Modeling and Hardware-in-the-Loop (HIL) Simulation of an Intelligent Electro-Hydraulic System for a Wheel Loader

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ABSTRACT

Wheel loaders are versatile earth moving construction equipments. In addition to the drive train, a significant portion of the power generated by a diesel engine is distributed through hydraulics. Emission regulation and high cost of fuel have driven OEMs to develop the next generation hydraulic system with the same performance but with improved fuel efficiency.

In this paper, an intelligent electro-hydraulic valve system used in wheel loaders is presented. The uniqueness of the valve system is Valvistor technology. The pilot operated proportional poppet valve has the large flow rating and automatically reduce the throttling loss. Four Valvistors connect to each work port of a hydraulic cylinder to pressure and tank separately. The decoupling of the meter-in and meter-out flow for tilt and lift work functions significantly improves operational efficiency and control flexibility. This is in particular important for mid to large size wheel loaders. Moreover, the sensors and the microcontrollers are embedded in each function and integrated via the network. Such modular design eases not only mechanical integration, but also software development and validation. The distributed sensors also enable electronically controlled load sensing technology that intelligently manages power based on the transmitted pressure sensor signals. All of these features are integrated together for improving efficiency.

For such a complex system that can only function with all the controllers in place, it is quite difficult and costly to develop and validate the software on the actual vehicle platform. Hardware-in-the-loop (HIL) platform would be very useful to serve as a virtual system in a wheel loader interacting with electronics and ECU. Development, assessment and validation of control and diagnostic algorithms being conducted on such a platform would save time, man power, and cost. In this paper, modeling approaches for the above hydraulic components and system are presented. The HIL platform that enables hardware in the loop validation for the wheel loader intelligent hydraulic system is presented in detail.

1. Introduction

Wheel loaders have been widely employed in digging, transporting, and loading of construction materials. The short loading cycle (V-cycle) is typically used to represent the wheel loaders' characteristic driving pattern in many applications. The short loading cycle consists of several phases: filling bucket, leaving bank, retardation and reversing, driving to dump truck, emptying bucket, leaving dump truck, retardation and reversing, driving to bank, retardation to bank.

In order to meet the typical duty cycle, wheel loaders need the combination of various functions of driving, steering, and lifting. The power generated from a diesel engine needs to be split into either the transmission path to achieve the driving function, or the hydraulic path for the steering and working functions. A typical power usage profile for various phases in a short loading cycle can be seen in **Figure 1** [Filla, 2005]. Regardless of significant variation from one phase to another, it is clear that the hydraulic system accounts for a significant portion of total power consumption. As the wheel loader market has continuously driven for improving fuel efficiency and reducing emissions, the hydraulic system in a wheel loader has remained unchanged for decades. There is an ever increasing need to improve hydraulic efficiency.

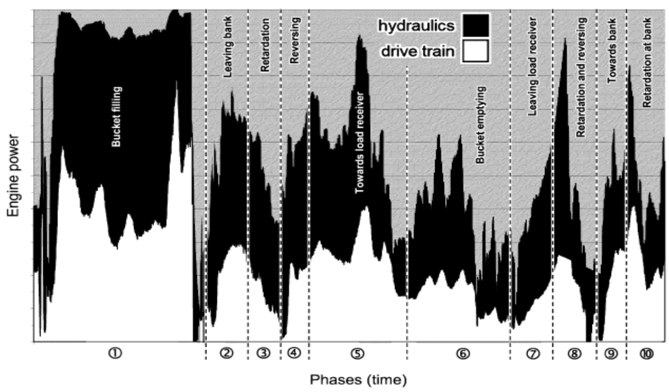


Figure 1: Power profile (hydraulic vs drive train) for a short loading cycle

One of the major causes for the energy loss in the conventional system is the single power source needing to meet multiple load requirements. The mismatch of the pressure and flow leads to throttling loss. In this paper, an intelligent electro-hydraulic valve system used in wheel loaders is presented in which by combining Valvistor and other technologies, the hydraulic system throttling loss can be dramatically reduced. A Valvistor [Eaton, 1999] [Eriksson, 2007] is a valve with high flow rating with a pilot stage and a main stage (a poppet). Its main poppet moves proportionally to the pilot flow, which can be directly controlled by an electromagnetic actuator. Therefore, the Valvistor acts as a flow amplifier, much like a transistor in the electronics world. High flow rate with the small pressure drop allows Valvistor to be good candidate to reduce throttling loss. Secondly, similar to Ultronics, a spool valve that can decouple meter-in and meter-out flow [Yuan and Lew, 2005], four valvistors can connect to a hydraulic cylinder with the independent control. A typical configuration is in “H” bridge format. The decoupled meter-in and meter-out flow increases power management flexibility, and enabling power and flow regeneration. For example, previous researches have shown 25% efficiency improvement for the short loading cycle when some of the proposed technologies are employed [Eriksson, 2007]. Moreover, the pressure sensors and spool position sensors can be embedded in the system, thus simplifying integration drastically as well as enabling the electronically controlled load sensing system that intelligently manages the power requirement from the steering and work circuit. All these features are integrated together for improving efficiency.

