TECHNISCHE UNIVERSITÄT WIEN

DIPLOMARBEIT

„Analysis and Optimization of a Hydraulic Hybrid Vehicle with numerical simulation methods“

ausgeführt zum Zwecke der Erlangung des akademischen Grades eines

„Master of Science“ unter der Leitung von

Univ. Prof. Dr. techn. Bernhard Geringer

und

Assoc. Prof. Dr. Peter Hofmann

Institut für Fahrzeugantriebe und Automobiltechnik

eingereicht an de Technischen Universität Wien

Fakultät für Maschinenwesen und Betriebswissenschaften

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Wien, am 12. Mai 2016 B16015

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Diplomaufgabe

Herrn Jesus Conde, Matr. Nr.: 1328233 wird folgende Diplomaufgabe gestellt:

"Analyse und Optimierung eines Hydraulik-Hybrid-Fahrzeuges mithilfe numerischer Simulationsmethoden"

Folgende Arbeiten sind durchzuführen:

Es soll ein Simulationsmodell eines Hydraulik-Hybrid-Fahrzeuges erstellt und hinsichtlich Kraftstoffverbrauch analysiert und bewertet werden. Fokus der Untersuchung soll die Modellierung des Hydraulik-Flüssigkeitskreislaufes inklusive der hydropneumatischen Speicher sein. Für die realitätsnahe Umsetzung sollen möglichst Daten von realen Komponenten aus dem automotive Bereich und spezifizierte Fahrzyklen verwendet werden.

Im Einzelnen sind folgende Punkte zu bearbeiten:

- Literaturrecherche bezüglich Hydraulik-Hybrid-Fahrzeugen
- Komponentenauswahl für den Hydraulik-Hybrid-Antriebsstrang
- Erstellung eines Längsdynamikmodells in GT-SUITE
- Analyse unterschiedlicher Betriebsstrategien und Komponentendimensionierungen
- Analyse und Bewertung des Kraftstoffverbrauchs in spezifizierten Fahrzyklen
- Vergleich des Modells mit einem Parallel-Elektro-Hybrid Konzept

Ein gebundenes Exemplar und eine Version der Diplomarbeit auf Datenträger sind am Institut für Fahrzeugantriebe und Automobiltechnik der Technischen Universität Wien abzugeben.

Dauer: März 2015 - Oktober 2015

Kennzahl: E 700

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Eidesstattliche Erklärung

Ich habe zur Kenntnis genommen, dass ich zur Drucklegung meiner Arbeit unter der Bezeichnung

„Analyse und Optimierung eines Hydraulik-Hybridfahrzeuges mithilfe numerischer Simulationsmethoden“

nur mit Bewilligung der Prüfungskommission berechtigt bin. Ich erkläre weiters an Eides statt, dass ich meine Diplomarbeit nach den anerkannten Grundsätzen für wissenschaftliche Arbeiten selbständig ausgeführt habe und alle verwendeten Hilfsmittel,

insbesondere die zugrunde gelegte Literatur genannt habe.
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Wien, am 3. September 2016 ....

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Abstract

Hydraulic hybrid vehicles (HHV) use a hydraulic circuit with one or more hydraulic machines and a gas to store the braking energy and us it as an alternative power source to the internal combustion engine (ICE).

HHVs are characterized by a lower cost of the components than the ones from the hybrid electric vehicles, as well as by the high power density of the hydro-pneumatic accumulator, which makes it capable of recuperating a high percentage of the braking energy.

Until now, all the developed HHVs have had a series or parallel configuration. The power-split HHV modelled in this project tries to combine the advantages from this type of this hybrid arrangement with the suitable use of the HHVs for urban driving.

The main advantage of the power-split hybrids is the ability of setting the ICE into its most efficient point, due to the CVT function of the two hydraulic machines, called Pump and Pump/Motor, and the engine connected to a planetary gear set.

The size of hydraulic machines is optimized in order to save as much fuel as possible in the NEDC, which is the standard European test to assess the fuel consumption, and, at the same time, being able to drive correctly the US06 cycle, which is distinguished by high accelerations and speeds.

Although being the initial objective to select hydraulic machines with a similar nominal power to the electric machines of the Toyota Prius 1st generation, the restricting parameter of these machines is the maximal displacement, so that the implemented ones count with a higher power limit, even if those values are never reached.

Other important empirically optimized parameters are involve the control of the state-of-charge of the accumulator, the use of regenerative or friction braking or the gear rates to keep the speeds of the machines within the wanted range.

In some situations, when power-split hybrids drive at high speeds, hydraulic machines work the inverse way they should (pumping instead of motoring, and vice versa),

dismissing the efficiency of the overall power train and making the fuel consumption higher than the one of a conventional vehicle.

Although, it does not solve the problem, by limiting the speeds of the hydraulic machines this behaviour can be enhanced.

With these conditions, the HHV is able to cut the fuel consumption of the conventional vehicle by more than a third in the NEDC. In the US06 cycle the fuel consumption is also lower, but the difference is minimal due to the little number of braking events.

These results show that the HHV has its best usage in urban driving and in vehicles, whose service involves frequent braking events.

List of abbreviations

BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
e-CVT	Electronically controlled Continuously Variable Transmission
NEDC	New European Driving Cycle
HEV	Hybrid Electric Vehicle
HHV	Hydraulic Hybrid Vehicle
HLA	Hydraulic Launch Assist
HP	High Pressure
HRB	Hydrostatic Regenerative Braking
ICE	Internal Combustion Engine
LP	Low Pressure
OEM	Original Equipment Manufacturer
PGS	Planetary Gear Set
PMot	Pump/Motor
RPM	Revolutions Per Minute
SOC	State Of Charge
SFTP	Supplemental Federal Test Procedure
THS	Toyota Hybrid System

Table of Contents

1. Introduction

In the last decades, concern about the environmental impact of the transport sector has increased, since it is responsible for around 20% of the total EU emissions of carbon dioxide (CO_2) and the major cause for urban air pollution [1].

Moreover, the dependence on oil as the sole source of energy for passenger vehicles has economic and political implications. The oil reserve of the world diminishes, the total number of vehicles keeps on rising (especially in countries like China and India) and many oil-exporting countries are located in geo-political unstable parts of the world.

In the European Union the fleet average of every OEM must achieve an emission level of 95 gCO_2/km by 2021, which is the equivalent of a fuel consumption of 4,1 L/100km for petrol engines and 3,5 L/100km for Diesel engines [2].

Achieving this target requires a combination of improved fuel efficiency, new types of vehicles, alternative fuels (e.g. bio-fuels, hydrogen, etc.) and a higher efficiency of vehicle’s transmission or rolling and air resistance.

For these reasons, many governments and governmental organizations promote the adoption of hybrid and electric vehicles as an important part of the portfolio of technologies required to reduce greenhouse gas emissions [3].

It is a fact that the electric vehicle will be the prevalent vehicle type in the future, either using batteries or fuel cells. However, there is still a long way to go, until the necessary facilities for these vehicles (e.g. charging stations) are accessible to everyone.

For this reason, hybrid vehicles are seen as an immediate solution to reduce the emissions of the transport sector, till the prevalence of the zero-emission vehicles is a reality.

2. Hydraulic Hybrid Vehicles

Hybrid Hydraulic Vehicles (HHV) differ from Hybrid Electric Vehicles (HEV) mainly in the type of energy used in the hybridization of the powertrain.

While HEVs include electric machines (motors and/or generators) and batteries, HHVs incorporate hydraulic machines (motors and/or pumps) and hydro-pneumatic accumulators.

As in HEVs, the way the hydraulic machines and the ICE are placed and connected within the power rain characterizes the configuration of the HHVs (Figure 2- 1).

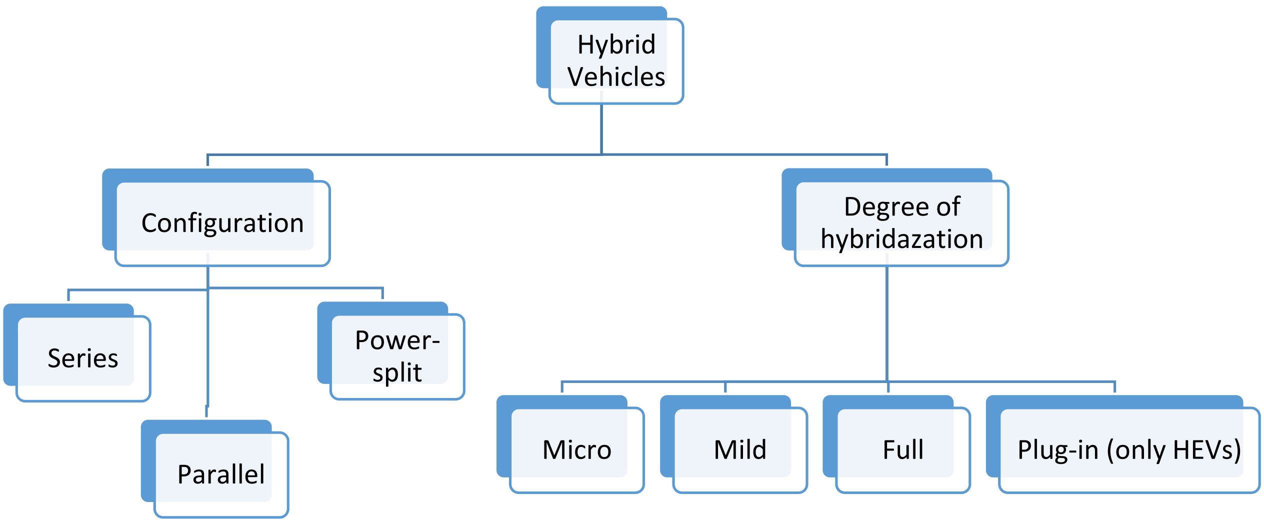

Figure 2- 1: Classification of hybrid vehicles

2.1 Configuration of Hydraulic Hybrid Vehicles

Hydraulic hybrid power train arrangements can be basically classified into three categories based on their configurations: series, parallel and power-split.

2.1.1 Series HHVs

In series HHV (**Figure 2- 2**), sometimes called “pure hydrostatic system”, the mechanical power output of the ICE is converted into hydraulic power at the pump.

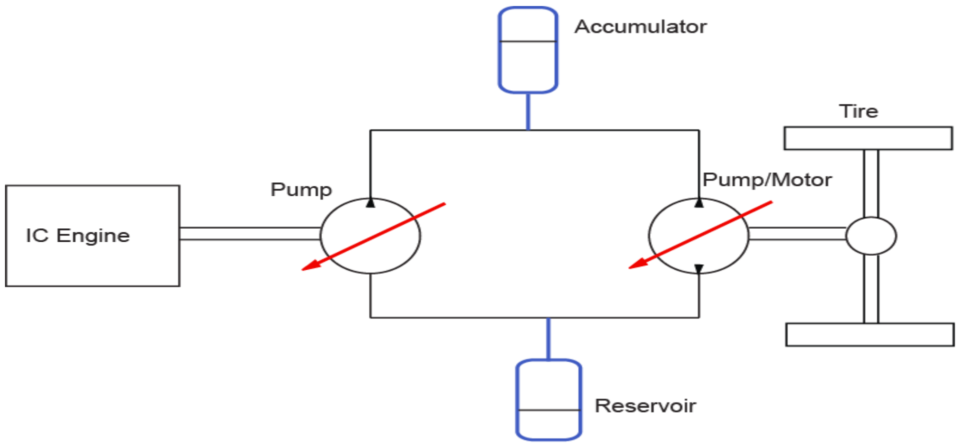

Figure 2- 2: Series HHV [4]

The high-pressure fluid is able to charge the accumulator or to flow directly into the pump/motor at the wheel-end to propel the vehicle.

This configuration permits the vehicle’s ground speed and the ICE speed to be decoupled, allowing the ICE to be controlled at its most efficient operating points and shut down when it is not needed.

However, this system requires the ICE output to be converted into hydraulic energy at the pump and, later, transformed back to mechanical energy at the hydraulic motor. This reduces significantly the efficiency.

2.1.2 Parallel HHVs

In a parallel system, the ICE and the hydraulic pump/motor are both directly connected to the drivetrain. It can be chosen either to drive only with the ICE, only with the pump/motor or combine both at the same time (Figure 2- 3).

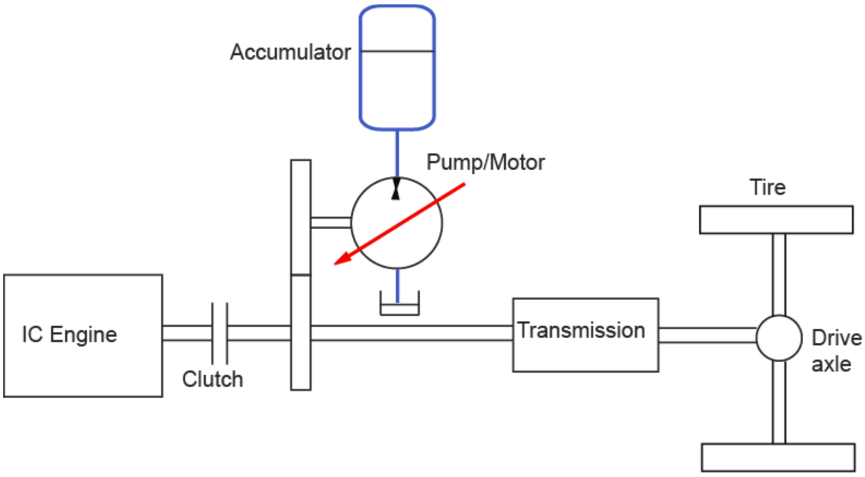

Figure 2- 3: Parallel HHV [4]

This pump/motor is capable of absorbing or providing hydraulic power from or to the mechanical system.

Parallel HHVs present some advantages:

- Only one hydraulic machine is needed (used as a motor and pump).
- The ICE can be used at high speeds (where it is more efficient) and the pump/motor at low speeds (e.g. for urban travel).
- Since the engine is connected directly to the wheels, it eliminates the inefficiency of converting mechanical power into hydraulic power and back, which makes these hybrids quite efficient on the highway.

However, compared to the serial system, the ICE does not normally work at a steady state, so that it is not possible to set it up to work at the point where emissions or fuel consumption are the lowest [5].

2.1.3 Power-Split HHVs

Power-Split HHVs (Figure 2- 4) offer the possibility to combine the advantages of the series and parallel layouts while minimizing their drawbacks, since it is based on the parallel type, but it allows limited series-like operation.

The engine power is divided along two paths: one goes to the pump to produce hydraulic power and one goes through a mechanical gear system to the drive wheels.

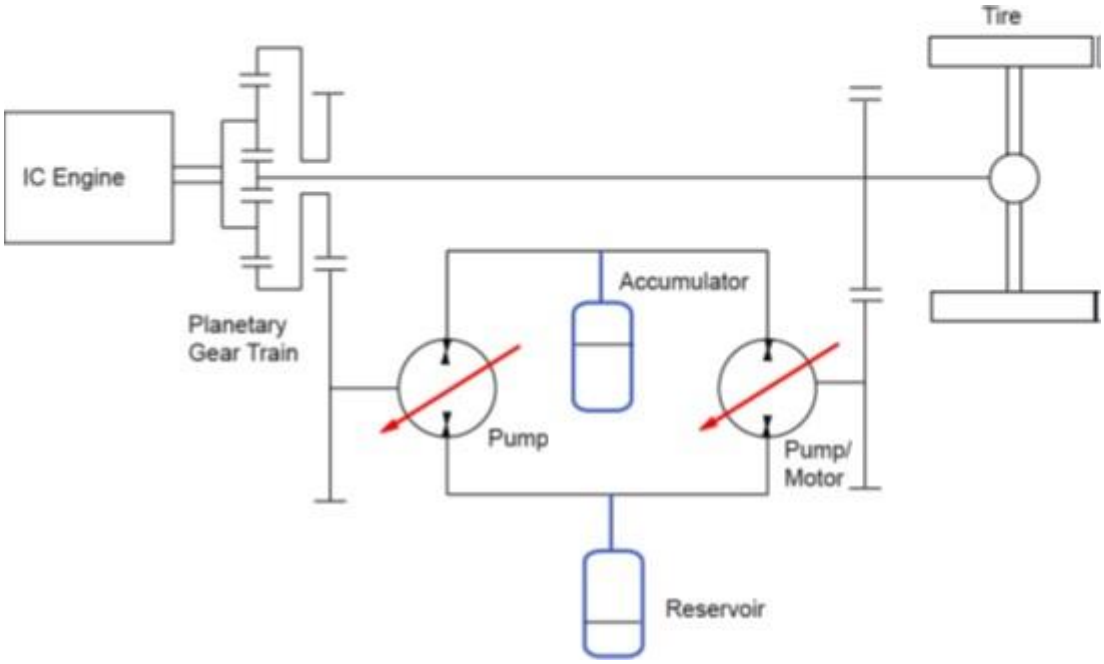

Figure 2- 4: Power-Split HHVs [4]

ICE, pump and pump/motor are connected through a planetary gear. As shown in Figure 2- 5, the ICE is connected to the planetary carrier (red), the pump to the sun gear (white), and the pump/motor to the ring gear (black).

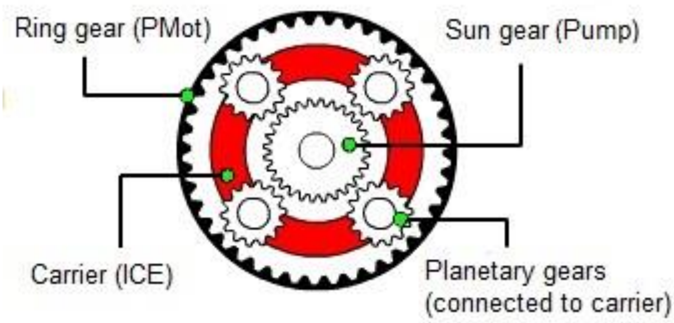

Figure 2- 5: Power-split gear [6]

With this power-split device, a wide range of linear combination of speeds of the three machines is possible. The orange straight from **Figure 2- 6** can “move” freely connecting the coloured of the pump, ICE and pump/motor which represent the possible working points of each.

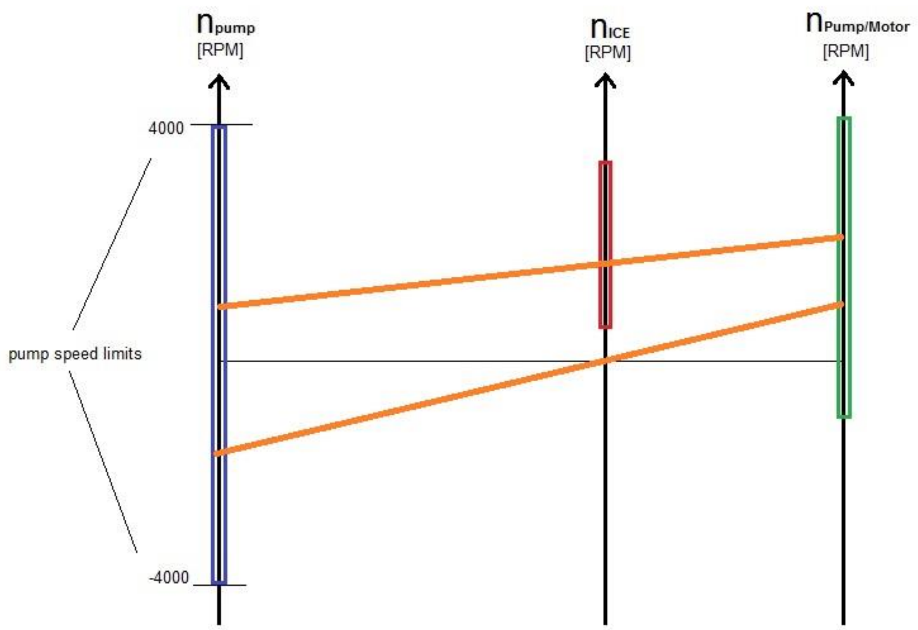

Figure 2- 6 Linear combination of speeds at the PGS

This gives a great liberty in terms of driving points of each machine and great optimization potential. It also makes control more complex. In HEVs, this system is often called "e-CVT" (electronically controlled continuously variable transmission), due to the continuously variable speeds and torque that it allows.

2.2 Degree of hybridization

The degree of hybridization is a term more commonly applied to HEVs, but it can also be used here.

Micro HEVs are similar to conventional ICE vehicles with start/stop systems. The difference usually lies on more powerful starter-generators and batteries, capable of regenerative braking. Accessories can be mostly powered by the battery and the generator does not start running until the battery has a low state-of-charge (SOC) [5]. This concept is hardly applicable to HHVs.

In mild hybrid vehicles, the drivetrain is fitted with regenerative braking and the electric or hydraulic machines are able to assist the ICE in some situations such as boosting.

Nevertheless, electric-only or hydraulic-only driving is very restricted [5]. Most of the developed HHVs until now could be classified in this category.

Full hybrid vehicles can be driven in three different modes: electric-only or hydraulic-only, ICE-only or combining both power sources. Nowadays, most of the HEVs include this characteristics. Although actual can HHVs are capable of driving on hydraulic-only mode, they are not capable of doing for long.

Plug-In HEVs (PHEV) are hybrid electric vehicles, whose batteries can be recharged from the electric power grid while it is parked [2]. This concept is not applicable to HHVs.

2.3 Components of a Hydraulic Hybrid Powertrain

For the hybridization of the powertrain, several components are necessary: hydraulic machines, hydraulic accumulators, gears and clutches necessary to combine both power sources.

2.3.1 Hydraulic Machines

The most common hydraulic machines used in HHVs are the axial piston pumps, since efficiencies can reach values of 95% and they are usually able to work at high pressures (up to 400 bar) [7]. In this project, the displacement of the pump is variable (Figure 2- 7).

This type of pump has a number of pistons in a circular array within a cylinder block (B). The angle of the swash plate (A), controlled through a hydraulic or electro-mechanical actuator, determines the displacement of the machine (h). The swash plate does not rotate, so that the piston absorbs the fluid from part “S” starting from a zero displacement, until it rises to its maximum displacement, and discharges the oil in part “D”.

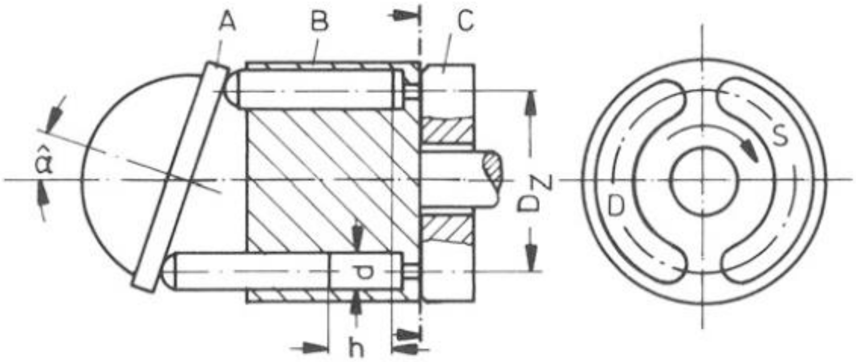

Figure 2- 7 Schematic view of an axial piston pump with variable displacement [7]

The following equations describe the behaviour of these machines when operating as a pump [8]:

$$Q = \frac{V_g \cdot n \cdot \eta_{vol}}{1000} \tag{1}$$

$$T = \frac{V_g \cdot \Delta p \cdot \eta_{vol}}{20 \cdot \pi \cdot \eta_{tot}} \tag{2}$$

$$P = \frac{Q \cdot \Delta p}{600 \cdot \eta_{tot}} \tag{3}$$

If the hydraulic machine is working as a motor, there are some changes in the equations:

$$Q = \frac{V_g \cdot n}{1000 \cdot \eta_{vol}} \tag{4}$$

$$T = \frac{V_g \cdot \Delta p \cdot \eta_{tot}}{20 \cdot \pi \cdot \eta_{vol}} \tag{5}$$

$$P = \frac{Q \cdot \Delta p \cdot \eta_{tot}}{600} \tag{6}$$

Where the symbols in the equations represent the following variables and have the corresponding units [8]:

Displacement	V_g	[cm^3]
Angular Speed	n	[RPM]
Torque	T	[Nm]
Pressure Difference	Δp	[bar]
Power	P	[kW]
Volumetric Efficiency	η_{vol}	[-]
Total (isentropic) Efficiency	η_{tot}	[-]
Flow Rate	Q	[L/min]

The volumetric efficiency expresses the losses due to oil leakage in the machine, while the total efficiency also counts the mechanical losses.

These equations are used in the regulation and control of the Pump and Pump/Motor (see chapter 3.5.1 and 3.5.2).

Table 1 shows a range of common technical data for these pumps.

Table 1: Typical values for axial piston variable pumps [7]

Nominal Pressure (bar)	Maximal Speed (RPM)	Total Efficiency
150 - 420	500 - 5000	0,85 - 0,95

2.3.2 Hydro-pneumatic Accumulator

The mission of the hydro-pneumatic accumulator is to provide the hydraulic machines with power. This power is stored in form of pressure by compressing a gas.

Hydro-pneumatic include two types of fluids:

- Liquid, which is the fluid which flows through the rest of the hydraulic circuit and machines. They are normally special oils determined by the requirements of the hydraulic machine.

- Gas, normally composed exclusively by nitrogen gas, which is needed in the accumulator due to the low (or zero) compressibility of the oils and, therefore, inadequacy to store hydrostatic energy [7].

