Improving System Design and Power Management for Hybrid Hydraulic Vehicles Minimizing Fuel Consumption

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To my family

Mahnaz & Daniel

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No man succeeds without a good woman behind him. Thanks to my wife for giving peace and hope during my life, who made all this possible, who gives me love and spiritual support when I was in low.

Duisburg, December 2014 Mohammad Ali Karbaschian

Kurzfassung

Im Fokus dieser Arbeit steht die optimale Gestaltung und Steuerung von hybridhydraulischen Fahrzeugen, um die gewünschten Eigenschaften des Systems, nämlich Kraftstoffverbrauch und Fahrbarkeit zu verbessern. Trotz unterschiedlich optimaler und suboptimaler Power-Management-Strategien sind typische Probleme im Umgang mit der Echtzeit-Anwendbarkeit von Power-Management-Strategien, speziell Rechenlast und Lastzyklus-Anforderungen noch diskutierte Themen in diesem Zusammenhang. Basierend auf den Modellen der hydraulischen Komponenten und Teilsysteme werden typische hydraulische Hybridfahrzeugtopologien entwickelt, verifiziert und mit Simulationsergebnissen aus technischer Software für die Modellierung der Hydrauliksysteme verglichen. Die parametrischen Modelle können für die Umsetzung des typischen optimalen Power-Management Strategien verwendet werden. Der Kern der Arbeit ist die Entwicklung, Anwendung, Optimierung und Auswertung von Druckregelstrategien, Power-Management-Strategien, sowie die Optimierung des Systemdesigns. Obwohl regelbasierte Druckregelstrategien suboptimale Ansätze darstellen, werden Kraftstoffverbrauch und Fahrverhalten in Form von Referenz-Geschwindigkeits-Trackings als wichtigste Kriterien für eine angemessene Bewertung und den Vergleich von optimierten Power-Management-Strategien im Rahmen dieser Arbeit berücksichtigt. Um eine Online-Optimale Power-Management-Strategie zu entwickeln, sind drei Arten von Multi-objective Multi-parametric optimalen Steuerungsprobleme entwickelt und im System angewendet worden. Hierbei werden verschiedene Optimierungsalgorithmen wie Dynamic Programming (DP), Non-dominated Sorting Genetic Algorithm II (NSGA II) und Model Predictive Control (MPC) angewendet. Der Hauptgrund für die Entwicklung von DP-basierten Leistungsverwaltungen ist die Entwicklung eines Off-line Power Management, um die Leistung von anderen entwickelten Algorithmen zu bewerten. Anschließend wird eine Instantaneous Optimized Power Management (IOPM) auf der Grundlage quasi-statischer Modelle des Systems entwickelt, welche auf dem momentanen Leistungsbedarf in jedem Zeitschritt basiert. Die entwickelten Power Management Strategien verbessern erheblich die gewünschten Systemeigenschaften, nämlich Effizienz und Fahrbarkeit Regelbasierte Druckregelstrategien sowie IOPM zeigen eine vergleichbare Performance. Basierend auf den entwickelten Off-line Power Management Strategien wird ein globaler Optimierungsansatz für die gleichzeitige Optimierung von Design-und Steuerungsparametern in diesem Beitrag entwickelt. Mit diesem Ansatz kann eine optimale Systemauslegung und deren Regelparameter an die angegebene Topologie für eine beliebige Fahrzeugklasse erreicht werden.

Abstract

The focus of this thesis is the optimal design and the optimal control of Hybrid Hydraulic Vehicles (HHV) to improve system desired characteristics, fuel consumption, and driveability. Despite the different optimal and sub-optimal power management strategies, typical challenging problems dealing with real-time applicability of the power management strategies particularly computational load and load cycle requirements are still discussing topics in this context. Based on the hydraulic components models as well as subsystems, typical HHV topologies are developed and verified compared to simulation results obtained from a technical software for the modeling of hydraulic systems. The parametric models can be used for the implementation of different optimal power management strategies. The core of the thesis is development, application, optimization, and evaluation of pressure control strategies, power management strategies, as well as the optimization of system design. Whereas rule-based pressure control strategies are sub-optimal approaches, fuel consumption and driveability in the form of reference velocity tracking are considered as the main criteria for a reasonable identical evaluation and comparison of optimized power management strategies developed within this thesis. In order to develop an on-line applicable optimal power management strategy, three types of multi-objective multi-parametric optimal control problems are developed and applied to the system. Hereby different optimization algorithms like Dynamic Programming (DP), Non-dominated Sorting Genetic Algorithm II (NSGA II), and Model Predictive Control (MPC) are applied. The main reason for the development of DP-based power management is the development of an off-line control trajectory in order to evaluate the performance of other developed algorithms. Subsequently, an Instantaneous Optimized Power Management (IOPM) based on the quasi-static model of the system is developed. This optimal power management operates based on instantaneous information about power demand without the consideration of time variation effects. The developed power management strategies significantly improve the system desired characteristics, namely efficiency and driveability. However, rule-based pressure control strategies as well as IOPM have comparable performances. Based on the developed off-line power management strategies, a global optimization approach for the simultaneous optimization of design and control parameters is developed within this contribution. Using this approach, optimal system design and control parameters related to the given topology for an arbitrary vehicle class can be achieved.

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Nomenclature

Symbol	Parameter	Unit
α	Swash plate angle	[rad]
β	Bulk modulus	[MPa]
η_m	Motor overall efficiency	[-]
η_p	Pump overall efficiency	[-]
η_{tm}	Motor mechanical efficiency	[-]
η_{vm}	Motor volumetric efficiency	[-]
η_{tp}	Pump mechanical efficiency	[-]
η_{vp}	Pump volumetric efficiency	[-]
ρ	Oil density	[kg/m^3]
ω_e	Engine speed	[rpm]
ω_{eoop}	Engine speed at EOOP	[rpm]
ω_{eool}	Engine speed at EOOL	[rpm]
ω_p	Pump speed	[rpm]
Δp	Pump pressure difference	[Pa]
d	Piston diameter	[m]
p	Pressure	[Pa]
p_{gas}	Accumulator gas pressure	[Pa]
p_{oil}	Accumulator oil pressure	[Pa]
x_p	Pump displacement ratio	[-]
v	Vehicle velocity	[km/h]
v_{ref}	Reference velocity	[km/h]
z	Number of pistons	[-]
C_f	Friction loss coefficient	[-]
C_h	Hydrodynamic loss coefficient	[-]
C_s	Laminar leakage coefficient	[-]
C_{st}	Turbulent leakage coefficient	[-]
C_v	Viscous loss coefficient	[-]
D	Pitch circle diameter	[m]
D_m	Motor volume displacement	[l/s]
D_p	Pump volume displacement	[l/s]
E	Energy	[J]
E_f	Frictional energy	[J]
F_{br}	Brake force	[N]
P_a	Accumulator power	[W]
P_d	Vehicle power demand	[W]
P_e	Engine power	[W]
Q_m	Motor flow rate	[l/s]
Q_{oil}	Accumulator oil flow rate	[l/s]
Q_p	Pump flow rate	[l/s]
R	Universal gas constant	[J/mol K]

Symbol	Parameter	Unit
V_{acc}	Accumulator volume	[l]
V_{gas}	Accumulator gas volume	[l]
V_{oil}	Accumulator oil volume	[l]
T	Accumulator gas temperature	[K]
T_e	Engine torque	[Nm]
T_{eoop}	Engine torque at EOOP	[Nm]
T_{eool}	Engine torque at EOOL	[Nm]
T_m	Motor torque	[Nm]
T_p	Pump torque	[Nm]

Abbreviations

DoH	Degree of Hybridization
DP	Dynamic Programming
ECMS	Equivalent Consumption Minimization Strategy
EOOL	Engine Optimal Operation Line
EOOP	Engine Optimal Operation Point
HEV	Hybrid Electric Vehicle
HHV	Hybrid Hydraulic Vehicle
IOPM	Instantaneous Optimized Power Management
MPC	Model Predictive Control
NSGA II	Non-dominated Sorting Genetic Algorithm II
PHHV	Parallel Hybrid Hydraulic Vehicle
SHHV	Series Hybrid Hydraulic Vehicle
SoC	State of Charge

1 Introduction

Nowadays, increase of the oil demand and decrease of the oil sources are critical problems of the world. Ground transportation system as the main fuel consumption has improved the economic aspects. However, the disadvantages of ground transportation system like greenhouse effects, low efficiencies, and high fuel consumptions are the challenging problems in the context of energy efficiency and environmental issues. In figure 1.1, the trends of vehicle ownership and oil demand for road transportation system are shown [OPE14]. Accordingly, fuel demand and air pollution will increase significantly in the next years.

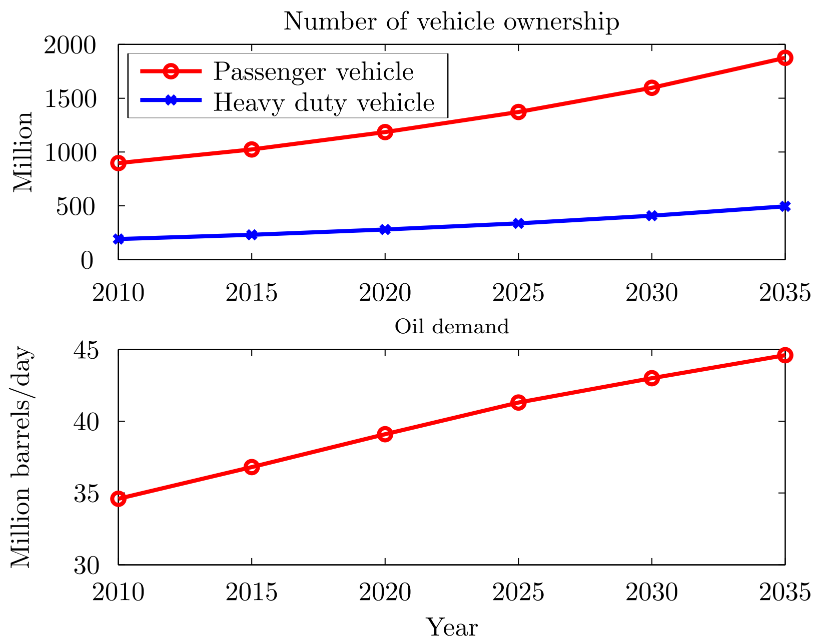

Figure 1.1: Vehicle ownership and fuel consumption trends [OPE14]

Therefore, efforts have been made to reduce energy consumption by reducing fuel consumption in automotive. Besides development of new technologies to improve vehicle safety, driver comfort, and driver assist systems, automakers compete to improve fuel economy of their products according to stricter standards such as the European emission standards. Developments of lightweight vehicles, replacement of hydraulic assist steering systems with electric ones, and application of turbochargers in diesel engines are some examples of these efforts. However, the main solutions are either using alternative fuels such as Ethanol and Gasoline, or improvement of the vehicle efficiency using alternative power trains. In order to use alternative fuels, fundamental changes in the engine structure of the conventional power trains are unavoidable. On the other hand, hybrid power trains are applicable to conventional vehicles with some modifications. Therefore, hybrid power train technology has a short-term practical solution.

1.1 Hybrid power trains

Hybrid power train technology as a potential solution for the improvement of vehicle efficiency is the central topic in the context of green power train technologies. By definition, hybrid power train includes two or more power sources combined either using a conventional transmission system or a hydraulic/electric transmission system to overcome power demand of the vehicle [EGGE05]. The functionality of both proposed hybrid electric vehicle (HEV) and hybrid hydraulic vehicle (HHV) are the same, despite their system components and different dynamics. The HHV or regenerative braking system which operates based on regeneration of braking energy, has been discussed since 1973 [OE73]. The most important case to use this kind of hybrid system is the case of Stop and Go vehicles like garbage transporter and shuttle buses [BR14b]. In figure 1.2, power density and energy density of hydraulic accumulator and typical electric batteries are compared. Accordingly, hydraulic accumulators have smaller energy density but larger power density compared to those of electric batteries. Due to chemical transportation processes, the charging and discharging processes of electric batteries cannot be compared with those of hydraulic accumulators. Here a compressible gas is compressed more or less ideally and more important, very fast. From an energy point of view it should also be stated that energy conversion efficiency using electrochemical processes (charging and discharging of batteries) is about 70% to 90% (per charging). In case of charging with high dynamics (fast charging/discharging) the efficiency is reduced, which is known as the rate-efficiency according to Peukert limited due to cell-chemistry [LL04]. For these reasons, dynamic response of hydraulic accumulator is faster than those of electric battery and can be used to improve the dynamic behavior of the vehicle significantly. In contrast, the low capacity of the hydraulic accumulator reduces its functionally to boost operation of short duration.

In figure 1.3, the structures of fully controllable series HEV and HHV topologies are illustrated. Electric generator, electric motor/generator, and buffer (battery, super cap, *etc.*), are the main components of HEV whereas, the main components of HHV are pump, motor/pump, and accumulator. The comparison of HEV and HHV is discussed in [LND08, KMMS13]. In [LND08] fuel consumption and vehicle performance of three parallel hybrid power trains applied to a bus are compared. The proposed hybrid vehicles are: a super capacitor-based HEV, a battery-based HEV, and a HHV. According to the results, HEV is more efficient than HHV. In addition, vehicle performance is compared for three criteria, maximum vehicle velocity, gradeability, and acceleration time. The super capacitor-based HEV can realize higher acceleration rate than HHV, while acceleration rate of the HHV is higher than battery-based HEV. However, gradeability of battery-based HEV is more than HHV and super capacitor-based HEV. In contrast to the system characteristics and power management control strategy, the effects of the driving cycle on the performance and efficiency of the HHV and HEV are not considered. Due to the low

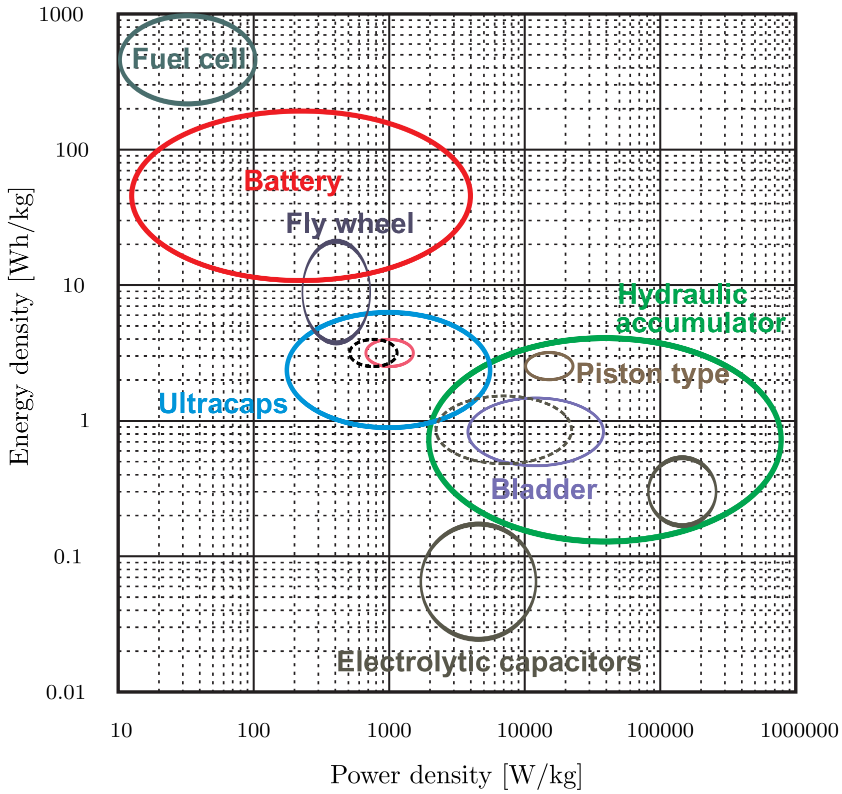

Figure 1.2: Power and energy density of typical energy storages [BEGK07]

power capacity of the hydraulic accumulator, it can only act as a temporary power source. In order to capture braking energy and due to the accumulator low capacity, accumulator should be discharged before each braking phase. On the other hand, fast dynamics of the accumulator makes it appropriate for stop and go driving cycles [ON114]. City driving cycles such as ECE-15 include more acceleration and deceleration phases in each period of time and the maximum velocity value is also lower than highway driving cycles. In contrast, charge sustainability of electric batteries as well as their large capacity are the reasons for the higher efficiency of the HEV than HHV in highway driving cycles.

According to figure 1.3, using the variable displacement pump, motor, and control valve to control the accumulator makes the system fully controllable. Here, fully controllable includes decoupled control of the power sources according to the vehicle power demand. The same statement is true for the proposed HEV topology when DC/DC converters are implemented. Although the dynamics of the HEV and HHV are different, their control concepts are equal.

Typical topologies of hybrid power trains are series, parallel, and power split. The parallel hybrid vehicle is also known as the power assist topology. It contains a hydraulic/electric motor coupled to the conventional transmission system. Gener-

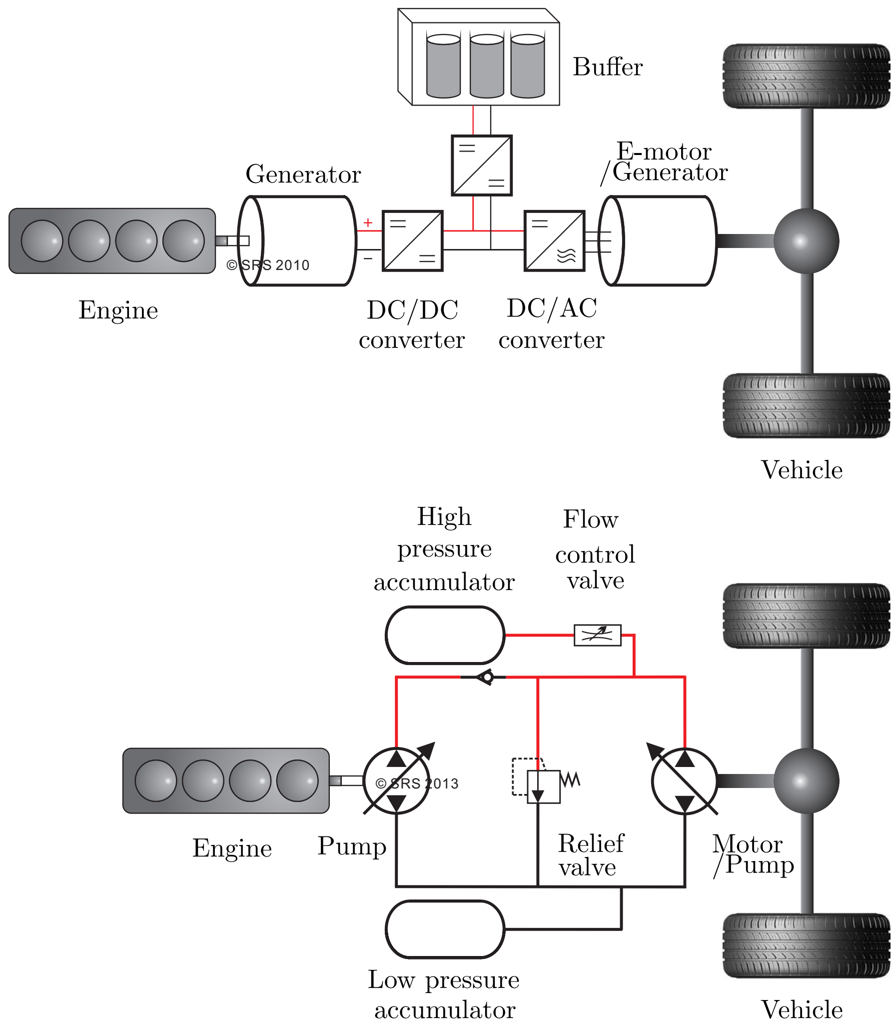

Figure 1.3: Series topology of, (up) hybrid electric vehicle; (down) hybrid hydraulic vehicle [KS14]

ally, the accumulator can be charged using braking energy or engine power. Also possible is the use of the engine power only for charging the accumulator (stationary charging mode). In series topology, the mechanical connection between engine and vehicle is replaced by a hydraulic/electric transmission system using direct coupling of the hydraulic/electric motor to the drive shaft and the hydraulic pump/electric generator to the flywheel. Due to the mechanical decoupling of vehicle and engine, optimal operation of the engine with an appropriate controller is realizable. To recapture braking energy, the hydraulic/electric motor connected to the drive shaft changes its operation mode to hydraulic pump/electric generator during deceleration phase. Using the valves in HHV and DC/DC-converters in HEV, several topologies for series hybrid vehicles can be realized. However, each topology has its individual characteristics and control strategies.

1.2 Application on automotive

1.2.1 Bosch-Rexroth HRB

For the first time in 2009, Bosch-Rexroth installed its first parallel Hydrostatic Regenerative Braking system (HRB) on Crane Carrier Company LET2 Truck and Heil Environmental refuse body and tested by New York City Department of Sanitation (DSNY) [BR14d]. Later on in 2011, it is applied to a Mack garbage truck. In its daily driving cycle, the reference garbage truck stops about 800 times. Depending on driving cycle and driver behavior, the capability of this truck to reduce fuel consumption is reported about 25% [BR14b].

1.2.2 Eaton HLA®

Eaton has developed the Hydraulic Launch Assist System (HLA) in 2007 [Eat14]. The operation modes of this system are: economy and performance mode. In economy mode, vehicle starts up only using accumulator power while in performance mode it starts up using combination of accumulator and engine power. The test results by applying HLA on Peterbit 320 truck show 28% fuel economy in economy mode and 17% in performance mode. Braking energy regeneration rate of this system is claimed about 70% available kinetic energy while its acceleration in economy and performance modes are claimed about 2% and 26%.

1.2.3 Artemis intelligent power

In 2008, Artemis presented the first prototype of Digital Displacement® hybrid hydraulic transmission technology applied on BMW 530i [Art14]. Traction force of the wheels is supplied individually. The emission reduction of this technology in a combined driving cycle is claimed about 30%. By elimination of transmission system, the interior space of the vehicle can be larger.

1.3 Contribution of the thesis

In this thesis, a methodology for the optimal design of the hybrid hydraulic power train is proposed. Moreover, typical control strategies are implemented to optimize the power distribution in a hybrid hydraulic power train. Several optimal control methods are applied with the aim to develop an on-line Instantaneous Optimal Power Management (IOPM). Some parts of this thesis are published as journal papers [KS14] or presented in international conferences [KS12, MKS13, KMMS13, KSeda, KSedb].

In chapter 2 [KS14], an overview about different power management approaches are given and evaluated. Typical power management approaches are classified based on applicability in real-time. The categories for the evaluation of power management approaches are off-line power management strategies (Dynamic Programming and Genetic Algorithm), on-line power management strategies (rule-based power management, fuzzy logic controller, and Equivalent Consumption Minimization Strategy), and real-time power management strategies (telematics technology-based predictive approach, predictive approach, and stochastic model predictive approach). Power management approaches are evaluated based on the results from publications.

In chapter 3 the models of elements corresponding to mechanical and hydraulic subsystems are investigated. This is performed using physical-based and logical modeling of the components considering components efficiencies, and the evaluation of system models. In order to evaluate components model as well as hybrid hydraulic power train model, simulation results of the models are compared to the ones getting from Hopsan.

In chapter 4 [KSeda], several control strategies for application within series hybrid hydraulic power train are designed, applied, and evaluated. Common aspects of the introduced control strategies are the low-level component-oriented control realization. In contrast to common supervisory control concepts, the low-level control strategies are more accurate because of the application of dynamic model of the system instead of static model. The engine on/off control strategy based on compensation of used accumulator power is compared to accumulator depleting strategy, in which demanded power is mainly supplied using accumulator power, as well as smooth engine strategy, in which the maximum power of the engine is bounded. The effects of control parameters on the performance and efficiency of the hybrid power train are investigated. Simulation results confirm the results and show significant improvement of fuel economy by applying engine on/off strategy as well as accumulator depleting strategy. In addition, the performance of the hybrid power train is detected significantly depend on the driving cycle.

In chapter 5 [KSedb], an IOPM algorithm using a two-level controller is introduced. Moreover, a Model Predictive Controller-based (MPC) power management is developed for comparing with the first one. Finally, both power management algorithms are compared to a globally optimized power management serves as benchmark reference.

In chapter 6, first typical topologies of HHV are investigated. Architectures as well as several design and control methods of the topologies are discussed in detail. Moreover, fuel economy of the typical topologies are compared based on simulation results. A new approach for the optimal design of hybrid hydraulic power train in order to apply to different vehicle classes is proposed. The criteria considered in this approach are based on instantaneous optimization of dynamical characteristics of the power train. The topologies of HHV, configurations, component sizes, and power management strategies are the proposed outputs of this global approach. In addition to system feasibility respecting to operation range of the system, effects of power train characteristics on the fuel consumption and driveability are considered.

The last chapter summarizes the main results and discusses future works.

2 Literature review

2.1 Introduction

Characteristics of hybrid power trains are defined by certain aspects like engine operation efficiency, power transfer efficiency, and regeneration of reversible energies. The efficiency of the entire hybrid power train drastically depends on the complex dynamics of the subsystems. Therefore, the design and application of an optimal controller to hybrid power trains is necessary. Power or energy management in hybrid power trains acts as a supervisory control algorithm to control the power split between engine and motor. Supplying vehicle power demand, objective functions such as vehicle fuel consumption are optimized. Therefore, the task of the power management strategy is the most important subject.

