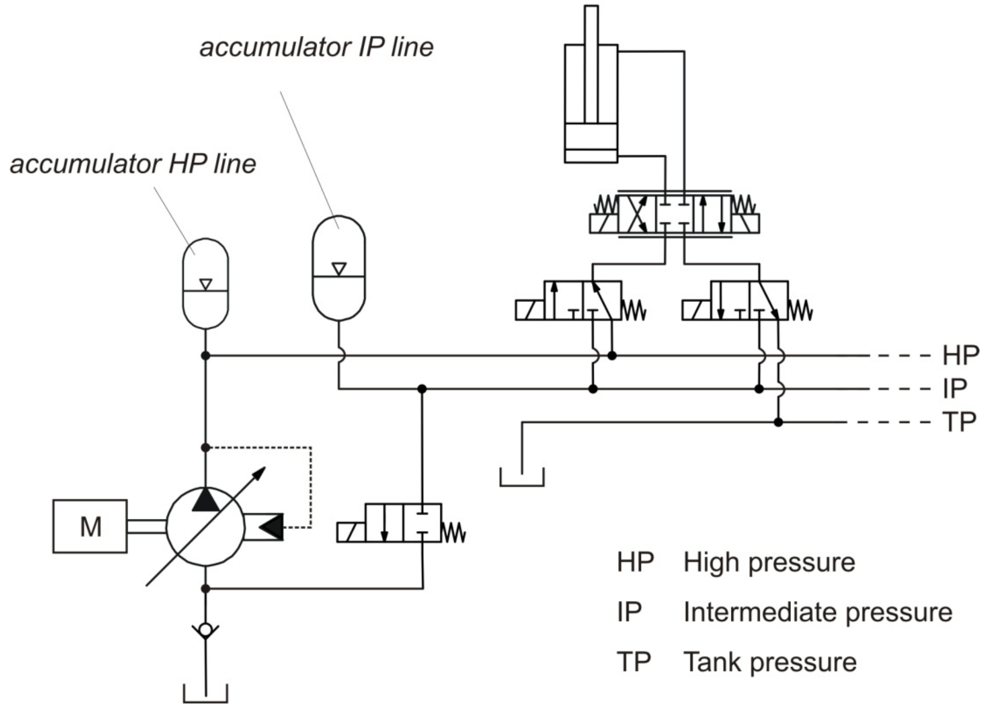

Figure 1: Constant pressure system with intermediate pressure line

Generally, the system allows four different switching states (see **Figure 2**). For very high loads, the switching valves connect the inlet port of the proportional valve to HP and the outlet port to TP and the system acts like a conventional constant pressure system. For lower loads the inlet port of the proportional valve is connected to the IP line and the accumulator is being discharged. The next lower switching state connects the inlet port of the proportional valve to the HP line and the outlet port to the IP line, the accumulator is being charged. For very low loads both ports of the proportional valve can be connected to the IP line.

Besides the hydraulic components the system needs a control algorithm to be operated. This control algorithm is programmed into a programmable logic controller (PLC) and determines the actual load at the cylinders by measuring the pressures in the chambers and chooses the appropriate switching state by actuating the switching valves. Once the switching state is set, the fine control of the piston is enabled by controlling the proportional valve according to the needed flow given by the operator. Furthermore, the algorithm has another element included, the switching strategy which performs actuation of the switching valves not only in terms of having enough force to move the piston but also in terms of global efficiency of the system. It allows therefore switching states which may cause high throttle losses but which allow to charge or discharge the accumulator in order to reach a more advantageous state of charge for future energy demand. Charging strategies are subject of intensive research in the field of hybrid drivetrains /7/, as it influences essentially the overall efficiency of the system and represent also in this system the most important part of the control architecture.

