Development and implementation of a control concept for a hydraulic load unit

REVIEWED

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Functionality and performance of novel hydraulic systems under real life stress can usually be examined in field tests only. In order to gather information about the behavior under stress during the development process as soon as possible, system components as well as systems get tested on test rigs. In hydraulics, applying passive loads e.g. to linear actuators can easily be done by throttling the outflow. For rotatory units, loads can either be applied with ropes and masses or other rotatory units. Especially applying active loads i.e. loads with the same orientation as the motion of the cylinder, is difficult and usually connected to a high complexity. At Karlsruhe Institute of Technology (KIT), a hydraulic load unit for hydraulic cylinders was developed to be used at various test rigs. The load units' controller design allows for the application of either active or passive loads in variable directions and intensities. The following paper introduces the load unit, its open- and closed-loop control concept and the verification results from simulation, which show the potential of the load unit.

Keywords: hydraulic load unit, simulation of active and passive loads, open- and closed-loop control concept
Target audience: Mobile Hydraulics, Mobile Working Machines, Test Bench Simulation

1 Introduction

Improving the efficiency of hydraulic systems is a very important task of modern system and component engineering, motivated by reduction of mobile machine emissions and protection of natural resources. /1/ Thus, various research projects focus on developing new concepts, systems and components for optimized mobile machines (e.g. /2/, /3/, /4/, /5/, /6/). Not every project leads to a fully grown mobile machine prototype and therefore test rigs are often used for testing. Benefits of test rigs are e.g. a higher reproducibility of conducted experiments and easier data acquisition compared to a field test with a mobile machine. /7/ Nevertheless, comprehensive and significant test results of a system and its behavior under real life stress can usually be conducted only during field operation tests of the machine. /8/

A load duty cycle usually consists of two types of loads: active and passive loads. In this paper, the terms active and passive loads will be used according to /9/. A load is called active (or overrunning) load, whenever the force applied to an object and the objects' direction of motion have the same orientation (Figure 1, left). For a passive load, force and motion have the opposite direction (see Figure 1, right). For hydraulic systems, active loads need to be considered especially during the design process, because hydraulic oil can only transmit compressive loads, but no tensile loads, which can lead to unsafe or dangerous situations in operation.

Since extensive field tests usually cannot be conducted in the early stages of development, different ways of applying loads and stresses on test rigs are available.

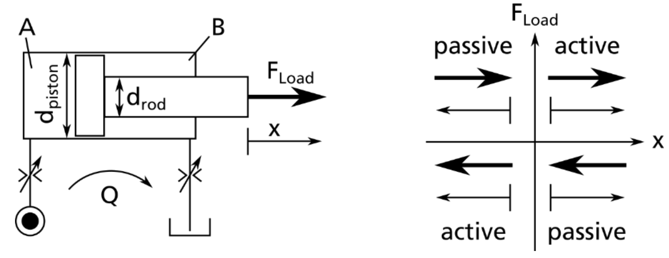

Figure 1: Visualization of Active (left handed) and Passive (right handed) Loads, according to /10/

A very common method to apply passive loads on hydraulic cylinders is a down-stream throttling valve, e.g. a *proportional pressure relief valve (PPRV)*. The increased outlet pressure caused by the PPRV results in a force acting on the piston, which has to be compensated by a force applied by the inlet pressure. For example, this effect is also used by lowering break valves, which help to avoid cavitation during lowering /11/.

The advantages of this method are an easy handling and implementation. By using a bypass valve and a PPRV in both cylinder ports, passive loads can be applied in both directions. As a disadvantage, no active loads can be applied this way.

Active loads can be applied on cylinders e.g. by using test masses /12/.With masses, however, active loads can only be applied in one movement direction – since the force of the test mass is caused by gravity, the load in the opposing direction is passive. Applying exchangeable active and passive loads in both movement direction without changing the setup of the test rig is often not easy and very complex /9/.

Another, more flexible method of applying loads is using a load cylinder and a test cylinder. In this configuration, the test cylinder simulates the motion while the load cylinder applies the intended load, preferably independent of the direction of motion. /8/

In order to improve the test capabilities, the *Chair of Mobile Machines (Mobima)* of the *Karlsruhe Institute of Technology (KIT)* developed a linear hydraulic load unit which can apply a load duty cycle, both variable in force direction and magnitude and thus interchangeable between active and passive loads. The main focus of the development was on the design of the load unit and its control concept, which is required to make possible a force application independent of the motion direction of the cylinders. Also, the control concept should prevent cavitation due to load magnitude and direction changes and allow for a high reproducibility and accuracy of applied load cycles. More detailed information about the control concept can also be found in /13/.

In this paper, the developed load unit and the implemented control concept will be introduced and discussed. The results of functionality verification by simulation will be shown and discussed. At the end of paper completion, the load unit was set up on the test rig, but was not put into operation yet.

