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Master’s Programme in Mechanical Engineering

Simulation-Based Study on Energy Efficiency of Different Hydraulic Actuators in Forestry Forwarder

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Abstract

This master's thesis focuses on the development of a dynamic hydraulic system model for a forestry machine using the Simscape simulation environment. The work is conducted in parallel with another thesis that has created a multibody model of the same machine. The objective of this thesis is to complement that mechanical model by providing a comprehensive simulation model of the machine's hydraulic system.

Two different systems are modelled. They are a load sensing hydraulic system and a direct drive hydraulics system. The objective is to create systems that work precise with the multibody model movements. The systems are then compared by their energy usage and energy efficiency. The modelling work concentrates on a load sensing hydraulic system, where multiple actuators are operated simultaneously under varying load conditions. The system's aim is to implement a control strategy that maintains constant actuator speeds regardless of load variations, while also minimizing the energy consumption of the system. Direct drive hydraulics system is modelled the simplest way. Closed-loop PID controllers are used to regulate the motion of the individual cylinders.

This thesis lays the foundation for simulation-based design and analysis of forestry machine hydraulics. The developed Simscape model serves as a platform for future research in control strategies, energy efficiency, and flow management. It also enables the study of system dynamics in various operating scenarios and facilitates the evaluation of component-level impacts on overall system performance.

Keywords Hydraulics, simulation, Simscape, forestry forwarder, load sensing system, direct drive hydraulics, PID-control, energy efficiency

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Tiivistelmä

Tässä diplomityössä kehitetään metsäkoneen hydraulijärjestelmän dynaaminen malli käyttäen Simscape-ohjelmistoa. Työ liittyy rinnakkaiseen diplomityöhön, jossa on laadittu metsäkoneen monikappalemalli, ja tämän työn tavoitteena on täydentää kokonaisuutta tuottamalla siihen hydraulijärjestelmän mallinnus. Hydraulimalli mahdollistaa metsäkoneen toimintojen, kuten nostolaitteiden ja muiden hydraulisesti ohjattujen komponenttien, käyttäytymisen analysoinnin ja optimoinnin simulaatioympäristössä.

Työssä mallinnetaan kaksi erilaista järjestelmää, jotka ovat kuormantunteva hydraulijärjestelmä sekä suorakäyttöhydraulijärjestelmä. Tavoitteena on luoda mallinnukset, jotka toimivat monikappalemallin liikkeiden mukaisesti mahdollisimman tarkasti. Näitä järjestelmiä vertaillaan keskenään niiden energiankäytön ja energiatehokkuuden mukaan. Mallinnuksessa keskitytään erityisesti kuormantuntevaan hydraulijärjestelmään, jossa useita sylintereitä ohjataan samanaikaisesti vaihtelevissa kuormitustilanteissa. Järjestelmän tavoitteena on toteuttaa ohjausratkaisu, joka pitää sylinterien nopeudet vakiona eri kuormituksilla sekä minimoi järjestelmän energiankulutuksen. Suorakäyttöhydrauliikan mallinnus luodaan yksinkertaisimmalla tavalla. Sylinterien ohjauksessa hyödynnetään suljetun säätösilmukan PID-säätimiä.

Työ tarjoaa lähtökohdat metsäkoneen hydraulijärjestelmän simulointipohjaiselle suunnittelulle ja säätöstrategioiden kehitykselle. Simscape-pohjainen malli luo perustan järjestelmän jatkokehitykselle, kuten ohjausalgoritmien optimoinnille ja virtauksen hallintaan liittyville tutkimuksille. Mallin avulla voidaan myös tutkia järjestelmän käyttäytymistä erilaisissa toimintatilanteissa ja arvioida eri komponenttien vaikutuksia kokonaisuuteen.

Avainsanat hydrauliikka, hydraulijärjestelmä, metsäkone, kuormatraktori, simulointi, energiatehokkuus

Table of contents

Preface and acknowledgements

This thesis was the final assignment made for the university to complete my studies. Creating the simulation models was successful and highly engaging. I want to thank my thesis advisor Jyrki for his continuous encouragement and professional advice during my thesis work. I have received priceless advice about structure, contents and professional language. Many thanks to M.Sc. (Tech.) Henri Wahlman for his earlier thesis work where my thesis could be continued fluently. Finally, I would like to thank my family and friends, who have supported and encouraged me throughout my studies so far.

Espoo, Finland, 21 November 2025
Topias Matero

Acronyms

CAD	Computer Aided Design
CAN	Controller Area Network
CP	Control Piston
CPM	CAN bus-Controlled Pilot Module
CTL	Cut To Length
DDH	Direct Drive Hydraulics
DS	Down Sample
ECU	Electronic Control Unit
LHV	Load Holding Valve
LS	Load Sensing
MB	Multibody
MBSE	Model-Based Systems Engineering
PID	Proportional-Integral-Derivative (controller)
PRV	Pressure Relief Valve

1 Introduction

Finland has a vast area of forests, and the forestry industry is a major sector providing work for over 100 000 people. Forestry is carried out using heavy-duty special vehicles that are agile, powerful and robust. The industry will benefit from improving these machines. [1]

This thesis deals with a forwarder. The main equipment used in such machine consists of diesel and electric motors, hydraulic actuators and valves. These parts allow the forwarder to rotate, bend and extend the main arm of the machine and to pick up and move wood. It is important to do accurate modelling in machine development, control design and energy efficiency. The usage of simulation tools, such as Simscape, has become an inevitable part of the early development. Having a better understanding of the system in development makes optimization and understanding complex hydraulic behaviour possible.

The forwarder usage cost is calculated by overall cost per cubic meter delivered to roadside. There are a few reasons why a forwarder should be developed to perform better. Efficiency of a forwarder should be made greater to reduce fuel consumption. Reducing fuel consumption leads to many benefits including economical and ecological aspects. Designing and simulation models are important processes to make the overall machine more efficient, more affordable, more reliable, more suitable and stronger. [2]

A parallel master's thesis developed a Multibody (MB) model of a forest machinery. The model of the forest machine is inspired by Komatsu 875. This is a heavy-duty industrial forwarder. It is designed to produce various heavy-weight wood log forest products and timber gathering works. [3] The parallel thesis adds a body to this thesis' hydraulic system, and the simulation movement will be more intuitive experience to understand with the visualization.

In this thesis two hydraulic simulation models of a multibody model are built. A dynamic model is built using a load sensing (LS) element, PID-controllers and a single pump. A direct drive hydraulics (DDH) model is simpler, consisting of a pump, valve and a PID for each hydraulic actuator. First, the chosen Simscape model is shown and explained, alternative model is presented, and the two models are compared as in simplicity, energy efficiency and simulation durability.

This thesis aims to generate two different hydraulic models that can be compared for their energy usage, simplicity and other merits. The designing work essentially includes hydraulic systems modelling. The two simulation models are used as the foundation for the different systems comparisons and performance tests. This thesis focuses on a general model of a forest machine. The focus is on modelling and simulating the system and not implementing anything physical. Control logic is limited to a basic-level PID which is used in the control systems.

This thesis continues as follows. The second chapter performs a literature review of hydraulics, forest machine and modelling approaches. The third chapter focuses on research material and methods. The fourth chapter handles the results. The fifth chapter reviews the successfulness of the work and presents various possibilities for further development. The final chapter presents a conclusion.

2 Literature review

This chapter gives information about already in use forwarders and their functions. Basic relevant hydraulic components are then presented. Lastly, simulation modelling approaches and methods are discussed.

Logging machine is a general form of a heavy machinery designed to harvest and transport wood. Along forwarders – the subject vehicle of the thesis – there are harvesters, skiddlers, feller brunchers, log loaders and -stackers, shovel loggers and mulchers used in different logging tasks. Forwarders are logging vehicles that are designed to carry and move logs from the forest to roadside. [4]

Forwarder is used to pick up and move the logs that a harvester has already processed. The harvester's processing includes harvesting the tree, removing its branches, cutting it to a desired length (cut to length, CTL) and piling logs up ready for a forwarder to pick up. Harvesters and forwarders are now a day fully mechanized, which accentuates the need to enhance their efficiency. [5]

2.1 Forwarder logging machine

A laden large forwarder can weight over 50 000 kg when carrying up to 25 000 kg of logs and are typically eight-wheelers [2]. Forwarders are often rated by their payload loading capacity. They are divided into light, medium and heavy forwarders, being under 10 t, 10 – 14 t and over 14 t respectively [6].

Forwarder's working principle can be divided into three subsystems which are presented in Figure 1. The subsystems are categorized as powertrain, hydraulic and electrical systems. They all have different functions to ensure forwarders full operational functions. This thesis concentrates on the Hydraulic Subsystem, but the other sections are briefly discussed in this chapter.

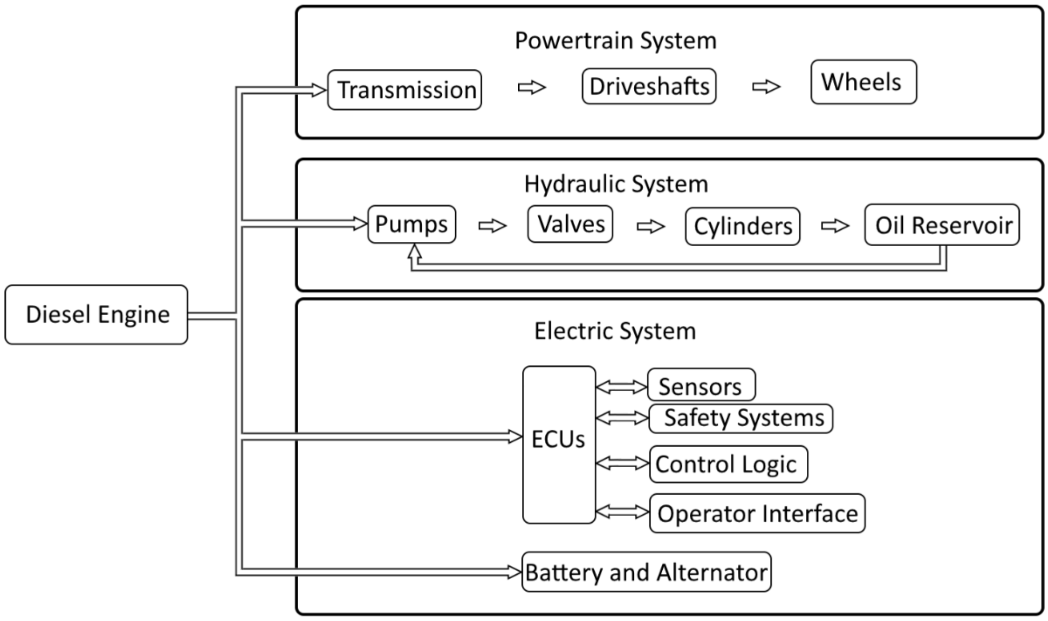


Figure 1. Forwarders general subsystems.

A diesel engine is currently the most used source of power, but fully electric forwarders are already developed. Ponsse in collaboration with Epec has built a fully electric forwarder PONSSE EV1 as a concept which is not yet on consumer production. This is a more sustainable harvesting solution compared to traditional diesel-powered machines. Epec has developed a Epec Flow electromobility system solution through simulations, where they have optimized energy consumption, productivity and usability. Currently the forwarder batteries can be charged with a separate combustion engine called Range Extender. [7] In a diesel forwarder, the power from the engine is directed with an alternator for the pump-motors. [8]

In a typical forestry forwarder, the prime mover is a turbo-charged diesel engine which connects to a transmission or hydrostatic/hydro-mechanical drive unit that distributes torque to the wheel axles and bogies. Transmission drives an 8 wheel-drive bogie system with high tractive force of up to 270 kN. [9] Driveshafts or portal axles carry the torque to the wheels, and the articulated frame and bogie design allow the machine to maintain off-road mobility on rough terrain. Research also shows that the drive system for forwarders is optimized for low speeds under load and high traction rather than high vehicle speed. [10]

In forwarding machines, the hydraulic system is critical for both travel (steering, bogie articulation) and work (boom, grapple, loading). Typically, a load sensing or proportional hydraulic pump supplies hydraulic oil under high pressure (200–350 bar) through valves to cylinders. Hydraulic cylinders provide the mechanical motion for the crane and grapple, and control valves regulate flow and direction. [11] Modern components in a hydraulic system are actively digitally controlled. The benefit of digitalization is deterministic operation of the simple components and their programmability. [12]

In many European machines, the electric system in a forwarder operates on a nominal 24 V network carrying power for lighting, instrumentation, ECUs and controls, and charging via alternator/battery. For example, the Rottne F13C model features a 24 V electrical system with 2× 115 Ah batteries and a 100 A alternator, and CAN-bus (Controller Area Network) machine control system. [13] In modern machines, ECUs are integrated to CAN-bus which allows to coordinate functions, monitor system health and support remote diagnostics [14].

Forwarder's work can be tracked as work cycles. First phase of a work cycle is travelling empty to the work zone. Second phase is the loading, where the crane begins to move and load logs into the forwarder. This phase lasts until the forwarder bunk is full. Third phase is travelling loaded, when the forwarder bunk is full and heading to the landing. In the fourth phase the crane unloads the logs from the bunk. [15]

2.2 Hydraulic components

Hydraulics are all about transferring mechanical work. Hydraulic fluid transfers a pump's shaft output to an actuator which changes the hydraulic power back to mechanical energy, for example moving a piston in a cylinder. Hydraulics allow us to produce high forces with linear or circular motion which makes it convenient to be used in moving work machines such as excavator or forwarder. Heavy machinery such as a forwarder usually has one source of power, where using tubes and pipes makes it easy to transfer power hydraulically to the actuator compared to a mechanical way of power transmission. [16]

Hoisting machines use a lot of hydraulics for their good weight to power ratio. Small actuators produce great number of forces. Hydraulic systems are part of a power transmission system, where mechanic power is transferred into hydraulic power, hydraulic power is transferred to actuator and then actuator converts hydraulic power back to mechanical power. Hydraulic power is transferred by hydraulic fluid's pressure and mass flow to an actuator. Hydraulics adds more freedom of power transmission line designing via its possibility to use flexible tubing and pipes. [16]

Hydraulic systems can be divided into open and closed systems. In an open system, a pump takes fluid from tank and pumps it into an actuator, and then the fluid flows back to the tank. In an open system it is typical that the pump spins only in one direction and the actuators are controlled by valves. Also, the system has high-pressure side from pump and low-pressure side returning to the tank. In a closed system, there is no tank inline. The hydraulic fluid returns from the actuator back to the pump. This means the system has no divided sides for high and low pressures. These systems often use a hydraulic accumulator as a power storage to transfer hydraulic power to the actuator. The benefit of using a closed system is usually its small size and weight due to lack of tank and additional valves. The actuator can be controlled simply by changing the pump's direction and speed. On the other hand, the downside will be a challenging maintenance and liability of overheating due to significantly smaller amount of hydraulic fluid compared to a tank. [17]

Common hydraulic components are pumps, motors, cylinders, a tank and various types of valves. On top of that there are different sensors used to monitor flow, pressure, temperature, force and position. This data gathered from various places is used to control the system. Traditional open system direct driven hydraulic system can be seen in Figure 2. It consists of the components of pump, tank, valve and actuator.

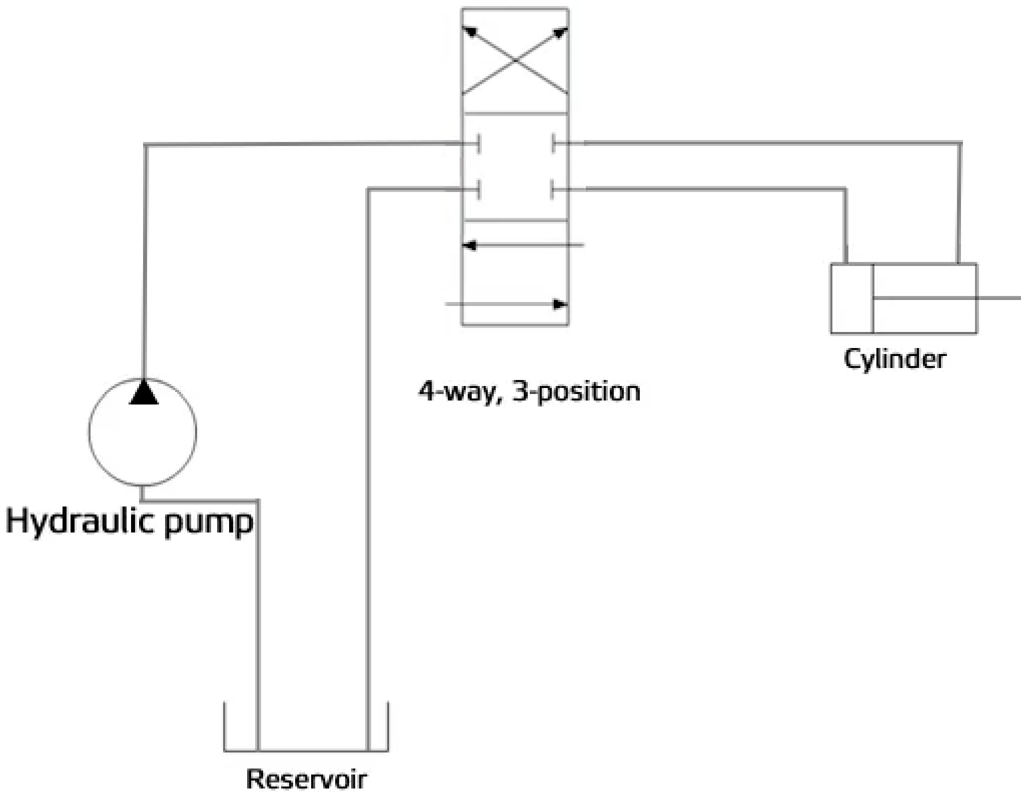


Figure 2. Drawing of a 4-way 3-position valve controlled direct drive hydraulic system for one cylinder.

The hydraulic components convert the hydraulic and mechanical power into another, as said earlier. By controlling these components, it is possible to change a diesel motor shaft power into cylinder power, which ultimately leads to a boom arm movement and potential energy. A hydraulic actuator's efficiency is defined by power. A hydraulic power P_{hyd} is defined by pressure difference p over the component times mass flow Q through it.

$$P_{hyd} = p * Q \qquad \text{eq. 1}$$

Rotating actuators such as motors' and pumps' mechanical power

$$P_{mek,p} = M * \omega \qquad \text{eq. 2}$$

is defined by moment M times shaft rotational speed ω. Linear actuators such as cylinders power can be derived as

$$P_{mek,l} = F * v \, , \qquad \text{eq. 3}$$

where F is cylinder force and v is the cylinder piston velocity. Total efficiency η_{total} of a component can then be derived by dividing power in P_{in} by power out P_{out}

$$\eta_{total} = \frac{P_{in}}{P_{out}} \, . \qquad \text{eq. 4}$$

The powers in and out can be powers from equations (1-3) depending on the component.