For such a complex electro-hydraulic system, however, the software development efforts are tremendous. The control algorithms are needed for every individual function, and should work seamlessly for the entire system. The diagnostics and fault tolerance control are much preferred by customers to improve reliability and safety. A rigorous assessment and validation process is undoubtedly necessary to guarantee the software quality and coverage. Hardware in the Loop (HIL) is a proven validation process in which an ECU equipped with the developed software will interact, not with the actual hydraulic system in a wheel loader, but with a “virtual” hydraulic system being modeled in a real time simulator. In this case, without the presence of the real wheel loader, one can put the plant models into a HIL simulator, which transmits the pressure and position signals to the ECU and receives the corresponding actuation command from the ECU. Obviously, developing a robust and fast executable plant model is very important to support HIL testing. There have been some research efforts in developing hydraulic circuits for a wheel loader using Bond Graph [Remero, 2008] and other software tools [Park 2009]. However, these models are not suitable for HIL process. In this paper, the detailed models for various components in wheel loader hydraulic system are presented. It is critical to keep in mind in the early phase that these models will be executed in real time eventually, thus a trade off between fidelity and simplicity must be taken into account.

The paper is organized as follows. First, the hydraulic architecture is presented, followed by the description of the models for the key components. The models consist of electronic load sensing pump, priority valve, hydraulic steering unit, and lift/tilt valves. The wheel loader hydraulic circuit also consists of auxiliary circuits which will not be covered here. The controllers have been developed based on these models, and the simulation results of the entire system are presented. Finally, the system architecture for the Hardware in the Loop simulation is discussed.

2. ElectroHydraulic Architecture

This section explains the architecture of the wheel loader hydraulic circuit, description of different components and subsystems. **Figure 2** shows the higher level schematic diagram of the wheel loader hydraulic circuit. The hydraulic circuit contains hydraulic steering unit, lift circuit, tilt circuit, pump controller, load sensing pump and priority valve. Other auxiliary circuits are not included.

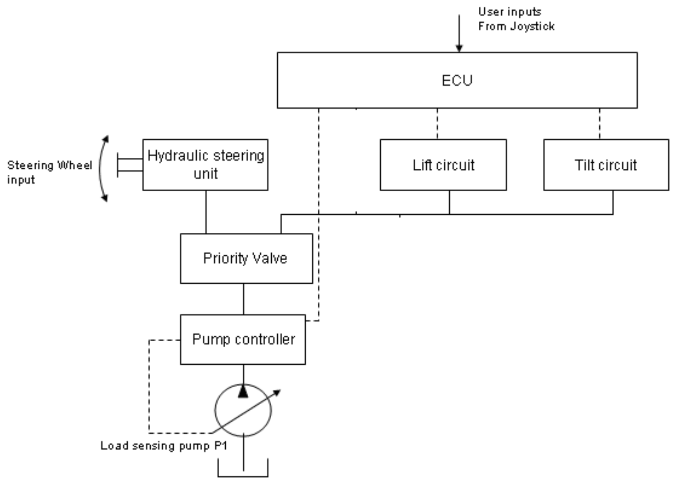


Figure 2: Schematic diagram of wheel loader hydraulic circuit

The load sensing pump is a variable displacement pump which supplies flow to load. Load sensing pump P1 supplies flow to the steering circuit, as well as to the lift or tilt circuit if there is no demand from the steering unit. It is the priority valve which prioritizes the flow to steering from the work circuit. Pump controller controls pressure at pump outlet to match the load request by electronically controlling load sensing pressure line connected to the pump.

The Electronic Control Unit (ECU) in Figure 2 is shown in a centralized manner, but in reality it is implemented as a distributed architecture with several microcontrollers that are collocated with each function. It calculates command signals to various valves in different subsystems, from the demands received from operator and measurements from the sensors. The details of the lift circuit, tilt circuit and steering unit are discussed in following sections.

3. Dynamic Modeling of ElectroHydraulic Circuit

Since the focus of the research is to develop and verify a complex intelligent hydraulic system for wheel loaders. It is important to capture both the steady state and dynamic behavior of the system when developing the models. The details can be seen as follow.

3.1. Electronically Load Sensing Pump

The load sensing pump reduces the power losses in the hydraulic circuit, yielding higher efficiency. It is a variable displacement pump [Schoenau 1990] which senses the load and adjusts flow rate and pressure demanded by the load. The flow rate is controlled by varying the swash plate angle. **Figure 3** shows the load sensing pump and its control circuit. The control elements which control flow rate are the pressure flow compensator, the control piston and the swash plate. The pressure flow compensator spool is controlled by the pressure differential between the load sense pressure and supply pressure at the pump outlet. The pressure compensator regulates the flow in and out of the control piston which controls the motion of the swash plate angle.

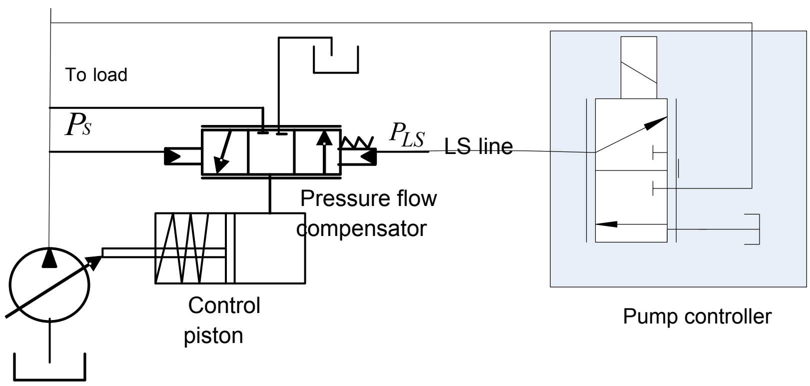


Figure 3: Load sensing pump and its control circuit (unlike the conventional LS pump, LS line connects to the pump controller where a valve is used to control LS line pressure)