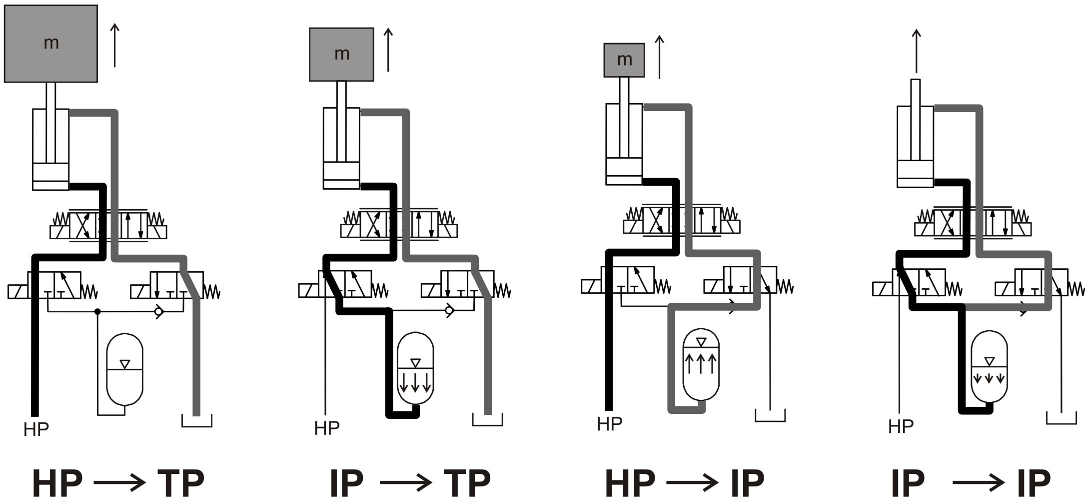

Figure 2: Different possible switching states of the system

2. Energy Analysis of Duty Cycles

As reference system a wheel loader was used which was equipped with a LS system. To the shaft of the diesel engine two (isolated) hydraulic circuits are mounted: a closed circuit for driving and an open circuit LS system supplying lifting cylinder, tilt cylinder and steering cylinder. Further supplementary working functions can be connected to this circuit. The system was analyzed for lifting and tilt cylinder, the steering cylinder was not considered in this project.

To identify the energy saving potential of a constant pressure system with an intermediate pressure line typical duty cycles of the wheel loader were measured and an energy analysis was executed. It allows to visualize energy flows in the system and to determine the efficiency improvement compared to a LS system. In the following paragraphs the described energy analysis shows only theoretical energy flows, based on measured duty cycles. For a final evaluation of the system the diesel engine and the losses in the pump and valves need to be considered as well as dynamic effects but are neglected in this stage.

For the test machine two major duty cycles were identified: loading of pallets and digging/loading. In **Figure 3** the so called “load and carry” cycle is shown which can be composed in different sections as described in /8/. In this cycle the loader loads the bulk material into the bucket and carries it to a transport vehicle where it is dumped. The duty cycle is composed of five different stages:

1. The wheel loader drives from the starting position (A) to the material (B). The bucket is lowered and aligned to the ground.
2. The machine drives into the material and loads the bucket. In the following the wheel loader transports the material to transport vehicle at (C).
3. The machine drives to the transport vehicle and lifts the bucket when arriving.
4. As the bucket is above the bed of the transport vehicle the tilt cylinder is actuated and the bucket is dumped.
5. The loader backs off and the operator brings the bucket back into the starting position.