In the context of vehicle longitudinal dynamics, several definitions for the performance of the vehicle exist. Acceleration rate, maximum vehicle velocity, and gradeability are known as typical criteria defining vehicle performance in practice. In the context of hybrid vehicles, one of the typical definitions describing vehicle performance is reference velocity tracking. In other words, for the improvement of vehicle performance, it can be expected that the hybrid vehicle can also track the expected driving cycle. For this reason, one of the objective functions in optimal control of power management controller belongs to vehicle velocity and reference velocity difference minimization. This chapter is published in the form of a scientific paper [KS14].

2.2 Power management controller

In review papers, several classifications for power management approaches are given. In [SG07], optimal power management approaches are additionally classified based on the used mathematical optimization methods into two categories namely, numerical optimization methods such as Dynamic Programming (DP) and analytical optimization methods such as Pontryagin minimum principle [YTFP13]. Static optimization, DP, Quadratic Programming (QP), Model Predictive Control (MPC), linear programming, Hamilton-Jacobi-Bellman equation are introduced as the usual optimization methods used for power management optimization. In [HSDS07], power management approaches are classified into optimal and sub-optimal control strategies. The global optimization methods such as DP are proposed as reference for other algorithms done to their reliable solutions. The other categories include heuristic control, rule-based power management control, and Equivalent Consumption Minimization Strategy (ECMS) [WHS+13, XCSZ14].

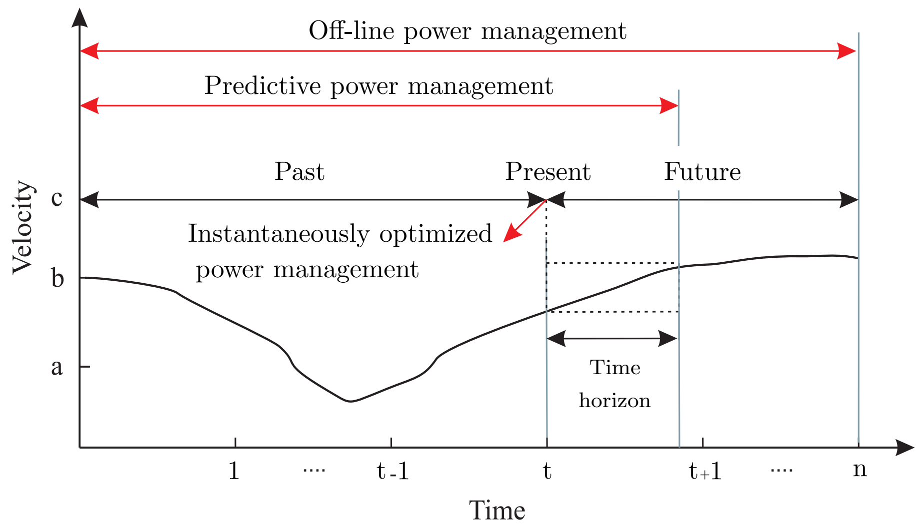

Figure 2.1: Optimization time domain for off-line, on-line, and real-time power management approaches [KS14]

Whether a power management strategy is applicable to an on-line hybrid power train depends on the available information regarding vehicle power demand as well as the future vehicle velocity. For off-line power management optimization, the past, present, future information about the vehicle velocity is needed as depicted in figure 2.1. While for the on-line Instantaneous Optimized Power Management (IOPM), just the instantaneous information at present is needed. Availability of partial information about vehicle velocity in the future makes it feasible to optimize the power management approaches in future time. Therefore, using predictive approaches, vehicle velocity can be predicted for a specific time horizon. Power management strategies can consequently be optimized in a limited time. The relation between different power management approaches like off-line, predictive, and instantaneous approaches are illustrated in figure 2.1. An overview about the proposed power management approach classification is presented in Table 2.1.

In the following sections typical off-line power management optimization methods and algorithms are explained. Further, the on-line and real-time power management methods are reviewed. Finally, the power management methods are evaluated and compared based on different criteria and published results. An overview of proposed power management classification is presented in table 2.1.

2.3 Off-line power management

Off-line power management is usually used to define optimal power-management control parameters in advance. Therefore, the approaches use knowledge of the vehicle driving cycle, briefly speaking: it is assumed that the knowledge is exact

Table 2.1: Classification of power management algorithms [KS14]

Off-line	**On-line**	**Real-time**
Dynamic Programming	Rule-based controller	Telematics predictive strategy
Genetic Algorithm	Fuzzy logic controller	Model predictive strategy
	Equivalent Consumption Minimization Strategy	Stochastic Model Predictive Strategy

and known in advance. Depending on the class of power management, both static and dynamic optimization methods can be applied to off-line power management. Static optimization methods use the quasi-static model of the power train to be controlled, such as instantaneous power balance method. In contrast, dynamic optimization methods such as DP, use dynamic models of the subsystems during the optimization process [WLF+04, KMMS13]. In this case, the mathematical formulation of the dynamics of the subsystem such as engine, vehicle, pump, motor, and accumulator are used to realize the dynamical model of the power train topology. The dynamic model of hybrid power train is complex and nonlinear. The nonlinearity results from the multi-mode operation of the hybrid power train. Due to the dynamical complexity of the power train, simplification of power management optimization, linearization, and discretization of the model are unavoidable. Feasibility of the solution is the other problem of power management optimization. Due to limited optimal operation of power train components and restrictions dictated by objective functions, different equality as well as inequality constrains are applied to this optimization problem. As some of these constrains depend on the states of the problem, they change during time. Therefore, feasibility check of the solution is also unavoidable.

Because of the necessity of prior knowledge about the vehicle driving cycle, in off-line power management strategies, both global and local optimization algorithms are applicable. However off-line globally optimized power management approaches are more optimal in comparison to the locally optimized power management, because global optimization methods use a wide range of input information for optimization, while local optimization methods define the optimal control based on individual working points.

2.3.1 Dynamic Programming-based power management

Dynamic Programming is a numerical sequential global optimization approach based on Bellman's principle of optimality [BD62]. The underlying concept of DP is optimization of a sequence of decisions to optimize each related subsequence. Therefore,

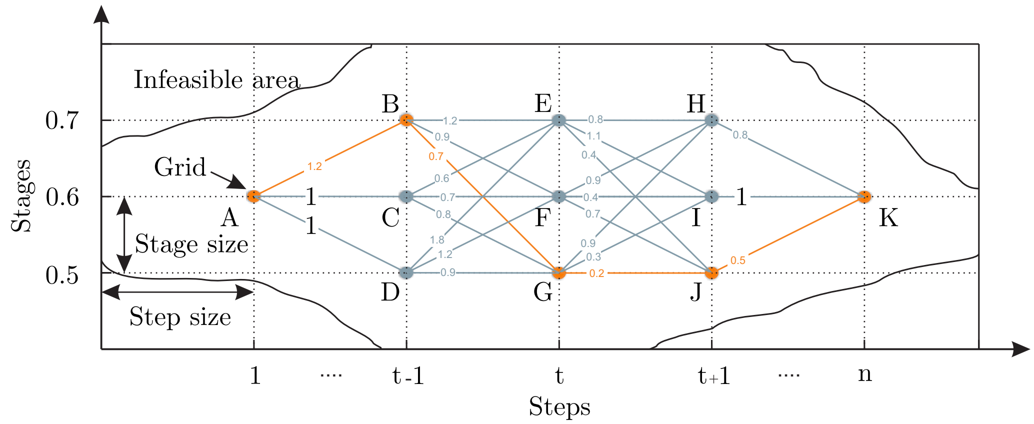

Figure 2.2: Dynamic Programming expression [KS14]

DP results in an optimal trajectory of sequential decisions within predefined initial and final conditions, realizing a global optimization of the objective functions as depicted in figure 2.2.

The optimality resulting from DP depends on the number of grid points on the trajectory plan of the state of the system. Decrement of the size of stages as well as steps increase the number of grid points and the optimality of the DP is increased consequently. However, the increase in the number of grid points leads to an increment of the computational load. To decrease the computational load, infeasible states can be omitted as depicted in figure 2.2. Although DP-based power management belongs to the category of off-line power management approaches, it establishes a benchmark for the evaluation of the optimality of the other developed power management approaches, particularly rule-based power management [LE13]. Moreover, the results of DP-based power management are typically used for improving the optimality of the hierarchical power management controllers [LKGP01].

In [WL12], the formulation of DP-based power management optimization for application in three typical topologies of HEVs, namely series, parallel, and power split, is given. Additionally, the sensitivity of the optimality of DP-optimized power management to the initial and final values of the system states is investigated. The results such as desired trajectories of accumulator State of Charge (SoC) for several driving cycles are saved in a lookup table to be implemented as real-time power management. A rule-based power management is developed and optimized using lookup table. The comparison of the tuned rule-based controller and DP-optimized power management confirm the improvement of the system's efficiency, showing closeness of the results to the DP-optimized power management. In [SMAH08], the same comparison between three typical topologies of hybrid hydraulic power train using DP-optimized power management is realized. For parallel topology transmission, gear ratio and engine state are considered as the control variables, while for series topology, engine state and engine speed are the control variables. For power-spit, control variables of both parallel topology as well as series topology are considered as control variables. The results show that the power-split appears to be the worse topology with respect to fuel consumption reduction for both urban and highway driving cycles. These results are in contradiction to the conclusion given in [DCLC13]. One reason for this conflict may be the consideration of subsystem efficiencies in both modeling and optimization in [DCLC13]. In [SMAH08], fuel consumption of the power split topology is reported to be close to series topology, while in [DCLC13], fuel consumption of both series and parallel topologies are significantly more than power split. Based on these results, parallel topology is slightly more efficient than series in the urban driving cycle but is significantly more efficient than series in the highway driving cycle [SMAH08]. The reason for low efficiency of both series and parallel topologies in highway driving cycle is the unavailability of braking energy for regeneration. The reason for different result in [SMAH08, DCLC13] may be due to the differences in the models. Furthermore, the substantial difference between fuel consumption of parallel and series topologies is not justified.

In [JBF11, TGP08], stochastic DP (SDP) method is used for optimization of the power management of a parallel hybrid hydraulic truck. Using this method, instead of using prior knowledge about driving cycle, the driving cycle is approximated using a Markov chain. A Markov chain is a qualitative mathematical approach to describe finite states of systems, assuming stochastic transition probabilities between the states. Using Markov chains, the probability to reach upcoming states can be calculated, the main assumption is, that the transition only depends on the state itself (and not on the former transitions history). The result of power management optimization for several driving cycles are calculated and saved in a lookup table. For the application of the results to a real-time power management, the instantaneous velocity of the vehicle is recognized to be close to those in the lookup table [MFC11]. Thereafter, the related control inputs are extracted using interpolation techniques.

A comparison between rule-based power management and two variables SDP power management applied to a parallel hybrid hydraulic power train in [JBF11], shows a significant fuel consumption reduction for SDP power management. The control inputs are the power distribution and gear shifting ratios. In [LP08], the performance of SDP-based power management and ESMC are compared with a DP-based power management, applied to a power split HEV. A two level controller approach consisting of power distribution control and engine optimization control is implemented. Vehicle velocity and SoC are considered as the state variables of the power train system. According to the results, both SDP and ECMS decrease the fuel consumption significantly close to benchmark.

In [SG09], a new boundary-line DP power management is developed. The new one-dimensional DP approach explicitly solves the issues of infeasibility, accuracy, and computational load. It is demonstrated that using pre-calculation and elimination of infeasible state variables, the computational load as well as the feasibility problem can be solved. As depicted in figure 2.2, boundary lines of the infeasible areas do not pass necessarily over the grid point of the system state. By definition, grid points are the intersection of stage and step lines which contain the values of the state variables. Therefore, an exact interpolation between grid values improves the accuracy of the DP-based power management. For this reason, the boundary line method is proposed for the exact extrapolation of the values of state variable as well as cost function. The accuracy of the DP optimization depends on the grid size of the state-space. It is investigated that using boundary line method, the effect of state-space resolution can be minimized. Finally, the computational effort to calculate the values of the states on grid points in infeasible area is decreased by the elimination of infeasible areas of the state. Therefore, the computational load of the boundary-line DP power management is less than usual DP power management.

2.3.2 Genetic Algorithm-based power management

Genetic Algorithm (GA) is a numerical optimization approach for constrained, multi-objective multi-parametric, and complex nonlinear problems. In the process of GA optimization, instead of using analytical methods, techniques such as selection, encoding, mutation, and crossover are used. Through an iterative loop, a batch of variable is generated and for each batch, the objective function is evaluated with respect to a given objective function. Finally, during a multi-loop iteration process, the algorithm defines those samples which converge to the optimal solution with respect to the given objective function. Although, GA can be used for the optimization of complex nonlinear multi-input multi-output (MIMO) hybrid power trains, its computational load is high. Because of convergence toward an optimal answer, all possible combinations of samples must be iteratively evaluated. For this reason, GA is only appropriate for application in off-line power management [KMMS13, CMX$^+$13]. It is also used for the adjustment of the parameters of other control strategies.

A GA-QP-based power management for a power split HEV is developed in [CMX$^+$13]. Using the QP method, the relation between the battery current and fuel-rate is defined. Using the relation between two variables of the problem, the problem can be simplified by reducing the number of variables. The GA is used to find the power threshold of the engine in combination with the Lagrange method. The outputs of the controller are engine-on power threshold and the accumulator power current. This algorithm is applied to optimize the power management instantaneously in both urban and highway driving cycles. The efficiency of the new power management is demonstrated by comparing the results with that of conventional power train. Lack of validation of the new simplified model reduces the reliability of the results. Additionally, GA parameters, namely number of population, elitist amount, and mutation rate, influence the optimality of the problem. Therefore, optimization of the GA parameters instead of their experimental-base selection may improve the efficiency of power management.

The threshold-based energy control strategy is a rule-based on-line power management discussed in the on-line power management section. This strategy involves two parameters, torque difference and pressure limit of active-charging-pressure, related to the output torque of the engine and demanded torque, as well as pressure of the hydraulic accumulator. In [TJSF13], GA is implemented for the static adjustment of the parameters of the threshold-based energy control strategy. The results show substantial reduction in fuel consumption compared to the conventional threshold-based energy control strategy. The parameters of the controller are adjusted off-line. Nevertheless, due to the dependency of the results on the driving cycle, the controller cannot to be used in combination with unknown driving cycles.

A Non-dominated Sorting Genetic Algorithm (NSGA) based on GA is applied for optimization of power management of parallel HEV and HHV [KMMS13, DPAM02]. The control variables are the power distribution ratio between the engine and the secondary power source and the parameters of PI driver controller. Vehicle performance and fuel consumption are the objective functions of the considered optimization problem. The results show a substantial improvement in vehicle performance and fuel consumption reduction. The same control strategy using conventional GA is applied to a HHV in [QJF10] for sizing of the components. The objective functions are weighted using Decision Alternative Ratio Evaluation System (DARE) law. The simulation results show that the larger accumulator size improves the fuel economy but decreases the vehicle performance. Moreover, larger pump/motor size improves the fuel economy and vehicle performance.

2.4 On-line power management

The application of power management strategies to on-line or real-time systems involves solving basic problems, namely those of unknown upcoming power demand trajectory and computational load. Lack of the knowledge of future power demand trajectories, makes global optimization of the power management impossible in real-time. Therefore, only optimization methods based on instantaneous vehicle speed data can be implemented. However, efficiency and performance of the on-line power management strategies are consequently lower than those resulting from off-line power management strategies. Development of on-line power management approaches start with realization of rule-based controllers [LFL+04]. Conventional rule-based power management approaches are non-optimal with respect to off-line globally optimized power management. The optimization of rule-based power management using off-line methods results in a desirable solution but it depends strongly on the information used to describe the upcoming driving requirements, briefly speaking the driving cycle necessary for optimization is unknown resulting in a decrease in system efficiency. In the following sections, the most important on-line power management strategies are explained in detail.

2.4.1 Rule-based power management

Rule-based power management methods are simple to design and implement. Based on an on-line controller for implementation of supervisory power management strategies, the rules can be typical heuristic experiences or results combining if-then conditions. The concept of rule-based power management is an instantaneous determination of power split ratio based on logical rules and local constrains. Therefore, rule-based power management cannot be optimal. As the logical rules depend on the system characteristics, topology, and design goals, an unique method for synthesizing the rules does not exist. The first step to design a rule-based power management is often based on the determination of vehicle operational phases. Using driver command the operational phase of the vehicle is divided to four phases, namely acceleration, deceleration, stop, and constant velocity. Depending on the SoC and demanded torque, different sub-modes may be selected e.g., braking energy regeneration and conventional braking combination during deceleration, accumulator charging during acceleration, using only the engine in acceleration phase *etc.* [PR07].

The threshold-based energy control strategy is one of the conventional rule-based power management controller. In [TJSF13], a threshold-based energy control strategy is developed based on the thresholds of the engine efficient operation and SoC. In this power management controller, the engine efficiency map is divided into three areas with the help of two constant power lines, namely engine-on and motor-assist power lines as depicted in figure 2.3. These two reference power lines are used to control the state of the engine in acceleration phase. If the vehicle power demand is between zero and engine-on power line, only the motor supplies power. If the power demand is between engine-on and motor-assist power, only the engine supplies power. Finally, if the power demand is more than motor-assist power line, motor assists the engine to dominate the power demand. In deceleration phase, braking energy must be recaptured to charge the accumulator as much as possible. Accordingly, the most efficient operation point of the engine is around the point with engine speed 2250 rpm, engine torque 280 Nm, and engine power about 66 kW.

In [KPB11], a map-based power management controller is developed based on a DP-based power management and using comprehensive extraction method. In addition to fuel consumption, emission of the engine is considered to be minimized. Therefore, emission is modeled using engine thermal dynamic and after-treatment dynamic models. Three control strategies namely engine bang-bang, gear-shifting, and power split based on the DP optimization map are proposed. The thresholds of these controllers are cold-start and hot-start emission of the engine. The results justify

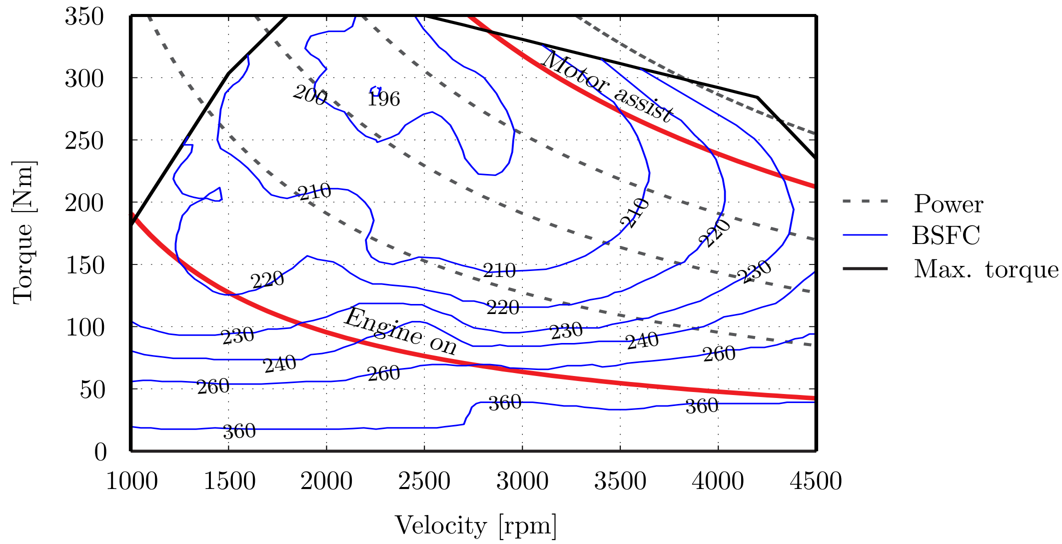

Figure 2.3: Engine threshold-based control strategy. Blue curves: the brake specific fuel consumption (BSFC) which represents the fuel consumption of the engine at each operation point; Dashed-line: output power of the engine [KS14]

the emission reduction of the vehicle using map-based power management controller.

Optimization of the applied logic of rule-based power management using the results of globally optimized power management such as DP is usual. The difference between demanded torque and efficient torque of the engine as well as the SoC threshold are the main control variables of rule-based power management [TJSF13]. According to the results, fuel consumption and dynamic performance of the power management are improved. Moreover, optimization of the rule-based power management controller using DP-based power management is realized in [FLD+04]. Based on the DP optimization results, a new gear-shift strategy is extracted for the HHV power train. Moreover, the logic-based rules for switching between operational modes are tuned using the DP results. For example if the SOC is high, charging of the accumulator with engine power must be avoided. The result shows a possible reduction of fuel consumption up to more than fifty percent using the optimized rule-based power management.

The bang-bang controller (also known as two points switching control, is in [EGGE05, FH12] denoted as thermostat strategy) is based on accumulator power preservation. In this control strategy, an effort is made to operate the engine at the most efficient points subject to the SoC of the accumulator [KF07]. The engine on and off strategy is based on hitting the minimum and maximum SoC of the accumulator respectively. However, in this strategy, engine switches between either idle/supply modes or on/off modes. The frequent switching of the engine is undesirable because it increases fuel consumption due to the transient operation of the engine. To

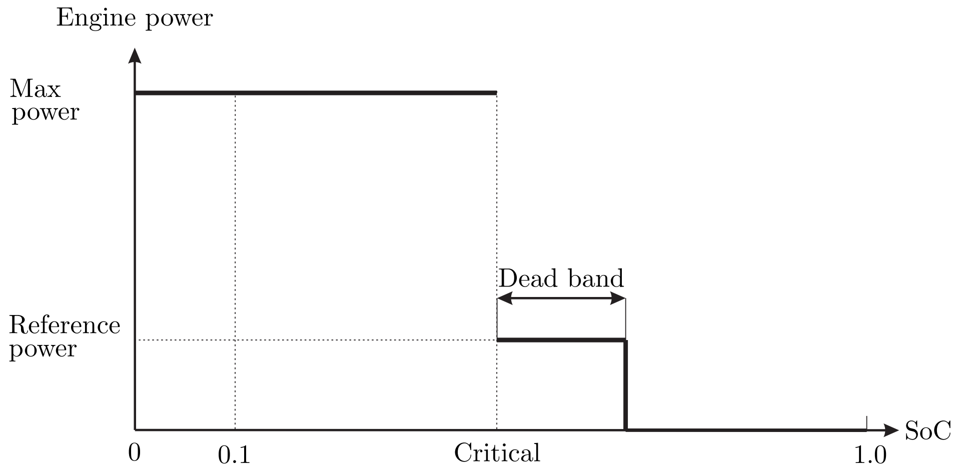

Figure 2.4: Two-points switching management controller [KS14]

overcome this drawback, the charging threshold of the accumulator can be limited using a SoC dead band [KF07]. In figure 2.4, the key parameters of the thermostat strategy namely, dead band, threshold power, and threshold SoC are presented. The critical SoC is the value in which the engine starts to charge the accumulator. In other words, when the SoC is higher than critical value, accumulator is the only power source while for values lower than critical SoC, the only power supply is the engine. To charge the accumulator, the engine operates at the shown maximum power point which is the most optimal operation point of the engine.

In [FH12], the effects of key parameters of the two points control strategy on the performance and efficiency of the power management are investigated. Accordingly, increase in the reference power value increases the number of undesirable switching of the engine between on/off modes or idle/supply modes at a certain time. An increase in dead band value, however, results in fuel consumption reduction. Moreover, the value of critical SoC has a significant effect on the amount of braking energy that is captured during deceleration phase. In other words, in two points control strategy, effort should be made in keeping the minimum SoC as low as the critical SoC. Therefore, depletion of the accumulator just before braking phase in order to increase of capturing braking energy may not be guaranteed. Consequently, algorithms defining the optimal values of control parameters are needed. For this reason, a static adjustment of control parameters is proposed in [FH12]. Although, this method improves the fuel economy relative to the conventional threshold strategy, using optimization algorithms may improve the performance and efficiency of the two points control strategy.

In [CVH13], three rule-based power management strategies are developed called namely by the authors: pure-thermostatic, acceleration-thermostatic, and with-idle-thermostatic strategies. The pure-thermal strategy uses SoC as the only state variable of the power train. In this case, the accumulator is charged with the engine whenever the accumulator has enough capacity. In the acceleration-thermostatic strategy, both SoC and driver command are used as state variables. In this case, in addition to the accumulator capacity, the power train must operate in acceleration phase to charge the accumulator. The only difference between acceleration-thermostatic and with-idle-thermostatic strategies is that, in the second strategy, the accumulator can be charged whenever the vehicle drives with constant velocity. The results justify the superiority of the with-idle-thermostatic strategies to decrease fuel consumption. The reason is that the accumulator can be charged in engine idle mode and it has enough capacity at the beginning of deceleration for capturing the braking energy.

Although rule-based power management controller is applicable to on-line and even real-time power management, it is not an optimal power management. Despite ability to improve the efficiency of the rule-based power management, the efficiency of the explained method is always less than both global and local optimization algorithms. Therefore, the rule-based power management controller is a sub-optimal, on-line and real-time-applicable power management.

2.4.2 Fuzzy logic controller

In contrast to rule-based power management, fuzzy-based power management controller is based on partially true logics. In other words, the rules are not necessarily true or false rules. Fuzzy controller contains a series of linguistic rules and each rule contains one antecedent and two consequences. The fuzzy logic controller consists of fuzzification, inference engine, rule base, and defuzzification. Application of partially true logics in fuzzy-based power management, causes a smooth transition of the dynamic behavior of the power train in contrast to true or false logics in the rule-based power management. Therefore, fuzzy controller may guarantee the robustness of the power management.