The ideal average pump output flow rate is given as

$$Q_{pump} = \frac{V_d n}{60} \tag{1}$$

Whereas volumetric displacement is calculated as

$$V_d = 2A_P RN \tan\alpha \tag{2}$$

The spool displacement of the pressure flow compensator is calculated as

$$M_f \frac{dx_f^{\,2}}{dt^2} = (P_S - P_{LS})A_f - B_f \frac{dx_f}{dt} - K_f x_f - F_{0f} \tag{3}$$

Flow going to the control piston through the pressure flow compensator is calculated as

$$Q_{in\_comp} = C_d A_1(x_f)\sqrt{\frac{2(P_S - P_c)}{\rho}} \tag{4}$$

Flow coming out of the control piston to the tank is calculated as

$$Q_{out\_comp} = C_d A_2(x_f)\sqrt{\frac{2P_c}{\rho}} \tag{5}$$

The pressure developed in the control piston is calculated as

$$\frac{dP_c}{dt} = \frac{\beta}{V_{0c} + A_c x_c}\left(Q_{in\_comp} - Q_{out\_comp} - \frac{dV_c}{dt}\right) \tag{6}$$

The displacement of the control piston is calculated as

$$M_c \frac{dx_c^{\,2}}{dt^2} = P_c A_c - B_c \frac{dx_c}{dt} - K_c x_c \tag{7}$$

The swash plate angle is determined by the movement of the control piston is approximated as

$$\alpha = \tan^{-1}\left(\frac{x_c}{L_c}\right) \tag{8}$$

Conventionally, to transmit the pressure information, load sensing lines run from the valve work port to the control input of the variable displacement pump. For the electronic load sensing system, the pressure transducer measure the pressure at the valve work port and transmit the load pressure signal electronically to the pump controller that converts it to a hydraulic representation locally via the electronic control. In this way, the system is of more accuracy, greater stability, increased efficiency, and higher flexibility. There are many different implementations of electronic load sensing pumps. One example is a proposal [Backe 1991] in which the original compensator has been replaced by a new proportional valve. However, a more robust and simple architecture keeps the compensator structure as described above, but adds a valve to generate LS signal [Eaton Corp, 2010] . In this way, the hydraulic load sensing mechanism proven in the past decades can be leveraged.

3.2. Priority Valve

The priority valve is used for prioritizing flow to the steering unit from load sensing pump P1. **Figure 4** shows the schematic symbol of static load sensing priority valve. The operation of the priority valve depends on the pressure differential between steering circuit supply pressure P_{CF} and load sense pressure coming from steering unit P_{LSS}.

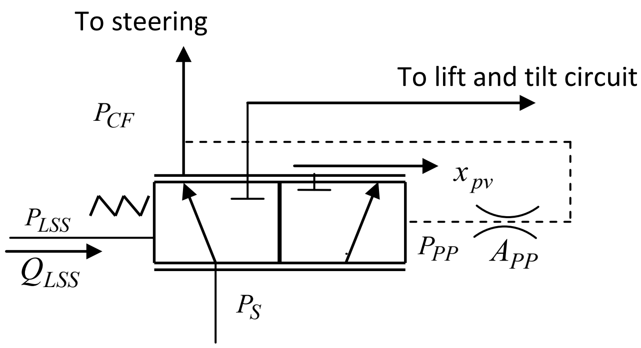


Figure 4: Schematic diagram of static load sense priority valve

The flow going to PP chamber is determined as

$$Q_{PP} = C_d A_{PP} \sqrt{\frac{2(P_{CF} - P_{PP})}{\rho}} \tag{9}$$

The spool displacement is calculated as

$$M_{pv} \frac{dx_{pv}{}^2}{dt^2} = (P_{PP} - P_{LSS}) A_{pv} - B_{pv} \frac{dx_{pv}}{dt} - K_{pv} x_{pv} - F_{0pv} \tag{10}$$

Pressure at PP port is computed as

$$\frac{dP_{PP}}{dt} = \frac{\beta}{V_{0PP} - A_{pv} x_{pv}} \left(Q_{PP} + \frac{dV_{PP}}{dt} \right) \tag{11}$$

Pressure at P_{LSS} port is computed as shown below.

$$\frac{dP_{LSS}}{dt} = \frac{\beta}{V_{0LSS} + A_{pv} x_{pv}} \left(Q_{LSS} - \frac{dV_{LSS}}{dt} \right) \tag{12}$$

There are two flow paths, one is from pump to steering circuit called Control Flow (CF), and other is from pump to work circuit called Excess Flow (EF) path.

The two flow rates are calculated as

$$Q_{CF} = C_d A_{CF}(x_{pv})\sqrt{\frac{2(P_S - P_{CF})}{\rho}} \tag{13}$$

$$Q_{EF} = C_d A_{EF}(x_{pv})\sqrt{\frac{2(P_S - P_{EF})}{\rho}} \tag{14}$$

3.3. Hydraulic Steering Unit

Hydraulic steering unit is used to steer the wheel loader to right or left as demanded by the operator. It consists of steering valve, gerotor, steering cylinders, check valves and relief valves [Eaton, 2006] [Yuan, 2008]. The steering valve spool is attached to the steering wheel which is controlled by the operator. The steering wheel rotation controls the fluid flow from the pump to the steering cylinder.

The steering valve is the most critical component. **Figure 5** shows the simplified representation of the steering valve connected to the cylinder, when it is steered to the right. The gerotor is a hydraulic motor used for smooth operation of steering unit. It contains different orifices to control the flow from the pump to the steering cylinder. The area of openings A_1, A_4, A_5, A_{LSD}, A_{LS} depend on the operator steering wheel.