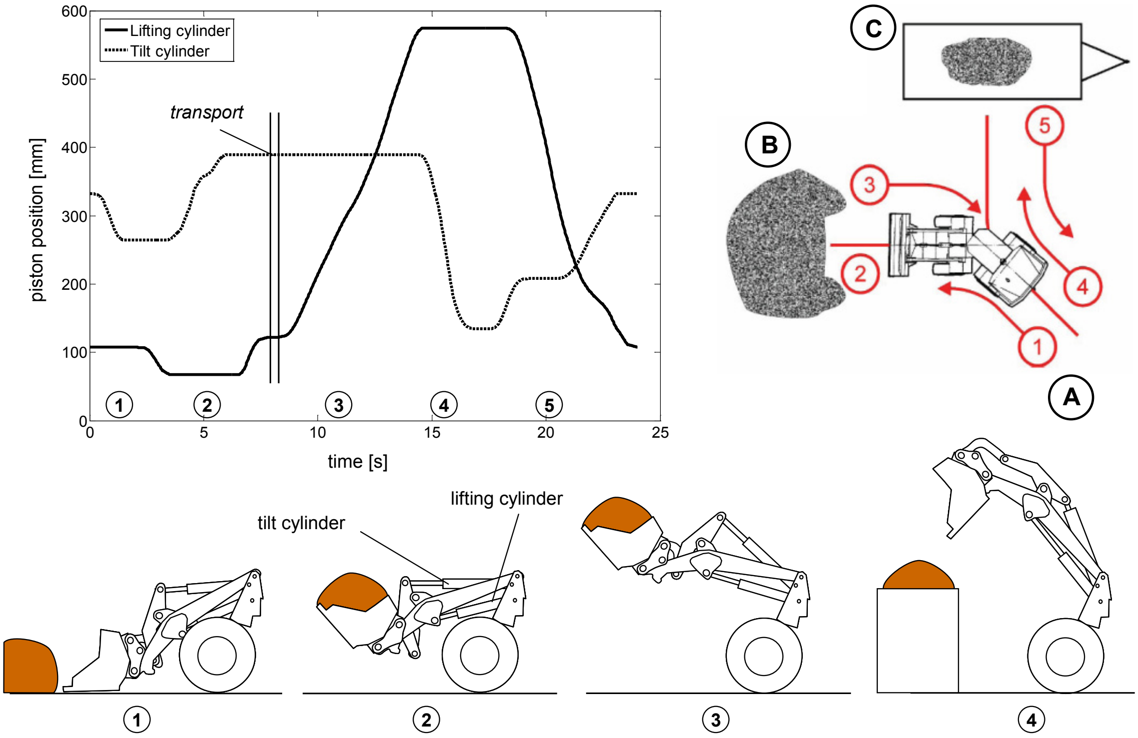

Figure 3: “Load and Carry” duty cycle according to /8/

The described movement was repeated during tests about 90 times at realistic digging conditions. For an energetic analysis one representative duty cycle was chosen and the load pressures were transformed into forces acting on the pistons. This allows identifying the needed forces and to calculate the needed energies in the different stages.

In a LS System the pump power is composed by two parts: one part is conducted directly to the hydraulic cylinders, the other part are power losses due to the constant pressure difference of pump pressure to load pressure (typically 20 bar).

Figure 4 shows graphically the energy flows of a LS system at a simple example (constant velocity of the piston at constant load). Above the x-axis the input energy is located which is equal to the pump energy as the pump is the only power source in LS systems. Below the x-axis the output energies are shown which are composed in used energy (needed to lift the load) and throttle losses. Both energy types, input energies and output energies, are always in equilibrium.

