Deterministic Control Strategy for a Hybrid Hydraulic System with Intermediate Pressure Line

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ABSTRACT

The paper introduces a new hydraulic system for mobile machines based on a constant pressure system with the aim to increase the efficiency of actuation of hydraulic cylinders. 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.

Due to the discrete pressure potentials of the system a control strategy is required which reduces throttle losses at the proportional valves and allows maximum recuperation of potential energy. Best results can be obtained if the future loads on the cylinders are predictable. For this reason a Model Predictive Control (MPC) was developed for a wheel loader which was used as a reference system. By using its specific geometric properties the MPC allows a precise load prediction as a function of the piston’s position. Using the criteria of Bellman, an analytic online calculation of the optimum sequence of pressure potentials and their durations for one complete cylinder stroke can be effectuated using Dynamic Programming. This leads to a deterministic algorithm which is easy to handle and which can be implemented into an online control loop of the wheel loader. The paper furthermore shows how an optimal switching sequence and the optimal accumulator parameters can be found offline using multi objective optimization and closes with simulation results showing an increase of efficiency of 13% compared to a LS system at the example of a typical duty cycle of a wheel loader.

1 INTRODUCTION

Constant pressure (CP) systems are one of the simplest hydraulic systems. They are 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/. Furthermore secondary controlled four quadrant drives can be operated in pump mode while braking which enables recovery of potential or braking energy which is stored in a hydraulic accumulator for later use. Both aspects, high system efficiency and energy recovery, make Advanced Constant Pressure Systems to a suitable drive system for hydraulic hybrid drives for mobile machines. 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. One concept of an efficient integration of hydraulic cylinders into a constant pressure system is the secondary controlled linear actuator which allows an at least discrete adaptation of the piston area to the present load /4/. Another way is the so called “hydraulic transformer” which consists of a constant flow pump connected to a secondary controlled drive /5/. Both systems have in common that they need components which are costly or which are simply not available at the market. Another approach for the efficient integration of linear actuators in a CP system by the use of standard components is shown in **Figure 1**.

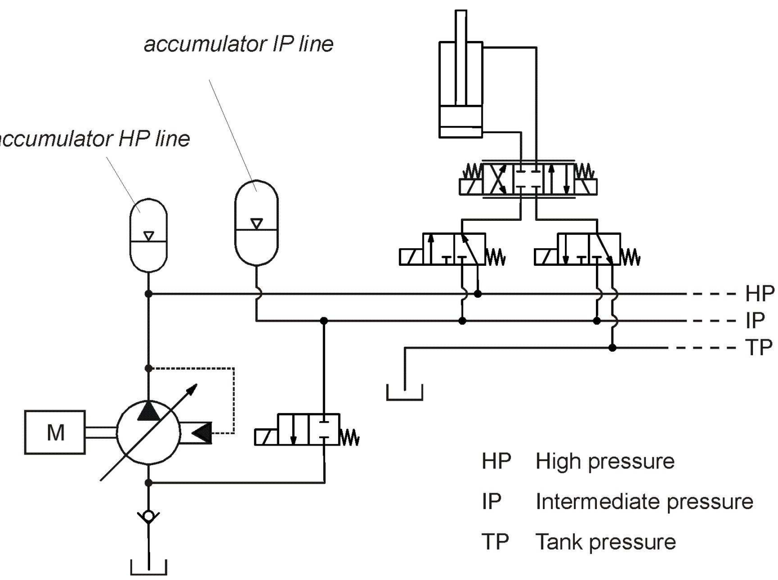


Figure 1: Basic layout of a CPIP system

In this system multiple discrete pressure potentials are used to reduce pressure losses at the proportional valves. These can be created if a supplementary pressure line is used. The pressure level of the additional line is located between high pressure (HP) and tank pressure (TP) and is therefore called “intermediate pressure (IP) line” /6/ so the system is called “Constant Pressure System with Intermediate Pressure Line” (CPIP). A hydraulic accumulator is connected to the IP line to enable recovery of 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.