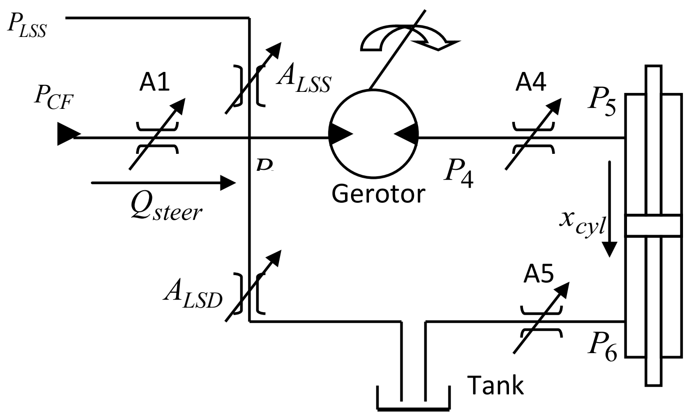


Figure 5: Simplified representation of steering valve

The pressure P_{CF} is calculated as

$$\frac{dP_{CF}}{dt} = \frac{\beta}{V_{0CF}}\left(Q_{CF} - Q_{steer}\right) \tag{15}$$

The flow going to the steering unit is calculated as

$$Q_{steer} = C_d A_1\sqrt{\frac{2(P_{CF} - P_3)}{\rho}} \tag{16}$$

The pressure P_3 is calculated as

$$\frac{dP_3}{dt} = \frac{\beta}{V_{01}}\left(Q_{steer} - Q_{LSS} - Q_{LSD} - Q_{in\_cyl}\right) \tag{17}$$

The flow rate for the load sense and load sense drain are calculated as

$$Q_{LSS} = C_d A_{LSS} \sqrt{\frac{2(P_3 - P_{LS})}{\rho}} \tag{18}$$

$$Q_{LSD} = C_d A_{LSD} \sqrt{\frac{2P_3}{\rho}} \tag{19}$$

The gerotor is replaced with a constant orifice in the simulation. The flow rate into the cylinder port is calculated as

$$Q_{in\_cyl} = C_d A_{4eq} \sqrt{\frac{2(P_3 - P_5)}{\rho}} \tag{20}$$

The pressure P_5 developed in the cylinder piston head side chamber is calculated as

$$\frac{dP_5}{dt} = \frac{\beta}{V_{0h} + A_h x_{cyl}}\left(Q_{in\_cyl} - \frac{dV_h}{dt}\right) \tag{21}$$

The flow rate going out of the cylinder chamber is calculated

$$Q_{out\_cyl} = C_d A_5 \sqrt{\frac{2P_6}{\rho}} \tag{22}$$

The pressure P_6 is calculated as

$$\frac{dP_6}{dt} = \frac{\beta}{V_{0r} - A_r x_{cyl}}\left(\frac{dV_r}{dt} - Q_{out\_cyl}\right) \tag{23}$$

3.4. Lift/Tilt Circuit

The lift and tilt cylinders are connected to the wheel loader's lift arm, lever, and other attachment. The motion of the cylinders leads to the corresponding motion of lift arm and bucket. The wheel loader contains two cylinders for each of the service.

As aforementioned, Valvistors are the key components used in this subsystem. "Valvistor" is actually derived from "valve" and "transistor". In a Valvistor, a main poppet amplifies a small flow through the pilot valve. The Valvistors can be categorized as A type or B type. For A type, the fluid flows from the end of the poppet (port A) to the side (port B); while for B type the fluid flows from the side of poppet (port B) to the end (port A). The Valvistors can also be further designed to allow or disallow reverse flow [Eaton, 1999]. Two Valvistors' used in the system are illustrated In **Figure 6**.

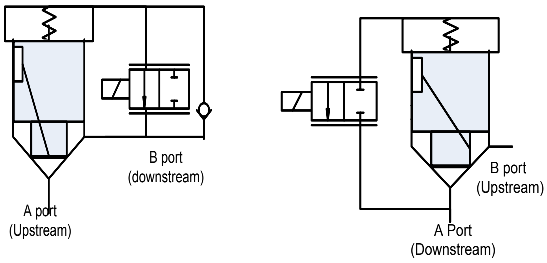


Figure 6: Valvistor configurations. Left: A type (flow from A port to B port) without the reverse flow; Right: B type (flow from B port to A port) allowing the reverse flow.

Regardless of the type of Valvistor, the working principle is pretty similar: the motion of the poppet is controlled via a pilot valve. The pilot valve is actuated by a solenoid. The spool displacement in the pilot stage will proportionally displace the main poppet. Without the loss of generality, the dynamics of a A type Valvistor is described as follows:

The pressure in the upper chamber of the Valvistor is determined by variable volume dynamics

$$\frac{dP_{top}}{dt} = \frac{\beta}{V_{0mt} - A_{mt}x_m}\left(Q_{slot} - Q_{pilot} + \frac{dV_{mt}}{dt}\right) \tag{24}$$

The flow going through the slot is calculated as

$$Q_{slot} = C_d A_{slot}(x_m)\sqrt{\frac{2(P_{in} - P_{top})}{\rho}} \tag{25}$$

The displacement of the main poppet is calculated as

$$M_m \frac{dx_m^{\ 2}}{dt^2} = P_{in}A_{ms} - P_c A_{mc} + P_{out}\left(A_{mc} - A_{ms}\right) - K_m x_m - B_m \frac{dx_m}{dt} - FF_m \tag{26}$$

