



## COMPONENT SIZING STUDY FOR A LIGHT-DUTY SERIES HYDRAULIC HYBRID VEHICLE IN URBAN DRIVE CYCLES

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**Abstract.** *With the focus on energy efficiency for many different kinds of vehicle applications, hybridization is considered a possible solution to reduce fuel consumption. While hybrid electric concepts are already available for passenger vehicles, and also considered for heavier applications, hybrid hydraulic alternatives have been mainly limited to the latter, which benefit most from the higher power density available. To study the different hybrid architectures and applications, a modeling framework for the system design is developed using the simulation tool Hopsan from Linköping University. Previously, the model of a series hydraulic hybrid vehicle was introduced, a light-duty vehicle simulated over two standard urban drive cycles, and its potential for further work established. In this paper, the model is extended by including a simple combustion engine power management to provide for more realistic propulsion of the hydraulic drivetrain, showing the potential to operate a series hydraulic hybrid vehicle's engine in more efficient regions. Additionally, the design is studied concerning the effects of a variation of key component sizes on the accuracy and energy efficiency objectives. Instead of subjecting the system to (multi-objective) optimization, at this stage the individual component's influence is studied, and the objectives are dealt with separately from each other to eliminate the need for compromise between them, both to gain a better understanding of the interdependencies.*

**Keywords:** *Series hydraulic hybrid, Hopsan modeling*

### 1. INTRODUCTION

Hybridization of a vehicle's drivetrain aims to improve its fuel efficiency by integrating a second energy form. Thus it becomes possible to recuperate braking energy, to lower power demand peaks to the engine and consequently to operate the combustion engine more efficiently, and – if considered during the design phase – it can allow for down-sizing of the combustion engine as well. While less common as of now, using hydraulics as a secondary power source can be advantageous over electrics especially due to a higher power density allowing for fast charging and discharging, though at the cost of a lower energy density leading to a shorter range.

Originally, hydraulic hybrid concepts were therefore considered especially suitable for heavy vehicles whose typical drive cycles include frequent stops, such as urban busses and refuse trucks (Yan, *et al.*, 2010; Baseley, *et al.*, 2007), benefitting from the high masses to accelerate and decelerate in short times. Recent research and development, though, has been extended to passenger and light-weight vehicles as well (e.g. Stelson, *et al.*, 2008; Kim and Filipi, 2007), even leading towards commercially available passenger vehicles with hydraulics announced for the foreseeable future (PSA Peugeot Citroën, 2013).

This paper is part of the development process of a modeling framework for hybrid hydraulic vehicles with the overall goal of designing and comparing different hybrid hydraulic architectures for given vehicles and duty profiles. The model for a series hydraulic hybrid vehicle previously presented contained a simple constant speed power source supplying the charge pump which can be switched on and off; with the pump operating at either zero or full displacement. Modifications to this power source allow now for different strategies of operation, which will here be limited to the assumption of running the combustion engine in its most efficient operating point. An initial design for a light-weight application is subjected to component size variations to study how those affect key resulting properties. These performance criteria concern whether the vehicle can follow its duty profiles, and whether and to what extent the inclusion of a secondary energy storage shows a potential for energy recuperation as desired. As both are not necessarily correlated, a full design optimization would require a weighing of the goals to express preferences, and require more additional design criteria to be considered. Instead, the objectives are treated separately to study the effect of different component sizes.

### 2. BACKGROUND

Depending on the placement of the hydraulics in the drivetrain, there are three different basic architectures for hybrid hydraulic vehicles (Fig. 1). Keeping a mechanical connection between the internal combustion engine and the

vehicle as in the parallel concept, including some form of gearbox, allows for a more efficient power transfer compared to a series hybrid. This same mechanical connection between the engine and the wheel speed. The split and connect

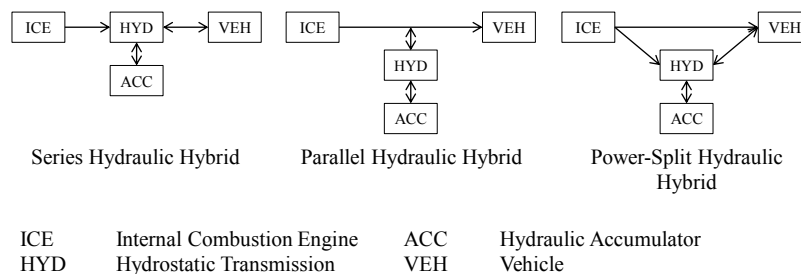


Figure 1. Hybrid hydraulic drivetrain architectures (schematic, compare similarly (Stecki and Matheson, 2005)).

