Systematic Data Analysis for Optimal System Design

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As part of the trend towards greater digitalization the number of sensors installed in mobile machinery is increasing each year. OEMs are consequently now capable of collecting large amounts of component measurement data, which they unfortunately do not have time to analyze or are not capable of interpreting. This is quite a pity, because when used in the right way such information can be used to develop a much better understanding of the machine and to develop new systems with lower fuel consumption and improved performance. The following paper introduces an approach used at Linde Hydraulics to analyse and assess large amounts of data with the goal of systematically identifying potential and designing new and improved hydraulic systems.

Keywords: Mobile hydraulics, excavators, data analysis, system optimization
Target audience: Mobile machine manufacturers

1 Introduction

When an OEM decides to bring a new excavator to the market, the design of the machine is usually based on that of the previous generation. Rarely, is such a development started from scratch. In order to decide which changes should be made to the hydraulic system, OEMs need a fast and systematic approach to not only analyze but also optimize the hydraulics. Judging the effect of small changes on performance and fuel consumption is not simple, as the physical relations are all highly nonlinear /1,2/. Such effects are even more pronounced when the interfaces between the individual subsystems are changed, for example the diesel engine or the kinematics. Performing real-life tests with different machine setups is far too time and cost intensive. Only an approach based on data analysis coupled with system simulations is feasible. This paper presents an overview of the approach used at Linde Hydraulics to quickly and effectively help OEMs find and define a hydraulic system that meets their requirements.

The approach is shown in Figure 1 and can be summarized as follows. To begin with, measurements of the typical duty cycles are collected. The measurement data must include hydraulic actuator pressures, actuator displacements (cylinder translation and swing rotation), pump pressures and the joystick commands. Using this data, the flow of power through the machine and the location and extent of the individual loss mechanisms are determined and visualized in a series of histograms, allowing a more systematic and intuitive interpretation of the data. This analysis is followed by a simplified system simulation of the machine. A major advantage of the simulation is that it provides access to pressures and flow rates throughout the whole machine that are not included in the measurement data. With this additional information it is possible to systematically change individual components and test different system configurations.

The paper begins by introducing the reference machine as well as the dataset used for the analysis and then goes on to explain how the measurement data is post processed and interpreted. This is followed by a discussion of the simulation approach used to create a model of the reference system, which will serve as a benchmark for all further steps. The ability of this model is then demonstrated by using it to evaluate the fuel consumption of an optimized hydraulic system and compare it to the reference system.

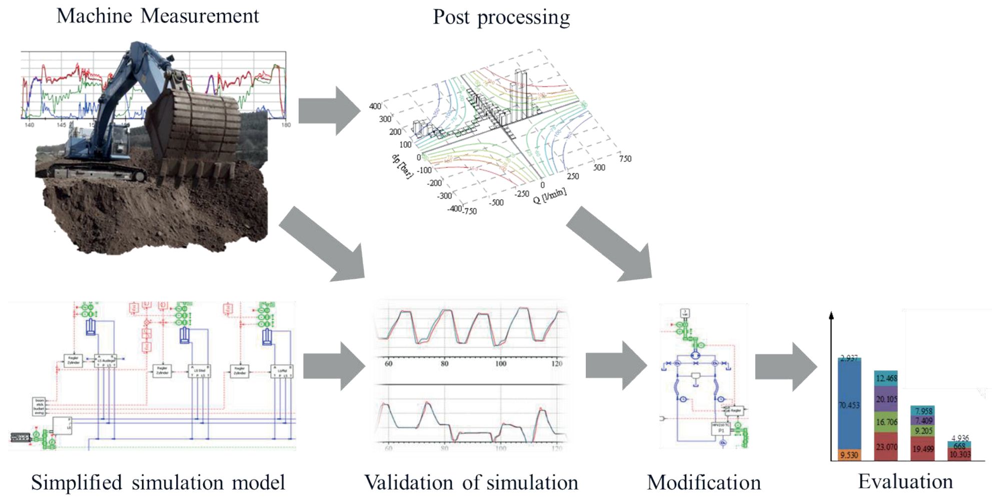


Figure 1: Approach based on data analysis and simplified system simulation

2 Example Dataset

To illustrate the approach, exemplary measurement data obtained from a 36 t crawler excavator with a single circuit Linde Load Sensing Control (LSC) hydraulic system are analyzed /4/. The hydraulic setup of the machine's implement system is shown in Figure 2. The cylinders (boom, arm and bucket) are controlled using valves with downstream pressure compensators. To ensure the swing is always supplied with oil an upstream compensator is used for this actuator. For clarity the figure does not show the compensators.