2 Functional Description

Figure 2 shows the scheme of the developed load unit and its components. It consists of two collinear hydraulic differential cylinders, henceforth called load cylinders, which can apply stress on the test cylinder. Both load cylinders and the test cylinder are connected to a mechanical carriage, see Figure 2 and Figure 3. The mechanical carriage can be moved horizontally and loaded with additional weights, e.g. metal plates. By this, additional inertia can be applied to the load unit. The flange of the mechanical carriage is exchangeable to fit multiple test cylinders. /14/ Both load cylinders can be controlled by a *4/3 proportional directional valve (4/3 PDV)* with upstream pressure compensators. The load can be adjusted by two PPRVs in closed-loop pressure control. /14/

Since active loads can be applied, the load unit needs to be powered by an additional power supply during operation. At Mobima, it will be powered by a central pressure supply with *closed-loop pressure control (CPS)*, q.v. /15/. /14/

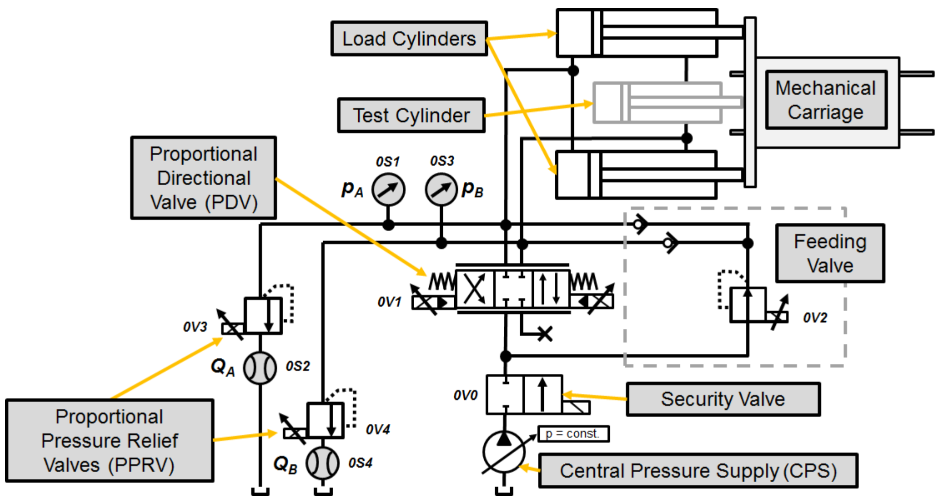

Figure 2: Scheme of Load Unit, according to /13/, /14/

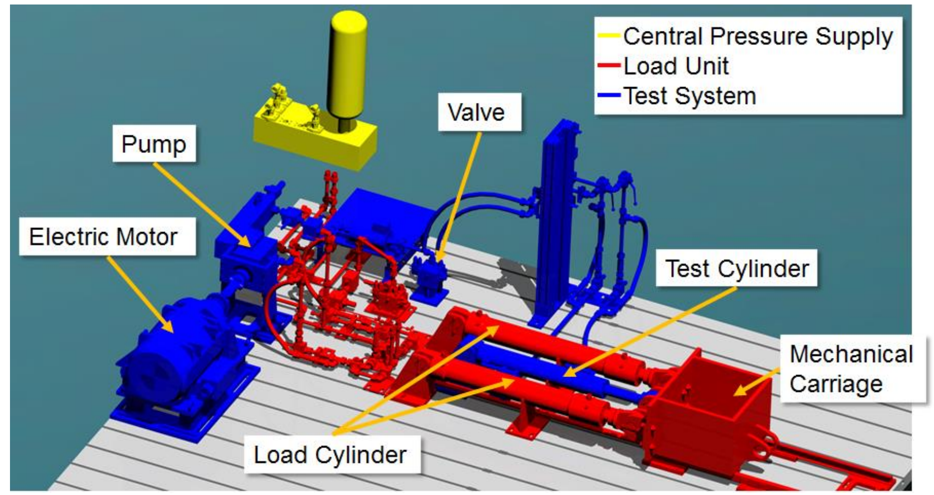

Figure 3: Set-up of the Hydraulic Load Unit

To avoid cavitation, the entire load unit system is preloaded with a pressure of 20 bar. Furthermore, the load cylinders can draw oil via a bypass feeding valve (0V2), as soon as the pressure drops below the preload pressure. This happens if the load cylinders are resided in an active load situation. In case of failure or emergency, the load unit can be disconnected from the power supply by an additional security valve (0V0) immediately. During normal operation, the security valve (0V0) is fully opened. /13/, /14/

In Figure 2, several pressure transducers (0S1, 0S3) and flow rate transducers (0S2, 0S4) are depicted. They measure the quantities necessary for closed-loop control and status monitoring. 0S1 detects the pressure in the piston chamber, 0S3 the pressure in the rod chamber of the load cylinders. 0S2 measures the flow rate from the piston chamber, 0S4 the flow rate from the rod chamber of the load cylinders.

In the current configuration of the load and test cylinders, the maximum load cylinders extension velocity is approximately $v_{out} = 0.109\ \mathrm{m/s}$, the maximum retraction speed is $v_{in} = 0.184\ \mathrm{m/s}$. Both maximum speed levels can be increased by changing either the installed 4/3 PDV (0V1), by using smaller load cylinders or by increasing the maximum flow rate of the CPS.

Figure 3 shows the hydraulic load unit (red), an exemplary test system (blue) and part of the central pressure supply (yellow) of Mobima / KIT. The test system consists of an electric asynchronous machine driving a hydraulic load sensing pump with variable displacement, conductive components like valves, pipes and hoses and a test cylinder. The test cylinder is connected to the mechanical carriage, which is also connected to the load cylinders of the load unit.

Since mobile machines and their hydraulic systems come in various sizes, setting up an equally scaled test system is not always possible and reasonable. Thus, scaling of system and load parameters to a viable magnitude is necessary. By respecting principals of dimensional analysis (e.g. Buckingham π theorem, see /16/), gathered test results of scaled models still can be deemed as valid.

3 Control Concept

Figure 4 shows a flow chart of the load unit control concept.