Of the three main hydraulic hybrid architectures, the series hydraulic hybrid concept was chosen as the first to be modeled as its hydrostatic transmission and combustion engine can be treated fairly separately from each other, the focus here still being the hydraulics of the drivetrain. This independence also allows for more energy-efficient engine management.

In current literature, modeling and optimization of series hydraulic hybrid concepts for passenger vehicles (e.g. Deppen, *et al.*, 2012; Johri and Filipi, 2010) and light- and medium-sized trucks (Kim and Filipi, 2007; Sun, 2010) are frequently addressed. While some focus is upon the optimization of the drivetrain's components (Sun, 2010), much research revolves around power resp. energy management strategies and their optimization (e.g. (Wu, *et al.*, 2004) for a rule-based approach; (Kim and Filipi, 2007) with a thermostatic state-of-charge control; (Johri and Filipi, 2010) utilizing Stochastic Dynamic Programming; (Deppen, *et al.*, 2012) using Model Predictive Control).

### 3. FRAMEWORK FOR A SERIES HYDRAULIC HYBRID VEHICLE MODEL IN HOPSAN

An earlier version of the model in this paper was presented previously (Baer, *et al.*, 2013) and set up and tested for a light-weight vehicle over to urban drive cycles to confirm its ability to follow the prescribed velocity profile accurately and its theoretical potential for reduced energy input and improved energy efficiency.

#### 3.1 Hopsan

The simulation model was created in a development release of Hopsan (Eriksson, *et al.*, 2010; Axin, *et al.*, 2010), a multi-domain simulation tool by the Division of Fluid and Mechatronic Systems at Linköping University, which is available for download free of charge. It utilizes transmission line modeling technique (Auslander, 1968) which bases pressure and flow calculations on wave characteristics (Krus, *et al.*, 1990). While the original Hopsan dates back to the 1970s, it has been launched in its current version in 2010 and is being continuously developed further.

#### 3.2 Previous model

The main components the model consists of (Fig. 2) are a one dimensionally-modeled vehicle, a variable pump/motor and a variable charge pump, as well as a gas accumulator. The vehicle is described through its mass  $M_{veh}$ , which takes the mass of the main hydraulic components into account, its effective frontal area  $c_d A$ , and its effective wheel radius  $r_{wheel,eff}$ , which includes the actual wheel radius  $r_{wheel}$  and the differential gear ratio  $i_{diff}$ . An additional gearbox between hydrostatic transmission and vehicle could allow for smaller components and protect the pump/motor against too high shaft speeds. While included in some models for series hydraulic hybrid vehicles (e.g. (Johri and Filipi, 2010)), it is not considered as of now. The mass of the main hydraulic components is derived from the data of commercially available products and extrapolated based on the installed displacement. Both hydraulic machines include efficiencies models (Rydberg, 1983) taking into account their displacement setting angle, speed and pressure difference. Furthermore, the modeled components allow theoretically for switching the high and low pressure side; this feature, however, is not utilized currently. The accumulator is modeled as being loss-free, assuming an adiabatic process with the polytropic exponent adjusted to the system's pressure level (Rydberg, 1984), and directly connected to the hydrostatic transmission. A combustion engine is not specifically modeled, for the charge pump a constant angular velocity is assumed instead.

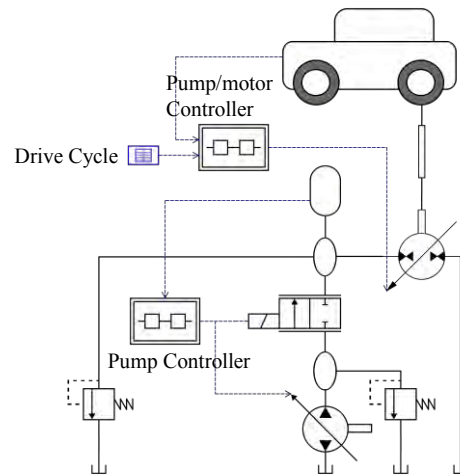


Figure 2. Basic structure of the series hydraulic hybrid model in Hopsan.

