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# **Control Aspects of Complex Hydromechanical Transmissions**

with a Focus on Displacement Control

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*To Kurtan* ✖

” Do I really look like a guy with a plan?

The Joker



# Abstract

This thesis deals with control aspects of complex hydromechanical transmissions. The overall purpose is to increase the knowledge of important aspects to consider during the development of hydromechanical transmissions to ensure transmission functionality. These include ways of evaluating control strategies in early design stages as well as dynamic properties and control aspects of displacement controllers, which are key components in these systems.

Fuel prices and environmental concerns are factors that drive research on propulsion in heavy construction machinery. Hydromechanical transmissions are strong competitors to conventional torque-converter transmissions used in this application today. They offer high efficiency and wide speed/torque conversion ranges, and may easily be converted to hybrids that allow further fuel savings through energy recuperation. One challenge with hydromechanical transmissions is that they offer many different configurations, which in turn makes it important to enable evaluation of control aspects in early design stages. In this thesis, hardware-in-the-loop simulations, which blend hardware tests and standard software-based simulations, are considered to be a suitable method. A multiple-mode transmission applied to a mid-sized construction machine is modelled and evaluated in offline simulations as well as in hardware-in-the-loop simulations.

Hydromechanical transmissions rely on efficient variable pumps/motors with fast, accurate displacement controllers. This thesis studies the dynamic behaviour of the displacement controller in swash-plate axial-piston pumps/motors. A novel control approach in which the displacement is measured with an external sensor is proposed. Performance and limitations of the approach are tested in simulations and in experiments. The experiments showed a significantly improved performance with a controller that is slightly more advanced than a standard proportional controller. The implementation of the controller allows simple tuning and good predictability of the displacement response.



# Acknowledgements

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Finally, I want to thank my family and friends for always being there for me and making me remember that there are more things to life than displacement controllers and lead-compensators. A special thank you to Jens for standing my endless dwelling and snapping me out of it when needed. Thank you Kurtan for only eating the notes I have already thrown in the trash can.

My greatest gratitude goes to my parents, Anna and Anders, who always believed in my ability.

Linköping, August 2017  
Viktor Larsson





# Abbreviations

CVT	Continuously Variable Transmission
GUI	Graphical User Interface
HMT	Hydromechanical Transmission
HST	Hydrostatic Transmission
HWIL	Hardware-in-the-loop
ICE	Internal Combustion Engine
ICPS	Input-Coupled Power-Split transmission
IVT	Infinitely Variable Transmission
OC	Open Circuit
OCPS	Output-Coupled Power-Split transmission
OCV	Open Circuit with Valve
P-controller	Proportional controller
P-lead-controller	Proportional controller with lead compensator
PWM	Pulse-Width Modulation
TLM	Transmission Line Modelling



# Nomenclature

$A_1$	Control piston area	$[\text{m}^2]$
$\alpha$	Swash plate angle	$[\text{rad}]$
$\alpha_{max}$	Maximum swash plate angle	$[\text{rad}]$
$C_q$	Valve flow coefficient	$[-]$
$d$	Valve spool diameter	$[\text{m}]$
$D$	Displacement	$[\text{m}^3/\text{rad}]$
$D_{max}$	Maximum displacement	$[\text{m}^3/\text{rad}]$
$\delta_a$	Desired relative damping of closed loop system	$[-]$
$\varepsilon$	Relative displacement	$[-]$
$\varepsilon_1$	Unit 1 relative displacement	$[-]$
$\varepsilon_2$	Unit 2 relative displacement	$[-]$
$\varepsilon_{2,min}$	Unit 2 minimum relative displacement	$[-]$
$\varepsilon_{2,ref}$	Unit 2 relative displacement, reference	$[-]$
$F_{ext}$	External piston force	$[\text{N}]$
$i_t$	Total transmission speed ratio ( $\omega_{out}/\omega_{in}$ )	$[-]$
$i_v$	Variator (HST) speed ratio ( $\omega_2/\omega_1$ )	$[-]$
$i_{v,max}$	Variator (HST) maximum speed ratio ( $\omega_2/\omega_1$ )	$[-]$
$J$	Rotational inertia	$[\text{kgm}^2]$
$K$	Control gain	$[-]$
$K_{PWM}$	Static solenoid gain	$[\text{m}/\%\text{Duty-cycle}]$
$K_q$	Valve flow gain	$[\text{m}^2/\text{s}]$

$K_s$	Static system gain	[1/%Duty-cycle]
$L$	Control piston lever (swash plate radius)	[m]
$\omega_1$	Variator (HST) unit 1 speed	[rad/s]
$\omega_2$	Variator (HST) unit 2 speed	[rad/s]
$\omega_a$	Desired resonance of closed loop system	[rad/s]
$\omega_d$	Displacement controller break frequency	[rad/s]
$\omega_{F,1}$	Controller lower break frequency	[rad/s]
$\omega_{F,2}$	Controller upper break frequency	[rad/s]
$\omega_{in}$	Transmission input speed	[rad/s]
$\omega_{out}$	Transmission output speed	[rad/s]
$\omega_v$	Spool valve break frequency	[rad/s]
$p_1$	Transmission pressure 1	[Pa]
$p_2$	Transmission pressure 2	[Pa]
$p_b$	Boost pressure	[Pa]
$p_c$	Control piston pressure	[Pa]
$p_T$	Tank pressure	[Pa]
$Q$	Volumetric flow	[m <sup>3</sup> /s]
$q_1$	Volumetric flow of unit 1	[m <sup>3</sup> /s]
$q_2$	Volumetric flow of unit 2	[m <sup>3</sup> /s]
$q_{acc}$	Accumulator inlet flow	[m <sup>3</sup> /s]
$\rho$	Oil density	[kg/m <sup>3</sup> ]
$s$	Laplace operator	[1/s]
$T_1$	Variator Unit 1 torque	[Nm]
$T_2$	Variator Unit 2 torque	[Nm]
$\tau$	Time constant	[s]
$T_{in}$	Transmission input torque	[Nm]
$T_{out}$	Transmission output torque	[Nm]
$t_r$	Rise-time (time to realise $\varepsilon : 0 \rightarrow \pm 1$ )	[s]
$U_\alpha$	Swivel angle signal	[V]

$U_{PWM}$	Pulse-width-modulated voltage	[% Duty-cycle]
$X_p$	Control piston displacement	[m]
$X_v$	Spool valve displacement	[m]
$F$	Controller transfer function	
$G_C$	Closed loop transfer function	
$G_O$	Open loop transfer function	
$S_{fwd}$	Clutch used in mode F1	
$S_h$	Clutch used in mode H	
$S_{rev}$	Clutch used in mode R1	



# Papers

The following papers make up the basis of the thesis. They are arranged in chronological order of publication and will be referred to by their Roman numerals. All papers are printed in their originally published state with the exception of some minor changes in text and figure layout in order to maintain consistency throughout the thesis.

In all papers, the first author is the main author, responsible for the work presented, with additional support from the co-writers. An exception is paper [I], in which the three first authors are the main authors, responsible for the work presented, with additional support from the fourth author. An overview of the topics addressed in each paper is provided at the end of chapter 1. A short summary of each paper is given in chapter 8.

- [I] Karl Pettersson, L. Viktor Larsson, K. Viktor Larsson, and Petter Krus. “Simulation Aided Design and Testing of Hydromechanical Transmissions”. In: *Proceedings of the 9th JFPS International Symposium on Fluid Power (JFPS’14)*. Matsue, Japan, 2014. ISBN: 4931070108.
- [II] L. Viktor Larsson, Karl Pettersson, and Petter Krus. “Mode Shifting in Hybrid Hydromechanical Transmissions”. In: *Proceedings of the ASME/BATH 2015 Symposium on Fluid Power and Motion Control (FPMC2015)*. Chicago, Illinois, USA, 2015. ISBN: 9780791857236. DOI: 10.1115/FPMC2015-9583.
- [III] L. Viktor Larsson and Petter Krus. “Modelling of the Swash Plate Control Actuator in an Axial Piston Pump for a Hardware-in-the-Loop Simulation Test Rig”. In: *The 9th FPNI Ph.D. Symposium on Fluid Power*. Florianópolis, SC, Brazil, 2016, V001T01A044. ISBN: 978-0-7918-5047-3. DOI: 10.1115/FPNI2016-1570.
- [IV] L. Viktor Larsson and Petter Krus. “Displacement Control Strategies of an In-Line Axial-Piston Unit”. In: *The 15th Scandinavian International Conference on Fluid Power (SICFP’17)*. Linköping, Sweden, 2017.

## Other publications

The publications listed below are not included in the thesis but constitute an important part of the background. In [V], both authors are the main authors and responsible for the work presented.

- [V] L. Viktor Larsson and K. Viktor Larsson. “Simulation and Testing of Energy Efficient Hydromechanical Drivelines for Construction Machinery”. Master’s thesis. Linköping University, 2014.
- [VI] L. Viktor Larsson. “Modellering och simulering av en hydraulmekanisk power-splittransmission med två moder”. In: *Hydraulikdagarna*. Linköping, Sweden, 2015 (In Swedish).



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## Appended Papers

<b>I</b>	<b>Simulation aided Design and Testing of Hydromechanical Transmissions</b>	<b>57</b>
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# 1

## Introduction

Heavy construction machines are used all around us to shape and build a world that fits our human needs. These machines, such as wheel loaders, demand efficient, robust transmissions for vehicle propulsion. Lately, there has been an increased interest in complex Hydromechanical Transmissions (HMTs) to replace conventional (torque converter) drivelines in construction machinery. This thesis treats control aspects of complex HMTs. A method for evaluating control concepts in early design stages as well as dynamic models and control strategies for key components of HMTs are presented and discussed.

### 1.1 Background

Fuel prices and environmental concerns are factors that drive research on energy-efficient transmissions for heavy construction machinery. The hydrodynamic (torque converter) transmission commonly used in heavier applications has several benefits in terms of cost, robustness and well-known behaviour for the operators. The component does, however, suffer from substantial losses [1], and there is a consensus regarding the need to find alternatives.

HMTs have several benefits that make them competitive with torque converter-based transmissions. Apart from a higher efficiency of the transmission itself, it decouples the engine speed from the vehicle speed, thereby enabling optimal diesel engine operation [2]. Power-split HMTs that combine a traditional Hydrostatic Transmission (HST) with a mechanical power path allow high efficiencies and wide torque/speed conversion ranges [3]. If multiple modes (gear shifts) are introduced, the range may be increased even further [4]. Another important feature of HMTs is that they may be hybridised using hydraulic accumulators [5]. This extension enables energy recuperation, for example during vehicle braking, thus making further fuel savings possible [6].

One challenge in the development process of HMTs is the large design space due to the high number of possible configurations. To handle this challenge,

simulation-based design optimisation has been proven to be an important tool [7]. As the transmission matures, low-level control and dynamic properties become increasingly important, and the computer-based simulations need support from hardware. Late in the development process, this hardware could be a prototype available for experiments. Building prototypes, however, is costly and time-consuming, and the big design space could lead to a high number of interesting concepts that have to be realised. For these reasons, Hardware-in-the-loop (HWIL) simulations could be considered as a middle-way alternative. The concept blends simulation and prototyping by including hardware in the simulations in real time. This aims to improve the reliability of the results while maintaining the versatility and relatively cheap process of software simulations [8]. A benefit specific to transmissions is that a real vehicle is not necessary, which facilitates the testing procedure [9]. HWIL simulations have been used successfully for control evaluation of both hybrid [10] and non-hybrid [11] HMTs.

Central components in HMTs are the hydrostatic machines (pumps/motors). One important aspect is high efficiency over a wide operating range [12]. From a control perspective, a critical condition is that the hydrostatic machines are variable, that is, they must be able to change their displacement. For non-hybrid HMTs, the displacement settings of the hydrostatic machines are the control input signals, which in turn determine the total transmission torque/speed ratio [13]. In the case of hybrid HMTs, the displacement settings control the power flows between the accumulator, engine and vehicle [14]. To ensure high productivity and acceptable behaviour for the operator of a heavy construction machine, fast, accurate control of the displacement is of great importance. Until recently, there has been little progress in the development of displacement controllers in hydrostatic machines for HMTs. In the long term, one approach could be to change the design and principle of operation compared to conventional axial piston machines, see for example [12, 15]. In the short term, the trend is to equip conventional machines with integrated sensors and controllers [16, 17]. This gives greater freedom in the controller design, with potential performance improvements, but at the same time choosing a proper control strategy becomes more difficult. An understanding of the dynamics of the displacement controller is therefore needed in the controller design process.

## 1.2 Aim and Research Questions

The overall aim of this thesis is to increase the knowledge of important control aspects to consider during the development of HMTs to ensure transmission functionality. The big design space and drive for low cost of HMTs motivate control evaluation in early design stages. HWIL simulation is a promising enabler for this, as it may move hardware tests to earlier design stages compared to pure, expensive prototyping. Pump/motor displacement settings constitute the major control signals in HMTs and their dynamic performance may therefore be expected to have great influence on the transmission control in hybrid systems in particular. This influence concerns both the control in a single mode and the highly transient shifts between modes. Knowledge of the displacement controller dynamics and suitable displacement control strategies is therefore necessary in the development of HMTs. More specifically, the aim may be translated into the following research questions:

**RQ1:** How may HWIL simulations be used to evaluate control of complex transmission concepts in early design stages?

**RQ2:** How is the mode shift in hybrid HMTs affected by the dynamic performance of the displacement controllers?

**RQ3:** What are the dominating dynamic characteristics in displacement controllers?

**RQ4:** What is a suitable control concept for displacement control?

## 1.3 Delimitations

This thesis concerns low-level control aspects of complex HMTs. Other aspects, such as high-level control (energy management), energy efficiency, manufacturing and marketing are not considered. In the dynamic analysis of displacement controllers, only swash-plate axial-piston pumps/motors are considered.

