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SIMULATION AIDED DESIGN AND TESTING OF HYDROMECHANICAL TRANSMISSIONS

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ABSTRACT

This paper demonstrates the use of high-speed simulation in transmission conceptual design and presents a transmission test bed for hardware-in-the-loop simulations of hydromechanical transmission concepts. Complex transmissions, such as multiple-mode hydromechanical transmissions and hydraulic hybrid transmissions, present new difficulties and costs in the development process. There is today a greater demand for more efficient product development and more work has shifted towards simulation. The Hopsan simulation package allows robust, high-speed simulations suitable for both offline and hardware-in-the-loop simulation. New simulation models for hydromechanical transmissions are developed and used to simulate a known two-mode transmission concept. The same concept is also tested in hardware-in-the-loop simulations in the proposed transmission test bed. Results show good agreement with the hardware tests and highlight the proficiency of the simulation tools.

KEYWORDS

Hydromechanical Transmission, Hardware-in-the-loop, Hopsan

NOMENCLATURE

B	Viscous friction
D	Hydraulic displacement
F	Force
J	Inertia
P	Power
R	Planetary gear ratio
T	Torque
i	Spur gear ratio
α	Elevation angle
ε	Displacement setting
η	Efficiency
ω	Angular speed

INTRODUCTION

There is today an increased need for more efficient product development methods to shorten lead times and lower costs. One way to accomplish this is to shift development work from prototype testing to simulation using model-based product development. This way, further knowledge can be gained earlier in the development process without physical testing and expensive prototype demonstrators. Moreover, it is possible to push the design decisions forward in the development process and apply a concurrent engineering approach [1].

In the development of high-efficient drive transmissions, such as multiple-mode Hydromechanical Transmissions (HMTs) and hydraulic hybrids, system complexity and the control effort required are greater than for conventional drive transmissions. For construction machines, there are also additional power consumers in the working hydraulic system, which further complicates the control task. In the development of multiple-mode HMTs, there is a great variety of possible transmission concepts due to the logarithmic increase in possible configurations for every additional mode. Complex HMTs contain a large number of mechanical gears, which makes them difficult to design [2]. For these reasons, model-based product development is central to a successful transmission development process.

To further decrease the need for vehicle testing Hardware-in-the-loop (HWIL) simulation is a powerful tool. Here, parts of the system can be tested and other parts of the system simulated. By having the simulation models interact with hardware subsystems in real-time, a more realistic test case is achieved. Furthermore, the test environment can be more flexible since it is easier to alter a simulation model than a hardware installation. HWIL simulations refers traditionally to the testing of the control code and communication interface of a physical control unit (the hardware) in a simulation environment, sometimes denoted "Controller-in-the-loop". Today, the use of HWIL simulation is rapidly growing and includes testing of more subsystems in earlier phases. In fact, Fathy et al [1] state that a paradigm shift has occurred where HWIL simulation is transformed from a controller prototyping tool to a method for system synthesis. Here, a high amount of power is active in the simulation which differentiates "Power-in-the-loop" from HWIL. In the development of drive transmissions, it provides a "middle way" between pure virtual prototyping and full vehicle tests and is suitable for testing identified critical components of the transmission. One early study including Power-in-the-loop is [3], where HWIL simulation was used to study the stability characteristics of load-sensing hydraulic systems using a simulation model of the load-sensing pump to control a servo valve connected to a real crane.

Figure 1 shows a simplified transmission development process where hardware tests make up an increasing part of the product assessment throughout the development process. For an efficient development process, it is proficient to have high-speed simulation models that can be reused throughout the design process without compromising on fidelity. It is also important to have the same controller interface in both offline simulations as well as in HWIL and full vehicle tests. This way, the control code can also be reused throughout the development process.