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. 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. This results in six switching states with different use of pump and accumulator energy. **Figure 2** illustrates the energy flows for a piston movement at constant speed and constant load when a switching sequence is used which contains all possible switching states.

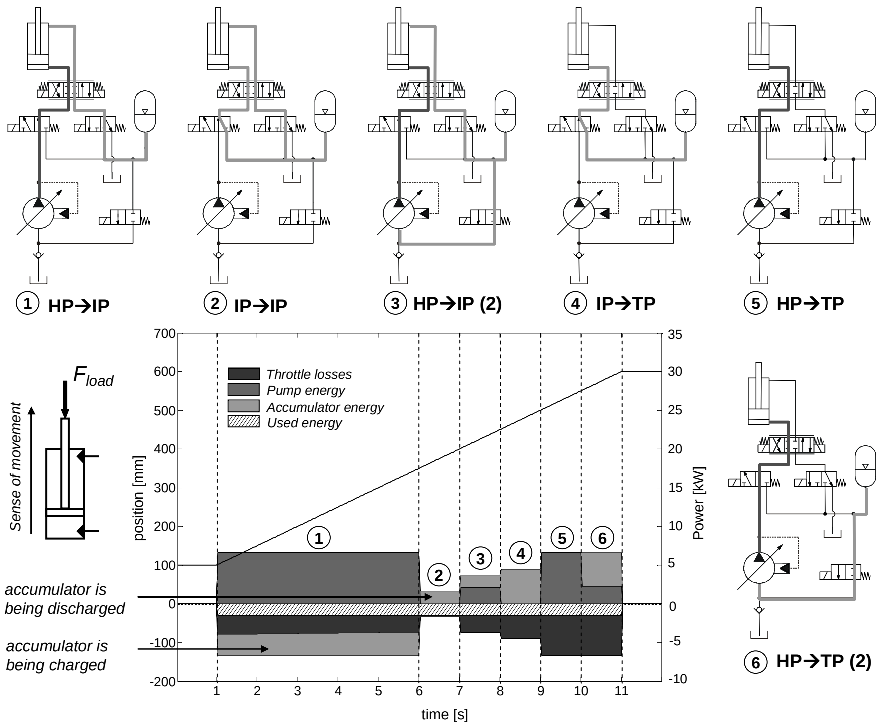


Figure 2: Possible switching states to effectuate a piston movement

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 the position of the piston 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 with the proportional valve according to the needed flow given by the operator. Furthermore, the algorithm has a 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 the global efficiency of the system. It therefore allows switching states which may cause high throttle losses at the moment but allow charging or discharging the accumulator in order to reach a more advantageous state of charge for future energy demands. These control strategies for hybrid drives are subject of intensive research /7/ as they essentially influence the overall efficiency of the system and represent the most important part of the control architecture, also in this system.

2 SYSTEM IMPROVEMENT WITH MULTI OBJECTIVE OPTIMIZATION

To develop a switching strategy which is able to find an optimum switching sequence a duty cycle with cylinder strokes and loads is needed. For this reason, the CPIP system was analyzed for the working hydraulics of a 4.7 t wheel loader. The considered actuators of the working hydraulics are two hydraulic cylinders, one for lifting and one for tilting. A typical duty cycle for this application is the so called “Load & Carry” cycle. In this cycle the loader loads the bulk material into the bucket and carries it to a transport vehicle where it is dumped. It can be composed in five different stages as illustrated in **Figure 3**:

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 the transport vehicle at (C).

3. The machine drives to the transport vehicle and lifts the bucket when arriving.

4. When 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.

The described cycle was repeated during tests about 90 times at realistic digging conditions. For the system optimization one representative duty cycle was chosen and the load pressures were transformed into forces acting on the pistons of the cylinders. This allows identifying the needed forces and calculating the needed energies in the different stages with the aim to get an energy profile of the duty cycle as shown exemplarily in Figure 2. To obtain an efficient system a sequence of switching states must be found which allows performing the movements of the duty cycle but with a lower energy demand than the current LS system.