Next, basic formulas for calculating hydraulic motors flow [l/min]

$$q = \frac{D * n}{1000 \; * \; \eta_v} \, , \qquad \text{eq. 5}$$

torque [Nm]

$$M = \frac{D * \Delta p * \eta_{hm}}{63} \, , \qquad \text{eq. 6}$$

power [kW]

$$P = \frac{q * \Delta p * \eta_t}{600}, \qquad \text{eq. 7}$$

and overall efficiency

$$\eta_t = \eta_v * \eta_{hm}, \qquad \text{eq. 8}$$

where D is displacement [cm³/rev], n is shaft speed [rpm], η_v is volumetric efficiency, Δp is differential pressure between inlet and outlet [bar] and $\eta_{hm}$is mechanical efficiency. [18]

2.2.1 Hydraulic cylinders

A general hydraulic cylinder is a hydromechanical actuator converting hydraulic power into linear piston movement's mechanical power and force. A hydraulic cylinder consists of a piston moving in a cylinder and seals as seen in Figure 3. There are two types of cylinders: single acting and double acting. Single acting cylinders can produce force only in one direction and must be returned to zero position by an external force. Double acting cylinders can be moved and produce force in both directions. This thesis' cylinder use focuses on double-acting cylinders. The cylinder has ports for mass flow in and mass flow out of the cylinder chambers. The piston can be moved by having mass flow into a chamber and flow out from the other chamber. The seals between cylinder and piston cause leakage and friction. These preferences vary from the application.

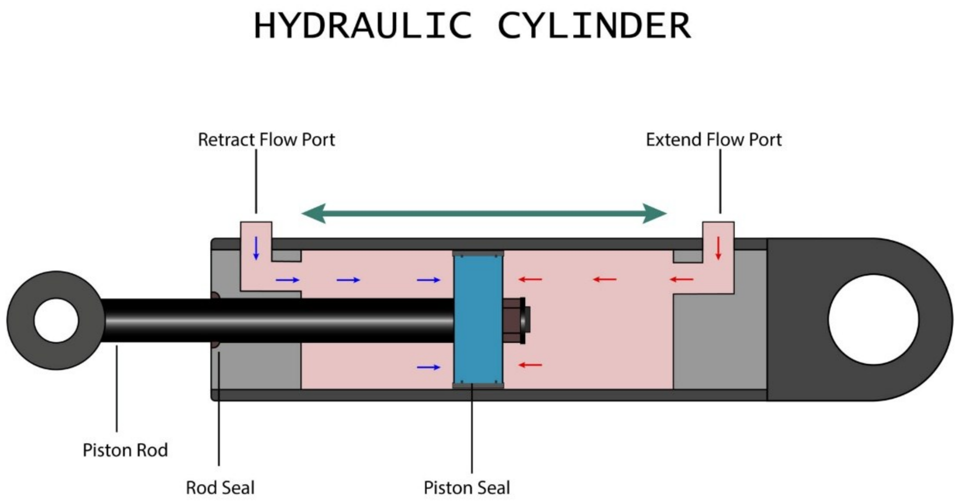


Figure 3. Simple model of a hydraulic cylinder. [19]

Cylinder force $F = pA$ is dependent of pressure over cylinder chambers and the area of the piston surface. By increasing pressure, the cylinder produces more force. The piston velocity v_{piston} depends on the fluid mass flow Q and A_{piston}. A cylinder is often a differential cylinder, because the piston rod on the other side reduces the effective area of the piston and makes the piston areas different on each side. The areas are referred as $A_{piston,A}$ on the extend flow port side and $A_{piston,B}$ on the retract port side. There are also available cylinders that have the same piston area on both sides. [16]

Cylinder velocity is dependent on mass flow directed in with a valve. To achieve a desired velocity v, mass flow

$$Q = \frac{A_{piston} * v}{\eta_{vol}} \approx A_{piston} * v, \qquad \text{eq. 9}$$

is needed, where volumetric efficiency $\eta_{vol} \approx 1$. The volumetric efficiency reflects the leakage between cylinder chamber and piston, which is often considered very minor with now a day's good piston seals. [16]

For a hydraulic actuator like a cylinder

$$F = p * A \qquad \text{eq. 10}$$

where F is force produced by the actuator [N], A is effective piston area [m²] and p is pressure acting on the area [Pa]. This way useful mechanical power will be

$$P_{useful} = F * v = (p * A) * v \quad \text{eq. 11}$$

and from using eq. 9 we get

$$P_{useful} = p * Q. \quad \text{eq. 12}$$

This shows that useful mechanical power is linearly dependent on pressure.

A symmetric cylinder means that its cylinder chambers have the same effective piston area. When $A_{piston,A} = A_{piston,B}$, the piston can be driven in a closed system fully to both ends without it beginning to cavitate. A common way of making such cylinder is to make the piston rod to go through both chambers. Such symmetric hydraulic cylinder is called double-rod cylinder and is presented in Figure 4. This type of cylinder is used in a direct drive hydraulic system presented in 3.1.

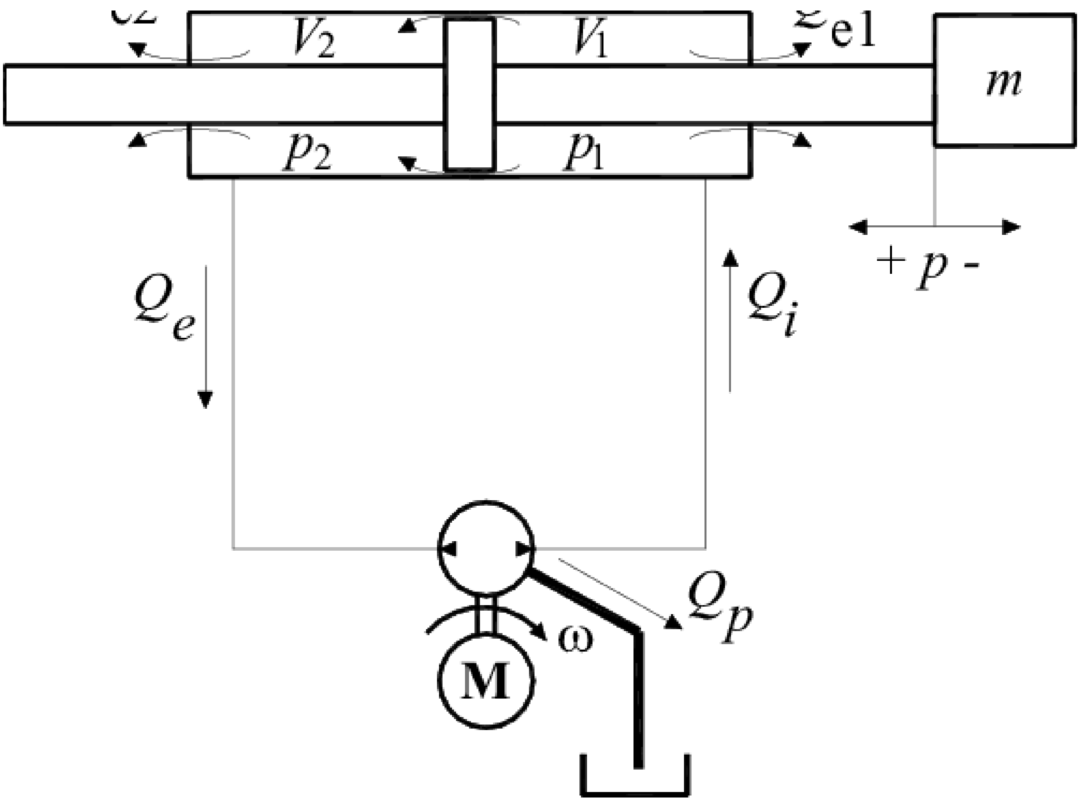


Figure 4. A simplified scheme of a symmetric double-rod hydraulic cylinder in a closed hydraulic system. [20]

2.2.2 Valves

The most used valves can be divided into directional-, pressure- and flow valves. Directional valves are used to allow hydraulic fluid mass flow direction and thus actuator's direction of movement. The simplest forms of directional valves are shut-off valve and check valve. A shut-off valve shuts the flow completely when closed and check valve always allows the mass flow only for one direction. When more demanding control is required for an application, directional control valves are used. There are many different configurations of them, which the most used are 2/2, 3/2, 4/2 and 4/3 directional control valves. First number is the number of connection ports, and the second number indicates the number of alternative valve positions available. Drawing symbols of different directional control valves are shown in Figure 5.

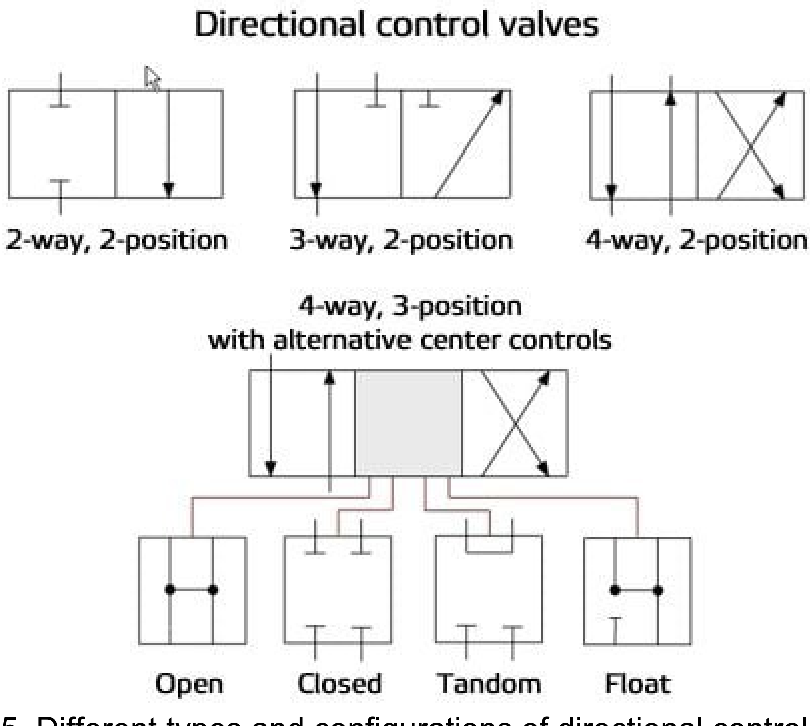


Figure 5. Different types and configurations of directional control valves. [21]

Pressure control valves are used to regulate or limit pressure within a hydraulic circuit. This way the circuit's components can be protected from damaging amounts of pressure. A pressure relief valve (PRV) among other pressure control valves is shown in Figure 6.

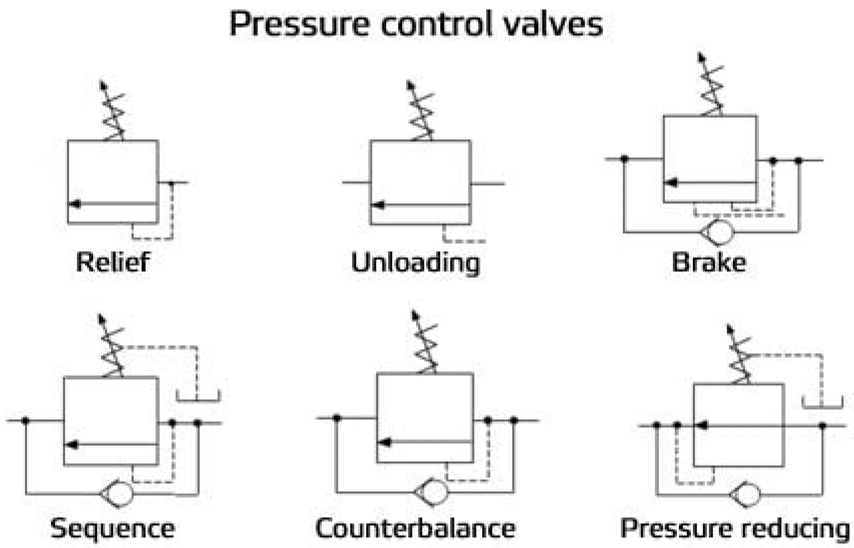


Figure 6. Pressure control valves. [21]

Flow control valves are used to limit the amount of hydraulic fluid into the circuit or actuator. Check valve among other valves is shown in Figure 7. Actuator velocity can be adjusted with controlling the mass flow. The flow control valve's working principle is to restrict flow over it, when the remaining mass flow is often directed back to tank or a reservoir. When flow is redirected back to tank, the system is not doing any mechanical work and thus producing energy waste.

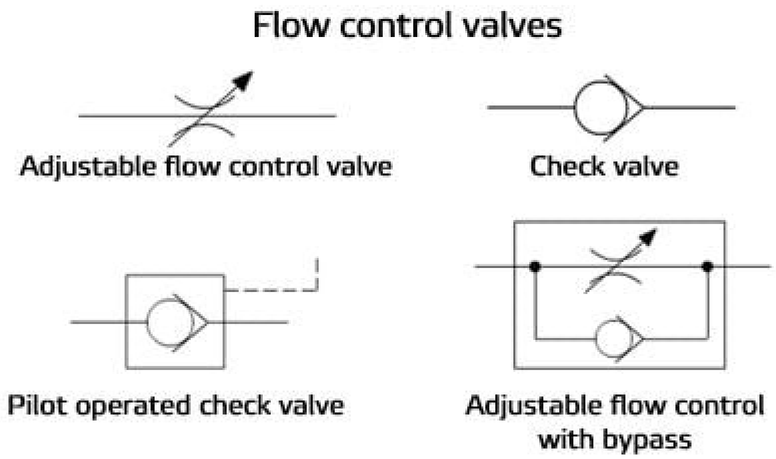


Figure 7. Flow control valves.

Energy waste in valves is caused by pressure losses which depend on mass flow through the valve, density of the fluid and its viscosity. The bigger the mass flow running through the valve, the bigger the power losses are.

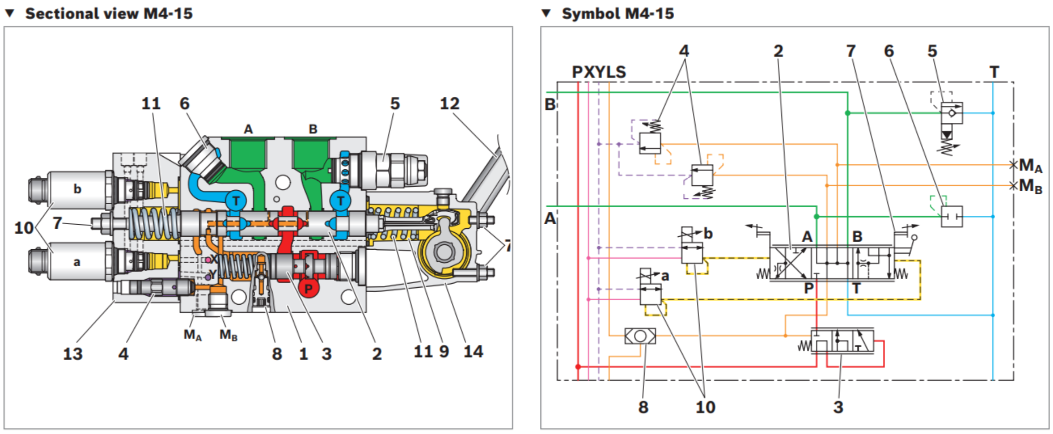
Pressure losses directly cause power losses in the circuit, because the forces produced by the system are linearly dependent on pressure as follows. Pressure drops causes power loss and the total power balance is

$$P_{pump} = P_{useful} + P_{loss}$$
$$p_{pump} * Q = p_{load} * Q + \Delta p_{loss} * Q$$
$$P_{loss} = (p_{pump} - p_{load}) * Q \qquad \text{eq. 13}$$

This equation shows that pressure losses cause power losses linearly because of force F on eq. 10 is linearly dependent on pressure.

Pilot control operations can be implemented in some adjustable valves. The pilot signal adjusts the valve and can come from a proportional valve actuator controlled by a PID controller for example.

The valve presented in Figure 8 could be used in the load sensing hydraulic system. Rexroth M4-15-series load-sensing control block monitors the highest load pressure occurring in the system and send a corresponding pilot signal back to the pump or controller. This way the variable-displacement pump sets its displacement regarding to the need so that mechanical input work stays at minimum improving work efficiency. Precise and load-independent metering is achieved by pressure-compensators which provide controlled flow. This makes the flow to an actuator essentially independent of load pressure. The valve can be used with fixed or variable-displacement pump. The actuation options are mechanical, hydraulic, electrohydraulic or electro-pilot with on-board electronics. Some variants offer load-holding functions and integrated pressure limiting or pilot relief valves in the inlet element for system safety and stable load holding.



1 Housing
2 Control spool
3 Pressure compensator
4 LS pressure relief valve
5 Pressure relief valve with feed function
6 Threaded plug
7 Stroke limiter
8 LS shuttle valve
9 Spring chamber
10 Pressure reducing valve
11 Compression spring
12 Hand lever
13 Cover A side
14 Cover B side

Figure 8. Load-sensing control block M4-15 by Rexroth. [22]

As can be seen from Figure 8, the control block has many valve types presented earlier in this chapter. Port T connects to tank and has low pressure. P is the high pressure produced by the pump. It is used in pressure compensator (3) to produce desired flow into the main 4/3 valve. Pressure reducing valves (10) use pressures of ports X and Y, where X is an external pilot oil supply port providing high pilot pressure signal for CAN bus-controlled pilot module (CPM) and Y is a low pressure signal which is eventually connected to tank. X and Y pressure define the pressure reducing valves (10) behavior which controls the main valve spool position (2). LS port is a load sensing signal from the previously connected control block. Ports A and B are consumer ports used for the cylinder chambers for example. The pressures of A and B define the forces encountered by compression springs (11) and thereby define the control spool (2) position. When a hydraulic system has many actuators, these control blocks can be stacked on each other. Then the LS shuttle valves (8) of each block are connected in series which provides the greatest pressure occurring in the system. The elements and functioning occurring in this control block are utilized in the LS model presented in chapter 3.2.

2.2.3 Hydraulic pumps

Hydraulic pumps pump oil into the system. Pumps can have control options such as flow or pressure compensators [21]. Pumps have different working principles and parameters. A smaller modern forestry forwarder Malwa 560F uses variable load sensing axial piston pump with torque limiter operating at 195 bar, 1700 rpm and pumping 148 l/min [23]. A bigger modern forestry forwarder Komatsu 875 uses a variable axial single circuit piston pump with load sensing element, producing 360 l/min at 2000 rpm [3].