The displacement setting angle of the pump/motor is set through a PI-controlled velocity feedback, comparing the current vehicle speed to the reference velocity of the drive cycle. The propulsion of the system depends on the state-of-charge of the hydraulic accumulator, which is calculated based on the current gas volume (Wu, *et al.*, 2004): if the current state-of-charge is lower than a lower boundary, the pump's displacement setting is fully opened and charges the system, until an upper boundary is reached, causing the displacement setting angle to be closed. Thus, it is modeled to switch the propulsion on and off. However, it does not allow for a more controlled pump flow beyond maximum or zero.

Two resulting properties were specified, depicting the accuracy in following a prescribed drive cycle and the energy recuperation potential, respectively. The accumulated relative velocity error (ARVE, Eq. (1)) integrates the vehicle velocity  $v_{veh}$ 's deviation from the reference velocity  $v_{ref}$ , and puts it into relation to the total distance covered by the drive cycle,  $x_{max}$ , a low value corresponding to little deviation from the drive cycle. The adjusted energy input ratio, ( $EIR_a$ , Equ. (2)), is calculated as the energy input into the system over the drive cycle,  $E_{in,system}$ , reduced by a state-of-charge difference of the accumulator between start and end of the cycle,  $\Delta E_{acc}$ , relative to the system's energy demand, i.e., the vehicle's energy demand  $E_{in,veh}$  plus losses in the hydraulic components,  $E_{loss}$ . The lower this ratio becomes, the more of the system's energy demand can be met by energy recuperated in the accumulator instead of energy provided by the system's propulsion.

$$ARVE = \int_t |v_{ref} - v_{veh}| dt / x_{max} \quad (1)$$

$$EIR_a = (E_{in,system} - \Delta E_{acc}) / (E_{in,veh} + E_{loss}) \quad (2)$$

As test application, a light-weight vehicle (see Tab. 1 for the vehicle parameters) was simulated over two urban standard cycles, the EPA Urban Dynamometer Driving Schedule FTP-72 (UDDS FTP-72) and the New York City Cycle (NYCC) (United States Environmental Protection Agency, 2013). These velocity profiles differ in length, maximum velocity, maximum acceleration, and general driving profile (see Tab. 2).

Table 1. Light-duty vehicle parameters.

Vehicle Property	Individual Value	Combined Value
Vehicle mass $M_{veh}$ (half-loaded, without additional components)	2700 kg	-
Frontal area $A$	3.75 m <sup>2</sup>	1.78 m <sup>2</sup>
Aerodynamic drag coefficient $c_d$	0.475	
Wheel radius $r_{wheel}$	0.33 m	0.1 m
Differential gear ratio $i_{diff}$	3.3 [-]	

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Table 2. Drive cycle characteristics.

Drive Cycle Property	UDDS FTP-72	NYCC
Length	1369 s	598 s
Maximum velocity	25.4 m/s = 91.3 km/h	12.38 m/s = 44.6 km/h
Average velocity*	10.8 m/s = 38.9 km/h	4.9 m/s = 17.6 km/h
Maximum acceleration*	1.48 m/s <sup>2</sup>	2.68 m/s <sup>2</sup>
Total distance covered*	12.0 km	1.9 km

\* approximated

For the modeled vehicle and drive cycles, promising values could be achieved for both accuracy and energy recuperation potential. Note, however, that the absolute values are still subject to change the more detailed and refined the model becomes, and rather indicate a tendency. Furthermore, it is to be pointed out that the intention is not to compare the two drive cycles to each other, but rather to expose the vehicle configuration to different driving requirements.