Hydro-pneumatic accumulators are mainly classified in the way the liquid and the gas are separated: bladder, diaphragm and piston type (Figure 2- 8).

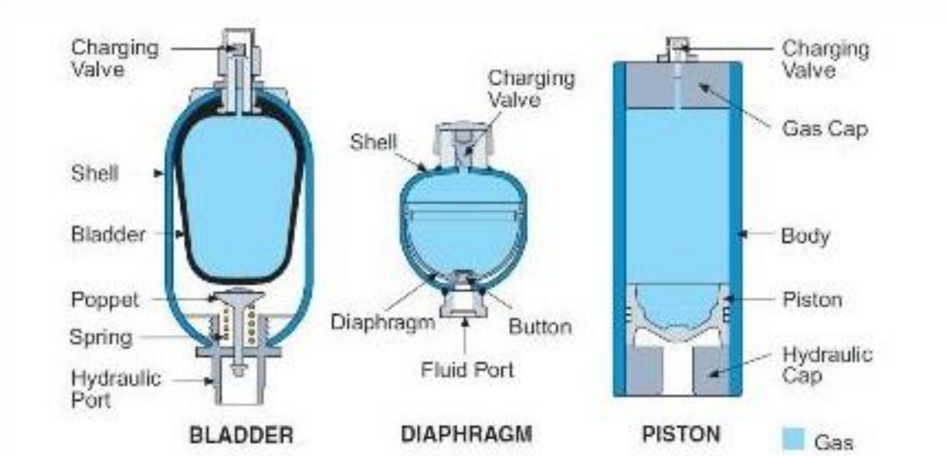

Figure 2- 8: Types of hydro-pneumatic accumulators [9]

When it comes to energy storage, there are two parameters which are used to compare the different types of systems (batteries, fuel cells, flywheels, electric double-layer capacitors or hydro-pneumatic accumulators). These parameters are the power density (W/kg) and the energy density (Wh/kg), which are often plotted in a “Ragone diagram” (Figure 2- 9).

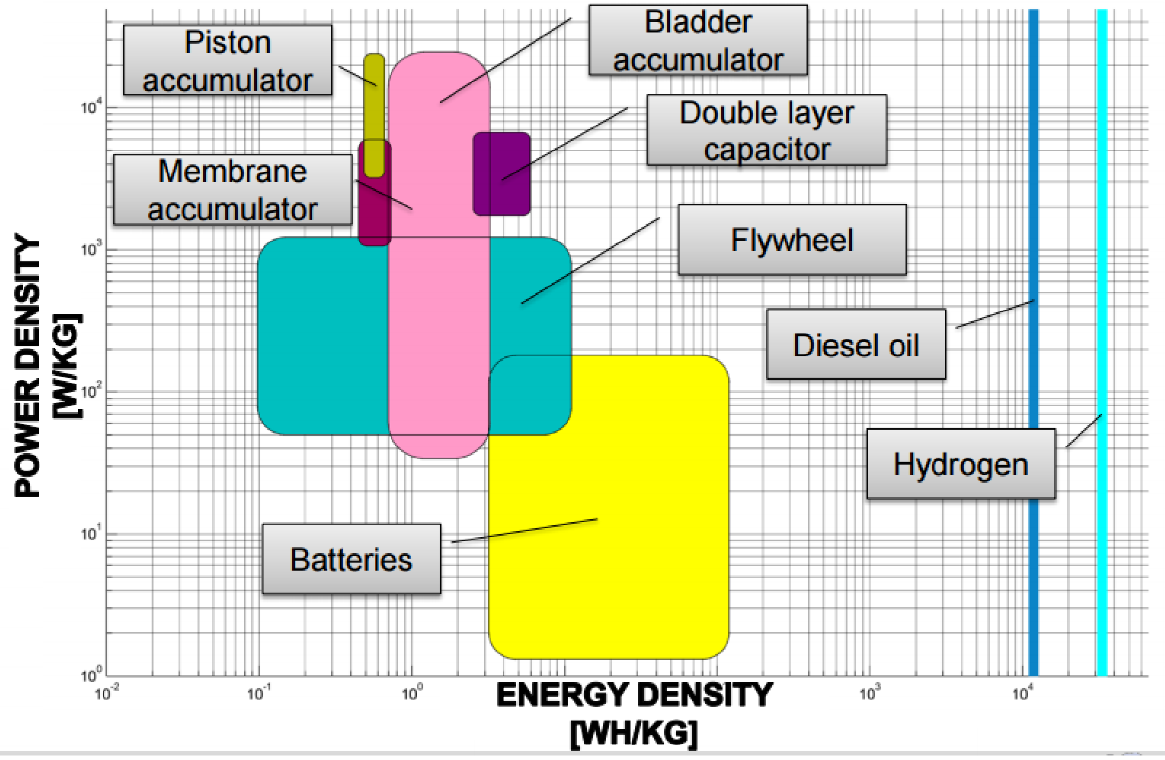

Figure 2- 9: Ragone diagram [10]

Hydro-pneumatic accumulators are characterized, by a high power density, but low energy density. This distinguishes the driving behaviour of HHVs, which are not normally able to operate for a long distance on hydraulic-only mode, due to the low energy density.

Nevertheless, a high power density makes hydro-pneumatic accumulators able of regenerating a high percentage of the braking energy and use is it later to help the vehicle to boost. This is why HHVs a very appropriate for urban driving and for services with many braking events.

Moreover, hydro-pneumatic accumulator have a higher life expectancy than other systems (Figure 2- 10).

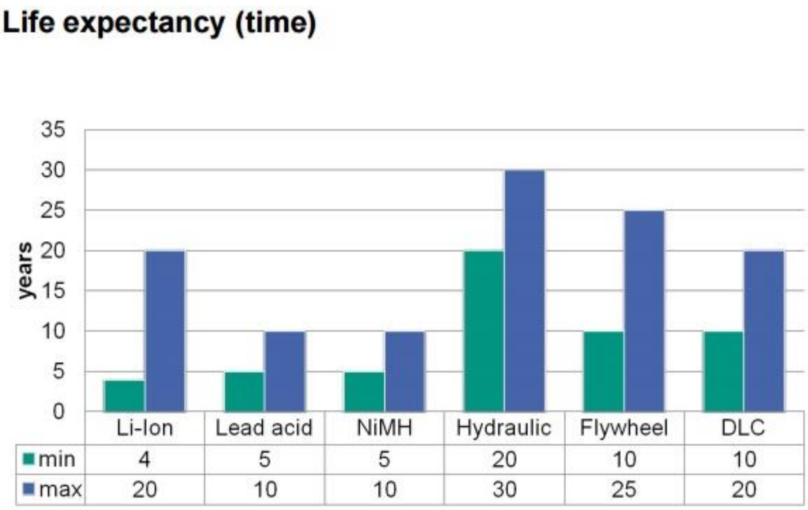

Figure 2- 10: Life expectancy of energy storage systems [10]

Figure 2- 11 shows a qualitative comparison between the main energy storage systems, showing that the only significant disadvantage of the hydro-pneumatic accumulator is the low energy density.

Technology	Electrical		Mechanical	Hydraulic
Energy storage device	Double layer capacitor	Lithium ion battery	Flywheel	Hydraulic accumulator
Energy density	–	+	o	–
Power density	+	o	o	+
Ageing / Capacity loss	o	–	o	+
Temperature sensitivity	–	–	o	+
Self discharge	–	o	–	+
Packaging	o	o	o	o
Cost effectiveness	–	o	o	+

Figure 2- 11: Qualitative comparison between energy storage systems [11]

The dynamic model of the accumulator is based on a polytrophic process described by Equation 7, which corresponds to the charging and discharging actions of the accumulators (processes 1-2 and 3-4 of Figure 2- 12).

$$p \cdot V^n = const. \tag{7}$$

Where the symbols in the equation represent the following variables and have the corresponding units:

p	Pressure	[Pa]
V	Volume	[m³]
n	Polytropic exponent	[-]

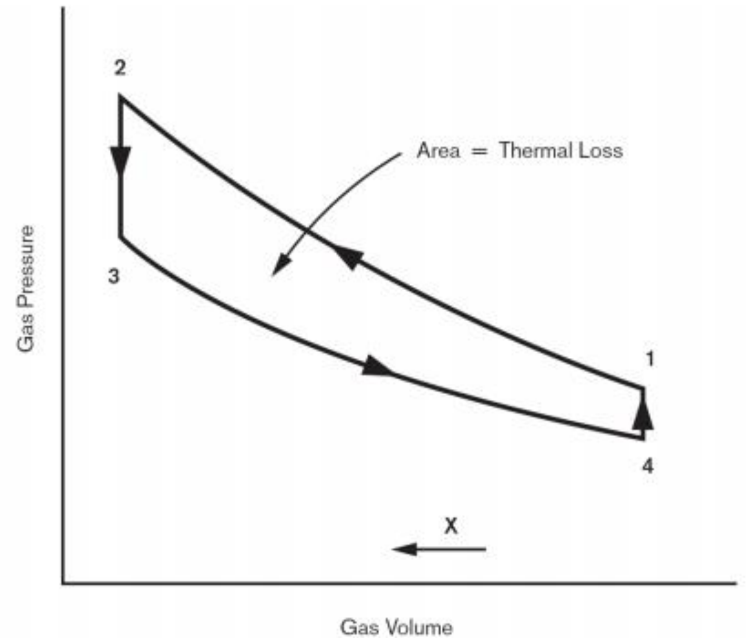

Figure 2- 12: Thermodynamic process of the gas in the accumulator

The polytropic exponent ("n") takes values from 1 (isothermal) to 1,4 (adiabatic). In the model this process is defined as adiabatic, since it happens very quickly and heat exchange is low.

The oil volume in the accumulator ("V_L") is calculated by resting to the total volume of the accumulator ("V_0"), which is constant, the calculated gas volume ("V") (Equation 8).

$$V_L = V_0 - V \tag{8}$$

In processes 2-3 and 4-1 of Figure 2- 12 there is a pressure drop in the accumulator due to heat transfer to the ambient, which represents a thermal loss.

An important variable of the accumulator is the state-of-charge (SOC), which is defined as the division of the current pressure in the high-pressure accumulator and the maximal pressure that it can hold (e.g. 350 bar) (Equation 9).

$$SOC = \frac{p_{current}}{p_{maximal}} \tag{9}$$

2.3.3 Internal Combustion Engine

The engines used in HHVs are basically the same as in conventional vehicles. The difference is when and how they are used.

Engines are more efficient when they operate at high loads and low speeds. Depending on the hybrid configuration, it may be possible to decouple sometimes the current vehicle speed from the requirements and operate the ICE at a more efficient point.

The concept "downspeeding" stands for the process of reducing the ICE speed in order to increase its efficiency (Figure 2- 13).

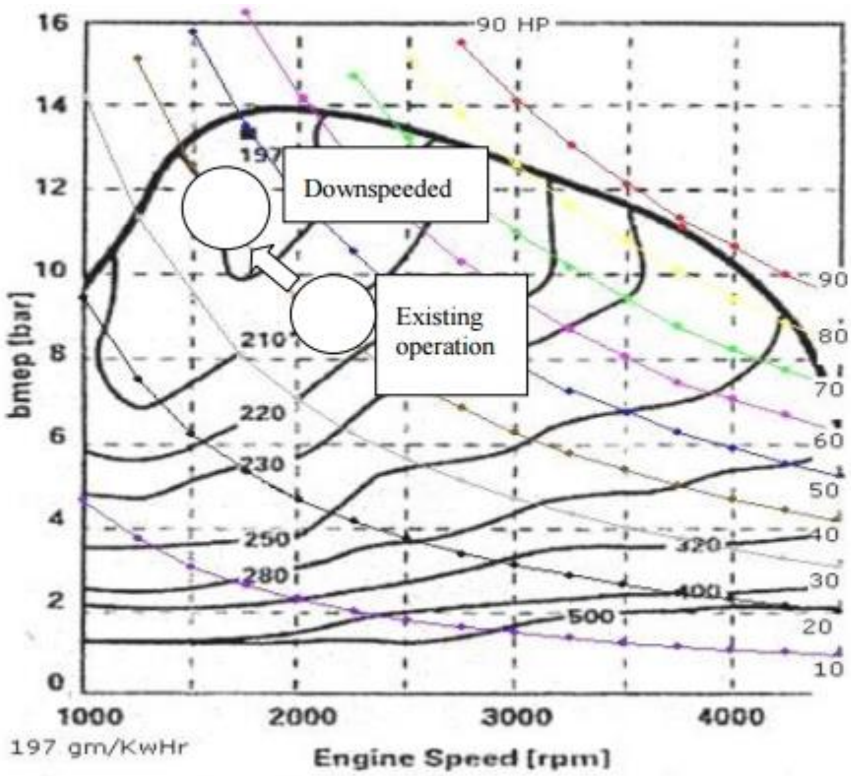

Figure 2- 13: Downspeeding of an engine [12]

One of the aims a power-split hybrid powertrain is to set the ICE at its most efficient point at every moment. Figure 2- 14 shows the operating points of a Toyota Prius, a power-split HEV, during the New European Driving Cycle (NEDC). It can be seen how the operating points are close to the red line, which represents the most efficient working points for a given power value.

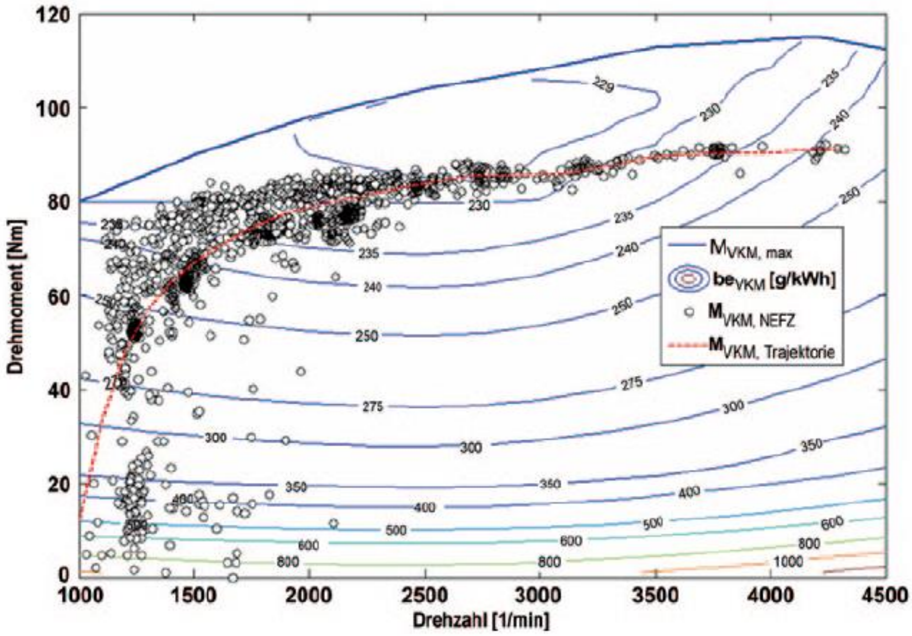

Figure 2- 14: ICE operating points in the NEDC and the resulting trajectory [5]

2.3.4 Hydraulic Fluid

The liquid flowing through the hydraulic circuit are special oils, whose required characteristics are often specified in the catalogue of the pumps.

There are two important criteria for their use in hybrid vehicles:

- High and low-temperature performance (e.g. from -40°C to +150°C).
- Avoid thermal breakdown (causes a drastic viscosity change) and oxidation.

For this purpose, hydraulic circuits count with an oil conditioning system to keep the oil clean and under temperature.

2.4 State of the Art

HHVs are less common than HEVs. However, there have been several projects and developments of such power trains.

2.4.1 Hybrid Air

PSA Peugeot-Citroën is developing together with Bosch a new concept of an urban HHV: the Hybrid Air [13].

This patented technology consists of a full hybrid, which implements a power-split configuration, as it is shown in Figure 2- 15.

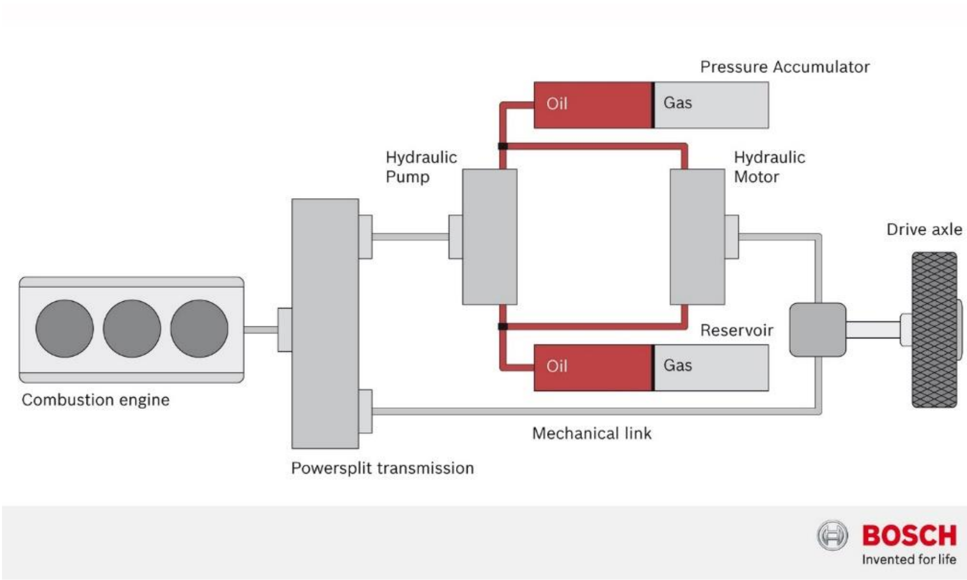

Figure 2- 15: Power-split hybrid powertrain by Bosch [14]

PSA expects that this vehicle will achieve an emission level of 69 g CO2/km and a fuel consumption of 2,9 L/100km in combined-cycle driving. This data correspond to the technology implanted in already existing models such as the Citroën C3 or the Peugeot 208, without any specific adaptations.

The conventional Citroën C3 has a fuel consumption of 5,1 L/100km and emissions of 104 g CO2/km [15], which means that the HHV could reduce the fuel consumption by a 43%.

Figure 2- 16 shows how the components of the hybrid power train are placed in the version given by PSA in 2013, although the newest version, from December 2014 (Figure 2- 17), places both accumulators on the rear axle.

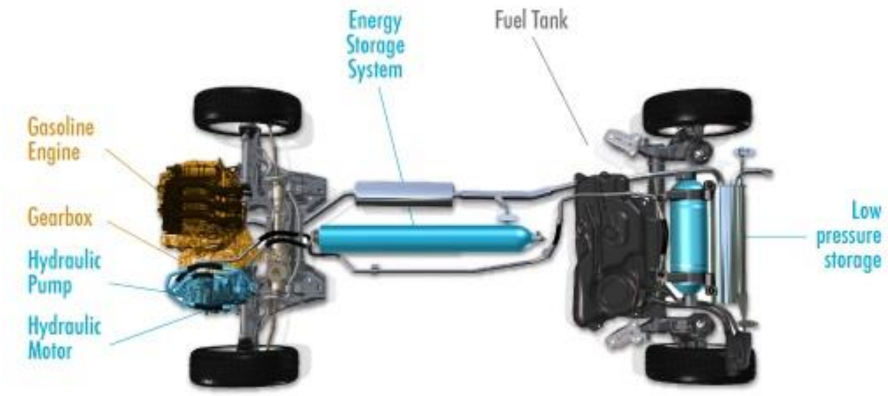

Figure 2- 16 Placing of the components within the Hybrid Air (2013) [16]

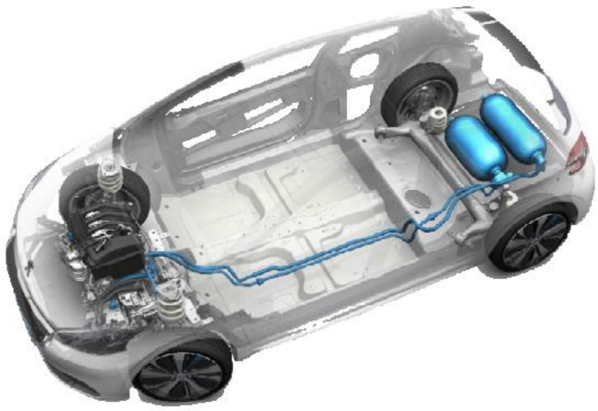

Figure 2- 17 Placing of the hydropneumatic accumulator (2014) [17]

Figure 2- 18 displays the different driving modes of the Hybrid Air:

i) ICE-only , for long journeys or when driving at higher speeds
ii) Hydraulic-only, fort short journeys
iii) Combined power
iv) Regenerative Braking

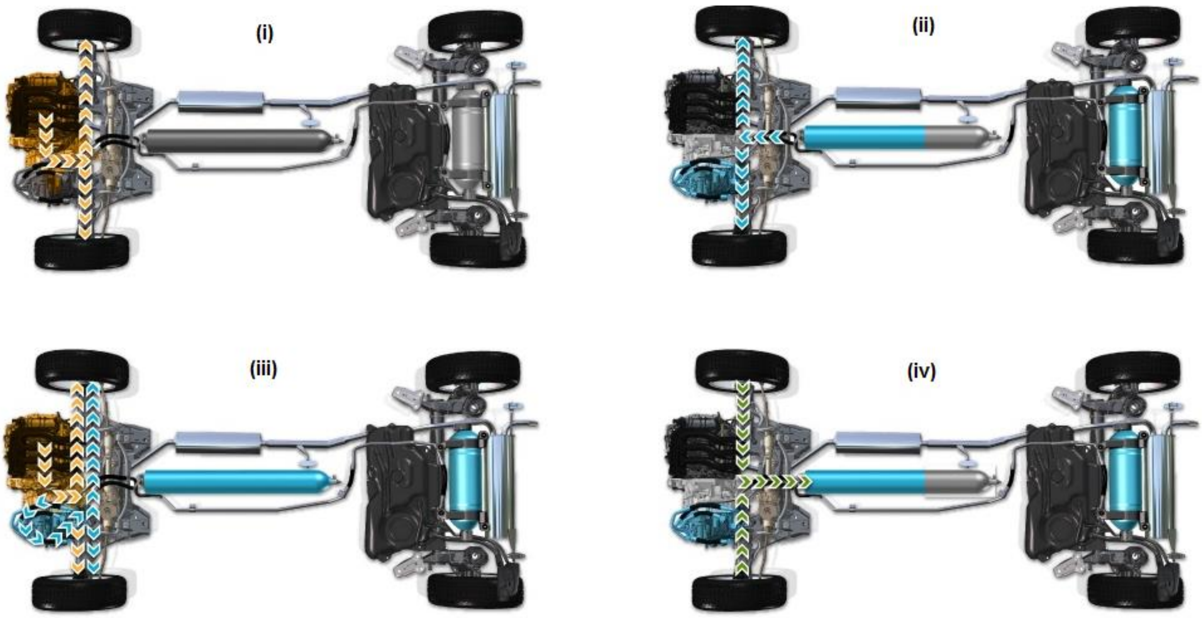

Figure 2- 18 Driving modes of the Hybrid Air [16]

PSA's statements about the pricing of the Hybrid Air have set it to a similar level of a conventional Diesel vehicle (Figure 2- 19).

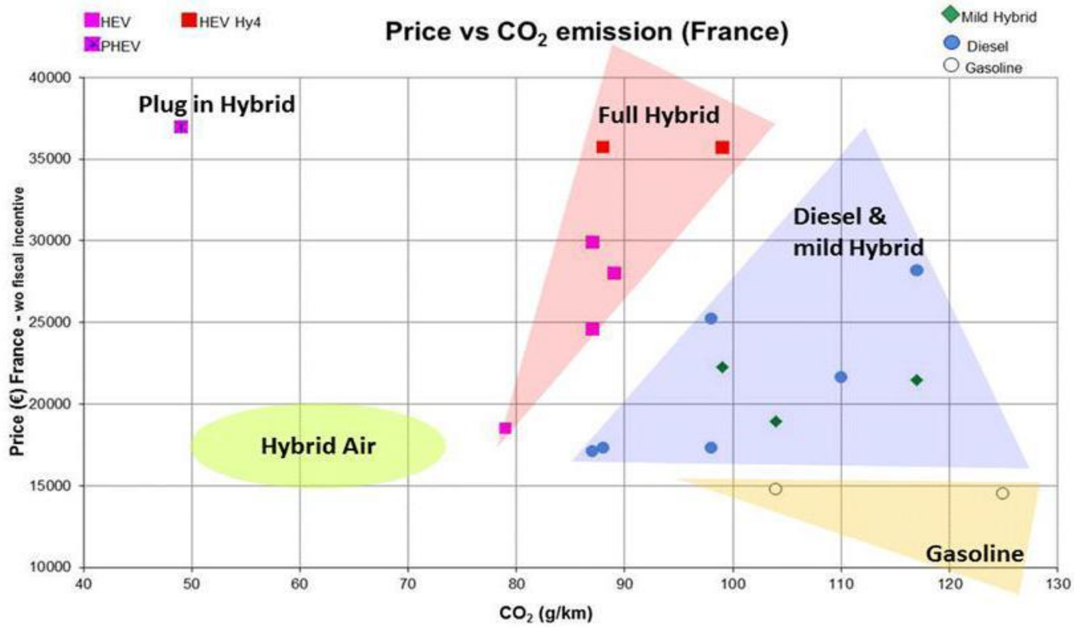

Figure 2- 19 CO2 emissions vs. Price [17]

This Hybrid Air is part of the "2L goal"-project developed by the French government, which also includes researches in new materials, reducing drag and tyre rolling resistance [17].