## 1.4 Contribution

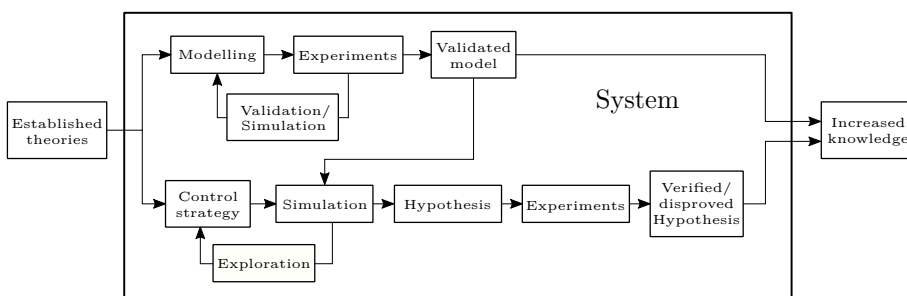
This thesis shows how HWIL simulations may be used to evaluate complex HMTs from a control perspective in early design stages. The problem of handling mode shifts in hybrid HMTs due to the dynamics of the displacement controllers is identified. A deeper understanding of the dynamic behaviour of displacement controllers in swash-plate axial-piston machines is provided, and control strategies for the displacement are proposed. The dynamic aspects and control strategies for the displacement controller are studied in analytical models and verified in hardware tests.

## 1.5 Method

The method used in this thesis may be summarised by three key concepts:

- Modelling
- Simulation
- Experiments

Established theory contributes methods to mathematically model different aspects of a studied system. Typically, the dynamic behaviour of the included components is of interest from a low-level control perspective. In order to make use of the model, it is validated against experiments in hardware, thereby adapting it to reality. This process is iterative, in which new insights concerning the system require further enhancement of the model and thus further experiments. By validating the model, it may be used with a higher level of trust. This makes the controller design process more efficient, since it can be performed in simulation to a large extent, thereby reducing the number of experiments. Just like the modelling process, the controller design process is iterative, although of a more exploratory and creative nature. Different control strategies may be tested and evaluated in the simulation environment. Eventually, this process leads to a hypothesis, for example that one control strategy is superior to another. Since the model is still only a simplified representation of the real system, final experiments are conducted in order to verify or disprove the hypothesis. These experiments are then carried out on the system with the proposed control strategies implemented. If the control strategy deals with a full vehicle, this final experiment could consist of HWIL simulations. If a simpler system is studied, simpler, isolated tests might be more applicable. This is, for example, the case when displacement controllers are evaluated in this thesis.



*Illustration of the method used in the thesis.*

## 1.6 Overview of Appended Papers

An overview of the methods used and the topics addressed in the appended papers is given in the table below. The relationships between the papers and the research questions listed in section 1.2 are also illustrated. See chapter 8 for a short review of each paper.

Paper	[I]	[II]	[III]	[IV]
<b>Method</b>				
Simulation	X	X	X	X
Experiments	X		X	X
HWIL-simulations	X			
<b>Topics</b>				
HMTs	X	X		
Hybrid		X	X	X
Displacement control		X	X	X
Control	X	X		X
<b>RQ</b>	1	2	3 and 4	3 and 4



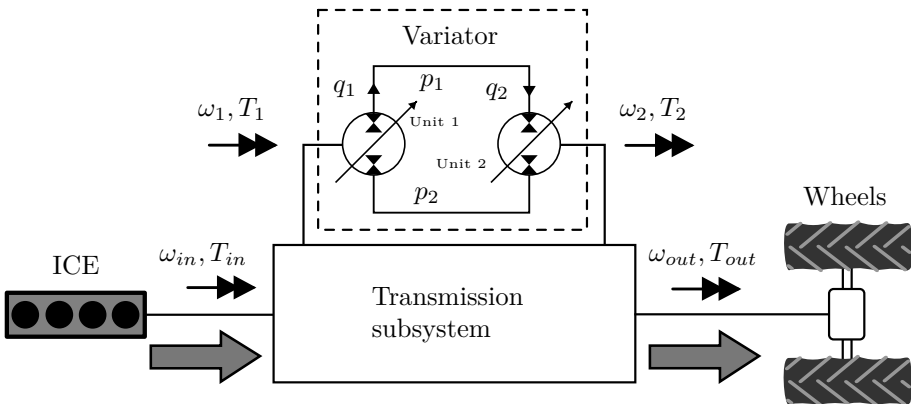


# 2

## Hydromechanical Transmissions

Continuously Variable Transmissions (CVTs) are characterised by the ability to continuously change their speed/torque ratio. Infinitely Variable Transmissions (IVTs) allow infinite variation of the speed/torque ratio, which means that the output shaft can have zero speed even if the input shaft rotates. Both these features are desirable in transmissions for heavy construction machinery [18].

A general CVT consists of a mechanical transmission subsystem connected to a *variator* which enables the continuously variable torque/speed ratio [4]. When the variator is hydraulic, the transmission is an HMT [19]. Figure 2.1 shows a general HMT that transfers power from an Internal Combustion Engine (ICE) to the wheels.



**Figure 2.1** General HMT, adapted from [4].

## 2.1 Basic Equations

To facilitate the understanding of figures and equations in this chapter, some basic equations are presented in this section. No losses are included, since the relationships aim to show the fundamental properties.

Referring to figure 2.1, the total transmission and variator speed/torque ratios may be defined as in equations (2.1) and (2.2), respectively.

$$i_t = \frac{\omega_{out}}{\omega_{in}} = -\frac{T_{in}}{T_{out}} \quad (2.1)$$

$$i_v = \frac{\omega_2}{\omega_1} = -\frac{T_1}{T_2} \quad (2.2)$$

The flows of the hydrostatic units may be calculated according to equations (2.3) and (2.4).

$$q_1 = \omega_1 D_1 \varepsilon_1 \quad (2.3)$$

$$q_2 = \omega_2 D_2 \varepsilon_2 \quad (2.4)$$

The torque at the shafts of the hydrostatic units may be calculated according to equations (2.5) and (2.6).

$$T_1 = (p_1 - p_2) D_1 \varepsilon_1 \quad (2.5)$$

$$T_2 = (p_1 - p_2) D_2 \varepsilon_2 \quad (2.6)$$

Since losses are ignored,  $q_1 = q_2$ , and the variator speed/torque ratio may be related to the relative displacements according to equation (2.7).

$$i_v = \frac{\omega_2}{\omega_1} = \frac{D_1}{D_2} \cdot \frac{\varepsilon_1}{\varepsilon_2} \propto \frac{\varepsilon_1}{\varepsilon_2} \quad (2.7)$$

The variator speed/torque ratio,  $i_v$ , is thereby controlled by varying the displacements of Unit 1 and 2. Depending on the transmission subsystem, this will in turn control the total transmission speed/torque ratio,  $i_t$ .

## 2.2 HMT Configurations

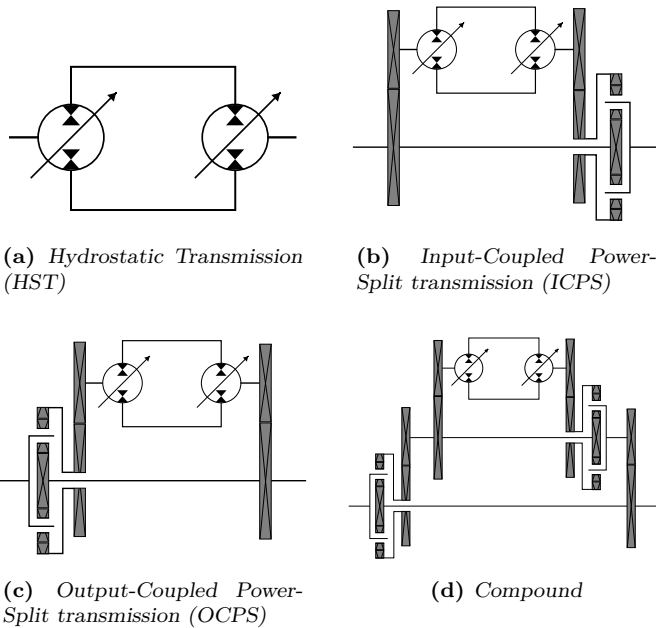
HMTs may be divided into single-mode transmissions and multiple-mode transmissions. Multiple-mode transmissions include clutches that enable switching between different single-mode configurations [7]. Some single-mode configurations commonly mentioned in the literature are shown in figure 2.2.

A simple single-mode HMT is a pure hydraulic variator, commonly referred to as a Hydrostatic Transmission (HST), shown in figure 2.2a. This transmission

consists of two hydrostatic units connected in closed circuit. Extensions to the HST are *Power-split* single-mode HMTs. These transmissions include epicyclic (planetary) gear trains that can split the power between an HST path and a mechanical path.

Power-split HMTs are usually classified based on where (to which shaft) the HST is connected. Consequently, an Input-Coupled Power-Split transmission (ICPS) (figure 2.2b) is defined by the fact that the HST is connected to the transmission input shaft while the HST is connected to the transmission output shaft in an Output-Coupled Power-Split transmission (OCPS) (figure 2.2c) [3]. It is possible to combine an ICPS and an OCPS to form a *compound* or *variable-bridge* configuration [5], shown in figure 2.2d.

Single-mode HMTs have different properties that make them advantageous in different applications. Pure HSTs are competitive with torque converter transmissions in terms of efficiency [20], but suffer from oversized hydrostatic units for high-power applications [1]. Power-split single-mode HMTs offer higher efficiency thanks to the mechanical path, and may be realised with smaller hydrostatic units compared to an HST [21]. Another aspect of power-split transmissions is power circulation, which can have a negative impact on both efficiency [3] and control effort [21].

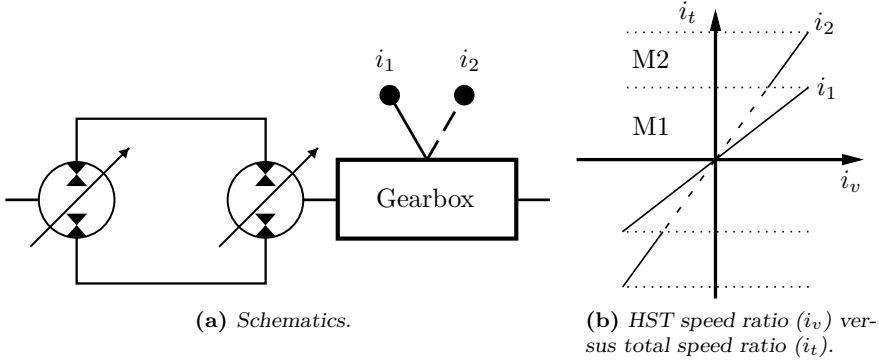


**Figure 2.2** Examples of single-mode transmissions [7]. Input shaft on the left.

### 2.2.1 Multiple-Mode Transmissions

By adding clutches to the transmission, several single-mode HMTs may be combined to form a multiple-mode HMT. The basic idea is to utilise the best properties of each single-mode configuration and increase the conversion range. If properly designed, a multiple-mode HMT thereby allows a wide speed range with high efficiency and moderate sizes of the hydrostatic units [7].

A simple multiple-mode solution is an HST and gearbox connected in series, as shown in figure 2.3. The two gears,  $i_1$  and  $i_2$ , introduce two modes, M1 and M2, which increase the transmission conversion range. The shift between the modes is non-synchronised, as a step in hydrostatic speed ratio is required. An HST with a gearbox in series is a common solution in smaller wheel loaders [1].

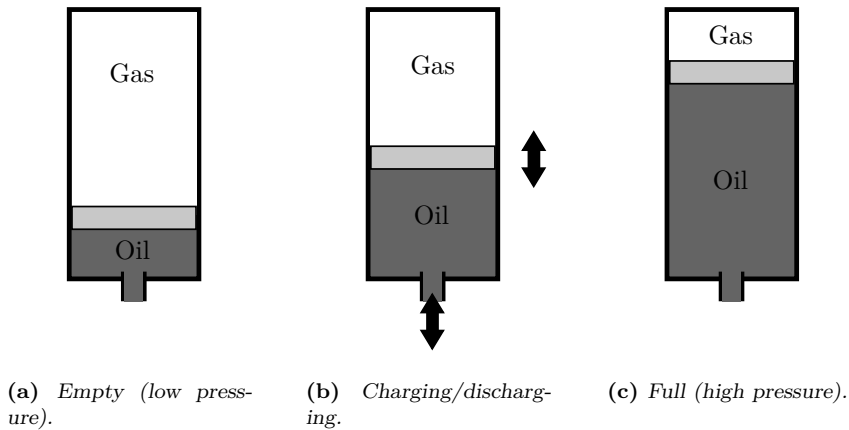


**Figure 2.3** Simple multiple-mode HMT realised by placing a gearbox with gear ratios  $i_1$  and  $i_2$  in series with an ordinary HST, resulting in two modes,  $M_1$  and  $M_2$ .

In the case of larger wheel loaders, more complex solutions may be applicable so that one or several of the modes are of the power-split configuration. One advantage with this approach, is that synchronised mode-shifts may be achieved. This means that no step in HST speed ratio is needed, which minimises the disturbance in tractive force during the mode shift [22]. Power-split multiple-mode HMTs are suitable for heavy applications and commercialised examples may be found for both agricultural tractors [19, 23] and construction machinery [24, 25]. A simple power-split multi-mode HMT with synchronised mode shifts is described in more detail and simulated in chapter 3.

