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ARTICLE

Framework for simulation-based simultaneous system optimization for a series hydraulic hybrid vehicle

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ABSTRACT

Hybridisation of hydraulic drivetrains offers the potential of efficiency improvement for on – and off-road applications. To realise the advantages, a carefully designed system and corresponding control strategy are required, which are commonly obtained through a sequential design process. Addressing component selection and control parameterisation simultaneously through simulation-based optimisation allows for exploration of a large design space as well as design relations and trade-offs, and their evaluation in dynamic conditions which exist in real driving scenarios. In this paper, the optimisation framework for a hydraulic hybrid vehicle is introduced, including the simulation model for a series hybrid architecture and component scaling considerations impacting the system's performance. A number of optimisation experiments for an on-road light-duty vehicle, focused on standard-drive-cycle-performance, illustrate the impact of the problem formulation on the final design and thus the complexity of the design problem. The designs found demonstrate both the potential of energy storage in series hybrids, via an energy balance diagram, as well as some challenges. The framework presented here provides a base for systematic evaluation of design alternatives and problem formulation aspects.

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ABBREVIATIONS

ARVD: Average Relative Velocity Deviation; BSFC: Brake Specific Fuel Consumption; NEDC: New European Driving Cycle; SHHV: Series Hydraulic Hybrid Vehicle; UDDS: Urban Dynamometer Driving Schedule; WLTP3: Worldwide harmonised Light vehicles Test Procedure (class 3)

1. Introduction

In the quest for increased energy efficiency in vehicular transmissions to lower fossil fuel consumption, hybridisation is generally seen as a viable solution. A hybrid transmission contains at least two power sources and typically allows for storage of recuperated energy. In vehicular hybrid drivetrains, the traditional internal combustion engine (primary power source) combined with a mechanical, an electric or a hydraulic secondary energy storage (flywheel, battery or hydraulic accumulator, respectively). Hybrid electric vehicles are already found in many commercial applications, such as passenger vehicles (Yang *et al.* 2016). The hydraulic alternative is interesting for a number of reasons, the higher power density of hydraulic components compared to their electric counterparts in particular. Consequently, hydraulic hybridisation of drivetrains lends itself most easily to existing hydraulic transmissions (off-road machinery), or to high power transient applications, such as city buses, (inner-city) delivery trucks and refuse collecting vehicles, whose usage profile is characterised by frequent braking and acceleration manoeuvres.

The main components of a traditional vehicle transmission, combustion engine and mechanical gearbox are typically dimensioned according to maximum performance requirements (Guzzella and Sciarretta 2013). This approach is usually applied to (hydraulic) hybrid transmissions as well (e.g. Surampudi *et al.* 2009), at times supported by parameter studies (e.g. Bowns *et al.* 1981). Research then focuses on determining optimal or near-optimal control strategies for fuel efficiency and other targets (e.g. Filipi *et al.* 2004, Stelson *et al.* 2008).

Alternatively, the sizing of the system's components and the control strategy can be viewed as a more integrated optimisation problem (Reyer 2000), ranging from the previously described sequential approach to an entirely simultaneous design process. Typically, a variant of the bi-level optimisation is employed: Karbaschian (2014) for example separates sizing and control aspects of a series transmission and initiates a new design iteration based on a final objective function evaluation. Li and Peng (2010) conduct a three-step process including feasibility check, control strategy and component optimisation for a hydraulic power-split transmission. An example of

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simultaneous optimisation is described in Kim (2008), where optimal control is applied to previously optimised designs in a second step. Although traditional engineering approaches undoubtedly lead to solid designs, optimisation can be used to explore, for example, design trade-offs (especially between component sizing and control parameters) and the impact of various design problem formulations in terms of variables, objectives and constraints.

This study addresses simultaneous component and control design for a hydraulic hybrid vehicle through simulation-based optimisation (Carson and Maria 1997, Krus 2003). Simultaneous component and control optimisation for (hydraulic) hybrid transmissions is computationally expensive and therefore often used only for systems with problems of reduced complexity, limiting the amount of detail and design parameters of the mathematical model used to evaluate a system. For the same reason, gradient-based optimisation algorithms are frequently used; they are however more likely than, for example, population-based methods to end in locally optimal solutions. This paper describes a general simulation-based optimisation framework, and implements as an example a series hydraulic hybrid vehicle (SHHV) transmission. It uses a dynamic simulation model, includes a rule-based control strategy and optimises nine design parameters primarily for fuel economy subject to additional constraints, using a non-gradient-based optimisation algorithm.