#### 4. INFLUENCE OF DIFFERENT PROPULSION STRATEGIES ON THE ENGINE'S OPERATING POINTS

The internal combustion engine in the previous design was reduced to a constant speed source providing the desired torque for the hydraulic pump at either full or no displacement, effectively cancels out the pump's variability of displacement. Because of the varying pump pressure and consequently load torque, the engine visits a wide range of operating points, especially operating in little efficient points when under a low load (see Fig. 3(a)). Torque variations stem from different pump pressures, operating points of a lower speed than targeted and torque outliers are transient. A wider span between the state-of-charge parameters which are responsible for the pump's actuation yields even further spread operating points due to a larger difference between lowest and highest pump pressure, see Fig. 3(b) with a lowered  $SoC_{low}$  compared to the default parameters.

As a series hybrid allows more than the other concepts for a highly efficient combustion engine operation independent from the current wheels' speed, this current propulsion strategy is not desirable. To operate the combustion engine more efficiently, the most intuitive alternative is to run it for as much as possible in its most efficient point, the so-called "sweet spot", with a defined speed and torque. By including a simple diesel engine model, it is possible to run the engine in a number of different ways, such as at a constant speed as before, at a constant power or varying the power supply according to the system's demand. Implementing a constant power source with the speed derived from the current load situation still yields different operations. If however next to the power also speed and torque are kept (close-to-)constant as well by adjusting the pump's displacement setting according to the current pressure, it is possible to reduce the range of visited points (Fig. 3(c)). Torque variations stem mostly from the start-up phase when the system pressure is lower than in the normal operating range due to the assumption of a discharged accumulator at the beginning of each drive cycle.

Through appropriate system control and dimensioning of the internal combustion engine, more fuel-efficient operation can be achieved once fuel consumption in the individual operating points is integrated in the component model and engine and hydrostatic transmission are matched. It has to be pointed out, though, that while for the individual combustion engine component this power management strategy aiming at the "sweet spot" yields a high efficiency, the system's most energy efficient operation does not necessarily have to concur with this, as other components' lower efficiencies could potentially reverse the engine's positive effect (Kim and Filipi, 2007; Johri and Filipi, 2010; Deppen, *et al.*, 2012).

The modeling of the internal combustion engine, albeit simplified as of now, also allows for more advanced input power management strategies to provide a power level modulated according to the system's state and demand (e.g. Kim, 2008), as well as modeling other hybrid architectures. Next to their implementation, once the engine is refined with fuel consumption data, these different operation strategies can also be studied e.g. concerning their energy, as well as the possibility of engine down-sizing compared to non-hybrid solutions.