The flow going from the inlet to the outlet is calculated as

$$Q_{out} = C_d A_{out}(x_m)\sqrt{\frac{2\left(P_{in} - P_{out}\right)}{\rho}} \tag{27}$$

The total output flow is the sum of the pilot flow and the flow coming from the inlet chamber i.e.

$$Q_{out\_tot} = Q_{out} + Q_{pilot} \tag{28}$$

It is worth mentioning that there is a check valve added into the type A Valvistor as shown in Figure 6. This prevents the fluid from reversely flowing from port B to port A if the port B's pressure is higher. Due to the check valve, the high pressure of Port B is able to propagate to the top chamber, thus locking the poppet in the closed position. It implies that if the cylinder port pressure is higher than the supply pressure, there will be no risks of falling back the load. It is a simple manner to improve the system safety. On the other hand, for Type B Valvistor used connecting the tank to the work port of the cylinder, the reverse flow is allowed. It is implied that the tank can back fill the work chamber if the tank is pressure is higher than the work port pressure. Since not all of the fluid needs to be supplied from the pump, it is a convenient way to improve the fuel efficiency.

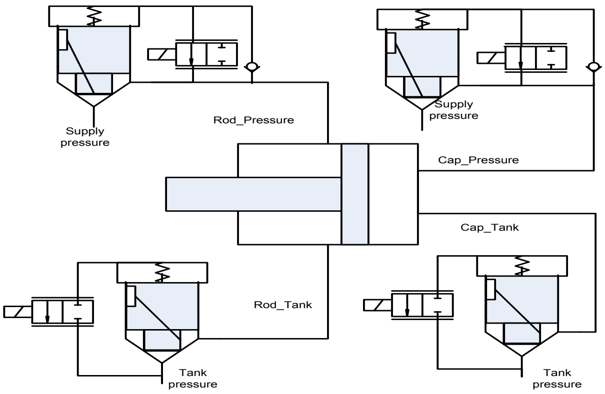


Figure 7: Hydraulic circuit for lift function

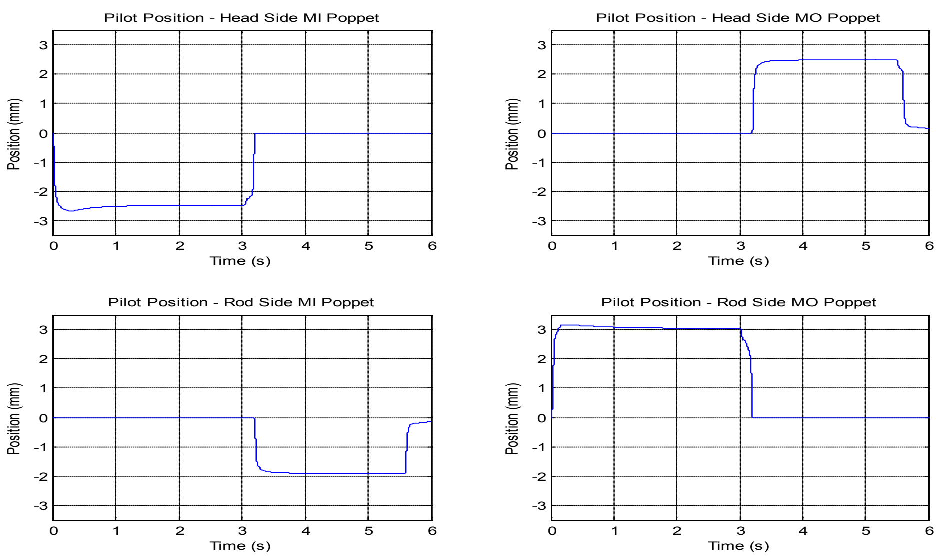
Figure 7 shows the schematic diagram of the lift circuit, which contains a set of valves controlling flow in and out of cylinder. The fundamental building block for the valve is the Valvistor. Two A-type valvistors without the reverse flow connect the work ports to the supply line, while the other two B-type valvistors allowing the reverse flow connect the work ports to the tank. In other words, each cylinder port is connected with one A-type and one B-type. Even though a similar independent metering architecture can be seen in [Eriksson, 2007] [Heybroek, 2008], the main differentiation here are 1) the Vavistors with several configurations are used in either pressure port or tank port to support the additional safety feature; 2) various fuel saving features can be enabled by high flow rated Vavistor valves; 3) the commercially available pressure and position sensors are directly embedded into the system, thus enabling more robustness and control flexibility from the system;

4. Simulation Results

The above equations represent all of the components of the electro hydraulic circuit of the wheel loader. These equations are built and solved in Matlab/Simulink. The models are calibrated from experimental data. For example, one critical parameter in the models is the function of the flow rate as a function of pressure drop, the spool opening, and the temperature. Such maps are developed from the bench testing. It is worth mentioning that the models may not reflect all the dynamic behaviors captured in testing. One particular consideration is that the models should be able to run in real time. The trade-off between fidelity and simplicity is a key to make the HILS practical and useful.

Once the plant models are in place, then the controller for each module is developed correspondingly. Different control modes, such as pressure control and flow control, are defined for various modules. Pressure control involves comparing the pressure command with actual pressure measured by sensor, which is given to the controller which will then generate a PWM signal for the pilot valve of a Valvistor. In the flow control strategy, a flow command is converted to a position command for a given pressure drop using position-to-flow maps.