2. Optimisation framework

The general structure of the simulation-based optimisation framework is shown in Figure 1. At the core of the framework is the simulation model with its modelling assumptions and parameters (*independent system parameters*), described in Chapter 3. The model is embedded in an optimisation routine, which determines the *design parameter* values according to the optimisation algorithm used. These are to be tested through simulation for the specified application. Explicit design relations capture properties derived from these design parameters, such as effects of component scaling (*derived system parameters*). The results of the simulation (*system characteristics*) are evaluated with regard to the fulfilment of specified performance requirements and the objective function. Chapter 4 details how the optimisation aspects are implemented for the example application.

3. System modelling

At the centre of this study is a forward-facing simulation model of a series hydraulic hybrid transmission in Hopsan. Hopsan is a free simulation tool for multi-domain systems, developed at Linköping University (Eriksson *et al.* 2010).

Figure 1. Simulation-based design optimisation procedure (adapted from Krus 2003).

3.1. System architecture

Three basic configurations (topologies) are common for combining the different power sources in hybrids (Stelson *et al.* 2008). In a series hybrid as used in this study, the secondary technology is placed between primary source (engine) and power outtake (vehicle). Despite inherent power conversion losses, the decoupling of engine and vehicle operation offers a high potential for energy efficiency improvements by balancing the engine load and allowing engine shut-down if sufficient energy is stored.

The hydraulic hybrid vehicle is modelled as a generic, one-dimensional road vehicle, experiencing aerodynamic drag and rolling resistance, both assuming constant coefficients, and additional load for uphill driving. The vehicle is driven by a series hybrid architecture (Figure 2), containing a full hybrid hydraulic transmission with final drive between a diesel engine and the vehicle. The hydrostatic transmission consists of a variable hydraulic pump, driven by the diesel engine, a variable hydraulic pump/motor for power transfer to and from the vehicle, and a hydraulic accumulator for temporary energy storage, modelled as an open-circuit arrangement. The hydraulic circuit contains two pressure relief valves, and a check valve to prevent back flow from the accumulator to the pump, but also eliminating the possibility of starting the combustion engine from the energy storage (Kargul *et al.* 2015). The accumulator is permanently connected to the hydraulic circuit. A friction brake is implemented, which is only to be used when the braking capacity of the system is limited due to high system pressure.

For the system design optimisation, it is necessary to consider how to scale the major vehicle components. Variation of the component sizes affects both power properties and other attributes and thus the performance of the entire transmission. To ascertain the scaling relationships for the hydraulic components, published data from major manufacturers of

components suitable for the desired pressure range were analysed (see Baer 2018 for more details).

3.1.1. Hydraulic pump and pump/motor

Both hydraulic pump and pump/motor are axial piston machines, the pump being of in-line type, whereas the pump/motor is of bent-axis design. Volumetric and hydro-mechanical losses are captured through efficiency models. Rydberg's (1983) steady-state efficiency model has been parameterised to represent measurements made for an in-line pump and a bent-axis motor (Figure 3). For the pumping mode of the pump/motor the motor coefficients are reused, an assumption agreeing with trends in measurements made by Kargul *et al.* (2015). Dynamics for the displacement actuators are modelled with first-order low-pass filters.

The maximum speed–volumetric displacement relationship obtained for component scaling matches previous studies (e.g. Manring *et al.* 2014, see for example Figure 4 for the pump); overspeed operation under less-than-maximum displacement setting (Macor and Rossetti 2011) is analysed for one machine of each type due to varying manufacturer specifications. Efficiency model adjustments are not considered during scaling. The hydraulic pump/motor scaling is based on data for hydraulic motors as suitable mass-produced component data is unavailable, with an adjusted component mass to take more complex component design into account.

3.1.2. Hydraulic accumulator

In the hydraulic accumulator, the state of the compressed gas (nitrogen) is central in modelling the component's behaviour. To capture the behaviour more accurately than through a polytropic process, the Benedict–Webb–Rubin equation is implemented. Although heat transfer from the nitrogen to the oil is not modelled directly, the heat exchange with the accumulator's environment is characterised by the accumulator's thermal time constant.

Figure 2. Basic system layout (simplified).