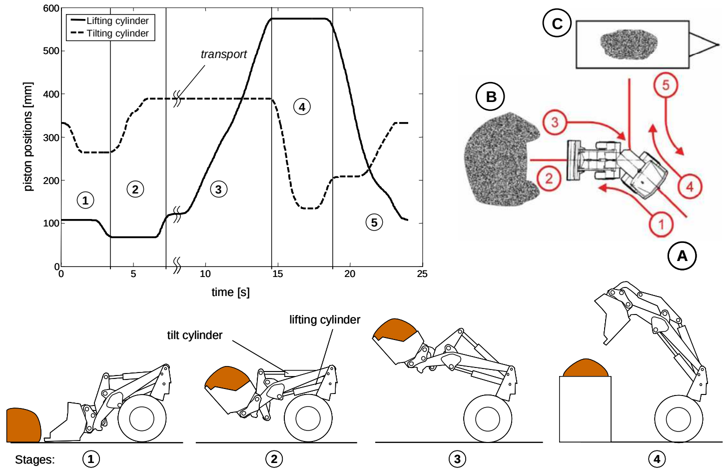


Figure 3: "Load and Carry" cycle of a wheel loader according to /8/

2.1 Optimal switching sequence

An optimum switching sequence for the given duty cycle is the one with the lowest use of energy delivered by the pump. As the pump directly feeds the high pressure line all switching states taking oil from the HP line like HP/IP, HP/IP (2), HP/TP and HP/TP (2) increase the energy demand of the system. These switching states have energy "costs" in form of the hydraulic energy E_{Pump} generated by the pump. Considering a switching sequence with n switching decisions the costs for HP/TP and HP/IP at decision i can be calculated (supposing $1 \leq i \leq n$). Assuming a constant pressure p_{HP} in the HP line the costs for HP/TP and HP/IP can be calculated according to the oil volume $\Delta V_{Cyl,HP,i}$ taken from the HP line at a cylinder stroke effectuated at i.

$$\Delta E_{Pump,i} = \Delta V_{Cyl,HP,i} \cdot p_{HP} \qquad (1)$$

Considering the entire duty cycle the total energy demand of the system after n switching decisions can be described as

$$E_{Pump} = \sum_{i}^{n} \Delta E_{Pump,i} \qquad (2)$$

The switching states IP/IP and IP/TP have no energy costs according to the definition given above as they only take energy from the accumulator. When used instead of HP/TP or HP/IP equation (2) is minimized as $\Delta E_{Pump,i}$ becomes zero. However, IP/TP and IP/IP reduce the state of charge (SOC) of the accumulator by $\Delta V_{Cyl,IP,i}$ so the used potential energy can be calculated as

$$\Delta E_{Acc,i} = \int_{V_{IP1}}^{V_{IP2}} p_{IP,i}(V_{IP})\, dV_{IP} \qquad (3)$$

$$\Delta V_{Cyl,IP,i} = V_{IP1} - V_{IP2} \quad (4)$$

$\Delta E_{Acc,i}$ is negative if the accumulator is being discharged and positive if the accumulator is being charged. This potential energy must be created by charging the accumulator using HP/IP or by lowering high loads with TP/IP (recuperation mode) or medium loads with HP/IP (2). The total accumulator energy after *n* switching decisions is therefore

$$E_{Acc} = \sum_{i}^{n} \Delta E_{Acc,i} \quad (5)$$

The switching states HP/IP (2) and HP/TP (2) combine pump energy and accumulator energy to reduce their costs as the accumulator of the IP line is connected to the suction side of the pump. The costs for HP/IP (2) and HP/TP (2) can be given as

$$\Delta E_{Pump,i} = \Delta V_{Cyl,out,i} \cdot p_{HP} + \int_{V_{IP1}}^{V_{IP2}} p_{IP,i}(V_{IP})\, dV_{IP} \quad (6)$$

The optimum switching sequence balances switching states with high costs and high generation of potential energy and switching states with low or zero costs and high consumption of accumulator energy in a manner that the global costs, e.g. the total energy demand E_{Pump} for the given duty cycle, become minimal. This is an optimization problem which can be solved with a multi objective optimization considering both energy types E_{Pump} and E_{Acc} as equivalent. The optimal decision changes the current energy state into a Pareto optimum as shown in **Figure 4**.