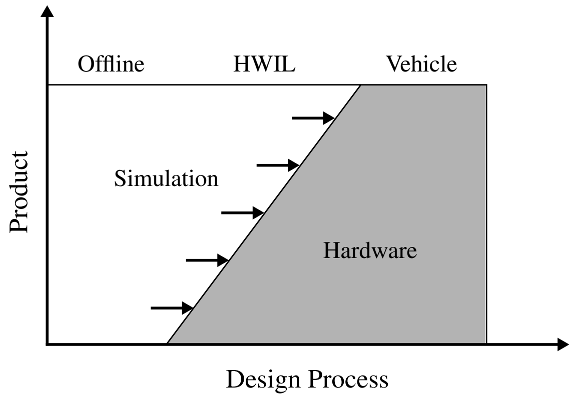


Figure 1: Idealized transmission development process, adapted from [4].

Hydromechanical Transmissions

HMTs are transmissions that transfer power both hydraulically and mechanically, either in series, like a Hydrostatic Transmission (HST) with mechanical gear steps, or in parallel. The parallel *power-split* architecture divides the power into a mechanical branch and a hydrostatic branch. In high-power applications, single-mode transmissions are often not enough to meet the high tractive force requirements. Instead, the basic hydromechanical transmissions can be combined into a multiple-mode transmission by using clutches. With multiple modes, a higher efficiency can be achieved since less power is transferred hydraulically. The number of possible multiple-mode concepts increases logarithmically with the number of modes and the variety of possible transmission concepts is consequently very large.

Today, hydrodynamic transmissions (torque converter) dominate the market for high-power classes due to its cheap and robust design and well-known behavior to the operators. There is, however, a clear market trend to replace the conventional transmission in wheel loaders in particular. Multiple-mode HMTs are strong candidates in this respect [5]. In this paper, a known two-mode HMT concept is considered, see Figure 2. The first mode (H) is purely hydrostatic, where all power flows through the HST. This mode is used for both forward and reverse motion by controlling Unit 1 over centre. The second mode is input-coupled power-split and allows for an increased range and higher efficiency at high vehicle speeds. This concept is relatively simple in its functionality and requires one over-centre variable pump/motor (Unit 1) and one fixed or variable pump/motor (Unit2).

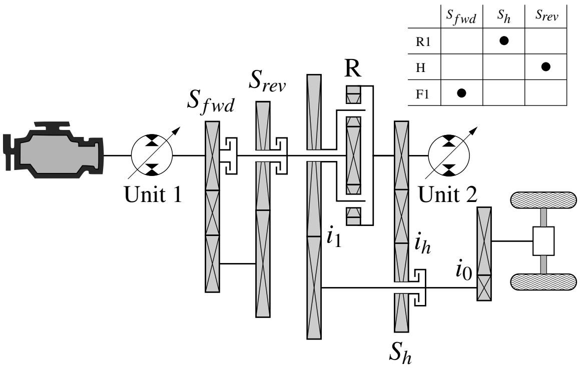


Figure 2: The considered transmission concept and clutch arrangement.

Hopsan

Hopsan is a multi-domain system simulation tool developed at Linköping University [6]. It is based on the transmission line element method, so that sub-models can be decoupled by transmission line elements and thereby computed independently using distributed equation solvers. This results in robust, scalable simulations with good performance. It also makes the simulations naturally parallel and thus suitable for multi-core processors [7]. Other features include optimization, sensitivity analysis tools and connectivity with other software. Real-time simulations are performed by interfacing with LabView.

Contributions

This paper demonstrates the use of the Hopsan simulation package for transmission conceptual design in early design phases and presents a transmission test bed for HWIL simulations of HMT concepts. New Hopsan component models have been developed and used to simulating a known two-mode transmission concept used in a small construction machine. Control algorithms have been developed and tested in simulation and in the transmission test bed. The simulation results have also been confirmed in HWIL tests on the test bed, where the mechanical part of the transmission is simulated in real-time and the HST constitutes the hardware test object.

MODELLING AND SIMULATION

This section describes the main component modelling done in Hopsan and the complete simulation model for the considered transmission concept. The modelling in Hopsan is based on wave propagation using Transmission Line Modelling (TLM) [8]. The component models derived in this section are previously known models adapted to TLM. The models not described here are taken directly from the standard component library in Hopsan [9].