Figure 3. Efficiency maps for in-line pump ((a) and (b)) and bent-axis motor ((c) and (d)) with displacement settings of 1.0, 0.67 and 0.33.

The selection of accumulators focused on standard bladder-type accumulators, based on their favourable power density over piston accumulators. An experimentally obtained accumulator time constant (Nyman and Rydberg 2001) is scaled to take into account variations in the accumulator surface area and the gas mass, the latter depending on the pre-charge conditions. A second accumulator is considered for the transmission mass as oil reservoir. Reduced weight using carbon-fibre and improved heat retention through fillings (most commonly foam, Otis and Pourmovahed 1984) are beyond the scope of this study.

For the fuel consumption evaluation, the simulation of the SHHV assumes a fully discharged accumulator at the beginning of a driving mission, but ends with residual charge after the final braking. To take this asymmetry into account, the accumulator's energy content E_{acc} for a given charge pressure can be determined approximately based on a polytropic process assumption as

$$E_{acc} = \frac{p_0 V_{0,acc}}{n-1} \left[\left(\frac{p_1}{p_2} \right)^{\frac{1-n}{n}} - 1 \right] \quad (1)$$

where n is the polytropic exponent, $V_{0,acc}$ the accumulator volume, p_0 the pre-charge pressure, and p_1 the minimum operating pressure (Bowns *et al.* 1981), with an assumed isotherm process between p_0 and p_1 . p_2 is for this estimation the accumulator pressure after full heat exchange between accumulator content and environment. Considering the wide pressure range necessary for hydraulic transmissions and variable ambient temperatures, the polytropic exponent is obtained from interpolation between Korkmaz (1975) tabular values. The accumulator energy content is then converted into an equivalent amount of diesel fuel based on its lower heating value.

3.1.3. Four-stroke diesel engine

For diesel engines, high-fidelity, but often computationally heavy, models based on the thermodynamic and combustion processes are available in the literature (e.g. Assanis and Heywood 1986, Wahlström and Eriksson 2011), including aspects such as dynamics for fuel injection, the modelling and dynamics of a possible turbo-charger, and temperature effects on the engine. Instead, the model is simplified here (similar to, e.g. Kumar *et al.*

Figure 4. Pump scaling relationships for component mass and maximum continuous speed as functions of component displacement.

2007) to a speed-controlled engine, with internal first-order dynamics to take combustion dynamics into account (Tsai and Goyal 1986). A steady-state map of the Brake Specific Fuel Consumption (BSFC, Digeser *et al.* 2005) is used to determine torque limitations (wide open throttle curve) and fuel consumption based on the current operating conditions. The inertia of the engine's moving parts and flywheel is modelled as lumped inertia on the pump component.

The engine model assumes the fuel consumption during start-up to be equivalent to 10 s of idle operation (Burgess *et al.* 2009) and acceleration time to idling speed to be 0.5 s (Kim 2008). The start-up behaviour is simplified to a start-up torque matching the flywheel inertia and desired acceleration.

The diesel engine's power is considered to be directly proportional to its maximum (rated) torque, as the speed range for a line of combustion engines is typically fairly similar, and assumed to be constant here. The engine's wide open throttle curve is scaled in accordance with the rated engine torque. The fuel consumption of the diesel engine is captured through a scaled BSFC map without considering component-size-dependent adjustment. Assanis *et al.* (1999) discuss the shortcomings of this method, though their simulation experiments illustrate limited impact of more accurate scaling on final results. For the engine mass, data for various diesel engines were analysed;

the inertia of the moving parts is scaled according to mass and geometry considerations.

3.2. Control strategy

3.2.1. Control approaches

Besides dimensioning the main components of the hydraulic hybrid transmission, their appropriate control is of utmost importance for realising a good fuel economy via energy recuperation or load balancing. In the supervisory control, control concepts address the power split between prime mover, secondary storage, and vehicle, explicitly or implicitly defining modes of operation for the hybrid. The lower level control deals with the operation of individual transmission components.

Control strategies can for example be classified according to their optimality and their online and real-time applicability (Karbaschian and Söffker 2014). For globally optimal power management, perfect information about the driving sequence is required. The optimisation of all control variables results in high computational load. Consequently, optimal control cannot be implemented online, and serves best as a benchmark or basis for other approaches.