The control algorithms interact with the plant models. The control algorithms take the input from operator, and the sensor signals from the plant, and calculate the desired PWM command, which is then transmitted to plant so that the response of the hydraulic system for the next time step will respond accordingly. A co-simulation result for Lift circuit in the closed loop manner can be seen in **Figure 8**.



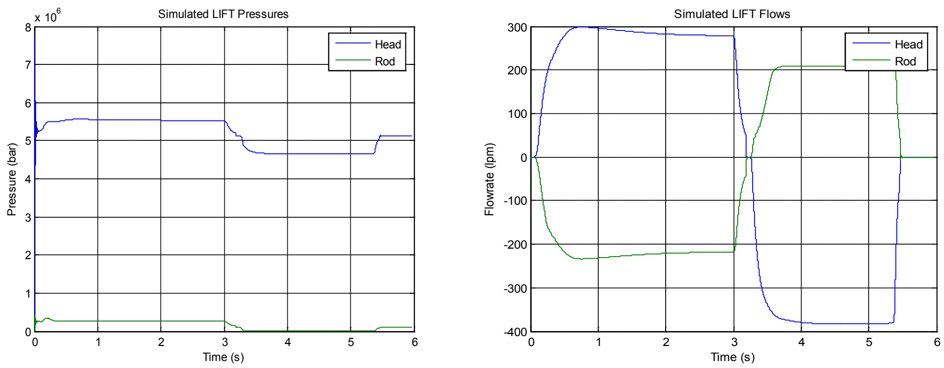


Figure 8: Simulation results for lift cylinder

5. Hardware in the Loop (HIL) Simulation

In this section, we will discuss the implementation of the above developed models into the Hardware in the Loop simulator. The main objective is to verify the software codes for the intelligent hydraulic system solution without the need of having a real wheel loader.

As can bee seen in **Figure 9**, the HIL simulation system consists of three parts: a Personal Computer, a (customerized) ECU, and the hardware in the loop simulator (HILS). In the following section, we will introduce each portion's function in detail.

5.1. HIL Simulation System

1) PC. The PC has multiple functions in a HILS system. The most basic purpose is to load Matlab/Simulink and Dspace HILS IO library and to build the executable for HILS. See Figure 10 [top] for such an example. The PC has the high speed cable connected to the HILS in order to download the executable to the HIL simulator. The PC then commands execution to begin.

In addition, through Control Desk software, the PC is able to monitor plant parameters in real time. Control Desk is a robust virtual instrument platform that integrates dSPACE with Matlab. Essentially the plant model that has been built for the HIL simulator is imported into Control Desk and real-time access to plant variables is granted to the user. The user is then able to construct visually intuitive GUIs for the application being run in real time on the dSPACE simulator. In addition to monitoring parameters in the plant, the user can also change parameter values in the system in real time. This grants the ability to perturb the plant, inject disturbances, and otherwise observe the robustness of the controller in real time. In **Figure 10** [mid], a GUI interface for one function has been illustrated.

Figure 9: The picture of the HILS system setup. Left: PC + ECU + HILS; Right: the interface between HILS and ECU.

Moreover, a PC based software is downloading the embedded software codes to the ECU, and monitoring and configuring the control parameter and signals through a J-1939 CAN interface.

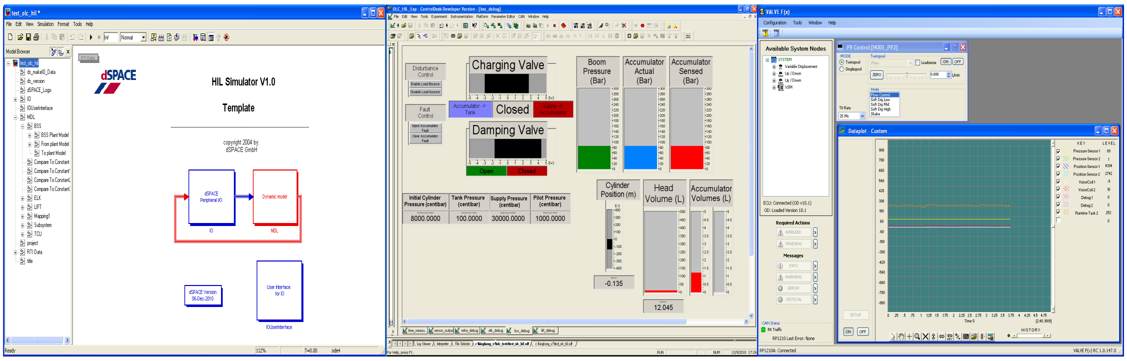


Figure 10: Screenshot of the software graphic interface in PC: Left: Plant model and HILS IO library in Matlab/Simulink; Middle: a control desk virtual instrument; Right: configuration software GUI for ECU;

2) HIL Simulator: The HIL simulator is manufactured by dSPACE. This simulator includes standard IOs, such as ADC, DAC, PWMin, PWMout, Resistance simulation, DigitalIn, DigitalOut, and CAN. In addition, the simulator has the extended conditional boards for LVDT sensor simulation, mV sensor simulation, and current monitoring and measurement. The combination of the standard IOs and customized functions allow this HIL simulator be applied on a variety of applications.

The ECU includes more than one microprocessor, each of which corresponds to different function for the hydraulic system, such as steering, lift, tilt, and so on. In the lower right corner of Figure 9, three nodes are illustrated.