2.2.4 Tank and hydraulic fluid

Tank and reservoir are places to store the hydraulic fluid when it is not circling in the system. The difference between a tank and a reservoir is that a reservoir is just a concept of storing fluid with no additional parameters and thus can only be used in simulations. A tank has several parameters such as volume, outlets and fluid surface height. [16]

2.3 Modelling approaches

Model-based systems engineering (MBSE) has the process and methods that will be part of an overall systems engineering processes in the future. Currently document-centric approach is generally practiced, but model-centric approach is expected to overrun that. MBSE is defined as follows: "(MBSE) is the formalized application of modelling to support system requirements, design, analysis, verification and validation activities beginning in the conceptual design phase and continuing throughout development and later life cycle phases." Model centric approaches include MBSE being in a role of a long-term trend. [24]

Modelling a forwarder is cheaper and faster than using prototypes. A virtual prototype allows quick testing of different improvement proposals in design phase and therefore studying their effects. A great need of improving has occurred since requirements of productivity, safety, environment caring and ergonomics has increased. [25]

The modelling can be done in various software applications. A comparative study between Simscape and Simcenter Amesim was carried out and found that both models are suitable for desired simulations and had major differences in simulation times. [26]

Hydraulic models can be used for sensitivity analysis. Jim W. Hall et al. carried out a study that illustrated deficiencies of standard regression coefficients in nonlinear models, which in this case was a bend pipe. The sensitivity analysis is used for various reasons. It can identify factors that have the most effect on model output, uncertain factors that need more studying and factors that provide no value and can be left out from further analysis. It can be found out if an analysis provides known influences from the simulation process and identify maximum variations between inputs and outputs. It can be identified if a factor or factor groups interact with each other. As in good modelling practice, sensitivity analysis is considered to be a necessary part of it. There are different approaches to sensitivity analysis - derivative-based sensitivity indices, linear regression, fourier amplitude sensitivity test, measure of Morris and many other methods. [27]

In the previous study example of a bent pipe [27], it was found out that the inlet pipe diameter parameter had the most influence in the results. This kind of information is useful for this thesis's simulation part as well. Although not committing a full sensitivity analysis, the correlations between pipes and accumulators with modified parameters can be observed in different parts of the hydraulic system.

It is convenient that the software can model both the multibody simulation and the hydraulics system. Suitable options are for example ADAMS [26] and Matlab with Simulink toolboxes. The parallel master's thesis about a forwarder's multibody simulation was made with Matlab and Simulink, so this software was used also for the hydraulics system simulations to be compatible for further testing and integrations.

There are two basic ways to make a suitable hydraulics system for the forwarder. One is a direct driven hydraulics system, and the other is a load sensing hydraulics system. These systems are taken a closer look in Chapter 3.

Johan Forsberg has compared energy efficiencies between two different harvesters in his thesis. The thesis conducted of a Simulink and Simscape models of the harvesters. The simulation models were simplified and for example did not have hydraulic accumulators or load sensing systems. [5]

3 Simulation models

This section explains the general Simscape blocks that are used in both the direct driven hydraulics (DDH) and load sensing (LS) systems. In the following sections 3.1 and 3.2 the case individual blocks are presented in detail.

The chosen method to make a hydraulic system was to make a simulation. Simulation modelling is often used method to test a system before implementing it. This is risk-free and usually cheap to make. It basically only requires the software, modelling skills and simulation hardware. This thesis' simulations are run with a PC with main hardware of NVIDIA RTX 3070 Ti, Intel i9-10900k @5,1 GHz, M.2 SSD and 32 BG of DDR4 RAM.

The hydraulic simulation models were made with Matlab, using Simulink and Simscape toolbox. Simscape toolbox provides all hydraulics related features, for example pumps, actuators, valves and sensors. The software was chosen, because it has all the needed functions and features, it is already licensed by Aalto University and because of that, professors are familiar to use it as well.

The simulation model goal was to make the hydraulic system to move the forwarder arm. The arm has three joints, which are moved with three hydraulic cylinders individually. The used forwarder model from the parallel thesis can be seen in Figure 9. The pistons (also referred as cylinders) are numbered and parts named accordingly.

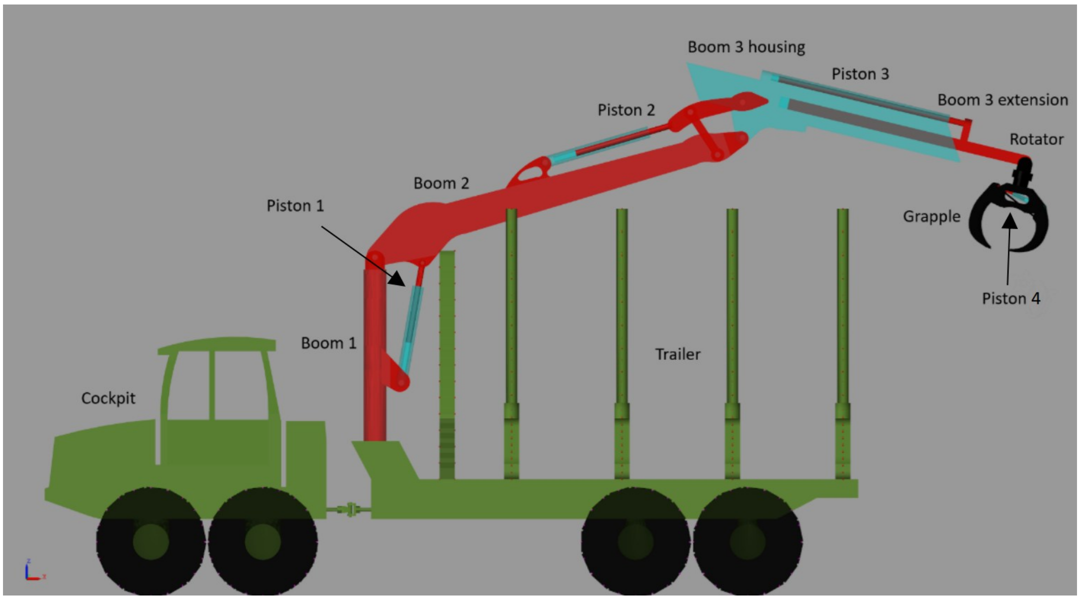


Figure 9. Multibody model of the forwarder with numbered parts. [28]

Two simulation models are made: first one is a simple traditional DDH system, and the second one is a more advanced system with a single pump and an LS system. The hydraulic circuits are simplified versions including only the necessary features for wanted simulation results.

Crane parts used in the models are considered as being rigid to simplify the models. The crane model only includes the related actuators to keep the scope of the simulation task light. The forwarder can be seen in action in Figure 10.

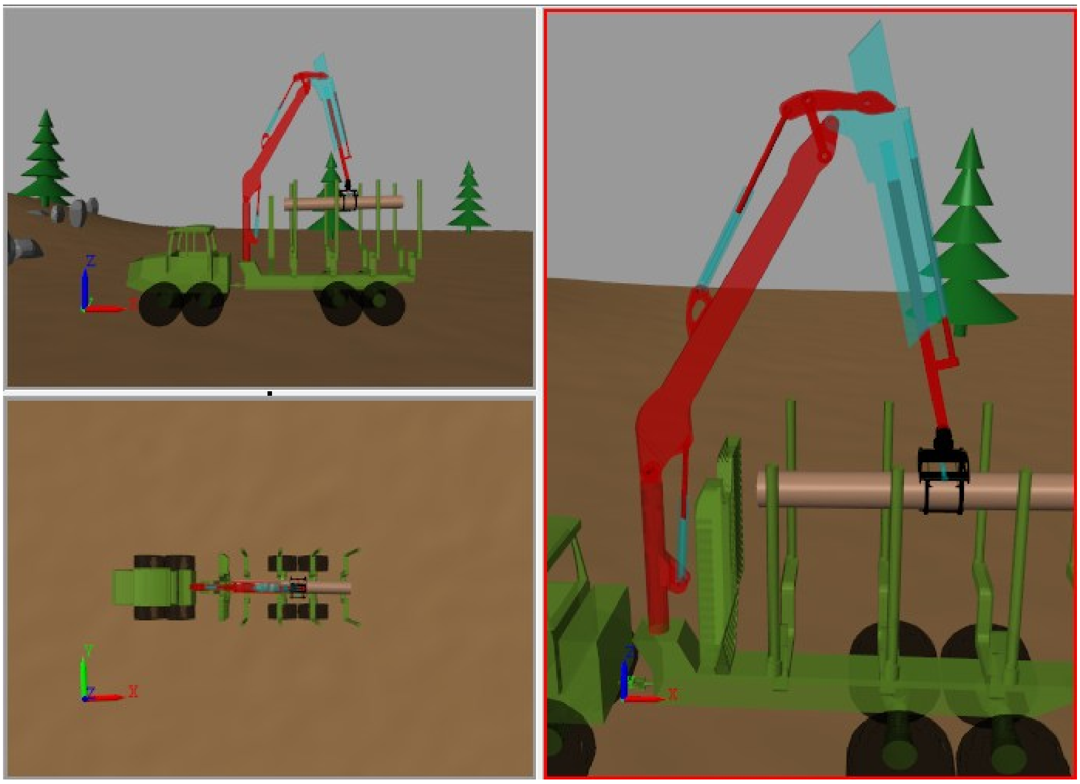


Figure 10. Multibody model of the forwarder during work cycle in simulation environment.

The masses of forwarder bodies are presented in Table 1. The masses affect the forces that each piston concerns.

Table 1. Mass of forwarder bodies. [28]

Body	Mass (kg)
Cockpit	6000
Trailer	6500
8 Wheels	2800
Boom 1	900
Boom 2	600
Boom 3 housing	250
Boom 4 Extension	150
Grapple	300

A hydraulic piston velocity

$$v_{piston} = \frac{Q_{piston,A}}{A_{piston,A}} \qquad \text{eq. 14}$$

is calculated as maximum for each cylinder, where mass flow Q and area A are values from cylinder's A-chamber. Piston maximum velocities are defined as shown in Table 2. The simulation is run with these defined, so the hydraulic simulation parameters are set accordingly.

Table 2. Maximum piston velocities for simulation. [28]

Piston	$Q_{max}\ (\frac{l}{min})$	η_{flow}	$d\ (mm)$	$v_{max} (\frac{m}{s})$
Piston 1	360	0.4	100	0.306
Piston 2	360	0.4	90	0.377
Piston 3	360	0.4	80	0.477
Piston 4	360	0.2	60	0.424

The forces are exported to file from the simulation. The forces can be seen in Figure 11. Forces come from moving the pistons which are moving the forwarder boom arm bodies and carrying the log. The friction in cylinders is small.

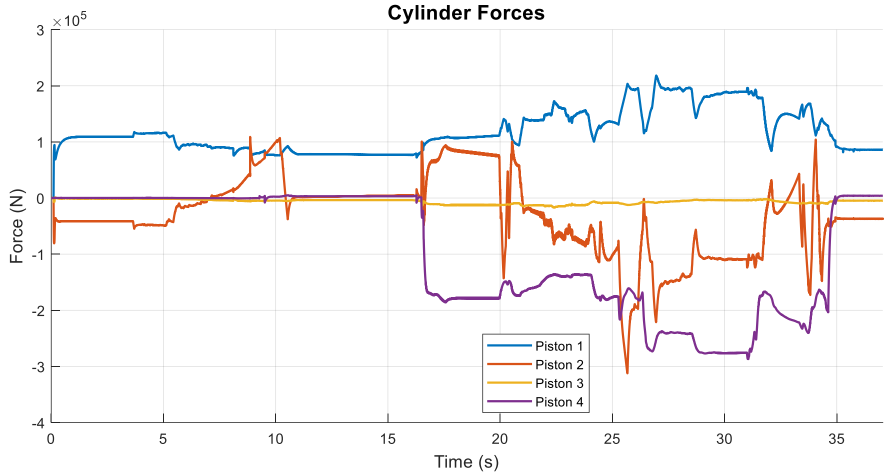


Figure 11. Cylinder forces from the simulation.

Hydraulic fluid has a major role in how the system performs. In the controlled simulation environment, the most significant property of used hydraulic fluid system is bulk modulus, because it defines how easily the fluid moves in the system and thus how fast the accumulators behave. Other fluid properties such as thermal properties come into question when applying the system in a realization prototype, as it can then be altered to changing conditions such as temperature changes caused by wind, sun, radiator or work duration. In Simscape, the fluid properties are set into an *Isothermal Liquid Properties (IL)* block. Parameters are set to be a custom hydraulic oil as follows in Figure 12.

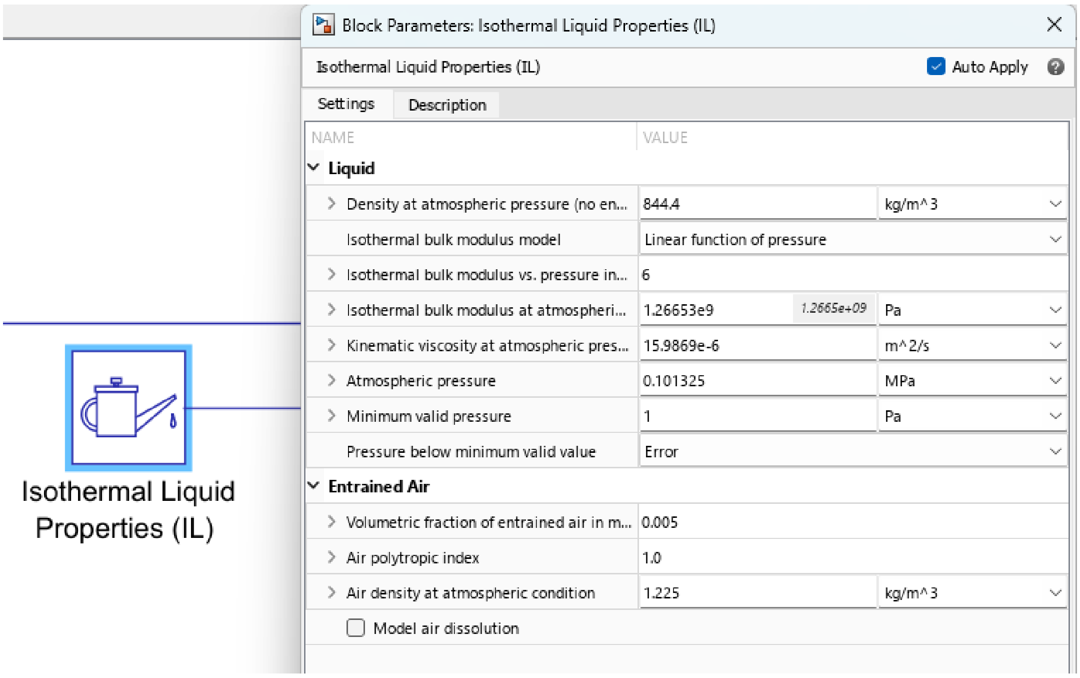


Figure 12. Hydraulic fluid properties used in every simulation.

There are various valves used in different sections of hydraulic systems. The main valve in this thesis is 4-way 3-position valve as seen in Figure 2. This valve is considered to be the main valve, because it directs the flow for the cylinder. The name comes from its capability of allowing flow in four directions by having three positions. The positions can vary depending on the model, but in this case the positions are shown in Figure 13 and opening characteristics in Figure 14.

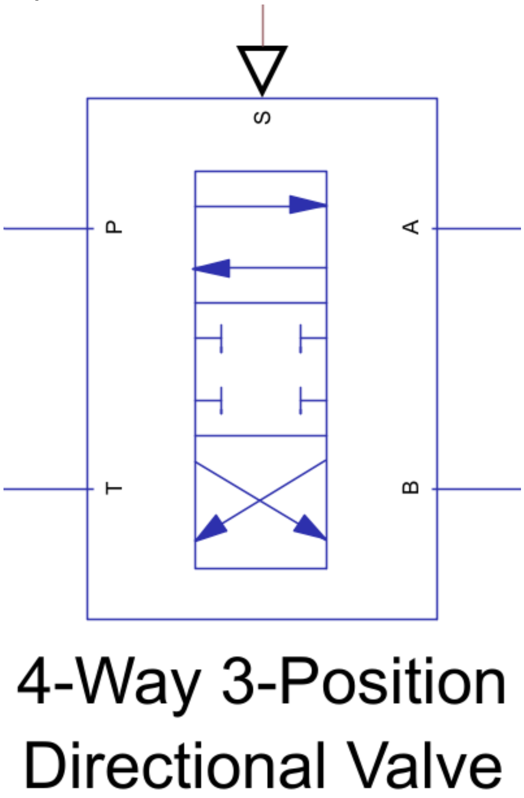


Figure 13. Valve positions behaviour.

In this thesis the valve opening characteristic is shown as in Figure 14. The first position will have open flow from port P to A and B to T. The second or the middle position the flow is blocked in every direction. The third position allows flow from port P to B and A to T. The valve position is controlled by a solenoid. The solenoid can be electronically controlled, and its behavior can be defined by application.

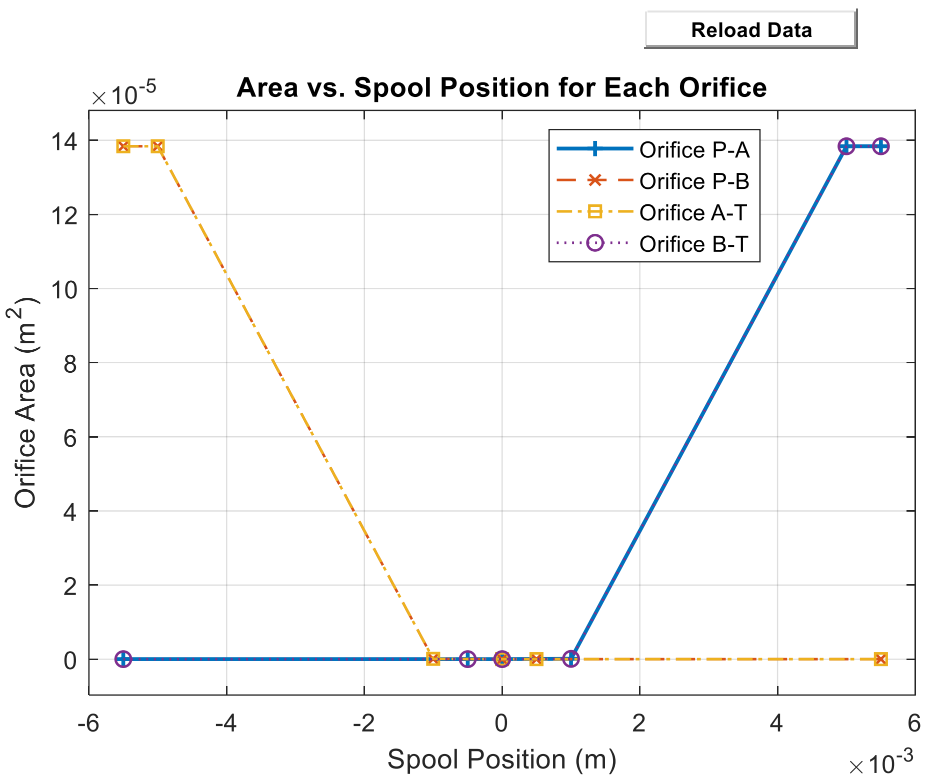


Figure 14. 4-way 3-position directional valve characteristics.