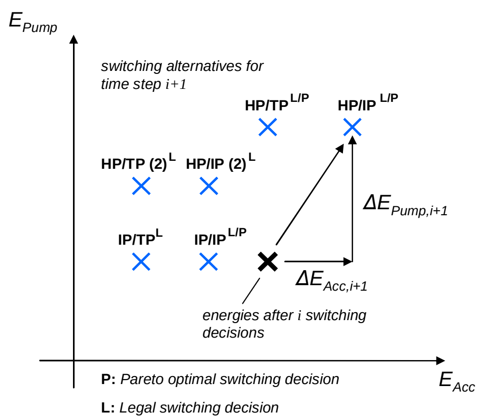


Figure 4: Pareto optimal decisions

To apply multi objective optimization the duty cycle is divided in *n* discrete time steps. At each time step *i* all “legal” switching states are identified. A switching state is legal if it generates a force which is high enough to move the piston in the current load situation.

Furthermore switching states which take oil from the accumulator can only be legal if the SOC is high enough to effectuate the cylinder stroke in the given time step. Starting from a state of total energy demand E_{Pump} and total accumulator energy E_{Acc} at a time step i the energies ΔE_{Pump} and ΔE_{Acc} for the next time step $i+1$ are calculated for all legal switching alternatives and added to E_{Pump} and E_{Acc}. The optimal switching decisions are those which do not worsen one energy type without improving the other one and therefore called Pareto optimal (see Figure 4). They are added to the already existing sequence of Pareto optimal switching decisions. If there is more then one optimal switching alternative the existing sequence must be duplicated by the number of optimal alternatives. The calculated energies E_{Pump} and E_{Acc} are used as starting energies for the next time step and the procedure is repeated. At the end of the duty cycle a set of points is found with different energies E_{Pump} and E_{Acc} following a Pareto frontier which begins with very low values for E_{Pump} and E_{Acc} and ending with very high values for both energy types. The searched sequence of optimal decisions is the one with the lowest pump energy demand and is shown in **Figure 5**.