5.2. Sensor and Plant Real Time Simulation

Since the pressure sensors and the position sensors are wired to the ECU, it has been given particular consideration to verify the sensor interfaces in this HILS study.

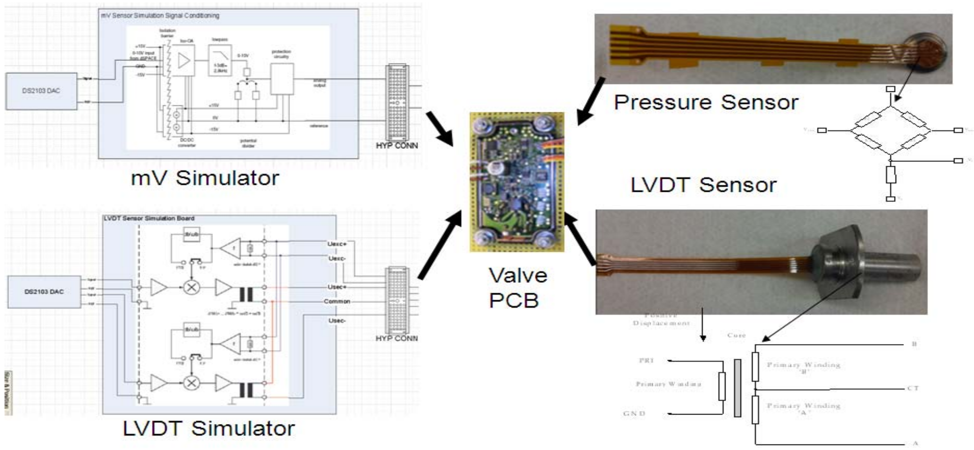


Figure 11: Sensors in a EH system can be simulated in HILS

Figure 11 shows the overall approach of simulating various sensors. Obviously the working principles are different for different sensors. For instance, a LVDT position sensor employs the principle of inductance. It consists of one primary coil and two secondary coils. The primary coil is excited with a sinusoidal signal with a certain frequency and amplitude which will generate the sinusoidal responses out of two secondary coils via the core. The core is physically attached to a spool and thus is indicative of the spool position. In the center position, the identical amplitude of the sinusoidal response is expected from the secondary coils. As the core moves away from the center, one secondary coil will generate the response with higher amplitude than the other. Traditionally, it is not easy to simulate the LVDT raw signals for various spool position. The HIL simulator contains a LVDT board with the similar modules – one primary coil and two secondary coils. There are ADC signals from DS2211 that can control the amplitudes of the two responsive sinusoidal signals to simulate the changing position. The ADC value is then assigned based on the plant model real time output. In this manner, we use HILS to simulate both plants and sensors.

The ECU responds to these sensor inputs by driving PWM signals to actuate the pilot spools for any valves it is intended to control.

It is worth mentioning that there is a great challenge for the hardware in the loop simulation for a complex system. On the one side, since we are addressing a intelligent electro-hydraulic system, the dynamics of the hydraulic components have to be considered to understand the performance with a reasonable fidelity. On the other hand, the system gets quite complex when adding the multiple functions together. Due to the processing capability constraints, the sample rate would get slower when more models have to be considered. When the sample rate reaches a threshold, the system would potentially have the numerical stability issue. The trade-off between fidelity and numerical stability have to be carefully conducted.

One of the advantages of using the HIL setup is the ability to accelerate development and validation of the control algorithm software without the actual hardware. The engineer can make changes to the controller algorithms, and see how it affects the plant behavior immediately. In addition, the engineer can perturb plant variables within the plant model to understand performance sensitivity. This allows the engineer to measure robustness of the controller under a set of operating conditions.

6. Conclusion

Wheel loaders are versatile earth moving construction equipment. Their fuel efficiency is becoming increasingly critical due to regulation and rising fuel costs. Hydraulics accounts for nearly 50% of their total fuel consumption. In this paper, an intelligent electro-hydraulic valve system that could improve efficiency by leveraging the Valvistor and electronically controlled load sensing system is presented. The focus of the paper is not to discuss the details of the function. By contrast, the authors are mainly concerned the software development process for such a complex system. Hardware-in-the-loop (HIL) platform would be very useful to serve as a virtual wheel loader interacting with electronics and ECU. In the paper, the authors present the HILS process deployed for a wheel loader electrohydraulic system. The development, assessment and validation of diagnostics and control algorithms have been conducted on such a platform. The codes developed in HILS have been successfully migrated into an actual wheel loader system with significant saving on time, man power, and cost.

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8. Nomenclature

A_f	Spool end area of pressure flow compensator spool	m^2
A_P	Cross sectional area of Piston	m^2
$A_1(x_f)$, $A_2(x_f)$	Area of openings in pressure flow compensator.	m^2