In [AC12], three control strategies are combined. The first strategy reduces fuel consumption using optimization of the power split ratio. The second control strategy maximizes the braking energy captured during deceleration phase, and the last control strategy sustains the SoC to increase the robustness of the control system. Three sub fuzzy controllers are designed to control the operation of the engine, on/off switching of the engine, and braking energy distribution. Based on accumulator energy level, different membership functions are selected which define the switch time between operation modes. The number of membership functions for braking energy distribution controller is five while for the two other controllers is seven.

The fuzzy controller uses driver command as well as SoC as the control inputs. By definition, the relation of captured braking energy to the available braking energy is regeneration ratio. The effect of this factor on the efficiency of the power management is studied and on-line control of this factor in order to improve vehicle performance is confirmed. The control strategy is applied to Toyota Pirus model, the used driving cycle is the combination of three driving cycles by means of long trip realization. Simulation results show efficient operation of the engine. The undesirable engine on/off switching is detected in the one segment of the driving cycle which contains more stop and go maneuvers. Comparison of the SoC shows better sustainability of SoC in the fuzzy-based power management in comparison with Prius HEV. The similar controller is used in [KBK10] to reduce the fuel consumption and emission.

A generalized fuzzy logic-based power management controller for three typical topologies of the HEV, namely parallel, series, power split is developed in [CMJJ10]. The state of the engine is determined by the power demand of a scaled vehicle, as well as engine velocity, engine temperature, SoC, vehicle speed, and acceleration. Vehicle power demand as well as vehicle torque are determined by PID driver controller. Each hybrid topology needs its individual power management. Therefore, for each topology individual power management is developed based on sustaining SoC strategy. These power management algorithms are combined using fuzzy rules to form a generalized power management. The power split ratio is the output of the generalized fuzzy logic-based power management. The results show that power split topology increases the fuel economy more than that by series and parallel topologies. Although the approach is a generalized power management controller, application of it to other topologies requires many modifications. A generalized fuzzy logic-based power management controller for three typical topologies of the HEV, namely parallel, series, power split is developed in [CMJJ10].

A new fuzzy logic controller based on travel distance information is developed in [AAF12]. The two inputs of the controller are SoC and vehicle power demand, the output of the controller is power split ratio. Considering the dependency of the SoC to the travel distance necessitates the need for using GPS data. The results show a comparable fuel consumption with engine on/off strategy. Comparison of the results shows substantial fuel consumption decrement. The presented results for a driving cycle show depletion of the accumulator at the end of the driving cycle and sustain requirement of SoC for the next driving cycle. Moreover, each driving cycle ends with braking so that brake energy is captured at the end of the driving cycle. Due to the point that brake energy losses are accepted and included, this approach cannot really be accepted as an hybrid power train approach.

2.4.3 Equivalent Consumption Minimization Strategy

The Equivalent Consumption Minimization Strategy (ECMS) is an instantaneously optimized power management strategy. In ECMS, the accumulator is considered as a subsidiary reversible fuel tank. Based on the ECMS strategy, the fluctuation of the energy level in secondary power source will be compensated by replacement

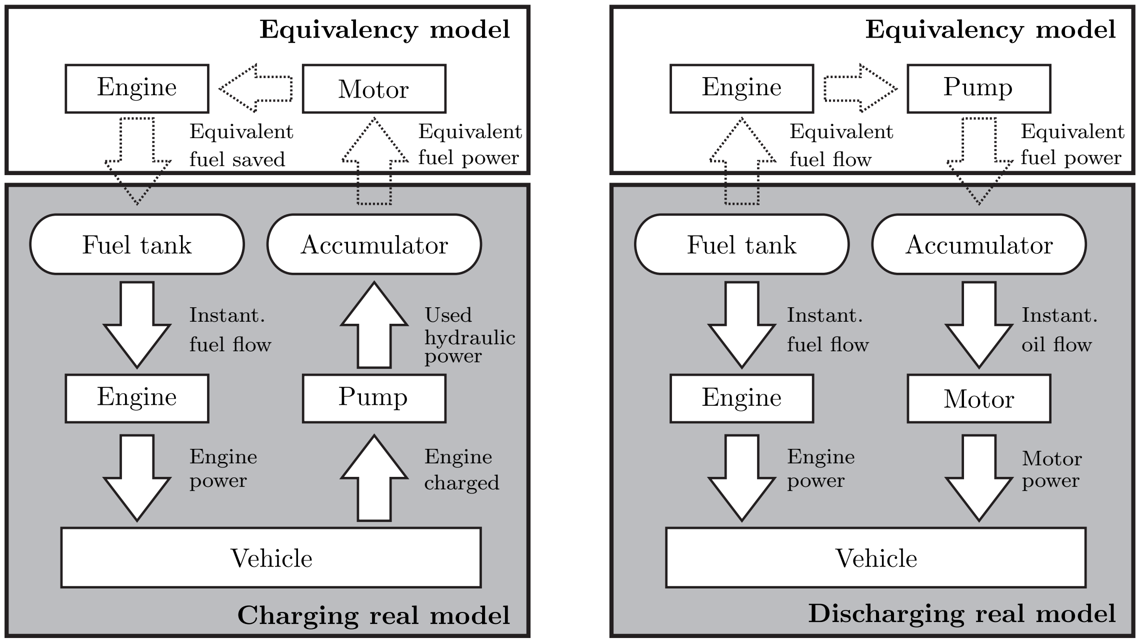

Figure 2.5: Power flow in the Equivalent Consumption Minimization Strategy (ECMS) algorithm in, (left) accumulator charge; (right) accumulator discharge [KS14]

of the equivalent fuel power in the future [PGD+00]. For equivalency of the fuel power and accumulator power, the instantaneous efficiency of the power line must be calculated as depicted in figure 2.5. Accordingly, the equivalent amount of fuel to compensate the used accumulator energy in discharging mode and equivalent amount of fuel to produce as much as power saved in the accumulator in charging modes is calculated. For this reason, energy losses at certain operation points of the system must be considered. Based on this definition, the power management is optimized by minimization of both engine fuel consumption and equivalent fuel consumption of the secondary power source. In contrast to other power management optimization methods, the control variable in ECMS is the equivalency factor. By definition, the equivalency factor is the relation between the used energy of the secondary power source to the power demand. Therefore, increase in the value of equivalency factor, increases the use of the auxiliary power source energy and vice versa.

Using the equivalency factor as the only control variable, proper charge and discharge of the accumulator considering variable boundary conditions cannot be guaranteed. In other words, accumulator may be overcharged or undercharged. To avoid this problem, using an iterative loop, the value of SoC is compared to the boundary values and if needed the equivalency factor is corrected. Due to difference between efficiency of the power transmission in charge and discharge operational modes, a method is developed to distinguish equivalency factor in charge and discharge modes of the secondary power source in [SBG04]. For an on-line application of this method, two different equivalency factors are converted to one variable which is the probability factor. The evaluation of the probability factor without using prediction methods results in on-line calculation of the equivalence factor for ECMS. However, the presented method increases substantially the performance and optimality of the power train, it is not applicable to real-time power management, because the power management is optimized for a specific driving cycle.

Performance of ECMS is preferred in comparison with the global optimization algorithms such as simulated annealing in [PGD+00, DGP+01] and DP in [SBG04]. An Adaptive Equivalent Consumption Minimization Strategy (A-ECMS) is developed in [MRS05] to increase the optimality of ECMS. Using GPS data and comparing with vehicle velocity, the equivalency factor can be adjusted instantaneously using this adaptive ECMS strategy. To decrease the computational load in order to apply the A-ECSM in real-time, the two dimensional adaptive problem is converted to an one dimensional one by substitution of a unique equivalency factor. The real-time implementation of this method is approved using vehicle velocity prediction method and GPS data for the evaluation of the road load. The results of A-ECMS are compared with DP-based power management as well as a conventional ECMS. The comparison shows less fuel consumption for A-ECMS relative to conventional ECMS. Furthermore, results justify the computational load decrement of the A-ECMS without alteration of the optimality.

A new strategy to improve the efficiency of the ECMS is presented in [LP06]. The improvement is brought about by relating the equivalent factor and the vehicle speed using more accumulator power. The results show relative fuel consumption reduction for city driving cycle, but not for the urban driving cycle. In [KB10], a new A-ECMS with predictive Dynamic Programming approach is developed which is called DP-ECMS. In this approach, power and torque demand of the vehicle are predicted. The gear ratio and SoC of the accumulator are considered as the two states of the controlled system to decrease fuel consumption. The comparison of the results with DP-based power management approves the near optimal fuel consumption of the new A-ECMS. Besides the robustness of this approach, its optimality is shown on comparison with a number of other A-ECMS approaches. However, high computational load of DP makes it inappropriate to be used for real-time power management. To overcome this drawback, an iterative method based on Pontryaginâs approach is developed in [KB11] called global Pontryagin optimization (GPO). Between different methods proposed to solve this problem, bisection method is proposed as the simplest and the most robust one. Comparison between the results of DP-ECMS and GPO for seven different driving cycles shows substantial decrement in computational load but no effect to fuel consumption reduction. Although low computational load of the GPO makes it appropriate in real-time power management, both GPO and DP-ECMS are off-line power management approaches and therefore not applicable.

In [GR06], the equivalency factor is estimated based on the recognition of driving pattern. The driving pattern is estimated using past drive patterns. Based on recognition of the driving pattern, the equivalency factor is controlled adaptively. In addition, a PI controller is utilized to sustain SoC close to nominal value. The results show good recognition of the driving cycle as well as SoC control. However, high computational load hinders the application of this method for real-time power management. Three A-ECMS, namely adaptive PI controller approach, discrete time method, and proportional controller, are compared together using SoC feedback [OS11]. The results confirm the robustness and optimality of the adaptive PI controller and discrete time method. Although, the results of driving cycle recognition approach are close to those resulting from model predictive approaches, the applicability of this method depends on the size of the accumulator. In other words, this method is valid in operational situations where the accumulator does not fully charge and discharge.

2.5 Real-time power management

Inaccessibility of the driving cycle in future is a crucial problem for the realization of real-time power management approaches. The assumption of prior-knowledge about the “driving cycle” is unrealistic and it is used just for off-line optimization of power management. To overcome this drawback, methods such as driving cycle recognition [LPJL04], and the use of GPS data [MWH07, KMS09] are developed and applied to real-time power management. However, they are based on the past and present vehicle velocity information. Furthermore, only local power management optimizations are realizable using these approaches. Finally, there is a time delay between current power demand signal and output controls of the power management.

2.5.1 Telematics technology-based predictive approach

Telematics technology is one of the new developments within the field of intelligent vehicle technologies. In general, this technology realizes the communication of the vehicles with other vehicles and environment. Using the information getting through this system, different intelligent systems such as vehicle tracking can be realized. Telematics technology can be used for adjustment of instantaneous vehicle velocity in urban environment [MWH07]. The comparison between telematics-equipped vehicle with conventional vehicles shows a significant reduction in fuel consumption [MWH07]. Also the efficiency of the telematics-equipped vehicle and HEV are comparable. Based on the information gathered from telematics technology, the vehicle velocity in the next partial time is provided [KMS09]. Combination of these information with the ECMS power management results in an on-line power management. The power management formulation is solved using sequential QP. Comparison of the results for different traffic preview lengths, show higher fuel economy in short time preview. The combination of the vehicle velocity and vehicle position to predict the power demand is presented in [BKS13]. It is shown experimentally that the position and velocity profiles of the stop and go vehicle such as garbage trucks follow a pattern. Based on this consideration, other factors such as driver behavior, road grade variation, and vehicle mass variation effect the power demand of the vehicle. The iterative learn prediction profile method calculates the vehicle power demand by comparison of driver behavior to the given data. At the stopping phase of the vehicle, using the GPS data, the next velocity and position profile for prediction of vehicle power demand is searched. Additional to predictive power management, a conventional power management is considered where the prediction of vehicle velocity is impossible. For application of predictive power management in real-time, synchronization of the predictive velocity and reference velocity is very important. If the data interval of the vehicle acceleration is larger than the predicted one, acceleration phase control will be applied in real-time instead of braking phase control. Therefore, power management does not operate appropriately.

2.5.2 Model predictive approach

Model Predictive Control (MPC) is a mathematical method for calculation of the system input trajectory to optimize the output of the system in the future. Based on the current dynamics of the system, the future of the system for a specific prediction horizon is predicted and the control inputs are calculated. The performance of the MPC drastically depends on the prediction window. The application of this method to hybrid power train can decrease the fuel consumption by prediction of the system inputs such as power split factor. If the computational load of this method is decreased, this method can be used as a real-time power management.

A two level nonlinear MPC-based power management is developed for a power split HEV [BVP+09]. To ensure simplicity, vehicle velocity and power distribution controllers are decoupled. In the supervisory or high level control, power demand of the vehicle is predicted using a linear model predictive algorithm while in low level control, the power distribution is optimized instantaneously. To increase the accuracy of the power management as well as the application of linear MPC, the internal model, which is used for vehicle velocity prediction, is linearized around the instantaneous operation points. The parameters of the MPC are adjusted using a rule-based controller. Although the results demonstrate significant decrease of fuel consumption in comparison with rule-based power management, the results are not validated and just simulation results are presented. Moreover, the efficiency of this MPC-based power management can be improved by optimal adjustment of MPC parameters. The same control scheme with small differences is used in [DASM10] to a series HHV. Engine torque, pump, and motor displacement ratios are the input variables of the system. The goal of the controller is optimal operation of the engine in drive mode. Tracking the desired vehicle velocity has more priority than fuel consumption reduction. Limiting the operation of the engine to a constant torque, consideration of constant efficiency for hydraulic elements, lack of optimal adjustment of MPC parameters are the problems associated with this MPC-based power management. In [DASM11] the relation between dwell time, accumulator size, and engine on and off switch frequency is investigated. Higher dwell time increases the engine durability. Moreover, larger dwell time requires larger accumulator capacity.

The performance of nonlinear MPC-based power management in association with an adaptive prediction time horizon is presented in [MS12]. If the predictive velocity matched well with the reference velocity, time horizon is increased and vice versa. Therefore, the computational load of the power management can be decreased adaptively while the performance and efficiency of power management are increased. Comparison of the MPC power management with DP as well as a conventional controller shows substantial improvement in both fuel consumption and vehicle performance. An integrated predictive power management controller is developed in [CB11]. In this approach, velocity prediction method is combined with ECMS supervisory control in order to optimize the fuel consumption. In this power management, the equivalency factor is the control variable. The weighting ratios of the objective functions are the inputs to the power management strategy. In addition, the equivalency factor controller is heuristic. Although, the velocity prediction method is not accurate, the simulation results for eight driving cycles justify the potential of the new integrated predictive power management to reduce fuel consumption. Moreover, comparison of the results with a non-predictive ECMS power management as well as a DP-based power management shows improvement in optimality of the power management respectively.

2.5.3 Stochastic model predictive approach

In contrast to the MPC, which uses prior knowledge about driving cycle, Stochastic Model Predictive Control (SMPC) uses random information about vehicle velocity for optimization of power management. Using a Markov chain, the distribution of the power demand in the future can be assumed based on previous experiences. The integration of the stochastic power demand prediction with DP is already explained. Because of the high computational load of DP, it is not implementable as real-time power management. However, SMPC uses supervisory optimization methods such as linear or QP with low computational load. Therefore, it is a real-time power management strategy.

In [RBC+10], SMPC approach is applied to a series hybrid power train as on-line power management. Using two level integrated power management approach, the power demand of the vehicle is predicted stochastically and power distribution is optimized in the second level. At each time step using all possible Markov states, all possible power demand distribution in the next step are generated iteratively. Based on the known power demand, power distributions is optimized using quadratic optimization approach. The results of SMPC power management are compared to two other MPC approaches, namely frozen-time MPC and prescient MPC. In frozen-time MPC, a prior knowledge about driving cycle is necessary while prescient MPC uses constant actual power demand for the prediction of power demand in future. Therefore, frozen-time MPC is a deterministic approach. Comparison of the results shows the optimality of prescient MPC, SMCP, and frozen-time MPC respectively.

Similar power management is applied to a HHV. The control variables are pump/motor displacement ratio as well as engine power variation. The objective functions are operation of the engine, vehicle performance, and the brake energy regeneration. Based on two operational modes of the engine, namely engine on and off, two predictive models are developed. The states of the prediction model are vehicle velocity, SoC, and output power of the engine. For simplicity, the complex model of the power train is linearized and discretized. Comparison of the results with frozen-time MPC, and prescient MPC verifies the results presented in [RBC+10].

2.6 Discussion and evaluation

Comparison of different power management approaches show that except optimization of the power management, computational load, ability to easily implement, and information about upcoming vehicle velocity are the other important criteria for realization of a real-time power management. The first problem associated with the implementation of power management to real-time systems, regardless of control strategy, is the lack of information about upcoming vehicle power demand. Additionally, backward model of the power train is needed to calculate power demand regarding the vehicle velocity. The complexity and non-linearity of the power train model must be simplified by linearization, and simplification of the model. Although the recommended methods are able to solve these problems theoretically, reduction of computational load is unavoidable for real-time implementation of power management. Use of powerful processors or simplification of e.g., predictive model to decrease the computational load, sequentially increase the cost and inaccuracy. Therefore, between simplicity and accuracy of the power train backward model a weighted balance has to be considered. Nevertheless, optimality of power management is usually sacrificed. Moreover, unavailability of experimental results or even possibility of comparison respect to practical oriented benchmarks results makes it impossible to judge about power management strategies. Different power management optimization strategies have their individual drawbacks. In Table 2.2, discussed power management algorithms are compared from the real-time applicability and optimality points of views. In real-time applicable power management, the calculation time is short enough that can be realized in real time steps. In contrast, on-line power

Table 2.2: Comparison of different power management algorithms [KS14]

Algorithm	**Optimal**	**Sub-optimal**	**On-line**	**Real-time**
DP-based	yes	no	no	no
GA-based	yes	no	yes	no
Rule-based	no	yes	yes	yes
Fuzzy logic	no	yes	yes	yes
ECMS	no	yes	yes	conditional
Telematic-based	yes	no	yes	conditional
MPC	yes	no	yes	conditional
Stochastic-based	no	yes	yes	conditional

management has the same structure as the real- time power management but the calculation time is longer. In conditional cases, the power management algorithm definitively cannot be applied as real-time power management. It depends on its ability to impose conditions, such as calculation load reduction which are discussed in detail.

Rule-based power management methods are based on predefined constant logical rules and real-time applicable. Performance and efficiency of rule-based controller depend on a wide variety of rules, logics, and conditions. Rule-based controller improves the optimal operation of the individual components without consideration of the overall efficiency of the power train. Near optimal point and near optimal line operation of the engine, on and off switching of the engine, reducing transient operation of the engine, engine constant speed with variable torque, SoC bound control, and maximum recapturing of braking energy are usual control strategies. Applicability of a specific strategy depends on the type of hybrid power train, Degree of Hybridization (DoH), size of the components, and driving cycle. To improve the efficiency and performance of the rule-based power management, controller adjustment is unavoidable. However optimization of power management depends on driving cycle. Therefore, generalization of a unique rule-based power management to all driving cycles and driver behavior is impossible. These types of power management are simple to implement. It can be applied as on-line and real-time power management. Nevertheless, it is not an optimal power management. Moreover, the time-delay between the feedback signal of vehicle velocity and output signal of power management is unavoidable. This drawback proves the superiority of predictive power management to the IOPM. All in all, rule-based as an instantaneous and sub-optimal control approach, is the usual real-time power management in context of hybrid power trains.

The type of optimization method which is used for development of power management of hybrid vehicles, depends on the availability of vehicle power demand.

Using prior-knowledge about driving cycle, both local and global optimization algorithms such as GA, DP, and ECMS without consideration of computational load, are applicable. A usual instantaneous power management strategy is ECMS and it is applicable as both off-line and on-line power management strategy. In contrast to rule-based controller, ECMS provides optimal power management. However, the method is very sensitive to the control parameters mainly equivalency factor. It is a robust power management in context of sustaining SoC. For this reason, SoC must be penalized using an additional objective function. The optimality of ECMS in off-line application is close to globally optimized power management approaches. Although different approaches such as A-ECMS are developed to increase the optimality of ECMS, their computational load is huge. Therefore, other such optimization methods, suffer lack of optimality due to simplicity. Model Predictive Control is proposed as on-line and real-time power management. Unlike rule-based, which is sub-optimal power management and such as ECMS, MPC-based power management is optimal power management. Same as other optimization methods, MPC leads to large calculation load. However, simplicity of the model decreases the complexity and computational load while decreasing the optimality.

As global optimization methods such as DP need more time for computational effort involved, mostly they are not applicable as real-time power management. Among all developed optimal power management methods, DP-based approaches show the most optimal power distribution. Therefore it can be used as benchmark for the evaluation of other power management methods. However, it is based on backward calculation of power distribution using deterministic driving cycle with mostly time consuming computational load.

Unlike numerical optimization methods, analytical optimization methods based on minimum principals such as Pontryaginâs minimum principle reduce the computational load substantially [MS12]. Nevertheless, formulation of a complex nonlinear MIMO hybrid power management for application of analytical methods requires model simplification. In this case, optimal model simplification methods may decrease unavoidable errors [RGB99], therefore, the result is sub-optimal. Moreover, it requires no prior knowledge about optimization horizon.

3 Modeling the hybrid hydraulic power train

Modeling of dynamic systems is a usual preferred approach strategy for studying its behavior. Even if the real system or a test rig is available, modeling and simulation are preferred. Presentation of a physical system using mathematical equations in order to realize a system dynamic behavior is defined as the modeling of the system [Oga02]. The mathematical equations are extracted using physical rules such as Newton's rules.

Simulation of system behavior consist of the numerical solution of mathematical equations represent the dynamics of the system. Using numerical or analytical methods, the response of the system to a specific input signal can be determined. Due to model simplifications, linearization, and discretization simulation results may differ from the real system. Therefore, the accuracy of the model has significant effects on controllers developed and applied to the system. For this reason, the results must be either validated through comparison with the results getting from the real system and/or test rig, or verified by comparing the published results and/or results getting from other simulation tools. For these reasons, one of the following actions is necessary to decrease the simulation error; parameter adjustment, change of the assumptions, change in the model, change of the solver toolbox or the adjustment of the solver.

In the following sections, models of the HHV sub-systems namely hydraulic transmission, engine, and vehicle are presented. Furthermore, the models of the key hydraulic components are presented and verified. Finally, a typical series hybrid hydraulic power train is modeled and its simulation results are verified.

3.1 Model structure

Typical modeling methods in the context of hybrid power train and based on the power flow direction in the system are forward-facing and backward-facing models. In backward-facing models power flow is considered from vehicle to engine while in forward-facing models like real system, power flow is considered from the engine to vehicle. Using backward-facing models of the hybrid power train, the main objective functions, here vehicle velocity tracking error and system efficiency can not be optimized simultaneously. Considering deterministic vehicle velocity, the optimization problem is reduced to an only power distribution optimization problem. However, using forward-facing models, both objective functions mentioned above can be optimized simultaneously. Simultaneous optimization of conflicting objective functions in a multi-mode system is more complicated than the their optimization separately. In figures 3.1 and 3.2, HHV backward-facing and forward-facing models are illustrated respectively. According to figure 3.1, the vehicle power demand is determined using the closed loop control system containing vehicle and driver models

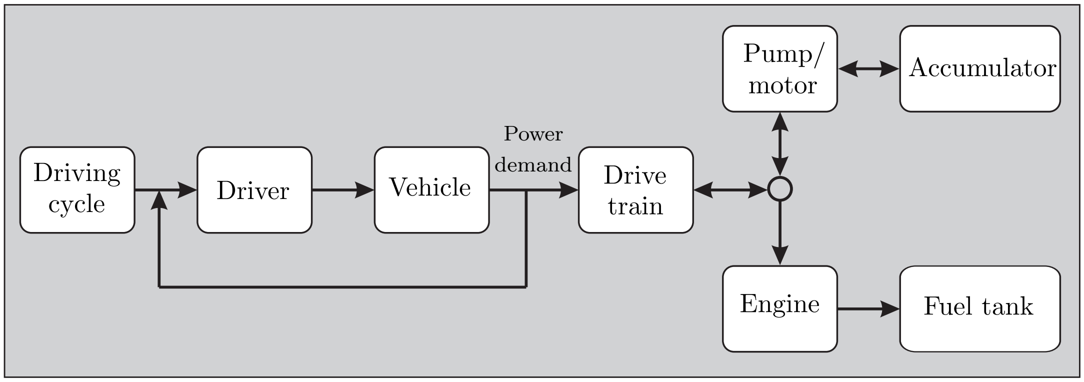

Figure 3.1: Backward-facing model

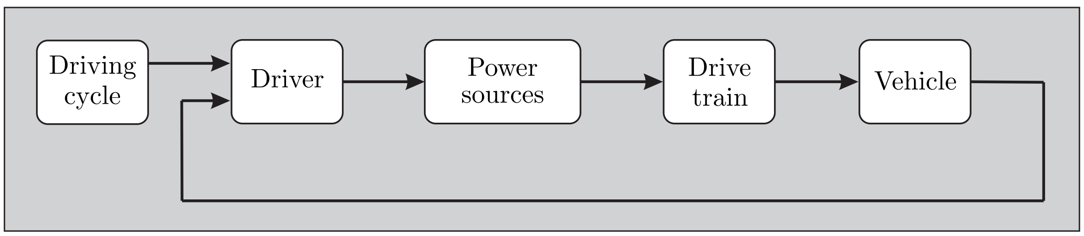

Figure 3.2: Forward-facing model

while driving cycle and load cycle are respectively the input and output of the control system. Based on the load cycle taken from control loop, power distributions between two power sources, namely engine and accumulator are optimized. The outputs of backward-facing approach are fuel consumption and accumulator State of Charge (SoC). According to figure 3.2, forward-facing model contains sub-systems in the form of a closed-loop control system. Therefore, using forward-facing models the conflicting objective functions explained above interact each other.