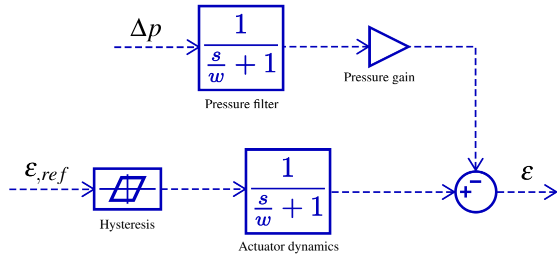
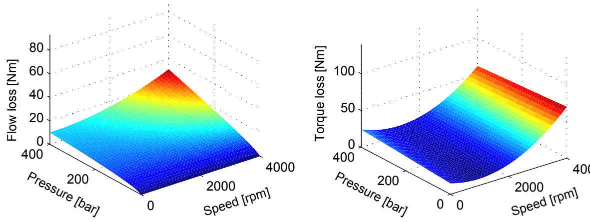


Figure 3: Model of the pump/motor control unit in Hopsan.



(a) Flow losses (b) Torque losses

Figure 4: Loss models for a hydraulic pump/motor of in-line design at full displacement with respect to pressure and angular speed.

Hydraulic Pump/Motor

The hydraulic pump/motors are created by modifying the standard Hopsan pump/motor model to consider control unit dynamics and a more advanced power loss model. The control unit is modelled by means of a simple first order filter to represent the actuator dynamics and a small hysteresis according to Figure 3. Tests have also shown that the system pressure has a minor influence on the displacement of the in-line machine, which is modelled with a filtered pressure feedback. The power losses are modelled using the POLYMOD method [10] with respect to pressure, speed, and relative displacement, see Figure 4. The number of polynomial coefficients have been reduced to keep the models simple and avoid a high number of numerical operations. Even though the precision of the model is compromised, validation tests have shown that it can be sufficiently accurate for the operating range of interest, see [11].

Planetary Gear

The three-shaft planetary gear component used in the considered transmission concept is modelled according to Figure 5. Each shaft carries a rotational inertia with viscous friction and the complete component is thus modelled as a restrictive "Q-component" [9]. The torque relations are described in Eq. (1) with the internal torque on

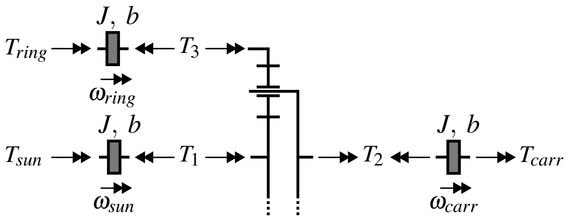


Figure 5: Model of the planetary gear with one inertia for each shaft.

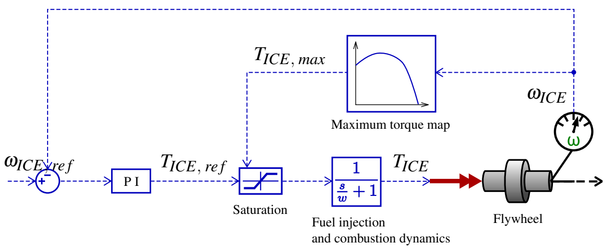


Figure 6: Simple engine model in Hopsan with maximum torque curve.

each shaft:

$$J\dot{\omega}_{sun} + b\omega_{sun} = T_{sun} - T_1 \tag{1a}$$
$$J\dot{\omega}_{carr} + b\omega_{carr} = T_{carr} - T_2 \tag{1b}$$
$$J\dot{\omega}_{ring} + b\omega_{ring} = T_{ring} - T_3 \tag{1c}$$
$$T_1 + T_3 + T_2 = 0 \tag{1d}$$
$$T_3 = -R \cdot T_1 \tag{1e}$$

The speed relation is described according to Eq. (2):

$$R = \frac{\omega_{sun} - \omega_{carr}}{\omega_{ring} - \omega_{carr}} \tag{2a}$$

(2b)

The complete TLM equations are not shown here since they do not provide a better understanding of the model.