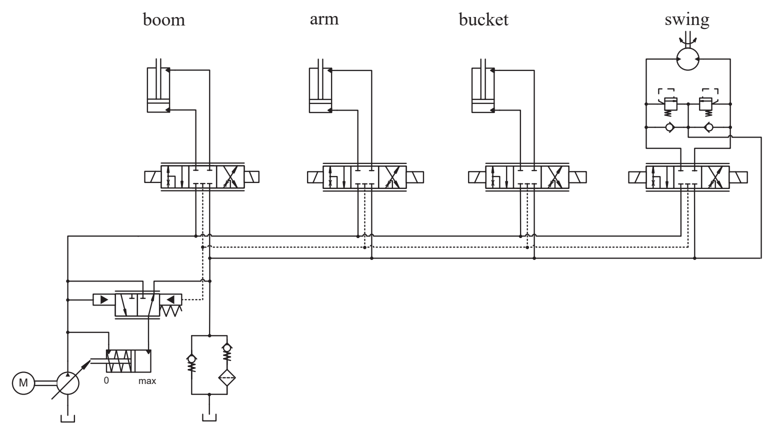


Figure 2: Hydraulic schematic

Although this machine is used for a variety of tasks, the paper will focus on data obtained from measurements of a 90° dig and dump cycle. An extract of the data is shown in Figure 3. The data includes measurements of the pump pressure, load sensing pressure, actuator chamber pressures as well as actuator displacements. Due to the large number of signals and their dynamic nature, evaluating system efficiency by just looking at the data as a function of time proves difficult /2,3/. In order to interpret the measurements and draw meaningful conclusions, the data must be postprocessed.

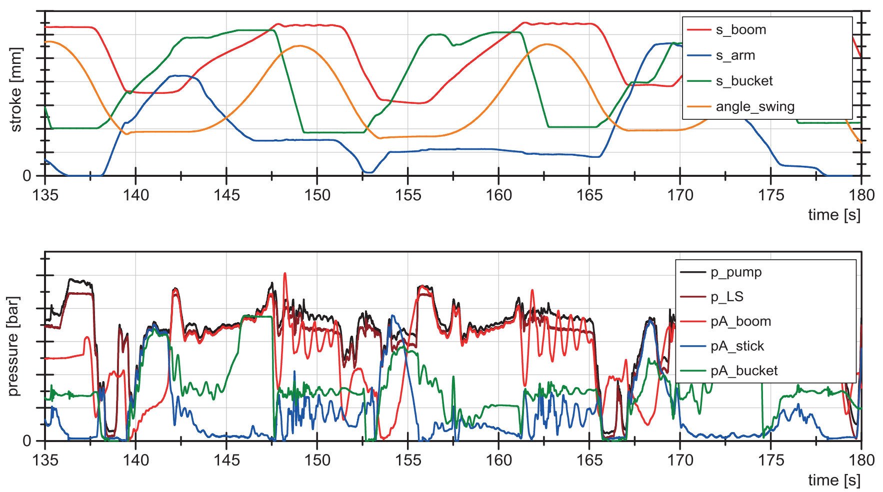
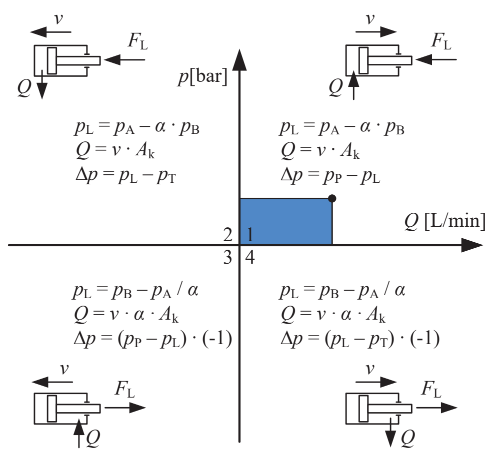