Rule-based control strategies offer a simpler implementation at the expense of sub-optimality. For hydraulic hybrid drives, these typically entail

a formulation of different operation modes and operating points based on the current power demand (e.g. Buchwald *et al.* 1979, Wu *et al.* 2002). More advanced power management approaches optimise the control for a limited prediction horizon (e.g. Deppen *et al.* 2010), or use an algorithm for power management that imitates previously derived optimal control (Sprengel and Ivantysynova 2016).

3.2.2. Implemented control concept

For the present study, a simple yet robust power management strategy is required for the simultaneous component and control optimisation. As general, supervisory control, the so-called thermostatic state-of-charge control is implemented, where predefined state-of-charge levels in the accumulator trigger specific engine operation modes (Wu *et al.* 1985), here with a pressure-based definition of the state-of-charge.

Following a similar control strategy as suggested by Kim (2008), the system's state-of-charge triggers engine operation at either a target speed (corresponding here to the engine's sweet spot) between a lower and upper limit, SoC_{low} and SoC_{high} , or at a speed corresponding to the maximum power below SoC_{low} . If the accumulator is sufficiently charged, the engine is switched off or set to idle. The general control strategy is extended to allow smoother operation and more flexibility with potentially higher performance:

- (1) A dynamic definition of the state-of-charge limits considers the current acceleration demand and vehicle speed. This improves vehicle performance and controller robustness, while taking increased energy recuperation potential at high speeds into account. The parameter p_{split} characterises the pressure level between the corresponding state-of-charge pressures towards which the limits converge at high speed.
- (2) The initially discontinuous transition between different reference speed levels results in a so-called 'bang-bang' control characteristic. It is replaced through gradual transition in an interval around the state-of-charge limits to obtain smoother engine operation.
- (3) At higher vehicle speeds, engine on/off operation becomes less desirable due to the time delay in power supply introduced by a start-up process. Consequently, total engine shut-down is prevented above a predefined speed, and the engine operates at the idling point instead if the accumulator is sufficiently charged.

The pump displacement setting is modulated to ensure engine operation close to the fuel-consumption-minimal line when transitioning between the target operating points, and limits the load on the engine if necessary to prevent stalling of the engine, but otherwise operates at full displacement to allow high efficiency. The pump load can lead to undesirably slow engine acceleration, which can be reduced by lowered pump displacement setting when accelerating.

As the pump/motor displacement setting determines the power transfer from transmission to vehicle, its controller is equivalent to a simple driver model. Its basic structure consists of a PI-controller, taking the reference speed from the drive cycle as well as the current vehicle speed into account. To prevent system failure, the pump/motor displacement setting is reduced – despite any current driver request – with the net flow out of the transmission relative to a design-dependent constant q_{ref} .

4. Implementation and application of optimisation framework

4.1. Vehicular application

As example vehicle, a compact truck or light commercial vehicle is chosen (Table 1). This vehicle category allows for a gross vehicle weight of up to 3500 kg. For fuel consumption evaluations, the vehicle will be considered to be half-loaded.

The vehicle performance requirements in the following design optimisation experiments are largely determined by standard drive cycles, the Urban Dynamometer Driving Schedule (UDDS), the New European Driving Cycle (NEDC) and the Worldwide harmonised Light vehicles Test Procedure (WLTP3), see Table 2. These speed profiles are closer to actual driving conditions in that they typically do not require extreme acceleration and maximum vehicle speeds, but remain for example within common legal speed

Table 1. Basic vehicle data.

Kerb weight/Load considered	1800 kg/850 kg
Effective front area	1.3 m ²
Effective wheel radius (incl. final drive)	0.1 m
Rolling resistance coefficient	0.02 [–]
Default diesel engine	
Maximum torque	367 Nm
Maximum power	130 kW

Table 2. Drive cycle comparison (Barlow *et al.* 2009, Tutuianu *et al.* 2013).

Drive cycle	t_{cycle} (s)	x_{cycle} (km)	v_{max} (km/h)	$\dot{v}_{max}$ (m/s ²)
UDDS	1369	12.0	91.1	1.5
NEDC	1220	11.0	120.0	1.0
WLTP3 (v5)	1800	23.3	131.3	1.6

limits. The three cycles utilised here have different countries of origin, but all include both inner-city and suburban driving. Both NEDC and WLTP3 also reach European highway speeds. Although the drive cycles also differ with regard to testing procedure, for example in terms of cold or hot start of the engine, these aspects are not addressed. Limits on the trace error are also specified but could be systematically exploited for favourable fuel consumption.