Combustion Engine

The combustion engine of the driveline is important in terms of both fuel consumption and controller design [12]. In this study, it is considered to be an electrically controlled direct injection engine and is modelled with an assembly of standard Hopsan components according to Figure 6. The model includes a maximum torque curve, first order control dynamics, and a flywheel to represent the complete engine inertia. Fuel consumption is not considered in this study. If required, however, the model can be easily adapted to include a fuel consumption map. An internal electronic speed control is used to track the reference speed given by the operator, see for instance [13, 14]. In this model it is represented by a simple PI-controller.

Clutch

The clutch model is a simple representation of a wet disc clutch where the input signal limits the maximum transmittable torque between the two ports. The clutch is modelled as a capacitive ”C-component”. When engaged, the transmittable torque is not restricted and the clutch acts as a mechanical rotational shaft. The simplicity of the model is motivated by the considered concept, where mode shifting occurs only at synchronised shaft speeds.

Vehicle

The vehicle model is based on a simple stiff mass moving in one dimension with rolling friction and a restrictive force from the considered loading material. The motion equation of the vehicle follows Eq. (3):

$$m_{veh}\dot{v}_{veh} = F_t - F_g - F_r - F_{load} \tag{3}$$

Here, F_t is the tractive force from the vehicle driveline according to:

$$F_t = T_0 \frac{\eta_0}{i_0 r_{tire}} - B_{axle} v_{veh} \tag{4}$$

where η_0 is the constant mechanical efficiency of the wheel axles, B_{axle} is the speed-dependent losses of the wheel axles, r_{tire} the wheel radius and T_0 the transmission output torque. The grade resistance, F_g, in Eq. (3) depends on elevation angle α according to Eq. (5).

$$F_g = m_{veh} g \sin(\alpha) \tag{5}$$

The rolling friction, F_r, in Eq. (3) is dependent on the elevation angle and the mass of the vehicle according to Eq. (6).

$$F_r = C_r m_{veh} g \cos(\alpha) \tag{6}$$

where C_r is the rolling friction coefficient. The restrictive force produced during digging operations, F_{load}, is more challenging to model accurately since it varies with, for instance, material properties, the geometry of the pile, environment etc. It is also highly dependent on the position of the bucket, which is not simulated here. To avoid a complex gravel pile representation, statistical data from real-world operation have been used to adapt an empirical one-dimensional model where the force varies only with the position and speed of the vehicle during the digging operation. The bucket is consequently assumed to be moved in the same way when excavating in all loading cycles. A more advanced model is shown in, for instance, [15], where a two-dimensional model is shown to be enough to accurately model the digging forces of a wheel loader.

Transmission Model

The complete model assembly of the transmission concept is shown in Figure 7. The inputs to the model are engine reference speed, reference displacement settings of

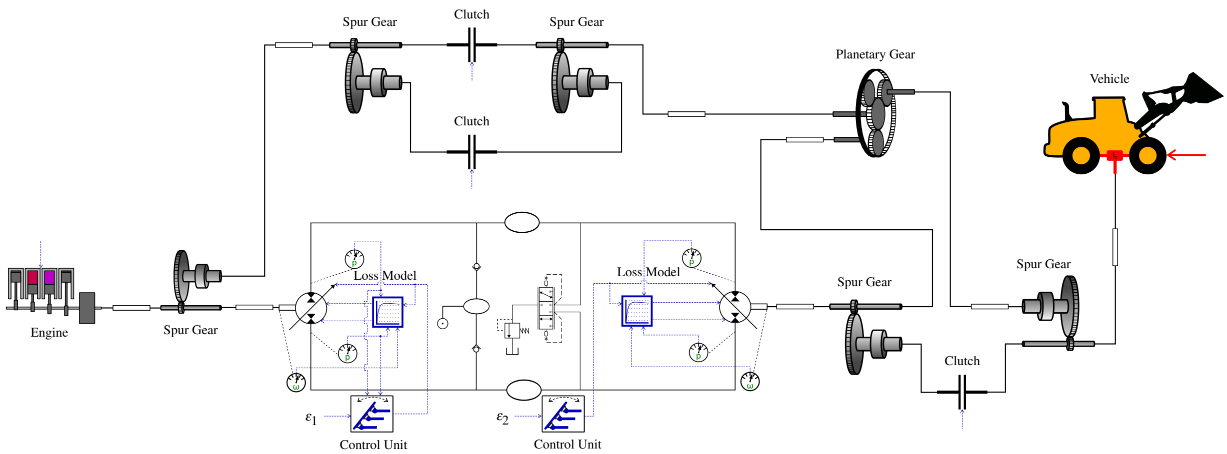


Figure 7: Complete Hopsan simulation model of considered HMT concept and vehicle.