## 2.3 Hybridisation

To further increase the fuel savings, hybridisation could be considered a logical next step. In an HMT, this is enabled by the use of a hydraulic accumulator. Hydraulic accumulators can be weight-loaded, spring-loaded or gas-loaded, of which gas-loaded is the most commonly used type in HMTs [27]. A gas-loaded accumulator stores energy in the form of pressure of a gas (commonly nitrogen) [14]. The gas is contained in a vessel and is separated from the oil by a piston, bladder or diaphragm [28]. The basic principle of operation of a gas-loaded piston accumulator is illustrated in figure 2.4.



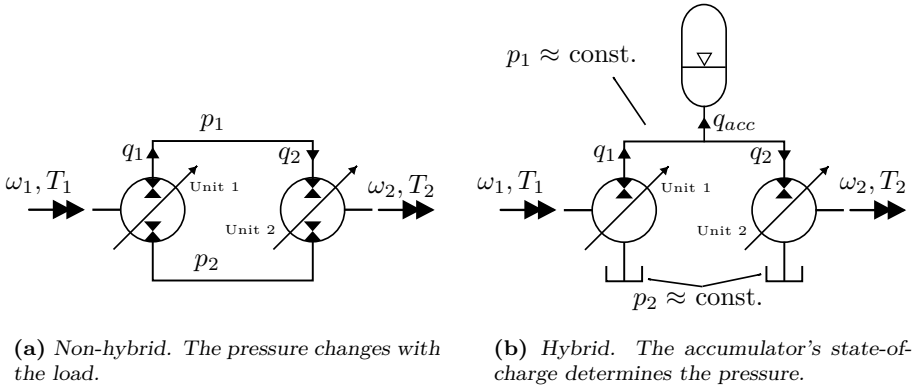
**Figure 2.4** Piston accumulator with different state-of-charge.

The accumulator has high influence on the performance of a hybrid HMT. The thermodynamic relationship between the pressure and the volume of the gas is very non-linear and gives rise to thermal losses as the gas is compressed [29]. These losses are highly affected by the frequency of charging/discharging [30], but may be reduced by using elastomeric foam in the gas volume [31]. Two other factors that affect energy recovery are the accumulator volume and precharge gas pressure [32].

Another important effect of the accumulator on the system is that its high capacitance causes the system pressure to be quasi-constant or *impressed* [33] by the accumulator's state-of-charge. Figure 2.5 shows how the HST circuit in an HMT has to be altered to enable hybrid configurations. The hybrid HST has dedicated high- and low-pressure sides, of which the high-pressure side is determined by the accumulator's state-of-charge. This is in contrast to a non-hybrid HST, in which the pressure sides may alter and the total

pressure difference is determined by the load. Since the accumulator may add/subtract flow to/from the circuit, equations (2.2), (2.1) and (2.7) are not always valid for a hybrid HST. The flow of each unit is instead determined by its shaft speed and displacement according to equations (2.3) and (2.4) and the difference between  $q_1$  and  $q_2$  results in a charge or discharge of the accumulator according to equation (2.8). The impressed system pressure means that the unit displacement determines the unit torques, according to equations (2.5) and (2.6), rather than the transmission speed ratio. This concept is commonly referred to as *secondary control*, which is discussed in section 2.4.

$$q_{acc} = q_1 - q_2 \quad (2.8)$$

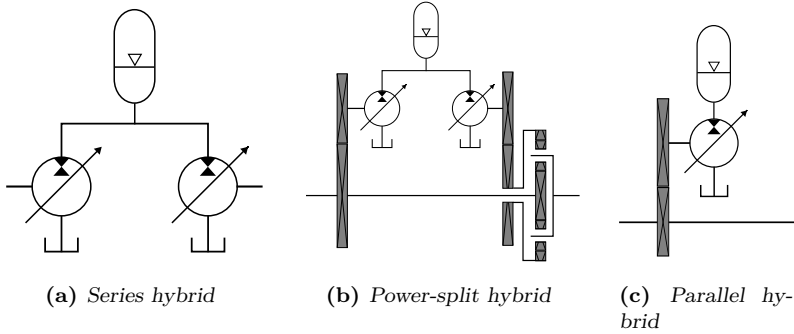


**Figure 2.5** A Hydrostatic Transmission (HST) is hybridised by adding a hydraulic accumulator. The high capacitance of the accumulator changes the behaviour of the circuit and the hybrid HST has one dedicated high-pressure side and one dedicated low-pressure side. The low-pressure side commonly has a low-pressure accumulator, here represented as tanks.

### 2.3.1 Hybrid Configurations

The HMT configurations presented in section 2.2 may all be hybridised by inserting an accumulator into the HST circuit as shown in figure 2.5. This results in a number of hybrid configurations. Figure 2.6 shows the hybrid configurations that are commonly mentioned and studied in the literature. The *series hybrid* is an ordinary HST with a hydraulic accumulator. *Power-split* or *complex* or *series-parallel* hybrids cover the hybrid versions of the power-split transmissions presented in section 2.2, both single-mode and multiple-mode types. The *parallel* or *add-on hybrid* does not have a non-hybrid equivalent

and is realised by adding a pump/motor and hydraulic accumulator to a conventional drive-line.



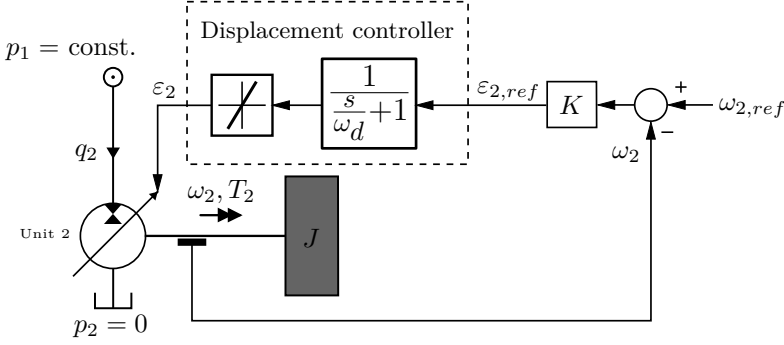
**Figure 2.6** Hybrid Hydromechanical Transmissions (HMTs) commonly mentioned in the literature.

As with non-hybrid HMTs, the different hybrid configurations have different properties that are advantageous in different applications. The series hybrid is commonly attributed the advantage of decoupling engine and vehicle speed, while the parallel hybrid allows more efficient operation as all power does not have to be transferred hydraulically [14]. Power-split hydraulic hybrids combine the benefits of series and parallel configurations but also present challenges in choice of configuration, component sizing and control effort [5].

Hybrid HMTs have the potential to improve fuel efficiency of wheeled vehicles and there are some examples of commercialised concepts for heavy commercial vehicles [34, 35]. For wheel loaders, hydraulic hybridisation still seems to be on an academic level but results to date seem promising. For a fully hybridised vehicle, including the working hydraulics, energy savings of up to 50% for the short loading cycle can be expected [36]. Also, power-split hybrid HMTs may utilise the complex power flows of the power-split configuration to handle energy management in both transmission and working hydraulics in an efficient manner [37].

## 2.4 Secondary Control

Secondary control is a concept within fluid power control that dates back to the 1980s, see for example [38]. The basic idea is that the speed of a hydraulic motor is controlled by the motor's own displacement [30]. This idea is enabled by subjecting the motor to a constant pressure, as illustrated in figure 2.7. By studying equation (2.6), repeated in equation (2.9), it is realised that the speed is controlled by controlling the unit's torque through the displacement. This



**Figure 2.7** Simple secondary-controlled motor with inertia load and proportional feedback. The displacement controller is represented as a first order system with a rate limiter.

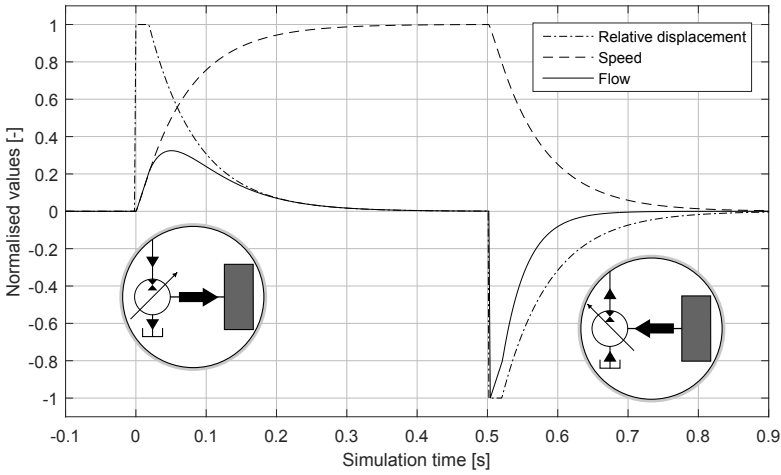
in turn requires active speed control [39] and four-quadrant operation and the ability to realise relative displacements within  $-1 \leq \varepsilon_2 \leq 1$  to enable braking of the load.

$$T_2 = (p_1 - p_2) D_2 \varepsilon_2 \propto \left/ \frac{(p_1 - p_2)}{\text{constant}} \right/ \propto \varepsilon_2 \quad (2.9)$$

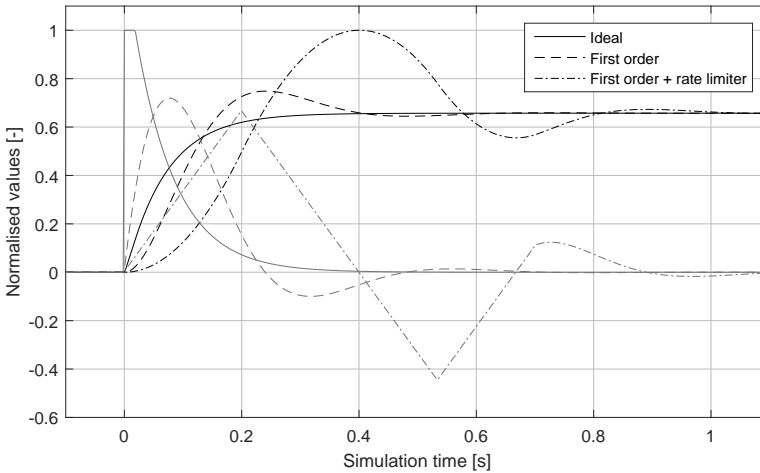
Figure 2.8 shows simulated step responses of the system in figure 2.7 without displacement controller dynamics. The speed is a consequence of the torque equilibrium of the flywheel, while the flow is determined by the product of the speed and the displacement according to equation (2.4). As the flywheel is braked to rest at 0.5 seconds, the flow changes direction (Unit 2 now acts as a pump) and the kinetic energy stored in the flywheel is sent back into the system, for example to an accumulator.

The displacement controller constitutes the innermost control loop in the secondary control loop and therefore has substantial influence on the dynamic response [40]. Apart from the dynamics of the displacement actuator, the maximum flow of the actuator control valve limits the maximum speed of the actuator [17]. This non-linearity does not cause instability in secondary-controlled systems but gives overshoots and delays the response [30]. This is illustrated in figure 2.9, which shows a step in speed of the system in figure 2.7 if dynamics and valve saturation are included in the model. The dynamics of the displacement actuator add overshoot in speed, which is amplified if a rate limiter is added to mimic the saturation in valve flow.





**Figure 2.8** Step response in speed of the system in figure 2.7, assuming an ideal displacement controller. The speed has been normalised with a value of 10.46 rad/s. The flow has been normalised with a value of  $1.64 \cdot 10^{-4} \text{ m}^3/\text{s}$ .



**Figure 2.9** Step response in speed (black) of the system in figure 2.7 with different displacement controller dynamics (grey). The speed has been normalised with a value of 15.93 rad/s.

**Table 2.1** Parameter values used in the simulations in figure 2.8 and 2.9.

Parameter	Value	Unit
$p_1$	300	bar
$K$	0.12	s/rad
$\omega_{2,ref}$	100	rpm
$J$	4.0	kgm <sup>2</sup>
$D_2$	100	cm <sup>3</sup> /rev
$\omega_d$	17	rad/s
$t_r$ (rate-limiter parameter, min time to realise $\varepsilon_2 : 0 \rightarrow \pm 1$ )	0.3	seconds

### 2.4.1 Secondary Control in Hybrid HMTs

As discussed in section 2.3, the impressed system pressure in hybrid HMTs transforms the system into a secondary-controlled one. There are, however, two main differences between “pure” secondary control as presented above and secondary control in a hybrid HMT.

First, the pressure is not constant but varies (slowly) with the accumulator’s state-of-charge. This means that the output torque and power are limited if the state-of-charge is low, which complicates the high-level control [37]. Varying pressure also means that secondary control relies on pressure control as the accumulator has minimum and maximum allowed pressure levels.

Second, the controller in the secondary control loop is the driver. This means that the driver sets the vehicle output torque to control the vehicle speed, a concept often referred to as *torque control* [5].

In series hybrid HMTs, Unit 1 is often used for pressure control while Unit 2 handles the output torque [41]. In power-split hybrid HMTs, a similar approach is commonly used in which one of the units controls the pressure while the other controls the output torque. However, the kinematic relationships in a power-split HMT is not as simple as in an HST and different configurations facilitate torque control to different degrees [5].