3.2 Hydraulic components models

In this section, the corresponding models of the hydraulic components used in HHV are presented. Additionally, key parameters, efficiency, and characteristics of the components are introduced.

3.2.1 Accumulator model

The accumulator is a temporary energy storage used in hydraulic systems in order to damp the pressure fluctuations or store energy in the form of Nitrogen gas compression in the accumulator. Compared to electric batteries, the hydraulic accumulator has larger power density while its energy density is smaller [DASM12]. Bladder type accumulator operates based on compression and expansion of Nitrogen gas. It contains a gas bladder filled with Nitrogen and a fluid chamber. By filling the oil chamber, bladder volume decreases and Nitrogen is compressed. Therefore, gas pressure as well as oil pressure in the accumulator are increased. The dynamics of the accumulator retrieved from ideal gas law, depends on the oil volume and initial pressure of the accumulator described by

$$\dot{V}_{oil} = Q_{acc}, \tag{3.1}$$

$$V_{gas} = V_{acc} - V_{oil}, \tag{3.2}$$

$$p_{acc} V_{gas} = nRT, \tag{3.3}$$

where Q_{acc} denotes oil flow into or out of the accumulator, V_{oil} oil volume in the accumulator, V_{acc} total volume of the accumulator, V_{gas} gas volume in the accumulator, p_{acc} pressure of the accumulator, T temperature of the gas in the accumulator, n number of moles, and R universal gas constant. Considering the gas compression process as adiabatic, right side of the equation 3.3 can be considered as constant. The flow rate of the accumulator is the function of accumulator and line pressure described by

$$Q_{acc} = \mathrm{sgn} C_d A_a \sqrt{\frac{2\left|p_{acc} - p_l\right|}{\rho}}, \tag{3.4}$$

and

$$\mathrm{sgn} \;=\; \begin{cases} +1 & p_{acc} < p_l \\ 0 & p_{acc} = p_l \\ -1 & p_{acc} > p_l, \end{cases} \tag{3.5}$$

where C_d denotes discharge coefficient, A_a accumulator throttling area, and p_l line pressure. Based on the above equations, accumulator pressure dynamics is described by

$$\dot{p}_{acc} = \frac{\mathrm{sgn} p_{acc}^2 C_d A_a}{nRT} \sqrt{\frac{2\left|p_{acc} - p_l\right|}{\rho}}. \tag{3.6}$$

Energy losses from the accumulator is due to heat transfer from accumulator to the surroundings. In [Pou90], it is demonstrated that isolation of the accumulator decreases the energy losses significantly. In case of fast charging and discharging of the accumulator, the state of the gas can be considered as adiabatic. In both processes, gas temperature remains constant [CB06]. Therefore, state of gas is determined by

$$p_1 V_1^k = p_2 V_2^k, \tag{3.7}$$

where k is the specific heat ratio. The value of k for isothermal process is 1 while for adiabatic process is 1.4. Another reason for energy losses in the accumulator is friction. Friction in the accumulator causes difference between gas and oil pressures. Although modeling of the frictional losses is not accurate, it can be considered proportional to the overall frictional energy losses described by

$$p_{gas} - p_{oil} = \pm \frac{p_{oil}}{2} \left(\frac{E_f}{E}\right), \tag{3.8}$$

where E_f represents the frictional energy loss, E accumulator energy, and p_{gas} and p_{oil} gas and oil pressure respectively [PBF92]. Positive and negative signs are related to the charge and discharge modes of the accumulator. In this contribution, accumulator charge and discharge are considered as adiabatic process. The amount of captured energy in the accumulator is directly related to the pressure of the included gas. The accumulator SOC monitors its instantaneous available energy. Here SOC is defined as the ratio of the current accumulator pressure to the maximum pressure of the accumulator

$$SoC = \frac{p - p_{min}}{p_{max} - p_{min}}, \tag{3.9}$$

where p_{max}, p_{min}, and p denote the maximum, the minimum, and the instantaneous pressure of the accumulator respectively.

3.2.2 Hydraulic pump/motor model

The variable displacement swash plate pump is the typical pump/motor used in HHV. Compared to a fixed displacement pump, swash plate pump has higher efficiency and it is able to operate under high pressure. In HHV, hydraulic motor performs the pump operation in energy regeneration mode. For realization of the vehicle reverse drive, the motor operates bi-directionally. In order to realize engine idle operation mode, motor freewheeling operation is necessary. Swash plate axial piston motor can realize the above operational modes. In this contribution, Wilson variable displacement pump/motor models are used for the representation of pump/motor models. The related equations are given in the sequence (equation 3.10 - equation 3.19) [PBF92]. Due to the mechanical coupling of the pump and engine as well as the motor and the transmission system, their dynamics can be integrated. For this reason, equivalent pump/motor inertia is integrated within vehicle and engine inertia. Therefore, the quasi-static models of pump/motor represent the transformation of mechanical energy to hydraulic energy and vice versa. A quasi-static model of the pump is given by

$$Q_p = x_p \omega_p D_p \eta_{vp}, \tag{3.10}$$

$$T_p = \frac{x_p \Delta p D_p}{\eta_{tp}}, \tag{3.11}$$

where Q_p denotes the oil flow of the pump, D_p maximum volume displacement of pump, x_p displacement ratio, ω_p pump speed, η_{vp} volumetric efficiency, η_{tp} mechanical efficiency, T_p pump torque, and Δp pressure difference across two chambers of the pump. The displacement ratio is the proportion of actual and maximum pump displacement ratios. Similarly, the motor model is described by The displacement ratio is the proportion of actual and maximum pump displacement ratios. Similarly, the motor model is described by

$$Q_m = \frac{x_m \omega_m D_m}{\eta_{vm}}, \tag{3.12}$$

$$T_m = x_m \Delta p D_m \eta_{tm}, \tag{3.13}$$

where Q_m and T_m denote motor oil flow and torque respectively. The pump/motor efficiencies include mechanical and volumetric efficiencies. Volumetric inefficiency of the pump/motor is due to internal leakages and cavitation while mechanical inefficiency is mainly due to friction of the pistons in the cylinder as well as hydrodynamic losses. Both mechanical and volumetric efficiencies are the functions of pressure difference, displacement ratio, and rotational speed. The mechanical efficiency of the pump is defined by

$$\eta_{tp} = \frac{1}{1 + \frac{C_v S}{x_p} + \frac{C_f}{x_p} + C_h x_p^2 \sigma^2}, \tag{3.14}$$

in which C_v denotes the viscous loss coefficient, C_f the frictional loss coefficient, and C_h the hydrodynamic loss coefficient [PBF92]. The volumetric efficiency of the pump is described by

$$\eta_{vp} = 1 - \frac{C_s}{x_p S} - \frac{\Delta p}{\beta} - \frac{C_{st}}{x_p \sigma}, \tag{3.15}$$

where C_s denotes the laminar leakage coefficient, C_{st} the turbulent leakage coefficient, β the fluid bulk modulus, and S and σ are the dimentionless values described by

$$\begin{aligned} S &= \frac{\mu \omega_p}{\Delta p}, \\ \sigma &= \frac{\omega_p D_p^{1/3}}{\sqrt{2\frac{\Delta p}{\rho}}}. \end{aligned} \tag{3.16}$$

Here ρ denotes the oil density. The mechanical and hydraulic efficiencies of the motor are described by

$$\eta_{tm} = 1 - \frac{C_v S}{x_m} - \frac{C_f}{x_m} - C_h x_m^2 \sigma^2, \tag{3.17}$$

$$\eta_{vm} = \frac{1}{1 + \frac{C_s}{x_m S} + \frac{\Delta p}{\beta} + \frac{C_{st}}{x_m \sigma}}, \tag{3.18}$$

where η_{tm} denotes the mechanical efficiency and η_{vm} the volumetric efficiency of the motor [PBF92]. The overal efficiency of the pump/motor is the ratio of the output energy to the input energy given by

$$\eta = \frac{E_{output}}{E_{input}} = \eta_t \times \eta_v. \tag{3.19}$$

As described, mechanical and volumetric efficiencies depend on different parameters. In this contribution, except individual parameters, namely pressure difference, displacement ratio, and rotational speed, the other parameters are considered to be constant. In the following section, effects of these variables on the pump efficiencies are presented. The coefficient related to pump efficiency are taken from [PBF92]. They are calibrated for application in industrial pump considering the operational conditions proposed in [BR14a] .

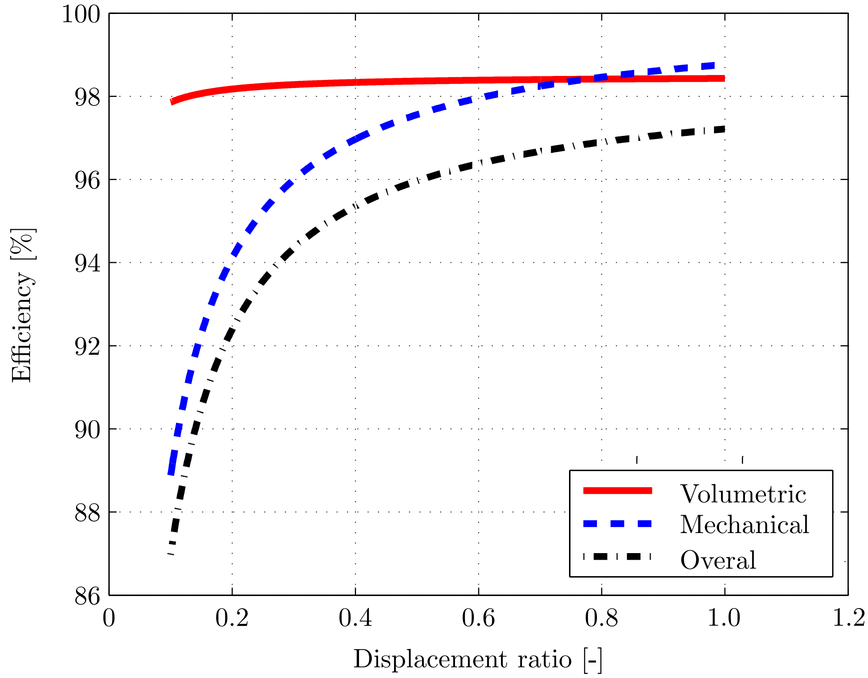

Figure 3.3: Effects of pump displacement ratio on the pump efficiencies (ω_p = 2000 rpm, p_p = 25 Mpa)

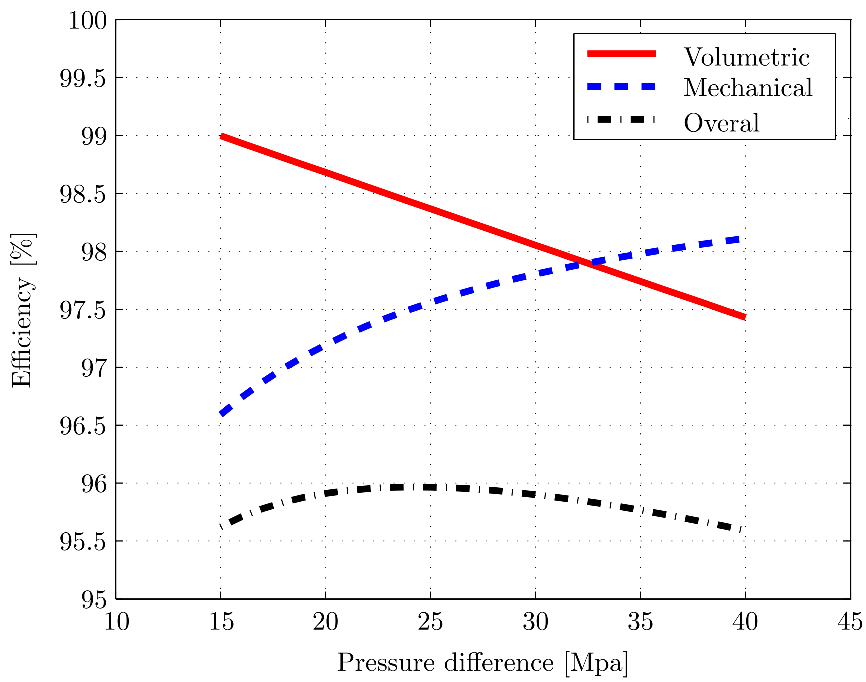

Figure 3.4: Effects of pump pressure difference on the pump efficiencies (x_p =1, ω_p =2000 rpm)

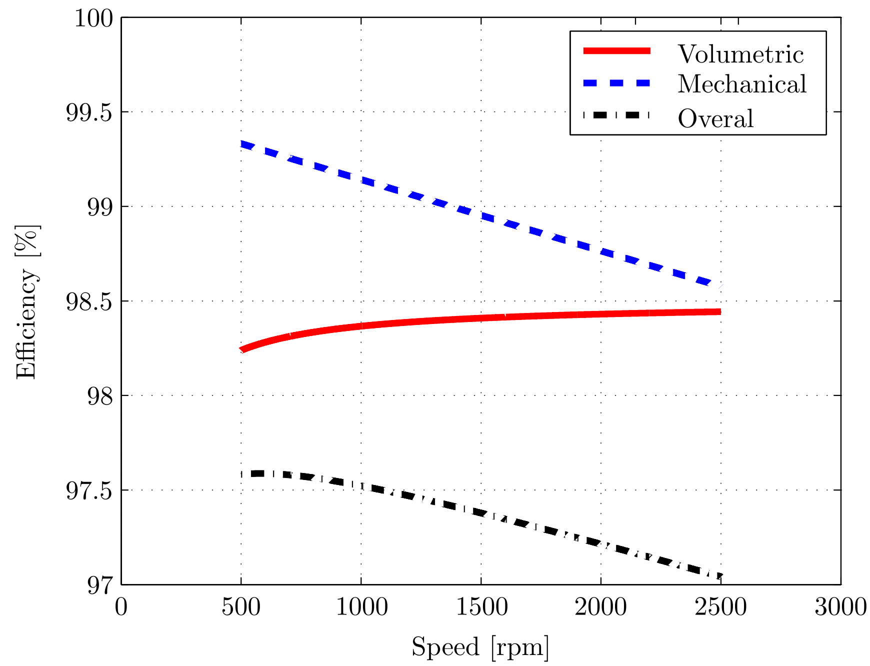

Figure 3.5: Effects of pump speed on the pump efficiencies (x_p =1, p_p =25 Mpa)

According to the results presented in figures 3.3 to 3.5, efficiencies of the pump significantly depend on related individual variables. According to figure 3.3, mechanical efficiency is more sensitive to displacement ratio than volumetric efficiency. By increasing of displacement ratio, both volumetric and mechanical efficiencies increase. Therefore, pump operation at large loads is more efficient. The effect of the pump pressure difference across two chambers on the pump efficiencies is shown in figure 3.4. Unlike mechanical efficiency, volumetric efficiency decreases by increasing of pressure difference. In figure 3.5 variation of the pump efficiencies in relation to pump speed are illustrated. Accordingly, in high speeds pump operation, its mechanical efficiency decreases in comparison to operation in low speeds. However, pump volumetric efficiency in high speeds pump operation is more than in low speed. The reason is the increase of the pump leakages in low speeds pump operation. It becomes clear that for system efficiencies improvement, optimal operations of pump/motor are unavoidable.

3.2.3 Pressure relief valve model

The task of the pressure relief valve (PRV) in hydraulic systems is the control of the system high pressure to prevent the hydraulic system from damage, overheating of the oil, and energy wastage. Direct operated PRV consists of a spool, spring, housing, and an adjusting screw as depicted in figure 3.6. System pressure acts against the spool and for pressure values larger than the valve set pressure, the

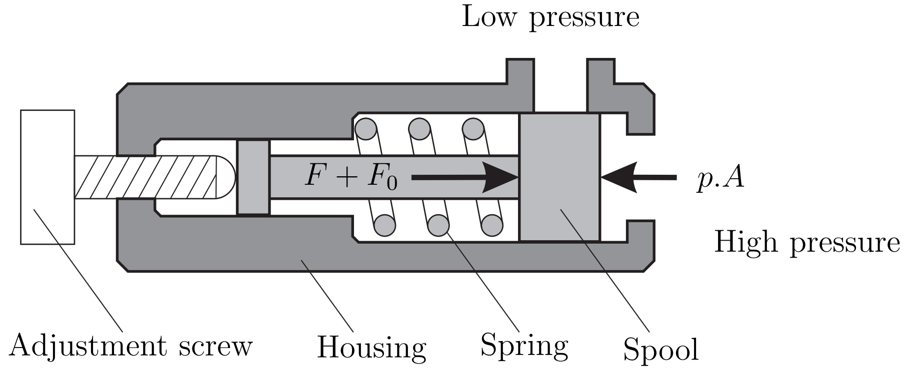

Figure 3.6: Schematic representation of pressure relief valve

spool moves. Therefore, oil is bypassed through the valve and system pressure set back. Minimum and maximum set pressure values are the key parameters of the PRV. The minimum set pressure depends on the pretension force of the spring while the maximum set pressure or maximum operational pressure depends on the maximum bypass flow through the valve. The initial force of the spring can be adjusted manually with adjustment screw. Therefore, the minimum set pressure value can be adjusted for different operational conditions. To keep the line pressure less than maximum operational pressure of the valve, difference between input and output oil flow in high pressure line must be equal or smaller than maximum bypass flow of the valve. The model of the PRV consist of spool dynamics and valve flow rate [Mer67]. The second order equation corresponding spool dynamics is described by

$$m_p \ddot{x}_p = p_l A_p - F_0 - k_s x_p - d_s \dot{x}_p, \tag{3.20}$$

where m_p denotes the mass of the spool, x_p the position of the spool, A_p spool area, F_0 pre-tension force, k_s spring coefficient, and d_s damping coefficient. It becomes clear that dynamics of the spool not only depends on valve parameters, but also on the line pressure dynamics. The bypass flow through the valve is proportional to spool position described by

$$Q_v = C_d A_v x_p \sqrt{\frac{2p_l}{\rho}}, \tag{3.21}$$

in which Q_v is the valve bypass flow and A_v the valve throttling area. It can be seen that the bypass flow depends on spool position and valve dimension. The verification of the valve model is presented in the last section of this chapter.

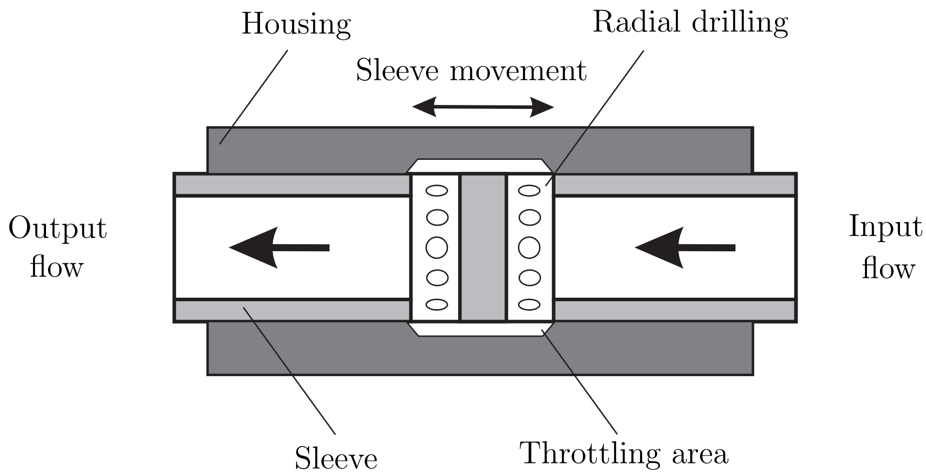

Figure 3.7: Schematic representation of throttling valve

3.2.4 Throttle valve model

The task of throttle valve in HHV is to control oil flow rate in order to regulate the accumulator power. In figure 3.7, the structure of throttle valve is shown. It consists of housing and sleeve adjuster. The throttling area is proportional to sleeve position. Therefore, valve flow is also proportional to sleeve position. The functionality of throttle valve in hydraulic systems is equivalent to resistance in the electric systems. Pressure difference as well as valve maximum flow are the key parameters of valve. On one hand, the oil flow through a fully opened throttle valve is the function of pressure difference in both side of the valve. On the other hand, the resistance of the valve causes energy loss. Therefore, pressure difference must be high enough to overcome the resistance of the valve. For this reason, minimum operational pressure is defined for throttle valve. Same as PRV, the flow rate through the throttle valve is governed by equation 3.21.

3.2.5 Connecting lines model

Line pressure in hydraulic pipes drops due to oil compression, line pressure, and friction. In addition to the dimensional parameters of the connecting lines and characteristics of oil flow, friction coefficients depend on the flow velocity. At low flow speeds, oil flow is laminar while in high flow speeds, it is turbulent. The difference between laminar and turbulent oil flow is distinguished by Reynolds number. Therefore friction model for laminar and turbulent flow are determined by two relations proposed in [MM80]. Considering laminar oil flow and lumped friction force in connecting lines, dynamic model of the connecting line is described by

$$\dot{p}_l = \frac{\beta}{V_h}(Q_{in} - Q_{out}), \tag{3.22}$$

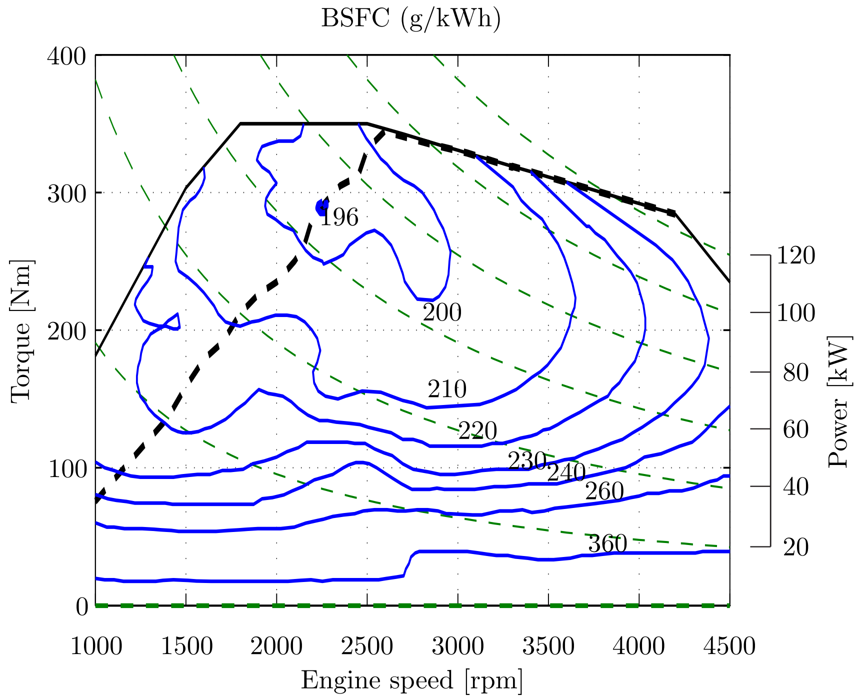

Figure 3.8: Engine efficiency map [Dat13]

where β denotes the Bulk modulus, V_h housing volume, Q_{in} input flow, and Q_{out} output flow of the line.

3.3 Engine model

In this section, the corresponding model of the engine as the primary power source of the HHV is presented. The subsystems of the engine are fuel system, intake system, injection system, cooling system, etc. [KN05]. The new engine technologies contain several controllers such as lambda control, idle speed control, knock control, etc. Modeling and simulation of the engine considering all subsystems and controllers increase the complexity of the problem. However for the purpose of power management strategy development a reduced order model of the engine may be appropriate. Therefore, using an appropriate simple engine model decreases the computational load. Nevertheless, the simplification of the model has some disadvantages. In this contribution, the simplified quasi-static model of the engine is used for the modeling of the engine as depicted in figure 3.8. The thermodynamic efficiency of the engine η_f, is defined as the relation between output and input power of the engine described by

$$\eta_f = \frac{P_e}{\dot{m}_f H_f} = \frac{T_e \omega_e}{\dot{m}_f H_f}, \tag{3.23}$$

where H_f denotes the specific energy of the fuel, m_f fuel mass, P_e effective engine power, T_e engine output torque, and ω_e engine speed.

The quasi-static model of the engine contains BSFC values of the engine at each engine operation point depending on engine torque and speed. In figure 3.8, the BSFC map of the engine used in this thesis is presented. It corresponds to a diesel engine with maximum power of 125 kW and maximum torque of 350 Nm at the speed of 2500 rpm [Dat13]. The maximum output torque of the engine is 350 Nm, which is achievable at engine speeds between 1750 and 2500 rpm. In figure 3.8, the black dashed line represents the minimum BSFC values at each engine output power. Operation of the engine at 2250 rpm and output torque about 285 Nm has the minimum BSFC value. This engine operation point is called sweet spot. Therefore, shifting the engine operation point close to minimum BSFC curve, particularly close to sweet spot point is the most important engine control strategy in the context of HHV [LFL+04].