The valve openings are set as shown in Figure 14. The solenoid type valve spool can move total of 10 mm. The spool controls an orifice opening area. The opening area changes linearly with spool movement but has small dead zones at the three main positions to reduce small vibrating piston control movements.

In Figure 15 the down sampled reference signal is divided into 50 points. This still follows the original signal well enough to simulate the movement accordingly. Increasing the number of points will drastically reduce the simulation speed.

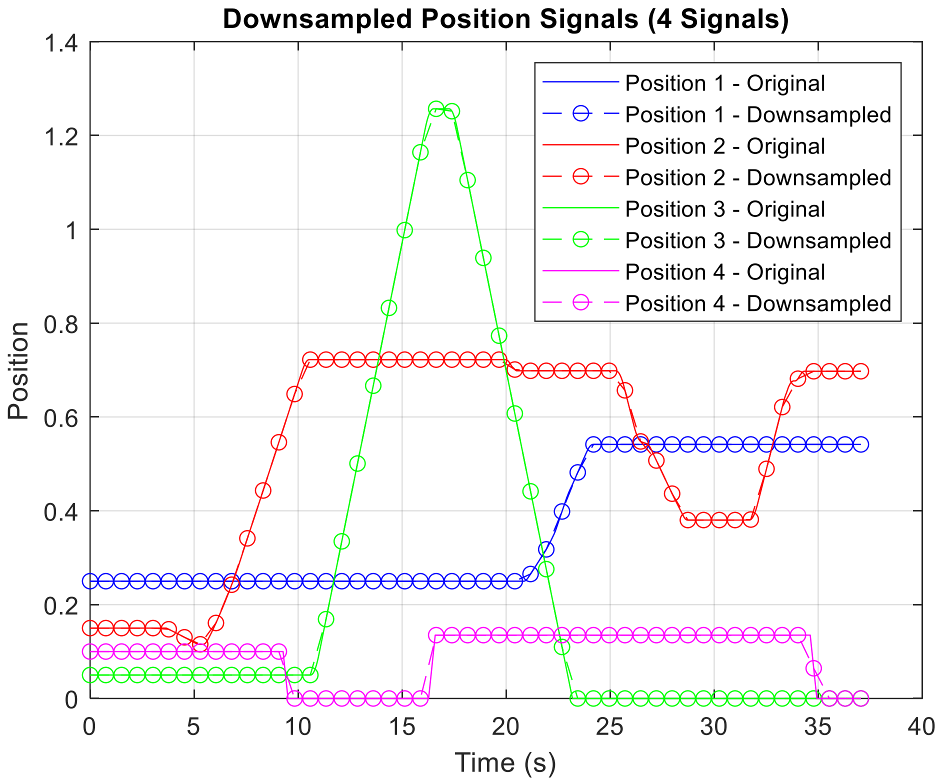


Figure 15. Down sampled reference signal of piston positions with 50 points.

In many scenarios the pump's angular velocity ω_p comes from the diesel engines crankshaft. In this model the ω_p is driven by PID-controller and no diesel motor is modelled separately. However, in chapter 5 the overall consumption power is considered with a fitting power source (diesel aggregate), and its fuel consumption is discussed.

The focus of the simulation is in one work cycle including movements of picking a log into the forwarders bunk as shown in Figure 10. The movement and forces of the crane are recorded from the parallel master's thesis into a CSV file. The file is then down sampled in Matlab, and the different sections of arm movement are separated for each piston movement references. The down sampling does not degrade the results, and is mandatory, because the simulation time would have taken way too much time. The hydraulic simulation tries to follow this reference signal the best way it can with PID- and other circuits. Reference signal is shown in Figure 15.

3.1 Direct Drive Hydraulics model

Direct drive hydraulics work by adjusting the pump motor's rotational speed and direction while keeping the pump displacement fixed to directly move the cylinder.

The DDH system consists of four identical systems only differing about parameter values. Each system has a fixed-displacement pump, symmetrical hydraulic cylinder and a PID-controller. The model top-view can be seen in Figure A1, where each pump-valve-cylinder combination runs individually. Pump subsystem is presented in Figure A2.

In Figure 16 the DDH system's piston positions are presented during the 37 seconds of the work cycle simulation. Pistons and down sampled (DS) references are numbered accordingly. Piston 1 has the reference of down sampled DS reference 1, piston 2 has DS reference 2, and so on. The actual piston positions follow the reference signal very well as can be seen from the signals being on top of each other's most of the time. The alternating force per cylinder seen in Figure 11 cause disturbance to the system which leads to errors and offsets about reference signals. The PID controllers try to maintain the piston positions on reference signal and the controllers' values are shown in Table 3. Note that integral and derivative members of the PID are not tuned and have a value of zero. The PID-controller's integral member had little to no effect, so it was decided to be zeroed on every controller. Thus, derivative members were not even tried to be tuned. The proportional gain's effect works well when it is set high enough.

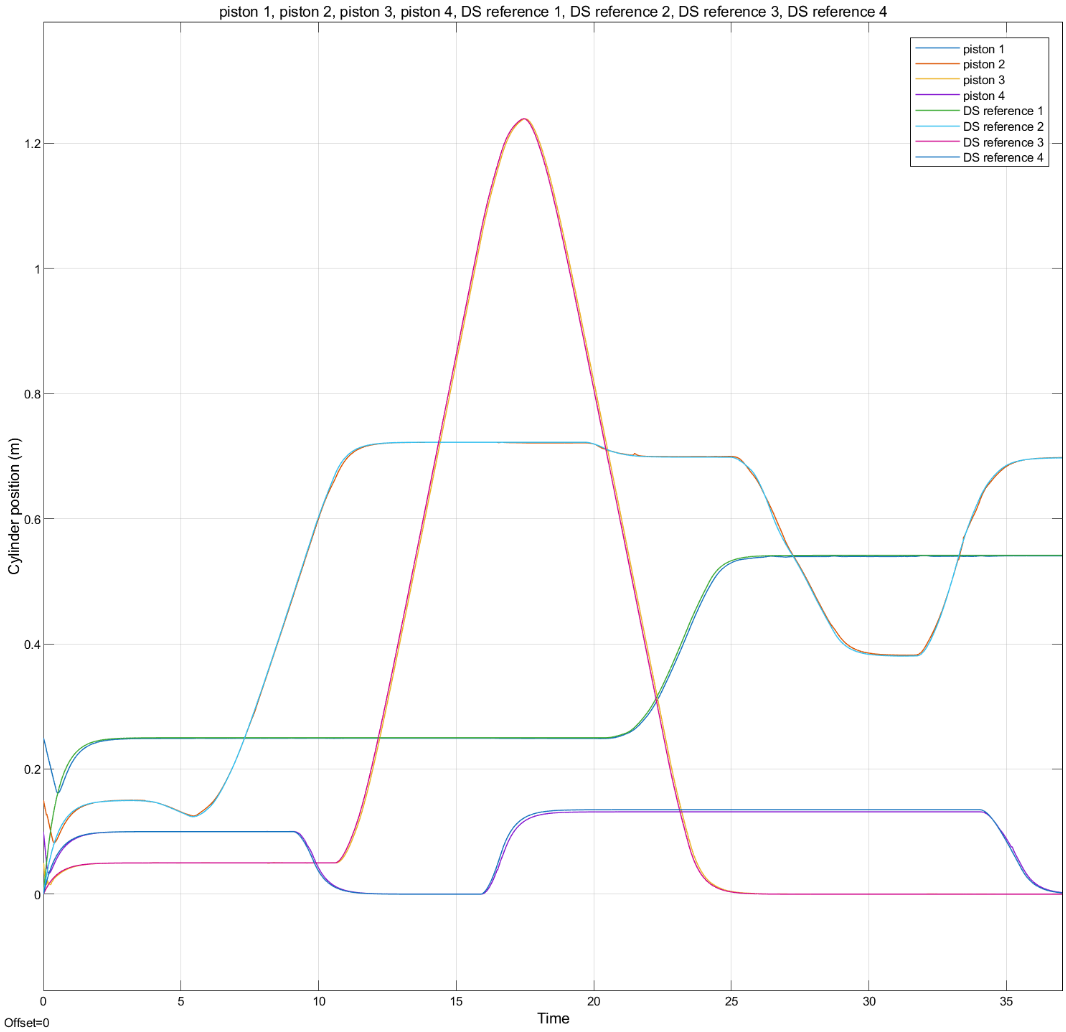


Figure 16. DDH piston positions of one work cycle.

Table 3. DDH system PID controllers' values.

Controller	P	I	D
Controller 1	40000	0	0
Controller 2	40000	0	0
Controller 3	40000	0	0
Controller 4	50000	0	0

In Table 4 the maximum torques are estimations based on eq. 6 at maximum rated pressure of 420 bar. The rated maximum torque is only informed at 100 bar pressure.

Table 4. Commercial pumps which properties are used in DDH simulation parameters. [29]

pump	make and model	maximum torque from eq. 6 [Nm]	theoretical rated maximum torque at 100 bar [Nm]	mechanical efficiency	volumetric efficiency
Pump 1	Parker F11-012	83,16	19,8	0,97	0,97
Pump 2	Parker F11-012	83,16	19,8	0,97	0,97
Pump 3	Parker F11-012	83,16	19,8	0,97	0,97
Pump 4	Parker F11-005	32,76	7,8	0,97	0,97

The 37 seconds simulation took around 80 seconds to complete. The simulation progresses at steady speed and has no slowdowns. Solver preferences are shown in Figure 17.

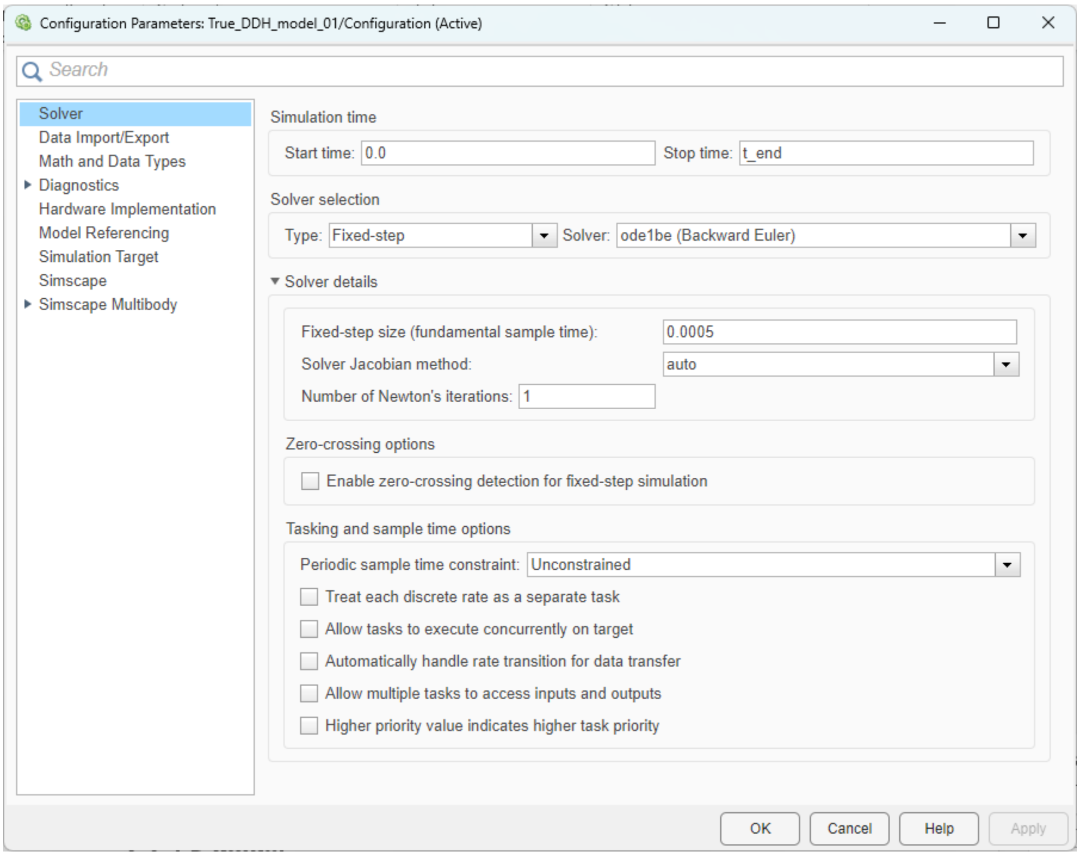


Figure 17. Solver settings used in DDH system simulation.

3.2 Load Sensing model

The second model is a load sensing and one hydraulic pump driven system. The model top-view can be seen in Figure A3, where each valve-cylinder combination runs by one variable-displacement pump. Pump subsystem is presented in Figure A4 and tank in Figure A5. Load sensing unit can be done in a traditional way or electronically. Both LS methods provide a well working LS module and are suitable for the system. For this system the LS system was made electronically. The electrical LS system was chosen, because a traditional LS system is more demanding to simulate and slows down the simulation significantly. The subsystem consists of switches that sort out the maximum pressure currently in the system. The numbered A and B signals are pressures from the corresponding piston chambers. The U value comes from PID controllers. It decides which pressure (A or B) will be considered. When PID gives positive U value, the piston is extending, and A chamber pressure is considered. With negative U value, the piston is contracting, and B chamber pressure is considered. This load sensing system is presented in Figure 18.

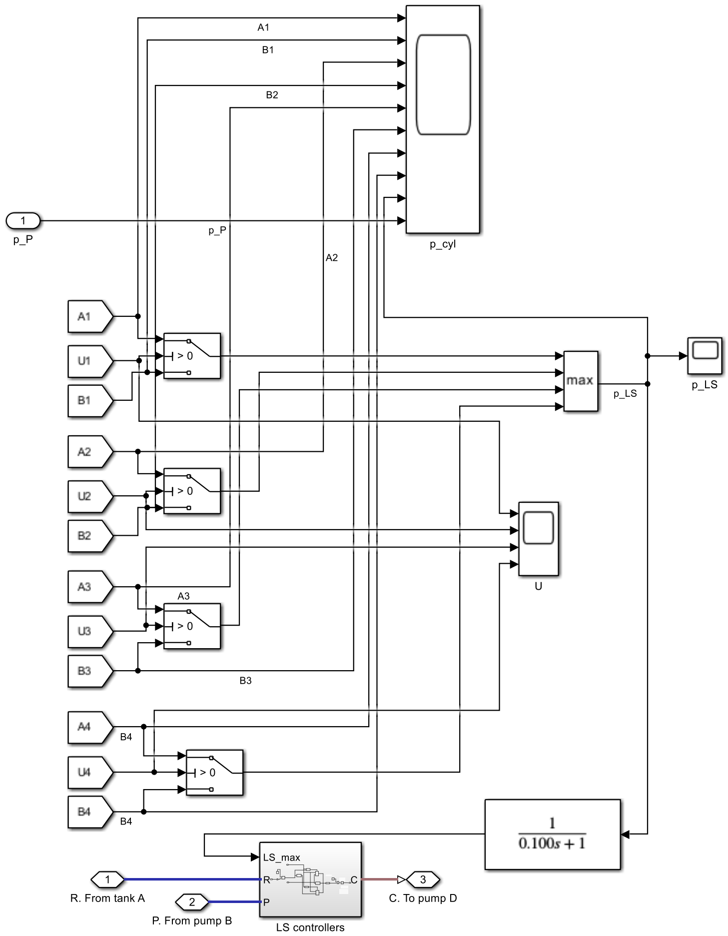


Figure 18. Load sensing system.

The maximum pressure is then sent to LS controller subsystem shown in Figure 19. The subsystem gets the maximum pressure occurring in the system from *LS_max* port. Port P is pump pressure and R is connected to tank. Double-acting pilot valve acts based on the pressure difference between *LS_max* and pump pressure. The pressure difference directly controls a 3-way directional valve spool that controls flow from pump to servo stroke pilot valve actuator, or from there to tank. The second 3-way directional pressure limiting valve is connected to pump, tank and controlled by pump pressure with a pressure limiting pilot valve actuator. The 3-way directional valves provide flow to an orifice (Control Piston, CP) and servo stroke controlling pilot valve actuator. The servo stroke pilot valve actuator subtracts its position from maximum stroke, has a gain block to product with pump nominal displacement divided by servo cylinder stroke and convert to radians per second. This output C signal is sent to the pump displacement port.

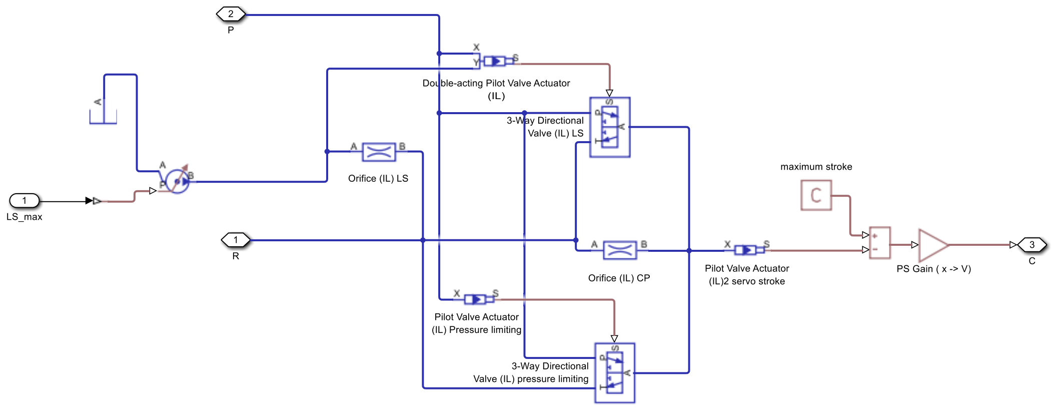


Figure 19. LS controllers in Load sensing system.

The pump used in load sensing application needs to be a variable-displacement pump, because the pump rotational speed will be fixed. One suitable pump is Parker PV092. It has suitable displacement range, speed and torque ratings. This pump's parameters are used in LS system simulation.

Figure 20 shows how the pistons move in 37 seconds work cycle simulation. Pistons and down sampled references are numbered accordingly. Piston 1 has the reference of down sampled DS reference 1, piston 2 has DS reference 2, and so on. The 37 seconds simulation took around 105 seconds to complete. The simulation progresses at steady speed and has no major slowdowns.