Until nowadays, this has been the only developed HHV with a power-split configuration.

2.4.2 Commercial Vehicles and Mobile Machinery

HHV have been especially employed in commercial vehicles which work in urban areas and in mobile machinery, since their service cycles involve many braking events. Moreover, such vehicles have enough space two incorporate large hydro-pneumatic accumulators.

Bosch Rexroth has developed the Hydrostatic Regenerative Braking (HRB) system in order to be used in refuse trucks, school buses or delivery vehicles [18].

The components of the HRB are the following [18] and could be implemented as in Figure 2- 20:

- A4VSO Axial Piston Unit + Gearbox
- Hydraulic Accumulator
- Pressure Relief Valve
- Valve Control Block HIC
- BODAS Controller RC

Figure 2- 20: Implementation of the HRB on a truck (red components) [18]

The HRB system can be carried out in a parallel or in a series configuration (Figure 2-21).

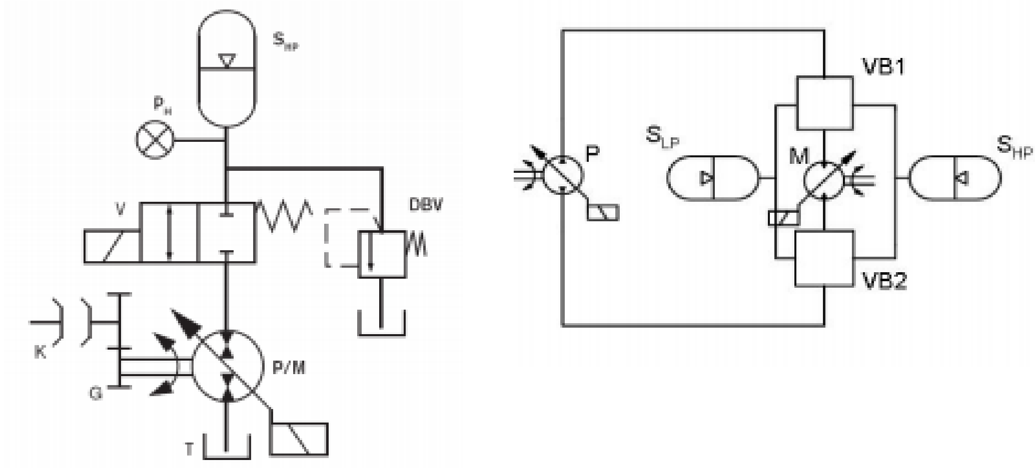

Figure 2- 21 Parallel (left) and series (right) configuration of the HRB [19]

Tests have been made, implementing the parallel HRB on a garbage truck and the series scheme on a forklift (lift truck). Table 2 displays data used in those test.

Table 2: Vehicle and component data for the simulation of HRB systems [19]

	Forklift (series)	Refuse truck (parallel)
Vehicle mass [t]	10	25
Required power [kW]	85	200
Max. HRB-Speed [km/h]	23	25
Hydr. Accumulator Volume [L] (each)	20	50

The cycle driven by the garbage truck consist of 100 metres of distance, achieving a speed of 25 km/h in the midway. This simulates the cycle between two collection points.

The cycle tested on the lift truck is the one showed in Figure 2- 22.

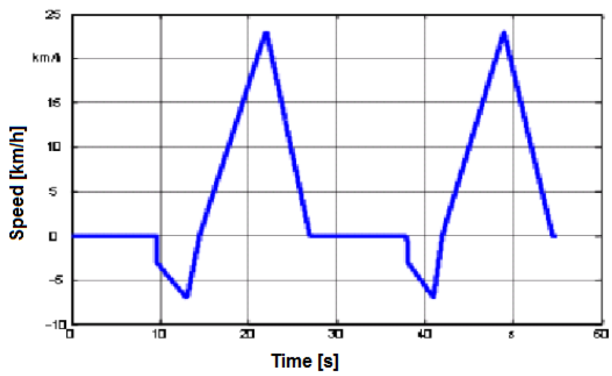

Figure 2- 22: Cycle driven by the lift truck with HRB [19]

In both cases, fuel consumption is reduced by approximately 25%, depending on the distances travelled by the vehicles between two stops. Maximal acceleration of both vehicles is also enhanced with the HRB [19].

Another example of commercial HHVs is the one designed by Environmental Protection Agency (EPA) of the United States for the delivery company UPS in 2009 (Figure 2- 23). It counts with a series configuration and, on real service tests, fuel economy has been improved by 60 - 70 percent, estimating the payback of the HHV in two to three years.

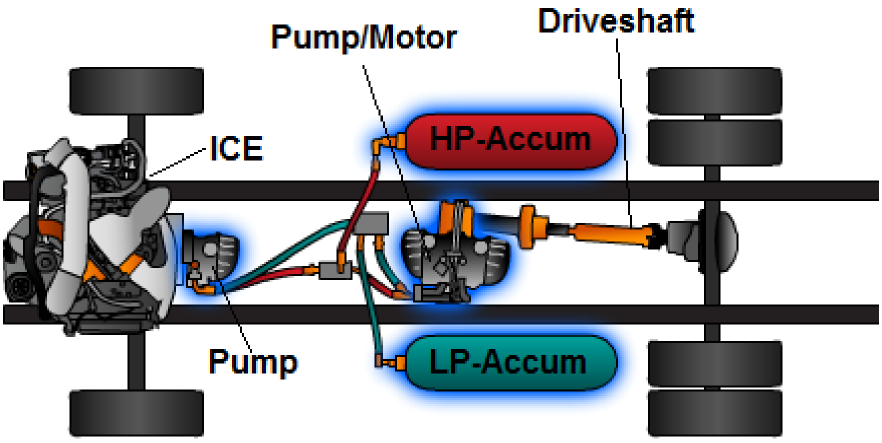

Figure 2- 23: HHV Layout of EPA [20]

More information about the demonstration HHVs of the EPA can be found on its website [20].

2.4.3 Hydrid

The Hydrid is a concept HHV with series configuration designed by the Dutch company Innas (Figure 2- 24).

It is a series HHV, but with the particularity that in includes four in-wheel hydraulic pump/motor. There are also two hydraulic transformers to control each axis.

The pumps used in the Hydrid function with the principle of the “floating cup”, which achieves efficiencies of up to 98% and enhances the behaviour of these machines at low-speeds or during start-up when the vehicle accelerates from standstill.

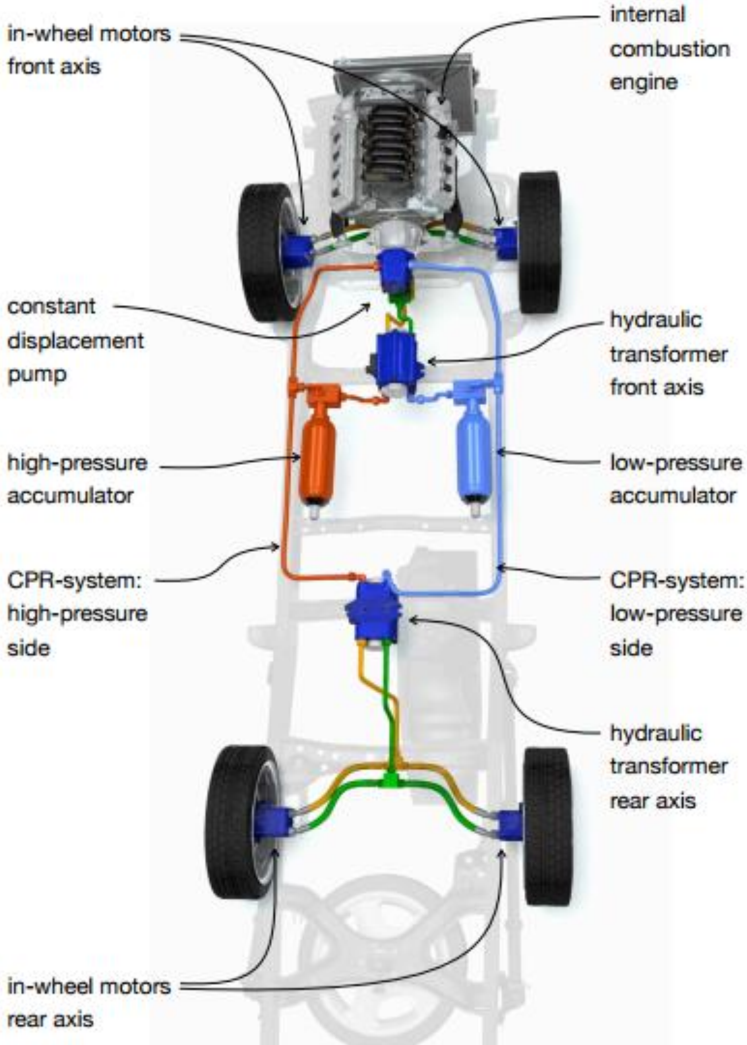

Figure 2- 24: Component's arrangement in the Hydrid [21]

Figure 2- 25 plots the results obtained by Innas for a vehicle with the same engine (100 kW, Diesel) with four-wheel-drive at the NEDC. The fuel consumption is cut by more than the half [21].

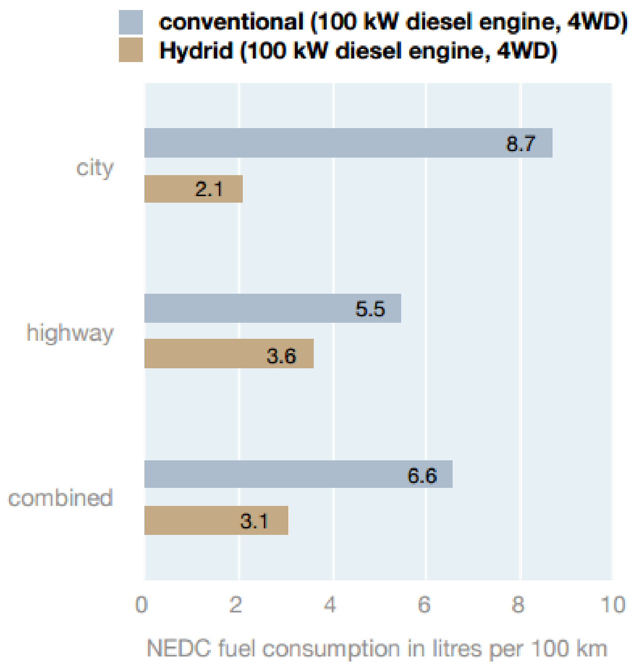

Figure 2- 25: Fuel Consumption results by the Hydrid [21]

3. Modelling

The software used to model and simulate the power-split HHV is GT-SUITE.

GT-SUITE is a simulation software tool with capabilities and libraries aimed to be implemented in applications in automotive engineering. It offers functionalities ranging from fast concept design to detailed systems or component analyses, design optimization and root cause investigations. GT-SUITE provides several tools to integrate models of multi domain systems [22].

The model developed to simulate the behaviour of a power-split HHV has several parts, including the hydraulic circuit, the engine, the vehicle (chassis, tyres, brakes, etc.), mechanical connection parts (gears and shafts) and the control units which regulate each machine (Figure 3- 1).

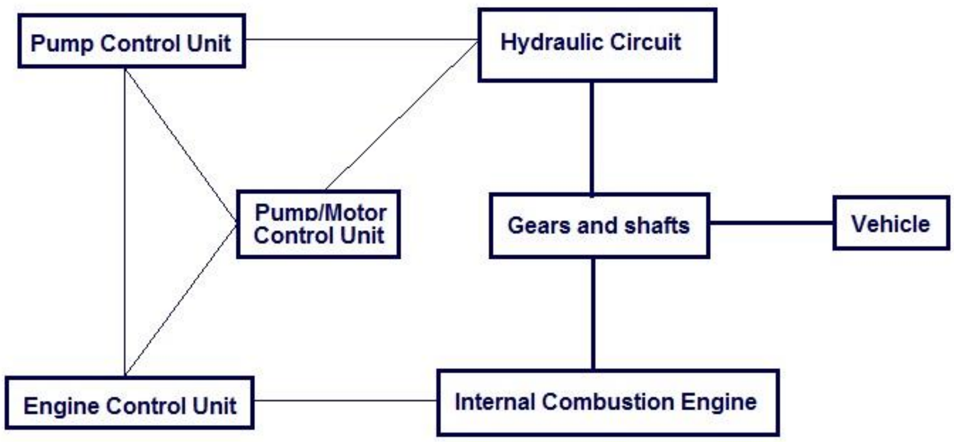

Figure 3- 1: General scheme of the model

3.1 Hydraulic Circuit

The hydraulic circuit in this model is composed by two hydraulic machines, two hydro-pneumatic accumulators and the necessary pipes to connect and control the flow between these components (Figure 3- 2).

How the corresponding shafts of these machines are connected is explained in chapter 3.4.

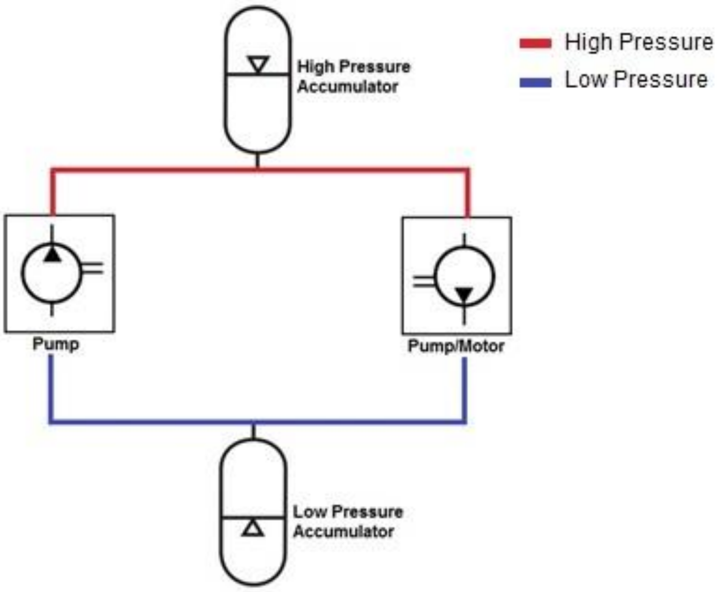

Figure 3- 2: General scheme of the hydraulic circuit

3.1.1 Hydraulic Machines

Axial piston pumps with variable displacement are characterized by parameters such as displacement per revolution, volumetric efficiency and total (isentropic) efficiency.

Pumps are regulated by the displacement, which is externally calculated in the Pump Control or Pump/Motor Control Units (chapters 3.5.1 and 3.5.2) to simulate a variable displacement pump/motor.

The total (isentropic) efficiency is read from a diagram, like the one showed in Figure 3-3.

The diagram used belongs actually to a pump which was used in a different project [23]. Furthermore, this efficiency map corresponds to a certain value of the displacement of the pump (maximal displacement in this case).

It would be desirable to have efficiency maps where the displacement was a third parameter, but this is not viable in the present work, since the calculation of the displacement already depends on the efficiency. This issue is explained in detail in 3.5.1.

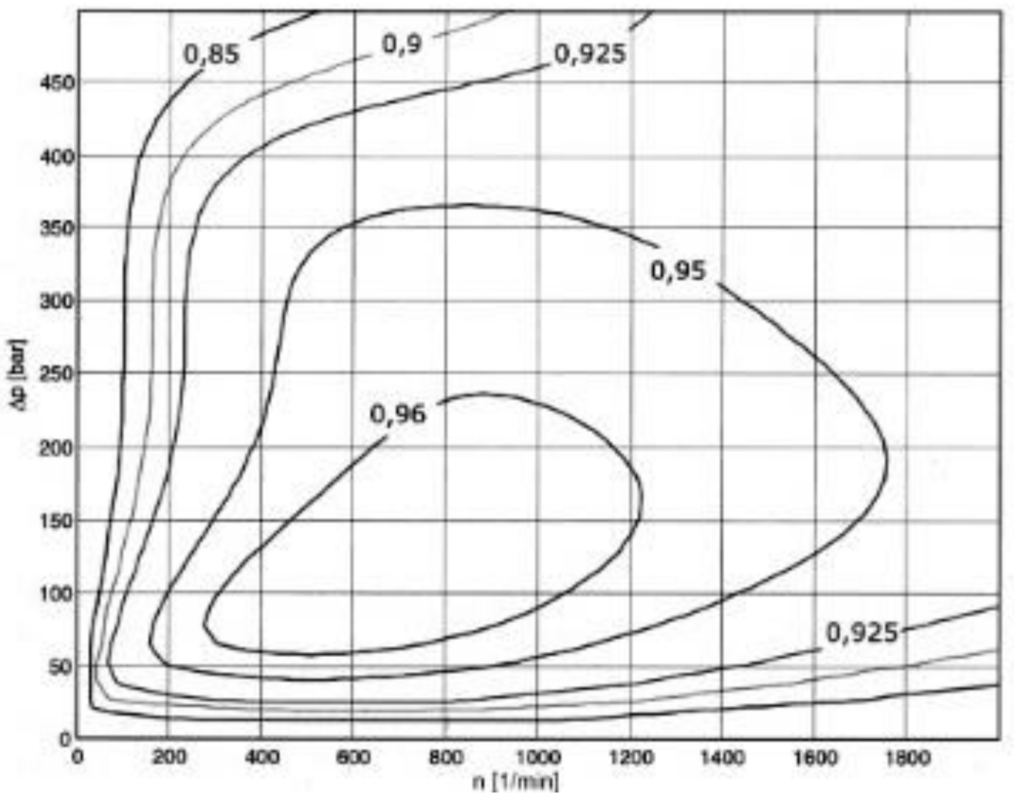

Figure 3- 3: Axial pump total efficiency [24]

The reason, why this diagram is used and not one example from real pumps, is the problem finding these data for real machines (from e.g. Bosch Rexroth).

The volumetric efficiency is set as constant for the same reason.

3.1.2 Hydro-pneumatic Accumulator

The hydro-pneumatic accumulator is modelled as a container embracing two fluids separated by a weightless membrane in a constant volume space.

The thermodynamic properties of both fluids (mineral oil and nitrogen gas) as well the initial state of each one (initial pressure, temperature and oil volume in each accumulator) and the total volume of each accumulator are necessary data for the simulation.

In this model, wall temperature is fixed to same value as in the ambient in order to ease the calculation of the exchanged heat. Nevertheless, these thermal, losses are very low and efficiency values normally reach 98 percent.

3.1.3 Pipes

Pipes are needed to connect the different parts of the hydraulic circuit.

For this model straight pipes with circular section are selected and their initial state (pressure, temperature) is the same as in the accumulators to which they are connected.

Another parameter required is the discretization length, which is needed by the numerical solver which calculates the changes in the state of the flow. The numerical method used to solve the differential equations which correspond to the flows is the implicit Runge-Kutta method.

3.2 Internal Combustion Engine

The mechanical output of the ICE, as the mean effective pressure (MEP), is read from a table when the values of the engine speed and accelerator pedal position are entered.

Two similar tables are used to simulate the engine friction and fuel consumption. The fuel consumption map (efficiency map) is important to set the engine into its most efficient point for a given power value (see chapter 3.5.3).

Other parameters which need to be determined are the engine displacement and the density and specific heating value of the fuel.

3.3 Vehicle

The performance of the vehicle, excluding the power train, is achieved with the interaction of the parts shown in Figure 3- 4.

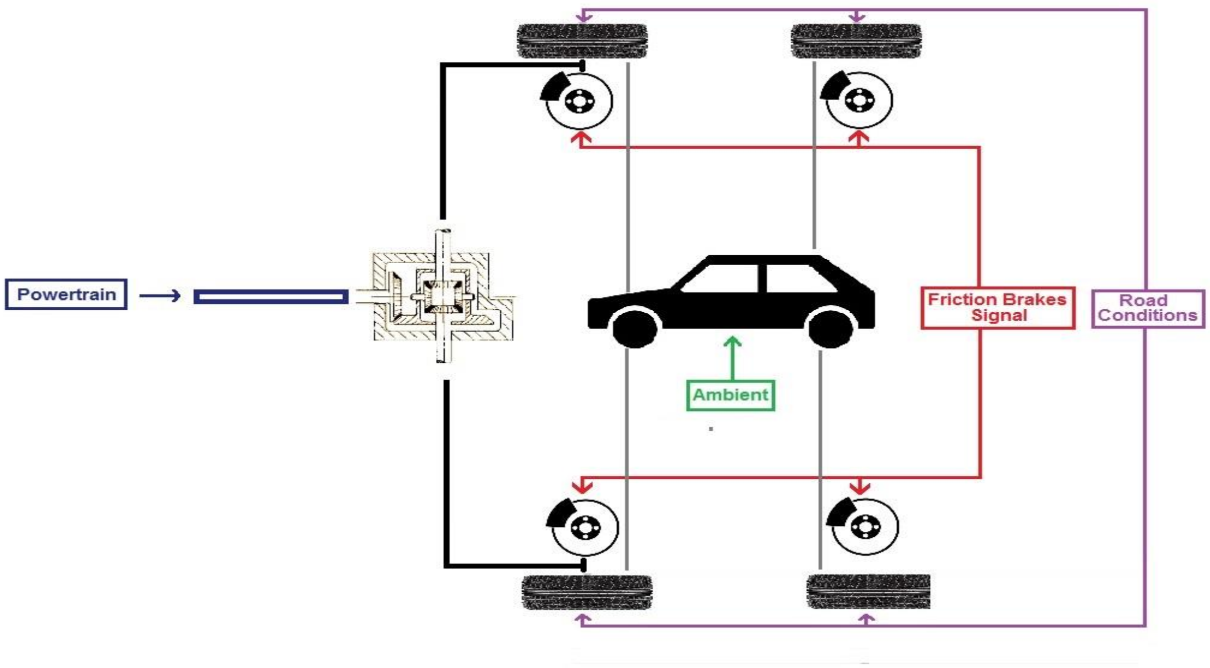

Figure 3- 4: Scheme of the vehicle

The output of the powertrain, consisting of the ICE and the two hydraulic machines, is passed though the driveshaft (blue in Figure 3- 4) into the differential.

The "final gear ratio" couples all the gear rates which can be found in a vehicle transmission together at the differential, and it is important to control the speeds of the Pump/Motor and the other machines. This is explained in detail in chapters 3.5.1 and 3.5.2.

The front-wheel-drive vehicle counts with four tyres, characterized by their resistance factor and the rolling radius, as well as with four friction brakes, which actuate when they receive a signal from the braking control unit (chapter 3.5.4).

It is also significant for the model, to introduce ambient conditions like the air pressure and temperature; and wind speed and direction.

Road grade and curvature take values of zero in this model.

The object represented by the car of Figure 3- 4 represents the vehicles geometry and is used to introduce parameters such as vehicle mass, frontal area, the aerodynamic drag or axle geometry.

3.4 Mechanical Connection Parts

Figure 3- 5 offers a general overview of the powertrain of the model and how the hydraulic circuit and the mechanical parts are connected. As it can be seen, the hydraulic machines are connected through shafts to the planetary gear set (PGS).

In order to keep the Pump speed within its limits and make the ICE capable of achieving its most efficient operating point at every moment, an extra gear is used between the carrier and the ICE (Figure 3- 5). The importance of this gear is explained in chapter 3.5.1.

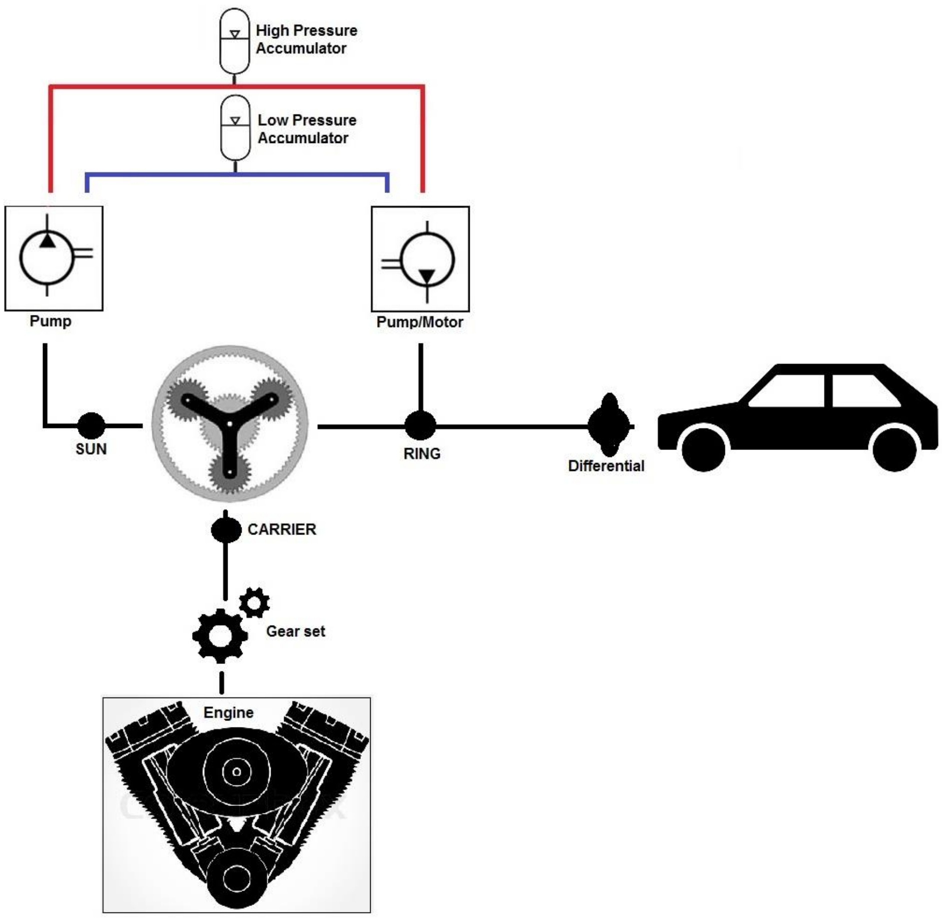

Figure 3- 5: Powertrain of the model

3.4.1 Planetary Gear Set

The planetary gear set (PGS) is the power-split device in the hybrid vehicle.