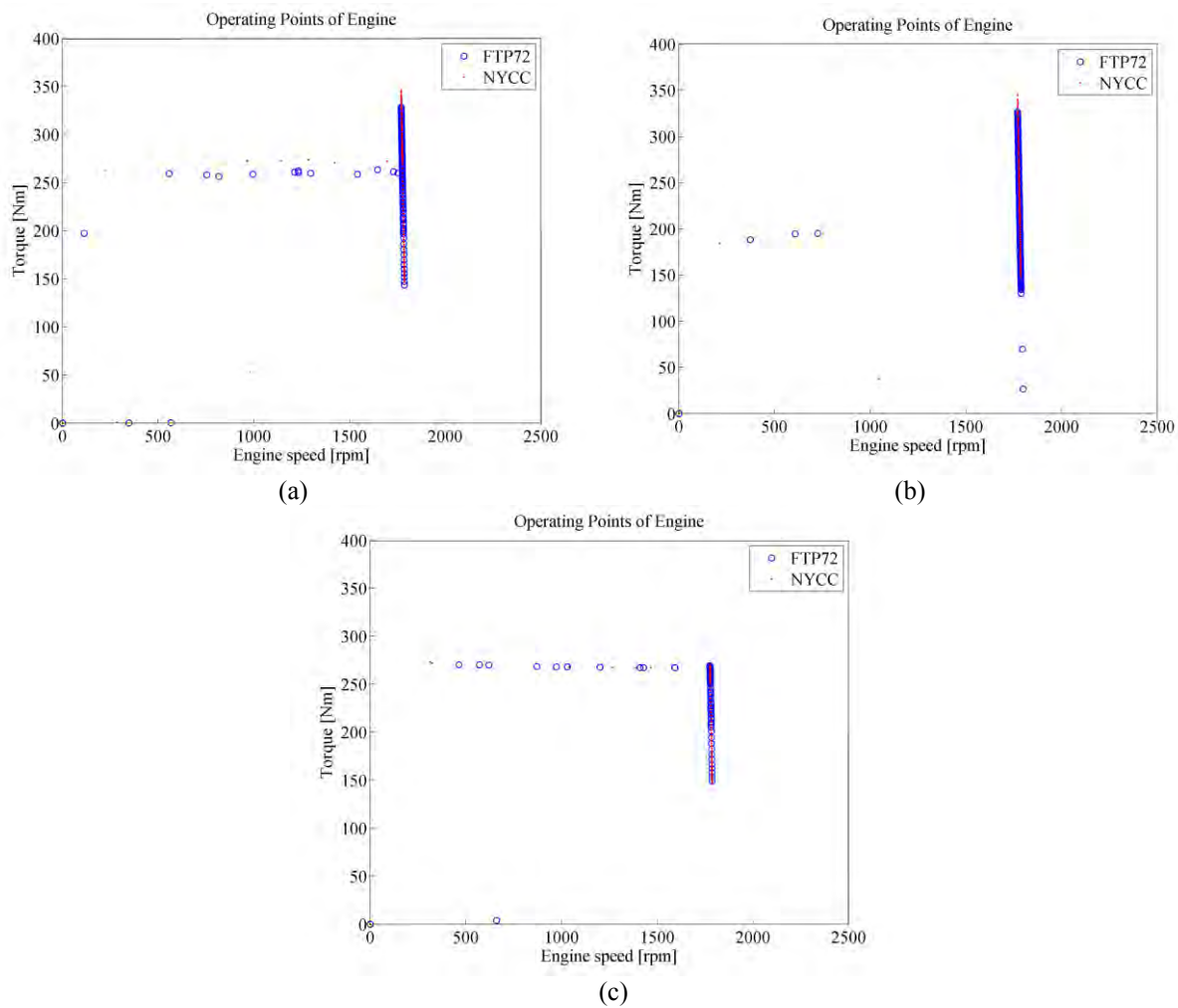


Figure 3. Operating points of combustion engine with constant target speed and default controller parameters (a) and lowered  $SoC_{low}$  (b), respectively, and constant target speed with adjusted displacement setting angle (c).

## 5. COMPONENT SIZE VARIATIONS

The initial components and some of the system's parameters are given in Tab. 3. While the previously presented results were promising, and the additional weight through the hydraulic components was taken into account, the system configuration is potentially oversized for a comparably small vehicle. Especially the accumulator shows potential for down-sizing without compromising on either accuracy or energy efficiency.

Table 3. Series hydraulic hybrid light-duty vehicle component sizes.

Component Size/Parameter	Value
Hydraulic pump displacement $D_p$	$100 \cdot 10^{-6} \text{ m}^3/\text{rev}$
Hydraulic pump/motor displacement $D_{pm}$	$150 \cdot 10^{-6} \text{ m}^3/\text{rev}$
Hydraulic accumulator volume $V_{0acc}$	$0.1 \text{ m}^3$
Constant pump (engine) speed	1800 rev/min
Maximum system pressure	33 MPa
Upper boundary state-of-charge $SoC_{high}$	0.71
Equivalent pressure for $SoC_{high}$	20 MPa
Lower boundary state-of-charge for $SoC_{low}$	0.55
Equivalent pressure for $SoC_{low}$	16 MPa

In this chapter, the effects of variation of one or more of the key component's sizes will be studied. The goal is neither yet to achieve an optimal configuration, nor to find a design satisfying multiple objectives. Instead, the properties introduced in chapter 3.2 will be considered separately, thus eliminating the problem of weighing the objectives against each other.

The following sections deal with the variation of the component sizes individually (section 5.1) while keeping the default values from Tab. 3, and the simultaneous variation of accumulator size and hydraulic machines' displacements (section 5.2). The effect of smaller components on system parameters other than the vehicle's mass including the hydrostatic transmission, e.g. higher allowed maximum pressure or pump shaft speed, has not yet been considered.