Figure 3: Extract of measurement data

3 Post Processing of Measurement Data

In a single circuit valve controlled hydraulic system, the major source of losses is the throttling across the metering edges due to differences between the supply pressure and load pressure of the individual actuators. By post-processing the measurement data the extent of these throttling losses can be visualized and quantified. The approach is explained in Figure 4. Instead of looking at the data as a function of time, the state of each actuator is described by showing the load pressure p_L as a function of load flow Q. The blue area enclosed by any point in the plane describes the amount of power, $P = p_L\, Q$, required to move an actuator or the amount of power that an actuator can release back into the hydraulic system. In quadrants one and three, the actuator has to be supplied with flow in order to overcome the load force, which opposes the direction of motion. In contrast, during operation in quadrants two and four the load force assists the motion and pressurized oil can be redirected back into the system.



with $\alpha = A_{ring} / A_{head}$ for cylinder, $\alpha = 1$ for swing

Figure 4: Definition of load pressure and load flow

To quantify the throttling losses Δp that occur when supplying an actuator with flow in quadrants 1 and 3, the difference between the pump pressure p_P and load pressure p_L has to be considered. The grey area shown in Figure 5 shows the power dissipated due to throttling. In quadrants 2 and 4, Δp is defined differently and describes the pressure drop between the actuator and tank, not between the pump and actuator. The grey area in these two quadrants is the recoverable energy that is throttled across the tank metering edge.

Due to their two dimensional nature, Figures (a) and (b) cannot be used to judge how often each operating point occurs. A third temporal dimension is, therefore, added to the diagram. The resulting columns in the histogram illustrate the frequency of occurrence of the individual points of operation, see Figure 5 (c).

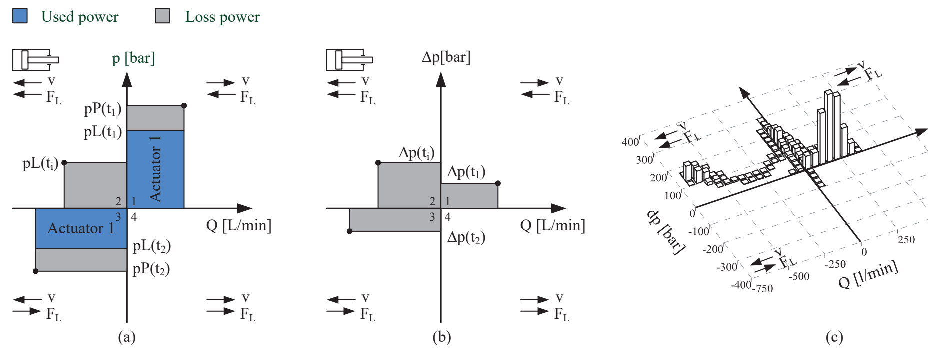


Figure 5: Analysis of load pressures and throttling losses

Using this procedure, measurement data from a dig and dump cycle is analyzed. Figure 6 shows the resulting histogram for the boom actuator. In order to judge the magnitude of the losses, constant power hyperbolas are added to the diagram. The losses occurring in quadrant 1 are all below 25 kW, indicating that the boom lifting motion is quite efficient. A look at quadrant 2 indicates that approximately 150 kW of power are dissipated across the tank metering edge during boom lowering. This recoverable energy comes in the form of a medium pressure, between 100 and 200 bar, and a high flow rate, more than 500 l/min, making it technically challenging to recover and reuse boom potential energy.

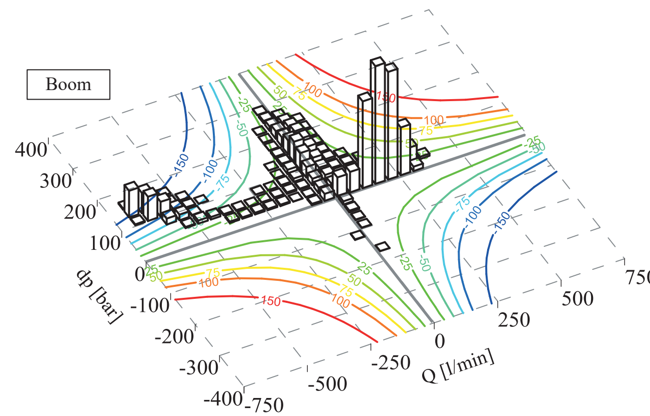


Figure 6: Throttling loss histogram of boom actuator

The histograms for all the actuators are shown in Figure 7. Regions with high losses are highlighted in red. These include swing braking in quadrants 2 and 4 and as already mentioned boom lowering in quadrant 2. The bottom left diagram indicates that up to 150 kW of power are throttled during bucket dump motion. This is due to the large flow of approximately 500 L/min passing through the A-T metering edge.