the hydraulic machines and clutch control signals. The HST is modelled with standard components and include a boost circuit with a fixed pressure source and a flushing circuit. The mechanical shafts connecting the components are from the standard component library and represent a stiff spring that is modified to achieve the desired numerical properties. The power consumed by the work functions will affect the control of the engine, but is not considered in the depicted model. Hopsan, however, is also suitable to model hydraulic systems such as the work functions of a wheel loader [6].

HARDWARE-IN-THE-LOOP TEST BED

A schematic of the HWIL transmission test bed is shown in Figure 8 and consists of an HST and two simulator systems for controlling the input and output shafts of the HST. The test bed was first developed by Jansson et al. [16] and then used by Lennevi [17] to develop control algorithms and study dynamic properties of hydrostatic transmissions. Today, the electronic control and measurement system has been updated to allow for more advanced HWIL simulations including multi-core support with the current version of Hopsan. The HST transmission has also been replaced with modern electronically controlled axial-piston machines. The test bed is today also equipped with hydraulic accumulators to enable tests of hybrid transmissions. This, however, is not included in this paper.

The left side (the drive side) is used to simulate the engine of the vehicle and a mechanical gear configuration depending on the considered HMT concept. The right side (the load side) is similarly used to simulate the load side of the HST including the mechanical gear configuration, vehicle dynamics and gravel pile. The drive side and the load side both operates in four quadrants with an ability to drive and brake in both directions. Since the test bed incorporates only an HST as test object, it allows for testing of different HMT concepts and designs by easily replacing the models of the mechanical configuration. This way, HWIL simulations can be used earlier in the design process and before the final concept design choice is taken.

The drive side is equipped with a servo valve controlled fixed displacement pump connected to a flywheel inertia. It is speed-controlled in closed loop to represent the static and dynamic behaviour of the diesel engine, as modelled in Figure 6. The inertia in the Hopsan model is in this case omitted since the physical inertia on the drive shaft represents the engine inertia. In this study, no mechanical gear configuration is modelled since Unit 1 is directly connected to the diesel engine in the considered concept.

The load side has the same configuration and is used to generate a load torque on the output of the HST. The flywheel inertia represents the inertia of the load side and the vehicle mass. As the experienced inertia changes depending on the current mode, the generated torque must compensate for this. Jansson et al. [16] show in theory and experiments a method for online compensation of the inertia within limits for this test setup. The torque control of the load side is challenging, since there is a strong coupling between shaft speed and torque due to the servo valve characteristics. In effect, one valve opening results in different load torques depending on the shaft speed. Expectations regarding the load side controllers are also very high due to the stiff behaviour of the gravel pile.

The control and measurement system of the test bed is schematically described in Figure 9. The system is based on National Instrument's LabView and includes a quad-core real-time PXI computer that handles the signal processing and interface with the Graphical User Inter-