## 5.1 Single parameter variation

Figure 4 contains the results for varying only one of the main hydraulic components sizes, while keeping the sizes of the other components as well as the system and controller parameters constant. Only the total vehicle mass varies depending on the component's weight.

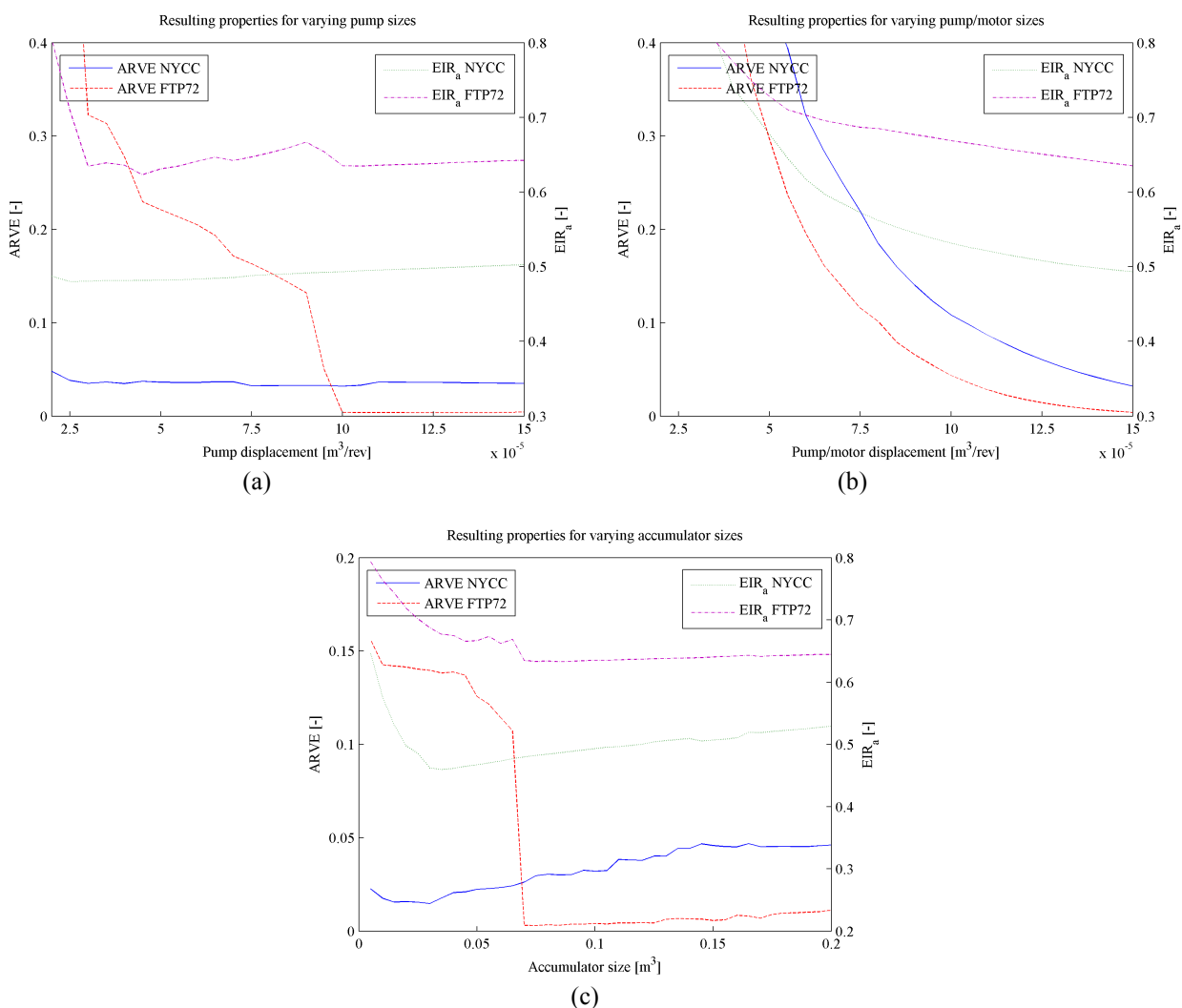


Figure 4. Individual component size variation: accumulator volume (a), pump displacement (b), and pump/motor displacement (c) for both drive cycles and resulting properties.