# 3

## Hardware-In-the-Loop-simulation of Hydromechanical Transmissions

This chapter summarises the primary parts of paper [I]. It shows an example of the use of HWIL simulations for control evaluation of HMTs. A multiple-mode transmission applied to a mid-sized heavy construction machine is considered, and the results are compared with corresponding offline simulations. Conclusions concerning the functionality of the specific transmission are drawn and the use of HWIL simulations for multiple-mode HMTs is discussed.

The basic idea of HWIL simulations is to blend traditional computer-based simulations and vehicle prototyping to speed up the product development process [8]. Essentially, the system of interest is included as hardware and its surroundings are simulated. This aims to increase the trustworthiness of the simulations, while versatility is maintained. Traditionally, HWIL refers to the testing of control code and the communication interface of a physical control unit (the hardware) in a simulation environment. In the context considered in this thesis, a high amount of power is active in the simulation, which differentiates “Power-in-the-loop” from HWIL. An early example of Power-in-the-loop is [42], in which a load-sensing pump is simulated in real-time for a mobile crane.

HWIL is motivated for HMTs for several reasons. One is that the big design space leads to many concepts as candidates for evaluation, and building a prototype for each one is both expensive and time-consuming. Another reason is that it allows testing of high degrees of control with high repeatability without the need for an actual vehicle [9].

The choice of controlled variables is important in the implementation of

HWIL. Sprengel et al. [9] use bond graph analogies to explain that for a rotating shaft, either the torque or the angular velocity should be controlled by the simulation while the other variable should be a consequence of the hardware. For example, a non-hybrid HST has a dominating speed-coupling between its input and output shafts due to the closed circuit. Consequently, Jansson et al. [11] let the controller set the speed of the input shaft and then control the torque at the output shaft. A hybrid HMT is torque-controlled by nature, and so in that case the approach would be to let the simulation control the output shaft speed [10]. In fact, Sprengel et al. [9] switch between torque and speed control in the HWIL-testing of their “Blended Hybrid” since this concept switches between a non-hybrid and hybrid HST configuration.

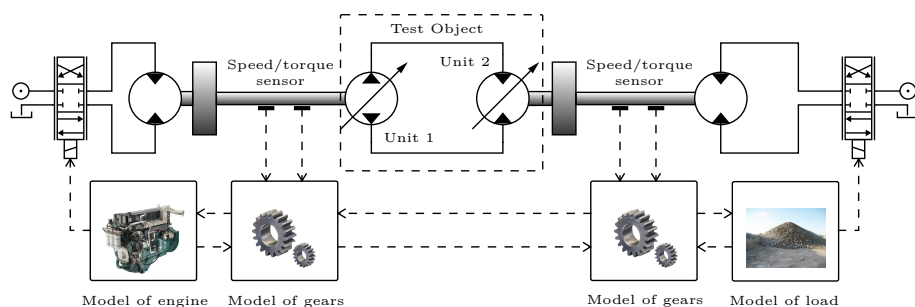
Another interesting problem is where to “cut” between the hardware and the models. For example, inertia may be simulated in software or represented as flywheels in the hardware. The former approach is versatile since it allows simulation of different inertia but puts high demands on accurate models and high controller and actuator bandwidths. The latter approach may produce a more realistic case, but limits the simulation to a specific vehicle. In the work presented in this thesis, a middle way is considered in which flywheels are used, but compensated for in the software.

### **3.1 Test Rig**

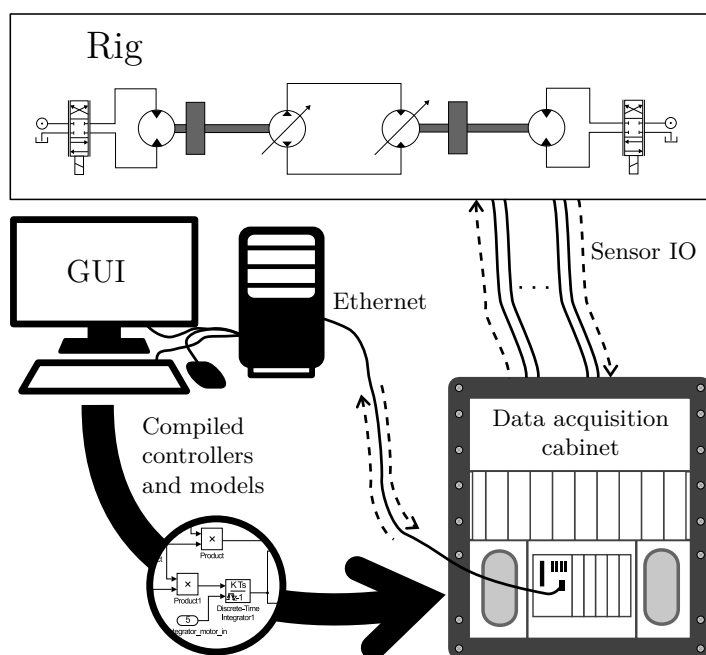
Figure 3.1 shows the layout of the test rig used for the HWIL simulations. The basic idea is that a servo-valve-controlled pump/motor is used on each side of an HST circuit to simulate its boundary conditions. By measuring and controlling the torque or the angular velocity on each shaft, the vehicle’s behaviour is simulated using models executed in real time in the data acquisition system. The models may represent the vehicle, the engine as well as the kinematic relationships in an HMT. For the transmission concept considered in this chapter, the rig’s left side represents a speed-controlled diesel engine. The valve on the right side controls the Unit 2 torque. The controllers are based on the work in [11]. Flywheels on the shafts are used to simulate engine and vehicle inertia.

The data acquisition system consists of a PXI-8110-RT real time computer from National Instruments (NI) which measures the sensor signals through a set of NI module measurement cards. Amplification of Pulse-Width Modulation (PWM) and voltage output signals as well as signal conditioning are done with a set of Prevas Gobi FISC (Fault Injection Signal Conditioning) boards.

The rig’s communication principle is shown in figure 3.2. The PXI computer runs as a server with LabView<sup>TM</sup> running in real time, with a sampling frequency of 1 kHz. A software called Viking is used to communicate with the PXI computer using an Ethernet interface [43]. On a PC, the Viking Graphical User Interface (GUI) allows controllers and models to be uploaded on the PXI computer and controlled during hardware tests. The controllers and models are compiled Simulink models that run in real time on the PXI computer.



**Figure 3.1** Hardware-in-the-loop transmission test bed for testing HMT concepts.



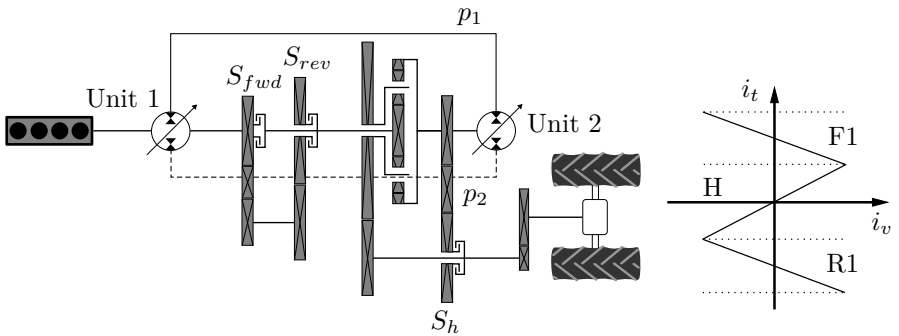
**Figure 3.2** Rig communication principle.

### 3.2 Transmission Concept

The considered transmission concept is shown in figure 3.3. The transmission is a simple version of the Jarchow concept [44] and has two modes in each direction of travel. The first mode (H) is purely hydrostatic, which enables easy reversal by controlling Unit 1 over centre. The second mode is of the ICPS configuration and increases the range and efficiency at higher speeds. The forward ICPS mode (F1) and reverse ICPS mode (R1) are identical reflections of each other. The mode shift is synchronised to limit the disturbance in transmission output torque during the shift. Table 3.1 shows the clutch arrangements for the different modes while figure 3.4 shows the transmission power flows in the different modes. In the hydrostatic mode, all power flows through the HST. At 7 km/h, clutch  $S_{fwd}$  ( $S_{rev}$  in reverse) is synchronised. At this point, clutch  $S_h$  is disengaged and clutch  $S_{fwd}$  is engaged. The transmission is then subject to negative power recirculation. As the speed increases, the speed ratio of the HST is decreased and the amount of power through the HST decreases. When  $i_v$  is zero, a full mechanical point is reached. Here, all power flows through the mechanical path with high efficiency. At speeds above the full mechanical point, the transmission is subject to additive power flow.

**Table 3.1** Clutch arrangements for the transmission in figure 3.3.

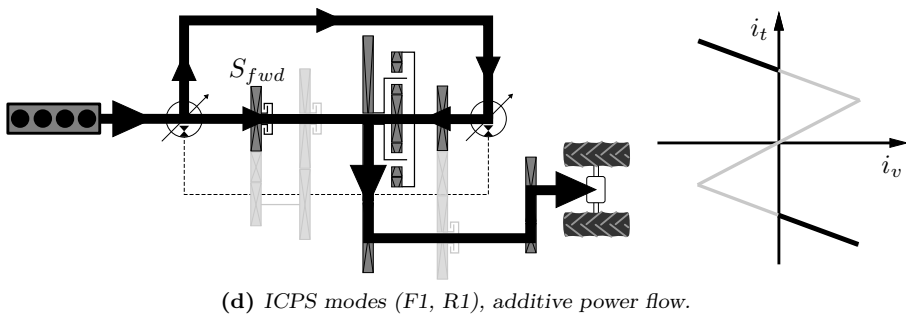
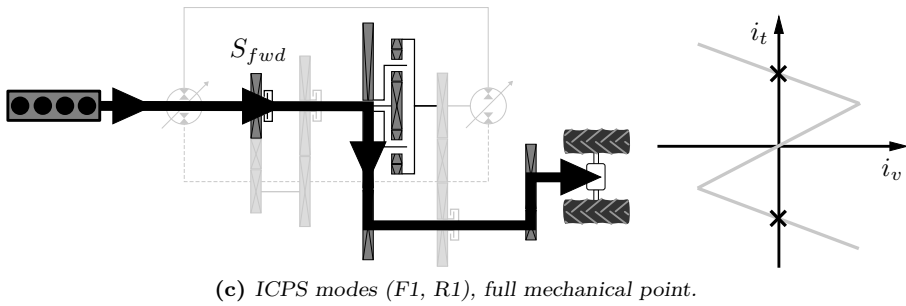
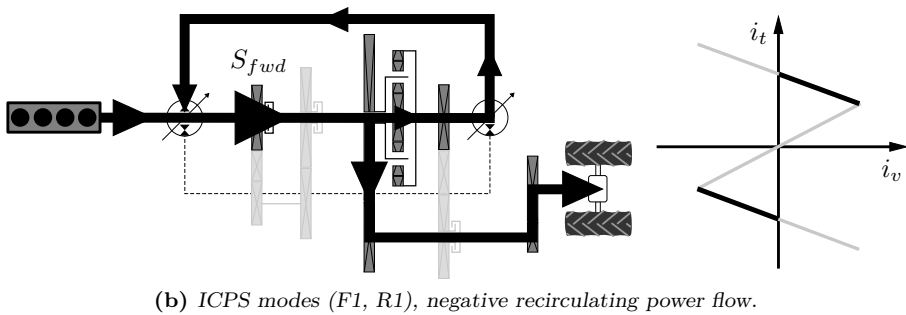
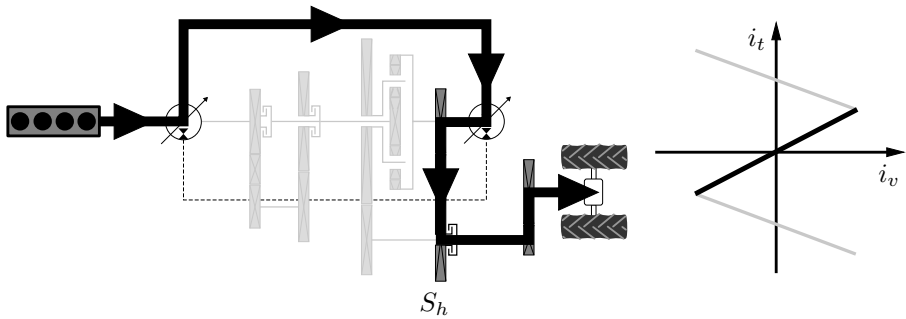
	$S_{fwd}$	$S_h$	$S_{rev}$
R1			•
H		•	
F1	•		



(a) Transmission layout.

(b) HST control for the different modes.

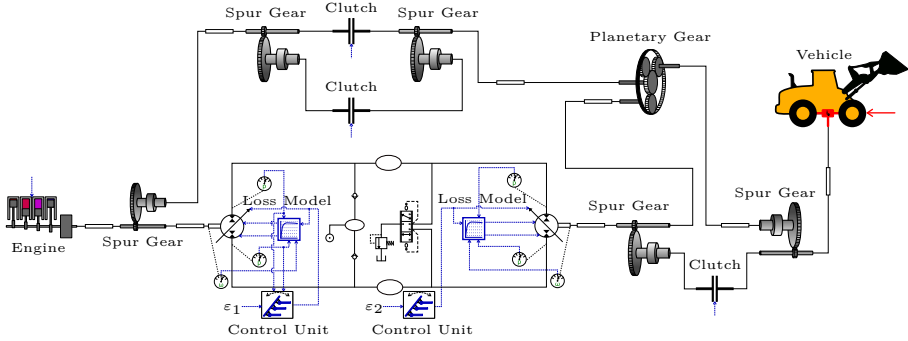
**Figure 3.3** Considered multiple-mode HMT concept.