Similar as in the Toyota Prius, the ICE is connected to the carrier, the Pump to the sun and the Pump/Motor ("PMot") to the ring gear (Figure 3- 6).

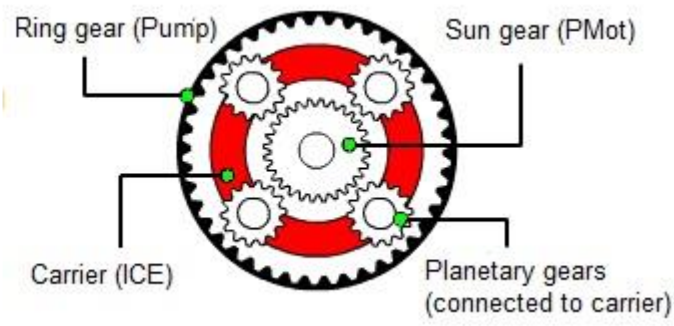

Figure 3- 6: Planetary Gear Set [24]

By introducing the values of the ring gear pitch radius, the number of teeth of the ring gear and sun gear, the geometrical relation of speeds and torques are defined.

Other parameters which have to be fixed are the number of planet gears, inertia value of each gear and mechanical efficiencies between the gears which are connected directly to each other.

In this model, the geometry of the PGS is the same for every simulation. The chosen values were the same that the ones of the Toyota Prius 1st generation (2001) [5]:

Ring gear pitch radius: 57.5 mm

Number of ring gear teeth: 78

Number of sun gear teeth: 30

3.5 Control Units

The control units are responsible of regulating each machine (Pump, Pump/Motor and ICE), as well as of defining the driving strategy to achieve a better fuel economy (e.g. Braking Control Unit).

3.5.1 Pump Control

As already mentioned before, the Pump is just regulated by the variable displacement. The aim of this Pump Control Unit is to calculate it, so that the ICE can be set into the most efficient points for a given power value, which are represented by the white line of Figure 3- 7.

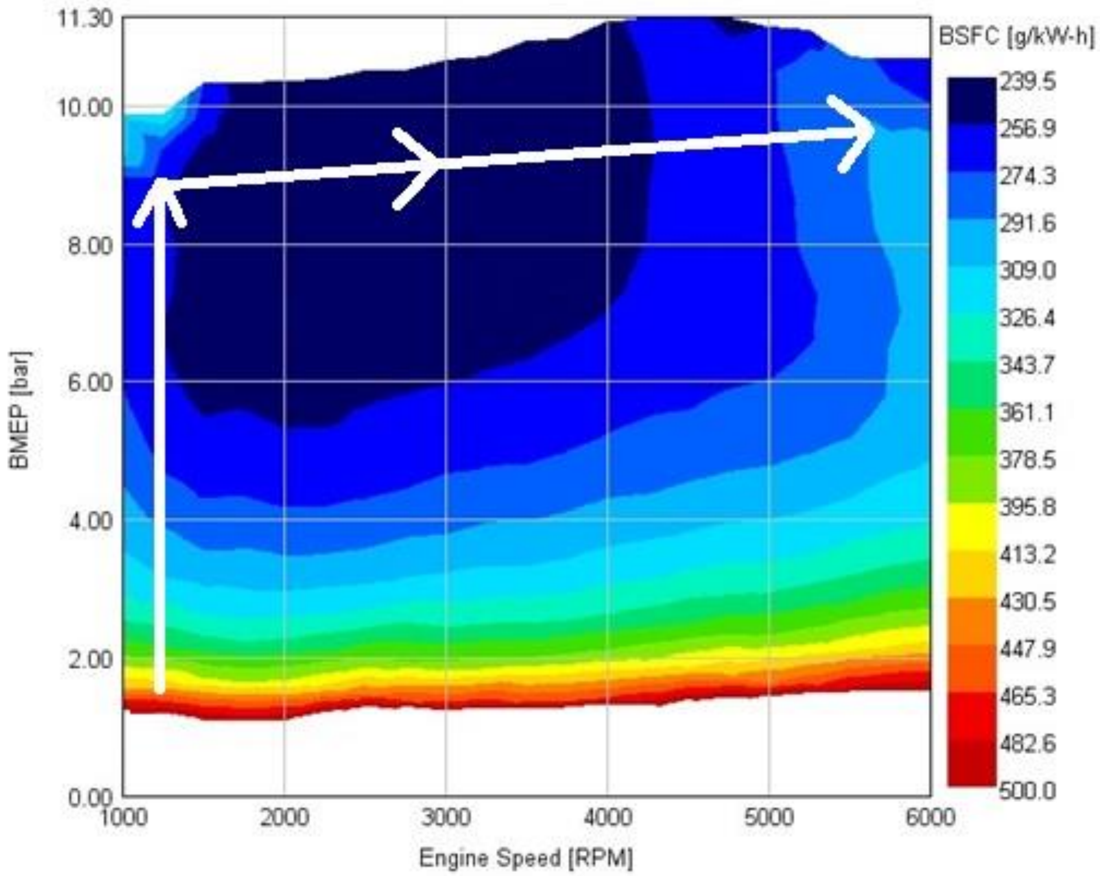

Figure 3- 7: Most efficient ICE operating points

In the first step of the Pump Control Unit, the Pump speed needed to set the ICE to operate in those points is calculated (Figure 3- 8).

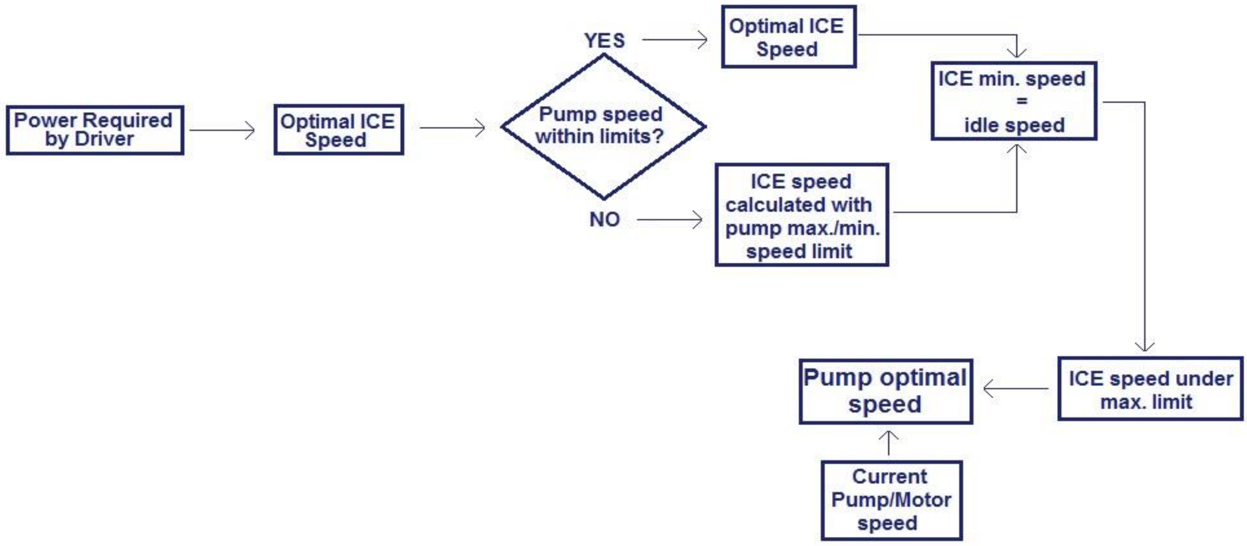

Figure 3- 8: Pump Control (a)

The speeds of the three machines are fixed related to each other through the gear ratios of the PGS, and this restricts the ICE speed to a certain value at every moment, which is expressed by Equation 10 and Figure 3- 9.

$$n_{current\_max/min}^{ICE} = \frac{1}{r_{ICE\_gear}} \cdot \left(\frac{z_{ring}}{z_{ring}+z_{sun}} \cdot n_{max/min}^{Pump} + \frac{z_{sun}}{z_{ring}+z_{sun}} \cdot n_{current}^{PMot} \right) \tag{10}$$

Where the symbols represent the following variables with the corresponding units:

$n_{current\_max/min}^{ICE}$	ICE current speed limit	[RPM]
r_{ICE_gear}	Ratio of the gear between ICE and Carrier	[-]
z_{ring}	Ring gear teeth number	[-]
z_{sun}	Sun gear teeth number	-
$n_{max/min}^{Pump}$	Pump speed limit	[RPM]
$n_{current}^{PMot}$	Pump/Motor current speed	[RPM]

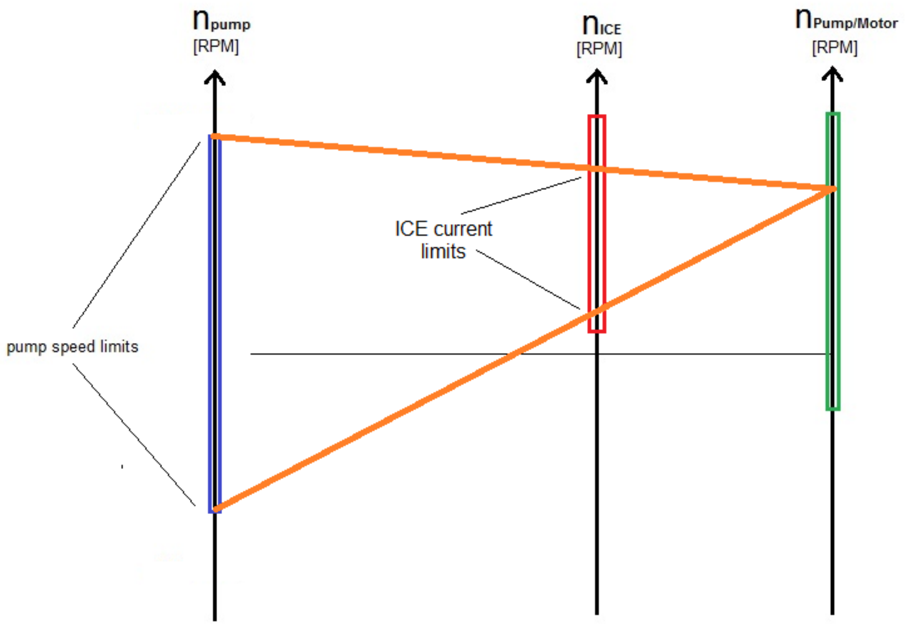

Figure 3- 9: ICE current speed limits

Once, the optimal ICE speed is calculated, it is possible to calculate the necessary speed of the Pump ("$n_{Pump,\ opt}$") (Equation 11).

$$n_{Pump,opt} = \left(1 + \frac{z_{sun}}{z_{ring}}\right) \cdot r_{ICE\_gear} \cdot n_{ICE,opt} - \frac{z_{sun}}{z_{ring}} \cdot n_{PMot\ current} \tag{11}$$

In the second step of the Pump Control Unit (Figure 3- 10), the necessary torque ("M_{Pump}") to change the current Pump speed into the optimal is calculated.

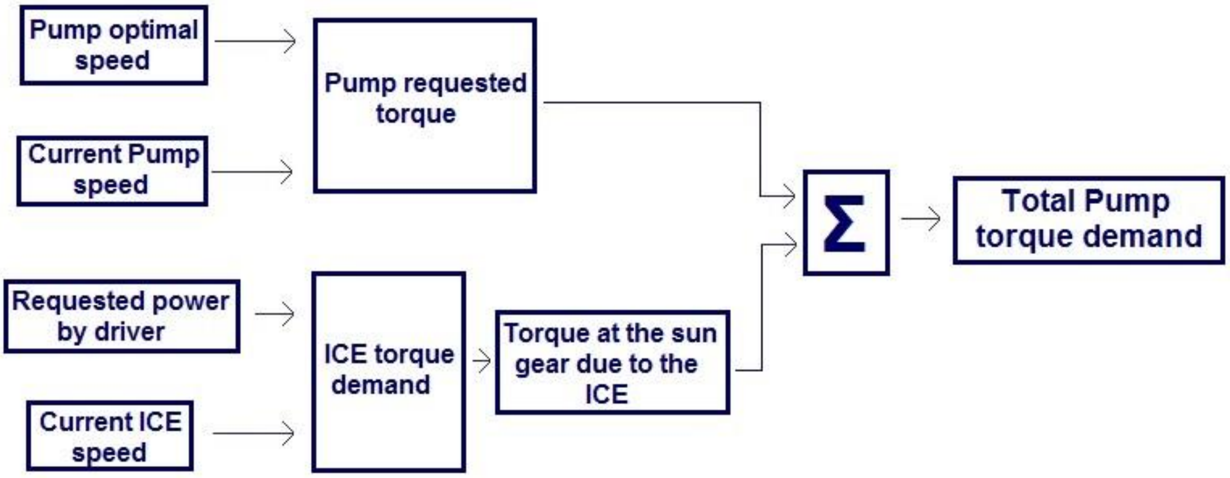

Figure 3- 10: Pump Control (b)

However, at the sun gear, which is directly connected to the Pump, the part of the ICE torque ("M_{ICE}") coming from the power-split also has to be absorbed by the pump.

The total torque requested to the pump ("M_{Sun}") is the sum of these two parts regarding the gear rates of the PGS (Equation 12) which has to be between the upper and lower torque limits of the Pump (e.g. +/- 200 Nm).

$$M_{Sun} = M_{Pump} + \frac{z_{Sun}}{z_{Ring}+z_{Sun}} \cdot M_{ICE} \tag{12}$$

The last part of the Pump Control Unit is dedicated to calculate the displacement of this machine (Figure 3- 11). If the Pump has to act as the ICE starter, the value of the torque is constant (e.g. 40 Nm).If not, the value is the one calculated in the previous step.

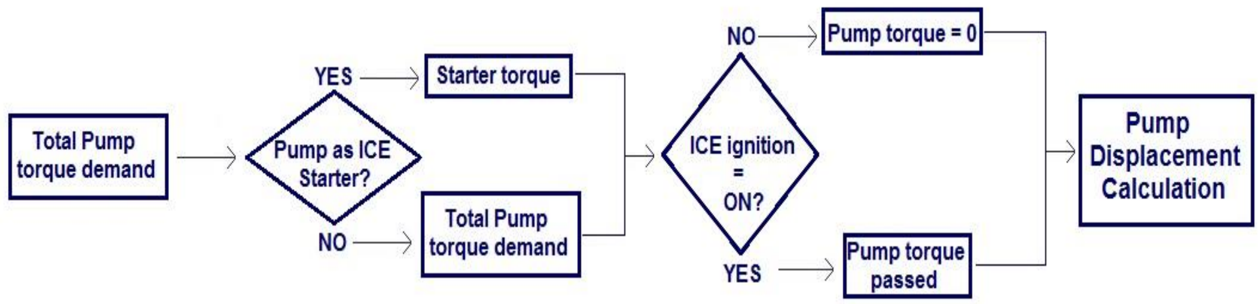

Figure 3- 11: Pump Control (c)

When the vehicle drives on hydraulic-only mode and the ICE does not operate, only the Pump/Motor works. In this case, the torque and, therefore, the displacement of the pump take a value of zero. This does not mean that the Pump does not rotate, though.

Once having transformed the Pump torque into a power signal, the displacement is calculated using Equation 13, which is obtained by combining Equations 1 to 3 from chapter 2.3.1 [25].

$$V_g = \frac{1000 \cdot 600 \cdot \mathrm{P} \cdot \eta_{tot}}{n \cdot \Delta p \cdot \eta_{vol}} \tag{13}$$

3.5.2 Pump/Motor Control

The Pump/Motor Control Unit is used to calculate the requested variable displacement of the machine (Figure 3- 12).

Since the Pump/Motor is directly connected to the ring gear, the torques coming from the Pump and ICE, have to be subtracted from the total demanded torque at the sun to drive the vehicle.

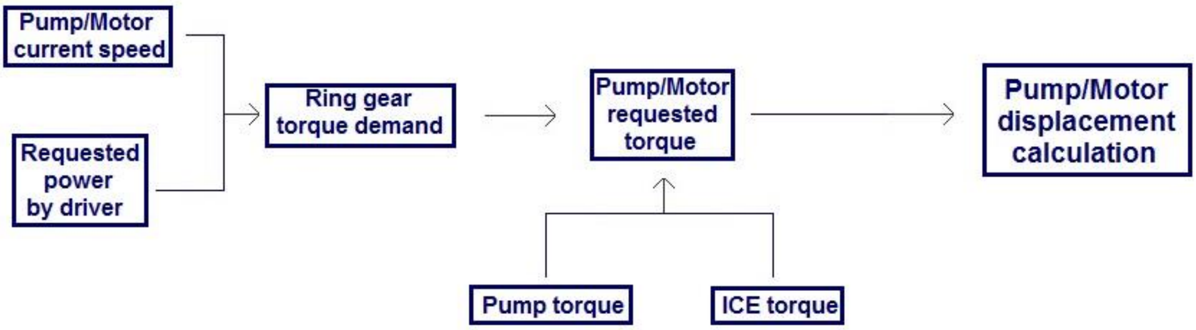

Figure 3- 12: Pump/Motor Control

The displacement is calculated as expressed in Equation 14. This equation is obtained combining Equations 4 to 6 from chapter 2.3.1 [25].

$$V_g = \frac{1000 \cdot 600 \cdot \mathrm{P} \cdot \eta_{vol}}{n \cdot \Delta p \cdot \eta_{tot}} \quad (14)$$

3.5.3 ICE Control

The ICE Control Unit has the mission of deciding when the engine has to be turned on or off, and, at the end, calculate the accelerator pedal position.

The desire of turning the ICE on is represented in the model by the propositional variable “ignition” (takes only the values “0” or “1”).

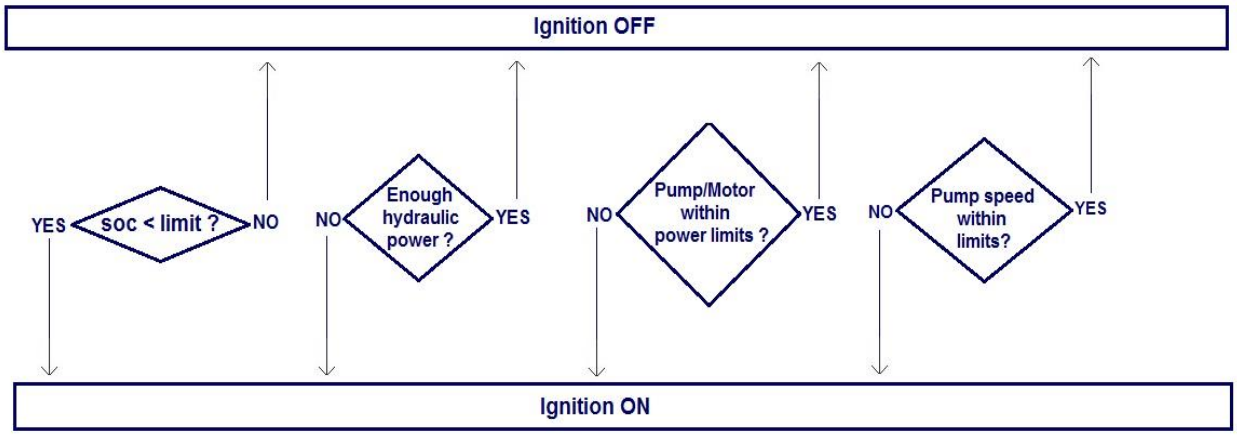

Figure 3- 13 Ignition of ICE

This variable takes the value "1" in the following cases (Figure 3- 13):

i) The SOC of the accumulator is below a certain limit (e.g. 0,2).The SOC should not take values below this limit, in order to avoid that both accumulators reach the same level of pressure.
ii) If the difference of pressure between both accumulators is low, the Pump/Motor is not able of supplying the vehicle with enough power to follow the driving conditions. This is made by comparing the power demanded by the driver and the current power of the Pump/Motor.
iii) The Pump/Motor overpasses its power limits.
iv) The Pump overpasses its speed limits and the vehicle has to accelerate (Figure 3- 14 ICE

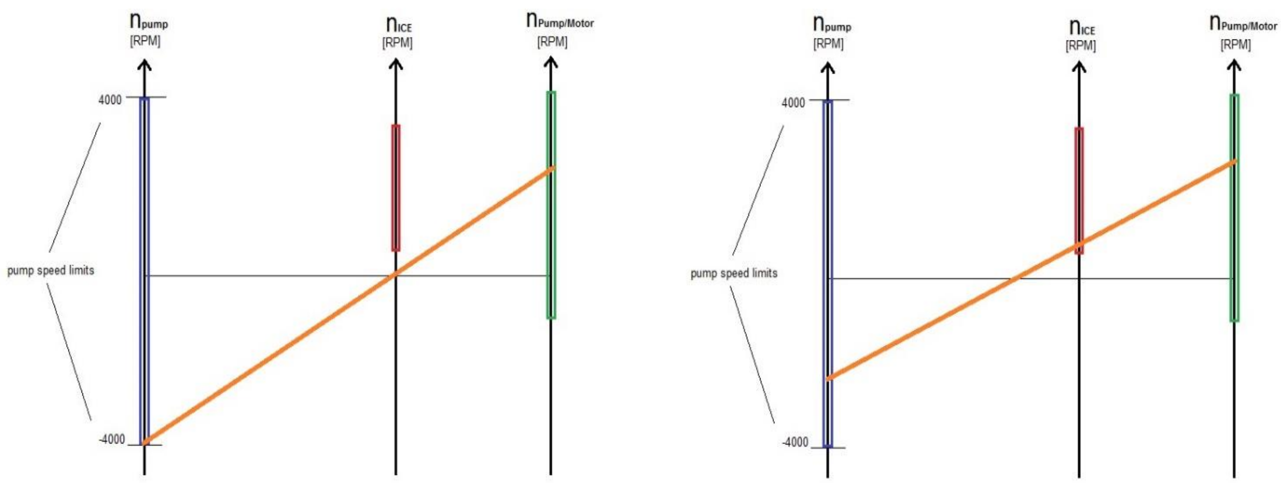

Figure 3- 14 ICE starts if the Pump has reached its speed limits

If the ICE has to be started, the Pump acts as the starter adapting its speed and torque to make it happen (see chapter 3.5.1)

In the next step, the accelerator pedal position is calculated by reading the values from a table when the power demand and speed are entered (Figure 3- 15).

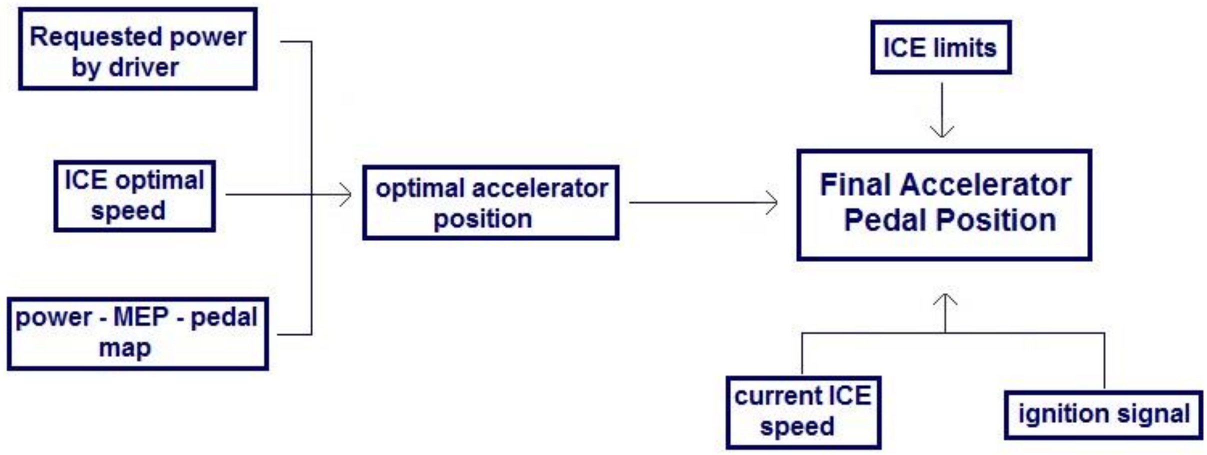

Figure 3- 15 Calculation of the accelerator pedal position

This first pedal position is corrected in case that the current ICE speed differs from the optimal or it is not within its limits.

3.5.4 Braking Control

This unit is used to calculate the brake pedal position of the friction brakes. This way it is determined how much braking power is absorbed by accumulator (regenerative braking) and the friction brakes.

The main factors to restrict the use of regenerative braking are a full accumulator (SOC>0,9) and that the braking power overpasses the power limit of the Pump/Motor (Figure 3- 16).