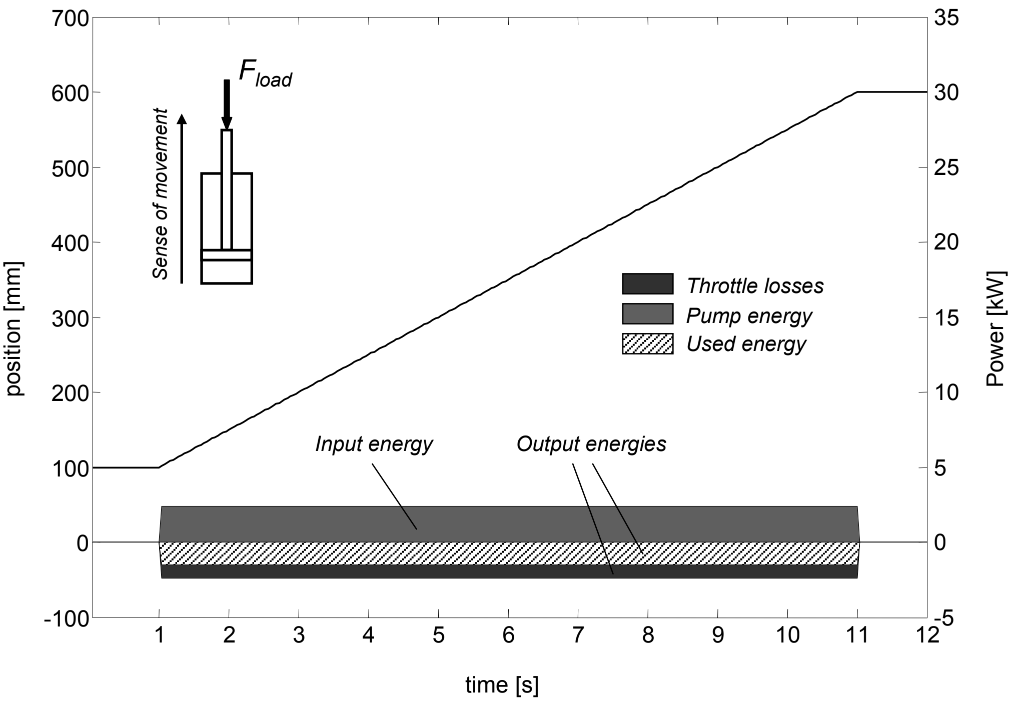

Figure 4: Energy flow of a LS system

For a constant pressure system with an intermediate pressure line there are two, or more precisely, three power sources which can deliver energy to the system separately or in combination. The pump remains the principal power source and produces high energies as the system pressure is very high. The second power source is the accumulator which has two directions of flow. When being charged, the IP line is connected to the outlet port of the proportional valve so the accumulator can be considered as a supplementary load to the actual load on the piston. When being discharged, it can move the piston independently or assist the pump when connected to the pump's suction side. The third very important power source is the potential energy. When the bucket is lowered the oil in the chamber of the cylinder can be used to charge the accumulator.

Figure 5 shows the same movement as presented in Figure 4, this time effectuated with a constant pressure system with an intermediate pressure line. The needed energy to move the piston is given by a sequence of switching states where each results in different energy flows and throttle losses. Additionally to the energy types “pump energy” and “losses” the accumulator energy is added which can be positive or negative. In this example the accumulator is charged by choosing first HP/IP until it has enough energy to be discharged in the stages IP/IP, HP/IP (2), IP/TP and HP/TP (2). The shown throttle losses appear to be very high compared to those of the LS system. This is due to the random switching sequence without the application of optimization algorithms.