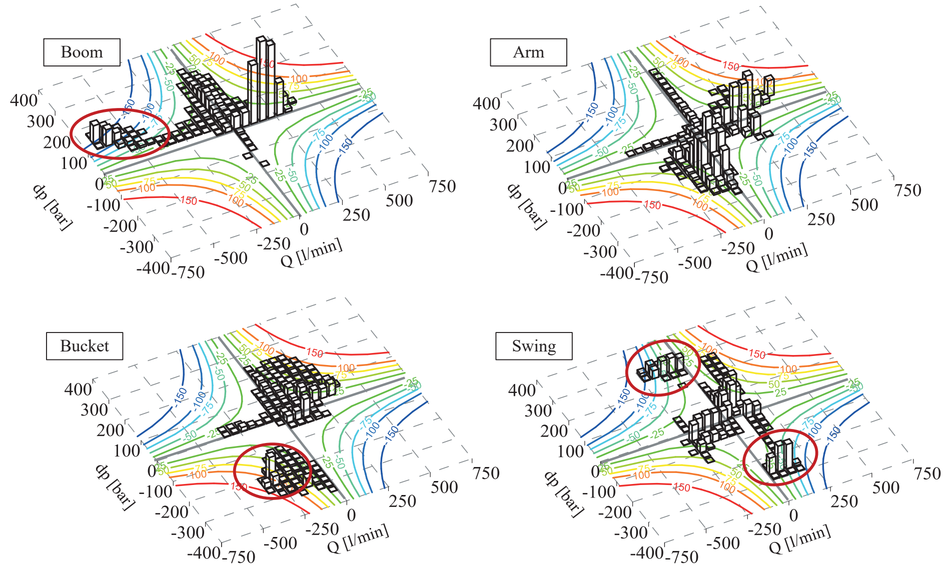


Figure 7: Delta p histograms of all actuators

4 Setup and Validation of Simplified Simulation Model

In the next step, a simulation model of the machine is created in AMESim. As shown in Figure 8, the model includes a diesel engine, hydraulic pump, main control valve and hydraulic actuators (bucket, arm, boom and swing).

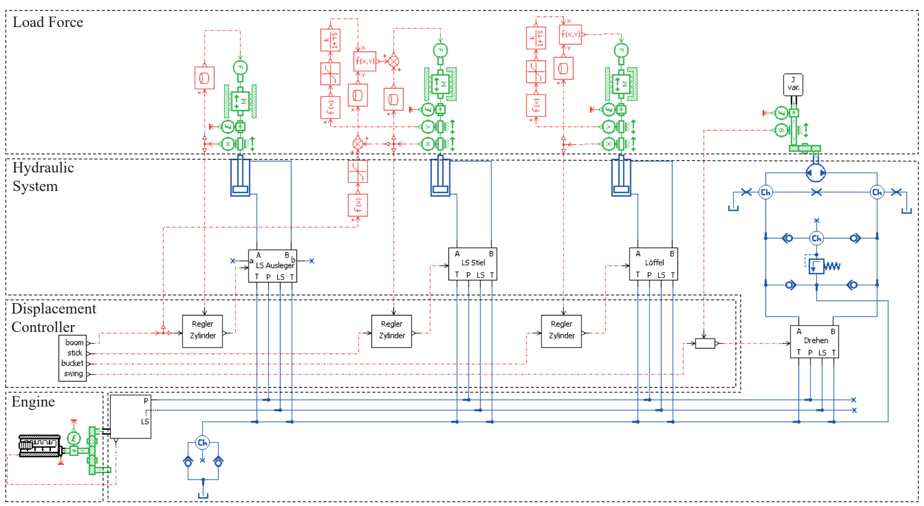


Figure 8: Simulation model of machine in AMESim

To replicate the actuator motion in the measurements usually only the measured pilot pressures are applied to the hydraulic valves in the model. This method unfortunately usually leads to large discrepancies between simulation and measurements, as it is difficult to capture the exact flow characteristics of each valve metering edge. To ensure the actuators closely follow the same trajectories as in the measurements, a closed loop displacement controller takes on the role of the "machine operator". Figure 9 illustrates the ability of such a controller to accurately follow the measured actuator displacements.