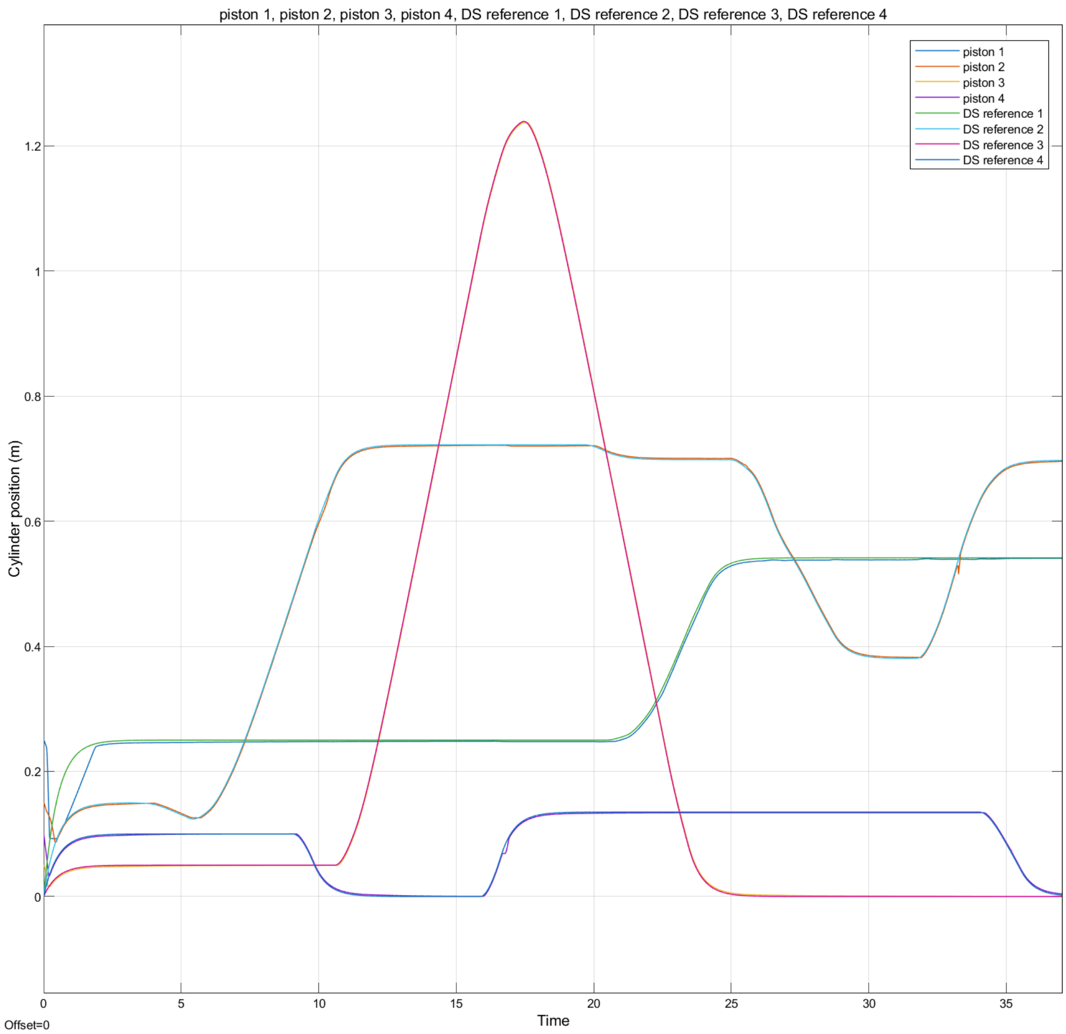


Figure 20. LS system's piston movements of one work cycle.

Table 5. LS system PID controllers' values.

Controller	P	I	D
Controller 1	500	0	0
Controller 2	1000	0	0
Controller 3	1000	0	0
Controller 4	1000	0	0

Parker PV092 variable-displacement pump's key parameters were used to simulate the LS model's motor behavior. They are presented in Table 6.

Table 6. Parker PV092 technical data. [30]

pump	displacement [cm3/rev]	maximum continuous operating speed [rpm]	torque [Nm]	mechanical efficiency	volumetric efficiency
Parker PV092	92	2300	533,3	0,93	0,97

The pressures during the work cycle can be seen in Figure 21. Maximum pressure over the pump is 28,2 MPa. The A side of the pump is connected to tank which has pressure levels between 0,1085 and 0,10875 MPa.

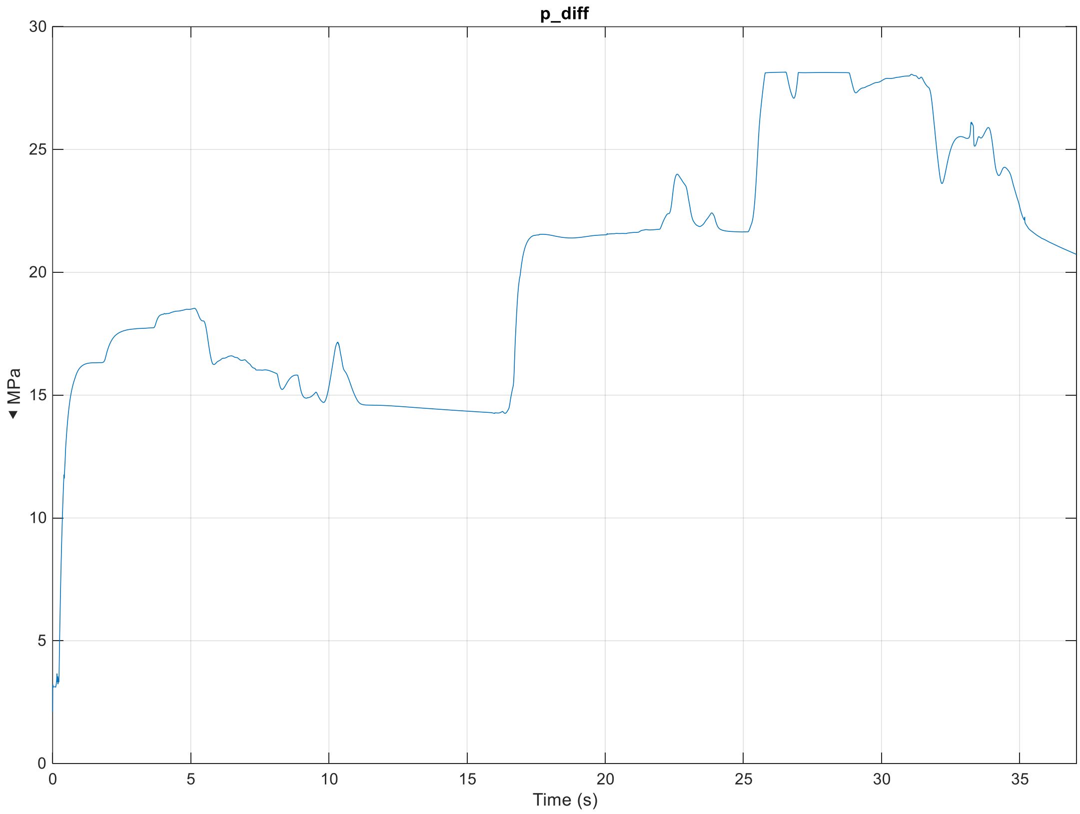


Figure 21. Pressure over variable-displacement pump in LS system.

Solver settings used in the LS model can be seen in Figure 22. The used step size gives fine results while having a decent simulation time. Solver settings are further discussed in Chapter 5.

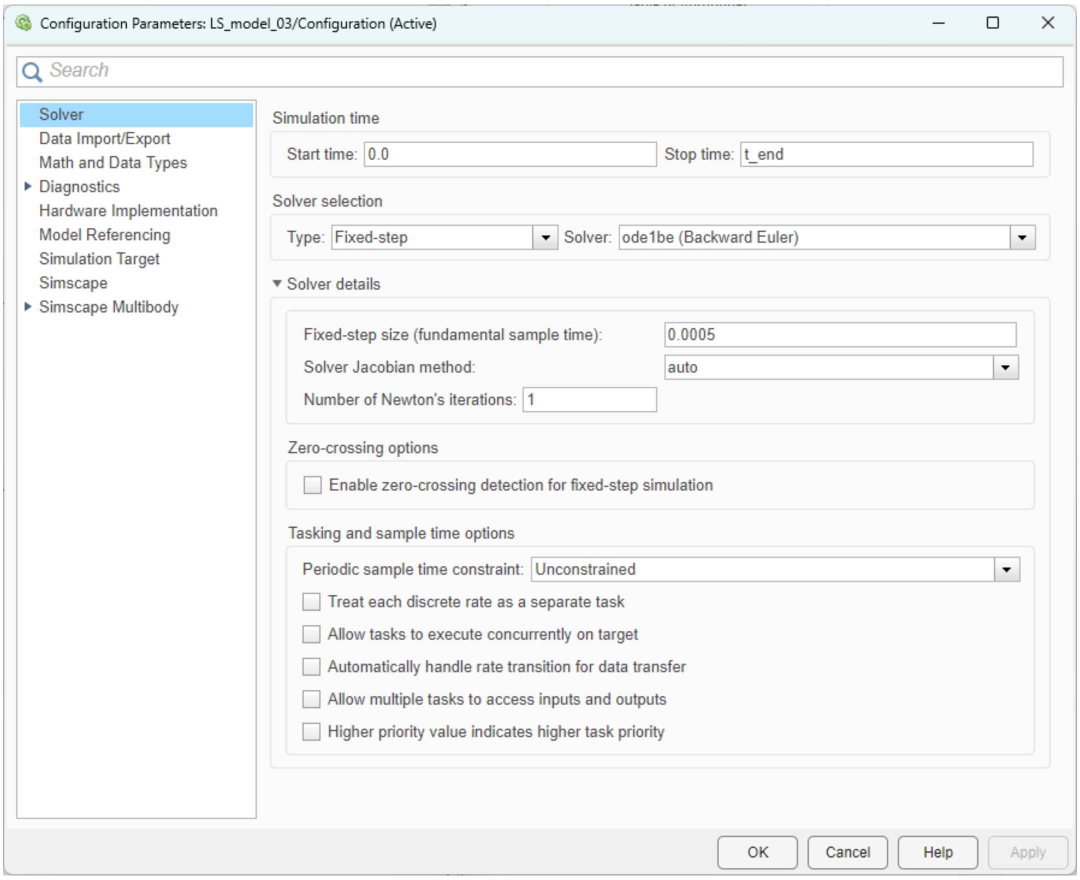


Figure 22. Solver settings for LS system simulation.

4 Results

The pump torques of the direct drive hydraulics system are presented in Figure 23. The different direction of the torque is caused by the forces that are encountered by the piston. To make a comparison of the simulated torque with a real-world pump, Parker F11 series pumps are used as shown in Table 4. As can be seen, the torques are manageable for the pumps 1-3, having maximum peak of 69 Nm. Pump 4 behaviour is further discussed in Chapter 5.

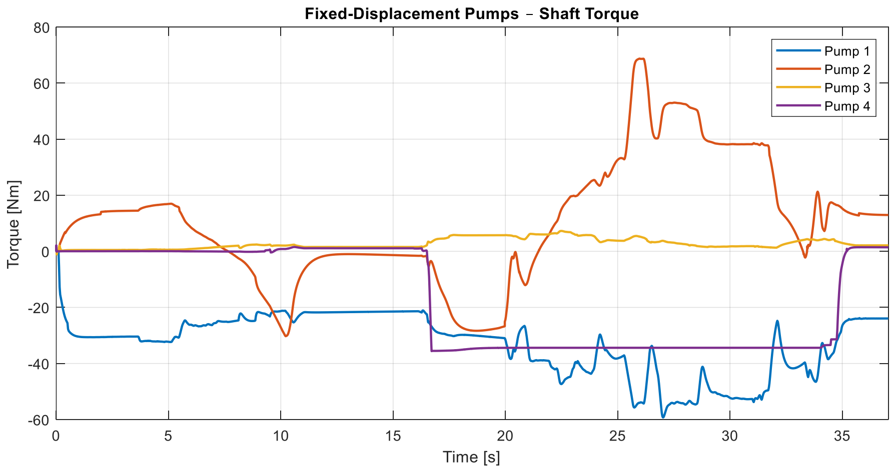


Figure 23. Fixed-displacement pumps shaft torques in DDH system.

The torques during the simulation are under the calculated maximum torque on pumps 1-3 peaking at 69 Nm, but pump 4 faces 35 Nm while grabbing the log. This error will not be fixed due to difficulties faced in simulation forces, which are discussed in Chapter 5.

The pumps rotating speeds can be seen in Figure 24. The rotation speed is an important factor effecting to torques and efficiencies. Too big displacement leads to low rotational speed, and too low rotational speed leads to high torque.

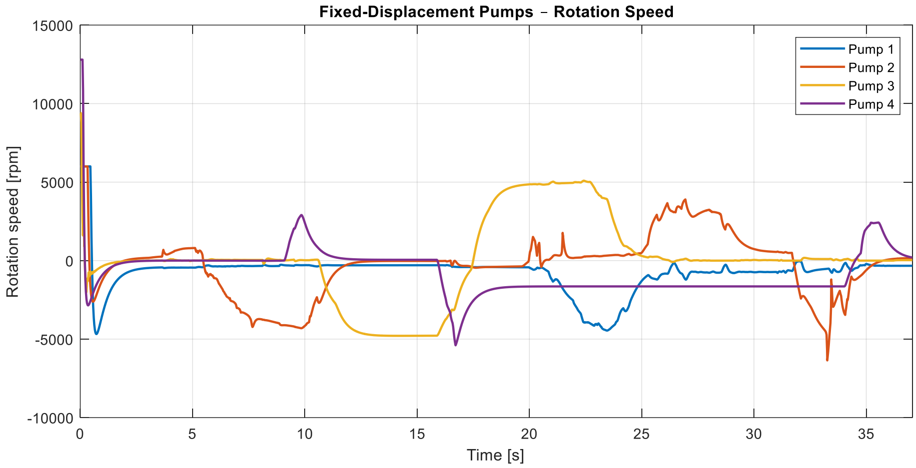


Figure 24. DDH pumps' rotational speed in rounds per minute.

Next, the LS model is discussed. The commercial pump which specifications are used in LS system's variable-displacement pump is Parker PV092. Its key specifications are maximum displacement of 92 cm^3/rev, maximum peak pressure of 420 bar, maximum input torque of 533,3 Nm and maximum speed of 2300 rpm [30]. This pump suits well in the LS system according to torques occurring during simulation run as seen in Figure 25.

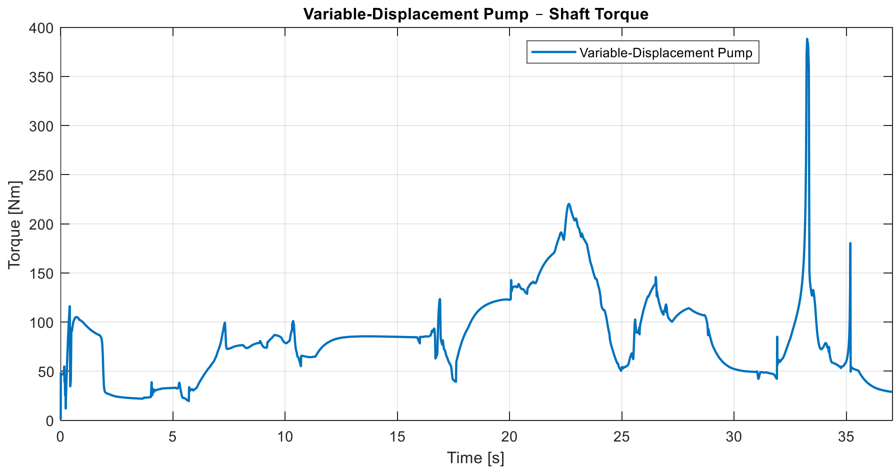


Figure 25. Variable-displacement pump shaft torque in LS system.

In Figure 25 the torque of the variable-displacement pump is presented. Majority of the time the torque is under 230 Nm. The PV092 has 533,3 Nm maximum torque rating which is not exceeded during simulation run. There is spike in torque around 33 seconds into simulation, which is caused by high stiffness of the simulation model and unsmoothed grapple rotation (fixed maximum speed with no/infinite acceleration), so it can be left out. This will be further discussed in Chapter 5.

Energy efficiency of each system cannot be calculated. The simulation run consists of varying directions of forces, velocities and thus powers. This makes the efficiency calculation complex, because the simulation run has many points where a cylinder is providing work and receiving work. The workload with gravity and cylinder position relative to ground and velocity direction defines the work power direction. This is problematic in comparison with the DDH and LS systems, because only DDH can turn "negative" work to pump into use. LS system will always lose possible potential energy from a cylinder, because the fluid flows to a tank and not to spin the pump's motor. DDH can have good regeneration efficiency (only friction and leakage losses of cylinder and components), but LS has always zero regeneration efficiency.

4.1 Energy usage

Energy usage can be examined in two ways: from the pump motors' input and output energy, and from pump motors' input and cylinders mechanical output energy. Total energy usage of the direct drive hydraulics system can be calculated by summing all the pumps' input mechanical energies. Total hydraulic output can be calculated by summing all the pumps' output hydraulic energies, or alternatively from cylinders' output energy. With the pumps' total mechanical input and total hydraulic output, overall system energy usage can be calculated since the cylinder has no leakage and friction effect is minimal. For the DDH system work cycle simulation, the pumps' total mechanical input energy is 324 kJ, total hydraulic output energy 115 kJ and cylinders total output energy 113 kJ. Average mechanical power of the pumps in simulation run was 8.73 kW and average hydraulic power 3.10 kW. Mechanical input power and hydraulic output power can be seen in Figure 26.

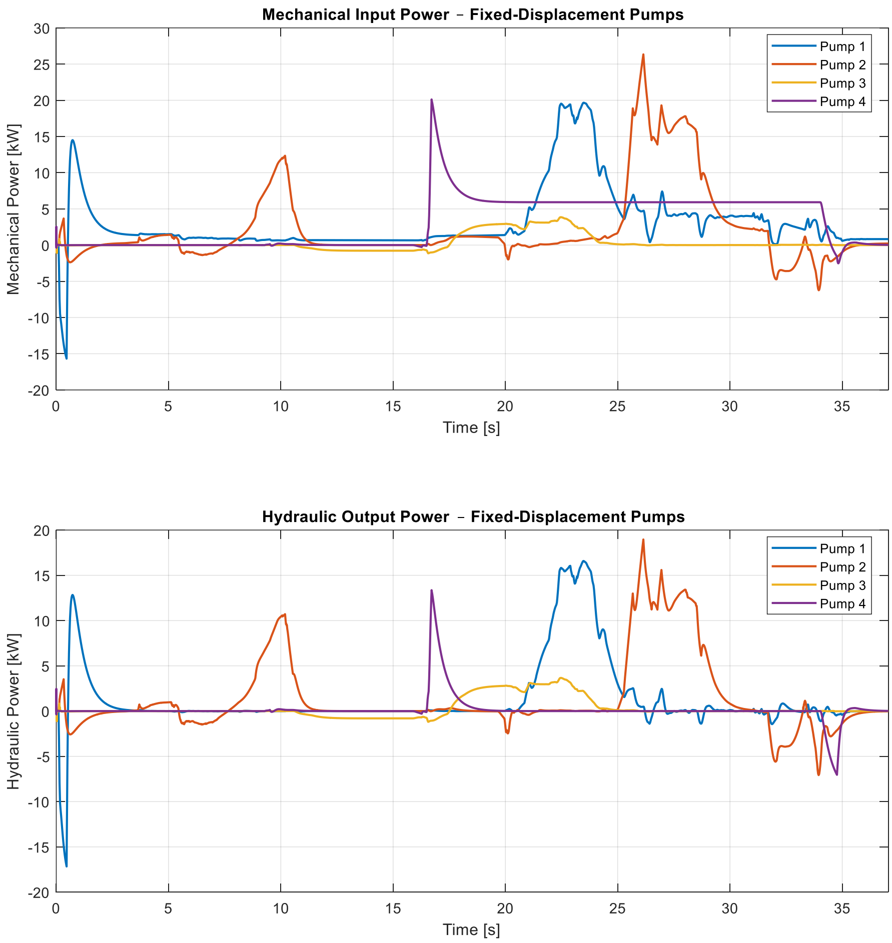


Figure 26. DDH system total input/output energy of the pumps.