3.4 Vehicle model

Power demand, traction force, and friction, are the subjects of vehicle longitudinal dynamics. Although, both lateral and vertical vehicle dynamics effect the vehicle traction forces, the contribution of lateral dynamics relative to longitudinal dynamics is negligible. For this reason, vehicle longitudinal dynamics is considered for the simulation of vehicle tractive force. Consideration of the tractive forces on the front or rear axle, and difference between rear and front axle braking forces have different effects on the traction and braking characteristics of the vehicle. Nevertheless, in this contribution, vehicle is modeled as a lumped mass with one degree of freedom. The main resistance forces applied to the vehicle, are aerodynamic drag force, rolling resistance force, and gradient resistance force. The aerodynamic drag force is described by

$$F_{drag} = \frac{1}{2}\rho_a C A v^2, \tag{3.24}$$

in which ρ_a denotes air density, C vehicle drag coefficient, A vehicle front area, and v vehicle velocity. The rolling resistance force depends on vehicle mass, road slope, and rolling resistance described by

$$F_{rolling} = M_v g C_{rr} \cos\alpha, \tag{3.25}$$

in which M_v denotes the vehicle mass, C_{rr} the rolling resistance coefficient, g gravitational acceleration, and α the road slope. Finally, the gradient resistance force results as function of road slope described by

$$F_{grad} = M_v g \sin\alpha. \tag{3.26}$$

Table 3.1: Vehicle parameters

Variable	Physical meaning	Value	Unit
A	Vehicle front area	4	m^2
C	Vehicle body drag coefficient	0.5	-
C_{rr}	Rolling resistance	0.008	-
M_v	Vehicle mass	1629	kg
R_w	Wheel radius	0.402	m

Therefore, the vehicle longitudinal dynamics containing tractive forces and resistance forces is described by

$$M_v \ddot{x}_v = \frac{T_m}{R_w} - \frac{1}{2}\rho C A \dot{x}_v^2 - M_v g C_{rr} \cos\alpha - M_v g \sin\alpha, \tag{3.27}$$

where x_v denotes vehicle displacement, R_w the wheel radius, and T_m the traction torque. In this thesis, a standard sedan vehicle is considered for application of HHV. The parameters of the vehicle model are presented in table 3.1.

In real-time, the vehicle parameters are not constant during vehicle drive. For instance, vehicle mass of the garbage cars changes due to loading and unloading of the vehicle. Moreover environmental effects such as wind velocity, road slop, etc. also vary during vehicle drive. All these parameters influence the vehicle power demand. However in this contribution, parameters related to the road conditions, environmental effects, and vehicle mass are considered as constant.

3.5 Verification of the components models

In this section, the model of important components are verified using comparison of numerical simulation results. For this reason, model simulation results are compared with the simulation results of the reference models developed in Hopsan software [HOP14]. Hopsan is a software developed for modeling and simulation of hydraulic, electric, mechanic and pneumatic systems. Here, the model of the key components, namely accumulator and pressure relief valve as well as typical series HHV are verified. The simulation conditions, as well as assumptions are explained and the results are presented and discussed.

3.5.1 Verification of the accumulator model

For the verification of the accumulator model, simulation results of repetitive charge and discharge of the accumulator using sinusoidal oil flow with three frequencies are

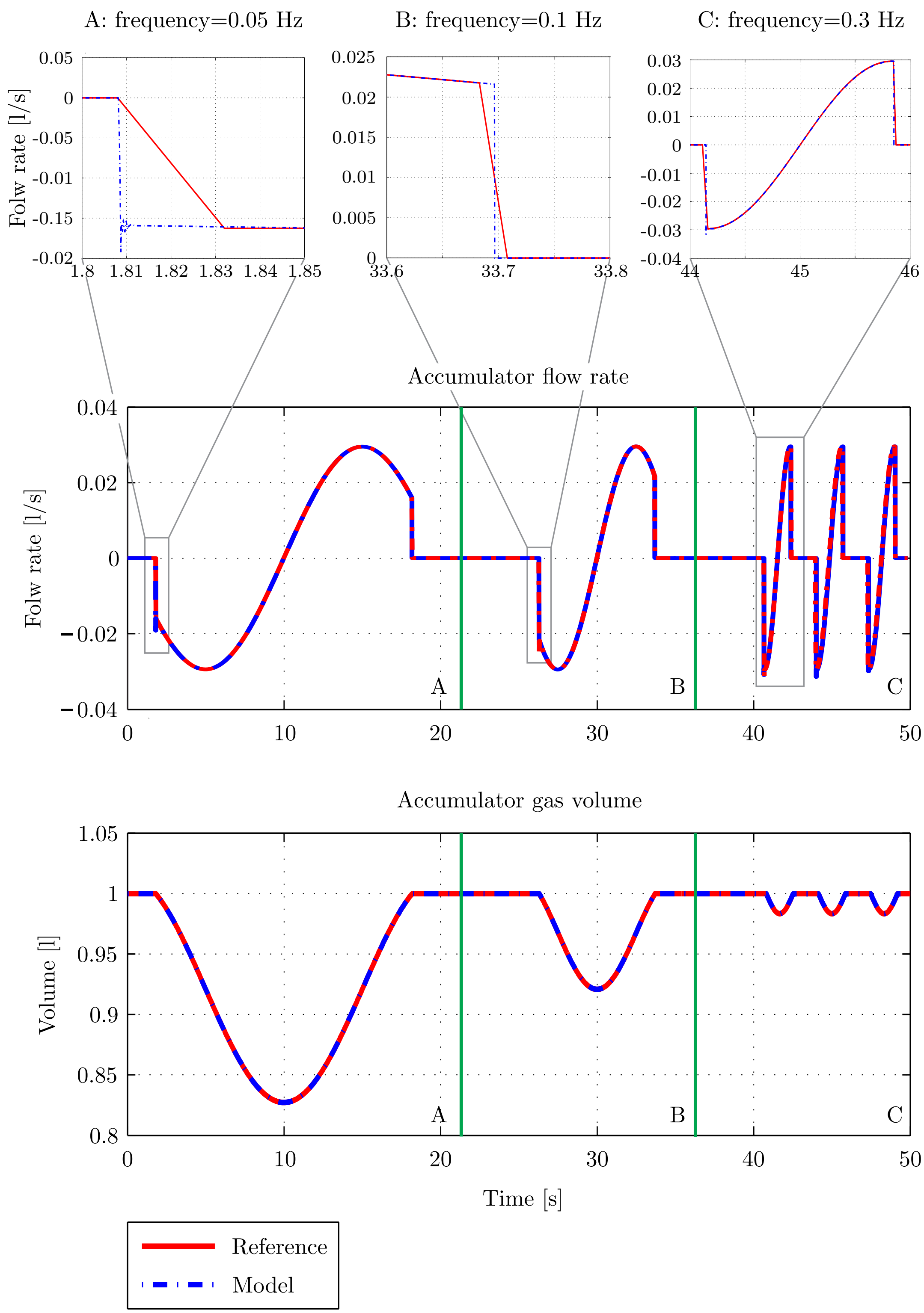

Figure 3.9: Accumulator oil flow and gas volume verification

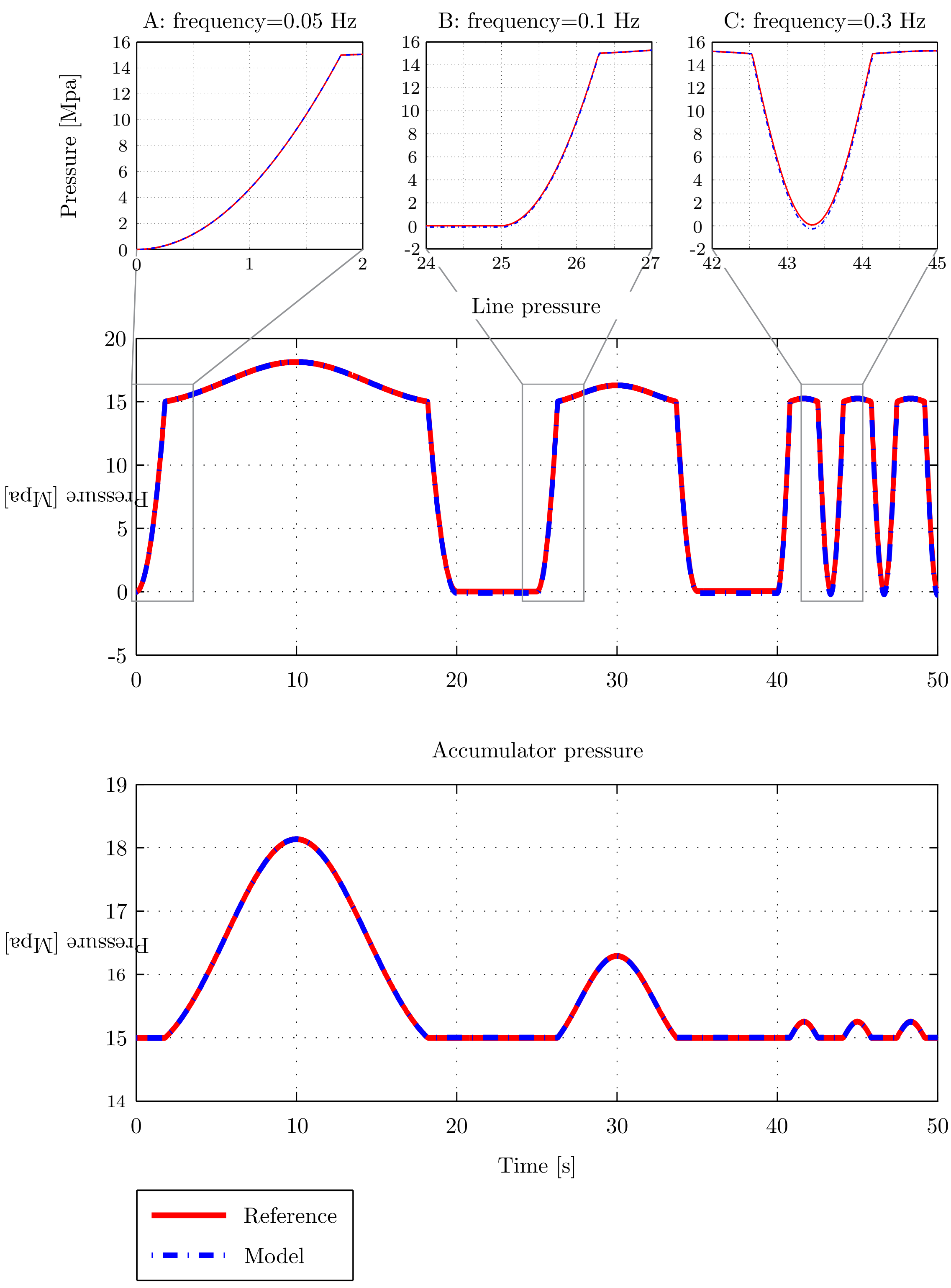

Figure 3.10: Accumulator and line pressure verification

Table 3.2: Accumulator model parameters

Variable	Physical meaning	Value	Unit
Q_{input}	Amplitude of input flow	0.03	l/s
f_1	First frequency of input flow excitation	0.05	Hz
f_2	Second frequency of input flow excitation	0.1	Hz
f_3	Third frequency of input flow excitation	0.3	Hz
p_{max}	Maximum accumulator pressure	20	MPa
p_{min}	Minimum accumulator pressure	15	MPa
p_{acc}	Initial accumulator pressure	15	MPa
V_{acc}	Accumulator volume	1	l
β	Bulk modulus	1000	MPa
V_h	Housing volume	1	l

used. The dynamics of the accumulator is modeled using equations 3.1 to 3.9. Also the dynamics of the main line pressure is modeled using equation 3.22. The parameters as well as initial and boundary conditions are identified using reference model in Hopsan. Due to sinusoidal variation of oil flow in the main line, the pressure in the main line changes dependently. The difference between line pressure and accumulator pressure values causes accumulator flows as shown in equation 3.5. As a result, accumulator is charged and discharged in accordance to the main accumulator flows.

The input of this simulation is the oil flow in the main line. The parameters of the accumulator model are presented in table 3.2. Accumulator flows, accumulator gas volume, accumulator pressure, and main line pressure are measured and compared to the results of the Hopsan model as reference. The comparison between the simulation results in Hopsan and developed model are shown in figures 3.9 and 3.10. The minimum operational pressure of the accumulator is considered to be 15 Mpa. Hence, if the main line pressure is less than 15 Mpa, accumulator flow is zero. Negative flow in figure 3.9 represents charge of the accumulator while positive flow represents discharge of it. Reverse fluctuation of gas volume in charge and discharge modes compared to the accumulator flow in figure 3.9 represents compression and expansion of the gas. Compression of the gas volume in amount of 7.5% results up to 1.25 Mpa increment of accumulator pressure as depicted in figure 3.10. In figure 3.10, it is shown that the line pressure significantly increases before hitting the minimum operational pressure of the accumulator. However in accumulator charging mode, the line pressure decreases till the accumulator flow cut off. Good adaptation of the simulation results of developed model and reference model for three frequencies justifies the integrity of the accumulator model.

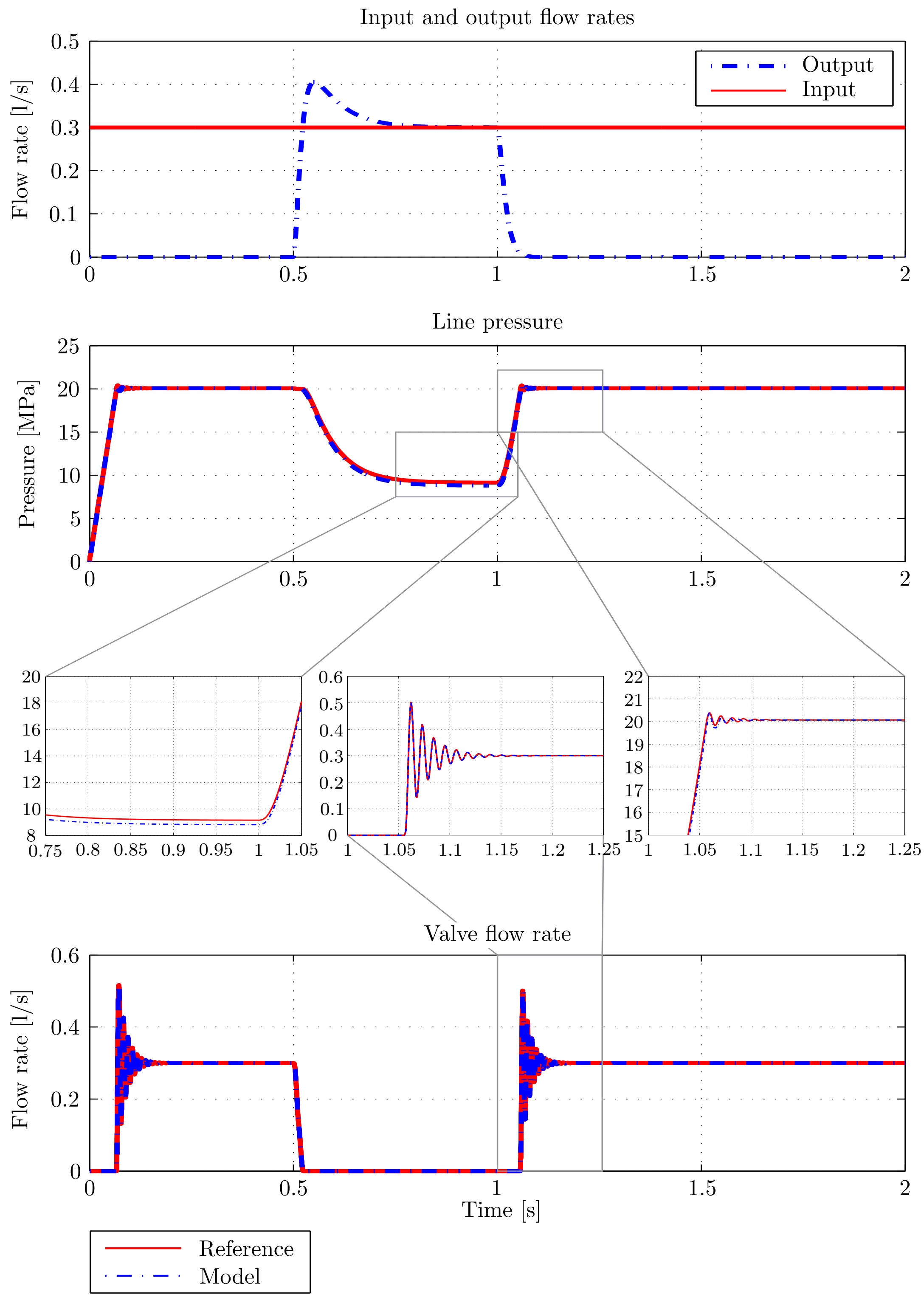

Figure 3.11: Dynamic behavior of the pressure relief valve verification

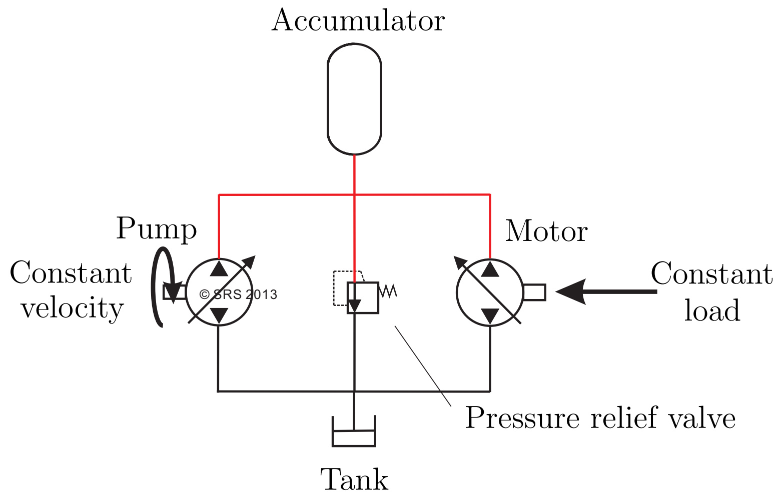

Figure 3.12: Topology of series hybrid hydraulic power train

3.5.2 Verification the pressure relief valve model

For verification of the PRV model, stepwise input flow rate to the main line is used. The dynamics of the PRV is modeled using the equations 3.20 and 3.21. Same as last section, dynamics of the pressure line is modeled using equation 3.22. All the valve parameters are not accessible in the supplier documents. Therefore, determination of the valve parameters specifically, damping and spring coefficients are challenging. For this reason, the PRV is verified qualitatively. Therefore, parameters of the valve are identified from the Hopsan model. Non-deterministic parameters are estimated using parameter estimation method. For this reason, the unknown parameters are identified using GA optimization method. In this iterative optimization loop, an effort is made to minimize the difference between model response and reference by varying the unknown model parameters. The stepwise variation of oil flow in main line causes pressure change in main line. In case of pressures larger than the valve set pressure, oil flow is bypassed through the valve.

In the first graph of figure 3.11, input and output oil flows in the main line used for this simulation are shown. In the second graph of figure 3.11, line pressure increases up to the valve set pressure value and oil is bypassed through the valve. In case of pressures larger than the valve set pressure value, the spool moves and the valve opens to bypass the additional oil. Therefore, line pressure decreases as depicted in the last graph of figure 3.11. Simulation results show good adaptation of the model and reference model of the valve. Nevertheless, in the third graph of figure 3.11 undesirable diversity of the results due to inaccuracy of the valve parameters identification are detected.

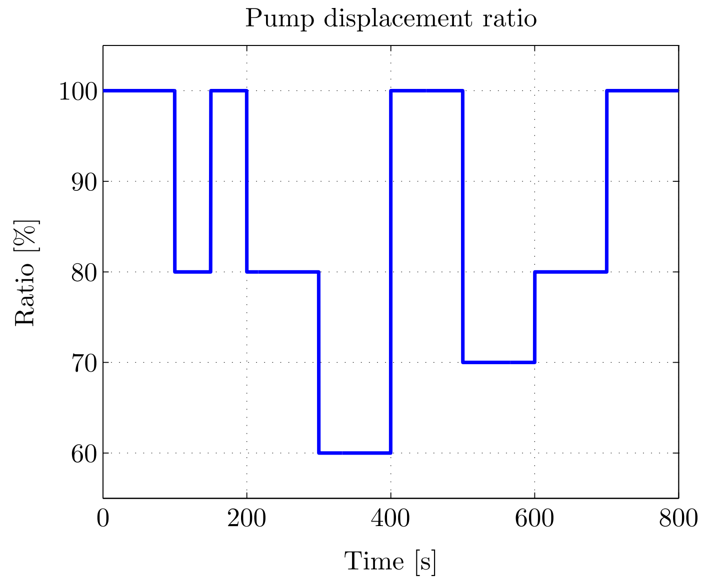

Figure 3.13: Arbitrary pump displacement ratio (input)

3.5.3 Verification of the hybrid system

After verification of the individual models of the components, the forward-facing model of the power train containing all the components is verified. The SHHV topology is modeled with the architecture shown in figure 3.12. This topology consists of a variable displacement pump, a variable displacement motor, an accumulator, and a pressure relief valve. Typical HHV topologies are investigated in chapter 6. In the simulation, pump speed is considered to be constant while an arbitrary input signal is applied to the displacement ratio of the pump as depicted in figure 3.13. The integrated components are verified with the corresponding input and output signals of the model and reference. The simulation results are depicted in figures 3.14 to 3.16.

In figure 3.14, high pressure line is shown. Line pressure is one of the system state which effects the dynamic response of the SHHV. A small offset which may be due to the elimination of temperature effects in high pressure line can be neglected. Accumulator pressure and oil flow rate are shown in figure 3.14 and 3.15 respectively. With a small difference in the pressure, model adapts satisfactorily to the reference. Moreover, fluctuation of the accumulator pressure due to oil flow fluctuation is reasonable. Due to the direct connection of the accumulator to high pressure line, accumulator pressure is in balance with the high pressure line. It is justified using the comparison of the graphs shown in the figure 3.14.

Oil flow rate of the pump and motor are depicted in figure 3.15. A small offset between model and reference in steady state parts is detectable. The difference is due to the large number of unknown parameters related to the efficiency of the pump model. Exact identification of the parameters related to the pump and motor efficiencies is impossible. The set pressure of the PRV is 40 Mpa. In order to use the whole range of the line pressure domain, the maximum valve bypass flow is considered to be equal or larger than maximum flow rate of the pump. The bypass flow rate of the PRV is shown in figure 3.16. A small offset between model and reference is detected. The difference is due to the offset between line pressure values of the model and reference. The comparison of the graphs in figures 3.14 and 3.16 show that the PRV is active in pressure values larger than set pressure.