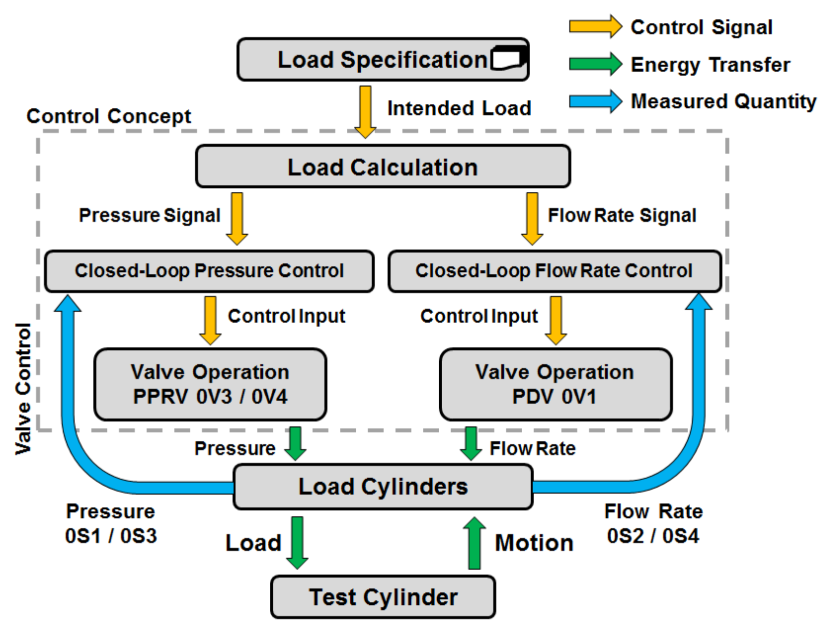

Figure 4: Flow Chart Control Concept Load Unit, adapted from /13/

Previous to testing, the load has to be specified, e.g. by using a time series of the load force applied to the test cylinder. Also, various parameters such as cylinder diameters, supply pressure and maximum pressure level have to be specified, since they have an influence on the *Load Calculation* module.

The control concept consists of three major components: the *Load Calculation* module, the *Closed-Loop Pressure Control (CLPC)* and the *Closed-Loop Flow Rate Control (CLFRC)* module.

The Load Calculation module processes the load specification data from the dataset into two separate control signals – a pressure signal and a flow rate signal. The maximum resulting load force F_{res} depends on the direction of movement and thus on the pressurized piston surface (piston or ring). If necessary, the intended load F gets limited and scaled within the permitted operation range in accordance with the specified parameters, cf. Equation (1).

$$F_{res,max/min} = k \cdot (F_{pA/B,max} - F_{pB/A,min}) \tag{1}$$

$$F_{pA,\mathrm{max/min}} = \frac{\pi}{4} \cdot \mathrm{d}_A^2 \cdot p_{A,max/min} \; ; \; with \; d_A = d_{piston} \tag{2}$$

$$F_{pB,\mathrm{max/min}} = \frac{\pi}{4} \cdot \mathrm{d}_B^2 \cdot p_{B,max/min} = \frac{\pi}{4} \cdot (\mathrm{d}_{piston}^2 - d_{rod}^2) \cdot p_{B,max/min} \tag{3}$$

$$F \in [\mathrm{F_{res,min}}, \mathrm{F_{res,max}}] \tag{4}$$

The resulting load force F_{res} is limited by the maximum pressure either of the loading or test cylinder. If it is assumed that for all configurations the load cylinders are always stronger than the test cylinder, the saturation depends on the test cylinder parameter $d_{piston}, d_{rod}, p_{A,max/min}$ and $p_{B,max/min}$, cf. Figure 1. To satisfy that the test cylinder movement will not be corrupted, the test cylinder force has to be larger than the resulting load force F_{res}. Therefore, the factor k reduces $F_{res,max/min}$ to include mechanical resistance and other losses, which can reduce the test cylinder force.

Then, both necessary chamber pressures of the load cylinders are calculated by using the transformed intended load F and the pressurized piston surfaces $A_A = \frac{\pi}{4} \cdot \mathrm{d}_{piston}^2$ and $A_B = \frac{\pi}{4} \cdot (\mathrm{d}_{piston}^2 - d_{rod}^2)$ with respect to the minimum load unit pressure of $p_{min} = 20$ bar, cf. Equation (5) and Figure 5. The calculated pressure signals $p_{A/B,set}$ then are forwarded to the CLPC module.

$$p_{A/B,set} = \frac{F - 2 \cdot (F_{pA,min} - F_{pB,min})}{2 \cdot A_{A/B}} + p_{min} \tag{5}$$

Additionally, flow rate signals are calculated. Depending on predefined load cases, cf. Figure 5, the chamber is chosen, which has to be supplied with oil and the corresponding valve signal is forwarded to the 4/3 PDV of the load unit. The threshold between both load cases depends on the load cylinder specifications as well as on the minimum system pressure of 20 bar. In the current configuration, the threshold force is approximately 20 kN.

Subsystem CLPC compares the current values of the pressure transducers (0S1, 0S3) with the calculated pressure values from the Load Calculation module. CLPC also uses the load case signal $LC_{1/2}(t)$ to compensate effects in active load situations. The deviation of both comparisons is then forwarded to the implemented control system. The controller adjusts the current pressure levels of the load cylinders to the nominal pressure levels and thus adjusts the intended force magnitude and direction.