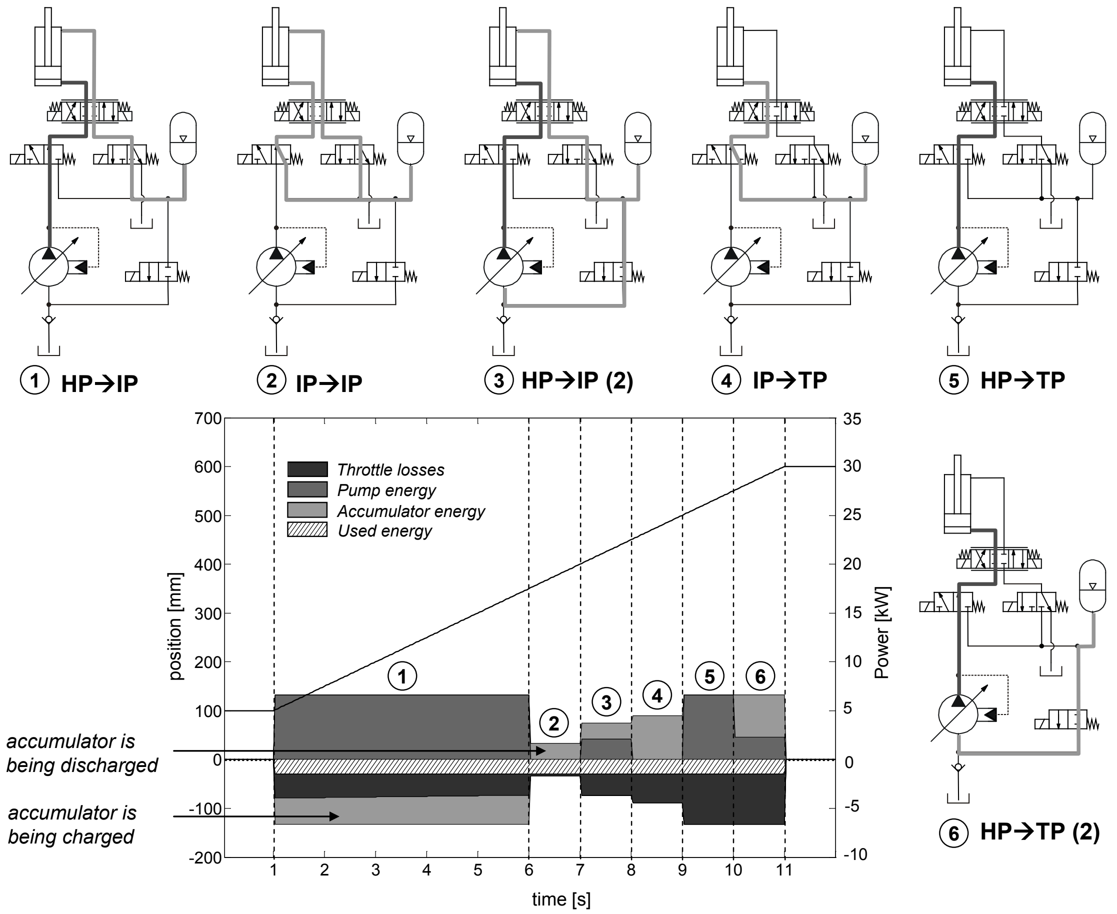

Figure 5: Energy flow in a constant pressure system with IP line

A great variation of different switching sequences is possible to effectuate the same movement. For the very simple example with only one piston the optimum sequence is quite easy to find. For a more complex movement with two actuators as the considered duty cycle the solution is much harder to find as the state of charge of the accumulator plays an important role. A higher pump energy “invest” may be more favorable in some situations to keep the accumulator charged. Both energy types can be considered as equivalent and the optimization problem can be solved using multi-objective optimization algorithms. The result of the multi-objective (Pareto-) optimization can be seen in **Figure 6** (in order to limit the calculation time the cycle was discretized in 0.5 s steps). It can be seen, that additionally to the pump and accumulator energy potential energy can be used when the bucket is lowered or dumped. This energy recovery distinguishes hybrid hydraulic systems and conventional hydraulic drivetrains and enables generally a higher efficiency.