**Figure 3.4** *Power flows in different modes in the considered concept.*

### 3.3 Model

For the offline simulations a model is implemented in Hopsan, a multi-domain simulation tool developed at Linköping University [45]. Hopsan is based on the Transmission Line Modelling (TLM) method, in which the transmission components may be modelled individually and separated by transmission line elements. This decouples the components from each other in time and enables fast parallel simulation using distributed equation solvers [46]. The full vehicle model is shown in figure 3.5. To represent the complete driveline, mechanical components (shafts, spur gears, planetary gear, clutches, engine and vehicle) and hydraulic components (hydraulic machines with loss models, displacement controllers, HST volumes, boost circuit and flush valve) need to be included. Detailed descriptions of the sub-models can be found in [V] and in paper [I]. [V] also includes a validation of the hydraulic components.

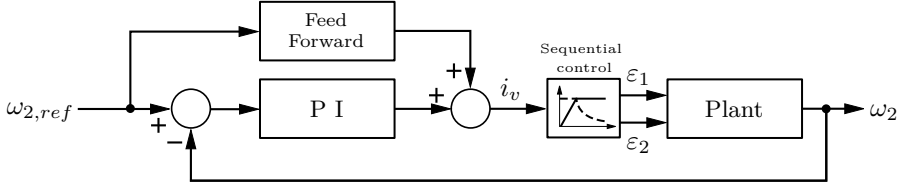


**Figure 3.5** Complete Hopsan simulation model of considered HMT concept and vehicle.

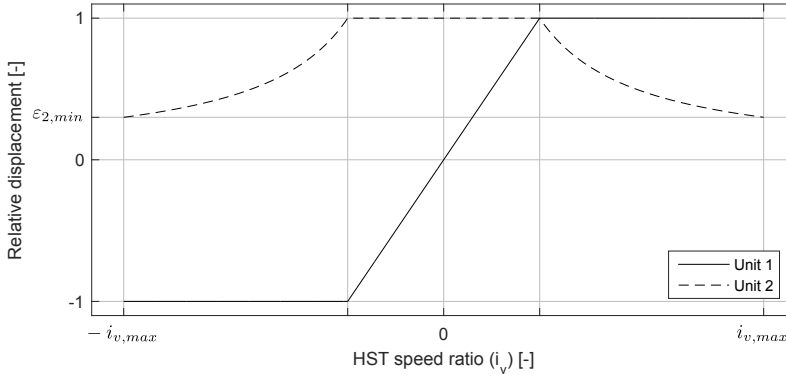
### 3.4 Control

The control algorithms are developed in Matlab Simulink, to which the Hopsan model is exported. In the considered tests the engine is controlled at constant speed which, means that the vehicle speed may be controlled by controlling the speed of Unit 2. The transmission is controlled by means of a simple PI-controller with feed-forward, as shown in figure 3.6. The reference speed of Unit 2 is calculated from the reference vehicle speed using the kinematic relationships for the current mode. The control signal is the speed ratio,  $i_v$ , of the HST, which is translated to relative displacement of the hydraulic machines with a sequential control mapping as shown in figure 3.7. The feed-forward part is based on the flow equation (2.7).





**Figure 3.6** Overview of the control concept for the HST.



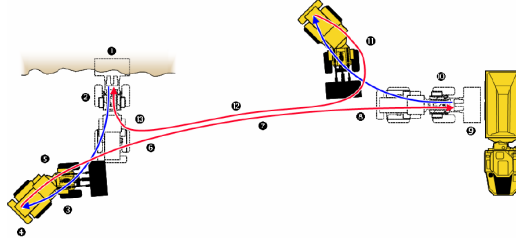
**Figure 3.7** Sequential control mapping for translation of HST speed ratio ( $i_v$ ) to relative displacements of Units 1 and 2, see for example [1]. Note that Unit 2 does not need to travel over centre.

### 3.5 Results

The developed control strategies are used to control the model in offline simulations in Hopsan as well as in HWIL simulations in the test rig. Numerical data for the considered transmission and vehicle is provided in table 3.2. In the HWIL simulations, the mechanical parts of the transmission are simulated in real time, while the HST is represented by hardware. The shaft connected to Unit 1 represents a diesel engine and is controlled at a constant speed of 1800 rpm. The torque on Unit 2 is controlled according to the vehicle and transmission model and the drive cycle. The drive cycle used is the load-carry cycle, shown in figure 3.8. Briefly explained, the machine digs in a gravel pile, reverses, transports the gravel and unloads it into a load receiver positioned some distance from the pile.

**Table 3.2** Vehicle and transmission data.

Operational mass	9000 kg
Engine power	55 kW
Maximum vehicle speed	40 km/h
Maximum tractive force at standstill	92 kN
Mode shift speed (at engine speed 1800 rpm)	7 km/h
Displacement Unit 1	110 cm <sup>3</sup> /rev
Displacement Unit 2	152 cm <sup>3</sup> /rev

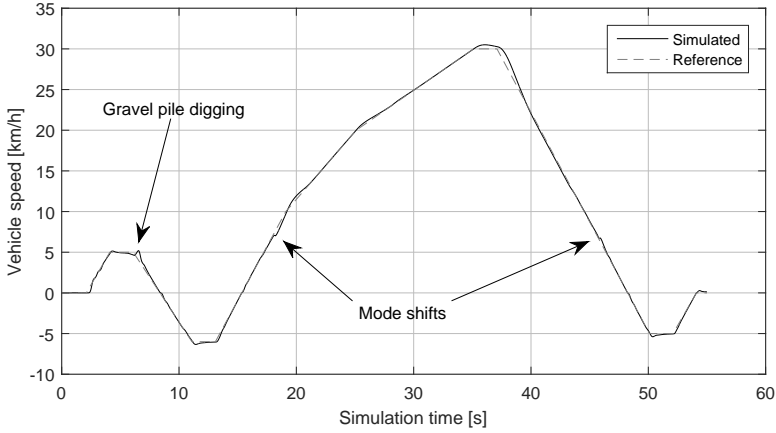


**Figure 3.8** Representation of the load-carry cycle used as reference for the constructed load cycle in the simulations, from [47].

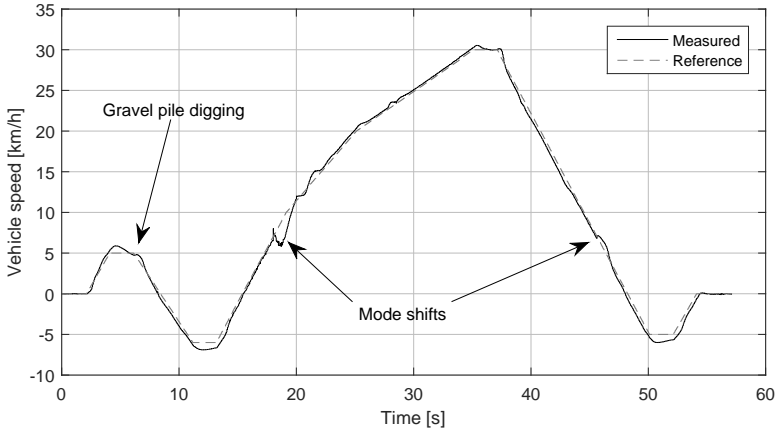
Figures 3.9 and 3.10 show the speed reference tracking and transmission pressures during the cycle, for offline and HWIL simulations. The mode shift takes place at around 7 km/h and may be noticed as a slight disturbance in speed at 18 and 46 seconds. The disturbance during the mode shift is affected by the timing of the clutches [V, 22]. Also, the instantaneous change in power flow in the HST when switching modes (see figure 3.4) means that the hydrostatic units switch between pump and motor mode. In turn, this requires a step in displacement on Unit 2 to compensate for the changes in efficiency, as shown in figure 3.11. Efficiency steps are used in commercial transmissions for improvement of the mode shift [18].

In general, the graphs for the HWIL simulations show the same pattern as the graphs for the offline simulations, although the hardware tests show a more transient behaviour, especially during the mode shift. The mode shift is a highly transient event that requires accurate dynamic modelling of the clutch engaging and disengaging, and is difficult to capture in the used setup. Another reason for the differences in behaviour is the difference in the inertia experienced by Unit 2. This will differ between the modes due to the gear configurations, as shown in table 3.3. To handle this problem, an inertia compensation is used in the HWIL simulations. Figure 3.12 shows the torques on the load side. The load side controller has a very high bandwidth and is able to track the highly transient behaviour of the reference torque, which is important in these tests. Due to the big difference in experienced inertia in the different modes, a high amount of compensation is needed. Here, the compensation is based

on the reference vehicle acceleration of the drive cycle, which means that the compensation is not particularly accurate when transient behaviour occurs. For HWIL simulations of multiple-mode concepts, the inertia compensation is very important, and to improve performance in this respect a compensation based on the measured shaft speeds could be considered. This has been done earlier in, for example, [11].

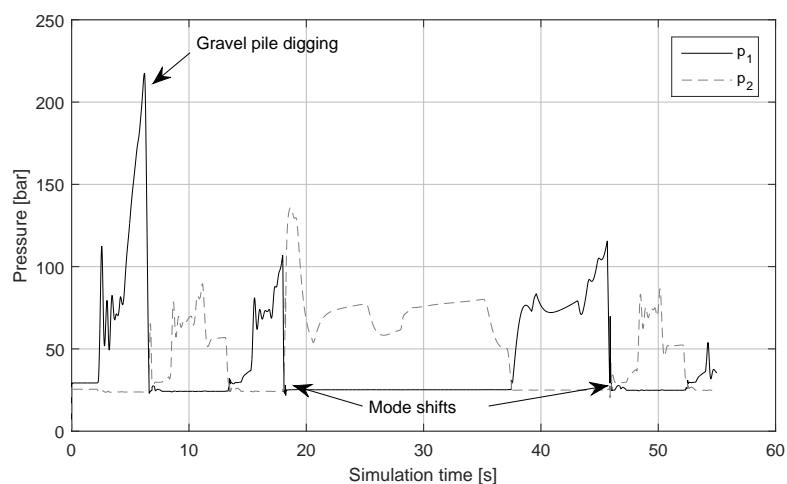


(a) Offline simulation.

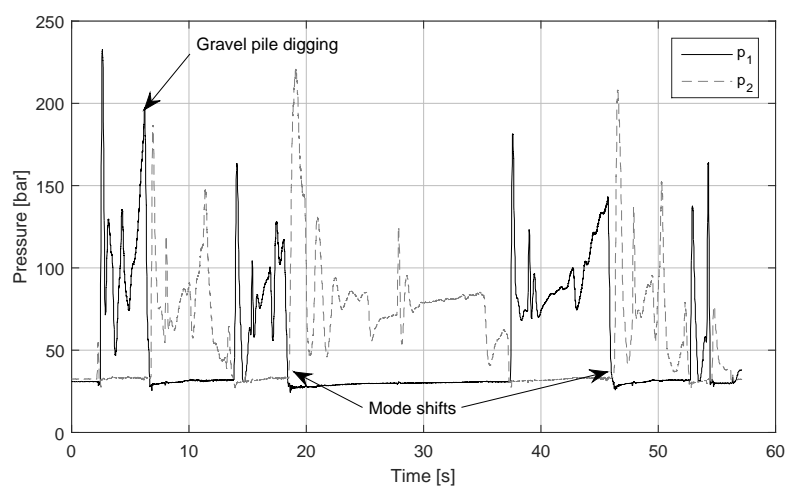


(b) HWIL simulation.

**Figure 3.9** Comparison between offline and HWIL simulations of the load-and-carry drive cycle. The mode shift ( $H \rightarrow F1$ ) takes place at 18 seconds, while the mode shift ( $F1 \rightarrow H$ ) takes place at approximately 46 seconds. The gravel pile is entered at approximately 4.5 seconds.

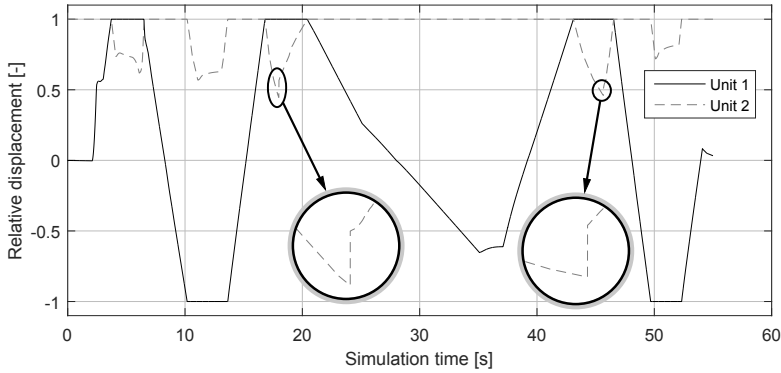


(a) Offline simulation.

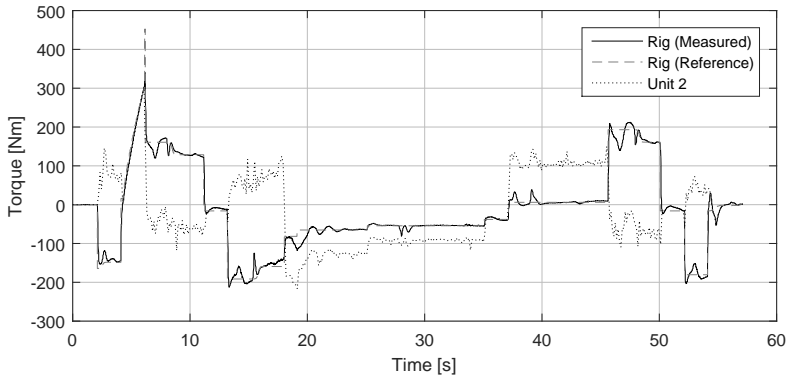


(b) HWIL simulation.