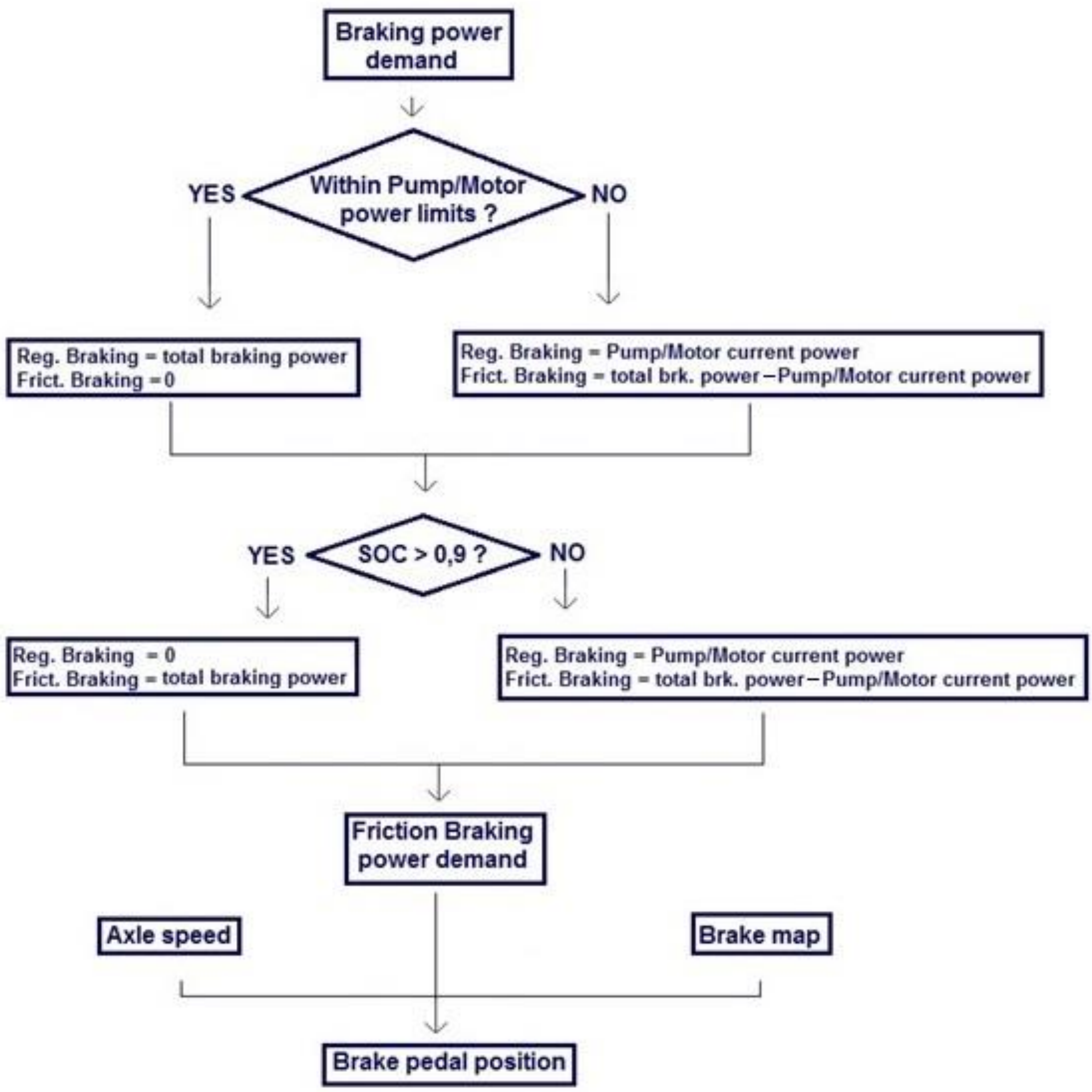

Figure 3- 16 Braking Control Unit

At the end, the demanded power to the friction brakes is transformed into a brake pedal position signal, considering the value of the axle speed and a table from which the pedal position is read.

3.5.5 Power Demand Unit

The purpose of this unit is to compute the power demanded by the driver to follow the velocity of the driving cycle (e.g. the NEDC or US06) at every moment. This is made regarding the vehicle's mass, the current of the vehicle and the target speed of the cycle.

3.6 Limitations and Simplifications of the Model

As every model, its use has to be restricted to the objective of the project and some simplifications have to be made to make the process easier and clearer.

The most important simplifications have to do with hydraulic circuit and machines.

It has already been mentioned that the efficiency maps used for the hydraulic machines correspond to machines used in a different study [23]. In the ideal case, there should be a different efficiency map for every value of the displacement, but in this model only one efficiency map is used, without regarding this aspect. Moreover, the same efficiency map is used for pumping and motoring situations.

Real hydraulic circuits include valves to regulate the flow. At first, the model incorporated these elements at each accumulator and hydraulic machine. Nevertheless, valves induce numerical instabilities which make the simulation much slower, so that they are not utilized in the model.

The absence of valves is compensated with the displacement is used, it does not flow anything if it takes values of zero.

In the area of heat exchange, the temperature of the wall of every pipe and the accumulators is fixed to 300 K (same temperature as ambient).

Moreover, the changes of the oil viscosity due to the temperature is not considered for calculation of the pump's efficiency.

Another important simplification is that the mass of the components needed for the hybridization of the power train is not taken into account. Both hydraulic machines weigh around 60 kg and the oil and accumulators around 120 kg.

4. Power management and components' optimization

Once that the model is completed, it is necessary to decide when to use exactly which machine of the hybrid power train.

Except for the ICE, which is a fixed component, there are several parameters from the components, which can be varied in order to find an optimal combination.

As the components size vary, the driving strategy has to be adapted, so that both processes have to be developed together.

4.1 Driving Strategy

The driving strategy's purpose is to decide when the ICE should be turn on and shut off, when the hydraulic machine should boost or when and how to use the regenerative braking, trying to exploit all the potential for saving fuel, which a concrete hybrid power train offers.

Nowadays, the driving strategy of hybrid vehicles can be very complex. Some systems are able to use current car-2-car information and predict the exact conditions which the vehicle will find. The on-board computer of these vehicles estimates future operating modes and the SOC changes regarding the future route [26].

Moreover, advanced optimization methods (e.g. heuristic methods or dynamic programming) can be applied to optimize the combination of variables which influence the driving strategy.

However, the driving strategy of this model was decided to be kept simple. The parameters involved directly in the driving strategy are optimized in order to save as much fuel in the NEDC (standard European cycle to test fuel consumption), while at the same time, being able to drive perfectly the US06 (cycle with high accelerations and speeds).

Another particularity of this model, is that there is not any “hydraulic boost” while the ICE is working. When the ICE is operating, the hydraulic machines only work to regulate the engine.

As previously described, the vehicle normally starts to drive on hydraulic-only mode and, then, ICE needs to be started if some of these situations happen:

i) $SOC < 0{,}26$

ii) $Power_{demand} - Power_{Pump/Motor} > 5\ kW$

iii) $n_{Pump} < -4000\ RPM$

iv) $P_{Pump/Motor} > 76\ kW$

Independently of the driving demands, condition i) expresses that the accumulators is almost empty. The value of “0,26” is chosen after trying several values close to zero. Experience with simulations supports this value, in order to avoid the problems that appear when the SOC approximates zero (numerical instabilities).

Condition ii) is totally conditioned by the driving demands. It happens when the Pump/Motor is no longer capable of propelling the vehicle on its own, e.g. during accelerations.

Conditions iii) and iv) belong to overpassing limits of the machines (lower limit of the Pump speed and power limit of the Pump/Motor).

While braking, regenerative braking should be used as often as possible to charge the accumulator and only restricted in the situations described in chapter 3.5.4 (SOC>0,9 ; P_{PMot}<-76 kW)

Moreover, regenerative braking only takes place below a vehicle´s speed of 70 km/h, in order to avoid regenerative braking at high speeds (e.g. 100 km/h). When this occurs, there is a high pressure rise in the accumulator in a very short of time, leading sometimes to overcharges of the accumulator (SOC > 1).

Figure 4- 1 plots the engine state (0=OFF; 1= ON) during the urban part of the NEDC, so that it can be seen when the HHV drives in hydraulic-only mode and when the ICE works.

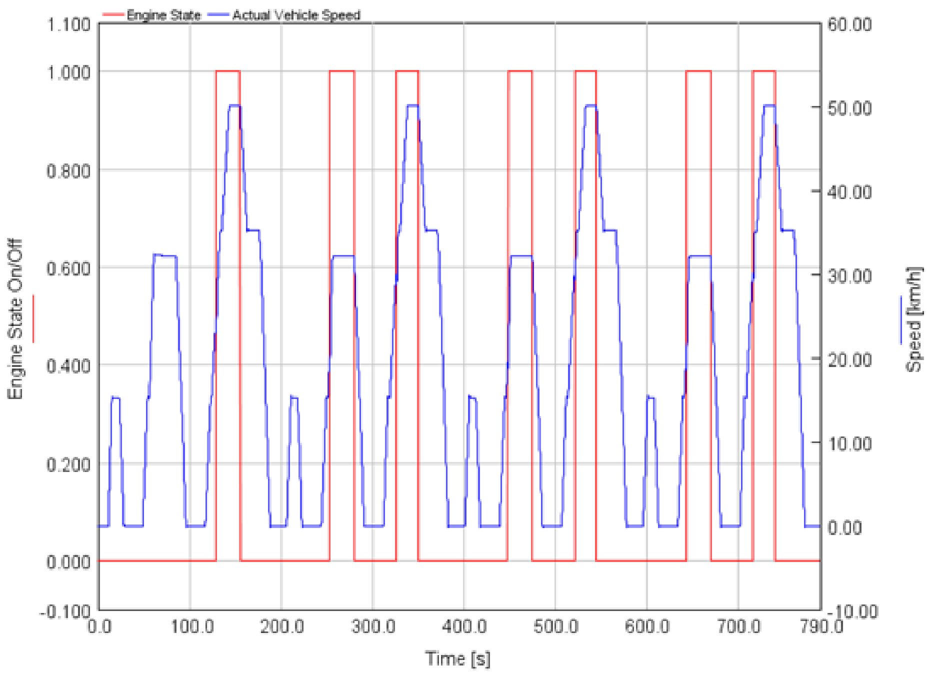

Figure 4- 1: Engine state during the urban part of the NEDC

4.2 Components sizing and optimization

As the aim of this work to compare a conventional and a hybrid power train, there are components and parameters of the model which stay unchanged during the process of optimization, such as the engine or the vehicle's dimensions.

The parts, whose parameters are varied are those involved in the hybridization of the powertrain (e.g. hydraulic machines, accumulator and parameters of the driving strategy).

In this part, no specific numerical optimization method is used, so that the combination found is just a coarse approximation to the optimum.

4.2.1 Vehicle

As described in chapter 3.3, the unit that simulates the behaviour of the vehicle, excluding the power train, is composed by several individual parts. The following tables (Tables 3 to 7) show the main parameters which characterize the chassis, tyres, ambient and road conditions.

It should be reminded, that these parameters are the same for the simulation of the HHV and of the conventional vehicle.

Table 3: Vehicle dimensions

Vehicle Mass	kg	1220
Drag coefficient "Cx"	-	0,32
Frontal Area	m²	2, 11

Table 4: Tyre characteristics

Rolling resistance factor	-	1220
Rolling radius	mm	298,2

Table 5: Ambient conditions

Ambient air temperature	K	298,15
Ambient air pressure	bar	1,01315
Wind velocity	km/h	0
Wind direction	deg	0

Table 6: Road Conditions

Road grade	%	0
Road curvature radius	m	0
Rolling resistance multiplier	-	1

Table 7: Axle moment of inertia

Axles' moment of inertia	kg-m²	1,25

The behaviour of the brakes is characterized by a table used to read the output values of the brake (Table 8) (rest of the values are interpolated).

Table 8: Brake map

		Axle speed [RPM]		
	Power [W]	**0**	**0,1**	**10000**
Pedal position [%]	**0**	0	0	0
	100	0	3000	30000

At the differential, all the parameters are kept constant, except the "final drive ratio". This parameter is varied during the optimization process, since it can directly vary the speed of the Pump/Motor (it is directly connected to the axle through the ring gear and the differential).

$$n_{Pump/Motor} = r_{final\_drive\_ratio} \cdot n_{axle} \quad (15)$$

At the end, a value of "3" is chosen, since it keeps the speed of the Pump/Motor within its limits.

This final gear ratio is also important for the ICE control, because the current speed of Pump/Motor can also influence the current limits of the ICE speed ($n^{ICE}_{current\_max/min} = \frac{1}{r_{ICE\_gear}} \cdot \left(\frac{z_{ring}}{z_{ring}+z_{sun}} \cdot n^{Pump}_{max/min} + \frac{z_{sun}}{z_{ring}+z_{sun}} \cdot n^{PMot}_{current}\right)$ (10(Equation 10 in chapter 3.5.1).

The extra gear set between the ICE and the carrier ("r_{ice_gear}") takes a value of 0,67 ($\frac{1}{r_{ICE\_gear}} = 1,5$). This value is also chosen empirically, so that the ICE is able of reaching its maximum speed limit (6000 RPM) if needed in Equation 10.

4.2.2 Internal Combustion Engine

The engine used for the simulations (of the HHV and the conventional vehicle) is a 1,4-litre-Otto engine with the parameters listed in Table 9.

Table 9: ICE and Fuel Parameters

Displacement	cm³	1364
Max. Speed	RPM	6000
Idle Speed	RPM	1150
Fuel Heating Value	MJ/kg	43
Fuel Density	m³/kg	749

The behaviour of the ICE is characterized by the mechanical output, the accelerator pedal position map and the specific fuel consumption map (Figure 4- 2), from which it is possible to build a table with the most efficient working points for a demanded power value (Table 10).

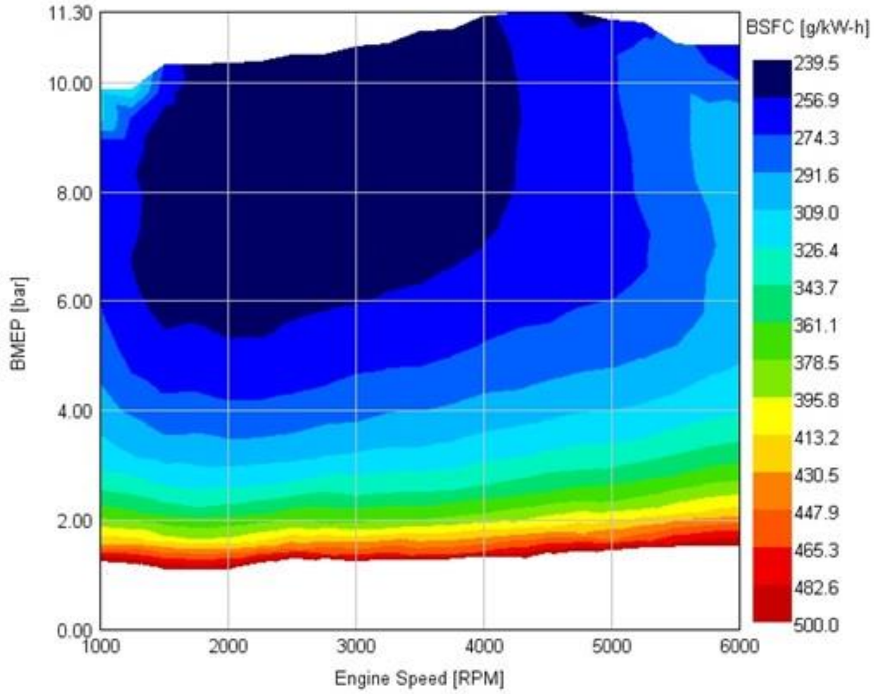

Figure 4- 2: Specific fuel consumption map

Table 10: Optimal ICE Speed

Power (W)	Opt. Speed (RPM)
0	0
1	1150
7891	1155
64238,8	6000

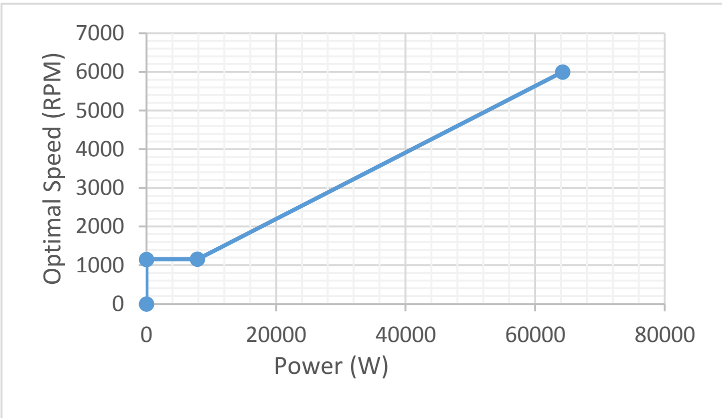

Figure 4- 3: Optimal ICE Speed

4.2.3 Hydraulic Machines

The criteria used to choose the parameters of the hydraulic machines, is to take similar values to the ones of models from the catalogue of Bosch Rexroth for axial piston variable displacement pumps [27].

Another criteria is that the nominal power should be similar to the one of the electric machines of the third generation of the Toyota Prius (2009): 60 kW [5].

Within the Bosch Rexroth catalogue, two series of pumps for closed circuits can be considered: series A4VG/32 and series A10VG (Table 11 and Table 12).

Table 11: Pump sizes of the series A4VG [28]

Size				28	40	56	71	90	125
Displacement		$V_{g\,max}$	cm³	28	40	56	71	90	125
Max. speed	at $V_{g\,max}$	n_{nom}	rpm	4250	4000	3600	3300	3050	2850
	intermittent [1)]	n_{max}	rpm	5000	5000	4500	4100	3800	3450
Flow	at n_{nom}	qv	l/min	119	160	202	234	275	356
Power	Δp = 400 bar	P	kW	79	107	134	156	183	238
Torque	Δp = 400 bar	T	Nm	178	255	357	452	573	796
Weight (approx.)		m	kg	29	31	38	50	60	80

Table 12: Pump sizes of the series A10VG [29]

Size				18	28	45	63
Displacement		V_g	cm³	18	28	46	63
Max. speed	at $V_{g\,max}$	n_{nom}	rpm	4000	3900	3300	3000
	intermittent [1)]	n_{max}	rpm	5200	4500	3800	3500
Flow	at n_{nom}	qv	l/min	72	109	152	189
Power	Δp = 300 bar	P	kW	36	55	76	95
Torque	Δp = 300 bar	T	Nm	86	134	219	301
Weight (approx.)		m	kg	14	25	27	39

The chosen machine is (at first) the A10VG size 45 for both machines (Table 13).

Table 13. Pump A10VG size 45 [28]

Displacement	cm³	45
Max. Speed	RPM	3800
Flow	L/min	152
Max. Torque	Nm	219
Max. Power	kW	76

However, the experience with simulations show that the limiting factor of this machines is not the power, but the maximal displacement. This is why a machine with a higher displacement was chosen for the Pump/Motor, being more critical for this machine, due to the responsibility of powering the vehicle on hydraulic-only mode.

The model chosen is the A10VG size 63, whose characteristic data is shown in Table 14.

.

Table 14: Pump A10VG size 63 [29]

Displacement	cm³	63
Max. Speed	RPM	3500
Flow	L/min	189
Max. Torque	Nm	301
Max. Power	kW	95

Although the theoretical power limit of these two models overcome 60 kW, as already explained, the parameter which restricts these machines is the displacement and values of 60 kW are never reached by them during the simulations.

The efficiency map used belongs to machines used for another project (Figure 4- 4) [23]. The reason for this is the lack of such data by OEMs, such as Bosch Rexroth or Eaton.

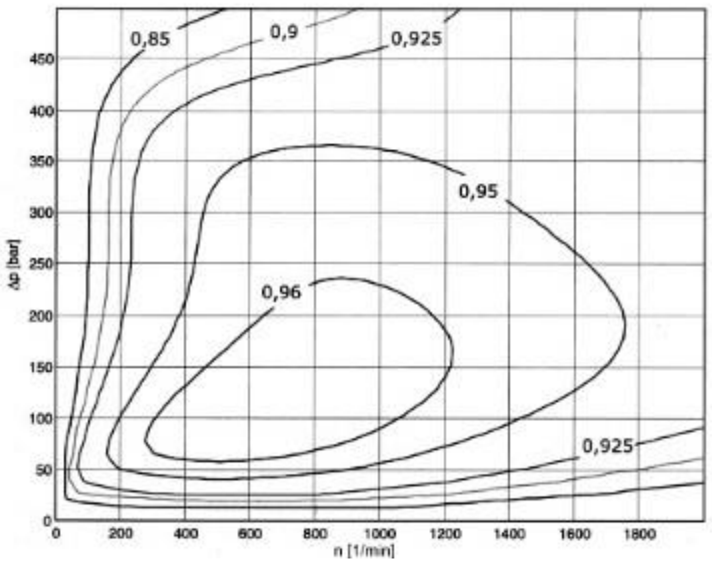

Figure 4- 4: Pump and Pump/Motor efficiency map [23]

The volumetric efficiency is set to a constant value of 0, 95 for similar reasons.

4.2.4 Hydro-pneumatic Accumulator

As with pumps, the hydropneumatic accumulator is chosen from a catalogue of Bosch Rexroth [30].

Bosch Rexroth offers one model of accumulator, which could fulfil the demands of the vehicle: the bladder accumulator HAB-4X. The size of this accumulator varies from 1 to 50 litres (Table 15).

Table 15: Bladder-type accumulator catalogue of Bosch Rexroth [31]

Hydraulic										
Nominal volume	V_{nom}	l	1	2.5	4	6	10	20	35	50
Effective gas volume	V_{eff}	l	1.0	2.4	3.7	5.9	9.2	18.1	33.4	48.7
Maximum permissible flow	q_{max}	l/min	240	600	600	600	900	900	900	900
Maximum permissible operating pressure	p_{max}	bar	350	350	350	350	330	330	330	330
Maximum permissible pressure fluctuation range	Δp_{dyn}	bar	200	200	200	200	200	200	200	200
Operating pressure and effective volume			See calculation on pages 6 to 9							
Hydraulic fluid			Hydraulic fluid according to DIN 51524; other fluids on request!							
Hydraulic fluids temperature range (others on request)			−15 °C to +80 °C (NBR) −35 °C to +80 °C (ECO)							
Pneumatic										
Filling gas			Nitrogen, cleanliness level 4.0, N_2 = 99.99% by volume							
Gas filling pressure	p_0		< 2 bar							

The estimation of the size of the accumulators is made regarding the space they would take within an existing passenger vehicle, without having to reduce much the space of the trunk or for the passengers, and without changing much the wheelbase or the track.

The thought behind this is not to increase the cost the HHV by having to design a full new vehicle. As a reference, it is taken Figure 4- 5, which shows the placing of the high- and low-pressure accumulators (blue) in the Hybrid Air by PSA.

Figure 4- 5: Accumulator placing in the 2013 prototype of the Hybrid Air [16]

Taking as an example the Citroën C3 from 2013, with a wheelbase of 2466 mm and a track of 1466 mm [32], the selected size for the accumulator is 20 litres (Table 16).

Table 16: Accumulator HAB-4X size 20

Nominal Volume	L	20
Effective Volume	L	18,1
Max. Flow	L/min	900
Max. Pressure	bar	350
Pressure fluctuating range	bar	200

The initial conditions of each accumulator at every cycle are summarized in Table 17 and Table 18.

Table 17: Initial Conditions of the High-Pressure Accumulator

Initial Oil Volume	L	13,195
Initial Pressure	bar	340
Initial SOC	-	0,97

Table 18: Initial Conditions of the Low-Pressure Accumulator

Initial Oil Volume	L	4,7125
Initial Pressure	bar	4

4.2.5 Pipes

The pipes take the same values in all simulations. The diameter is 20 mm, the length 200 mm and the material was defined as smooth .The discretization length (“dx“), necessary for the integration of the flow differential equations, is 20 mm.

4.3 Conventional Vehicle

The fuel consumptions of the HHV in several tests described in chapter 5 are compared with the one of a vehicle without any hybridazation.

This vehicle counts with the same ICE, vehicle dimensions (mass, drag coefficient, frontal area), tyres, brakes and drives under the same ambient conditions as the HHV (see chapter 4.2).

The main difference relies on the use of a gearbox connecting directly the crankshaft of the ICE with the differential.

4.3.1 Gearbox of the conventional vehicle

The gearbox used in the model consists of five forward gears (the reverse gear is not used) with the gear ratios indicated in Table 19.

Table 19: Gear Ratios

Gear	**1**	**2**	**3**	**4**	**5**
Ratio	3.538	2.125	1.360	1.029	0.720

The mechanical efficiency of the gearbox depends on the gear number, temeperature, speed (RPM) and torque, so that effiency maps are bulit with these parameters. In general, the efficiency values vary between 0,91 and 0,97.

In the driving tests, the conventional vehicle is provided with a shifting strategy, which is different for upshifting and downshifting. Table 20 and Table 21 show the point at which the shifting takes place for upshifting and downshifting respectively.

For each gear number, depending on the accelerator pedal position (from 0% to 100%), the gear is shifted (up or downwards), once the given ICE speed number is reached.

Table 20: Upshifting Point

		Gear Number				
	ICE Speed [RPM]	**1**	**2**	**3**	**4**	**5**
Accel. Pedal Position [%]	**20**	2200	2000	2000	2000	2000
	100	3500	3800	4000	4200	4400

Table 21: Downshifting Point

		Gear Number				
	ICE Speed [RPM]	**1**	**2**	**3**	**4**	**5**
Accel. Pedal Position [%]	**20**	0	1200	1200	1200	1200
	100	0	2800	2800	2800	2800

Note that it is only downshifted from the gear 1, when the ICE stops.