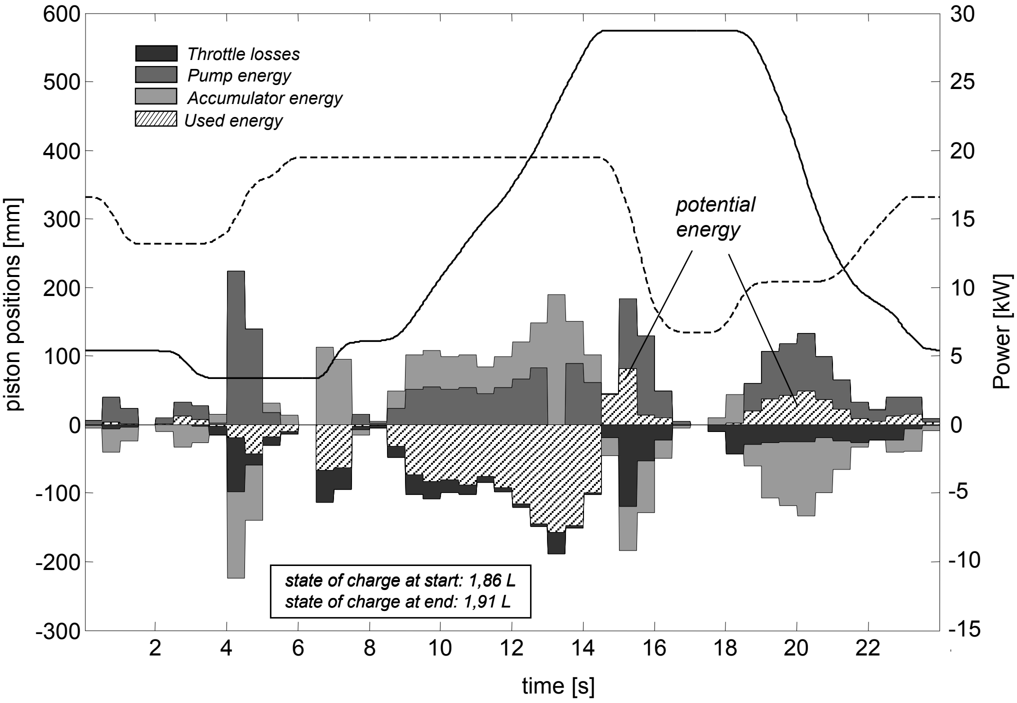

Figure 6: "Load & Carry" cycle with optimum switching sequence

In this case the result shows an energy saving potential of 20% compared to a LS system. As this analysis is a pure theoretical analysis the energy demand can be expected higher in real application due to higher throttle losses in the proportional valves.

3. Online Optimization using Model Predictive Control (MPC)

The energetic analysis of the duty cycle as described in paragraph 2 shows an optimum sequence of switching states with a considerably lower energy demand than a conventional LS system. This sequence was determined using a Pareto optimization algorithm which implies that the cycle is known. For a switching strategy which can be used online in real application an algorithm must be found which can determine an optimum switching sequence for the given cycle without "a priori" knowledge. The most promising results can be reached using a Model Predictive Control (MPC) /9/. MPC implies a mathematic model of the system and extrapolates the present state of charge for a definite time step into the future, called prediction horizon. Using this model the controller can approximate the future response of the system to interactions of actuators at the present instant of time which allows a positive influence on the system behavior for the prediction horizon.

In general, the piston movements of a wheel loader are very quick (a couple of seconds to lift the bucket) and can furthermore be interrupted at any instant by the operator. At these conditions a time extrapolation of the present load would be quite difficult to implement. Doing this, a time related MPC would lead to more instability of the system than to a gain of efficiency. For this reason a position related MPC was created which predicts precisely the evolution of loads on the lifting cylinder according to the present position of the piston. Due to the geometric properties of the wheel loader the load on the lifting cylinder is a linear function of the piston’s position. Once the arm is lifted and the present load and position are known the future evolution of loads is determined no matter which velocity will be chosen or whether the movement will be interrupted or not. Using this information the optimization problem is limited to the question at which positions the system needs to switch from one switching state into another with less energy use possible.

To find an answer to this question all six switching states are sorted in ascending order according to the maximum force which can be reached with this state beginning with IP/IP as the lowest and finishing with HP/TP as the highest force. For the given prediction horizon each section *s* of piston movement in one of these switching states needs to be determined to reach the total length h_{max}. Except HP/TP the maximum force of all switching states depends on the state of charge of the accumulator. When driving the piston at HP/IP for example, the accumulator is being charged and therefore the maximum possible force lowered due to the increasing pressure in the accumulator. Simultaneously the maximum force of IP/TP will increase as well as its possible duration because of the simple fact that there is more oil in the accumulator for effectuating the movement. A control algorithm needs to be found which is capable to determine the sections *s* for each switching state under consideration of its influence on forces and maximum reachable length of *s* for all following switching states. Furthermore the maximum reachable force of each switching state must never fall below the present load. The target function of the algorithm is the energy loss due to the difference of the maximum force of each switching state and the actual load. **Figure 7** illustrates the described optimization problem as well as the influence of the duration of the sections *s* to the respective maximum forces. Supposing the criteria of Bellman this problem can be solved analytically with the Dynamic Programming method /10/. This method determines precisely the sections *s* for the prediction horizon and avoids time expensive iterations which makes it suitable for an online optimization in a real application.