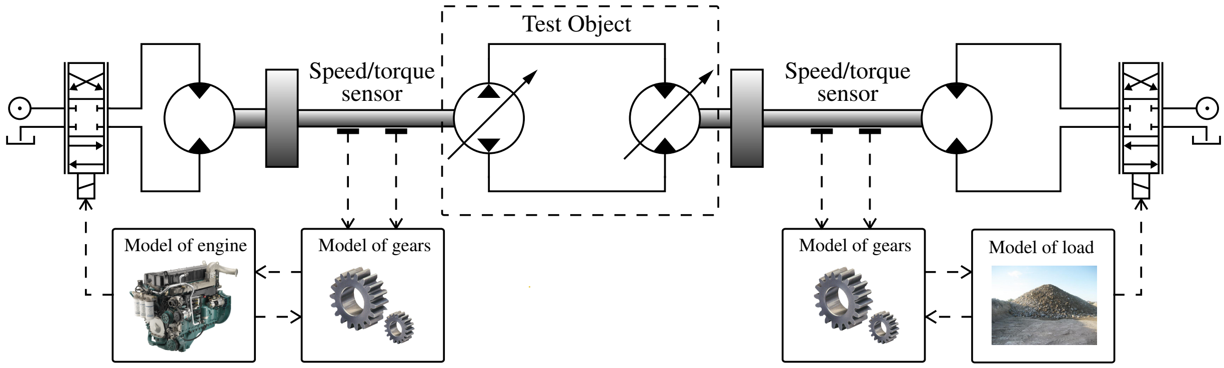


Figure 8: Hardware-in-the-loop transmission test bed for testing HMT concepts.

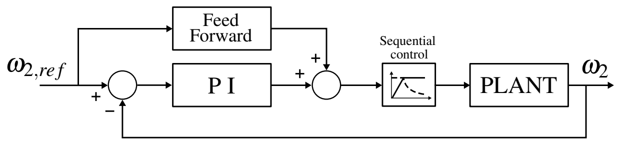


Figure 10: Overview of the control concept for the HST.

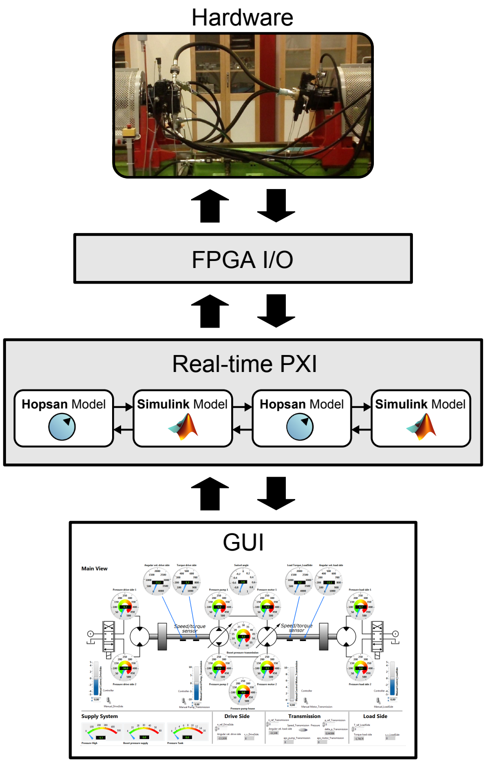


Figure 9: Schematic of the control and measurement system on the transmission test bed.

face (GUI). The sampling rate is 1 kHz and the system supports parallel simulations with compiled MATLAB Simulink and Hopsan models. For more information on the control and measurement system, see [11].

TRANSMISSION CONTROL

To control the transmission, the Hopsan model is compiled and exported to MATLAB Simulink, where the control algorithms are developed. The main transmission control is a simple PI-controller with a feed-forward link as described in Figure 10. The feed-forward link is based on known steady-state models of the hydrostatic transmission with constant efficiencies for the hydraulic machines. The input to the left is the reference speed of Unit 2. It is calculated from the reference vehicle speed using the known kinematic models for the gear configuration of the current mode. The control signals of the HST are coupled for sequential control of the displacements. The engagement and disengagement of the clutches are actuated when the shaft speeds on each side are synchronised to avoid loss of tractive force. Since the tests are run at a constant engine speed, the mode shifts occur at fixed vehicle speeds depending on the design of the gear configuration. The timing of engaging and disengaging the clutches influences the smoothness of the mode shifts, but is not studied in detail in this paper.