5. Tests

Several tests are carried out with the purpose of characterizing the HHV, such as driving cycles (NEDC and US06), or tests where the range or the maximal acceleration which can be performed with hydraulic power are proved.

5.1 Fuel Consumption

Fuel consumption (FC) is calculated with the help of the engine BSFC-map, the travelled distance and data about the fuel such as its density or heating value.

5.1.1 New European Driving Cycle

The New European Driving Cycle (NEDC) is used in the European Union to test and quantify the fuel economy of vehicles [31].

It consist of an urban cycle, which is made up of four repeated elementary urban cycles; and an extra-urban cycle [32] (Figure 5- 1).

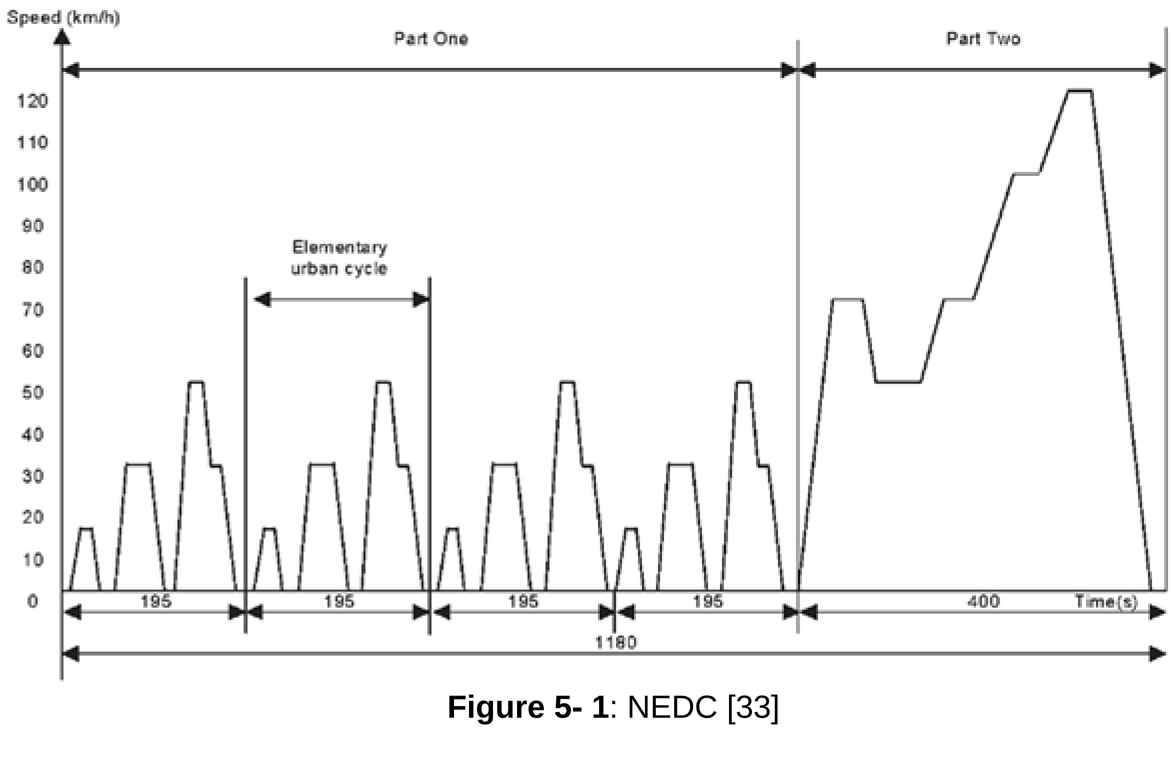

Figure 5- 1: NEDC [33]

5.1.2 US06

The US06 Supplemental Federal Test Procedure (SFTP) (Figure 5- 2) is a test designed to complete the weaknesses of the EPA Federal Test Procedure "FTP-75" in the representation of aggressive, high speed and/or high acceleration driving behaviour, rapid speed fluctuations, and driving behaviour following start-up [34]. For the same purpose is it used here: to check the behaviour of the HHV at high accelerations and high speeds.

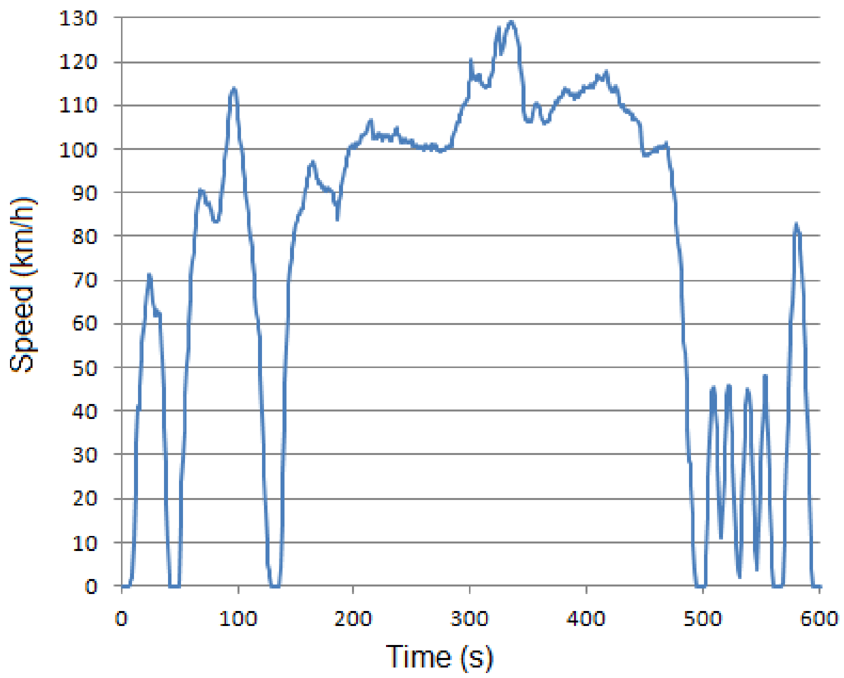

Figure 5- 2: US06 [35]

5.1.3 Constant Speed Driving

In order to see the fuel consumption and how the power supplied by the ICE is split at the PGS at different speeds, tests are carried out where the vehicle has to drive at a constant speed.

In power-split HEVs, such as the Toyota Prius, there is a known situation that comes out when the equivalent machine of the Pump starts to work as a motor and the equivalent of the Pump/Motor as a generator (machines work "inversely"), so that mechanical power from the ICE has to flow over the hydraulic circuit into the final gear (Figure 5- 3). This elevates amount of power flowing through the hydraulic circuit and decreases the overall efficiency of the power train.

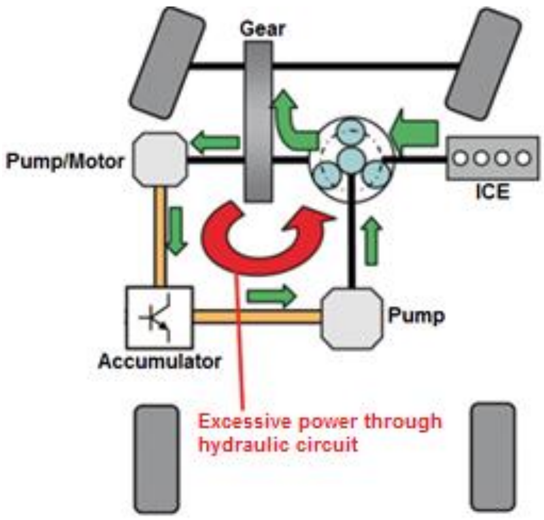

Figure 5- 3: Power flow when hydraulic machines work inversely [5]

The way to avoid that the hydraulic machines work inversely is to prevent the speed of the Pump becoming negative when the ICE is working.

The problem of this restriction is that the Pump is no longer capable of setting the ICE into its most efficient point. This introduces a new parameter to optimize: the lower limit of the Pump speed. The optimal limit has to combine the advantages of an efficient working point of the ICE and the disadvantages of high absorption of power by the hydraulic circuit.

The power absorbed by the hydraulic circuit is the one that comes out of the ICE, but does not arrive at the driveshaft.

In the tests, fuel consumption is only measured when the vehicle has already reached the target speed, so that there is not any acceleration more (Figure 5- 4).

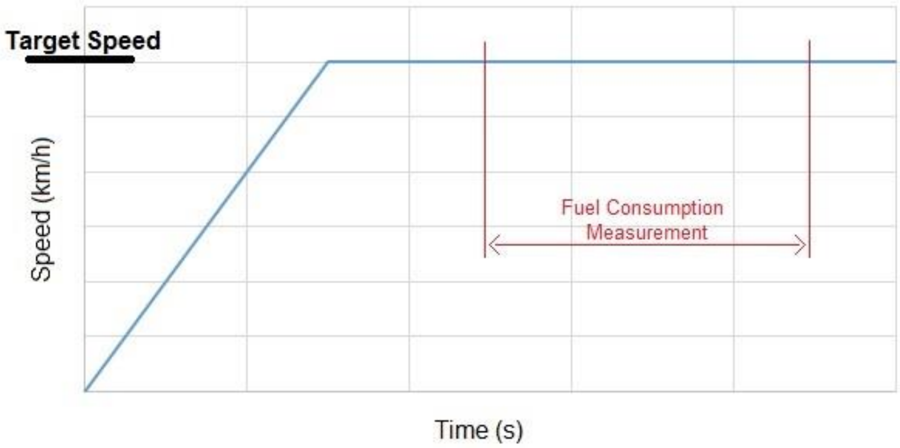

Figure 5- 4: Fuel Consumption at Constant Speed

Apart from varying the lower limit of the Pump speed, total efficiencies of both hydraulic machines are set as constant and increased to observe how this influences the fuel consumption.

5.2 Hydraulic Range

"Hydraulic range" has been defined as the distance which the HHV is capable of driving without turning the ICE on (own definition).

Since the hydraulic range depends on the driving profile, tests are made for a constant speed (10, 20, 30, 50, 70 and 100 km/h) until the Pump/Motor is no longer capable of powering the HHV, being the distance travelled at each speed the hydraulic range.

The initial speed of the vehicle is already the testing speed, so that the vehicle does not have to accelerate.

Moreover, the volume of the accumulators is varied during for this test to verify how it affects the hydraulic range.

5.3 Maximal hydraulic acceleration

These tests are made to check the maximal acceleration that HHV is able to do without switching the ICE on, and they are defined as the one needed to reach a certain speed in 10 seconds (Figure 5- 5).

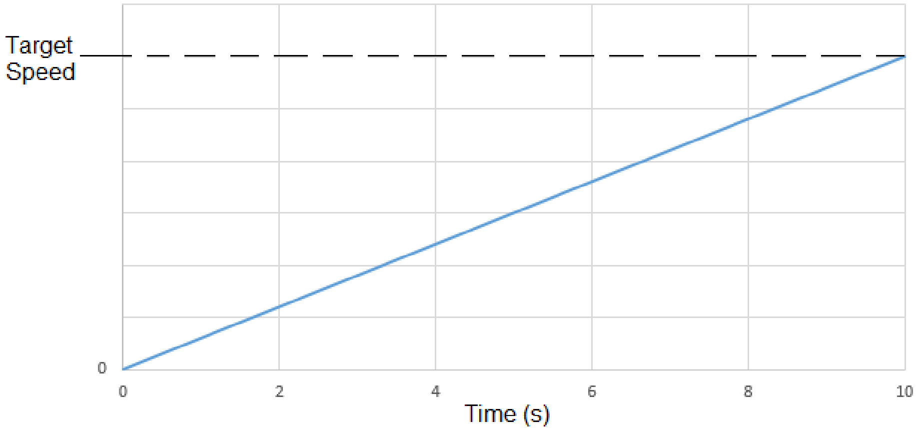

Figure 5- 5: Speed profile for the maximal hydraulic acceleration test

Several target speeds (30, 50, 70, 80, 100 and 130 km/h) are tested and observed if the HHV is able to follow the speed for at least two seconds (own criteria).

6. Results and Discussion

6.1 Fuel Consumption

At the NEDC and the US06 the fuel consumptions of three vehicles are compared: the HHV, the conventional vehicle and the conventional vehicle with an engine start-stop function.

Moreover, the evolution of the FC (in litres) of each vehicle during the cycles is depicted in diagrams.

6.1.1 Results of the Constant Speed Driving Tests. Table 22 shows a comparison of the fuel consumption of the conventional vehicle and the HHV with the original Pump speed limit (-4000 RPM) when driving at different constant speeds.

Table 22: Fuel Consumption at Constant Speed Driving (limit: -4000 RPM)

		HHV			Conventional Vehicle			
		ICE Speed [rpm]	ICE PME [bar]	Fuel Consumption [L/100km]	ICE Speed [rpm]	ICE PME [bar]	Fuel Consumption [L/100km]	Fuel Cons. Difference (%)
Speed (km/h)	**30**	1005	1.01	3.27	1502	1.22	3.77	**-13.26**
	50	1006	2.85	2.86	1907	1.73	3.46	**-17.34**
	70	1005	5.98	3.85	1852	3.85	3.46	**11.27**
	100	1580	9.52	5.21	2810	4.71	4.67	**11.56**
	130	2803	9.79	7.08	3400	6.91	6.46	**9.6**

Figure 6- 1 and Figure 6- 2 plot the operating points and the specific fuel consumption of the ICE of the HHV (original Pumps speed limit) and the conventional vehicle during this test.

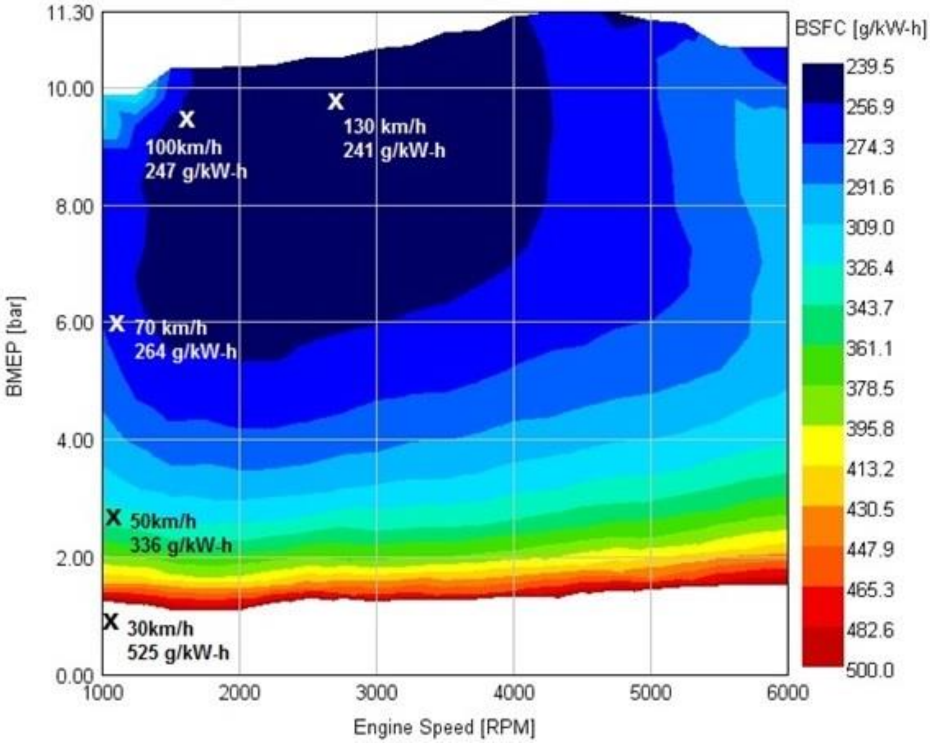

Figure 6- 1: HHV ICE working points at constant driving

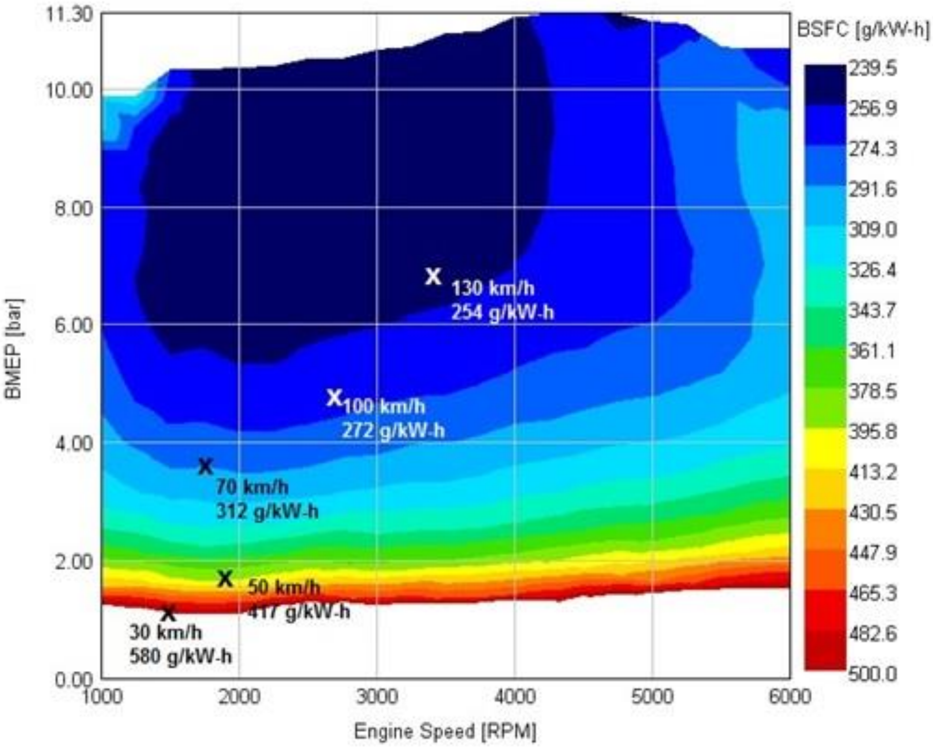

Figure 6- 2: Conventional vehicle’s working points at constant driving (limit -4000 RPM)

Influence of the total efficiency of the hydraulic machines

The results of the variation of the total efficiency (constant values during each test) of both hydraulic machines are displayed in Figure 6- 3 (Pump speed limit is -4000 RPM).

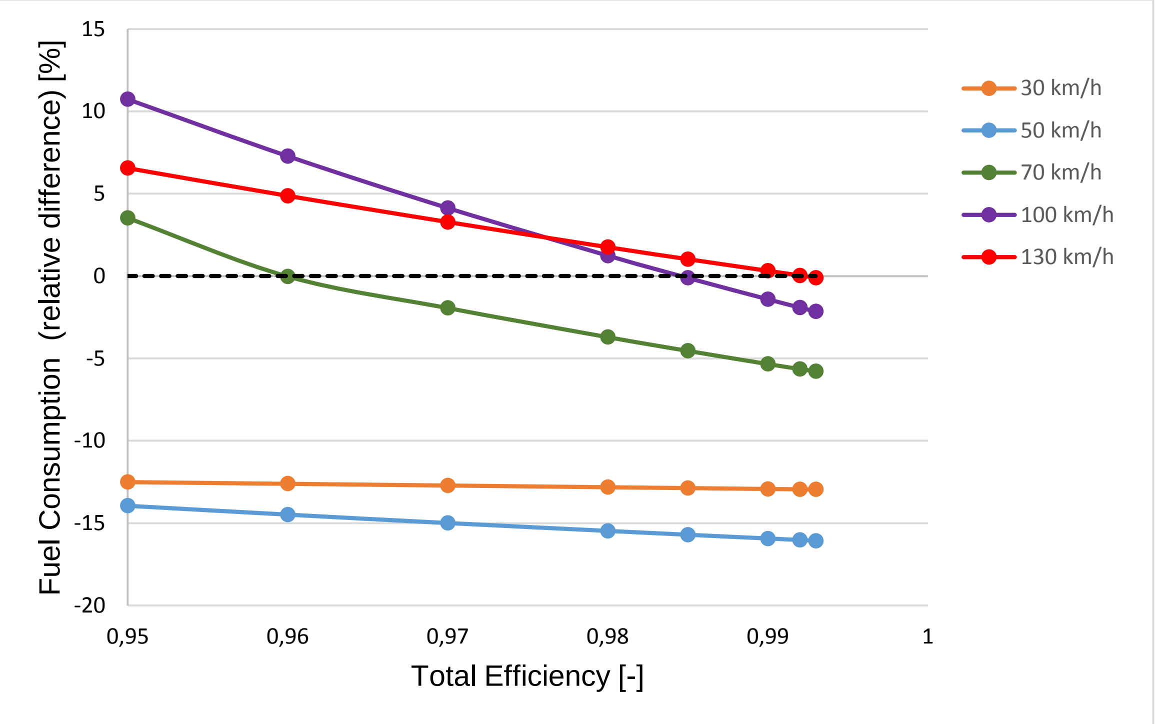

Figure 6- 3: Reduction of Fuel Consumption with the total efficiency

Influence of the Pump speed limit

The amount of ICE power absorbed by the hydraulic circuit when the Pump speed limit is -4000 RPM can be checked in Table 23.

The power of the hydraulic machine is defined as positive when it works as a pump, and negative when it operates as motor.

Table 23: ICE power-split at the PGS (limit -4000 RPM)

		Power [kW]				ICE Power absorbed by hydr. circuit [%]
		ICE	PMot	Pump	Mech. Output	
Speed [km/h]	30	1,41	-0,20	0,20	1,34	**4,77**
	50	3,42	1,69	-1,38	2,92	**14,72**
	70	7,88	9,28	-7,56	5,55	**29,59**
	100	17,49	19,86	-16,07	12,36	**29,33**
	130	30,26	20,84	-16,35	23,90	**21,00**

Motoring
Pumping

The value for the lower Pump speed limit is varied from +100 RPM (positive speed avoids machines working inversely) to -4000 RPM, which is the limit of the original Pump.

The fuel consumption differences compared to the conventional vehicle for every constant speed are showed in Figure 6- 4.

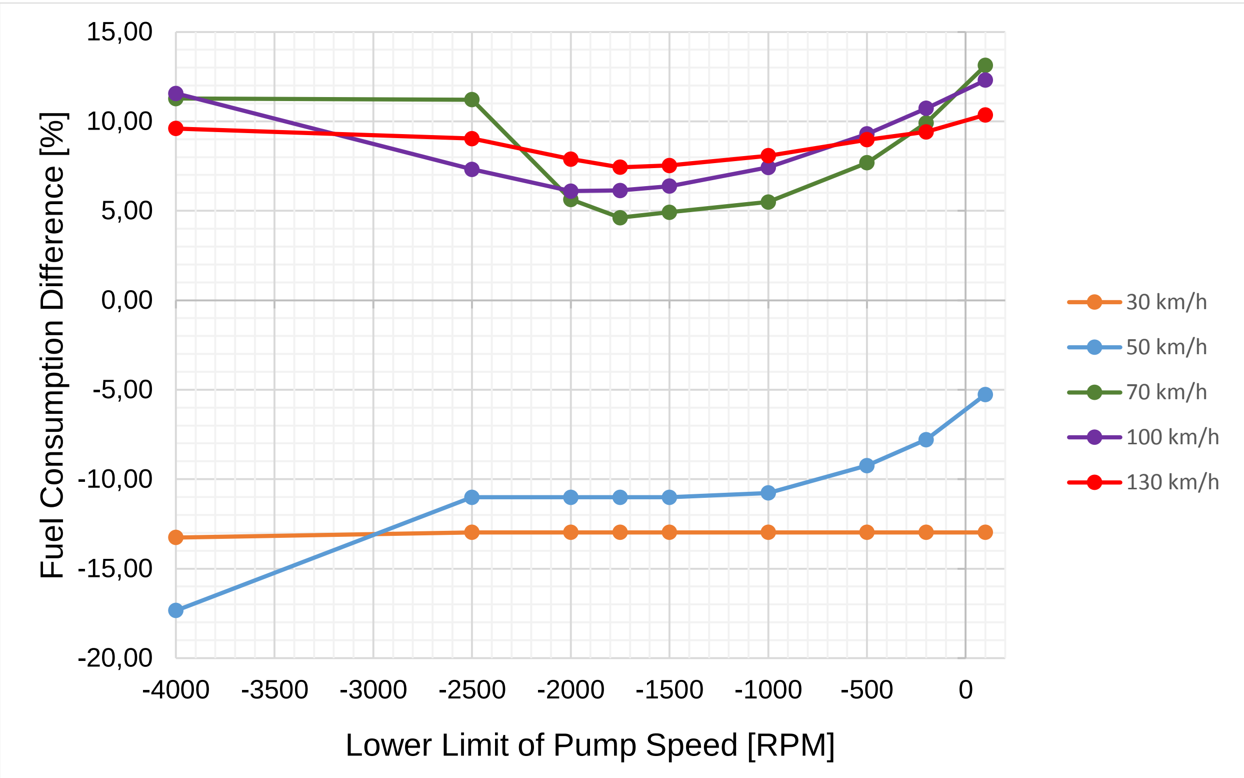

Figure 6- 4: FC-difference at different speeds vs. lower limit for the speed of the Pump

The optimum lower limit for the Pump speed corresponds approximately to a value of -1750 RPM (although at high speeds the HHV still consumes more than the conventional vehicle).

The exact results of the fuel consumption with this limit is shown in Table 24 and the power split at the PGS in

Table 25. The ICE operating points at the different speeds can be seen in Figure 6- 5.