DDH system pumps' total mechanical input power and hydraulic output power is presented in Figure 27. Pumps' total mechanical input power and cylinders' output power is presented in Figure 28. The difference is small as the output energy of pumps and cylinders are similar (115 kJ vs 113 kJ).

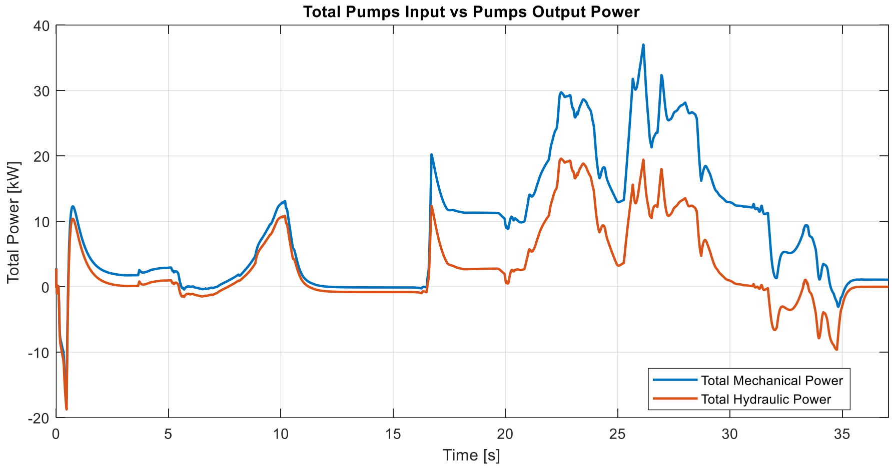


Figure 27. DDH system pumps' total input and output power.

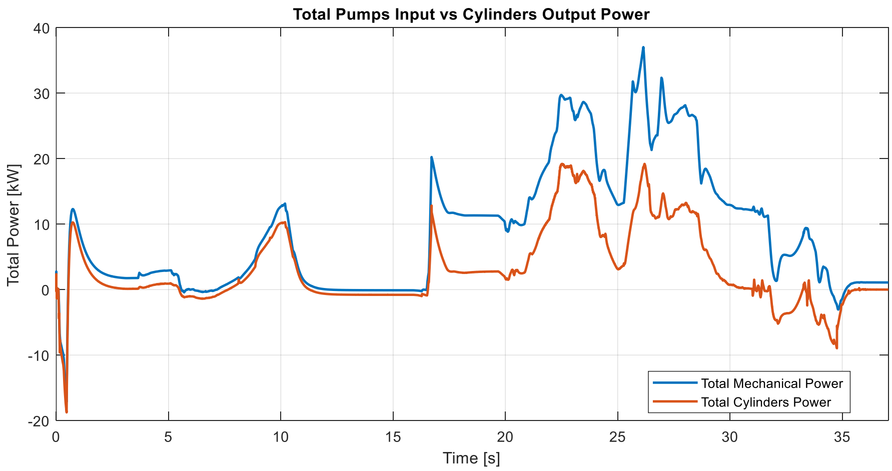


Figure 28. DDH system total pumps' input and cylinders output power.

DDH system with only cylinders 1, 2 and 3 needs only mechanical input energy of 210,38 kJ and has hydraulic output energy of 109,90 kJ. The cylinder 4 efficiency during simulation is poor, because it uses mechanical input energy of 113,19 kJ and hydraulic output energy of 4,96 kJ during simulation. It uses 5,9 kW just to keep the grapple closed as can be seen in Figure 26.

Total energy usage of the load sensing hydraulics system is the pump's input mechanical energy. With the pump's mechanical input and hydraulic output and cylinders' output, overall system energy usage can be calculated. The simulation gives pump's total mechanical input energy of 752 kJ and total hydraulic output energy of 529 kJ. Cylinders total output energy is 99,89 kJ. Average mechanical power of the simulation run was 20,30 kW and average hydraulic power 14,28 kW. Pump's mechanical input power and hydraulic output power can be seen in Figure 29. Pump's mechanical input power and cylinders' output power can be seen in Figure 30.

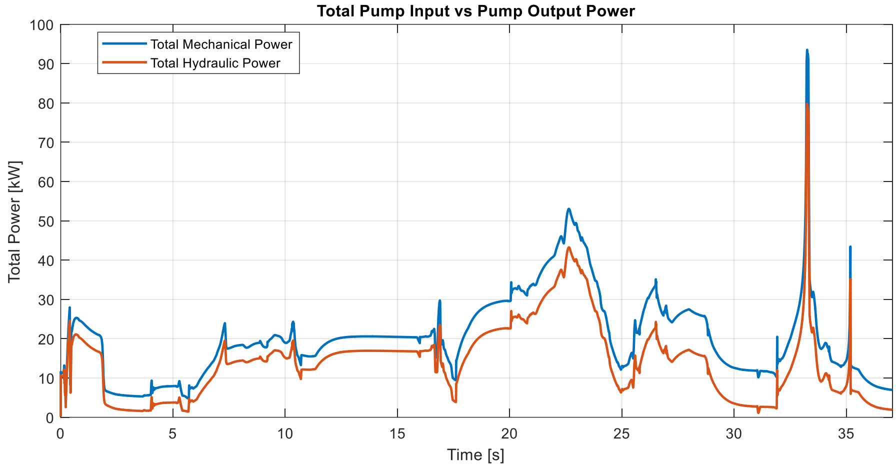


Figure 29. Variable-displacement pump's mechanical input power and hydraulic output power.

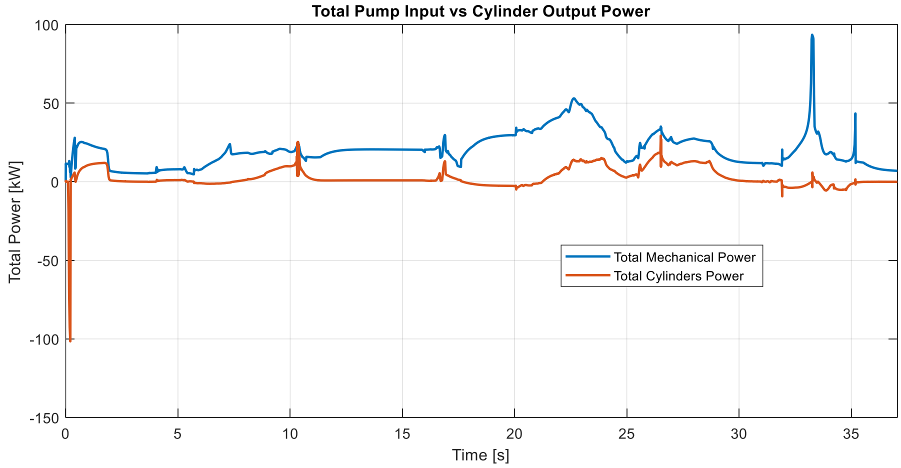


Figure 30. Variable-displacement pump's mechanical input power and cylinders' output power.

The pumps' total hydraulic output energy is 115 kJ on DDH system and 529 kJ on LS system. The DDH system has cylinder output energy of 112,84 kJ and the LS system has 99,89 kJ. The cylinder output energies are similar because the systems do the same piston movements with small differences in PID tuning and working principles. When looking at the pumps' output energies, it implies that the DDH system uses 78 % less energy with the pumps in the system to achieve the same piston movements during the work cycle. This great difference is mainly caused by power regeneration in DDH system and losses in hydraulic components in LS system. LS system has greatly more components that have leakage and thus losses. Also, when the pump is driven for the highest pressure needed, all components that do not need the high pressure are leaking more than they need to. Driving the pump on higher pressure causes energy waste in the system. Table 7 presents the main results about energy and power usage in DDH system and Table 8 in the LS system.

Table 7. Energy and power usage of each pump in DDH system.

Pump System	Pump's Total Mechanical Input Energy (kJ)	Pump's Total Hydraulic Output Energy (kJ)	Cylinder's Total Hydraulic Output Energy (kJ)	Average Mechanical Power (kW)	Average Pump's Hydraulic Power (kW)
1	114,12	48,20	47,37	3,08	1,30
2	83,51	50,20	49,50	2,25	1,35
3	12,74	11,50	11,12	0,34	0.31
4	113,19	4,96	4,85	3,05	0,13
1-4	323,57	114,87	112,84	8,73	3,10

Table 8. Energy and power usage of pump in LS system.

Pump System	Pump's Total Mechanical Input Energy (kJ)	Pump's Total Hydraulic Output Energy (kJ)	Cylinder's Total Hydraulic Output Energy (kJ)	Average Mechanical Power (kW)	Average Pump's Hydraulic Power (kW)
1	752,11	529,08	99,89	20,30	14,28

4.2 Energy comparison

The DDH simulation uses parameters from Parker F11 series pumps. Their volumetric and mechanical efficiencies are presented in Figure 31. To get reasonable results of efficiency, the pump parameters need to be set for volumetric (hydraulic) and mechanical efficiencies. The simulations use values 0,97 for both efficiencies as can be interpolated from Figure 31.

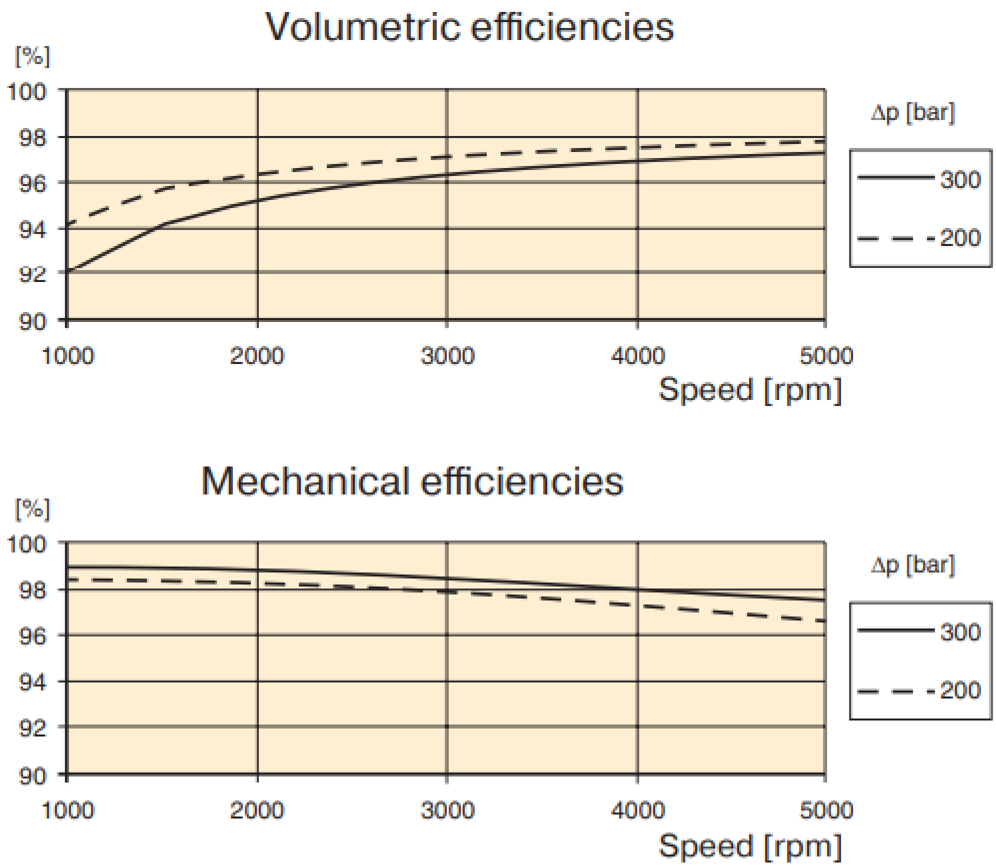


Figure 31. Parker F11 series efficiency maps [18].

For the load sensing system, the pump's mechanical and volumetric (hydraulic) efficiency is set based on Figure 32.

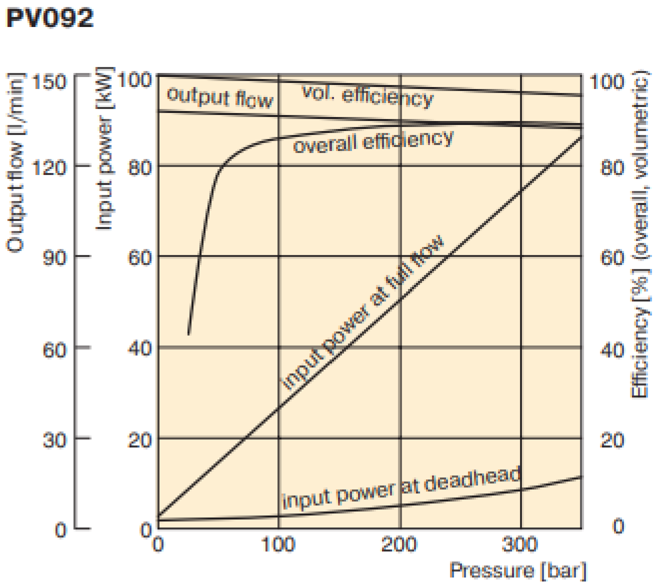


Figure 32. Parker PV092 pump efficiency map [30].

It shows that the DDH system performs better than the LS system. The amount of energy used for the simulation cycle varies significantly between DDH and LS systems. DDH used mechanical input energy only 323 kJ and LS used 752 kJ for the work cycle. This means that energy needed in this particular simulation work cycle in DDH system is only 0.43 times the energy needed in LS system. The total mechanical input power needs to be known to fully understand the system's energy consumption.

5 Discussion

A diesel engine is the most used power source in forestry forwarders today. Another power source is a battery in an electric forwarder. Using diesel as a fuel is easy compared to having a charging station in the middle of a forest. Using diesel has its benefits in a remote location. The fuel can be transported for the forwarder from a gas station, or a mobile diesel tank can be used. Electric forwarder needs to be charged, but electricity source for the charger is not necessarily available at the work zone. Ponsse has developed a Range Extender to manage this problem, but it is just a diesel aggregate [7]. In using both the power sources, diesel or electric, diesel still has its role.

The direct drive hydraulics model is as plain and simple as it can be. There could be pressure relief valves to be used to protect the components from over- and underpressurization to prevent cavitation. In the used work cycle there did not arise such incidents in the system. The DDH model could use an accumulator to store pressure from cylinder movement. It works as a tank and could provide high pressure fast for the cylinder which enhances its performance. The cylinders would really benefit from load holding valves. Currently the cylinders are not moving all the time, which means that the pump needs to do wok to keep the piston in place. These valves would greatly reduce the required pump output.

The DDH system energy usage would benefit from load holding valves shown in Figure 33. The concept allows the pump and motor to stop producing pressure when piston needs to stay still. This way by closing the load holding valves (LHV) the piston stays in position and does not need energy from the pump. The system improves overall DDH system efficiency in scenarios where piston is under load and not doing work (stays still). If this was used in the DDH simulation of this thesis, the saved pump energy would take place all the time that the piston positions stay the same (see Figure 16). This kind of system could be developed on this template. One method to control the load holding valves electronically would be to keep them closed when the controller input signal is zero (axis dead zone) and open the valve when signal is other than zero.

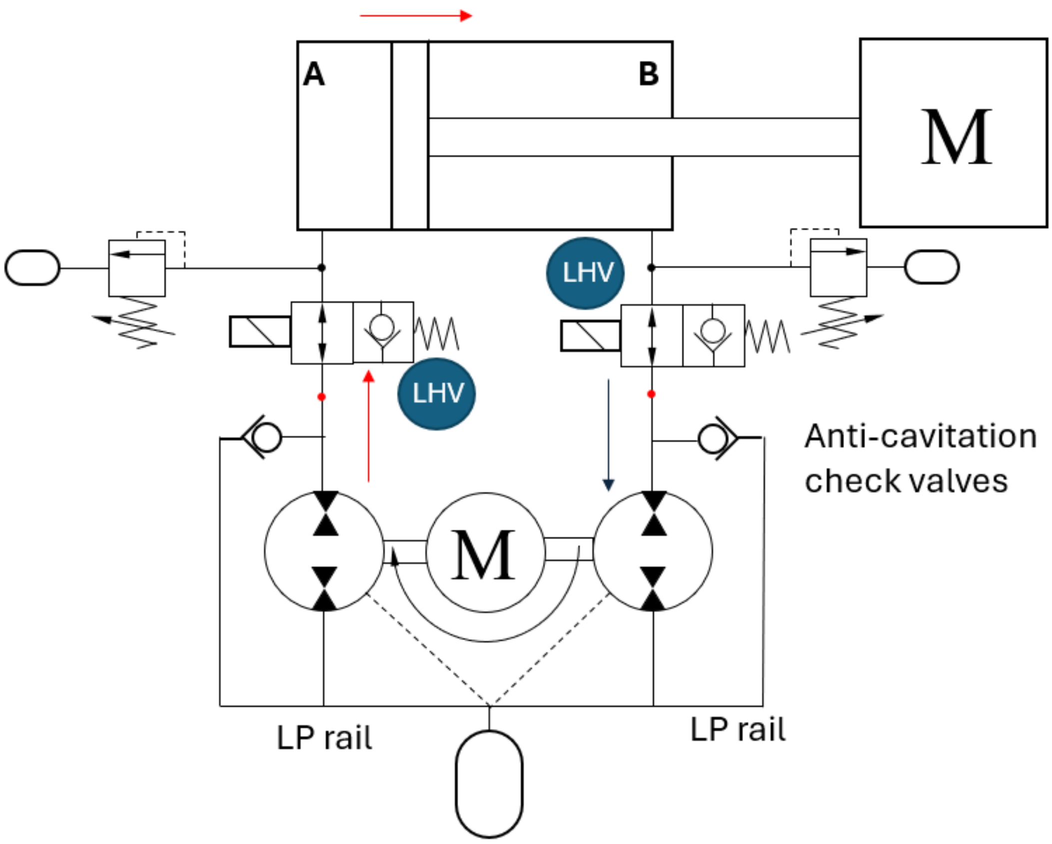


Figure 33. DDH system with load holding function (LHV, load holding valve).