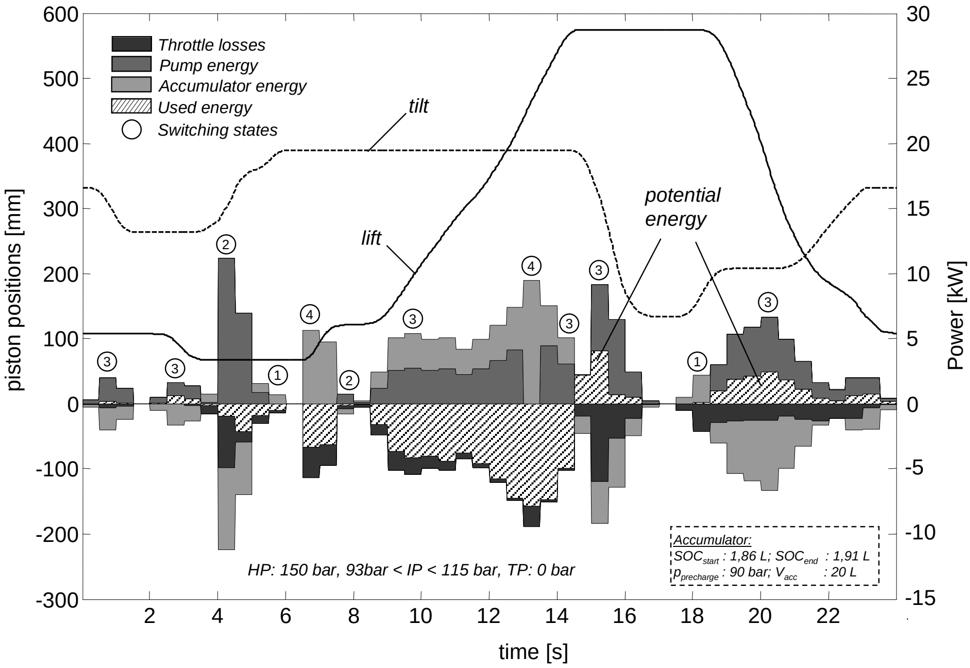


Figure 5: Optimal switching sequence for the given duty cycle

2.2 Optimal accumulator parameters

The described optimization method is done for a given duty cycle and fix accumulator parameters. In order to build up an optimized system the parameters *precharge pressure* and *accumulator size* can be found with the same method. Supposing a hydro-pneumatic accumulator with a polytropic gas behavior with the polytropic index κ the relation between gas volume and gas pressure can be given as

$$p \cdot V_{Acc}^{\kappa} = const. \qquad (7)$$

Applying the multi objective optimization on the same duty cycle by gradually changing the values for precharge pressure and gas volume of the accumulator a characteristic correlation between those parameters and the total energy demand can be observed in **Figure 6**. A valley can be identified showing low energy demands for all tested accumulator sizes at a precharge pressure of 90 bar. This valley has a smooth slope and reaches the global minimum of hydraulic energy consumption at 20 L gas volume and 90 bar precharge pressure. An accumulator with these parameters enables therefore the highest efficiency improvement of the system and was chosen to be mounted on the wheel loader.