Subsystem CLFRC adjusts the oil supply of both load cylinders by using the previously specified load cases and parameters. Furthermore, the CLFRC reduces the oil requirement of the system to a minimum. This feature is necessary for the load unit as long as it is powered by a power supply with closed-loop pressure control. Due to the feeding valve (0V2) the load unit will act like an open centre system. Thus, depending on the pressure difference between CPS and load unit pressure, an unnecessary flow rate would produce energy losses as well as unnecessary heating and stress for the hydraulic fluid. The flow rate is adjusted by a closed-loop control of the 4/3 PDV (0V1) to a nominal value of at least 20 l/min, in order to avoid cavitation and to prevent a pressure collapse caused by load or movement changes.

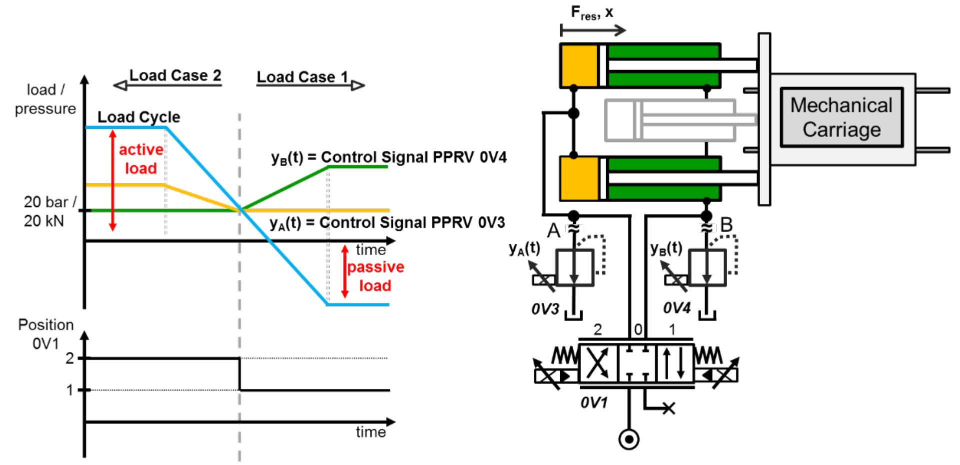

Figure 5: Nominal Pressure Signals and Spool Position of the 4/3 PDV (0V1) Depending on Load Case, adapted from /13/

The upper left diagram in Figure 5 shows an exemplary load progression (curve *Load Cycle*) and the corresponding nominal signals $y_A(t)$ (curve *Control Signal PPRV 0V3*) and $y_B(t)$ (curve *Control Signal PPRV 0V4*). The spool position of the 4/3 PDV (0V1) is depicted in the lower left diagram of Figure 5.

While in load case 2, the resulting load force F_{res} is oriented in extending direction caused by the pressure adjusted by PPRV 0V3. PPRV 0V4 provides the system minimum pressure of 20 bar. While the load magnitude decreases, the nominal signal of PPRV 0V3 also decreases. When the load drops below 20 kN, load case 1 is active. Thus, the nominal pressure of PPRV 0V3 is set to 20 bar, while the nominal pressure of PPRV 0V4 increases to the specified level. In this case, F_{res} acts in retracting direction. For example, in this case an extending test cylinder would be applied with a passive load (load case 1) and an active load (load case 2) during its movement.

In the upcoming sections, the modules *Closed-Loop Pressure Control* and *Closed-Loop Flow Rate Control* will be described in more detail.

3.1 Closed-Loop Pressure Control (CLPC)

The module CLPC, see Figure 6, calculates the deviation $\Delta p_{A/B}$ from the measured pressure $p_{A/B,current}$ and the nominal pressure $p_{A/B,set}$ calculated by the module *Load Calculation*. Then $\Delta p_{A/B}$ is forwarded to the appropriate PID-controller to adjust the pressure in the piston or rod chamber. The PID-controller's output $u_{A/B}(t)$ added to the nominal value $y_{A/B,set}(t)$ results in the control signal $y_{A/B}(t)$, which operates the PPRVs 0V3 and 0V4. Thus, each PPRV is operated by its own *closed-loop control system* (CLCS), cf. Figure 6.

The example in Figure 6 shows a load cylinder with a resulting load force F_{res} in extending direction as a result of forces F_{pA} and F_{pB}. If the load cylinder gets moved against the direction of F_{res}, here in retracting direction, the current pressure in the rod chamber (green) $p_{B,current}$ can drop below 20 bar due to an unsatisfactory supply with oil caused by friction losses in components, piping and others. This affects the counter force F_{pB} and thus changes the resulting load force F_{res}. Since the pressure control loop of PPRV 0V4 is not active at this moment, the rod chamber pressure cannot be raised by PPRV 0V4. In order to simulate the intended load F exactly, in this case, the deviation of the passive CLCS Δp_B is added to the deviation Δp_A via switch LC_1(t) (depicted as closed) right before the PID-controller, see Figure 6, thus altering the pressure in the piston chamber in an appropriate way to fit F_{res} to the intended load F.

Thus, changes of load force caused e.g. by changes of movement, can be compensated by the reciprocal interaction of the passive and active CLCS, depending on load cases LC_1 and LC_2.