RESULTS AND DISCUSSIONS

In this section, the control algorithms developed in MATLAB Simulink are used to control the offline simulation model in Hopsan as well as in HWIL. In the HWIL

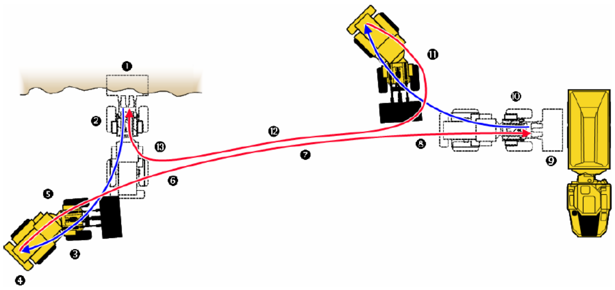


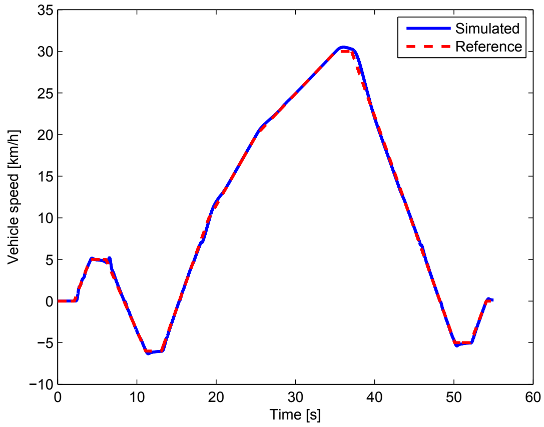
Figure 11: Representation of the load-carry cycle used as reference for the constructed load cycle in the simulations, from [19].

Parameter	Quantity	Value
Operational mass	m_{veh}	9000 kg
Engine power	P_{ice}	55 kW
Wheel radius	r_{tire}	0.65 m
Displacement Unit 1	D_1	110 cm^3/rev
Displacement Unit 2	D_2	152 cm^3/rev
Planetary gear ratio	R	-0.62
Hydrostatic gear ratio	i_H	0.28
Power-split gear ratio	i_S	1.73
Final gear ratio	i_0	1/20
Engine reference speed	$\omega_{ICE,ref}$	1800 rpm

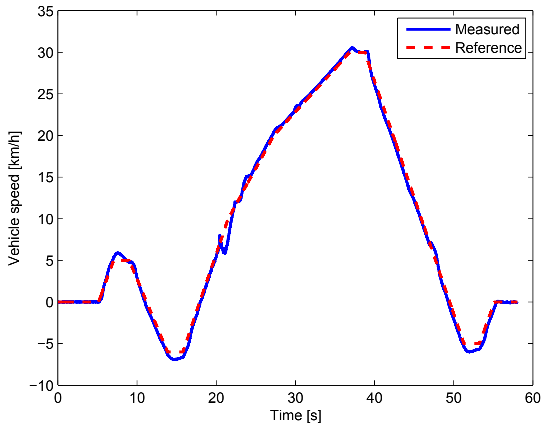
Table 1: Main simulation parameters

simulations, the engine, mechanical gears and vehicle are simulated, while the HST is tested in hardware. The control algorithms of the test bed are based on the work of Lennevi [17] and are further described in [18]. The reference cycle is an idealised load-carry cycle constructed with linear segments. A representation of the load-carry cycle is shown in Figure 11. 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. As a reference, the simulation time in Hopsan for the 55-second cycle described below was 2.5 seconds on a 2.7 GHz Intel Core i5-3340M processor with a 0.1 millisecond step time. Using multiple cores did not improve the simulation time due to the simplicity of the model.

Table 1 shows the main parameters of the simulated driveline for the considered construction machine application. The displacements of the HST correspond with the pump/motors installed in the test bed and are of reasonable sizes to meet the tractive force requirements of the vehicle application. Figure 12 shows the speed reference tracking from the offline simulation and the hardware experiments. At around 7 km/h, the mode shift occurs, causing a minor disturbance in vehicle speed. This problem is related to the clutch control and modelling and