A suitable load holding valve could be Sun Hydraulics's DACCMCN 2-way, 2-stage, solenoid operated directional poppet valve [31]. This 2/2 valve closes when piston needs to stay still. The valve is opened by pilot operated check valve or electronically when the piston needs to move. The piston and valves' leakage cause piston positions to be offset over time, so the pump may have to produce pressure to compensate for leakages. This small work is minor compared to a system without LHVs where pumps need to hold pressure all the time.

In Figure 34 the typical losses of energy can be seen. The simulation work cycle has moments where pistons stay still and thus cause losses. The LS system performs well when idling, but the DDH system uses input energy to maintain piston positions.

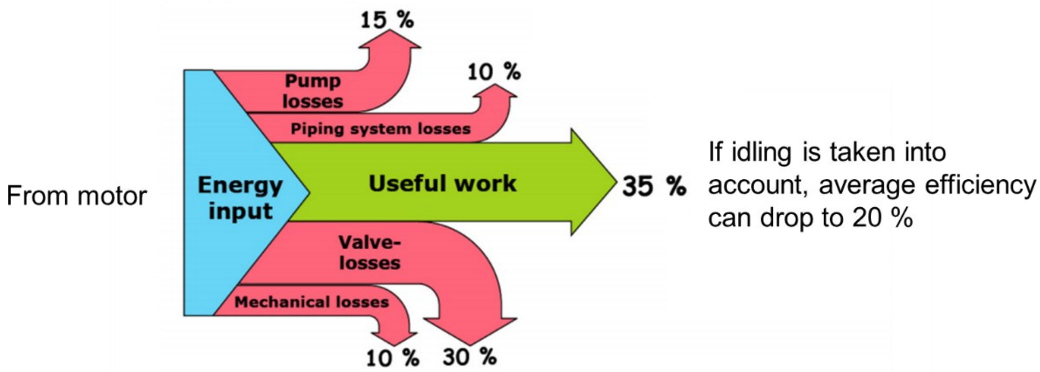


Figure 34. Typical loss levels in mobile valve controlled hydraulics with variable pump. [32]

In direct drive hydraulics model, there are two possible reasons for the high torque that the piston 4 encountered during the log's movement. The more obvious reason might be that the pump displacement is too big. The fixed displacement only requires the pump shaft to spin too low speed which increases faced torque. The second reason can be a matter with multibody simulation model itself. The piston 4 may face forces that should not be that high. This is due to lack of friction in the simulation or dimensions about the grapple and log. Piston 4 is experiencing forces to squeeze the log all its holding time, when in practice grapple geometry, gravity and friction help the piston 4 to hold the log in place. These factors are not considered in multibody simulation model, so the effect of reducing piston 4 forces cannot be estimated with current model.

In Figure 25 the load sensing system variable-displacement pump's torque appears to reach a peak of 388 Nm at 33 seconds. This peak happens when the log is dropped onto the forwarder bed by releasing grapple (cylinder 4 movement). The spike does not actually need to happen, because the effect of reducing log's weight from the cylinders does not need to be fixed that fast with PID controls. The negative force spike forces the PID controllers to act fast to match piston position to reference signal. In practice, the reference signal after releasing the log is not that crucial to achieve very fast by changing the pump's rotational speed unnecessarily quickly. The boom arm can be a few centimeters off from reference signal, but the simulation does not allow it. When reference signal was created, the forces did not cause any slowdown or additional movement to the cylinders.

If previously mentioned torque spikes did not happen, the LS-system's pump size could have smaller maximum displacement. This optimization could improve energy consumption by operating at better output efficiency.

A fixed-step solver was chosen for both the DDH and LS simulations, because a variable-step solver was significantly slower and caused errors during simulations. The solver still had problems with cylinder pistons moving to end position and caused pressure difference related errors. These errors could be avoided by decreasing step size or increasing dead volume in cylinder chambers.

The simulation models can be optimized more precisely by committing a sensitivity analysis as presented in chapter 2.3. This was not yet done in the simulation as the optimization was not the main target of the thesis, and the analysis would be too time consuming to make. These performance properties optimizations are not considered within the scope of this master's thesis work.

The modeled DDH system is so simple that pump's mechanical input power and hydraulic output power are enough to get reasonable efficiency results. Another method would be to use pump mechanical input power and cylinders mechanical output power but estimating cylinder properties such as friction is inaccurate and its effect is minor. In the following figures the closer the total mechanical power and total hydraulic power are of each other the better the efficiency is.

Pump 1 encounters the most force, because it is the first one in the boom arm. It has also the biggest cylinder. During 25 - 34 seconds the weight of the log requires more input power to hold the load and thus the efficiency during that timeframe is low.

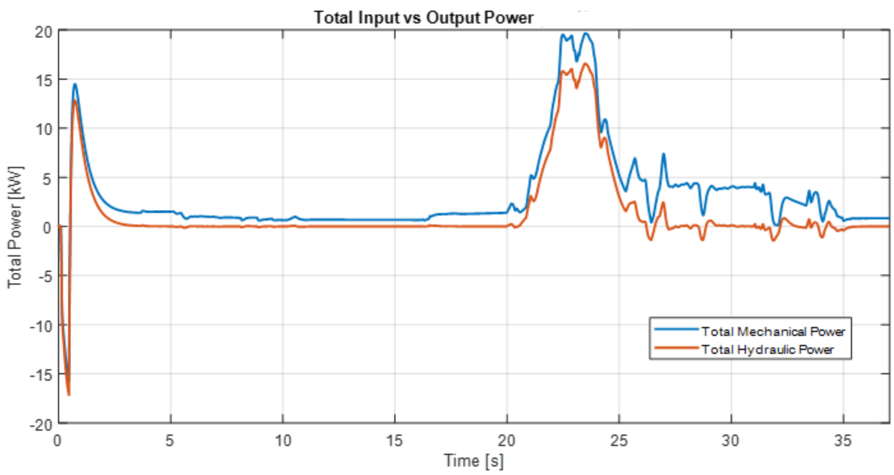


Figure 35. DDH pump 1 powers.

Pump 2 has good performance as seen in Figure 36. The piston moves a lot, so mechanical power is used effectively.

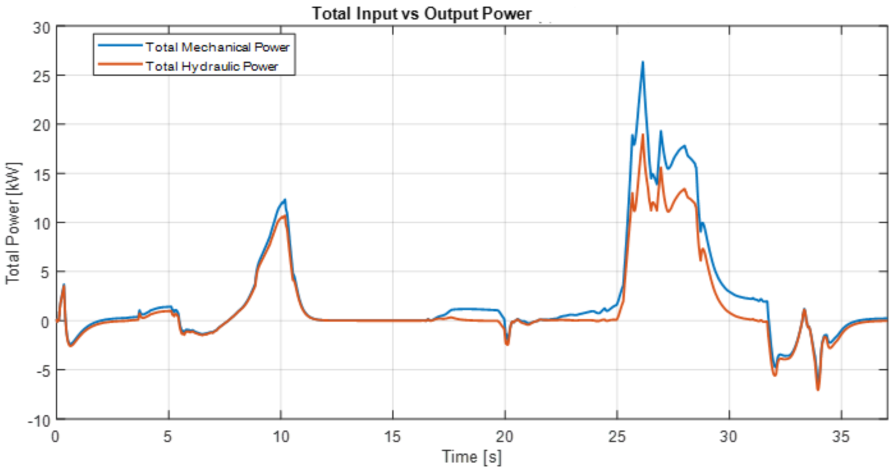


Figure 36. DDH pump 2 powers.

The pump 3 in DDH performance is presented in Figure 37. There is mechanical input power of the pump and hydraulic output power of the cylinder. As can be seen, efficiency is very high even with piston not moving over half of the time. This efficiency is achieved with the way the simulation was driven. The forces for cylinder 3 are minimal, because the piston was parallel with ground, leading to minimal force perpendicular to the piston rod. Thus, the pump does not need to produce high pressure to keep the piston in place.

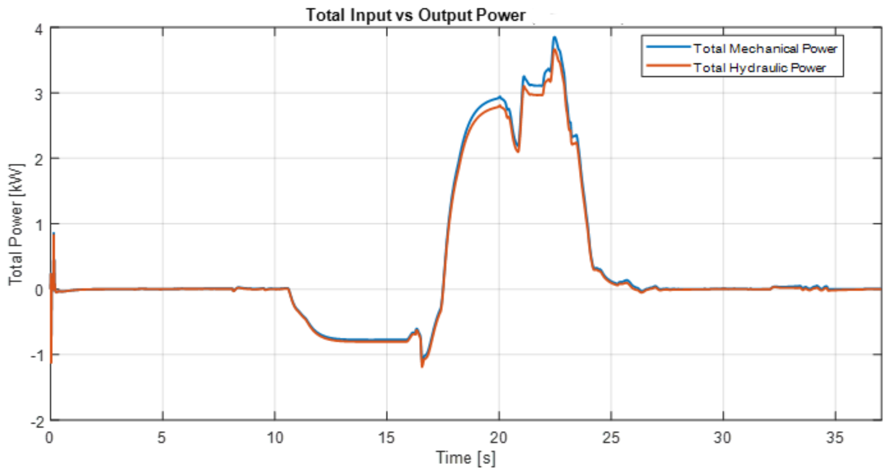


Figure 37. DDH pump 3 powers.

Pump 4 efficiency is the worst one. The piston stays still most of the time and encounters a great static force for a long time. It even uses almost as much input energy as pump 1 while being the smallest piston with the least output power as can be seen in Table 7.

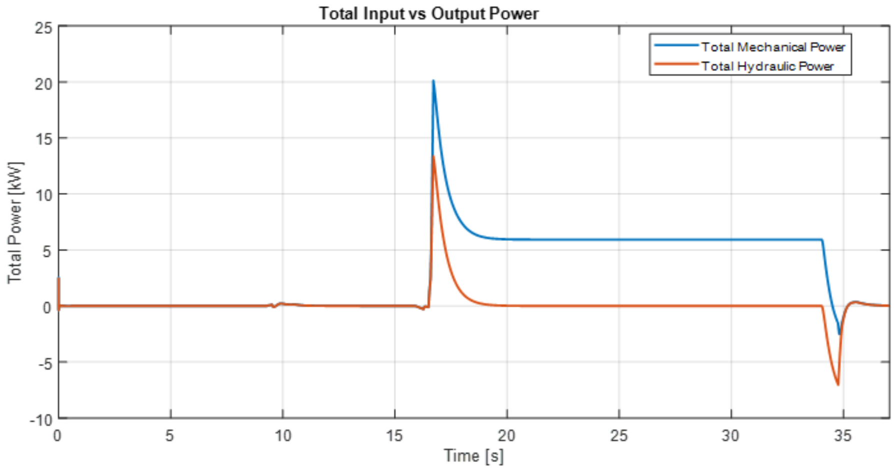


Figure 38. Pump 4 powers.

In Table 7 the results of energy, efficiency and power of each pump have been presented. As can be seen, pump 4 has a major negative effect on the overall DDH system performance. This is because of the period of holding the log using 5,9 kW, which is about half of the simulation time.

6 Conclusions

In this work energy efficiency of two hydraulic actuator systems, direct drive hydraulic (DDH) and load sensing (LS), were studied related to forestry forwarder's boom operations. The research was based on hydraulic system simulation.

The primary focus of this thesis was to determine efficiency and energy consumption of two different hydraulic systems for a simulated forwarder. To achieve that, key objectives were to build hydraulic simulation models to simulate forwarder's piston movements. They were built based on MATLAB's Simulink Simscape Isothermal Liquid Library. The simulation period of analysis was determined to be one work cycle which was the act of lifting of one log from roadside to forwarder bunk. The goal was to compare different hydraulic systems by their efficiency and energy usage with realistic hydraulic component parameters. The simulated outputs on both systems are comparable to real-life operation, though they currently reflect simplified models. More accurate results could be obtained by using more detailed simulation model.

The inputs for system simulation were from a previous study (parallel master's thesis by Henri Wahlman) in which a forwarder's multibody model was built without dedicated hydraulic actuators. This multibody simulation model allowed to export forces and positions of the hydraulic pistons. This recorded data was then used to carry out hydraulic simulation model runs with direct drive hydraulics system and load sensing system. Both the DDH and LS models were built to match the dimensions used in the multibody model. With the previously given cylinder sizes all the rest of the components could be chosen. The components have same or very similar properties that would be in a corresponding commercial component. Fundamentals of a DDH and LS systems were used as a basis for the models.

It was found that the direct drive and load sensing systems can be compared with the built simulators and the topic can be further investigated with them by adding necessary components, including new features, and changing system parameters.

Both DDH and LS controlled models are well built and provide great results in terms of inspecting and comparing efficiency and energy usage. Comparing the results showed that the LS system was taking roughly 2,3 times more input energy than DDH system during the work cycle. LS system also used about 4,6 times more pumps' hydraulic output energy. The pros of using a DDH system are that it is simple, easy to set up, has great performance and energy efficiency. The cons in DDH system are the need of having many pumps taking place and adding weight. This system also uses input power in situations when the actuator is just holding load without any motion. During those periods the pump needs to operate and maintain the actuator pressure which means significant energy usage. DDH model studied is very simple and working but would benefit from a load holding element which could be an electrically or hydraulically operated poppet valve with minimum leakage. To ensure this setup's operation, simulations and a real-world implementation with a prototype would be beneficial.

The pros of using a LS system are its load holding capability and need of only one pump. The cons of LS system are the pump's usage mainly at small displacements meaning low efficiency and the complex LS element with many components adding leaks and additional pressure losses. Usage of hydraulic actuators with different load pressures requires pressure matching which in LS systems is made by using throttling of flow leading to high pressure and power losses.

Improvements to the multibody simulation model would include a powertrain system. This subsystem would make the model produce more accurate results and to complete a forestry forwarder simulation model. Also, the hydraulic model(s) could be more seamlessly connected to multibody simulation run, maybe even in real time. This adduces the concern of simulation performance, as the simulation would be even more demanding task for the PC hardware. Currently neither the multibody model nor both hydraulic models can none be run in real time with high end gaming PCs. Optimization is possible and recommended, but it is outside the scope of this thesis.

This thesis shows that hydraulics modelling is a powerful tool for studying common hydraulic systems behaviour, energy consumption and efficiency analysis. The models created can be continuously improved further to give more accurate results. They provide a provide a solid foundation for further hydraulics research, as different components, parameters and other system solutions can be tested using this framework. The models do not limit to a forestry forwarder only but can be used in various mobile machine applications.

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A. Simulation models and Subsystems

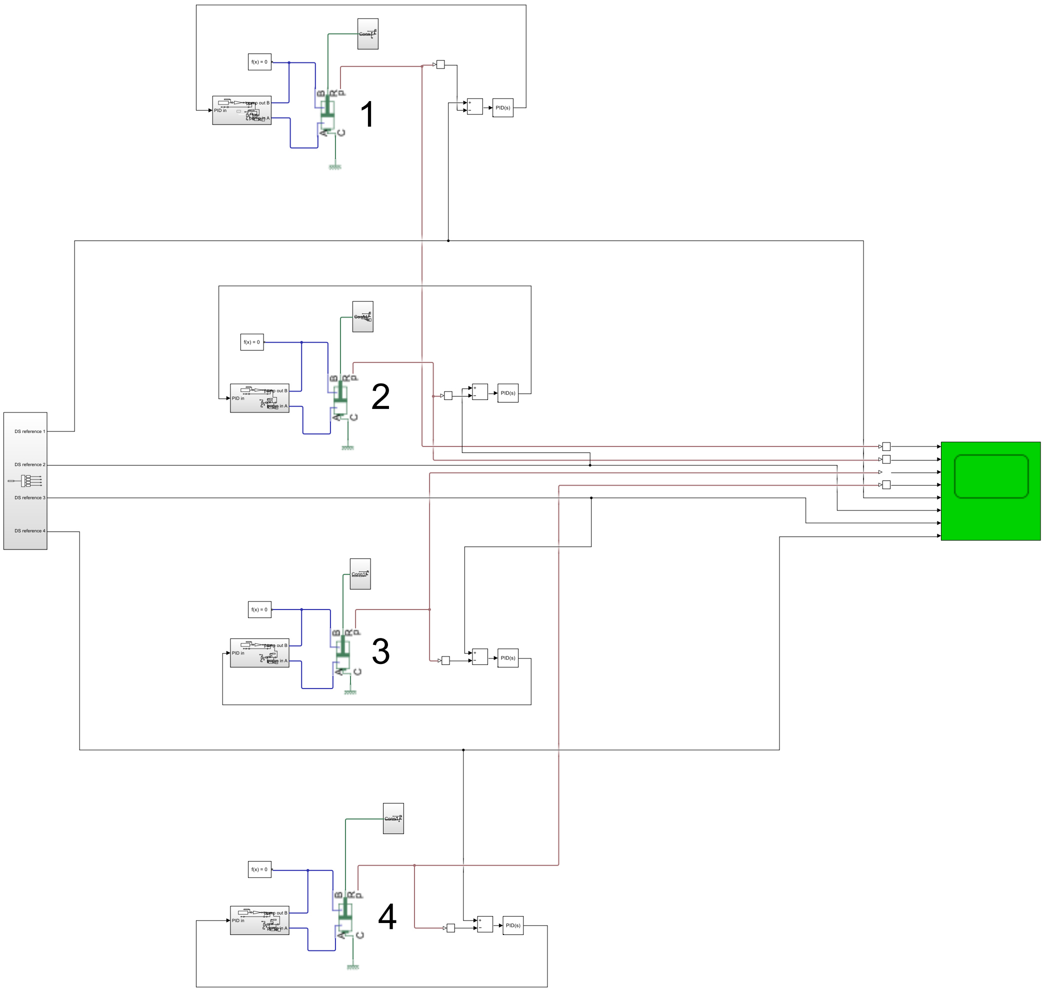

Figure A1. Top-level view of the DDH system simulation model in MATLAB Simulink with Simscape.

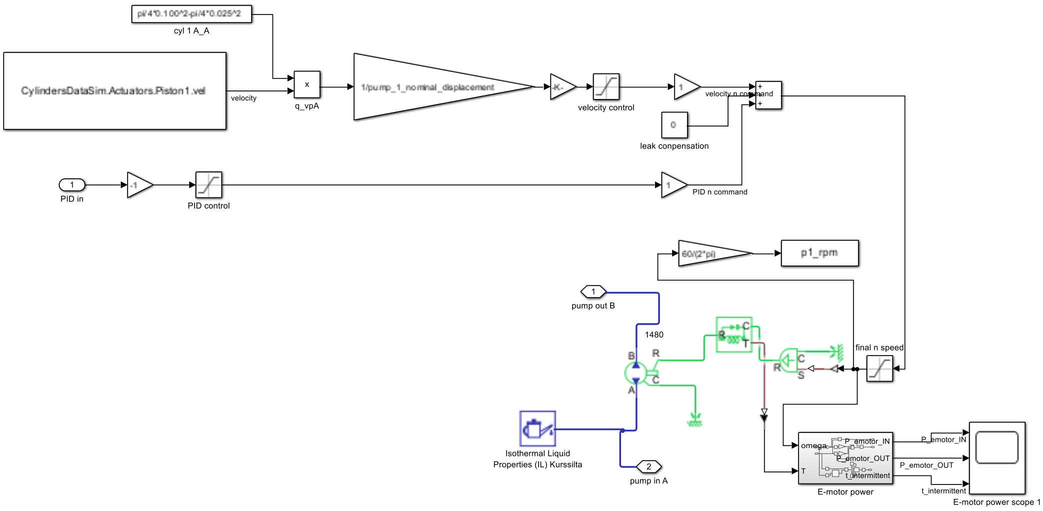

Figure A2. Pump 1 subsystem in DDH model. All the pump subsystems 1-4 are the same with differing parameter values.