Table 24: Fuel Consumption at Constant Speed Driving (limit –1750 RPM)

		Fuel Consumption [L/100km]		
		HHV	Conv. Vehicle	Difference (%)
Speed (km/h)	**30**	3.28	3.77	**-12.97**
	50	3.08	3.46	**-11.01**
	70	3.61	3.46	**4.61**
	100	4.96	4.67	**6.14**
	130	6.94	6.46	**7.43**

Table 25: ICE power-split at the PGS (limit -1750 RPM)

		Power [kW]				
		ICE	PMot	Pump	Mech. Output	ICE Power absorbed by hydro. [%]
Speed [km/h]	**30**	1.41	-0.20	0.20	1.34	**4.77**
	50	3.42	1.69	-1.38	2.92	**14.69**
	70	6.66	3.89	-3.17	5.55	**16.70**
	100	14.04	5.06	-4.10	12.36	**12.00**
	130	26.81	7.15	-5.62	23.97	**10.57**

Motoring
Pumping

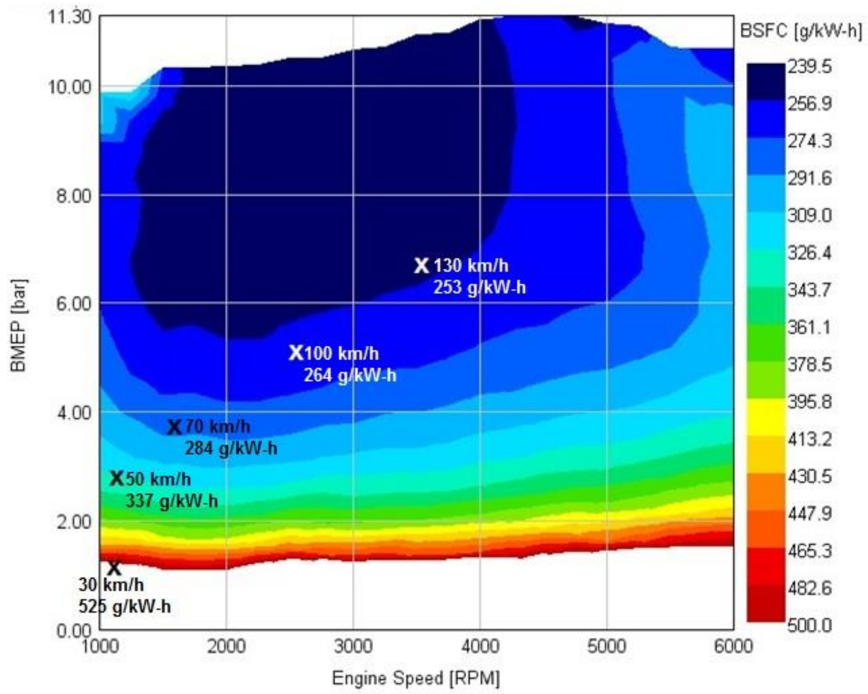

Figure 6- 5: HHV's ICE working points at constant driving (limit -1750 RPM)

6.1.2 Discussion of the Results of the Constant Speed Driving Tests

The results of the constant speed driving test show that for high speeds (70, 100 and 130 km/h) the fuel consumption of the HHV is higher than the one of the conventional vehicle, even though the operating points are more efficient (Figure 6- 1 and Figure 6- 2).

The HHV does not achieve the same fuel consumption as the conventional vehicle when driving at 130 km/h until the efficiency of both hydraulic pumps rises to 0,993. This is a very high and unrealistic value, which leads to say that the efficiency is not the reason for an elevated fuel consumption at high speeds.

In the analysis of the ICE power split at the PGS, it can be observed that only when driving at 30 km/h the hydraulic machines do not work inversely. At the rest of the speeds, the Pump is motoring and the Pump/Motor is pumping (Table 23). For these speeds the power absorbed by the hydraulic circuit increases significantly.

This suggest that the reason for the high fuel consumption is that the machines work inversely. At 50 km/h a better operating point of the ICE compensates the power loss at the hydraulic circuit. This does not happen when driving at 70, 100 and 130 km/h.

If negative speeds of the Pumps are avoided (lower limit of +100 RPM), the results are worse, due to the incapacity of setting efficient operating points of the ICE.

The lower limit of the Pump speed which combines best the advantages of a good operating point and the disadvantages of the high losses in the hydraulic circuit is -1750 RPM, although the HHV still consumes more at high speeds.

At next, the results of the fuel consumptions during the NEDC and US06 are shown for the model with the optimal limit of the Pump speed (-1750 RPM).

6.1.3 Results of the NEDC Test

Table 26 and Figure 6- 6 depict the fuel consumption values of the three vehicles during the NEDC. The HHV improves the fuel economy by 37,07 percent if the conventional vehicle has the start-stop function, and by 50,13 percent if it does not have it.

Table 26: Fuel consumption in the NEDC

	HHV	Conv. Vehicle Start-Stop	Conventional Vehicle
Fuel Consumption [L/100km]	3.75	5,14	5,63

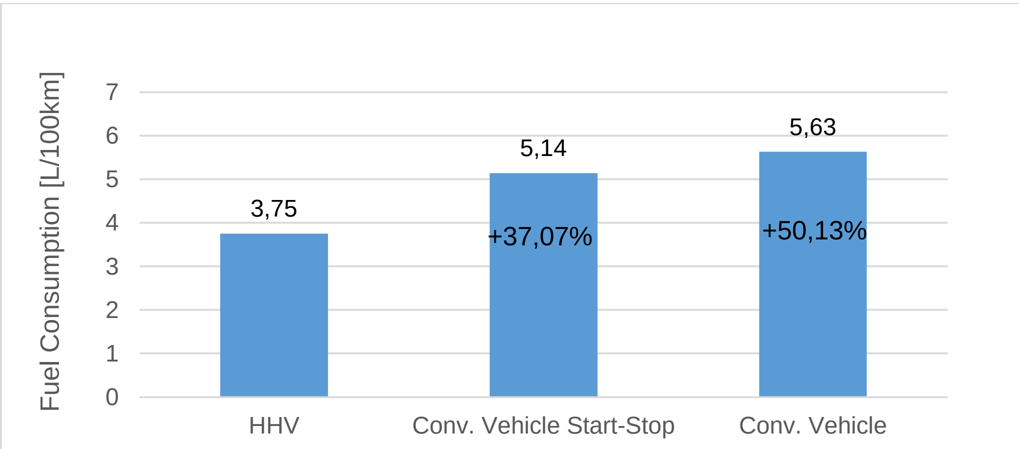

Figure 6- 6: Fuel Consumption in the NEDC

Figure 6- 7 shows the evolution of the fuel consumption (in litres) of the three vehicles during the NEDC.

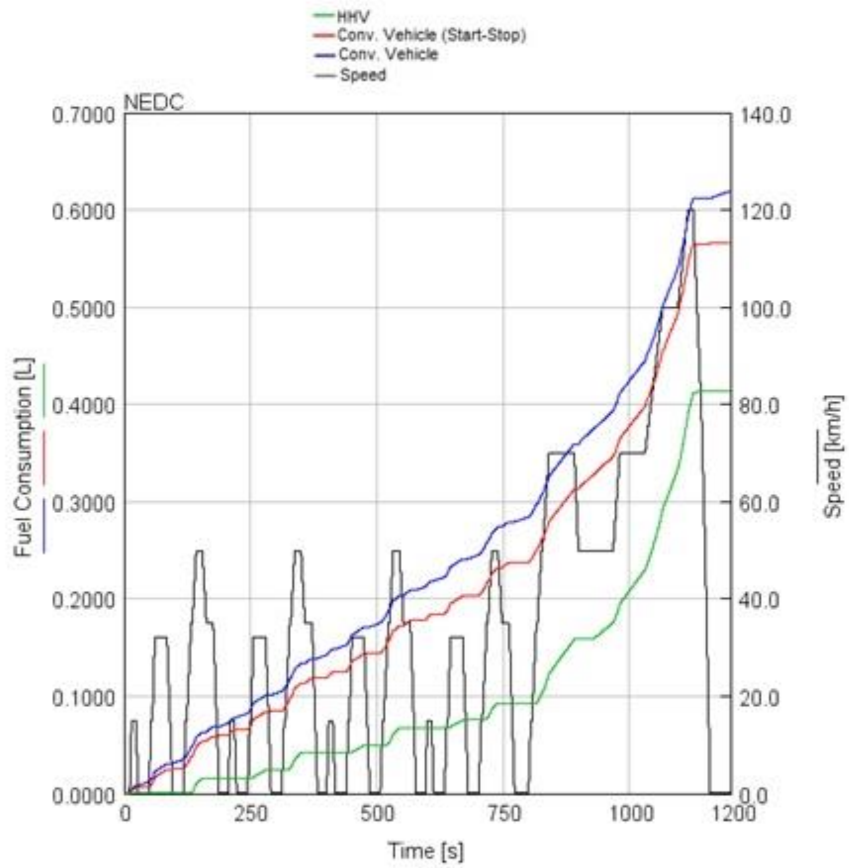

Figure 6- 7: Evolution of the fuel consumption in the NEDC

Figure 6- 8 displays the time distribution of the HHV's ICE operating points (in %) in the NEDC.

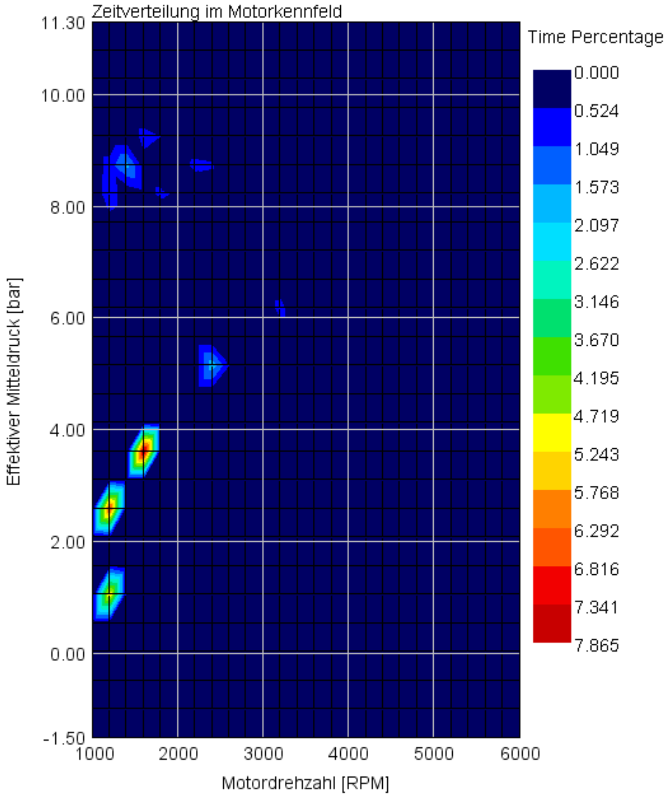

Figure 6- 8: Time distribution of the HHV's ICE operating points in the NEDC

Figure 6- 9 plots the evolution of the SOC during the cycle.

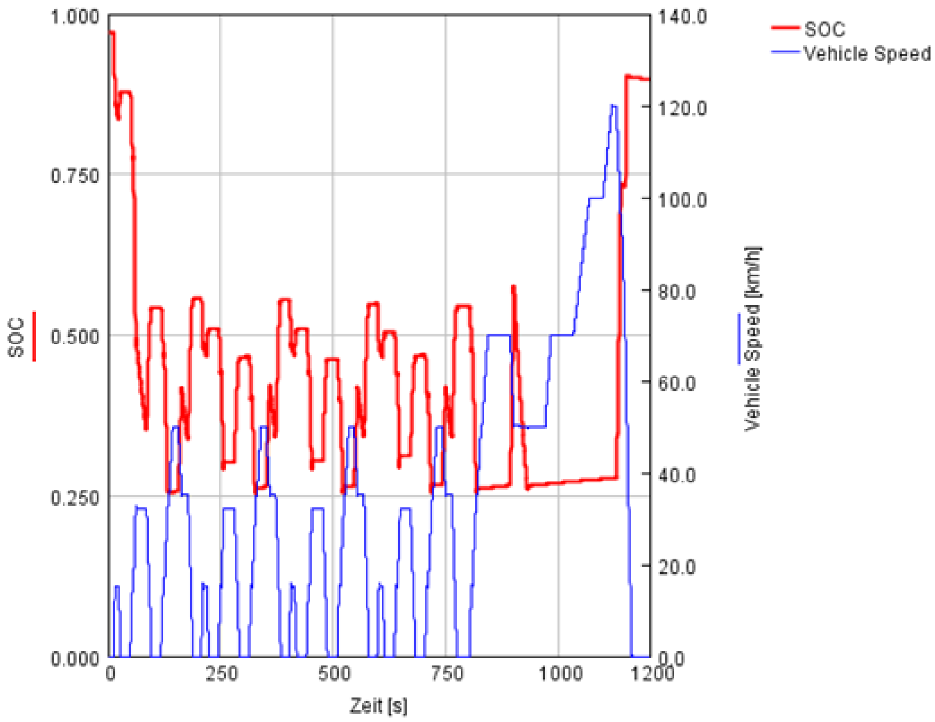

Figure 6- 9: SOC during the NEDC

6.1.4 Results of the US06 Driving Test

The results of the calculation of the fuel consumption in the US06 cycle are shown Table 27 in and Figure 6- 10.

Table 27: Fuel consumption in the US06 cycle

	HHV	Conv. Vehicle Start-Stop	Conventional Vehicle
Fuel Consumption [L/100km]	6.45	6,66	6,67

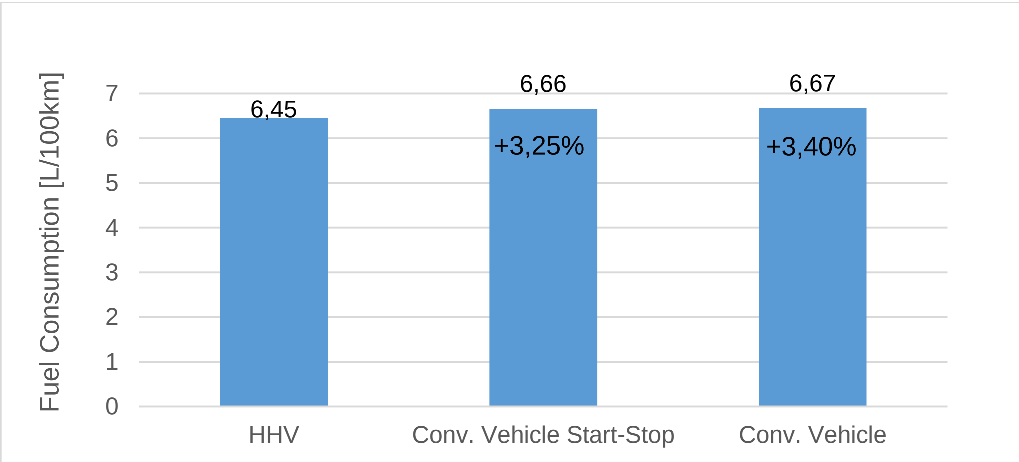

Figure 6- 10: Fuel Consumption in the US06

As in the NEDC, Figure 6- 11, Figure 6- 12 and Figure 6- 13 present respectively the evolution of the fuel consumption, the time distribution of the ICE operating points and the evolution of the SOC during the US06 cycle .

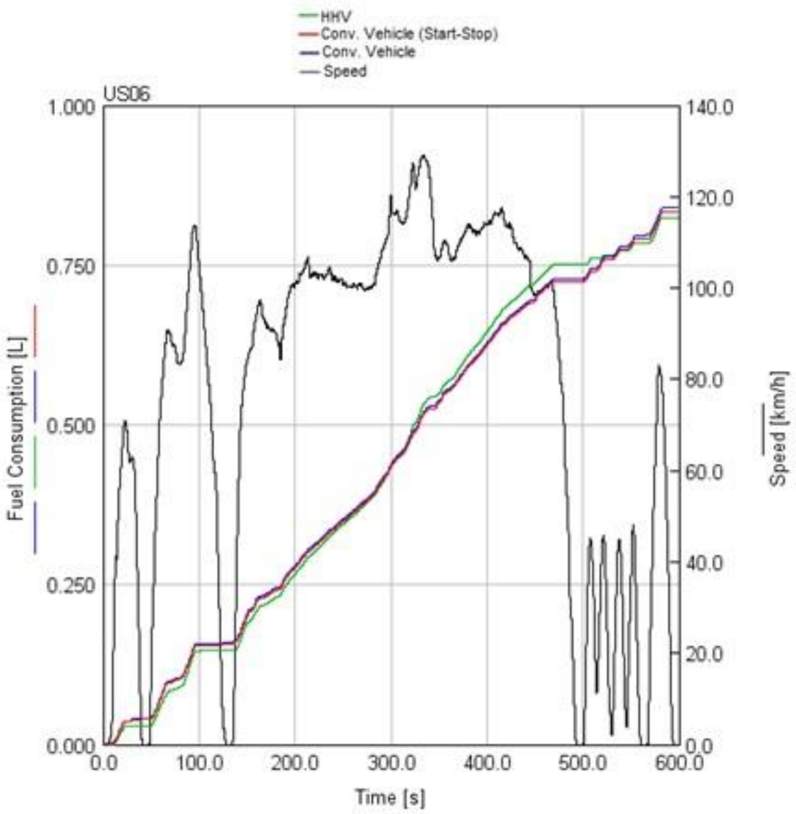

Figure 6- 11: Evolution of the fuel consumption in the US06 with a limit of -1750 RPM

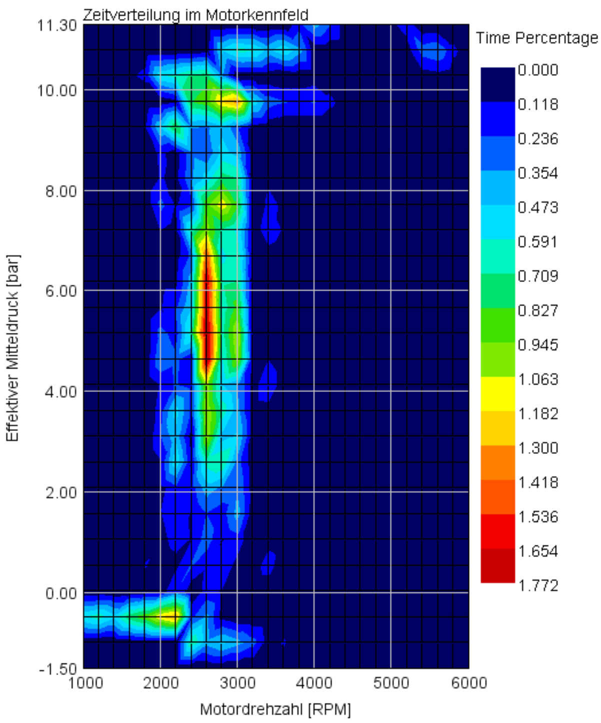

Figure 6- 12: Time distribution of the HHV's ICE operating points in the US06

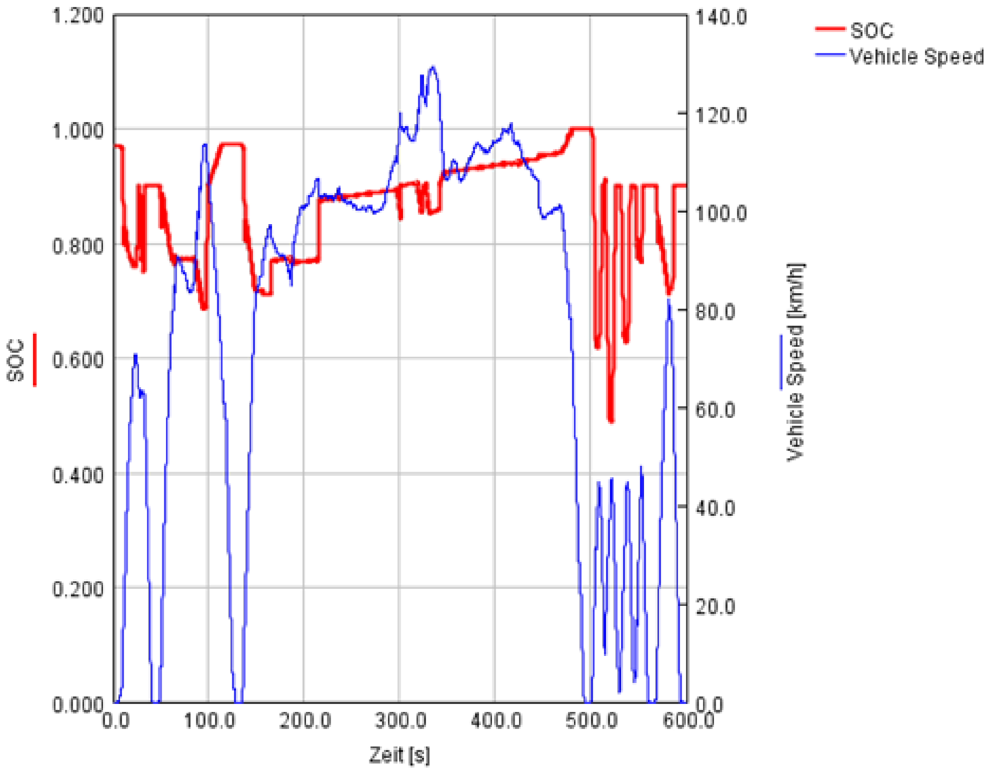

Figure 6- 13: SOC evolution during the US06

Figure 6- 14 is a comparison of the fuel consumptions at each cycle with the original and the optimal lower limit of the Pump speed.

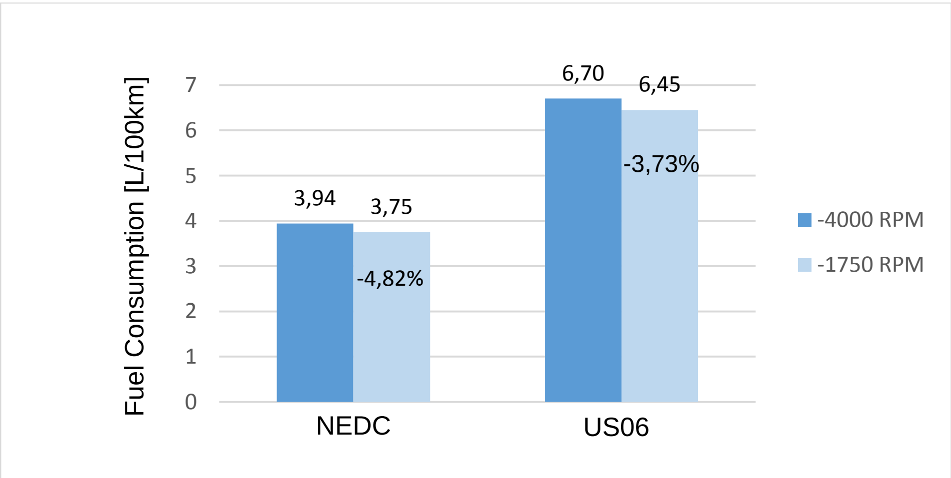

Figure 6- 14: Fuel consumption improvement with the optimal limit

6.1.5 Discussion of the Results of the NEDC and US06 Tests

Although the US06 is not used to assess the fuel economy, the results show that there is no big difference between the vehicles, mainly because there are not as many start-stops and barking events as in the NEDC, where the HHV saves fuel most.

However, Figure 6- 11 shows how the fuel consumption of the HHV (green line) gets separated from the other ones in the part where it is driven at high speeds and there is no braking. The reason is that machines work inversely in this part.

Nevertheless if the evolution of the fuel consumption in the US06 with the original limit (-4000 RPM) is compared with the one of the optimal one (-1750 RPM), it can be seen that the separation at high speeds is much higher in the original model (Figure 6- 15). This explains the enhancement of the fuel consumption in the whole cycle.

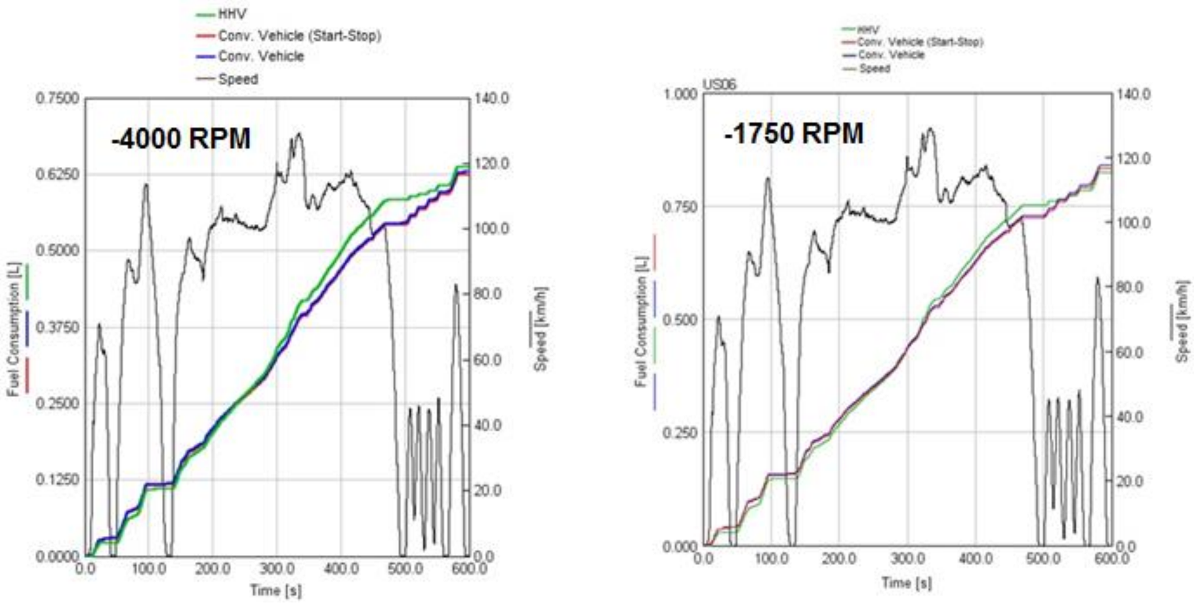

Figure 6- 15: Comparison of FC evolution in the US06 with different Pump speed limits

This high fuel consumption is caused by the inefficient operating points during the US06 shown in Figure 6- 12.