For that reason, as main performance measure the total absolute deviation between reference and actual vehicle speed, $v_{ref}(t)$ and $v_{veh}(t)$, relative to the total distance covered in the drive cycle, x_{cycle} , is defined as *Average Relative Velocity Deviation*, ARVD:

$$ARVD = \frac{\int_0^{t_{cycle}} |v_{ref}(t) - v_{veh}(t)| dt}{x_{cycle}} \quad (2)$$

As this measure will lead to drive-cycle-specific optimised designs, maximum speed and acceleration requirements will be reintroduced later to obtain more flexible designs.

4.2. Optimisation problem formulation

4.2.1. Design parameters

In a basic design optimisation, the main hydraulic components as well as the diesel engine plus its flywheel are considered on the hardware side. As the main defining control parameters, the stationary state-of-charge limits, defined through reference pressures p_{high} and p_{low} , are included. The reference flow limit, q_{ref} and the high-speed operating point, characterised through the target pressure level p_{split} , are also added as they depend on the general system design parameters. Wide design parameter limits (Table 3) are chosen to aim for convergence within the design space.

4.2.2. Optimization objectives

The optimisation focuses primarily on the vehicle's fuel consumption ($FC(\mathbf{x})$) over a given drive cycle. The total fuel consumption considers engine start-ups, engine operation, and for comparability also an approximated fuel equivalent to the accumulator state-of-charge after driving and standstill.

Secondary design objectives include the performance, ARVD, over the studied drive cycle, the fulfilment of external performance requirements where applicable, the avoidance of system failure, indicated through cavitation in the system, inadmissible design parameter constellations, or a violation of the hydraulic machines' speed limitations. These are formulated as constraints $C_l(\mathbf{x})$, resulting in penalties of varying weight c_p if violated. For the ARVD performance, a level of 1.0% is targeted.

The optimisation problem with m secondary objectives becomes then

$$\min f(\mathbf{x}) = FC(\mathbf{x}) + \sum_{l=1}^m C_l(\mathbf{x})$$

subject to

$$x_{i,min} \leq x_i \leq x_{i,max}, \quad i = 1, \dots, 9$$

with

$$C_l(\mathbf{x}) = \{0, 1\} \cdot c_{p,l}, \quad l = 1, \dots, m \quad (3)$$

where $\mathbf{x}$ is the vector containing all design parameters x_i with the limits $[x_{i,min} \ x_{i,max}]$.

4.3. Optimization algorithm

For the optimisation, the Complex-RF algorithm is used. This is a non-gradient-based method and an extension of Box' Complex algorithm. The Complex-RF contains random noise and a forgetting factor as extensions, aiming to prevent premature convergence on a possibly local optimum and favour results of recent iterations (Krus 2003). This method can provide a compromise between computational burden and a sufficient performance (Fellini 1998). By selecting a number of random starting parameter sets, the probability of exploring a major part of the design space is increased. The optimisation algorithm is integrated in the Hopsan simulation tool, which allows for optimisation experiments to be run in parallel in a Linux cluster configuration (Nordin *et al.* 2015). The algorithm parameters follow the values obtained through meta-optimisation by Krus and Ölvander (2013). An optimisation is run with a

Table 3. Design parameter limits.

Design Parameter	Unit	Lower Limit	Upper Limit
Pump displacement D_p	$10^{-6} \text{ m}^3/\text{rev}$	25	250
Pump/motor displacement D_{pm}	$10^{-6} \text{ m}^3/\text{rev}$	25	250
Flywheel inertia J_{FW}	kgm^2	0	1
Maximum engine torque $T_{ICE,max}$	Nm	75	400
Accumulator volume $V_{0,acc}$	10^{-3} m^3	10	100
Upper state-of-charge limit p_{high}	10^6 Pa	15	45
Lower state-of-charge limit p_{low}	10^6 Pa	12.5	44 ^a
Target pressure level p_{split}	10^6 Pa	p_{low}	p_{high}
Reference flow q_{ref}	m^3/s	0	0.05

^a Adjusted to max. ($p_{high} - 5 \cdot 10^6$) Pa.

maximum of 4000 iterations, and aborted if a convergence of 0.001 is reached for all design parameters.