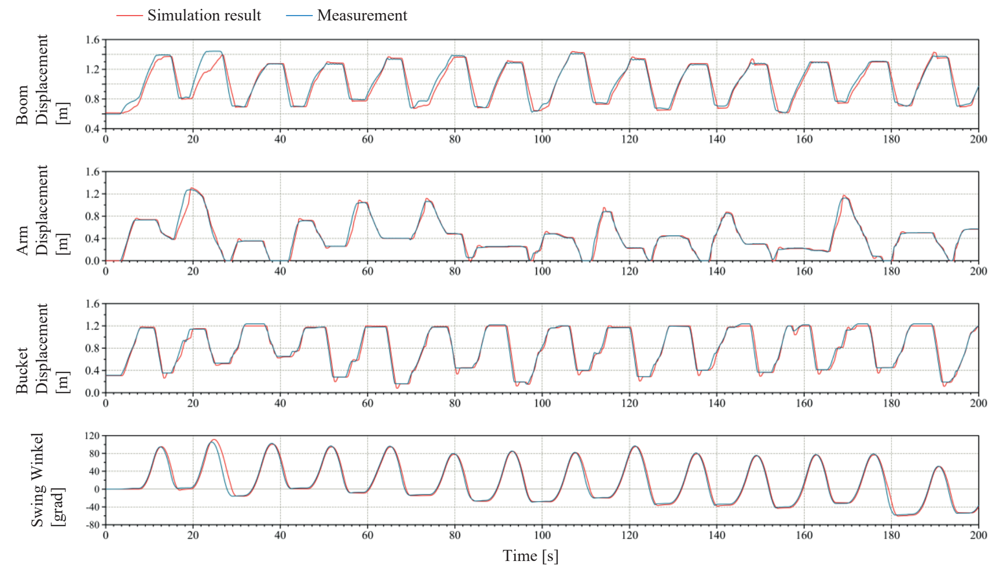


Figure 9: Validation of the closed loop displacement controller

A closer look at Figure 8 reveals that only the swing drive is connected to an inertial mass as in the real machine. To avoid long simulation times, no kinematic structure with inertia is used for the other actuators. The load is artificially created by applying the measured load forces to the individual actuators externally. Instead of applying the force only as a function of time, $F_L(t)$, the measurement data is used to create an averaged look-up table that applies the force as a function of actuator displacement, $F_L(x)$. This ensures that the correct force is applied even when the simulated and measured displacements do not match exactly. A closer analysis of the measurement data reveals that the arm load force is also a function of the boom actuator position, therefore a combination of two lookup tables is used for the arm $F_L(x_{Boom}, x_{Arm})$. A comparison of the measured and simulated load forces is shown in Figure 10.

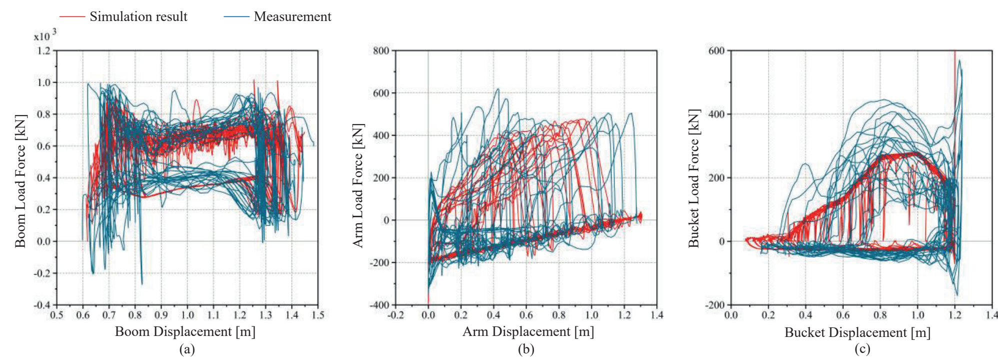


Figure 10: Comparison of the measured and simulated load forces

Figure 11 shows the histograms obtained from the simulation model. These show good agreement with the measurement results in Figure 7. The model is therefore validated and can be used to analyze and answer a whole range of questions. In contrast to the measurements, it is now possible to look at the pressures and flow rates throughout the whole system and not only at the locations where sensors are installed. The validated model is used as a reference to judge all further optimizations and system architectures.