A further step into a better solution could be to include in the power management the calculation of a different Pump speed limit for every demanded situation at every moment.

This would not solve the problem of the hydraulic machines working inversely. The solution to this problem is to find an optimal combination for the teeth number of the planetary gear set.

For example, if the teeth number of the ring gear is reduced, a negative speed of the pump could be avoided (Figure 6- 16).

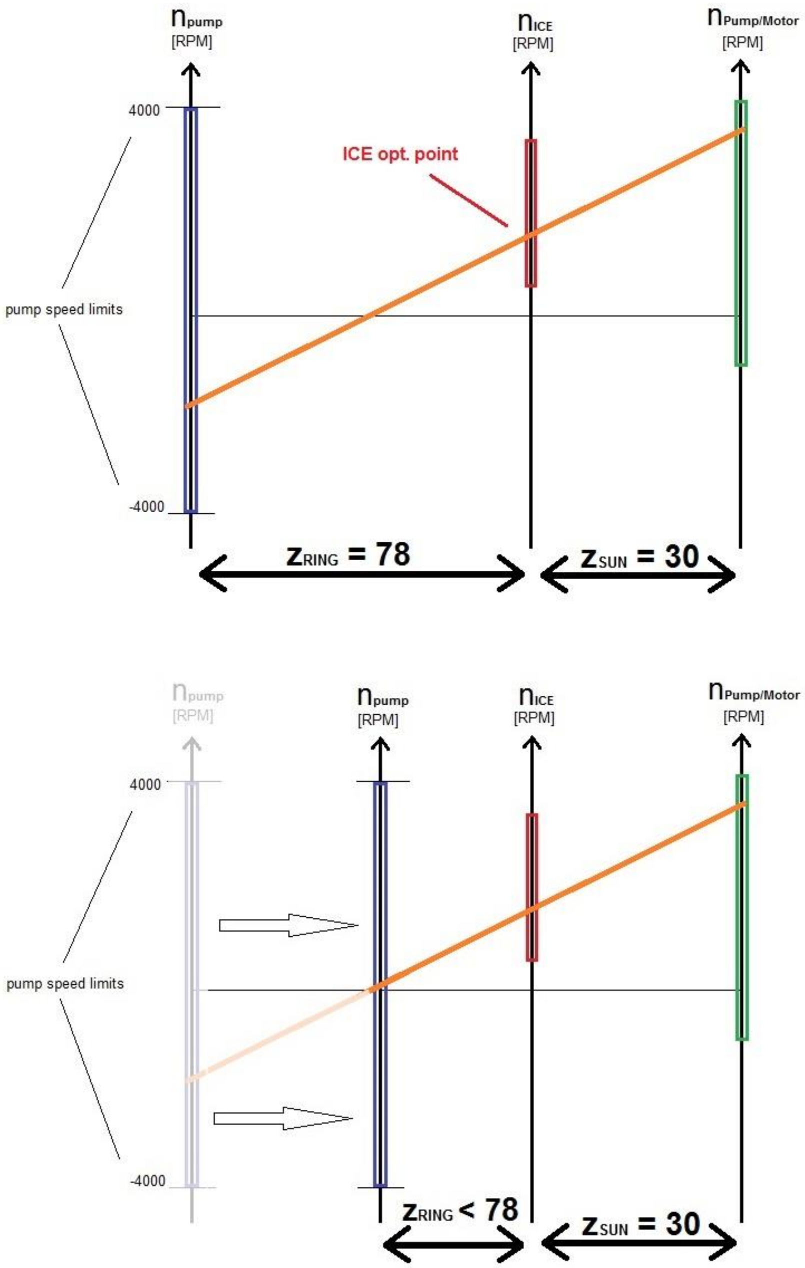

Figure 6- 16: Linear combination of speeds with original and reduced number of ring teeth

A similar solution would be to increase the number of the teeth of the sun gear.

Nevertheless, the search of an optimal combination of the number of teeth of the ring and sun gears is not simple. This changes affect other processes, such as the one described by Equation 10, used to calculate the current ICE speed limit (chapter 3.5.1).

This would mean that under certain conditions, the ICE would not be able of reaching high speeds, which would be the optimal ones at that moment, losing efficiency and performance.

For each Pump/Motor speed, which is proportional to the vehicle speed, and each ICE (optimal) speed, which depends on the demanded power, there is a gear teeth combination that avoids the Pump speed getting negative.

Moreover, for a different teeth number, there is, probably, a different model for the Pump and Pump/Motor with an optimal size.

The number of possibilities is quite large, so that the optimal combination would have to be calculated with numerical optimization methods.

6.2 Results of the Hydraulic Range Tests

Table 28 shows the distance which the HHV is capable of driving on hydraulic-only mode.

Table 28: Hydraulic Range

Speed [km/h]	Range [km]
10	1.19
20	1.11
30	0.99
50	0.72
70	0.48
100	0.26

To see how the hydraulic range increases if the volume of the accumulators (high- and low-pressure) is augmented, tests are made with volumes until 50 litres (Figure 6- 17).

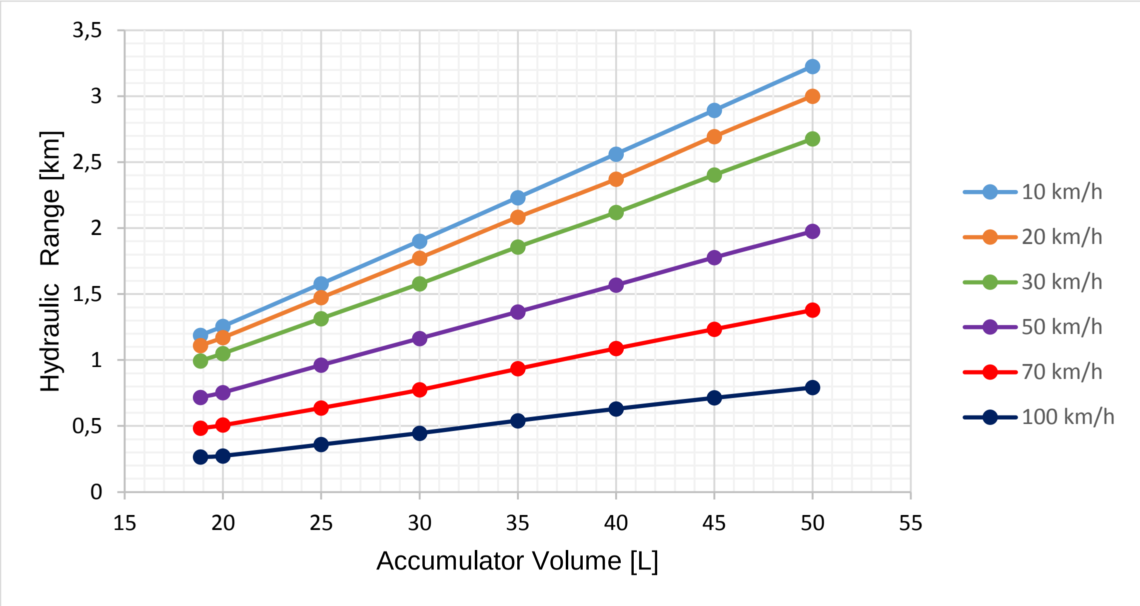

Figure 6- 17: Variation of the Hydraulic Range with the Accumulator Volume

In the NEDC, the HHV is able to drive 141 metres before starting the ICE with the original accumulator volume (18,2 litres).

6.2.1 Discussion of the Results of the Hydraulic Range Tests

The test of the hydraulic range demonstrates that it decreases when speeds are raised and that the HHV is not meant to drive at high speeds in hydraulic-only mode, since it only achieves 260 metres when driving at 100 km/h.

Figure 6- 17 points out the linearity between the range and the accumulator volume, although two accumulators of 50 litres each would be nonsense in a passenger car.

6.3 Results of the Hydraulic Acceleration Tests

Figure 6- 18, Figure 6- 19, Figure 6- 20, Figure 6- 21, Figure 6- 22 and Figure 6- 23 show the tested accelerations by comparing the target speed (red) and what the HHV is capable of, if only hydraulic energy is used (blue).

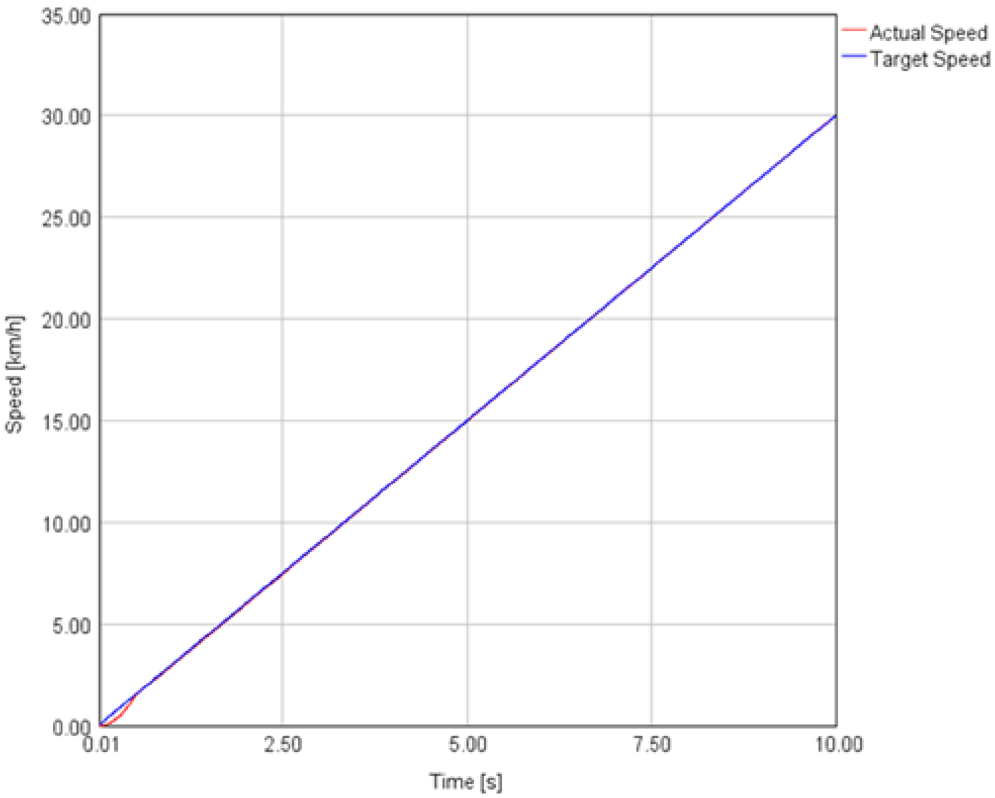

Figure 6- 18: Hydraulic Acceleration Test for 30 km/h

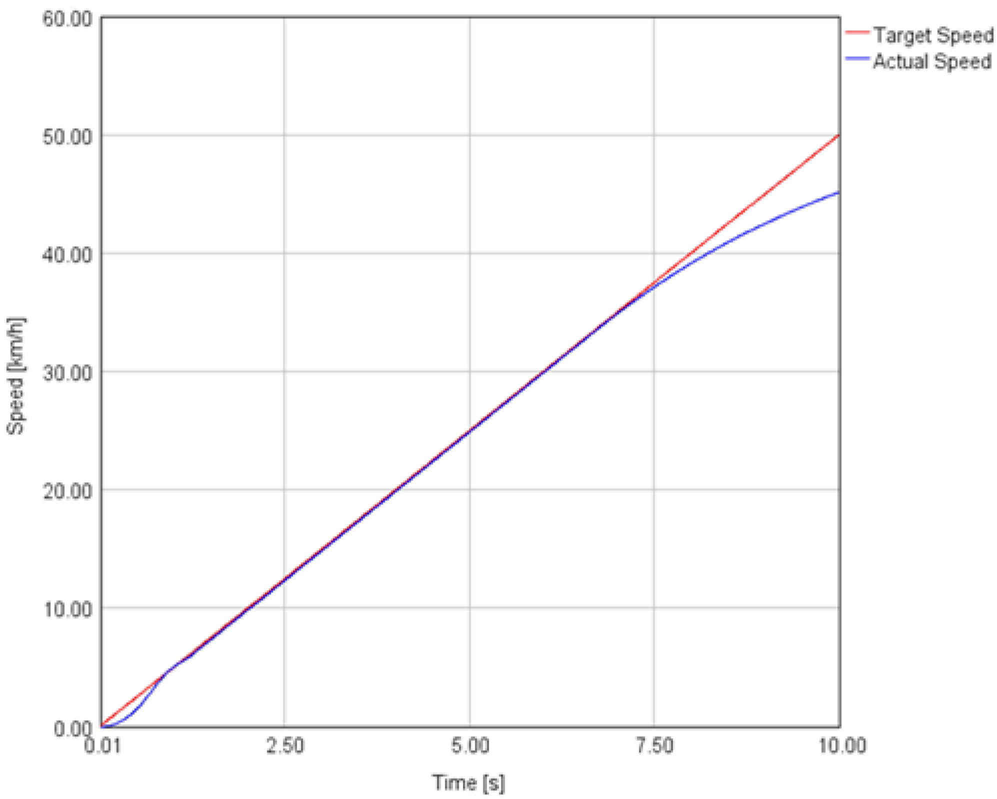

Figure 6- 19: Hydraulic Acceleration Test for 50 km/h

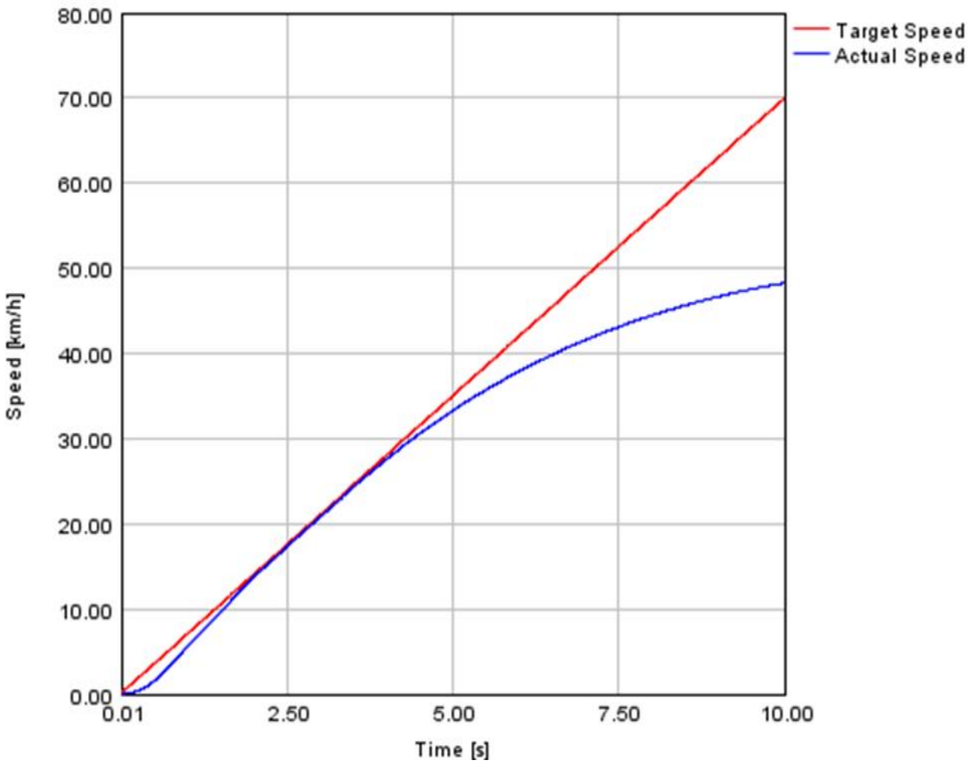

Figure 6- 20: Hydraulic Acceleration Test for 70 km/h

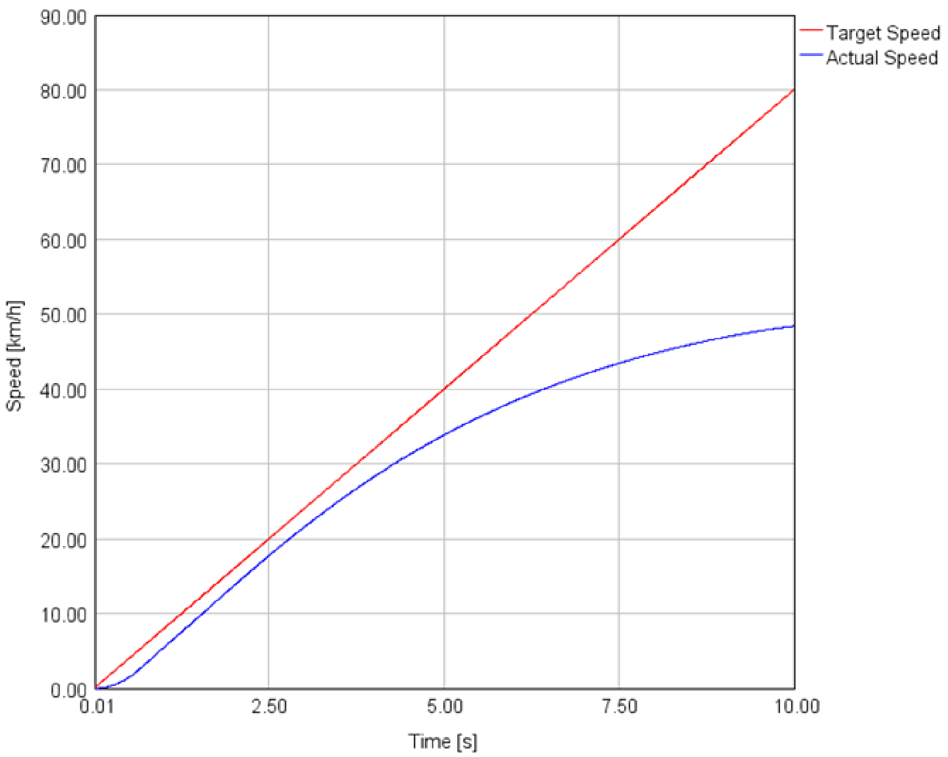

Figure 6- 21: Hydraulic Acceleration Test for 80 km/h

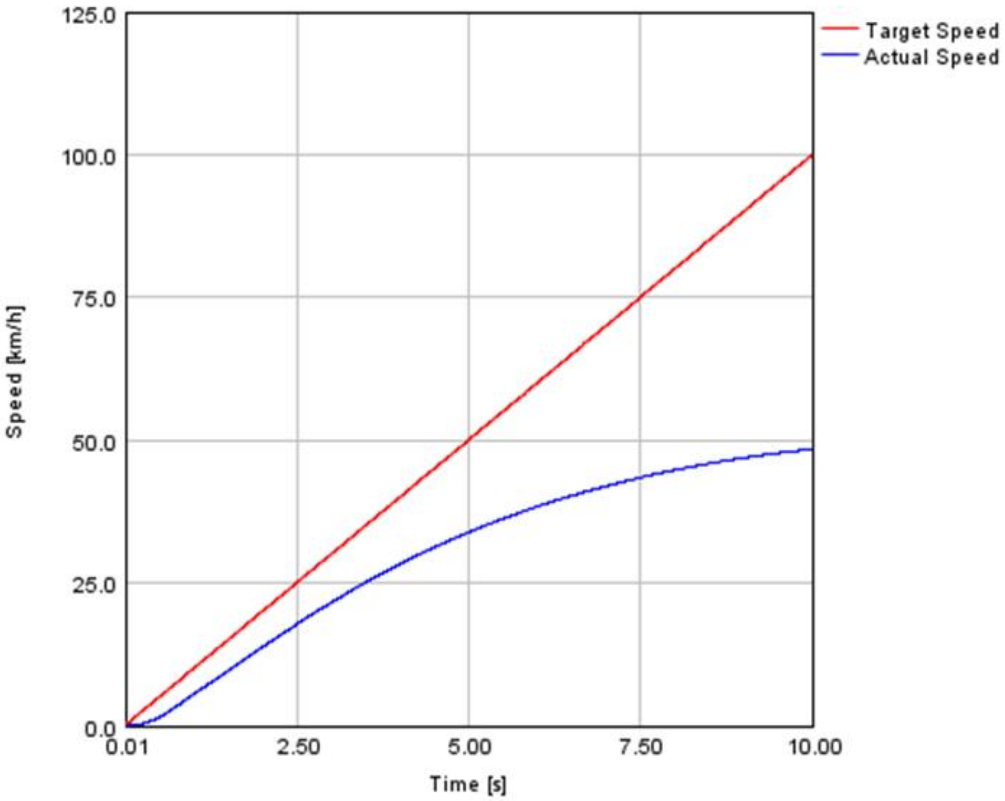

Figure 6- 22: Hydraulic Acceleration Test for 100 km/h

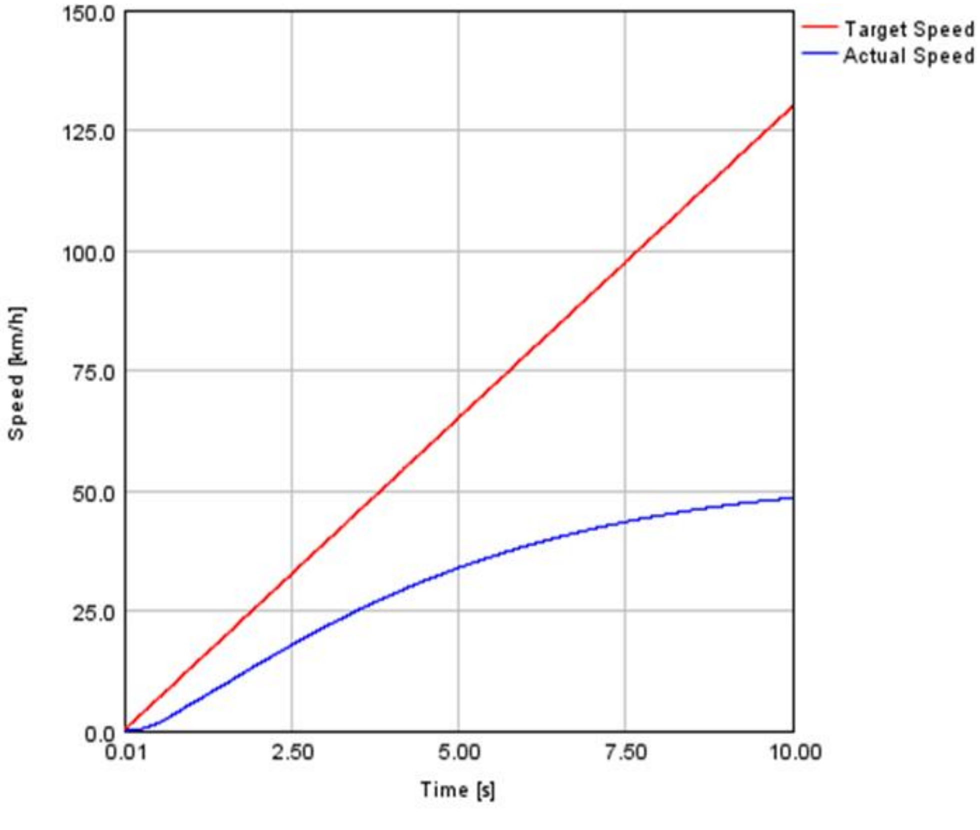

Figure 6- 23: Hydraulic Acceleration Test for 130 km/h

6.3.1 Discussion of the Results of the Hydraulic Acceleration Tests

The value to be taken as the maximal acceleration, depends on the applied criteria, which can be discussed, since it has been own defined.

Under the chosen canon, the maximal acceleration on pure-hydraulic mode is the one corresponding to the acceleration to 90 km/h in ten seconds (2,50 m/s^2). The approval of the acceleration to 100 km/h (2,78 m/s^2) to be accepted as the maximal one is debatable

7. Conclusion

The analysed powertrain for the HHV has proved to save an important amount of fuel, especially in driving cycles which include frequent braking events. On hydraulic-only mode, the vehicle has a limited use in terms of range and accelerations. The best usage of this vehicle would be in urban driving.

The vehicle consumes more at high speeds, being the root of the problem the excessive power flow through the hydraulic circuit when the Pump and the Pump/Motor work inversely.

The solution to this issue is to find an optimal geometry of the planetary gear set, although the fuel economy with the original PGS can be enhanced, if the Pump negative speed is limited to –1750 RPM, instead of the -4000 RPM limit of the original Pump.

Furthermore, it has to be mentioned that the hydraulic hybrid power train, although being a more affordable alternative than the HEVs, is not a technology whose developments and improvements will help the electric vehicles to become standard.

However, it can be considered as an immediate solution to reduce the emissions in urban areas, until electric vehicles are affordable for a greater spectrum of the population and the necessary infrastructure to recharge the batteries or the fuel cells are easily accessible.

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