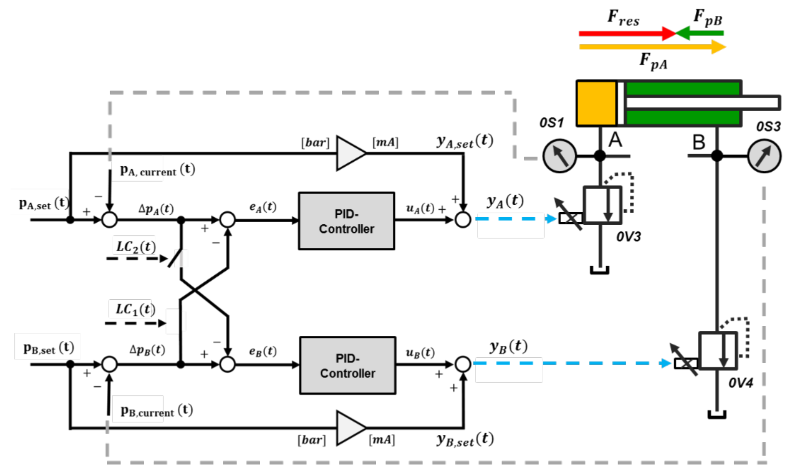

Figure 6: Module CLPC in Detail & Balance of Forces for a Load Cylinder, according to /13/

3.2 Closed-Loop Flow Rate Control (CLFRC)

The module CLFRC pursues two objectives: establishing a sufficient oil supply for the load cylinders independent from the movement and reducing the flow rate demand of the load unit and thus avoiding unnecessary temperature increase and its impact on the fluid. Both objectives are achieved by measuring the flow rate of both PPRVs and adjusting it either to the nominal flow rate or at least to a minimum level of 20 l/min by a closed-loop flow rate control system. Figure 7 shows the boundary conditions of the flow rate control.

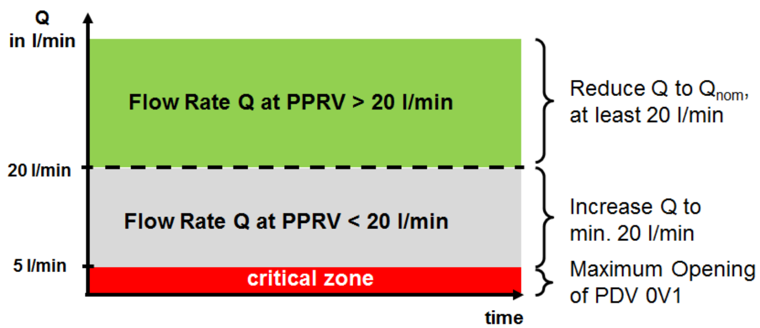

Figure 7: Boundary Conditions of the Closed-Loop Flow Rate Control, according to /13/

As previously described, a minimum flow rate of 20 l/min is always necessary for a smooth operation. In order to reduce the losses, every flow rate above 20 l/min gets reduced to its nominal value, always provided for the minimum flow rate. In some situations, e.g. a sudden change in direction or speed, the flow rate at the PPRVs can drop below its specified minimum. To avoid any resulting damage, a critical zone was defined, beginning at approximately 5 l/min. As soon as the flow rate drops into this range, valve 0V1 is opened to 100 %. When the flow rate is back in its designated range, the previous signal path is re-established and 0V1 returns to its intended opening.

Figure 8 shows the control concept implemented in the module CLFRC.

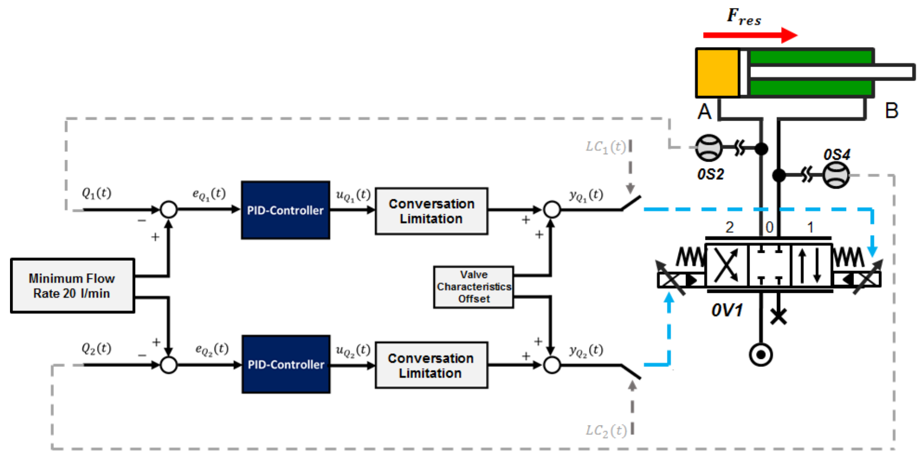

Figure 8: Module CLFRC to Operate the 4/3 PDV 0V1 Depending on Load Case, according to /13/

The current flow rate value Q_i and the nominal minimum flow rate are compared. The flow rate deviation $e_{Q,i}(t)$ then is fed forward into a PID-controller. Next, the output signal from the controller $u_{Q,i}(t)$ gets converted and limited in accordance with the predefined specifications. If necessary and to fit different valve characteristics, an additional correction value can be added to the control signal $y_{Q,i}(t)$ of the valve. In the current configuration, a linear valve characteristic was modelled and adjusted appropriately. Depending on the load case, see Chapter 3, the control signal $y_{Q,i}(t)$ is fed to the 4/3 PDV (0V1) which then moves into the intended position 1 or 2. In case of emergency, both control signals are interrupted so that the valve returns to idle position 0, which interrupts the oil supply of the load cylinders.