**Figure 3.10** Comparison between offline and HWIL simulations of the load-and-carry drive cycle. The mode shift ( $H \rightarrow F1$ ) takes place at 18 seconds, while the mode shift ( $F1 \rightarrow H$ ) takes place at approximately 46 seconds. The gravel pile is entered at approximately 4.5 seconds.



**Figure 3.11** Relative displacements of Unit 1 and Unit 2 during the load-and-carry drive cycle (offline simulation). A small step in relative displacement of Unit 2 is needed at the mode shifts to compensate for the difference in efficiency due to the instantaneous change of power flow direction.



**Figure 3.12** Due to compensation for the difference in inertia experienced by Unit 2, the reference torque sent to the rig controller is different from the torque experienced by Unit 2.

**Table 3.3** Values of the actual and experienced inertia for the two modes.

Installed Inertia	2.9 kgm <sup>2</sup>
Hydrostatic Mode (H)	0.75 kgm <sup>2</sup>
Power-split Mode (R1,F1)	4.2 kgm <sup>2</sup>

## **3.6 Summary**

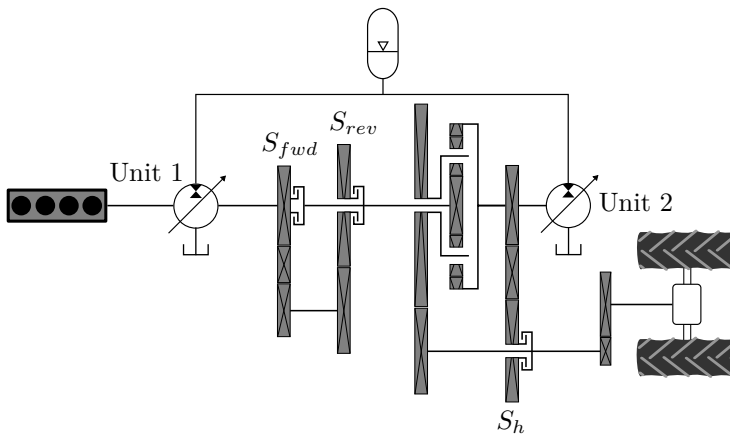
The proposed control strategy is sufficient for controlling the vehicle speed in the different modes. The mode shift is a critical event that requires accurate control of the clutches and the pumps/motors' relative displacements. This is due to the fact that the system dynamics change between the modes due to changes in experienced inertia and that the load on the hydrostatic transmission changes instantaneously during the shift.

The HWIL simulation is versatile since another transmission concept may be studied by changing the real-time model. To achieve a realistic test case, it is important that the rig inertia and the inertia in the studied system are properly matched. Also, the highly transient mode shift is difficult to simulate in the given setup. However, the results obtained from the HWIL simulation confirm the suitability of the proposed control strategy in the single modes, which is valuable information for the decision to continue with the suggested transmission concept or not.

# 4

## Mode Shifting in Hybrid Hydromechanical Transmissions

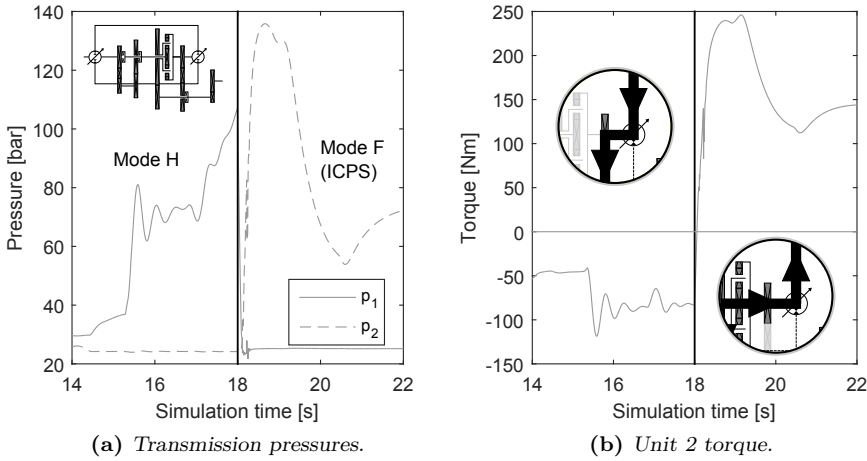
This chapter summarises the primary parts of paper [II]. It shows how the mode shift behaviour of a complex HMT is complicated if the HMT is hybridised. The HMT described in chapter 3 is used as an example and the relationship between displacement control and drop in vehicle speed due to the mode shift is illustrated.



**Figure 4.1** Hybrid version of the transmission presented in chapter 3.

In heavy construction machinery, it is of great importance to maintain high tractive forces at all times. The mode shift is therefore a critical event that must take place as smoothly as possible. As explained in the previous chapter, complex HMTs with power-split configurations allow the mode shifts to be synchronised, which facilitates disturbance-free tractive force control.

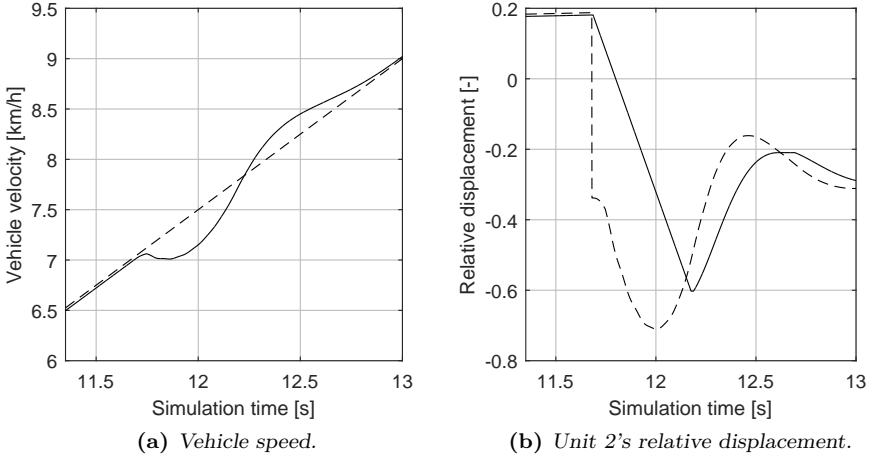
If a hybrid HMT is considered, however, the situation is not as straightforward. Figure 4.2 shows the transmission pressures and the torque on Unit 2 during a mode shift in the non-hybrid HMT studied in chapter 3. The mode shift demands a change in direction of power flow in the HST circuit. As the mode shift is synchronised, this is realised by altering the torques on the HST units. In figure 4.2 this is observable as a change in pressure sides during the shift and a change in torque on Unit 2. If a hybrid version of the transmission is considered, see figure 4.1, the pressure sides cannot switch and the high pressure is determined by the accumulator state-of-charge rather than the load. The hydrostatic machines are thereby secondary controlled and the machine torques are determined by the displacements. Instantaneous changes in displacement are therefore required to realise the mode shift. The displacement controllers are not infinitely fast and a disturbance in tractive force and vehicle speed is therefore to be expected during the mode shift.



**Figure 4.2** The power flow can change direction in the HST circuit during the mode shift in an HMT. The graphs in this figure show the transmission pressures and the torque on Unit 2 when switching from hydrostatic (H) mode to forward (F) ICPS mode (at  $T=18$  seconds) in the non-hybrid transmission described in chapter 3.



Figure 4.3 shows a simulation of a vehicle equipped with the hybrid HMT in figure 4.1 during the mode shift from hydrostatic to forward ICPS mode at constant acceleration. Unit 2 determines the output torque in both modes and is therefore responsible for controlling the vehicle speed. As seen in figure 4.3b, the mode shift implies a step in Unit 2's relative displacement from approximately 0.2 to -0.3. Due to the dynamics of the displacement controller, this does not happen instantaneously, resulting in a disturbance in vehicle speed.



**Figure 4.3** The dynamics of the displacement actuator cause a drop in vehicle speed during the mode shift. The simulation above shows the difference between reference (dashed) and simulated (solid) signals during a mode shift from hydrostatic to ICPS mode in the hybrid multi-mode HMT in figure 4.1. The model presented in paper [I] with modifications (to hybrid) according to paper [II] has been used to generate the plots.

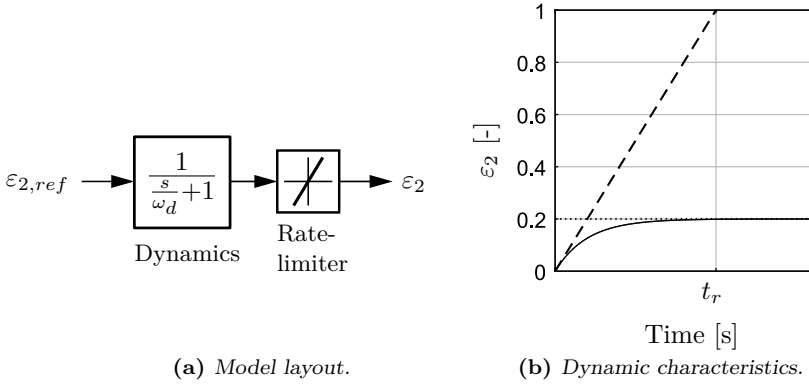
## 4.1 Mode Shifts and Displacement Control

To study the influence of the displacement controllers on the disturbance in output speed during the mode shift, the simulation in figure 4.3 is repeated for different responses of the displacement controller in Unit 2. For each simulation, the severity of the disturbance is quantified as the maximum relative velocity deviation after the shift, as calculated in equation (4.1), after steady-state errors have been removed.

$$v_{dev} = \frac{v_{veh,ref} - v_{veh}}{v_{veh,ref}} \quad (4.1)$$

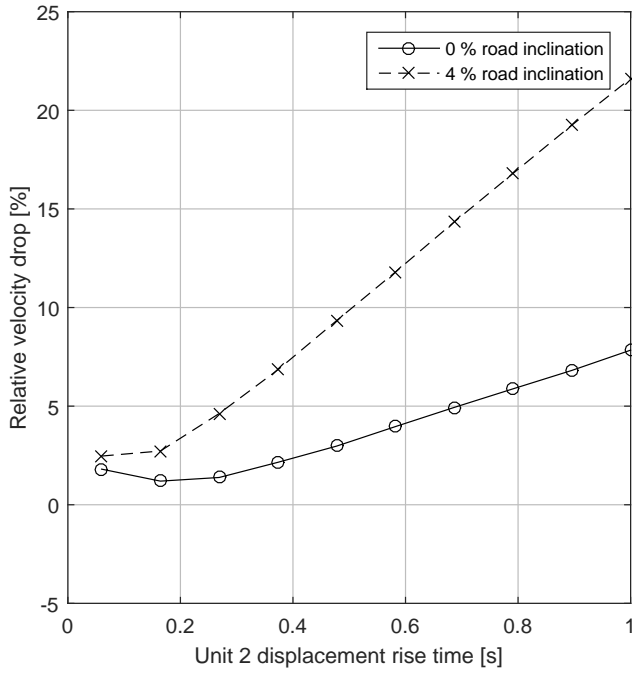
The closed loop displacement controller is modelled as in section 2.4, i.e. as a first order system with break frequency  $\omega_d$ , in series with a rate-limiter, see figure 4.4a. The rate-limiter is configured by defining the rise time,  $t_r$ , as the time required to realise a full stroke ( $\varepsilon_2 : 0 \rightarrow \pm 1$ ), see figure 4.4b. Assuming that the rate is saturated for steps in  $\varepsilon_2$  larger than 0.2,  $t_r$  may be related to  $\omega_d$  according to equation (4.2). Equation (4.2) is then used to scale the response for the different simulations, using  $t_r$  as input.

$$t_r = \frac{1}{0.2 \cdot \omega_d} \quad (4.2)$$



**Figure 4.4** Displacement controller model with rate limiter.

Figure 4.5 shows the relative drop in vehicle speed for different rise times of Unit 2. There is a clear trend in terms of more severe disturbance for a slower displacement controller. It may also be observed that the severity of the disturbance is related to the load, as larger output torques require larger relative displacement and therefore larger steps in relative displacement during the mode shift. Another noteworthy phenomenon is that the drop increases if the displacement actuators are too fast. This is explained by the fact that there are also some dynamics in the clutches, and if the displacement controllers are too fast they will affect the output torque before the clutches have changed the kinematic relationships. In turn, this stresses the importance of synchronising the clutches with the displacement controllers.



**Figure 4.5** Relative drop in vehicle speed due to mode shift for different responses of Unit 2. The drop is increased when the transmission is subjected to a higher load, here illustrated by simulating a road inclination of 4%.



# 5

## Displacement Control

As emphasised in the previous chapters, low-level control in HMTs relies on fast, accurate displacement controllers. In hybrid HMTs in particular, the displacement controllers become vital to manage power flows, torque control and mode shifts. An understanding of the dynamics and limitations of the displacement controllers is therefore central in the control of HMTs. This chapter summarises the main parts of papers [III] and [IV]. It describes the derivation of a dynamic model for the displacement controller in a prototype axial piston unit and compares two strategies for displacement control. The control strategies are confirmed in hardware tests in the test rig described in chapter 3.