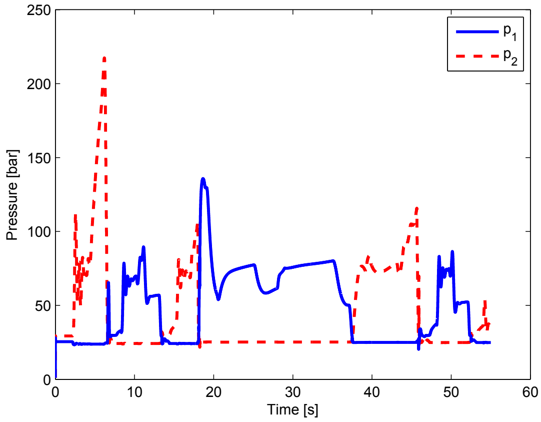
(a) Offline Simulation



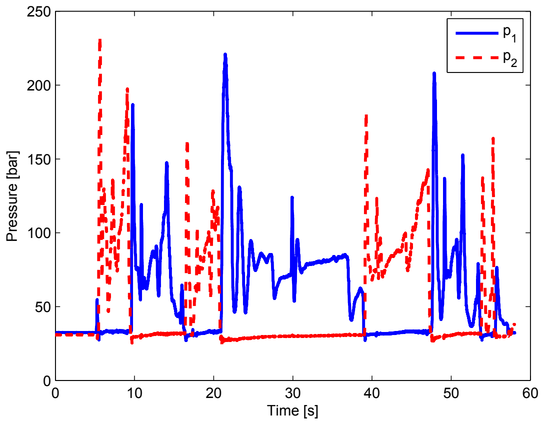
(b) HWIL Simulation

Figure 12: Vehicle speed reference tracking for HWIL and offline simulations for the load-carry cycle.

causes the tractive force to drop slightly. Figure 13 shows the corresponding system pressures of the HST from the simulated cycle. The graphs follow the same pattern even though the highly transient behaviour in the hardware is not completely covered in the models. At the mode shift, there is a clear difference which is due to difficulties in modelling the highly transient behaviour when the high and low pressure sides switch. This problem is also noticeable in Figure 12. As described above, the models are kept on a simple level and could be refined to better represent the detailed behaviour of the transmission, such as clutch friction and slip. Figure 14 shows the torque on the load side of the transmission and the actual experienced motor torque for the current mode. The latter includes the compensation of the physical inertia for the different modes. The bandwidth of the torque control is very high and it is able to track the stiff dynamics of the load side. Much of the generated shaft torque (on the right



(a) Offline Simulation



(b) HWIL Simulation

Figure 13: Transmission pressure for HWIL and offline simulations for the load-carry cycle.

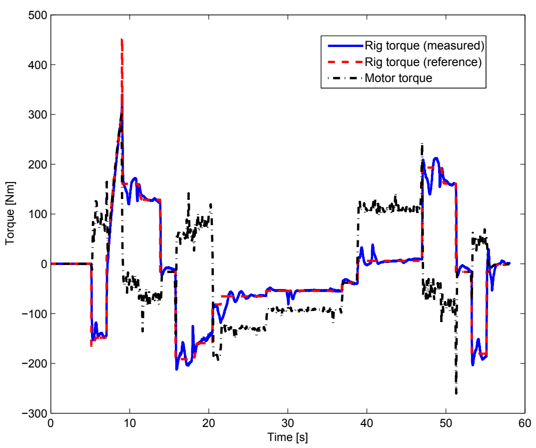


Figure 14: The load torque during the load-carry cycle.

Installed Inertia	2.9 kgm^2
Hydrostatic Mode	0.75 kgm^2
Power-split Mode	4.2 kgm^2

Table 2: Values of the actual and experienced inertia for the two modes.

side of the inertia in Figure 8) is inertia compensation based on the acceleration of the vehicle. This is a critical part of simulating multiple-mode concepts in the test bed. Table 2 shows the values of the experienced and actual inertia for the given application: So far, the compensation is based only on the reference signal and not on measured values as in [16]. See [11] for a deeper analysis of the dynamic effects of the change in inertia.

CONCLUSIONS

In this work, simulation models for simulating multiple-mode hydromechanical transmissions have been established and tested using high-speed system simulation. The simulations show good agreement with hardware tests and the control algorithms developed have been shown to accurately control the considered transmission concept in simulation and in hardware-in-the-loop simulations. The results show that Hopsan is a useful tool for simulating hydromechanical transmissions and that hardware-in-the-loop simulations of hydromechanical concepts are powerful and relatively simple to make with the proposed setup. Future work includes more advanced modelling and tests of different hydromechanical configurations with and without hydraulic energy storage.

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