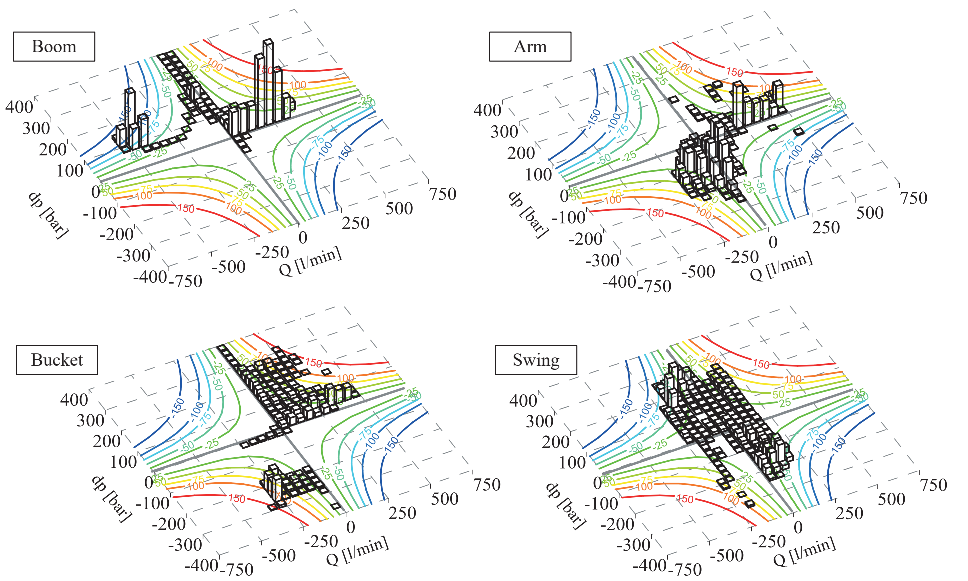


Figure 11: Simulated delta p histograms of all actuators

5 Optimized System Architecture

The simulation model can also be used to judge the energy saving potential and performance of new system configurations. The circuit shown in Figure 12 is used as an example.

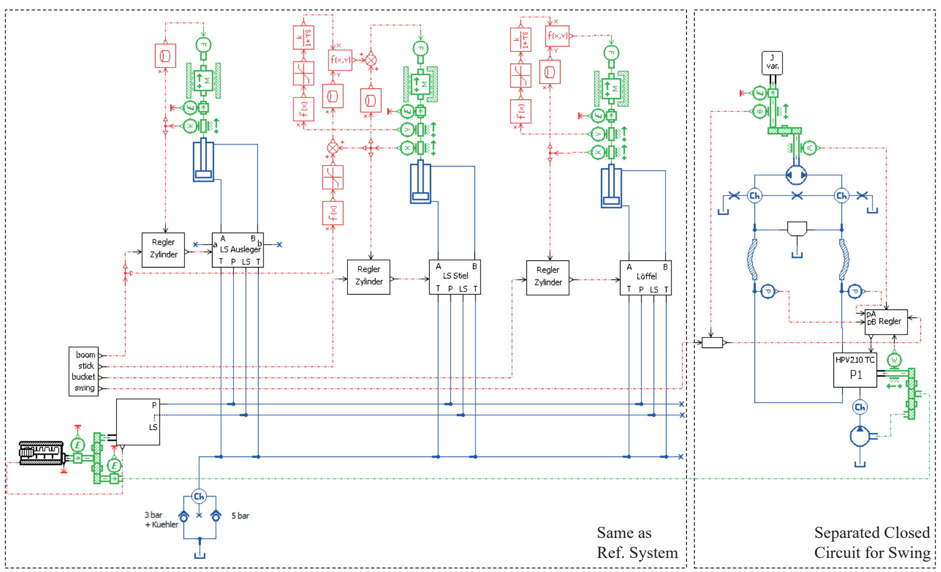


Figure 12: 1,5 circuit load sensing system

The setup is referred to as a 1,5 circuit LS system, which is basically a single circuit load sensing circuit with a closed loop swing. The advantages of the modified circuit are that a separate closed loop pump supplies the swing actuator with flow, thereby reducing throttling losses and enabling energy recovery across the engine shaft during swing braking.