### 5.1 Displacement Actuation

Different pump/motor types fulfil the demands on displacement controllers to different degrees. This thesis focuses on swash-plate axial-piston units as they are considered a suitable compromise between efficiency and displacement controllability. In swash-plate pumps/motors, varying the swash-plate angle varies the displacement. This action may be realised by different actuating systems which may be mechanical, electro-mechanical, hydraulic or electro-hydraulic. Of the four types, actuators that use hydraulics are preferable for fast response [48]. Conventional swash-plate pumps/motors commonly use internal feedback with a mechanical linkage from the swash-plate to the control valve. This is equivalent to a proportional controller and with a four-way valve solution the closed-loop response may be represented by a first order system with time constant,  $\tau$ , according to equation (5.1)[49].

$$\tau = \frac{A_1 L}{2K_q K} \quad (5.1)$$

In equation (5.1),  $A_1$  is the control piston area,  $L$  the swash-plate radius,  $K_q$  the valve flow gain and  $K$  the control linkage gain. The high stiffness of the mechanical linkage means that the valve dynamics may be ignored [50]. However, the mechanical linkage offers little freedom in the control design. By measuring the displacement angle with a sensor, an external feedback loop is created, thereby enabling more sophisticated controllers to be implemented in software. This approach, however, puts high demands on the control valve, which then dominates the dynamics. Many of the previous publications on this topic [17, 40] therefore use fast servo valves. Indeed, [17] reports very high performance, but servo valves are very expensive and are unlikely to be used in series produced pumps/motors in the short term. The approach in this thesis is therefore to study an axial piston unit with a simpler proportional valve and see how much the performance may be improved with a control strategy slightly more advanced than a standard proportional controller.

### **5.1.1 Swash-Plate Oscillations**

In the recent literature, there has been some discussion on oscillations that may occur in swash-plates. These oscillations are caused by an oscillating torque that acts on the swash-plate. The oscillations occur at a frequency proportional to the shaft speed and are caused by the pistons entering and leaving the high- and low-pressure zones of the valve plate [49]. The oscillations have been proposed to interfere with both the control [51] and for the floating-cup pump also with efficiency [52]. However, swash-plate oscillations seem to be machine-dependent and have not been observed to interfere with the displacement controllers of the axial piston units used for experiments in this thesis. Swash-plate oscillations due to the oscillating torque are therefore not considered in this work.

## **5.2 Application Example**

The Bosch Rexroth A11VO prototype pump, shown in figure 5.1, is considered for modelling and hardware tests. Some basic machine data for the specific units used in the thesis is given in table 5.1. The A11VO unit is intended to be used for secondary control in hybrid applications and therefore one modification compared to its previous version is the ability to realise negative displacement. The particular unit studied in this thesis, however, is equipped with a valve plate optimised for pump operation. All the tests have therefore been carried out with positive direction of the shaft.



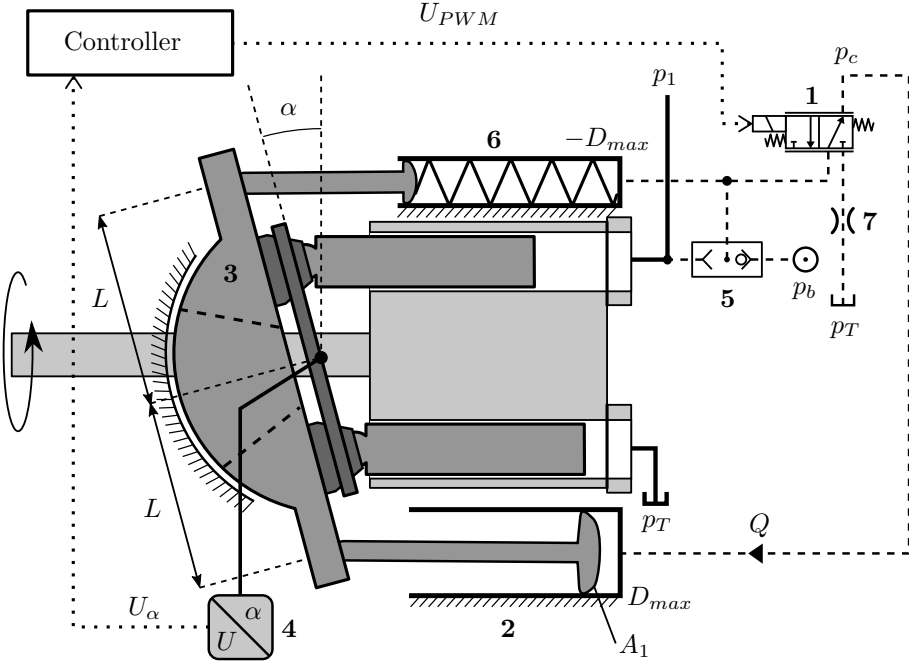
**Figure 5.1** BOSCH A11VO pump used for displacement controller modelling.

**Table 5.1** Specifications of the A11VO pump.

Property	Value
Type	In-line, axial piston
Operation principle	Open circuit
Maximum displacement, $D$	$110 \text{ cm}^3/\text{rev.}$
Displacement operating range	$-D$ to $+D$
Nominal operating pressure	350 bar
Maximum rotational speed	2800 rpm

### 5.2.1 Displacement Actuator

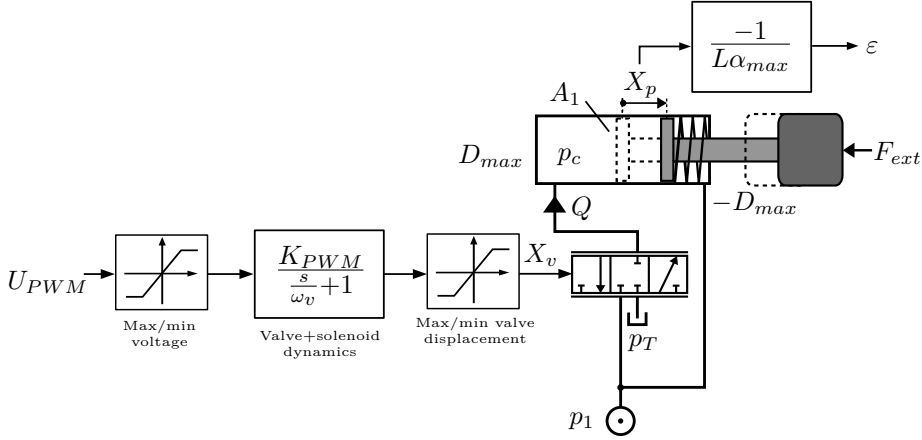
The displacement actuator circuit of the A11VO pump is shown in figure 5.2. A proportional spool valve (1) controls the flow to the control piston (2) that acts on the swash plate (3), thereby changing the displacement through the swash plate angle,  $\alpha$ . The proportional valve (1) is actuated with a solenoid that is fed with a Pulse-Width-Modulated (PWM) voltage signal,  $U_{PWM}$ . An orifice (7) is mounted at the valve tank port for extra damping of the control mechanism. The displacement angle is measured with a Hall-effect sensor (4) and sent to an external controller for closed-loop control of the displacement.



**Figure 5.2** Displacement actuator mechanism used in the BOSCH A11VO pump.

The supply pressure to the valve (1) is either an external boost pressure,  $p_b$ , or the system pressure,  $p_1$ , managed by the shuttle valve (5). Here,  $p_b$  is assumed to be used only during start-up and that  $p_1 > p_b$  during normal operation of the pump. Furthermore, a spring-loaded piston (6), supplied with the same pressure as the valve (1), acts on the swash-plate (3) in the opposite direction to the control piston (2). The spring forces the pump into maximum displacement in the case of insufficient supply pressure to the valve (1). This feature is useful in a pump as it allows pressure to build up from start-up by the pump itself. The usefulness of this feature could, however, be questioned if the unit is used for secondary control in a hybrid vehicle, as maximum displacement implies maximum output torque, which could be dangerous. It might therefore be necessary to investigate other configurations at start-up, as discussed in paper [IV].





**Figure 5.3** Displacement actuator mechanism used in the BOSCH A11VO pump represented as a three-way valve controlled piston. Note that the damping orifice ((7) in figure 5.2) has been lumped with the valve. To avoid too high solenoid currents, a limitation in valve input voltage is implemented. Also, the limited stroke of the valve is needed in the model as it limits the response of the actuator.

### 5.3 Model

To facilitate the modelling process, the displacement controller may be represented as a three-way valve controlled piston, as shown in figure 5.3. Due to the small volume in the control piston, the hydraulic eigenfrequency (as defined in [50]) is very high and dominated by the valve dynamics [17]. For the A11VO pump, it was found that a first order system with lumped valve and solenoid dynamics gave good agreement with the experiments [III]. The open loop transfer function from valve input signal,  $U_{PWM}$ , to relative displacement,  $\varepsilon$ , may then be written as in equation (5.2).

$$G_O = \frac{\varepsilon}{U_{PWM}} = \frac{K_s}{s \left( \frac{s}{\omega_v} + 1 \right)} \quad (5.2)$$

The static system gain,  $K_s$ , is lumped as shown in equation (5.3).

$$K_s = \frac{K_{PWM} K_q}{A_1 L \alpha_{max}} \quad (5.3)$$

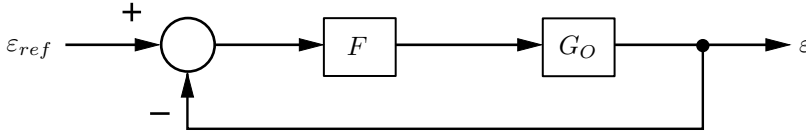
The valve flow gain,  $K_q$ , is determined according to equation (5.4).

$$K_q = -\frac{\partial Q}{\partial X_v} = \begin{cases} C_q d\pi \sqrt{\frac{2}{\rho} (p_c - p_T)}, & X_v \geq 0 \\ C_q d\pi \sqrt{\frac{2}{\rho} (p_1 - p_c)}, & X_v < 0 \end{cases} \quad (5.4)$$

Since the A11VO unit is assumed to be connected to an accumulator, the system pressure,  $p_1$ , is seen as impressed (i.e. constant from a control perspective). The tank pressure is ignored,  $p_T = 0$ . The control piston pressure,  $p_c$ , is a consequence of the control piston force equilibrium and the area ratio between the two pistons [III].

## 5.4 Control

The pure integrator in  $G_O$  requires a feedback loop for stabilisation. The controller is implemented as shown in figure 5.4.



**Figure 5.4** System ( $G_O$ , from eq. (5.2)) with controller ( $F$ ) and feedback loop.

Two candidates for  $F$  are studied. The first candidate is a standard Proportional controller (P-controller), defined in equation (5.5).

$$F = K \quad (5.5)$$

The second candidate is a Proportional controller with lead compensator (P-lead-controller), defined in equation (5.6).

$$F = K \frac{\frac{s}{\omega_{F,1}} + 1}{\frac{s}{\omega_{F,2}} + 1} \quad (5.6)$$

Neither candidate contains purely integrating elements. This is motivated partly by the existence of a pure integrator in  $G_O$  and partly by a desire to avoid problems caused by integrator wind-up when the controller is used as an inner loop in a full system. In other words, some static error is considered tolerable since it will be handled by an outer loop.

### 5.4.1 Parametrisation: Pole Placement

With pole placement, the controller is parametrised by placing the poles of the closed-loop system according to given requirements [53]. Here, it is assumed

that the input from the controller designer is desired resonance,  $\omega_a$ , and relative damping,  $\delta_a$ , of the closed loop system defined in equation (5.7).

$$G_C = \frac{F \cdot G_O}{1 + F \cdot G_O} = \frac{1}{\frac{s^2}{\omega_a^2} + \frac{2\delta_a}{\omega_a}s + 1} \quad (5.7)$$

### P-controller

Equations (5.2) and (5.5) with identification of  $\omega_a$  and  $\delta_a$  in eq. (5.7) yields that the controller gain,  $K$ , in the P-controller may be chosen with either  $\omega_a$  (equation (5.8)) or  $\delta_a$  (equation (5.9)) as input, which limits the flexibility of the controller approach.

$$K = \frac{1}{\omega_v K_s} \omega_a^2 \quad (5.8)$$

$$K = \frac{\omega_v}{4K_s} \frac{1}{\delta_a^2} \quad (5.9)$$

It may be shown that the relationship between  $\delta_a$  and  $\omega_a$  given in equation (5.10) applies.

$$\omega_a \delta_a = \frac{\omega_v}{2} \quad (5.10)$$

With a P-controller there is thus a compromise between damping and resonance (i.e. response), and to increase both of them simultaneously a faster valve is required.

### P-lead-controller

Equations (5.2) and (5.6) with identification of  $\omega_a$  and  $\delta_a$  in eq. (5.7) yields that  $\omega_{F,1}$  and  $\omega_{F,2}$  in the P-lead-controller may be chosen according to equations (5.11) and (5.12), respectively.

$$\omega_{F,1} = \omega_v \quad (5.11)$$

$$\omega_{F,2} = 2\omega_a \delta_a \quad (5.12)$$

The controller gain,  $K$ , may be chosen according to equation (5.13).

$$K = \frac{\omega_a}{2K_s \delta_a} \quad (5.13)$$

With the P-lead-controller, the damping and resonance are decoupled and may therefore be chosen independently of each other. Equation (5.11) means that there is a pole cancellation of the valve dynamics. With a  $\omega_{F,2}$  significantly higher than  $\omega_v$ , a significantly faster response with maintained damping is therefore possible with the P-lead-controller compared to the P-controller.