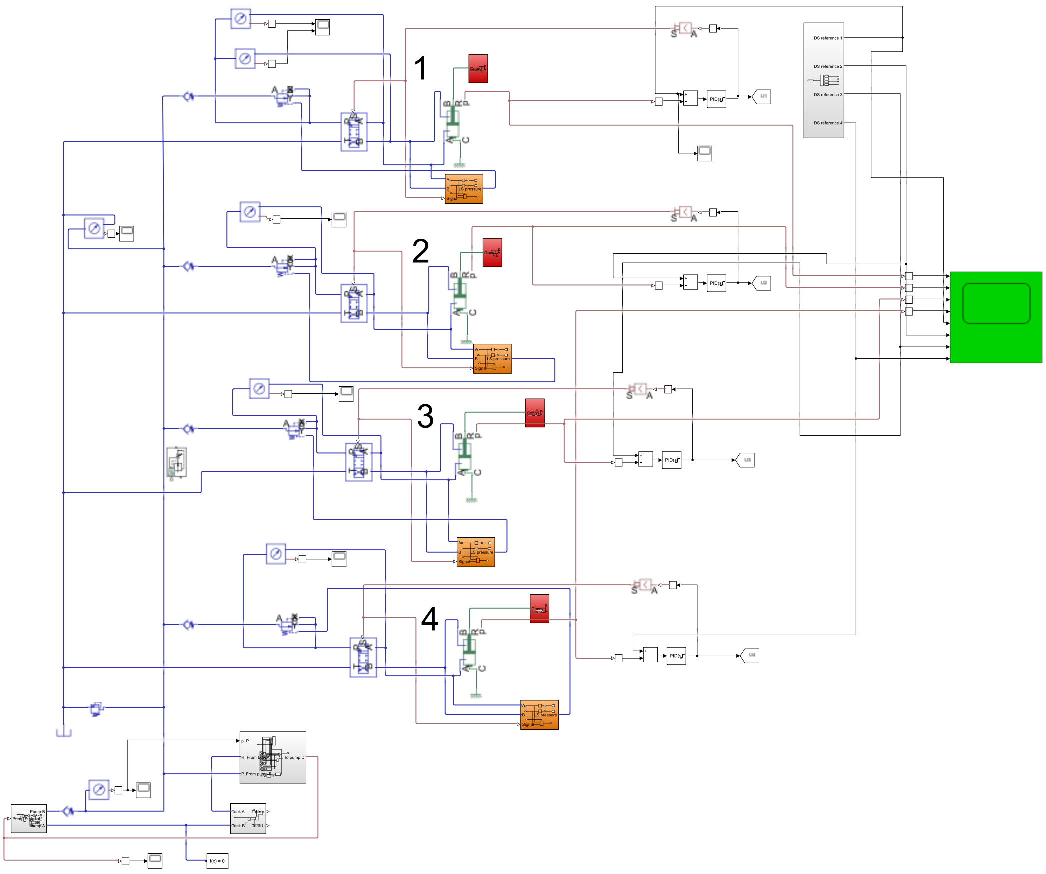

Figure A3. Top-level view of the LS system simulation model in MATLAB Simulink with Simscape.

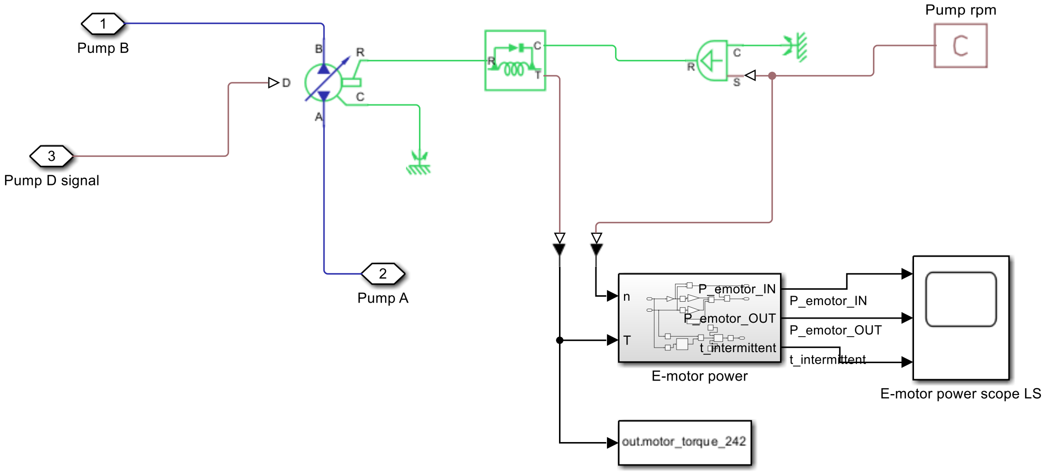

Figure A4. Pump subsystem in LS model. The pump is a variable-displacement pump.

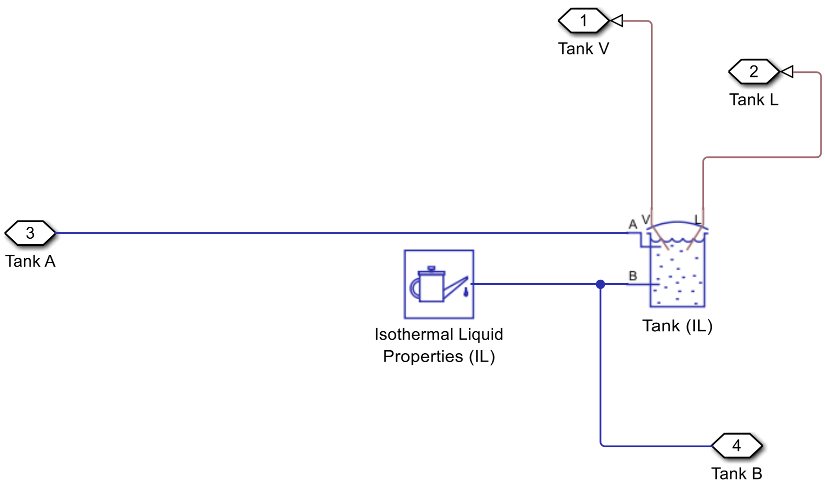

Figure A5. Tank subsystem in LS model. Hydraulic fluid properties are also given.

B. Matlab code

```
%parameters
%-----------------------
% open input command file created from piston movements.
% load downsampled_positions
load('C:\Users\topia\OneDrive - Aalto University\5 vuosi\Diplomityö\Ver-
sion1\recent_downsampled_positions.mat','downsampled_positions');
% load CylindersDataSim
load('C:\Users\topia\OneDrive - Aalto University\5 vuosi\Diplomityö\Ver-
sion1\CylindersDataFromSim.mat','CylindersDataSim');

% set simulation duration
t_end = CylindersDataSim.Actuators.Piston1.pos.Time(end);
%t_end = t_end + 1;

% Close Simulink Scope windows. Maalaa ja paina F9
shh = get(0,'ShowHiddenHandles');
set(0,'ShowHiddenHandles','On');
hscope = findobj(0,'Type','Figure','Tag','SIMULINK_SIMSCOPE_FIGURE');
close(hscope);
set(0,'ShowHiddenHandles',shh);

servo_cyl_stroke = 0.009;                                  %    Servo-cylin-
der stroke (m)          m
pump_stroke = 0.009;                                       %    Pump control
member stroke (m)       m
servo_cyl_area = 6.0e-4;                                   %    Servo-cylin-
der area (m^2)          m2
servo_cyl_preload = 1.4;                                   %    Servo-cylin-
der preload force (N)   N
servo_cyl_max_force = 62;                                  %    Servo-cylin-
der maximum force (N)   N
servo_cyl_spring_stiffness = (servo_cyl_max_force-servo_cyl_pre-
load)/servo_cyl_stroke;  %          Servo cylinder spring’s stiffness
            N/m
area_X = 2.0e-4;                                           %    Spool face
area X (m^2)            m2
area_Y = 2.0e-4;                                           %    Spool face
area Y (m^2)            m2
press_diff = 2.00e6;                                       %    Pressure dif-
ferential (Pa) (delta_pLS)          Pa
press_max = 2.80e7;                                        %    Maximum pump
pressure (Pa)           Pa
delta_p = 1.30e5;                                          %    Pressure dif-
ferential delta_p (Pa)  Pa
valve_max_area = 1.40e-5;                                  %    Valve orifice
maximum area            m2
spool_stroke = 0.005;                                      %    Spool stroke
(m)         m
Leakage_area = 1.0e-9;                                     %    3-Way valve
leakage area            m2
Orifice_area_CP = 0.2e-06;                                 %    Orifice area
CP
```

```
spring_stiffness_X = 2100/spool_stroke;                %     LS valve ac-
tuator      N/m
max_force_Y = (press_diff + delta_p) * area_X;         %     LS valve ac-
tuator      N
spring_stiffness_Y = delta_p*area_X/spool_stroke;      %     LS valve ac-
tuator      N/m
Orifice_area_LS = 6e-08;                               %     Orifice area
LS          m2

% Proportional control valve parameters (float center)
U_vector_1=[0, 0.001, 0.005];                          %valve's opening
vector PA and PB
U_vector_2=[0, 0.004, 0.010];                          %valve's opening
vector AT and BT
K_leak=0.45/60000/(0.7*sqrt(2*50e5/844.4));            %leakage parameter
K_full= 400/60000/(0.7*sqrt(2*20e5/844.4));             %full opening pa-
rameter. 40->190L/min, 35->20 bar paine ero
% A_vector=[eps, K_leak,K_full];                         %opening vector
[0, K_leak, K_full]
%
A_vector=[0.2e-7, K_leak,K_full];                         %opening vector
[0, K_leak, K_full]

% Mass for ...PID_LS.slx - unused
mass_on_piston = 500;   %kg

% Cylinders initial positions that are same as Henri's at the beginning
cyl_init_pos_1 = 0.250; % m
cyl_init_pos_2 = 0.150;
cyl_init_pos_3 = 0.050;
cyl_init_pos_4 = 0.100;

% Masses and pumps
mass1 = 10;    %kg
mass2 = 10;    %kg
mass3 = 10;    %kg
mass4 = 10;    %kg
pump_1_nominal_displacement = 12.5;     %F11-012 @12.5, 12.5cm^3/rev =
12.5e-6 m^2 = 12.5 lpm
pump_2_nominal_displacement = 12.5;     %F11-012 @12.5
pump_3_nominal_displacement = 12.5;     %F11-012 @12.5
pump_4_nominal_displacement = 4.9;      %F11-005 @4,9
pump_nominal_displacement = 92;         %PV092   @92, LS system main pump
pump_1_rpm = 9400;       % F11-012 PARKER
pump_2_rpm = 9400;       % F11-012
pump_3_rpm = 9400;       % F11-012
pump_4_rpm = 12800;      % F11-005 12800, F11-008
pump_rpm = 2300;         % PV092 @2300 --- in LS system

pump_rpm_2000 = 2000/60*2*pi;           %rad/s
pump_rpm_155 = 1480/60*2*pi;

p_A_load=mass_on_piston*9.80665/(pi/4*0.040^2);   %500 = mass

% Electric motor parameters
```

```
P_loss_fixed = 60;      % [W]
k_iron = 1.0554e-04;    % iron losses [Ws^2/rad^2]
k_copper = 3.810063;    % copper losses [W/(Nm)^2]

%save('F12_182_sim.mat','F12_182_LS_simulation')
```

Figure B1. MATLAB initialization code and parameters.

```
%% STEP 0
% Henrin parameters.m
% FORWARDER_FINAL_DataFile.m
% Pelaa peli: FORWARDER_FINAL2_xbox360.slx -> Run
% Aja tämä.
% avaa Parameters_DDH_and_LS.m -> Run
% True_DDH_model_03.slx TAI LS_model_03.slx -> Run

%% STEP 1
% Post-process Simscape log data
% Make sure simlog exists (from last simulation)
assert(exist('simlog','var')==1, 'No simlog found in workspace. Run the
model first.');

% Preallocate struct
CylindersDataSim = struct;

%The data for each piston:

% Piston 1
pos = actuator_position_cyl_1;
vel = actuator_velocity_cyl_1;
force = actuator_force_cyl_1;
power = force .* actuator_velocity_cyl_1;   % Power (force * velocity)
% käytä hydrauliikan nopeus dataa^^
CylindersDataSim.Actuators.Piston1.pos    = pos;
CylindersDataSim.Actuators.Piston1.vel    = vel;
CylindersDataSim.Actuators.Piston1.force  = force;
CylindersDataSim.Actuators.Piston1.power  = power;


% Piston 2
pos = actuator_position_cyl_2;
vel = actuator_velocity_cyl_2;
force = actuator_force_cyl_2;
power = -force .* actuator_velocity_cyl_2;   % Power (force * velocity)

CylindersDataSim.Actuators.Piston2.pos    = pos;
CylindersDataSim.Actuators.Piston2.vel    = vel;
CylindersDataSim.Actuators.Piston2.force  = force;
CylindersDataSim.Actuators.Piston2.power  = power;


% Piston 3
```

```
pos = actuator_position_cyl_3;
vel = actuator_velocity_cyl_3;
force = actuator_force_cyl_3;
power = force .* actuator_velocity_cyl_3;   % Power (force * velocity)

CylindersDataSim.Actuators.Piston3.pos    = pos;
CylindersDataSim.Actuators.Piston3.vel    = vel;
CylindersDataSim.Actuators.Piston3.force  = force;
CylindersDataSim.Actuators.Piston3.power  = power;


% Piston 4
pos = actuator_position_cyl_4;
vel = actuator_velocity_cyl_4;
force = actuator_force_cyl_4;
power = force .* actuator_velocity_cyl_4;   % Power (force * velocity)

CylindersDataSim.Actuators.Piston4.pos    = pos;
CylindersDataSim.Actuators.Piston4.vel    = vel;
CylindersDataSim.Actuators.Piston4.force  = force;
CylindersDataSim.Actuators.Piston4.power  = power;


% Save to MAT file
save("C:\Users\topia\OneDrive - Aalto University\5 vuosi\Diplomityö\Ver-
sion1\CylindersDataFromSim.mat",'CylindersDataSim');
disp('CylindersDataFromSim.mat created successfully.');




%% STEP 2
% Create Signal objects

%t = actuator_position_cyl_1.Time;
%timeSig = Simulink.SimulationData.Signal;
%timeSig.Values = timeseries(t, t);

sig1 = Simulink.SimulationData.Signal;
sig1.Values = actuator_position_cyl_1;

sig2 = Simulink.SimulationData.Signal;
sig2.Values = actuator_position_cyl_2;

sig3 = Simulink.SimulationData.Signal;
sig3.Values = actuator_position_cyl_3;

sig4 = Simulink.SimulationData.Signal;
sig4.Values = actuator_position_cyl_4;

% Combine into a Dataset
POS = Simulink.SimulationData.Dataset;
POS = [sig1; sig2; sig3; sig4];

% Optional: Save to .mat
```

```
save('C:\Users\topia\OneDrive - Aalto University\5 vuosi\Diplomityö\Ver-
sion1\POS_data_Matero_H1.mat','POS')
disp('POS_data_Matero_H1.mat created successfully.');

%% STEP 3
% downsampling the cylinder position command signal

load("C:\Users\topia\OneDrive - Aalto University\5 vuosi\Diplomityö\Ver-
sion1\POS_data_Matero_H1.mat");

% Load the POS dataset (make sure it's in the workspace)
if exist('POS', 'var') == 0
    error('POS variable not found in the workspace.');
end

% Extract time and all four position signals from POS
% Model gives negative values for pistons 2 and 4, because of hot it was
% modeled
time = POS(1).Values.Time;
position1 = POS(1).Values.Data;
position1 = position1 - 0.200; %dead volume on the A side bottom in Hen-
ri's model

position2 = POS(2).Values.Data;
position2 = abs(position2);
position2 = position2 - 0.150; %dead volume on the A side bottom in Hen-
ri's model

position3 = POS(3).Values.Data;
position3 = position3 - 0.050; %dead volume on the A side bottom in Hen-
ri's model

position4 = POS(4).Values.Data;
position4 = abs(position4);
position4 = position4 - 0.100; %dead volume on the A side bottom in Hen-
ri's model


% Define the number of desired points           <---
!!!!!!!!!!!!!!!!!!!!!!!!
num_points = 50;

% Generate the new downsampled time vector
time_down = linspace(time(1), time(end), num_points);

% Interpolate the position data to match the new time vector
% Subtract dead volume of each cylinder in Henri's model
position1_down = interp1(time, position1, time_down, 'pchip');
position2_down = interp1(time, position2, time_down, 'pchip');
position3_down = interp1(time, position3, time_down, 'pchip');
position4_down = interp1(time, position4, time_down, 'pchip');

% Combine downsampled signals into one variable
downsampled_positions = [time_down', position1_down', position2_down',
position3_down', position4_down'];
```

```
save('C:\Users\topia\OneDrive - Aalto University\5 vuosi\Diplomityö\Ver-
sion1\recent_downsampled_positions.mat', "downsampled_positions")

disp('recent_downsampled_positions.mat created successfully. Displaying
Figure...');

% Display the downsampled signals (all in one figure)
figure;
plot(time, position1, '-b', 'DisplayName', 'Position 1 - Original');
hold on;
plot(time_down, position1_down, '--bo', 'DisplayName', 'Position 1 -
Downsampled');

plot(time, position2, '-r', 'DisplayName', 'Position 2 - Original');
plot(time_down, position2_down, '--ro', 'DisplayName', 'Position 2 -
Downsampled');

plot(time, position3, '-g', 'DisplayName', 'Position 3 - Original');
plot(time_down, position3_down, '--go', 'DisplayName', 'Position 3 -
Downsampled');

plot(time, position4, '-m', 'DisplayName', 'Position 4 - Original');
plot(time_down, position4_down, '--mo', 'DisplayName', 'Position 4 -
Downsampled');

% Customize plot appearance
xlabel('Time (s)');
ylabel('Position');
title('Downsampled Position Signals (4 Signals)');
legend('show');
grid on;
hold off;

%% Next step
% avaa Parameters_DDH_and_LS.m
% DDH_model_01.slx TAI LS_model_01.slx
% Run
```

Figure B2. Steps to post-process Simscape log data, create signal objects file, create down sampled cylinder position command signal.