A_c	Control piston spool end area	m^2
A_{PP}	Orifice area connecting between P_{CF} and P_{PP}	m^2
A_{pv}	Spool end area of priority valve	m^2
A_{EF} , A_{CF}	Area of openings in priority valve	m^2
A_1 , A_4 , A_5 , A_{LSS} , A_{LSD}	Area of openings in steering valve	m^2
A_{4eq}	Equivalent orifice area of A_4 , Gerotor restriction	m^2
A_h , A_r	Cylinder piston head, rod side cross sectional area	m^2
A_{1p} , A_{2p} , A_{3p} , A_{4p}	Area of openings in Ultronics pilot valve	m^2
A_i	spool end area of Ultronics main stage valve	m^2
A_{1i} , A_{2i}	Area of openings in Ultronics main stage valve	m^2
A_{ms}	Valvistor poppet inlet surface area	m^2
A_{mc}	Valvistor poppet top chamber surface area	m^2
A_{mt}	Valvistor poppet load side surface area	m^2
A_{slot}	Orifice area of slot in Valvistor poppet	m^2
A_{out}	Area of opening in Valvistor	m^2
B_f	Damping coefficient of pressure flow compensator	kg/s
B_c	Viscous damping coefficient of control piston	kg/s
B_{pv}	Viscous damping coefficient of priority valve	kg/s
B_m	Viscous damping coefficient of Valvistor	kg/s
C_d	Discharge coefficient	
F_{0f}	Spring pre load in pressure flow compensator	N
F_{0pv}	Spring pre load force in priority valve	N
$Fvcoil_p$	Force produced by voice coil	N
$Fpre_{ib}$, $Fpre_{ia}$	Spring preload force in Ultronics main stage valve	N
FF_i	Flow forces in Ultronics main stage valve	N
FF_m	Flow forces on Valvistor poppet	N
K_f	Spring rate of the pressure flow compensator	N/m
K_c	Spring rate in the control piston	N/m
K_{pv}	Spring rate in priority valve	N/m
K_p	Equivalent spring rate in Ultronics pilot valve	N/m
K_{ib} , K_{ia}	Spring rates in Ultronics main stage valve	N/m
K_m	Spring rate in Valvistor	N/m
L_c	Perpendicular distance between shaft axis and control piston center axis	m
M_f	Mass of the pressure flow compensator spool	Kg
M_c	Mass of control piston	Kg
M_{pv}	Mass of priority valve spool	Kg
M_p	Mass of Ultronics pilot valve spool	Kg
M_i	Mass of Ultronics main stage valve spool	Kg
M_m	Mass of Valvistor poppet	Kg
n	Pump shaft speed	Rad/s
N	Number of pistons	Piston
P_S	Supply pressure developed at pump outlet	Pa

P_{LS}	Load sense pressure coming from load	Pa
P_c	Pressure developed in control piston chamber	Pa
P_{PP}	Priority valve PP chamber pressure	Pa
P_{LSS}	Priority valve LSS chamber pressure	Pa
P_{CF}	Steering circuit supply pressure	Pa
P_{EF}	Pressure downstream of priority valve connected to Lift, Tilt circuit	Pa
P_3, P_5, P_6	Intermediate pressures in steering valve	Pa
P_{in1}, P_{in2}, P_{out1}, P_{out2}	Input port, output port pressure of Ultronics main stage valve.	Pa
P_{top}	Valvistor top chamber pressure	Pa
P_{in}	Valvistor poppet inlet chamber pressure	Pa
P_{out}	Valvistor poppet outlet chamber pressure	Pa
Q_{in_comp}	Flow rate coming into control piston	m^3/s
Q_{out_comp}	Flow rate going to tank from control piston	m^3/s
Q_{pump}	Average flow rate supplied by load sensing pump	m^3/s
Q_{EF}	Excess flow rate from priority valve	m^3/s
Q_{CF}	Flow rate from priority valve to steering unit	m^3/s
Q_{PP}	Flow rate going to PP chamber of priority valve	m^3/s
Q_{steer}	Flow rate to steering unit	m^3/s
Q_{LSS}	Flow rate going to LSS chamber of priority valve	m^3/s
Q_{LSD}	Load sense drain flow rate from steering unit	m^3/s
Q_{in_cyl}	Flow rate going in to cylinder	m^3/s
Q_{out_cyl}	Flow rate coming out of cylinder	m^3/s
Q_{1p}, Q_{2p}, Q_{3p}, Q_{4p}	Flow rate at different ports in Ultronics pilot valve	m^3/s
Q_{1i}, Q_{2i}	Flow rate at ports of Ultronics main stage valve	m^3/s
Q_{slot}	Flow rate through slot in Valvistor poppet	m^3/s
Q_{pilot}	Pilot flow rate in Valvistor	m^3/s
Q_{out}	Flow rate to outlet of Valvistor from inlet	m^3/s
Q_{out_tot}	Total flow rate to Valvistor outlet.	m^3/s
R	Piston pitch radius of swash plate	m
x_f	Displacement of pressure flow compensator spool	m
x_c	Control piston displacement	m
x_{pv}	Priority valve spool displacement	m
x_{cyl}	Cylinder piston displacement	m
x_p	Ultronics pilot valve spool displacement	m
x_i	Ultronics main stage valve displacement	m
x_m	Valvistor poppet displacement	m
V_d	Volumetric displacement of the pump	m^3
V_{0c}	Initial volume of control piston	m^3
V_c	Instantaneous volume of the control piston	m^3
V_{PP}	Priority valve PP chamber instantaneous volume	m^3
V_{0PP}	Initial volume of priority valve PP chamber.	m^3
V_{0CF}	Constant volume for calculating P_{EF}	m^3

V_{01}	Constant volume	m^3
V_{0LSS}	Initial volume of priority valve LSS chamber	m^3
V_{LSS}	Instantaneous volume of Priority valve LSS chamber	m^3
V_{0h} , V_{0r}	Initial volumes on head, rod side of cylinder	m^3
V_h , V_r	Instantaneous volumes on head, rod side of cylinder	m^3
V_{0mt}	Initial volume of Valvistor top chamber	m^3
V_{mt}	Instantaneous volume of Valvistor top chamber	m^3
α	Swash plat angle	rad
ρ	Density of the fluid	kg/ m^3
β	Fluid bulk modulus	Pa