Histograms of the optimized system are shown in Figure 13. Compared to Figure 11, the first major difference is the distribution of operating points for the swing. As is expected, no throttling occurs in swing quadrants two and four.

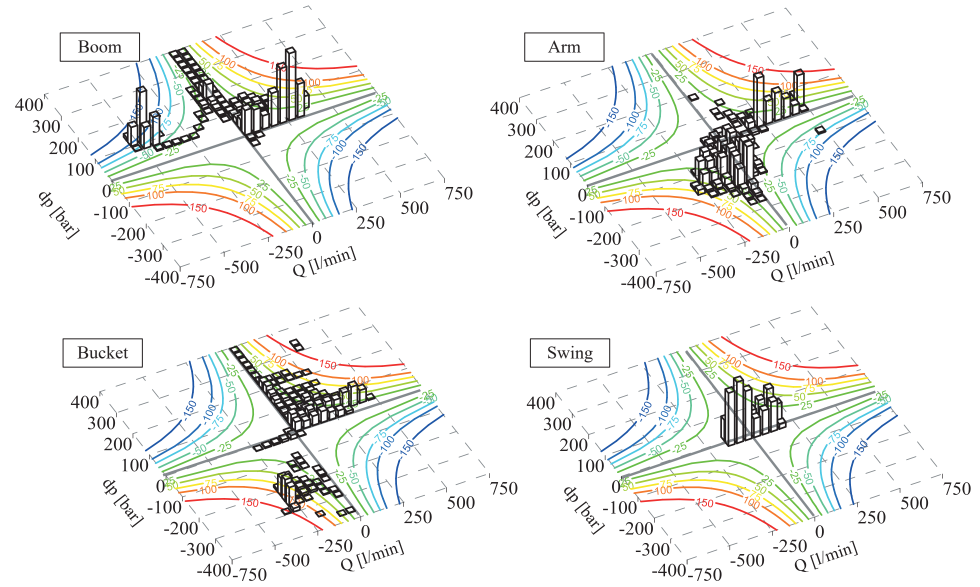


Figure 13: Simulated delta p histograms of optimized system

6 Simulation Results

An overview of the energy distribution of the reference system for a 90° dig and dump cycle is shown in Figure 14 (a). More than 260.000 kWs of energy in the form of diesel fuel enter the system. Due to the low efficiency of the internal combustion engine, only 33 % of this energy is converted into mechanical power (86.371 kWs). Another 4 % (11.967 kWs) is lost as the pump converts the incoming mechanical power across the shaft to hydraulic power, which is then sent to the main control valve. After throttling a further 12 % is dissipated, meaning that only 43.089 kWs actually reach the actuators. Approximately 36 % of this actuator energy never really leaves the machine because it is used to raise the boom and accelerate the swing, meaning that it can in fact be recovered during boom lowering and swing braking. In total, a mere 27.468 kWs or 11 % of the total incoming diesel energy is actually used to perform work on the surroundings and objects in contact with the machine.

Figure 14 (b) illustrates a more detailed distribution of the hydraulic power among the individual actuators. The main control valve delivers the most energy to the boom and bucket actuators, both approximately 21.000 kWs, followed by the swing and arm, both around 15.000 kWs. Very little energy is lost across the boom valve, as its load pressure is usually the highest in the system. Due to their lower pressure levels, only half the energy passing through the swing and arm valves actually reaches the actuator. Worst of all, only about a third of energy passing through the bucket spool actually reaches the bucket cylinder. This is one of the major disadvantages of a single circuit system, in which only one supply pressure is available.

A look at the negative actuator energy column in the figure reveals that approximately half of the energy used to raise the boom and more than 60 % of the energy used to accelerate the swing can be recovered during lowering and braking. Because the reference system is not capable of recovery or recuperation, this energy is dissipated as heat across the valves.