### 5.4.2 Choosing Resonance and Damping

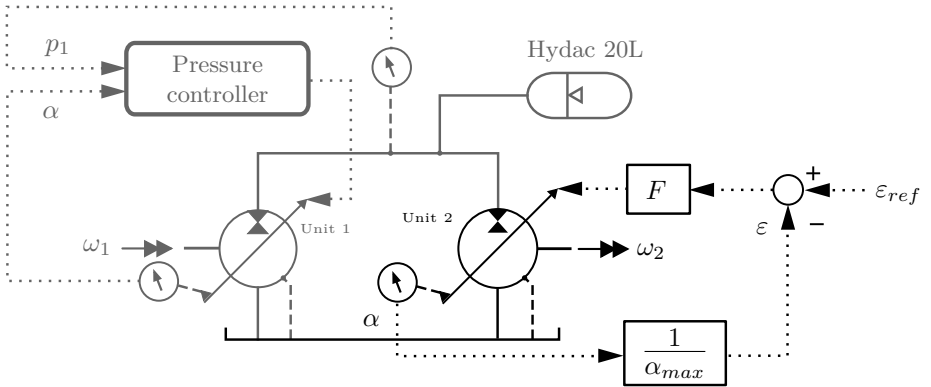
$\omega_a$  and  $\delta_a$  are achieved if the open loop system gain,  $K_s$ , and the valve break frequency,  $\omega_v$ , are known. If this is the case, the P-controller could be designed either by choosing  $\omega_a$  and accepting the resulting  $\delta_a$  or vice versa. With the P-lead-controller on the other hand,  $\omega_a$  and  $\delta_a$  may be chosen arbitrarily. This situation, however, is not entirely realistic as both  $K_s$  and  $\omega_v$  are subject to system operating point- and day-to-day-variations. These are primarily related to the shaft speed and the system pressure, which could be compensated for with gain-scheduling.

Also, the limitations in valve flow and PWM voltage (see figure 5.3) and signal noise (for the P-lead-controller) do not allow infinitely fast response. For the specific units studied here, a maximum value of  $\omega_a$  of approximately 90 rad/sec with  $\delta_a \approx 0.8$  was found adequate for the P-lead-controller.

## 5.5 Results

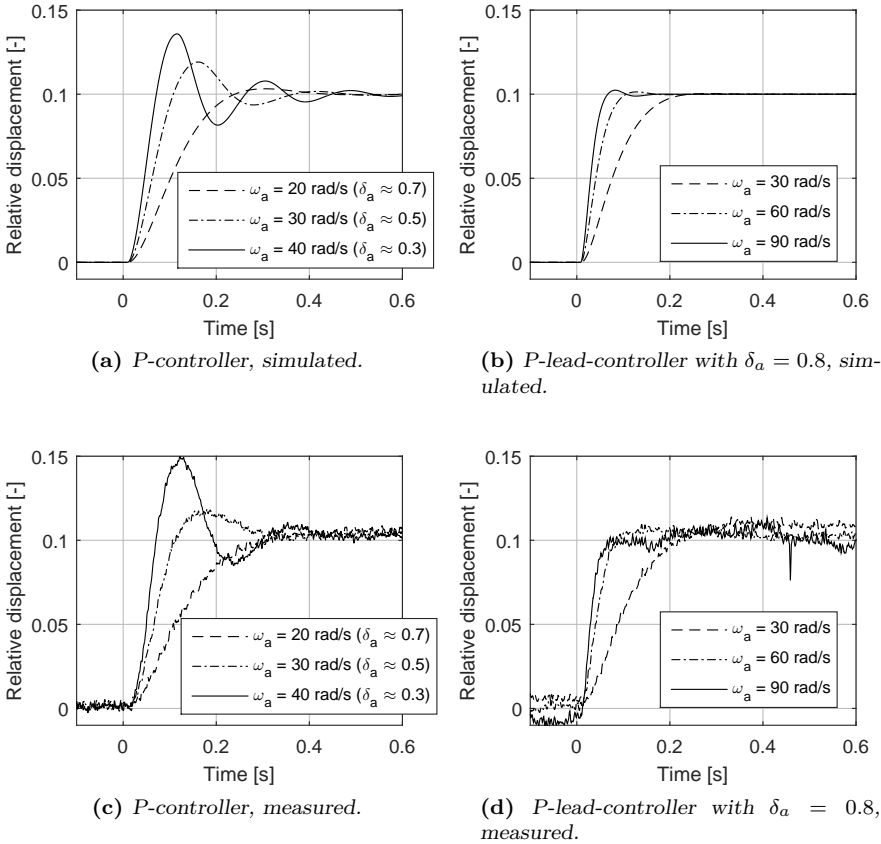
The model and the control strategies were implemented in Hopsan. The model and the proposed control strategies were also verified in experiments on an A11VO pump. This section deals with the control strategies. Experimental results from the model validation are provided in paper [III].

For the hardware tests, the test rig described in chapter 3 was modified. The modifications consisted of replacing the closed circuit hydrostatic machines with two A11VO units connected in an open circuit with an accumulator, as described in figure 5.5.



**Figure 5.5** Configuration of the transmission part of the test rig used in the experiments. Both units are controlled at constant speed and a pressure controller is implemented on Unit 1. Different control strategies could thereby be tested on Unit 2 for different speeds and pressures.

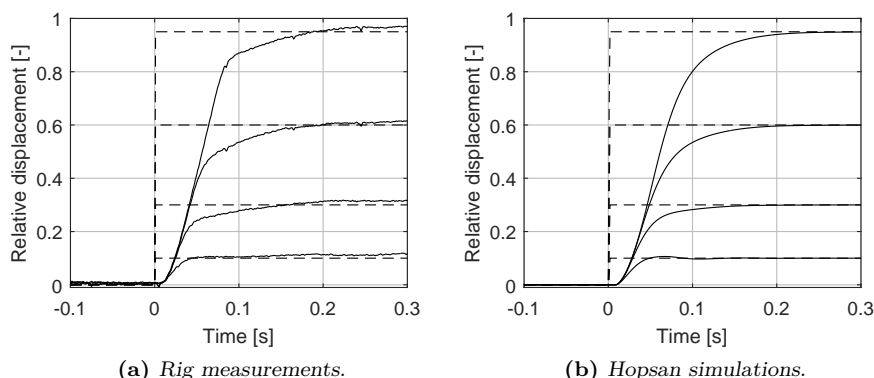
Figure 5.6 shows measured and simulated step responses for the P-controller and the P-lead-controller for different values of  $\omega_a$  and  $\delta_a$ . The P-controller was designed with a desired  $\omega_a$  as input, i.e. with equation (5.8). For both controllers,  $\omega_v = 26.6$  rad/s and  $K_s = 0.76$  were used. These values correspond to a shaft speed of 1000 rpm and a system pressure of 100 bar. The compromise with the P-controller is well illustrated in figure 5.6, as a faster response leads to decreased damping. In contrast, the response time of the P-lead-controller may be improved with a constant relative damping. The P-lead-controller manages responses with  $\omega_a = 90$  rad/s with a relative damping of  $\delta_a = 0.8$  while the P-controller is limited to  $\omega_a = 20$  rad/s for approximately the same damping.



**Figure 5.6** Simulated (top) and measured (bottom) step responses with different values of  $\omega_a$  with P-controller (left) and P-lead-controller (right). The results were obtained for a shaft speed of 1000 rpm and system pressure of 100 bar.

Figure 5.7 shows larger step responses at 200 bar with a P-lead-controller. Due to the relatively high  $\delta_a$  (0.9), the responses are still reasonably damped, even though the pressure is 200 bar (the controller was parametrised for 100 bar). The limitation in valve stroke may also be observed as a constant velocity in larger steps. Furthermore, the limitation in input voltage signal results in over-damped behaviour at the end of the larger steps.

Table 5.2 shows measured time constants for different step sizes and system pressures. It is clear that the displacement controller becomes slower as the step size increases, which is due to the limitations in voltage and valve stroke. For step sizes below 0.3, the pressure has little effect on the response. For larger steps on the other hand, increased pressure decreases the time constant primarily because of the maximum deliverable flow of the valve that increases with the pressure.



**Figure 5.7** Measured (a) and simulated (b) step responses at 1000 rpm and 200 bar with  $\omega_a = 90$  rad/s, and  $\delta_a = 0.9$  with a P-lead-controller.

**Table 5.2** Measured time constants (0→63%) in milliseconds for different step sizes and different pressures. The same controller as the one used for the results in figure 5.7 was used in all cases. All tests were carried out at a shaft speed of 1000 rpm.

↓Step size, System Pressure [bar]→	100	200	300
0.1	26	25	29
0.3	39	36	38
0.6	56	49	49
0.95	81	64	60

# 6

## Conclusions

The versatility of hardware-in-the-loop simulations allows the evaluation of control of many transmission concepts early in the design process. In turn, this evaluation yields valuable information when choosing a transmission concept. The mode shift is important to consider in the development of hybrid hydro-mechanical transmissions as it may interfere with vehicle functionality if poorly executed. However, if the displacement dynamics are known, the severity of this interference may be studied and evaluated at an early stage. This may in turn be enabled with the use of the displacement control strategy proposed in this thesis. Referring to the research questions listed in chapter 1, the following more specific conclusions may be drawn:

- RQ1:** Hardware-in-the-loop simulations may be used to verify or reject a transmission control strategy. As with any experiment, a hardware-in-the-loop simulation should be well aligned with its purpose. The example used in this thesis showed the difficulty of accurately simulating mode shifts in real time, due to both the highly transient behaviour of the mode shift and the difference in experienced inertia in the different modes. The tests should therefore concentrate on evaluation of control in a single mode.
- RQ2:** In hybrid systems, the impressed system pressure means that fast displacement controllers are needed to achieve fast torque control. Since mode shifts imply instantaneous changes in unit torques, too slow displacement controllers would cause disturbances in vehicle speed during the shift.
- RQ3:** The dynamic response of displacement controllers that use a sensor for external displacement feedback is limited by the dynamics and the maximum flow of the control valve. By using software-based controllers, more sophisticated algorithms may be used to improve the response of the displacement controller.

**RQ4:** For the pumps studied in this thesis, proportional-lead control showed significantly better performance and allowed more than 4 times as high resonance for the same relative damping compared to standard proportional control. If the valve dynamics are known, pole placement may be used to parametrise the controller using desired relative damping and resonance of the closed loop system as input. The controller may thereby be trimmed in a simple, intuitive way.



# 7

## Outlook

Hydromechanical transmissions have the potential to replace conventional transmissions in heavy construction machinery and this change is already taking place. In the quest for energy efficiency, hybridisation is a probable next step. In the short term, add-on solutions to conventional drivelines could help increase both efficiency and performance. For example, a hydraulic machine with accumulator could be added to the output shaft of the transmission and both recuperate energy and add extra torque to relieve the torque converter.

In the longer term, power-split hydraulic hybrids could be considered to increase both efficiency and productivity even more. A challenge is how to handle control earlier in the design process, as there may be many candidates for evaluation. In this respect, hardware-in-the-loop simulation could be a suitable method. The hybrid configuration also opens up for possibilities and reasons to consider all the system components, like engine, transmission and working hydraulics, simultaneously in the control.

An increased use of hybrid systems will imply a high dependence on displacement controllers. The majority of pumps and motors available on the market today have been designed for non-hybrid applications and the transition towards hybrid systems may change the performance requirements concerning displacement controllers. After defining these requirements, the use of integrated controllers and additional sensors seems like a rational path towards fast, accurate displacement control.



# 8

## Review of Papers

### Paper I

#### **Simulation aided Design and Testing of Hydromechanical Transmissions**

This paper shows the use of simulations for conceptual design of complex hydro-mechanical transmissions. A full-vehicle Hopsan model of a small construction machine with a two-mode complex transmission is presented. Control algorithms for the transmission are developed and tested in simulation and in a transmission test bed. The results are confirmed in hardware-in-the-loop simulations on the test bed, in which the mechanical part of the transmission is simulated in real time and the hydrostatic transmission path is represented as hardware. The benefits and challenges of hardware-in-the-loop simulations in the design process of complex hydromechanical transmissions are shown.

### Paper II

#### **Mode Shifting in Hybrid Hydromechanical Transmissions**

In this paper, the problem of shifting modes in hybrid hydromechanical transmissions is identified. The problem is linked to the hydraulic accumulator's influence on the pressure in the hydrostatic circuit. A black box model approach is used to identify mode shift categories and how the mode shift affects the torques on the hydrostatic transmission for the different categories. Two hydrostatic transmission circuit configurations are presented, one of which includes a high-speed switching valve. The transmission concept studied in paper [I] is hybridised and the two hydrostatic transmission configurations are evaluated in simulations. The benefits of the high speed switching valve and the significance of fast displacement actuators on the hydraulic units are shown.

## **Paper III**

### **Modelling of the Swash Plate Control Actuator in an Axial Piston Pump for a Hardware-in-the-Loop Simulation Test Rig**

Here, a model of a swash plate control actuator for a prototype pump of axial piston, in-line design is derived. The model is implemented in Hopsan and validated with rig measurements. A sensitivity analysis is made to identify the parameters of highest significance in the dynamic behaviour of the control actuator. The major limiting factors of the displacement response are identified as the maximum control valve opening area and the break frequency of the control valve spool.

## **Paper IV**

### **Displacement Control Strategies of an In-Line Axial-Piston Unit**

In this paper, two controller architectures (proportional and proportional-lead) for displacement control of the prototype pump modelled in paper [III] are proposed. The controllers are parametrised using a pole-placement approach, in which the desired relative damping and resonance of the closed-loop system are used as input. In simulations and hardware tests, the proportional-lead controller's performance is found to be superior to that of the proportional controller in terms of fast response with maintained damping. The response of the displacement controller is found to ultimately be limited by the maximum flow of the control valve, especially if the controller reference signal is a step with high amplitude.

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