A safety rule is implemented to avoid cavitation in the load cylinders caused by an active load from the test cylinder. If the PPRV flow rate drops below a value of 5 l/min (critical zone, see Figure 7), the PID controller signal gets overruled by an additional signal, causing the 4/3 PDV (0V1) to open to maximum. This results in an immediate increase of the supply flow rate supported by the power supply of the load unit. Thus, pressure break down in the cylinders and the accompanied problems like cavitation and sealing damage can be avoided effectively.

4 Simulation Results

The functionality of the load unit and its control concept was verified by simulation. The hydraulic system, the load unit itself and an exemplary test cylinder system were simulated by using DSHplus, by Fluidon. The control concept, see Figure 4, was modelled in Matlab / Simulink, by Mathworks. The verification was conducted by using coupled simulation between DSHplus and Matlab / Simulink, see Figure 9.

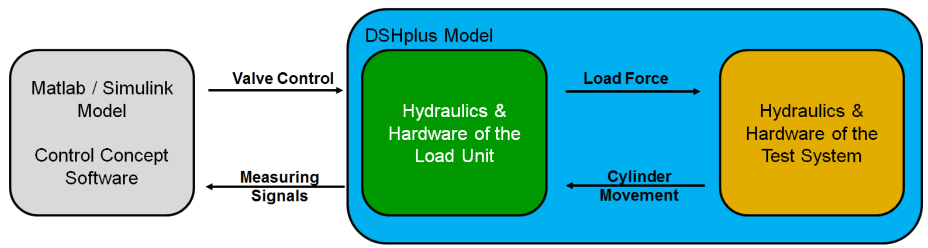

Figure 9: Scheme of the Simulation Model

For verification, the boom cylinder model of a forestry crane from /17/ was used, see Figure 10. The boom cylinder was equipped with an electro hydraulic flow-on-demand system, consisting of a variable displacement pump, an 8/3 PDV with downstream pressure compensators and an electronic control unit. In the simulation, the crane system was operated by a virtual user according to /18/. The virtual user is a closed-loop position control with weighted predictive behavior. Load and positioning trajectory were based on measurement data from /17/. The dataset was taken from the working process of the forestry crane while positioning a log and feeding it to a hydraulic debarker. The cycle had a duration of 30 s.

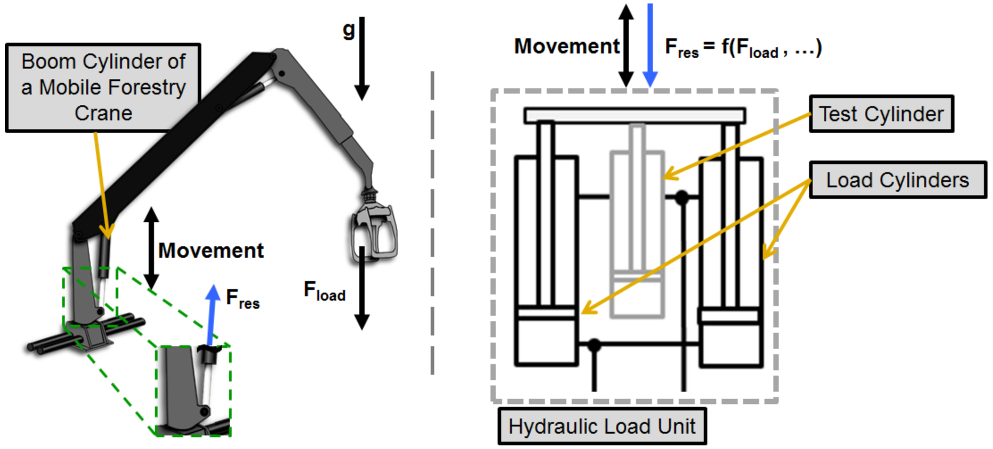

Figure 10: Adapting a Real Working Machine to a Test Rig Model

In Figure 11, the cylinder movement as well as the load on the boom cylinder caused by the working cycle can be seen. During retraction, the position trajectory has a negative gradient, during extension it has a positive gradient. Since the load always acts in the retracting direction, the cylinder is affected by a passive load during lifting and an active load during lowering the arm. The load trajectory shows the highest absolute loads with a magnitude of approximately 275 kN in the time ranges from 5 s to 10 s and from 16 s to 22 s. The high loads were caused by lifting the log, refer to /17/. For the rest of the time, the nominal load was caused by the kinematics of the machine itself and by dynamics.

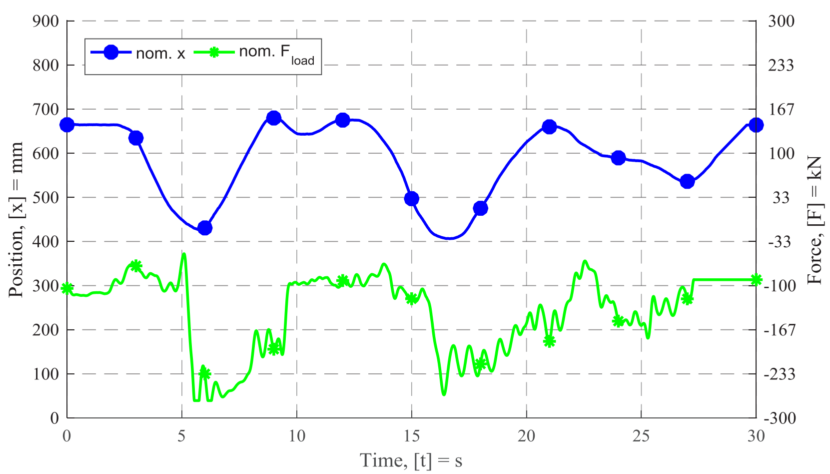

Figure 11: Reference Cycle Data –Position Trajectory and Nominal Load, according to /17/

The results of the simulation using the previously described model are depicted in Figure 12 and Figure 13.