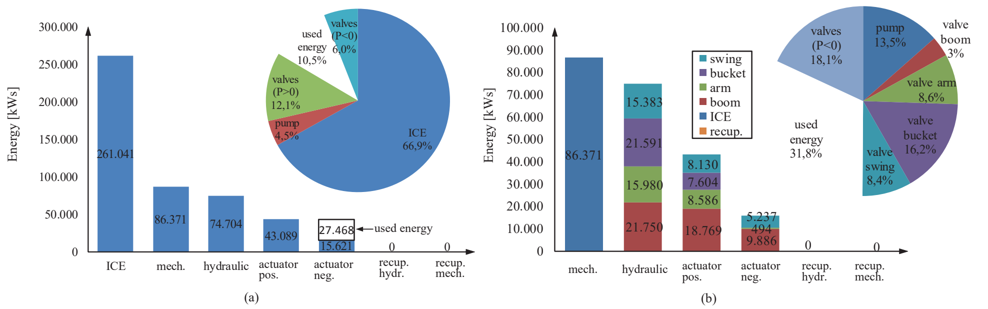


Figure 14: (a) Energy flow through reference system; (b) Energy consumption of individual actuators

Figure 15 shows the analysis for the optimized system with a closed loop swing architecture. A first look confirms that for the same cycle the optimized system consumes less fuel than the reference system, 247.575 kWs compared to 261.041 kWs. The total system efficiency increases from 10,5 % to 11,2 %. Using the closed loop swing 3.854 kWs of the available 4.936 kWs are recuperated during braking. Due to the efficiency losses in the displacement unit, only 2.937 kWs of mechanical energy actually reach the shaft. Because the other three actuators (boom, arm and bucket) are still connected to one pump, the throttling losses are comparable to those of the reference system.

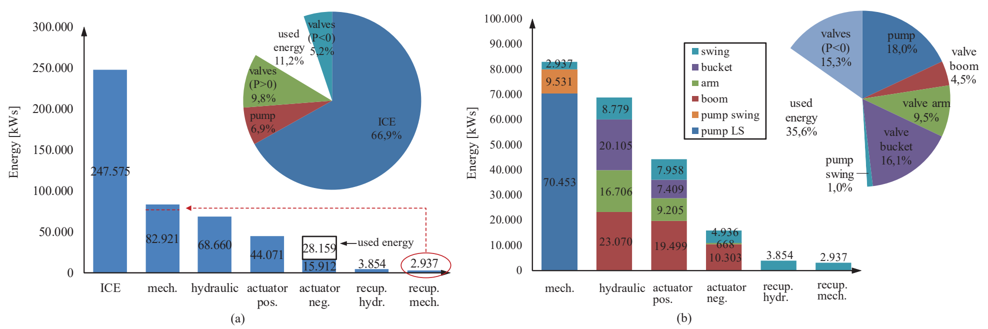


Figure 15: (a) Energy flow through optimized system; (b) Energy consumption of individual actuators

As the performed work for two the configurations is not identical, it makes sense to compare the relative fuel consumption [mL/kWs] and not only the absolute fuel consumption. These values are summarized in Table 1. The optimized circuit lowers the relative fuel consumption from 0,254 mL/kWs down to 0,235 mL/kWs, which is an improvement of 7,5 %.

	Reference system	Separated closed circuit for swing
Performed work [kWs]	27.468	28.159
Consumed fuel [L]	6,986	6,626
Rel. consumption [mL/kWs]	0,254	0,235

Table 1: Efficiency Comparison

7 Summary and Conclusion

The amount of machine and measurement data available to OEMs and their suppliers is increasing at a rapid rate. Knowing how to use and manipulate this data will prove valuable as it gives engineers the opportunity to develop a whole new level of understanding for the systems they design. The following paper has focused on the data analysis approach developed at Linde Hydraulics aimed at serving the needs of their customers more efficiently. Although a 36 t excavator was used to illustrate the methodology, the same tools can be used to optimize the hydraulic systems in a wide range of mobile machinery.

Nomenclature

Variable	*Description*	*Unit*
p_A	Actuator Chamber A pressure	[bar]
p_B	Actuator Chamber B pressure	[bar]
p_L	Actuator Load Pressure	[bar]
p_{LS}	Load Sensing Pressure	[bar]
p_P	Pump Pressure	[bar]
v	Actuator Speed	[m/s]
α	Actuator Area Ratio	[-]
Δp	Throttling Losses	[bar]

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