4.4. Optimization experiments

Two sets of optimisation experiments are conducted for this study, each optimising the SHHV transmission over the three drive cycles presented above.

- *Case I*: Designs are obtained focusing on a single drive cycles for driving performance requirements and fuel consumption evaluation.
- *Case II*: The resulting designs from the previous case are expected to be inflexible with regard to changes in usage profile. For more robust performance, drive-cycle-independent requirements are introduced. The targeted limits (an acceleration to 100 km/h in 15 s or less and a maximum continuous speed of 130 km/h) aim for real-life driving. The performance is measured for optimal operating conditions, that is, considering only the vehicle's kerb weight and a fully charged accumulator.

5. Optimization results

To increase the probability of successful optimisation leading to a close-to-optimal solution given the optimisation-algorithm-inherent stochastic elements, each individual experiment is run multiple times. No exact duplicate results were obtained, indicating the complexity of the design problem. In this chapter, for each case and drive cycle the best result obtained is considered, summarised in Figure 5. The optimizations converged on average within 1930 function

evaluations, each taking approximately 30 s on a typical desktop computer.

Considering for example the optimised design obtained for the UDDS in *Case I*, a very small engine (here: 121 Nm) delivers a sufficient performance for this optimisation's objectives. The engine operates largely in its most fuel-efficient region during propulsion, and in transition between target speed levels at maximum torque (see Figure 6).

An energy perspective (Figure 7) during the first 400 s of the UDDS illustrates the system's operation further:

Figure 7(b) shows the energy balance diagram for the system including the total combined energy, both kinetic and potential (accumulator). In an ideal, lossless case without restriction on energy storage the total energy should be constant, and there will solely be exchange between kinetic and potential energy. The figure verifies that the optimised system behaves as expected within the existing limitations: low accumulator charge triggers energy input via diesel engine and hydraulic pump. Once the engine has been shut down, vehicular and transmission losses cause a reduction in total energy despite relatively constant driving speed. In the high-speed part of the drive cycle, the engine's operation becomes more dynamic due to speed – and acceleration-dependent state-of-charge limit adjustments, and the prevented engine shutdown. The accumulator in this optimised design is capable of capturing approximately 57% of the vehicle's maximum kinetic energy during this drive cycle, the rest being lost through vehicle and transmission losses during braking, the hydraulic system's pressure relief valve and the friction brake. In comparison to a hypothetical non-hybrid case, the hydraulic hybrid vehicle's mean total energy content

Figure 5. Distribution of design parameters within design range (relatively) for different drive cycles and cases. Minimum and maximum for each design parameter given as absolute values (p_{split} is per definition relative).

Figure 6. Bubble plot of engine operating points for *Case I*, UDDS-optimised design, together with wide open throttle curve (maximum torque), scaled BSFC-map and fuel-consumption-minimal operating points ($BSFC_{min}$).

is increased through the addition of the hydraulic accumulator as energy buffer, but a lower standard deviation indicates how hybridisation helps even out drive cycle peak demands (Table 4).

The same optimisation experiment for the other drive cycles leads to differences in the designs obtained (see Figure 5). For the NEDC, a smaller pump/motor and smaller accumulator are sufficient. A higher upper state-of-charge limit results from less recuperation potential and moderate accelerations but requires a bigger engine. For the higher power requirements of the WLTP3, pump and engine need to be even bigger. A bigger accumulator and lower upper state-of-charge compared to the NEDC design allow more energy recuperation. For all designs, p_{low} is close to the lower design parameter limit, indicating that the dynamic state-of-charge definition limit regulates increased power demand sufficiently.

Introducing additional performance requirements in *Case II* has more effect on the UDDS – and NEDC-based designs than for the WLTP3-based one (see Figure 5 and Table 5). For WLTP3, the design parameters are not much affected by the additional requirements, indicating that the specified acceleration and speed targets come close to the performance required in this drive cycle. The fuel economy is even slightly improved compared to *Case I*. This is due to the particular solutions the optimisation algorithm converged on in each case; apparently

the additional performance requirements helped guide the optimisation to this solution.

For the other drive cycles, larger components are required than in *Case I* to fulfil the performance requirements: pump and pump/motor are similarly sized now for all designs, the engine in the UDDS test case and the accumulator in the NEDC case are larger than before to meet the increased power requirements. The overall larger and heavier transmission with control strategies in part tuned to meet performance requirements yield comparably higher fuel consumption than before (Table 5).