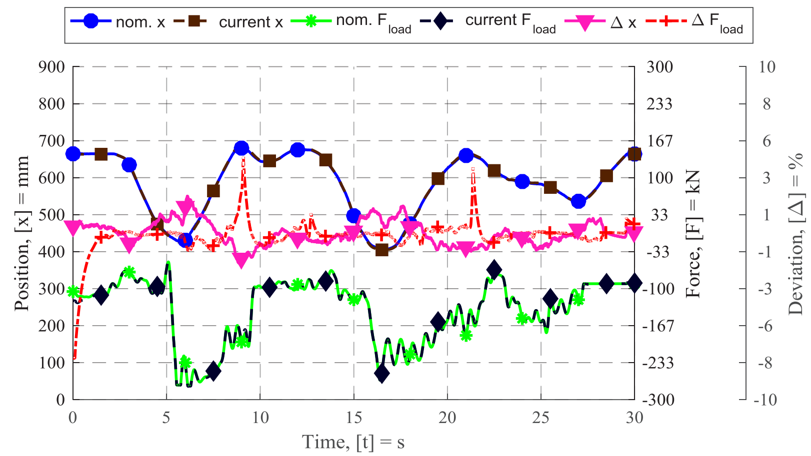

Figure 12: Nominal Load, Position and Load Deviation Trajectory, according to /13/

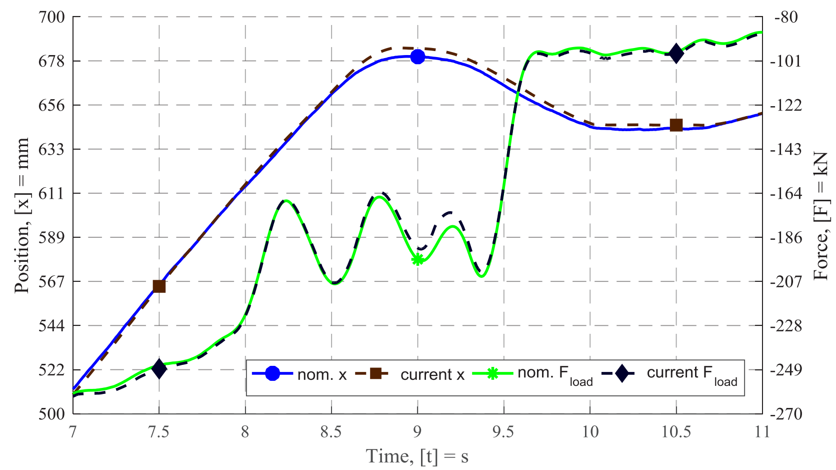

Figure 13: Load and Position in Detail

The cylinder position trajectories of reference and simulation have minor deviations of less than 3 % absolute, see curve Δx, Figure 12. Also changes in the movement direction can be compensated. The control concept thus allows for a high accuracy of position and velocity.
The load trajectories also have minor deviations of less than approximately 8 % absolute, which are mainly caused by changes in load and motion (curve ΔF_{load}, Figure 12). The deviation peak at the beginning of the cycle is caused by initializing effects of the simulation.

Figure 14 shows the pressure curve progression in the piston and rod chamber of the load cylinders.

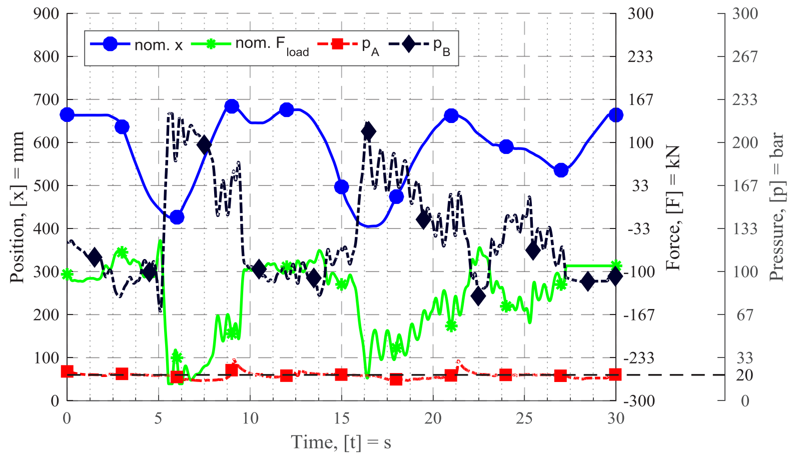

Figure 14: Pressure Trajectory of Piston (p_A) and Rod (p_B) Chamber of Load Cylinders, according to /13/

It can be seen, that the applied load is caused by the rod chamber pressure p_B, since the piston chamber pressure p_A remains at approximately 20 bar during the entire cycle. The stable chamber pressure also shows, that the load units flow rate supply is satisfactory for the flow rate demand resulting from the load cycle and the movement of the test cylinder. A closer look at Figure 14 furthermore shows, that in the time ranges 6 s to 9 s, 11 s to 12 s, 17 s to 21 s and 28 s to 30 s, the piston chamber pressure drops below the minimum pressure of 20 bar. In those time ranges, the test cylinder always extends, thus the load cylinders require additional oil from the power supply of the load unit. Nevertheless, the pressure deviations of the rod chamber are always smaller than ± 5 bar and thus no cavitation occurs.