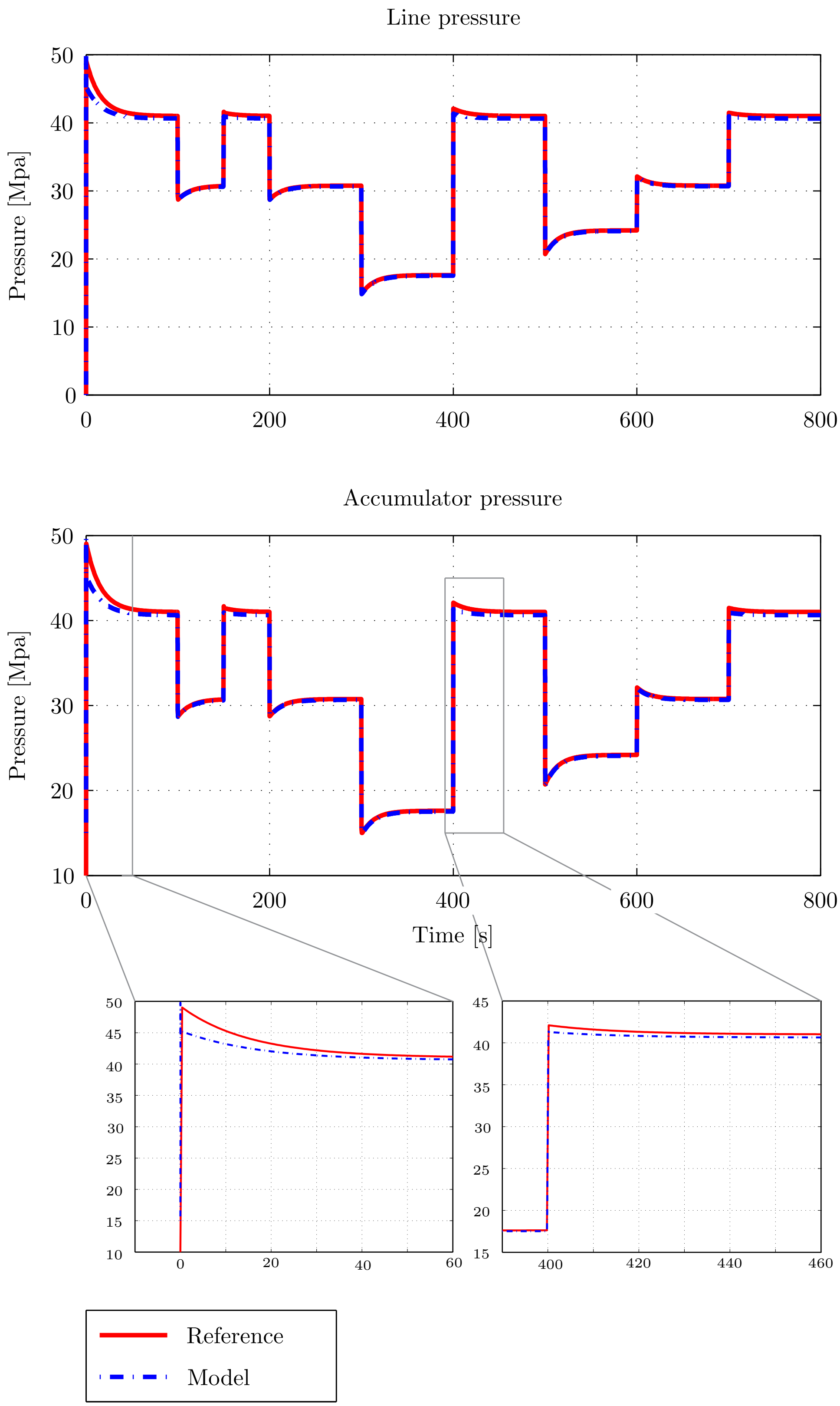

Figure 3.14: Pressures in series hybrid hydraulic power train verification

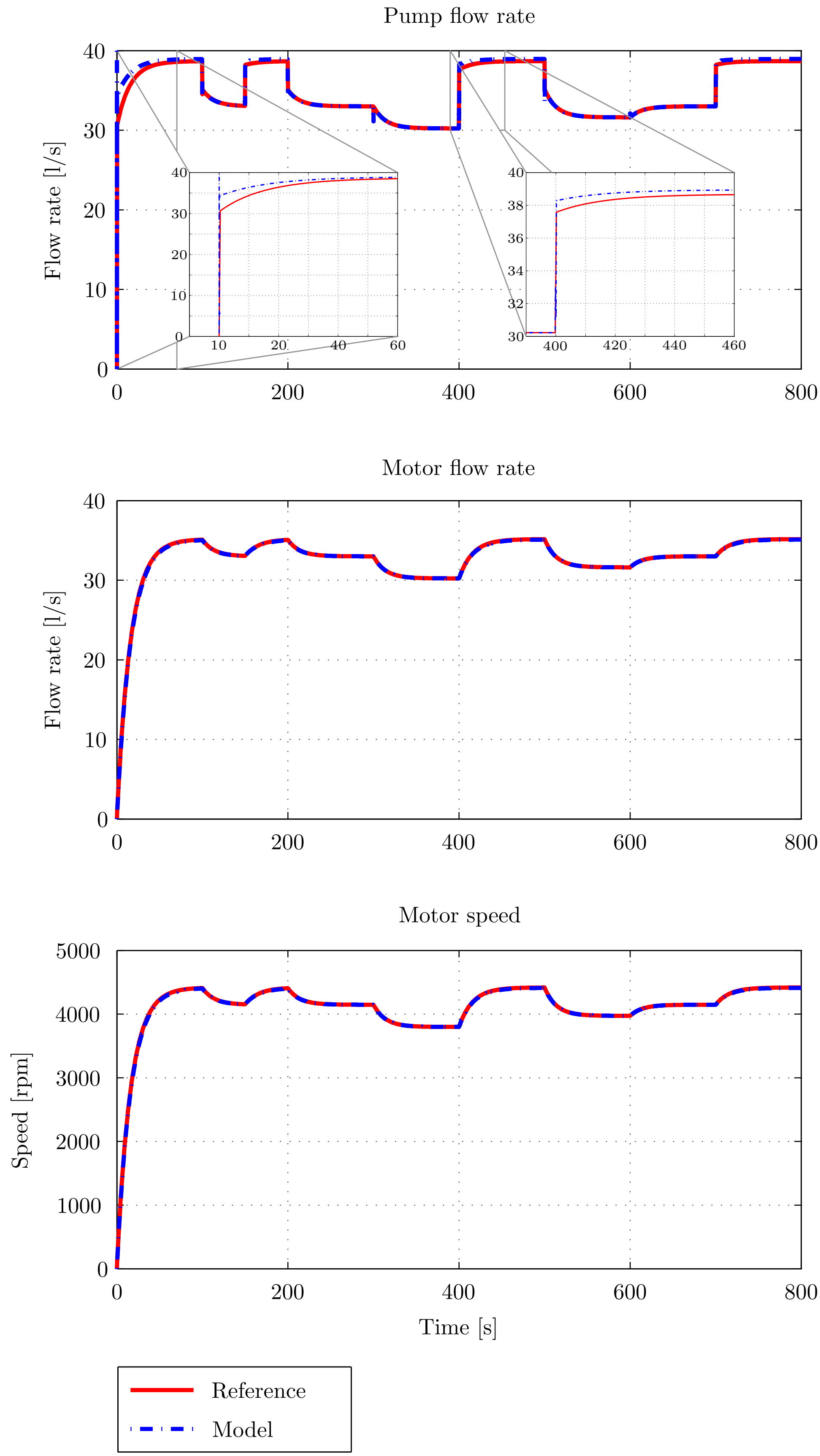

Figure 3.15: Flow rates in series hybrid hydraulic power train verification

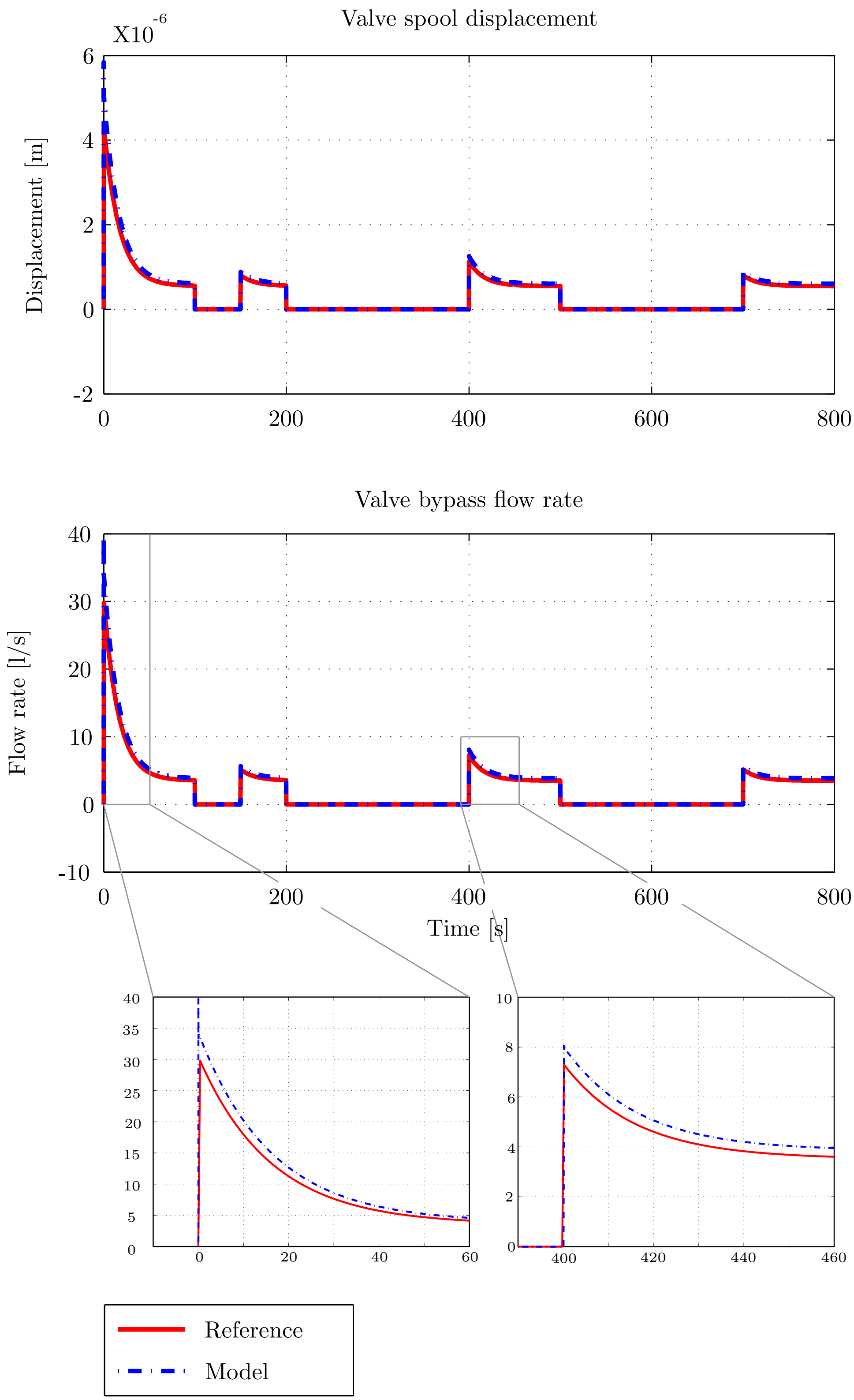

Figure 3.16: Valve dynamics in series hybrid hydraulic power train verification

4 Pressure control strategies for series hybrid hydraulic vehicles

Besides system topology design, element size optimization, and parameter adjustment, power management is other central aspect in HHV context. Operational characteristics of the HHV are directly affected by the control concept of the system. In the context of HHV controller design, supervisory control of the power distribution, and control of subsystems are the main aspects. Power management controls the power distribution between power sources to overcome demanded power somehow the fuel consumption, vehicle performance, emission, etc. are improved. On the other hand, subsystems control strategies deal with control of individual subsystems such as engine. However, both control strategies are internally dependent. In some control strategies, both power management and subsystems are controlled and optimized at the same level. Within other control strategies, power management and subsystem controller are optimized in two levels. Typical power management strategies are studied and evaluated in chapter 2. In this chapter, three rule-based control strategies namely, engine on/off, accumulator depleting, and smooth engine power are designed and applied to a SHHV. The optimal control parameters are selected based on the simulation results. Further, the performance and efficiency of the control strategies are compared and evaluated. Some parts of this chapter are published [KSeda].

4.1 Subsystems control strategies

4.1.1 Engine control strategies

Direct coupling of the hydraulic pump and engine in SHHV enables the engine to be optimally controlled. Two references for optimal control of the engine are engine optimal operation line (EOOL) and engine optimal operation point (EOOP) as depicted in figure 4.1. The EOOL contains the most efficient operation points of the engine at each level of engine output power, while the EOOP is the most efficient operation point of the engine. In other words, at these two references, the fuel amount for producing 1 kW power is less than the other engine operation points.

The first engine control strategy is based on shifting the engine operation points close to the EOOL as depicted in figure 4.1. The efficiency of hydraulic pump is the function of pump displacement ratio, line pressure, and engine velocity. It is described by

$$\begin{aligned}\eta_{tp} &= f_1(x_p, p, \omega_e),\\ \eta_{vp} &= f_2(x_p, p, \omega_e),\end{aligned} \tag{4.1}$$

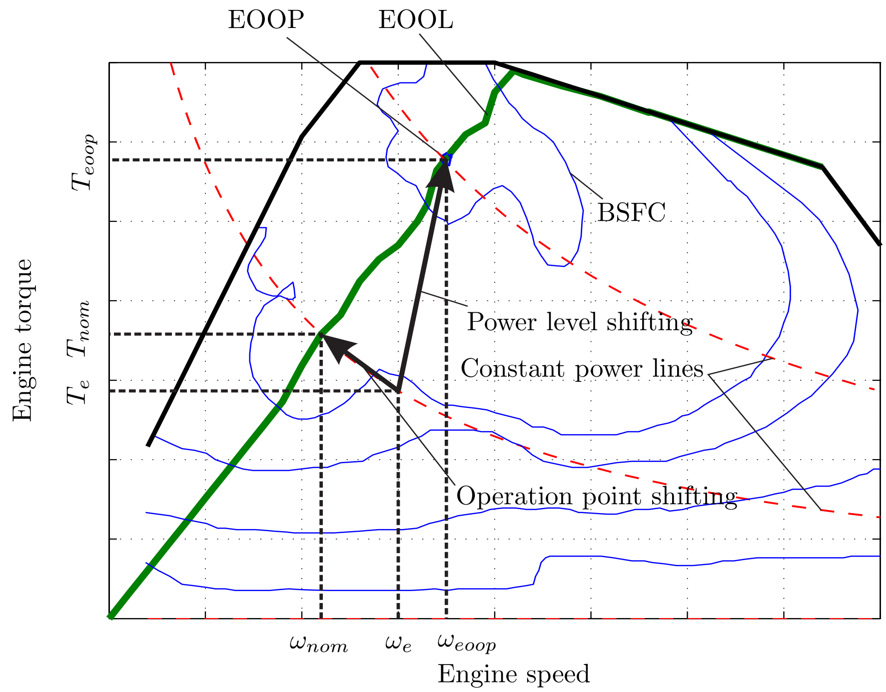

Figure 4.1: Engine operation points control concept based on [LKGP01]

where ω_e denotes the engine speed. On the other hand, engine power is a function of hydraulic power demand and pump efficiency as

$$\begin{aligned} P_e &= \frac{P_d}{\eta_{tp}\eta_{vp}}, \\ P_e &= T_e\omega_e, \end{aligned} \tag{4.2}$$

where P_e denotes engine power, P_d hydraulic power, and T_e engine torque. However, optimal operation of the engine at individual power level depends on the engine speed and engine torque values as depicted in figure 4.1. Due to complex interdependency of the engine power, engine torque, engine speed, and pump displacement ratio in this contribution an iterative-based optimal controller is used for optimal control of the engine. The resulting parameter is pump displacement ratio which is calculated by minimizing

$$f_{eool} = \int_0^T \sqrt{(T_e - T_{eool})^2 + (\omega_e - \omega_{eool})^2} dt, \tag{4.3}$$

in which T_{eool} and ω_{eool} represent respectively engine torque and speed related to EOOL. The concept of second control strategy is shifting the operation point and engine output power close to the EOOP as depicted in figure 4.1. This control strategy is only implementable when the accumulator is able to capture extra power of the engine. However, in comparison with electric batteries, the accumulator capacity is low and the accumulator must be depleted before upcomming braking phase. To realize EOOP, the control parameters are determined by minimizing

$$f_{eoop} = \int_0^T \sqrt{(T_e - T_{eoop})^2 + (\omega_e - \omega_{eoop})^2}dt, \tag{4.4}$$

in which T_{eoop} and ω_{eoop} represent respectively torque and speed corresponding to the EOOP.

4.1.2 Vehicle performance control strategy

In the context of vehicle longitudinal dynamics, different definitions of vehicle performance exist. Acceleration rate, maximum vehicle velocity, and gradeability are the typical criteria for vehicle performance evaluation. In the context of hybrid vehicles, one of the typical definitions describing vehicle performance is reference velocity tracking error. In other words, for the improvement of vehicle performance, it is expected that the hybrid vehicle can track the reference driving cycle. In order to optimize the vehicle tracking error the following objective function

$$f_{driv} = \int_0^T \sqrt{(v - v_{ref})^2}dt, \tag{4.5}$$

must be minimized in which v_{ref} denoted the reference velocity.

4.1.3 Transmission ratio control strategy

The most important advantage of SHHP topology is mechanical decoupling of engine and vehicle. In this case the engine can operate independent from vehicle. On the other hand, the transmission ratio between engine and vehicle can be changed continuously between various possible transmission ratios as depicted in figure 4.2. The transmission ratio of SHHV i_{trs}, is the function of pump and motor displacement ratios defined by

$$i_{trs} = \frac{x_m D_m}{x_p D_p} \times \frac{1}{\eta_{vp}\eta_{vm}}. \tag{4.6}$$

In figure 4.2, the gray pentagon represents all possible transmission ratios considering the predefined maximum vehicle velocity and engine speed. Accordingly, for a specific vehicle velocity, gear ratio as well as engine speed can be changed in a wide range, with respect to the maximum engine torque and vehicle torque demand. Therefore, the continues variable transmission system enables optimal control of the engine with respect to vehicle velocity, accumulator power, and demanded torque.

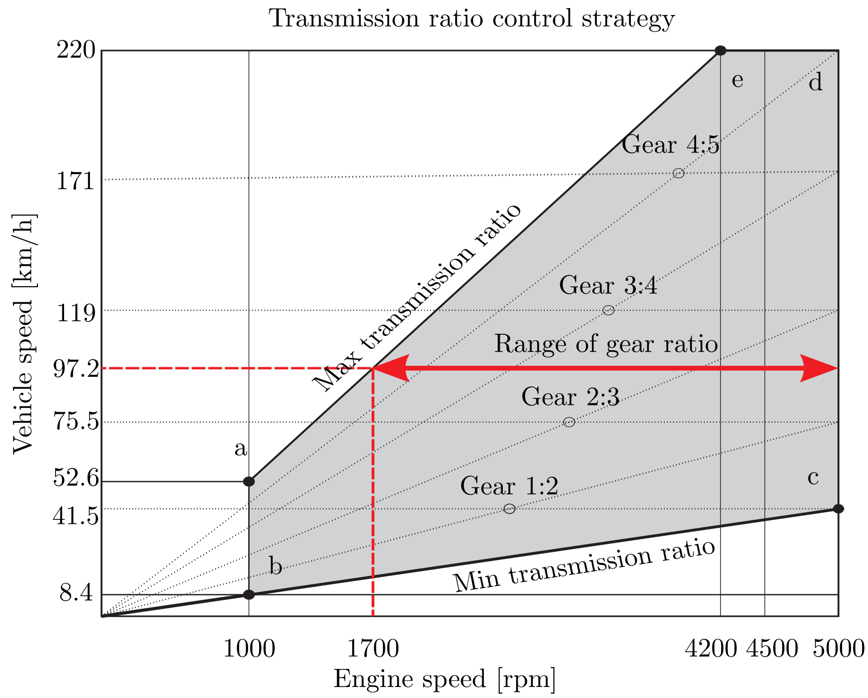

Figure 4.2: Transmission ratio control strategy [KSedb]

4.2 Conventional vehicle

In this contribution, the conventional vehicle is considered to be supplied only using engine power and equipped with manual transmission system. It has the same engine and vehicle characteristics as hybrid vehicle which are presented in table 3.1. However, it is equipped with a manual transmission. Optimal gear shifting strategy to improve driver comfort is implemented for the evaluation of the conventional vehicle performance. The optimal gear shifting strategy applied to the conventional vehicle is precisely explained in figure 4.3. The optimal points for gear shifting is the point in which the traction forces in two different gear ratios at the same speed have the same values. Based on this definition, the shifting points are considered on the intersection of transmission traction force related to each gear ratio at the specific vehicle velocity as depicted in figure 4.3. The effects of clutch and slippage in the clutch are not considered in this contribution.

4.3 Pressure control strategies

In this section, the operation modes of HHV as well as their individual control variables are described. Switching of the system between operation modes depends on the power demand and available power in the accumulator. In figure 4.4, different

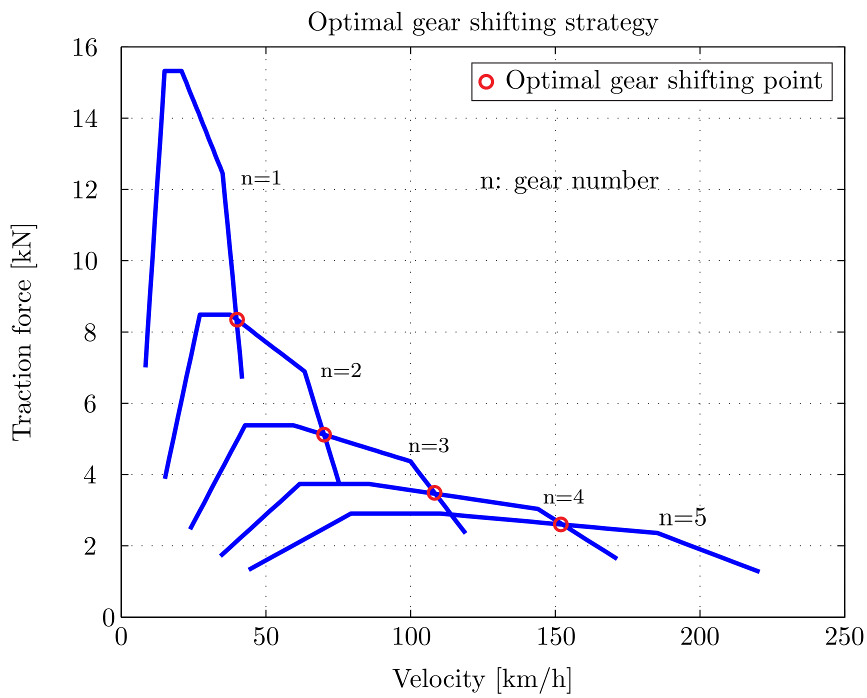

Figure 4.3: Optimal gear shifting strategy for manual transmission system [Dat13]

possible operation phases of a hybrid vehicle depending on demanded power P_d and accumulator power P_a values are shown. In propulsion phase, based on the contribution of the power sources for vehicle propulsion, four different sub-operation modes can be realized. In table 4.1, details of SHHV operation modes are described [Sch09]. Pure engine mode (PEM) relates to the vehicle propulsion using only engine power like in conventional vehicles. In this mode, the power of both engine and vehicle are positive while hydraulic power is zero. In this mode, SHHV operates as a hydrostatic transmission system. The control inputs of the system are pump and motor displacement ratios.

In pure hydraulic mode (PHM), vehicle is supplied only using accumulator power while engine is off or in idle mode. The only control variable of SHHV in PHM is motor displacement. In hybrid mode (HM) both engine and accumulator powers supply the vehicle power demand. Due to integration of both power sources to overcome vehicle power in HM mode, power management in this mode must be optimized. In this mode, system is fully controlled with control of pump and motor displacement ratios. In addition to HM, charging mode (CM) is also challenging. In CM, vehicle power demand is fully supplied using engine power. Also engine power is used to charge the accumulator. Vehicle power demand in braking mode is shown by negative sign. In braking mode, two operation modes of the system are braking energy recapturing mode (BERM) and conventional braking mode (CBM). Depending on the accumulator charge level, part of braking energy may be recaptured by accumulator and vehicle is decelerated using regenerative braking system. If accumulator

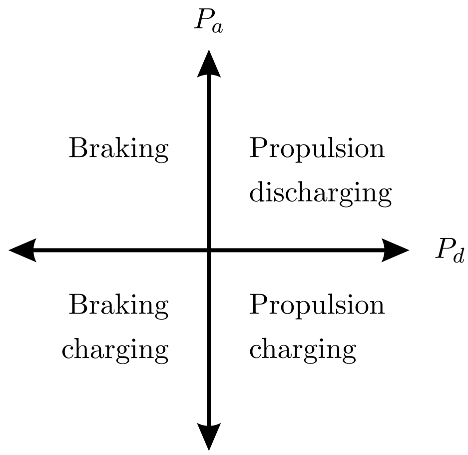

Figure 4.4: Different operation phases of the vehicle

Table 4.1: Sub-operation modes of SHHV [Sch09]

Vehicle phase	**Hybrid mode**	P_d	P_a	P_e	**Controlled variables**
Propulsion	PEM	+	0	+	x_p, x_m
	PHM	+	+	0	x_m
	HM	+	+	+	x_p, x_m
	CM	+	-	+	x_p, x_m
Braking	BERM	-	-	0	x_m
	CBM	-	0	0	F_{br}

PEM: The vehicle is supplied using only engine power.
PHM: The vehicle is supplied using only accumulator power.
HM: The vehicle is supplied using both accumulator and engine power.
CM: The extra power of the engine is captured by the accumulator.
BERM: The brake energy is captured by the accumulator.
CBM: Conventional braking without charging the accumulator is realized.

is fully charged, the conventional brake system decelerates the vehicle.

4.4 Rule-based pressure control strategies (PCS)

Development of rule-based pressure control strategies starts with the interpretation of the vehicle power demand which in real-time are the gas and braking pedal inputs. Depending on a given driving cycle, negative power demand is interpreted as brake pedal input while positive power demand is interpreted as the gas pedal input. In vehicle acceleration and cruising phases, all the system components are integrated in the HHV model, while in the decelerating phase, engine is excluded from the

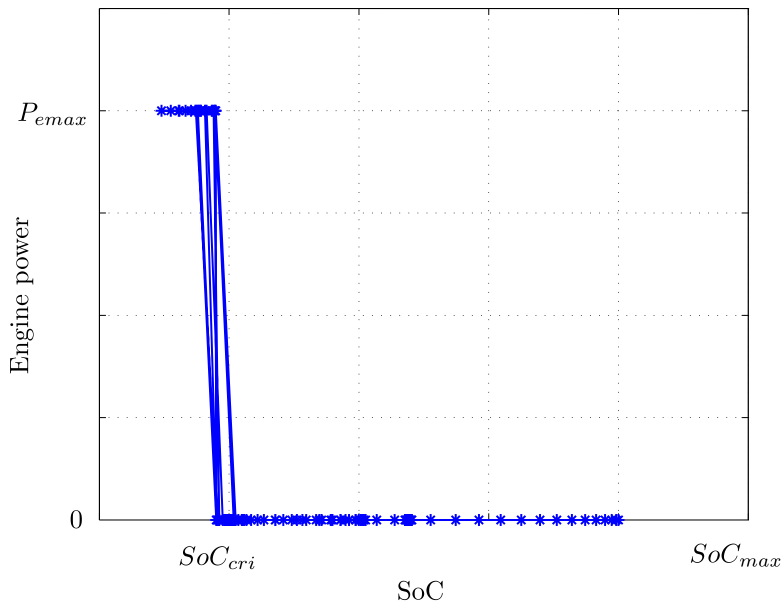

Figure 4.5: Concept of engine on/off control strategy [KSeda]

model. Another difference between acceleration and deceleration phases of SHHV is the drive motor operation. In acceleration phase, drive motor operates in motor mode while in deceleration phase it operates in pump mode. In this contribution, each operation mode of the SHHV is modeled individually and a switching system controls the modes transition of the system. Based on vehicle power demand, the control strategies which are a trade-off between energy balance and fuel consumption of the SHHV calculate the control inputs of the control system.

In this section, typical rule-based pressure control strategies namely, engine on/off strategy, accumulator depleting strategy, and smooth engine power strategy are presented and applied to SHHV. The fuel economy potential as well as performance of these pressure control strategies are compared and analyzed.