The results in Fig. 4(a) show that an accumulator size reduction can be achieved as expected, even when the hydraulic machine sizes are not adapted. The installation of more pump displacement (Fig. 4(b)) and hence higher hydraulic charge power leads to increased losses in the system and a higher total vehicle weight, thus resulting in a decreased energy recuperation potential. The absence of a volumetric limitation for the components used would in an energy efficiency-oriented optimization lead to a maximally sized pump/motor, due to the adjusted energy input ratio of the UDDS FTP-72 not converging as the other resulting properties do already for smaller machines (Fig. 4(c)). Lastly, the utilization of more than one drive cycle proves to be beneficial, as the UDDS FTP-72 cycle's accuracy is less robust

towards decreased component sizes due to higher velocities over longer periods, while the NYCC's profile is characterized by high accelerations but short distance and low maximum velocity, leading to its energy efficiency being affected more easily due to a comparably low total energy.

## 5.2 Multiple parameter variation

The previous section does not touch upon interdependencies between the components which have to be expected. Even without a concrete limit or penalty on the components' combined volume in place, the goal of the component sizing has to be a light, compact and hence less expensive drivetrain without compromising too much on the aforementioned objectives of accuracy and energy efficiency. Two systems are compared in the following considerations, one of which contains the original accumulator of  $0.1 \text{ m}^3$ , while for the second the accumulator gas volume has been reduced even below what has been suggested in Fig. 4. For each system, the displacements of the hydraulic machines are varied simultaneously, and the resulting properties evaluated for each system design.

As stated before, the vehicle's ability to follow the Urban Dynamometer Driving Schedule FTP-72 is more affected by accumulator size variation. Figure 5 shows that with a lower energy storage capacity, the system's accuracy decreases less than before when using a smaller pump/motor, while an under-sized pump leads to considerably higher accumulated velocity deviations. From the plots it can also be seen that while the original pump and pump/motor sizes' result is worse for a smaller gas accumulator, it is however possible to find a design with the same pump and a decreased pump/motor displacement with satisfying accuracy. It appears that part of the pump/motor's size depended on the machine to be matched to the discharge flow of the accumulator, which is lower with a smaller oil volume stored.

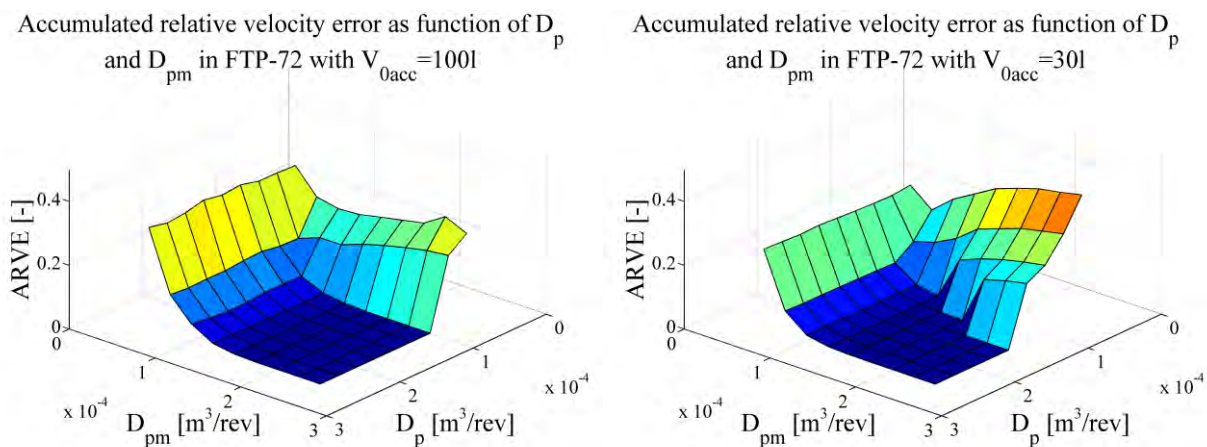


Figure 5. Hydraulic machine size variation for two different accumulator sizes. Accumulated relative velocity error in Urban Dynamometer Driving Schedule FTP-72.