Figure 15 depicts the pressure and position progression of the test cylinder. Durations of active and passive loads are highlighted: white background color indicates passive, yellow background color indicates active loads.

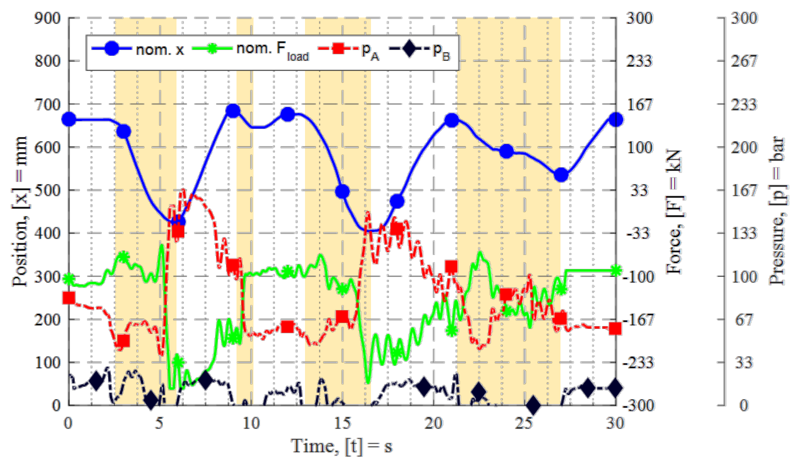

Figure 15: Pressure Trajectory of Piston (p_A) and Rod (p_B) Chamber of Test Cylinder, according to /13/

Like on the real machine, active loads only occur during retraction of the (boom) test cylinder. The simulation also shows, that at some moments during the work cycle, the rod chamber pressure drops to values near or approximately 0 bars, which means cavitation in the cylinder or at least undersupply by the system. Those undersupply situations need to be dealt with before the system can be examined on a test rig. If not, the test rig or components could be damaged.

Summing up, the simulation results show, that the hydraulic load unit, developed by Mobima / KIT, can apply a dynamic and variable load progression to a test cylinder. Passive and active loads can be applied independently of the direction of movement. By using the hydraulic load unit, more thorough and reliable assessments can be conducted on a system test rig. Also aspects of system safety and safety critical scenarios can be examined and analyzed on the test rig in an early state of a project.

5 Summary and Conclusion

Test rig experiments can only help to assess a limited number of parameters. Especially in relation to real life stress, the efforts are very high. For hydraulic actuators, active and passive loads should always be taken into account, since they usually occur during operation. In this paper, a hydraulic load unit was introduced, which allows for the application of dynamic and direction variable active and passive loads on a test cylinder with a high position and load accuracy.

The hydraulic load unit was designed with a high grade of flexibility and modularity and thus can be used for a vast range of experiments. By changing cylinders and / or valve components, the power range of the load unit can be adapted easily. In the current configuration, the load unit has a maximum extension velocity of $v_{out} = 0.184$ m/s, a maximum retraction velocity of $v_{in} = 0.109$ m/s and can apply loads in extension direction of up to $F_{load} = 235$ kN and in retraction direction of up to $F_{load} = 400$ kN.

A simulation of the hydraulic load unit stressing a test system showed, that the developed control concept is able to reproduce position and load trajectories at the test cylinder with an accuracy of nearly 92 %. Furthermore, the load unit system can be operated without cavitation and with reduced energy losses.

The hydraulic load unit can be used to examine hydraulic systems with linear motors in a more realistic test scenario. Apart from static and dynamic analysis, also safety aspects resulting of changes in loads can be addressed on the test rig, without having a full scale prototype available. Thus, valuable and important information can be gathered about the test system and its components in a very early state of the product development process.

Nomenclature

Variable	***Description***	***Unit***
A	Cylinder Port A	
B	Cylinder Port B	
CLCS	Closed-Loop Control System	
CLPC	Closed-Loop Pressure Control	
CLFRC	Closed-Loop Flow Rate Control	
CPS	Central Pressure Supply	
ECU	Electronical Control Unit	
KIT	Karlsruhe Institute of Technology	
LC	Load Case	
Mobima	Chair of Mobile Machines	
PDV	Proportional Directional Valve	

PPRV	Proportional Pressure Relieve Valve	
$A_{A/B}$	Pressurized Area of the Cylinder	
$e(t)$	Control Deviation	[bar]
F_{load}	Load Specification	[kN]
F_p	Force at Cylinder Port A/B	[kN]
F_{res}	Resulting Load Force	[kN]
ΔF_{max}	Maximum Force Deviation	[kN]
$\Delta F_{\emptyset}$	Average Force Deviation	[kN]
k	Safety Factor	[1]
$LC(t)$	Load Case Signal	[-]
p	Pressure	[bar]
$p_{current}(t)$	Measured Pressure	[bar]
$p_{set}(t)$	Predicted Pressure Signal	[bar]
Δp	Pressure Deviation	[bar]
Q	Flow Rate	[l/min]
$u(t)$	Controller Output Value	[mA]
$v_{in,out}$	Maximum Velocity of the Cylinder's Piston	[m/s]
x	Movement Direction	[mm]
$y(t)$	Control Signal	[mA]
$y_{set}(t)$	Nominal Value	[mA]

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