4.4.1 PCS I: Engine on/off control strategy

The engine on/off control strategy (in [EGGE05] and [FH12] denoted as thermostat control strategy) is a typical control approach applied to SHHV. In figure 4.5 the concept of engine on/off control strategy is shown. The main idea is to maintain energy level in the accumulator as well as to efficiently operate the engine in drive mode. Based on the accumulator available energy and vehicle power demand, the system operates in PHM, HM, and CM modes. According to the figure 4.5, the engine operates either in idle mode or in pre-determined maximum power level.

Critical SoC and critical engine power are the control variables of the engine on/off control strategy. In other words, for SoC larger than critical SoC engine operates in

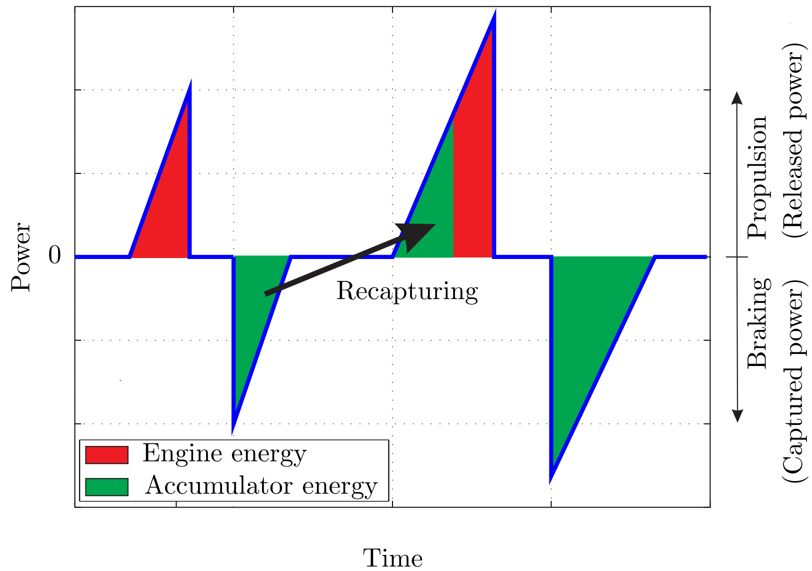

Figure 4.6: Concept of accumulator depleting strategy [KSeda]

off or idle mode and vehicle is propelled only using accumulator power. Otherwise, engine operates in its maximum power value. On the other hand, for the SoC smaller than critical value, even if the accumulator energy for vehicle propulsion is enough, engine switches on to maintain SoC.

Due to accumulator low energy capacity, engine switches frequently to maintain the accumulator energy. As a result of frequent transient operation of the engine between on and off modes, fuel consumption increases and engine durability decreases. These are the main disadvantages of PCS I. Decrease in the critical value of SoC decreases the switching number of the engine. Therefore, accumulator has more free capacity for energy capturing. However, accumulator power density at low pressure is not enough for the propulsion of the vehicle. Therefore, compatibility between the accumulator critical SoC value on the one hand, and vehicle performance and efficiency on the other hand may improve the performance of PCS II. Another important aspect is that in braking phase, the whole accumulator capacity for recapturing the braking energy is not free. However, operation of the engine in the EOOP is the main advantage of PCS I.

4.4.2 PCS II: Accumulator depleting control strategy

The idea of accumulator depleting control strategy is using accumulator power in the first acceleration phase as depicted in Figure 4.6. In this approach, accumulator is just charged using braking energy. In other words, accumulator charging using engine power is not considered in PCS II because discharged accumulator has enough free capacity to capture braking energy in the upcoming braking phase. Therefore,

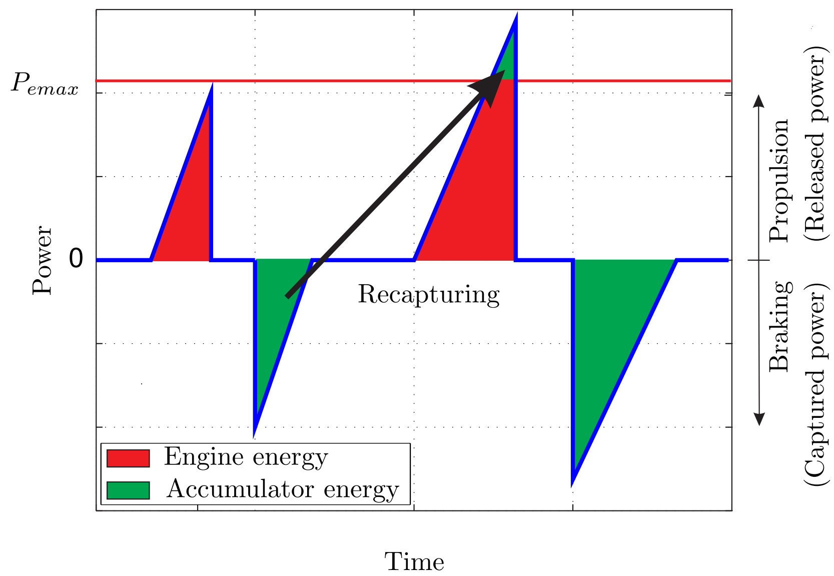

Figure 4.7: Concept of Smooth engine power strategy [KSeda]

accumulator is only a temporary power source for recapturing of braking energy. In this approach, in propulsion phases system operates mainly in PEM and PHM modes, while in deceleration phases it operates in BERM mode. Due to decoupled operation of power sources, this approach is simply implementable. Considering discharged accumulator in acceleration phases, fuel consumption can be decreased using optimal control of the hydrostatic transmission. Operations of two power sources are considered as decoupled because they do not operate at same time.

Due to indirect charging of the accumulator using engine power, the control of the engine in EOOP is not realizable within this approach. Therefore, operation of the engine in higher power demand is unavoidable. Besides disadvantages, the main advantage of this control strategy is its simple applicability to typical HHV topologies. Additionally, due to full depletion of the accumulator at each acceleration phase, accumulator operates in its whole operation range. This approach is suitable for vehicles undergoing stop and go driving cycles such as shuttle buses.

4.4.3 PCS III: Smooth engine power control strategy

Smooth engine power control strategy is based on the idea of limiting maximum engine power with assistance of accumulator power. In Figure 4.7, the concept of this control strategy is shown. Accordingly, the accumulator assists the engine whenever demanded power is larger than a predefined value which is shown by P_{emax}. In other words, this approach limits the maximum engine operation power. Therefore, this control strategy is a proper solution for engine downsizing. In this approach, depending on maximum engine power the operation of SHHV in acceleration phases, is divided into two modes; PEM for power demands less than maximum engine power, HM for power demands more than maximum engine power. Same as accumulator depleting control strategy, the accumulator is not charged directly using engine power.

In this control approach, performance of both power sources are coupled and significantly depend on the system pressure. Therefore, this control strategy is more complicated from design and application points of view. Smooth operation of the engine in small power range is the main advantage of this control strategy. Moreover, bounding the maximum engine power as well as fuel consumption capacity of this control approach are significantly dependent on the accumulator size. In contrast to other control strategies, the performance of this approach can be optimized using optimal control approaches.

4.5 Evaluation of pressure control strategies

After development of the components models and topologies, subsystems controllers, and pressure control strategies, in this section control strategies are applied to SHHV. The main goal is evaluation of three developed rule-based control strategies. Moreover, the effects of driving cycles on the performance of the control strategies are discussed. Sensitivity analysis of the control strategies performances to the related parameters is the other goal of this section. For these reasons, three control strategies are applied to SHHV and for three different driving cycles. Integrated fuel consumption and vehicle velocity tracking error considering the same characteristics of the conventional vehicle are compared. Additionally, engine operation points and power distribution between power sources are compared. The effects of control parameters related to the pressure control strategies on fuel consumption, velocity tracking error, and the number of engine switching are discussed using simulation results. Models and topologies of the system used for simulation are described in chapter 6. The criteria for the evaluation of system performance and efficiency are discussed in sections 4.1. For the assessment of the driving cycles effects on both system performance and efficiency, ECE-15, EUDC, NEDC, and US06 driving cycles, (figure 4.8) , are used for realization of the load profiles. For vehicle speed control, a driver model is used to determine the reference velocity. Because the upcoming load cycle is unknown, the value of the power demand is unknown. Therefore, it is reasonable to set the initial value of the accumulator energy to 50%. In this case, accumulator has energy for acceleration and also capacity for regeneration of braking energy. According to the engine EOOP control strategy, the EOOP of the engine model used in this contribution is 50 kW. For this reason, the maximum power values for the pressure control strategies are set to 50 kW.

The simulation results for the ECE-15 driving cycle are shown in figures 4.9 to 4.18. In figure 4.9, power distribution between power sources according to the

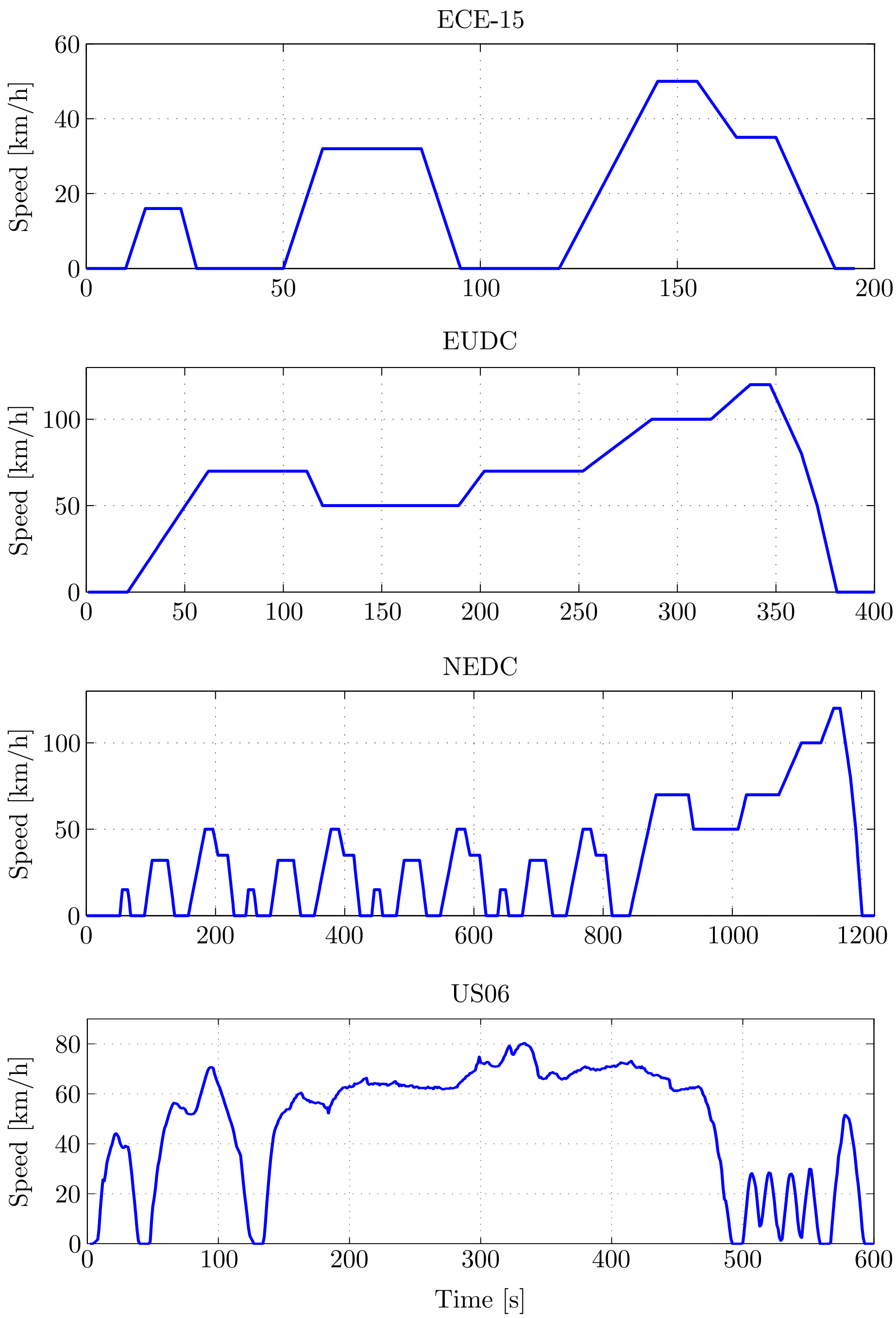

Figure 4.8: Applied driving cycles

vehicle power demand are shown. In other words, this figure demonstrates the contribution of the engine and accumulator to overcome vehicle power demand as well as contribution of the accumulator for recapturing of braking energy. In the first graph of figure 4.9, fluctuations of engine power between minimum and maximum values relating to PCS I are shown. Engine operation depends on the SoC critical

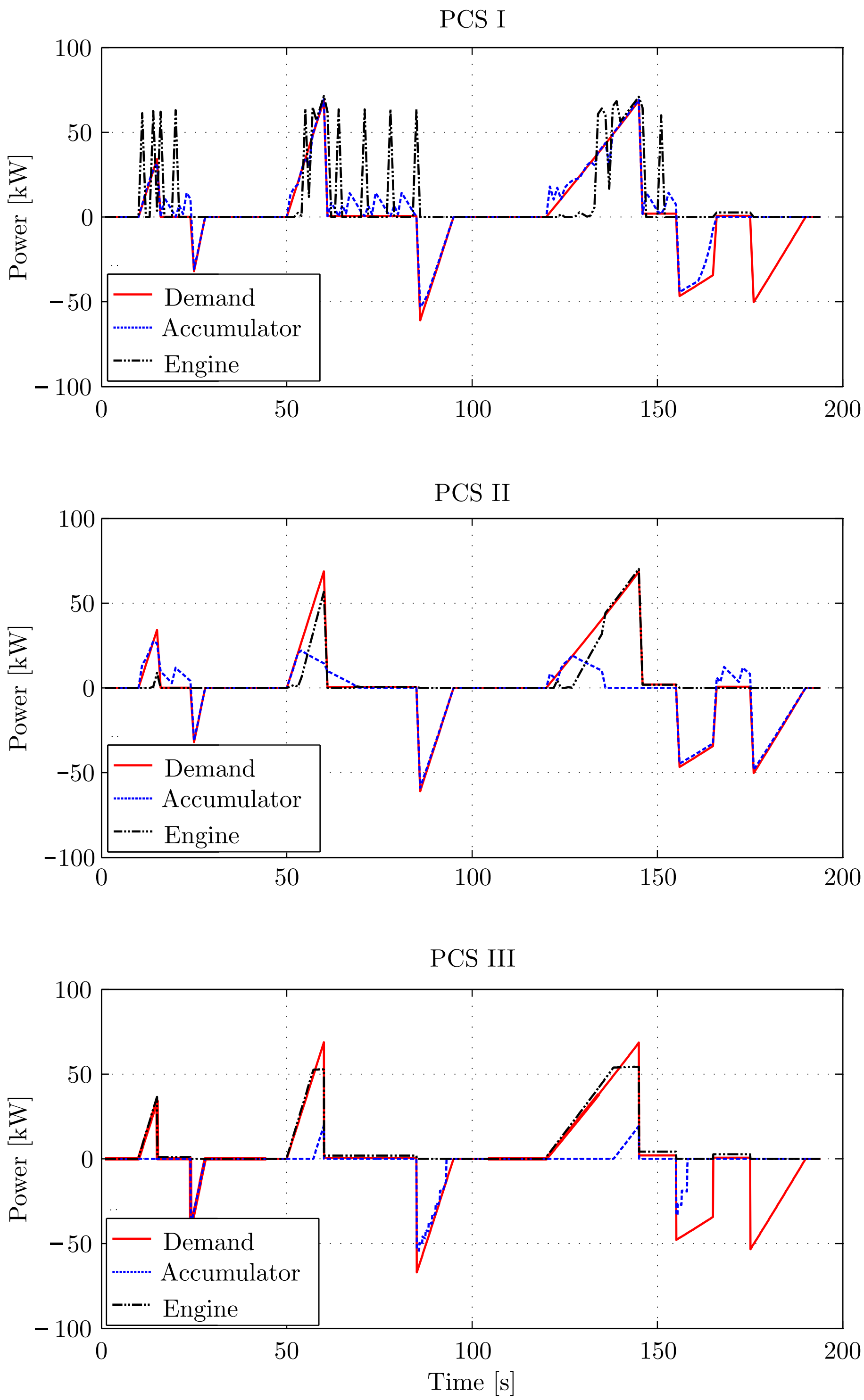

Figure 4.9: Power distribution for ECE-15 driving cycle

value. Due to variation of system efficiency, the maximum engine power is about

10% more than the set value of EOOP. The value of power supply at the beginning of acceleration phase is mostly more than power demand. It is due to low efficiency of SHHV at the beginning of acceleration phase. Comparison of the negative powers makes it clear that the accumulator does not capture all available braking energy due to its low capacity. Therefore, energy recapturing ratio of the system is the function of PCS I control variables. Accumulator low capacity for PCS I is more critical than two other pressure control strategies. The reason is keeping the minimum level of the accumulator energy to a preset value.

The second graph of figure 4.8 relates to PCS II. It becomes clear that accumulator is immediately depleted at the beginning of each acceleration phase. Therefore, it can capture more braking energy. The PCS II shows a simple accumulator charge and discharge process. Therefore, performance of PCS II significantly depends on accumulator size. Engine operation in all range of power indicates that the same engine as conventional vehicle must be used for the application of PCS II to SHHV. In other words, using PCS II engine downsizing is not reasonable. In contrast to other control strategies, PCS II recaptures more braking energy. Therefore, vehicle is driven using pure hydraulic power in cruising phases.

The final graph of figure 4.9, relates to the PCS III. It indicates that the maximum engine power is limited by the assistance of accumulator. Nevertheless, for the power levels less than maximum value, vehicle is driven based on pure fuel energy. On the other hand, PCS III braking energy recapturing rate is less than two other control strategies. The reason is non-use of accumulator energy in low power demands. Therefore, same as PCS I, PCS III recapturing ratio depends on the control variable.

In figure 4.10, the operation points of the engine for three pressure control strategies are shown. For all three control strategies, engine operation points are shifted close to EOOL. The main reason is mechanically decoupled operation of the engine and vehicle as described in section 4.1.3. The first graph illustrates that for PCS I, engine operates either close to idle or close to EOOP operation points. Although the BSFC in EOOP has the minimum value, energy losses due to conversion of the fuel energy to hydraulic energy are undesirable. In this control strategy, the operation of engine between idle and peak power values is due to the lack of accumulator energy to dominate vehicle demanded power. The second graph in figure 4.10 relates to PCS II. It indicates that engine operates in all power ranges. In contrast to two other pressure control strategies, operation of engine along EOOL is extended from minimum value to the EOOP value. Moreover, the engine operation points are mainly shifted close to EOOP. Comparison of the engine operation points for PSC II and PSC III shows that the engine operation is limited to the predefined maximum power, here 50 kW for PCS III. The goal of engine downsizing is limitation of the engine peak power. In this case, small engine size from the maximum engine power point of view can be used for vehicle traction without compromising performance of the vehicle. Based on the simulation results, PCS III is shown to be the suitable control strategy for engine downsizing.

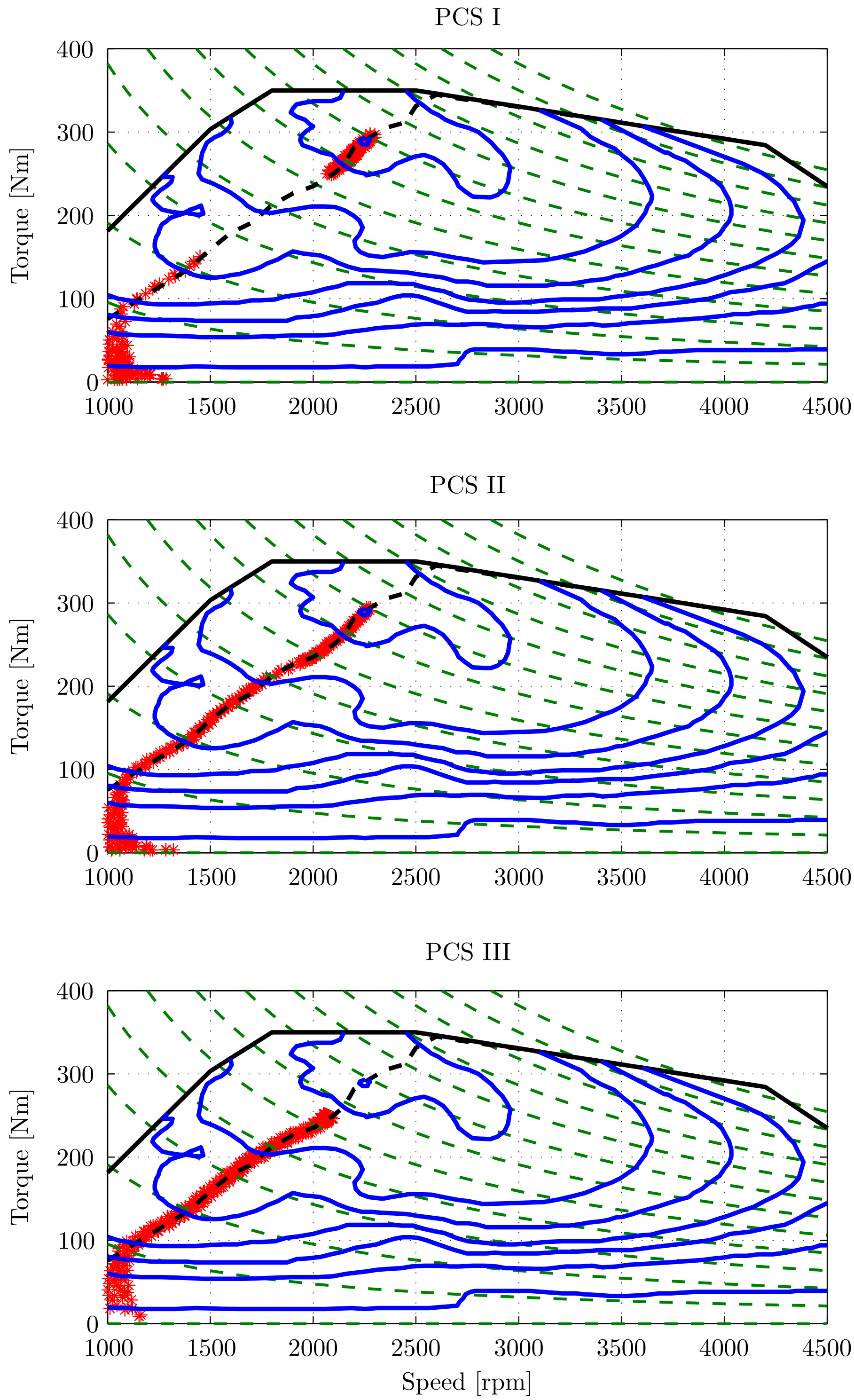

Figure 4.10: Engine operation points for ECE-15 driving cycle

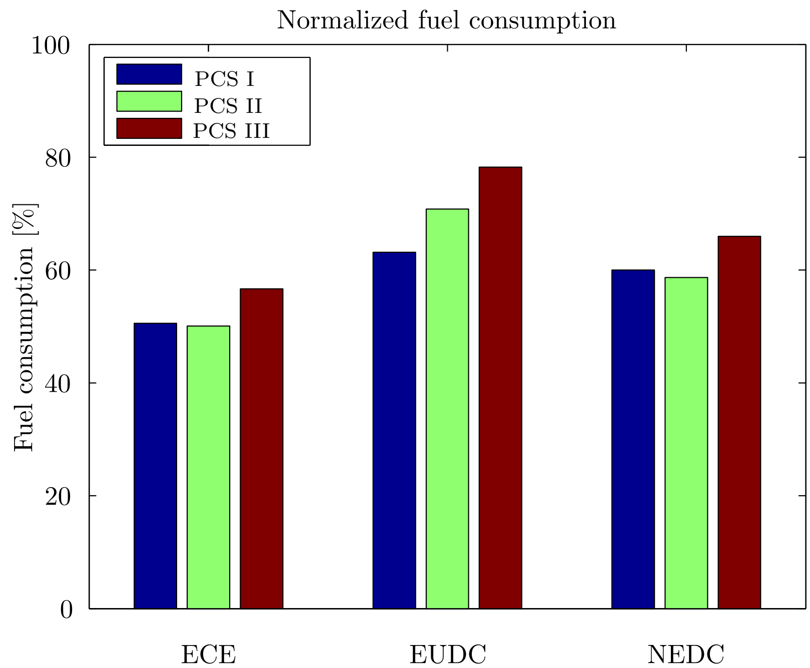

Figure 4.11: Comparison of fuel consumption for three driving cycles [KSeda]

For the evaluation of pressure control strategies performances, total normalized fuel consumption and velocity tracking error resulting from implementation of pressure control strategies to SHHV are respectively shown in figures 4.11 and 4.12. For a reasonable comparison of the pressure control strategies, the results are normalized based on characteristics of the conventional vehicle. In section 4.2, the conventional vahicle is described in detail. Due to difference between time period of the driving cycles, both fuel consumption and velocity tracking error are compared for a 100 km travel distance of the vehicle. It must be noted that the control parameters are setup according to the results presented in the last section.