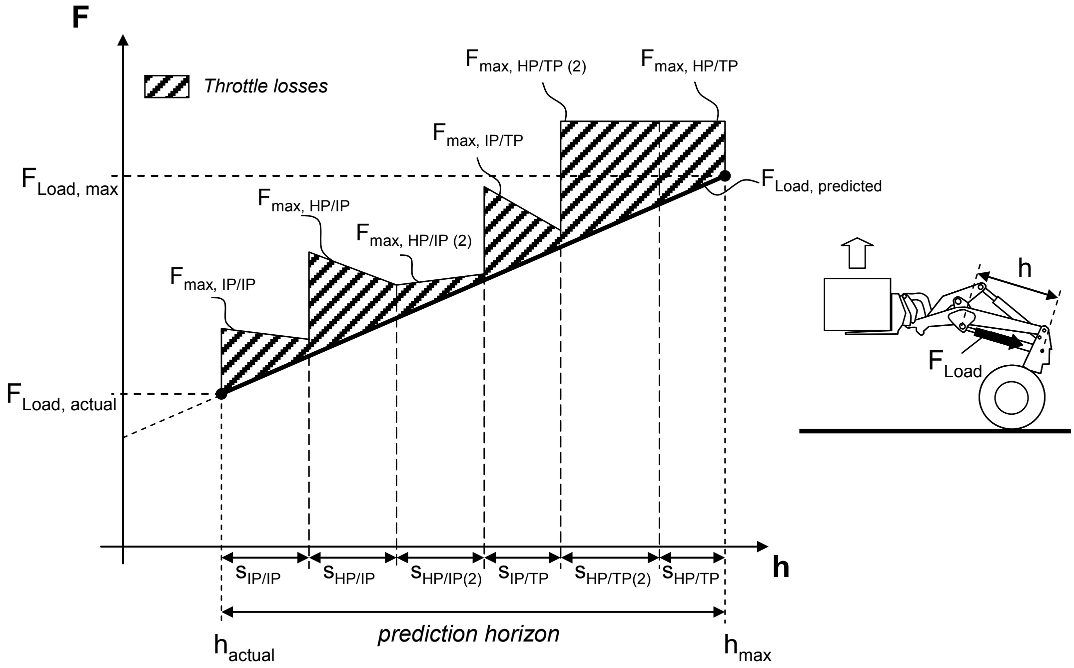

Figure 7: Prediction horizon for lifting

4. Summary and Outlook

The paper presents the idea of an energy efficient actuation of hydraulic cylinders in a constant pressure system when another line with intermediate pressure is used. This allows an integrated hydraulic system with secondary controlled hydraulic rotary drives and standard hydraulic cylinders which are operated at different pressure potentials between high pressure, intermediate pressure and tank pressure. Furthermore a theoretical improvement of efficiency of 20% compared to a standard LS system is shown when driven with an optimum switching sequence. An optimum sequence for a chosen specific duty cycle is determined by the use of a multi objective optimization. In order to reach best results in real application the switching strategy is programmed in a PLC with the Dynamic Programming method.

For proof of efficiency and improvement of the switching strategy the system will be tested in a virtual reality connecting the PLC with a simulation model of the wheel loader. The hydraulic system is modeled in the simulation tool *DSHplus* and transforms the output signals of the PLC into valve actuations and returns pressure signals back to the PLC /11/. The tested and improved control algorithm will be tested on a wheel loader in a real application. **Figure 8** shows the test machine with the mounted valve blocks and the accumulators for HP line and IP line. Each valve block consists of two switching valves and one proportional valve. The fuel consumption of the new system will be compared to the one of the LS system to prove the overall efficiency

Figure 8: Test machine with mounted valve blocks and accumulators

5. Acknowledgments

The project was performed by the Chair of Mobile Machines (Mobima) at the Karlsruhe Institute of Technology (KIT) in cooperation with ARGO-HYTOS GmbH, FLUIDON GmbH and Hermann Paus Maschinenfabrik GmbH. The project is financed by the Federal Ministry of Education and Research (BMBF) and supervised by the German Aerospace Center (DLR). The authors thank the BMBF and the DLR for the support of this project

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