Both the adjusted energy input ratio of the Urban Dynamometer Driving Schedule FTP-72 and the accumulated relative velocity error over the New York City Cycle are robust towards the accumulator size variation. The energy recuperation potential when driving the latter cycle with a smaller accumulator (Fig. 6) is slightly improved, especially for small pump and pump/motor, and can be explained through the lower total vehicle mass, the effect of which is more pronounced with this velocity profile in comparison to the FTP-72 cycle.



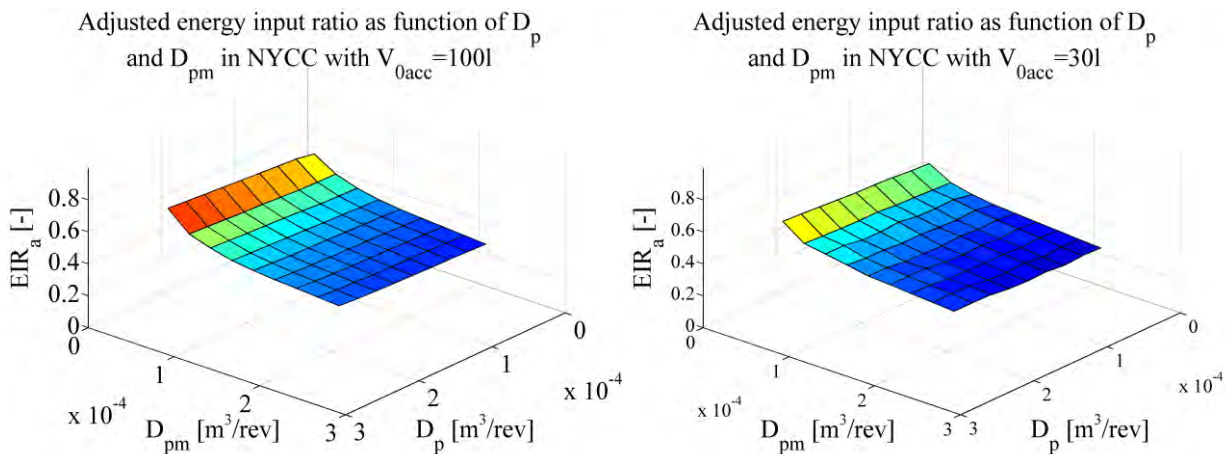


Figure 6. Hydraulic machine size variation for two different accumulator sizes. Adjusted energy input ratio in New York City Cycle.

## 6. CONCLUSIONS

In this paper, the existing series hydraulic hybrid model was extended to include a simple combustion engine component, and it was used to show the possibility to focus the operating points on a smaller range with the potential of more fuel-efficient driving. For the future, this allows in combination with fuel consumption data for more complex and more energy-efficient control strategies. The light-weight vehicle design was studied concerning the effect of its main hydraulic components' sizing, particularly in terms of down-sizing without compromising on the accuracy, i.e. drivability, and energy efficiency over urban drive cycles. It was shown that especially the original accumulator size could be greatly reduced (without claiming optimality), and the consequences for the pump and pump/motor's displacement choices were pointed out.

Besides the hydraulic component sizes (and their masses), further component and system specific parameters will have to be studied and considered for the design, such as the system's maximum pressure, the operating speeds of pump and pump/motor, the pump/engine control parameters  $SoC_{high}$  and  $SoC_{low}$ , the pump/motor controller gains and others, including their interdependencies amongst each other and with the hydraulic components. Increased operating speed of the hydraulic components would allow for smaller installed displacements and vice versa; with smaller machines and a smaller accumulator volume the system's maximum pressure could be increased, etc.

Besides the core resulting properties mentioned above, others objectives and requirements play a role in the system design as well. Restrictions such as the volume of the additional components installed and additional costs, but also minimum requirements not covered by the current drive cycles such as gradeability and driving at higher or maximum velocities need to be considered; while highway-driving might have less potential for energy recuperation due to less frequent acceleration and braking, the vehicle's drivetrain still needs to be designed for this task as well.

Lastly, model components such as the accumulator and the combustion engine contain simplifications that need to be at least partly resolved to improve the model to allow for a full system analysis and design.

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