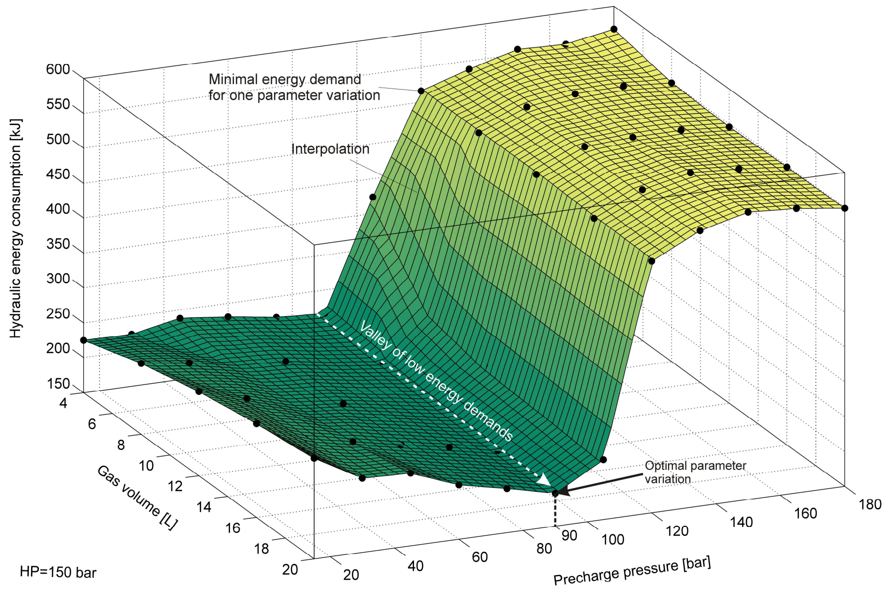


Figure 6: Results of parameter variation reveal optimal accumulator settings

3 MODEL PREDICTIVE CONTROL USING DYNAMIC PROGRAMMING

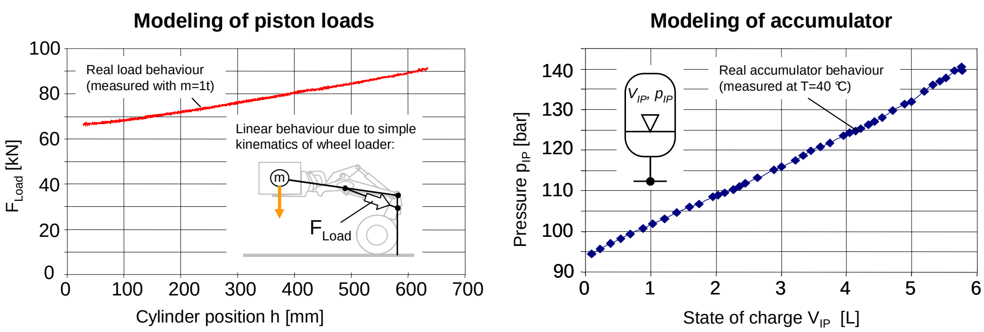
Besides an optimal switching sequence Figure 5 shows the mechanical energy which is needed in the hydraulic cylinders of the wheel loader. The working stages 1 and 2 (described in Figure 3) are characterized by load peaks which are relatively high but short so the needed energy is quite low. In these stages the switching valves must be fast enough to create the forces which are needed at the pistons but due to the spontaneous character of these peaks an optimization of the switching sequence can not really take place. At stage 4 potential energy occur at the tilting cylinder when the bucket is dumped and at the lifting cylinder when the arm is lowered. This energy must be recuperated in order to improve the energy demand of the system. At stage 3 it can be seen that most of the energy is needed for the lifting cylinder when the arm is lifted. Here the loads are high and the movement lasts long so an online optimization of the switching sequence can be effectuated. An optimization algorithm must be found which can take an optimal switching decision at any instant of the piston movement. This algorithm must be flexible, fast and easy to handle to use it as an online control algorithm. The most promising approach can be created with Model Predictive Control (MPC) /9/. MPC implies a mathematical 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 carry out. Doing this, a time related MPC would not improve the system as the precision of prediction would be very poor. For this reason a position related MPC was created which predicts precisely the development of loads on the lifting cylinder according to the present position of the piston.

3.1 Modeling of loads and of the accumulator

Due to the geometric properties of the wheel loader the load at 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 change from one switching state into another.

Another important aspect is the modeling of the accumulator. According to equation (7) it has a nonlinear behaviour which is difficult to handle. For this reason the accumulator behaviour needs to be linearized. One of the results of the multi objective optimization (shown in Figure 5) is that the accumulator is never charged with more than 5 L so a linear model represents a good approximation if compared to the real measured behaviour shown in **Figure 7**.



Load model:

$$F_{Load} = F_0(m) + b \cdot h$$

Accumulator model:

$$p_{IP} = p_0 + V_{IP} \cdot c$$

Figure 7: Modeling of loads and accumulator using measuring data

3.2 Online optimization algorithm with Dynamic Programming

Due to the linear correlation between load and piston position the loads on the lifting cylinder can be precisely predicted. The *prediction horizon* is therefore the cylinder stroke from the present position (which is measured) until its end position (which is estimated). A simplification of the optimization problem can be achieved if the variety of possibilities is limited by sorting the switching states in ascending order of maximal reachable force as shown in **Figure 8**. The switching sequence therefore consists of *n=6 periods* with different switching states for each period. The sequence begins with period *i=1* and IP/IP as the switching state with the lowest force and finishes with *i=6* and HP/TP as the one with the highest force. For the given prediction horizon the duration of each period *i* can be described with the length s_i representing the cylinder stroke driven with the respective switching state. The prediction horizon is therefore the sum of all lengths $s_1 \ldots s_6$. As almost all switching states influence the SOC of the accumulator, the choice of the length s_i at the period *i* is called *policy* as it influences all switching decisions in the succeeding periods and therefore the efficiency of the whole movement. When driving the piston at HP/IP for example, the accumulator is being charged and therefore the maximum possible force is continuously lowered due to the increasing pressure in the accumulator (which is considered as a linear function of the SOC). 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 to effectuate the movement. The online optimization algorithm must therefore determine the optimal policy $s_{opt,i}$ for each period *i* in a way that the throttle losses, which are described by the *objective function* E_{Loss} are as low as possible. The Dynamic Programming optimization method /10/ supposes the criteria of Bellman describing a policy in a period *i* as optimal if it minimizes the objective function of the same period *i* and the one of the succeeding period *i+1*.

$$E_{Loss,\min,i}\left(h_i, V_{IP,i}\right) = \min\left\{E_{Loss,i}\left(h_i, V_{IP}\right) + E_{Loss,\min,i+1}\left[T\left(h_i, V_{IP,i}\right)\right]\right\} \quad (8)$$