A test of each design over the non-optimised drive cycles helps to distinguish the results with regard to robustness to different driving conditions (Table 6). A number of these 'cross-tests' lead to a violation of the pump speed constraint, typically when a design is faced with more demanding drive cycles than optimised for. This indicates that diesel engine and pump operating speed range need to be matched better. Otherwise, the expected improvement in driving performance over varying driving missions, expressed as ARVD, can be observed for designs obtained with additional driving performance requirements (*Case II*).

6. Discussion

The results from the simulation-based optimisation framework of several design experiments with various

Figure 7. Reference velocity, energy content in system and instantaneous rate of fuel consumption for *Case I*, UDDS-optimised design during first 400 s of UDDS.

target driving scenarios illustrate how sensitive the final design is to the optimisation problem formulation.

Brake energy recuperation aims to improve fuel economy. The results in Table 5 do not support that yet compared to typical vehicle data. This can be attributed to a number of factors:

- The optimisation is based on limited and available component and efficiency data, where alternative

and targeted choices might yield performance and efficiency gains. Static efficiency maps, as utilised here, are only partially representative. As in a series hybrid architecture all power is transmitted through the hydraulic transmission path, which is less efficient than its mechanical counterpart, highly efficient machines are needed (see e.g. Kargul *et al.* 2015), and their accurate modelling is of great importance.

Table 4. Comparison of system's energy content for non-hybrid (kinetic energy of vehicle and flywheel) and hybrid case (kinetic energy of vehicle and flywheel plus energy content accumulator) for *Case I*, UDDS-optimised design.

	Mean energy content	Standard deviation
Non-hybrid	$1.8 \cdot 10^5$ J	$2.2 \cdot 10^5$ J
Hybrid	$4.4 \cdot 10^5$ J	$1.7 \cdot 10^5$ J

Table 5. Optimised vehicle designs' fuel consumption (g).

Drive cycle	UDDS	NEDC	WLTP3
Case I	814	844	2072
Case II	888	874	2066

Table 6. Comparison of ARVD for different designs and drive cycles.

Case	Drive cycle	Test cycle		
		UDDS	NEDC	WLTP3
I	UDDS	(1.0 %)	3.9 % ^a	8.4 % ^a
	NEDC	2.3 %	(1.0 %)	1.7 % ^a
	WLTP3	1.9 %	0.6 %	(1.0 %)
II	UDDS	(1.0 %)	0.6 % ^a	1.5 % ^a
	NEDC	0.7 %	(0.5 %)	0.9 % ^a
	WLTP3	0.8 %	0.4 %	(0.8 %)

^a Pump speed constraint violated

- The system architecture in this example does not include design options such as additional gearing and valves. Similarly, the hydraulic pump and pump/motor's operating efficiency can be improved by considering multiple smaller components in sequential control (see e.g. Kim 2008).
- A rule-based control strategy does not yield globally optimal control. Although the basic control strategy has been extended, a number of its refining parameters are set, but not optimised simultaneously.

However, the energy balance diagram (Figure 7 (b)) indicates that the optimised system utilises the energy storage as expected.

The design experiments presented here focused on variations in the optimisation problem by studying various target drive cycles and by including acceleration and maximum speed performance requirements suitable for typical drive cycles. Post-optimisation gradeability simulations for the designs obtained showed that moderate slopes could be driven by a fully-loaded vehicle at constant medium-high speeds, while steeper climbs were possible at lower speeds. The framework can be extended to include additional, more exceptional driving conditions.

Further research can examine other design problem variations for the given system architecture, for example in terms of design objectives (costs and business case (Kargul *et al.* 2015) and emissions (Tikkanen *et al.* 2017)) and additional parameters. It is possible to include and compare different architecture-enhancing features.

7. Conclusions

This paper introduced a simulation-based optimisation framework applied to a series hydraulic hybrid vehicular transmission. The simulation-based approach allows to evaluate the transmission performance under instantaneous power demand and reserve conditions, and the corresponding system dynamics. Through numerical optimisation, a large design space can be explored, including trade-offs between components and control parameters. The optimisation problem formulation including the driving mission, the optimisation objectives and boundary conditions, as well as the application under consideration and modelling assumptions thereby have a large impact on the results. The energy balance diagram was used as an indicator that the optimised system operates as could be expected.

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Notes on contributors

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