$s_{opt,i}$ is the solution for equation (8) and minimizes $E_{Loss,i}$ which are throttle losses and result from the difference of maximal forces of the states and the load at each period. Due to the dependence of the maximal force on the SOC $V_{IP,i}$ and the dependence of the load on the position h_i of the piston the objective function $E_{Loss,i}$ depends on both parameters which are variables describing the *states* at the beginning and end of each period. The throttle losses E_{Loss} at the stage *i* and the throttle losses at the succeeding stage *i+1* need to be linked to calculate the minimum at *i*. This link is done here with the *transition functions* $T[h_i, V_{IP,i}]$ which describe the linking of the SOC V_{IP} and the piston position *h* of two periods:

$$h_{i+1} = h_i + s_i \quad (9)$$

$$V_{IP,i+1} = V_{IP,i} + s_i \cdot A_{Piston,i} \quad (10)$$

As the accumulator is being charged or discharged in dependency of which switching state is presently used, the piston surface A_{Piston} and its sign depend on the present stage *i*.

With the easy initial calculation of the optimum length of the switching state at the last stage using the estimated final position h_7

$$s_{opt,6} = h_7 - h_6 \tag{11}$$

equation (8) can be solved top down calculating $s_{opt,i}$ for each period until arriving at index 1 with the initial state variables h_1 and $V_{IP,1}$ which are measured and therefore known. The results for $s_{opt,i}$ are functions which depend on the state variables h_i and $V_{IP,i}$ of each period and are stored in a table. If the initial state variables h_1 and $V_{IP,1}$ change, the optimum switching sequence is adapted immediately by recalculating bottom up all values s_i which makes this optimization algorithm very fast and deterministic as it does not approach iteratively the optimum but it calculates it analytically in a single run.

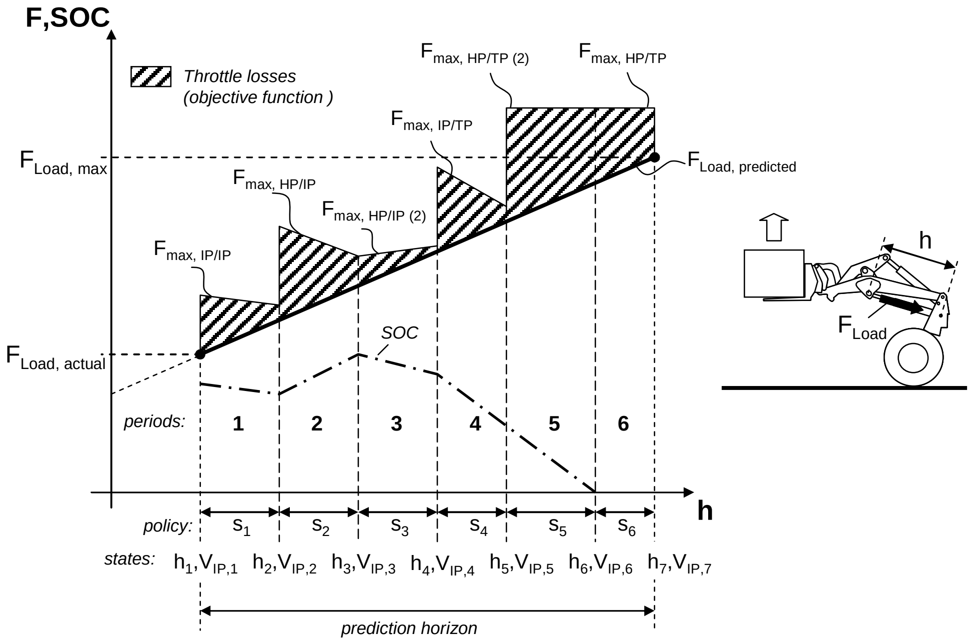


Figure 8: Principle of online optimization with Dynamic Programming

3.3 Simulation results

A big advantage of the online optimization algorithm is the deterministic calculation of global minima. The equations are simple as the used mathematical models to describe load and accumulator behaviour are linear. For each switching state of the lifting cylinder exists one equation which is stored in the controller of the system. At each calculation cycle of the controller the input data like piston loads, piston positions and intermediate pressure are transformed into loads, forces and SOC and the equations are solved. This very simple and fast online optimization algorithm is programmed into a Programmable Logic Controller (PLC) which will pilot the system of the experimental vehicle. Before being applied on the machine, the program code is tested in a simulation environment first. For this purpose a simulation model was created and validated with measuring data from the wheel loader. In order to make the virtual test environment as realistic as possible and to have an efficient tool to improve the program code of the PLC, the simulation model was connected directly to the program code via an Object Linking and Embedding for Process Control (OPC) interface as described in /11/. This enables an interchange of data between program algorithm of the PLC and the simulation model. A realistic system behaviour can be achieved allowing a first validation if the online optimization is working and how the system does improve the efficiency of the system. **Figure 9** shows the simulation results of four succeeding Load & Carry cycles. In this simulation the duty cycle is not known by the control algorithm, the controller need to find the optimal switching decision at any instant. Thanks to the online optimization algorithm the switching sequence is very close to the one found with the offline optimization method and an improvement of efficiency of 13% compared to a LS system could be determined.

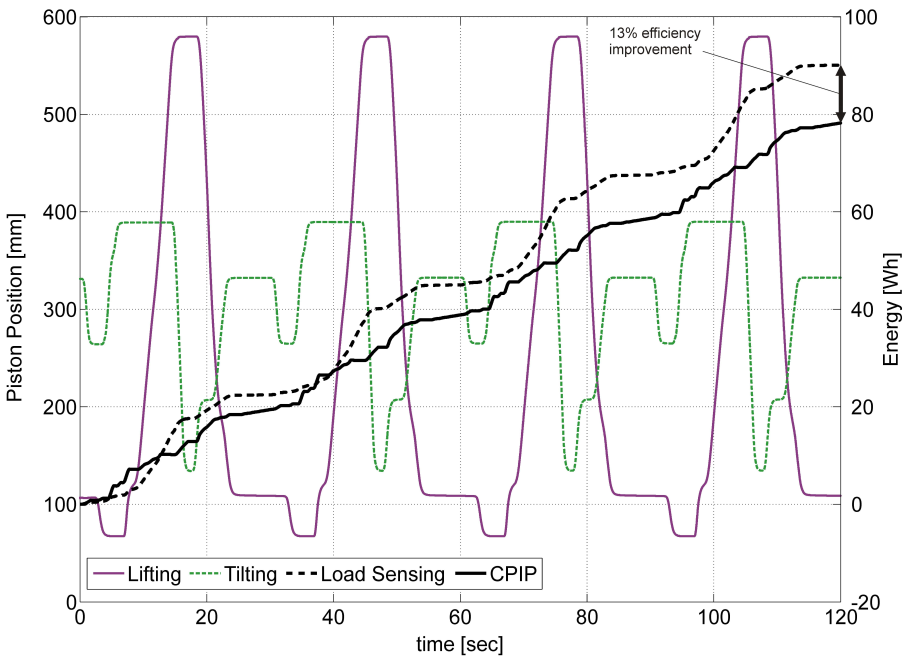


Figure 9: Results of simulation run of four cycles showing efficiency improvement

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 hybrid 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. An optimum sequence for a chosen specific duty cycle is determined by the use of a multi objective optimization. An online control algorithm was developed using the Dynamic Programming method and tested in simulation showing an efficiency improvement of 13% compared to LS.

The control algorithm which will be used for the logic controller will be tested and improved by the help of software-in-the-loop simulations /11/. The wheel loader is equipped with valve blocks and accumulators for HP line and IP line as shown in **Figure 10**. The fuel consumption of the new system will be compared to the one of the LS system to prove the overall efficiency.

Figure 10: Results Test machine with mounted valve blocks and accumulators

5 ACKNOWLEDGEMENTS

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