In figure 4.11 normalized fuel consumption of SHHV with application of three pressure control strategies and for three driving cycles are shown. Comparison of the results based on the driving cycles shows that the operation of SHHV in highway driving cycles such as EUDC is more inefficient than city driving cycles such as ECE and NEDC. On the other hand comparison of the fuel consumption based on the pressure control strategies shows dramatic results. The third control strategy, PCS III shows less efficiency than two others. The main reason is low braking energy regeneration rate for PCS III. According to figure 4.8, applying PCS III to SHHV, vehicle is supplied using mainly engine power. Decrease of the critical power for PCS III, the contribution of accumulator power can be increased. This is discussed in detail in the next section. However, the integration of accumulator power in power demands larger than the critical value in PCS III, prevents the accumulator from full depletion. In contrast to PCS II, accumulator does not have enough capacity

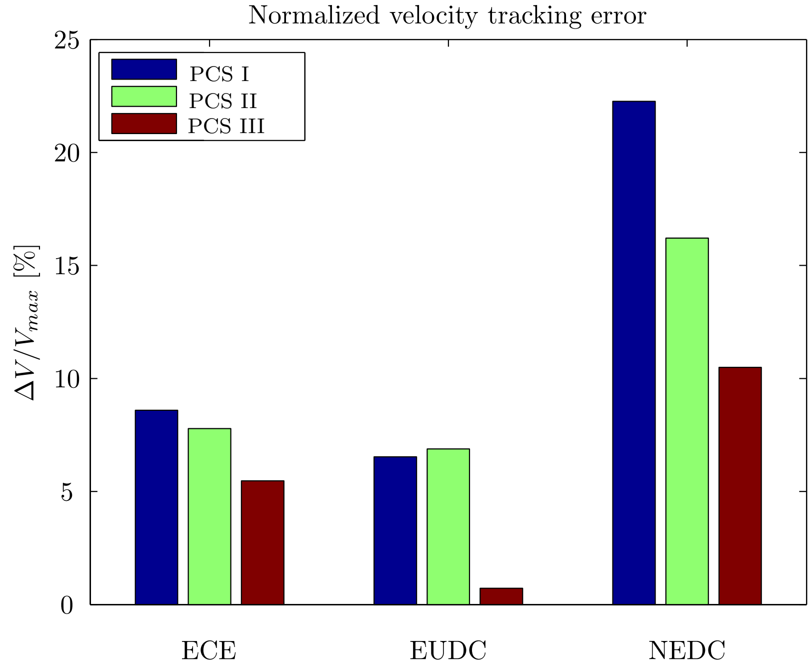

Figure 4.12: Comparison of velocity tracking error for three driving cycles

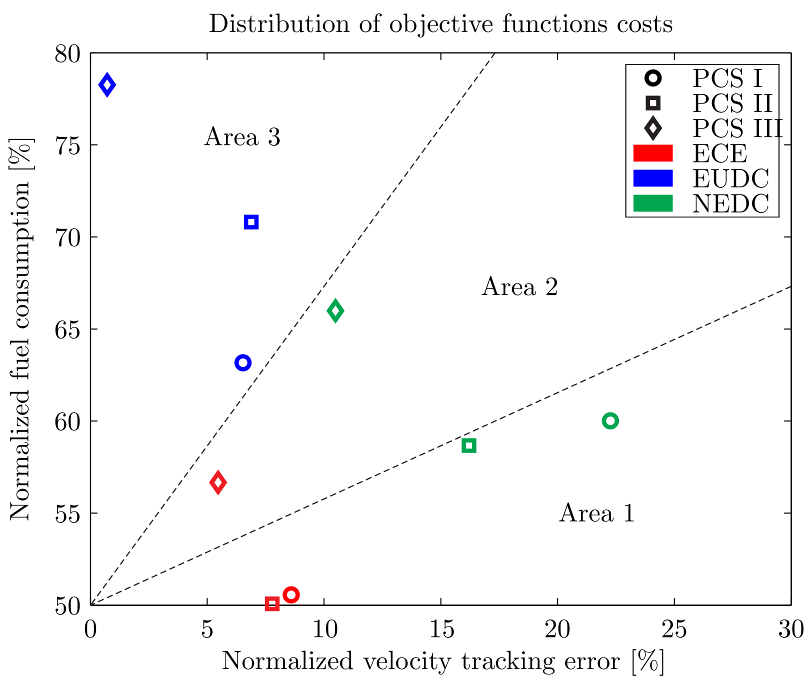

Figure 4.13: Comparison of three pressure control strategies for three driving cycles [KSeda]

for regeneration of the braking energy. The fuel consumption of PCS I and PCS II for city driving cycles ECE and NEDC are relatively the same while PCS I shows smaller fuel consumption than PCS II in highway driving cycle. The main reason is efficiency dependence of the PCS II to the driving cycles. In other words, PCS II shows better fuel economy in stop and go driving cycles because its energy braking regeneration rate significantly large.

In figure 4.12, SHHV normalized velocity tracking error with application of three pressure control strategies and for three driving cycles are shown. Comparison of the results based on the driving cycles shows that in contrast to fuel consumption, the velocity tracking error of SHHV in highway driving cycles such as EUDC is less than city driving cycles such as ECE and NEDC. The comparison of results based on the driving cycles shows that in contrast to fuel consumption, the velocity tracking error of SHHV in highway driving cycles such as EUDC is less than city driving cycles such as ECE and NEDC. It can be concluded that for comparison of such disadvantages, additional power supply can be the solution. However, it is in contrast with fuel consumption as the other evaluation criteria. On the other hand, comparison of the velocity tracking error of pressure control strategies shows that PCS III has smaller velocity tracking error than two other control strategies. The smallest velocity tracking error relates to EUDC driving cycle due to slight acceleration and not frequent velocity change in highways. The largest velocity tracking error can be observed for NEDC because of sever accelerations and larger maximum velocity compare to ECE-15. According to section 4.6, discharging of the accumulator must be compensated using engine power. Considering regular charge and discharge of the accumulator, vehicle power supply using both engine and accumulator powers is not enough. Due to the power drop, vehicle driveability and velocity tracking error deteriorate respectively.

In figure 4.13, the results are summarized based on both evaluation criteria. The weighting coefficient of objective functions are selected to normalized numerical values of the objective functions. Therefore the contributions of objective functions are the same. In figure 4.13, the results are categorized in three areas. The results in the area 1 relate to the controller and conditions in which improvement of the fuel economy are preferred to the vehicle driveability. It becomes clear that both PCS I and PCS II in the ECE and NEDC driving cycles have acceptable fuel economy, while their driveabilities are relatively worse than other conditions. The results in the area 2 represent satisfactory results for the simultaneous minimization of both objective functions. In this case, PCS III in both city driving cycles ECE and NEDC shows better fuel economy and driveability than other conditions. The area 3 represents the result of control strategies and driving cycles which tend to improve driveability more than fuel economy. It becomes clear that the driveability of the pressure control strategy depends significantly on the driving cycle. It can be concluded that the combination of pressure control strategies to improve SHHV characteristics in different driving cycles may be practical solution for overcoming their individual disadvantages.

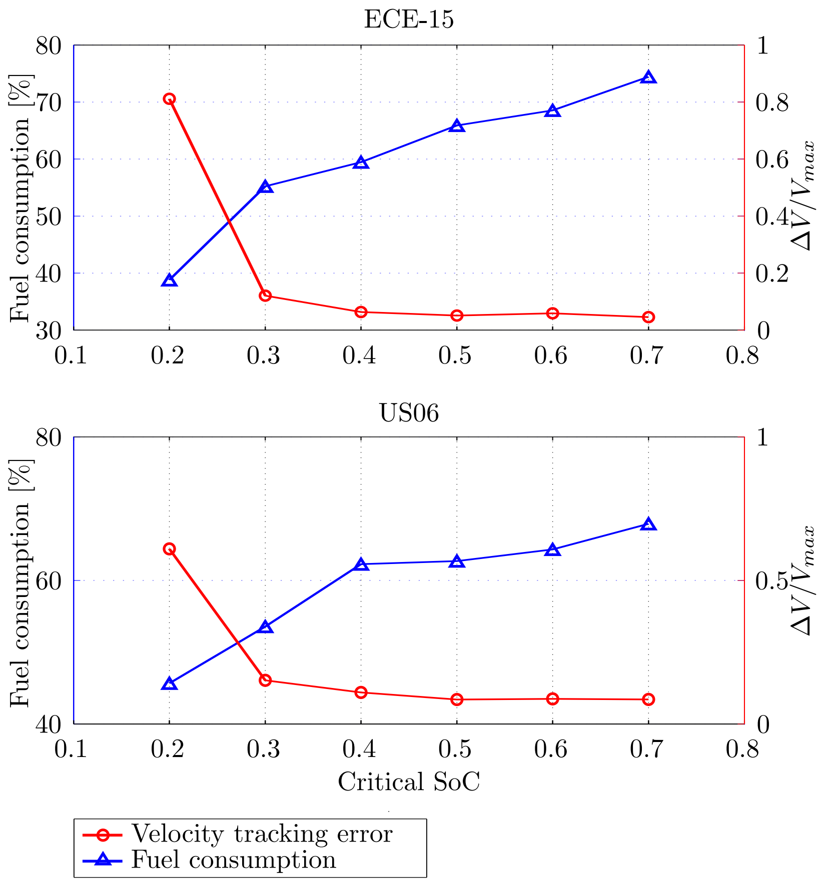

Figure 4.14: Effect of SoC on vehicle performance and fuel consumption (PCS I) [KSeda]

4.6 Effect of control parameters

In this section, the effects of control parameters related to the discussed pressure control strategies on the driveability and the fuel consumption of SHHV are discussed. For this reason, two driving cycles namely ECE-15 and US06 are used. It should be noted that accumulator depleting control strategy does not have any direct control parameter.

4.6.1 PCS I control parameter's effects

The first simulation result demonstrates the effect of two control parameters of the engine on/off control strategy on the driveability and fuel consumption of the SHHV. The setup of the model for all simulations is the same and the used driving cycles are ECE-15 and US06. The initial value of accumulator SoC is set to 0.5. In figure 4.14, the simulation results for different critical SoC show that the increase in critical

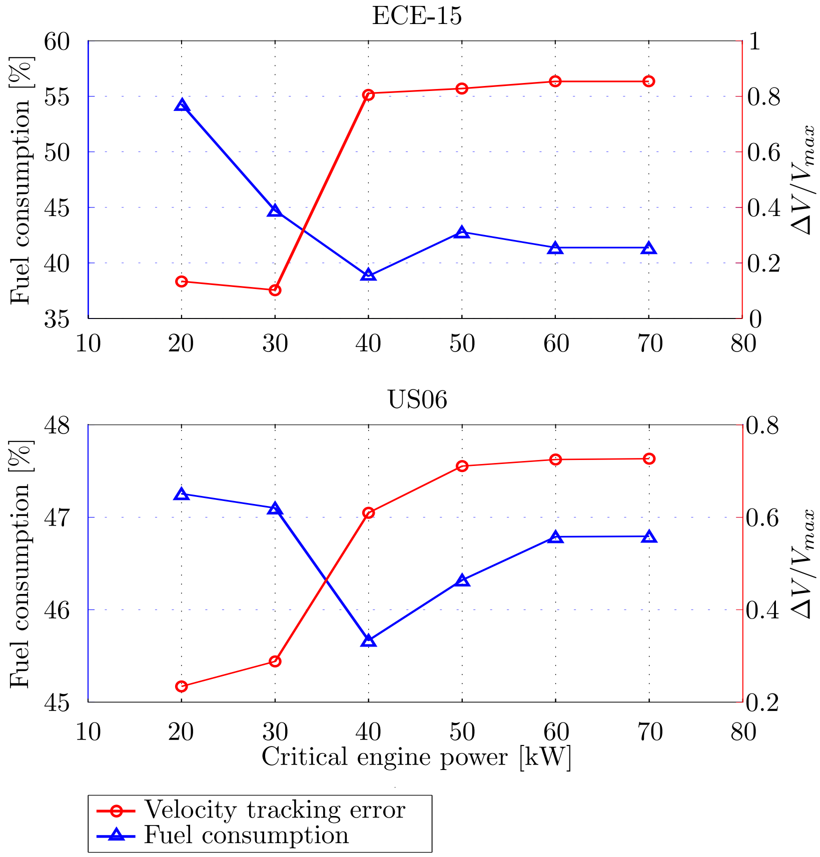

Figure 4.15: Effect of critical power on vehicle performance and fuel consumption (PCS I) [KSeda]

SoC results in driveability improvement, but it increases the fuel consumption. It must be noted that keeping the SoC close to maximum value causes the inability of the accumulator to capture braking energy. Therefore, vehicle is supplied mainly using engine power. The results of both driving cycles show the same trends. Also the optimal values of critical SoC for ECE and US06 driving cycles with a negligible difference are 0.262 and 0.271 respectively. It becomes clear that the optimal value of critical SoC in PCS I can be selected independent from driving cycle.

In figure 4.15, the effects of critical power of the engine on both performance and fuel consumption of the vehicle are shown. The critical SoC is set to 0.4. Increase of the critical engine power results increment of velocity tracking error while fuel consumption is decreased. The optimal values for engine critical power for ECE and US06 driving cycles are shown at 33 kW and 35.5 kW respectively. The main reason is the sufficiency of accumulator and engine resultant powers to overcome the demanded power. In other words, the operation of engine at large critical power values than the mentioned values may charge the accumulator. However, charging the accumulator considering hydraulic system efficiency is not always efficient.

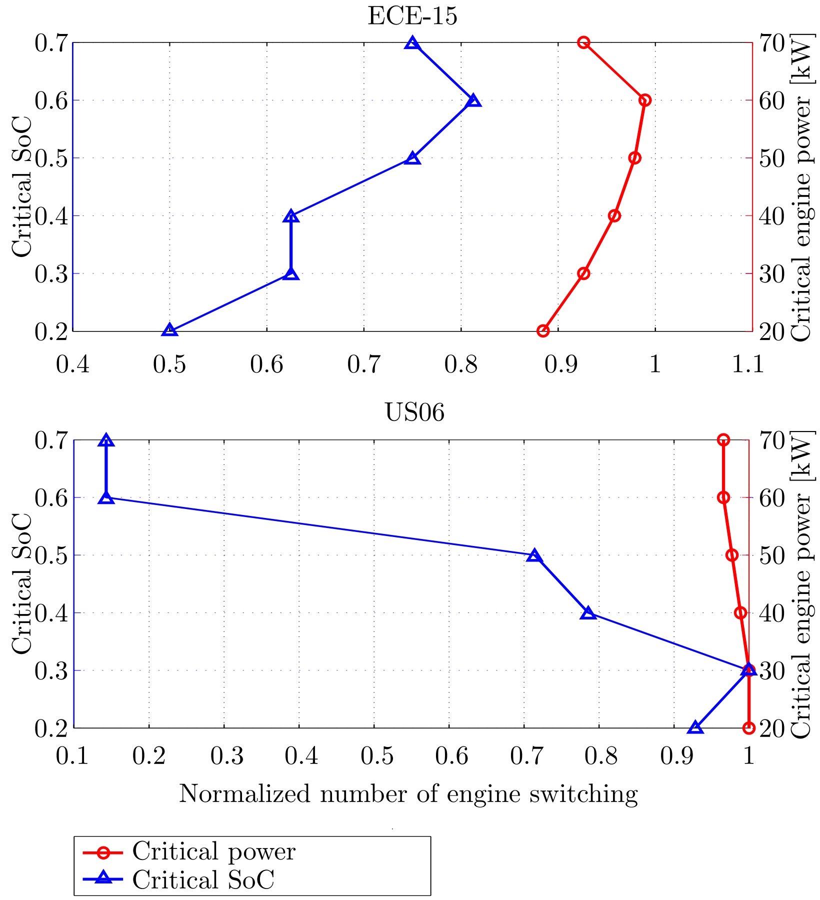

Figure 4.16: Effect of SoC and critical power on the number of engine switching (PCS I) [KSeda]

The effect of the PCS I control parameters on the number of engine switching between predetermined maximum power level and idle mode are shown in figure 4.16. The increase of the critical SoC and engine power has relative same effects on the engine switching number. The maximum value of engine switching number for ECE is at critical SoC value of 0.6 and engine critical power value of 60 kW while the minimum value of engine switching number is at SoC 0.2 and critical engine power 20 kW. This trends for the US06 is same as ECE driving cycles. However the maximum and minimum engine switching numbers are at different control parameters values. The reason is that large critical SoC decreases the operation band of the accumulator to critical SoC and maximum SoC. Therefore, accumulator is depleted very frequent and engine must compensate the accumulator energy. On the other hand, large engine power critical value guarantees the level of accumulator power close to optimal value with smaller engine switching number. In figure 4.17, SoC fluctuation in acceleration and cruising phases for ECE and US06 driving cycles are presented. The initial and final values of the SoC for both cases are 0.5. The equality of the

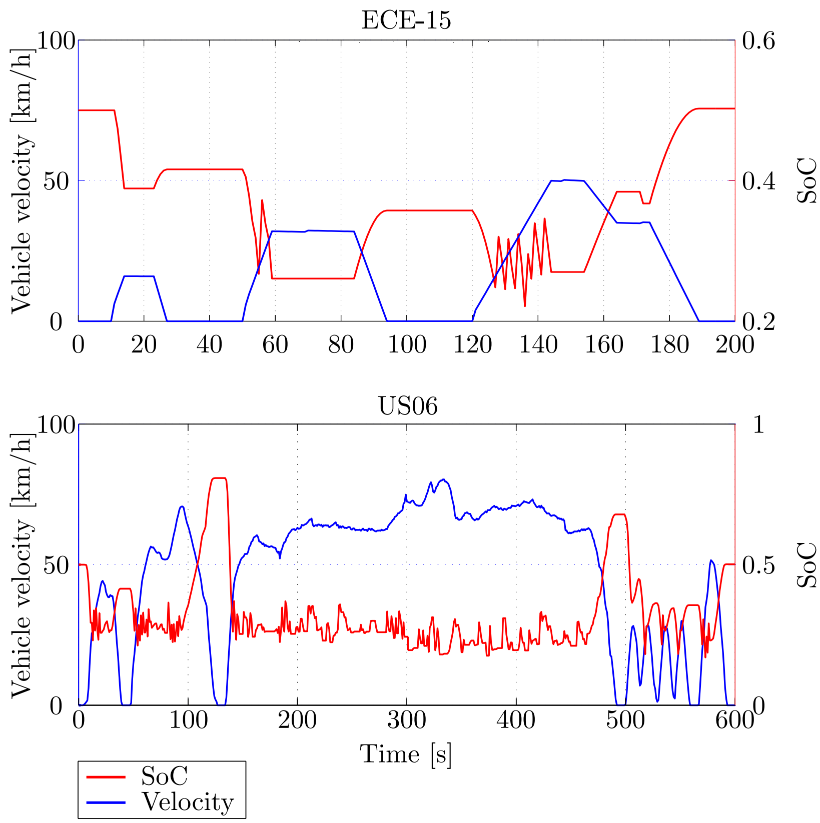

Figure 4.17: SoC fluctuation in acceleration and cruising phases (PCS I) [KSeda]

initial and final SoC indicates reasonability of the comparison because accumulator is considered as a temporary power source and its final power value is as same as its initial power value. Accordingly, accumulator power as well as engine power frequently switch during acceleration phases. The main reason of fast and frequent depletion of the accumulator is high value of power demand in acceleration phases.

4.6.2 PCS III control parameter's effects

In this section, the effect of the PCS III control parameters on the performance of the SHHV is discussed based on simulation results. In figure 4.18, the results performed by applying PCS III are shown. The only control variable of PCS III is the maximum power of the engine. It becomes clear that decrease of the engine maximum power decreases the fuel consumption. Here, the accumulator is large enough to supply vehicle power demand. The decrement of engine maximum power value, depends on the size and dynamic characteristics of the accumulator. In other words, the maximum engine power can be decreased depending on the maximum power that can be supplied by accumulator. Therefore, optimization of the hydraulic components size, particularly accumulator, for the application of the smooth engine

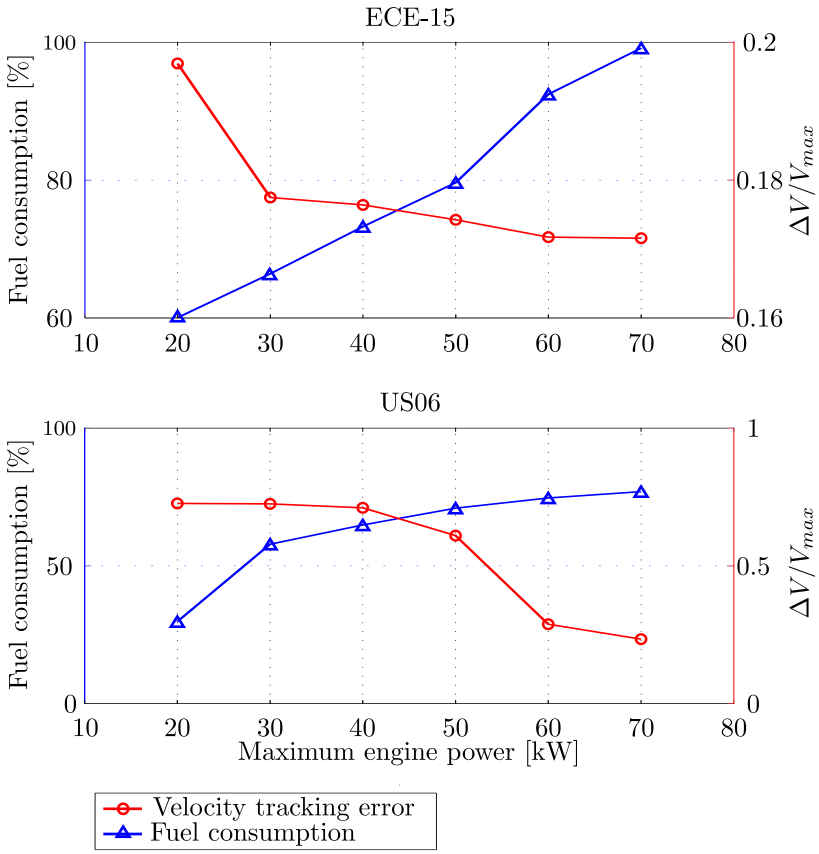

Figure 4.18: Effect of critical power on vehicle performance and fuel economy (PCS III)

power control strategy is unavoidable. The trends of the both characteristics, fuel consumption and velocity tracking error according to the increment of the maximum engine power are the same. The optimal values of the maximum engine power for minimization of both characteristic values are almost the same and about 43.7 kW. It can be concluded that optimal control parameter values of the PCS III are independent from driving cycle. During vehicle cruising phases, if accumulator power is insufficient, engine is the only power supply. In this case, fuel consumption can be decreased by continuous power transmission through the hydrostatic transmission. Depending on the vehicle power demand, engine can be shifted close to the optimal operation line. However, engine optimal operation is impossible. Having a wide range transmission ratio at each vehicle speed, makes it possible to control the engine efficiently regarding to the optimum BSFC map.

5 Optimal control of power management

Typical optimization methods applied for the optimization of the power management in HHV are described in chapter 2. Accordingly, optimized power management strategies can improve the characteristics of HHV. Implementation of optimization methods in real-time due to model complexity and large computational load is not straightforward. Application of rule-based power management approaches which are sub-optimal, optimization of rule-based power management approaches based on the results taken from off-line power management approaches, and development of optimized power management strategies using optimal control concepts are three typical methods for development of optimal power management approaches as shown in table 5.1. Based on the results presented in chapter 4, the application of power management strategies in HHV improves significantly its fuel consumption and efficiency. However, efficiency and performance depend on control parameters.

Within this chapter, first an off-line power management strategy based on DP is developed. The main reason is to use the result as benchmark for the evaluation of other power management strategies. In order to predict the vehicle load behavior, a MPC-based power management algorithm is developed and implemented to SHHV. The main reason for development of MPC-based power management is the prediction of vehicle load behavior in order to prepare information for the optimization of power management. Finally, an Instantaneously Optimized Power Management (IOPM) algorithm is developed and applied to SHHP. It contains a two-level controller for the optimization of power management. The main reason for development of IOPM is to reduce both computational load and complexity of problem formulation. Efficiency and performance of these three power management algorithms are compared and evaluated based on the simulation results. Some parts of this chapter are published [KSedb].

Table 5.1: Typical power management development methods

Power management	**Optimality**	**Algorithms**
Rule-based	Sub-optimal	Fuzzy
Optimized rule-based	Sub-optimal	DP, GA
Optimal control	Optimal	MPC, ECMS

5.1 Dynamic Programming optimization algorithm

Dynamic Programming is a direct optimization algorithm based on the known information about the vehicle load behavior and system state. However, it results relatively the optimal power distribution between HHV power sources. Therefore, in this contribution, DP is applied to develop a reference for the evaluation of other power management strategies.

The model used for the application of DP-based power management is backward-facing model. Deterministic DP assumes that the knowledge of the upcoming driving manoeuvre is given in advance. Based on the deterministic vehicle power demand, the power distribution is optimized. Therefore, vehicle torque demand and velocity are considered as known. The manipulated variables of the system are pump and motor displacement ratios. According to equation 3.6, accumulator flow rate as well as accumulator power are the functions of the line pressure. This dependency is shown in figure 5.1. Accordingly, negative power values indicate accumulator charging capacity while positive power values represent accumulator discharging capacity. For the accumulator initial pressure 20 Mpa, maximum accumulator power supply is about 30 kW at line pressure 14 Mpa while this value for the accumulator initial pressure 40 Mpa, is about 90 kW at line pressure 26 Mpa. Depending on the vehicle power demand shown by green line in figure 5.1, line pressure can be set for a wide range between maximum and minimum line pressure values. The line pressure set value for a specific power demand determines the power distribution range between engine and accumulator. Therefore, line pressure can be considered as manipulated variable for the purpose of power management optimization.

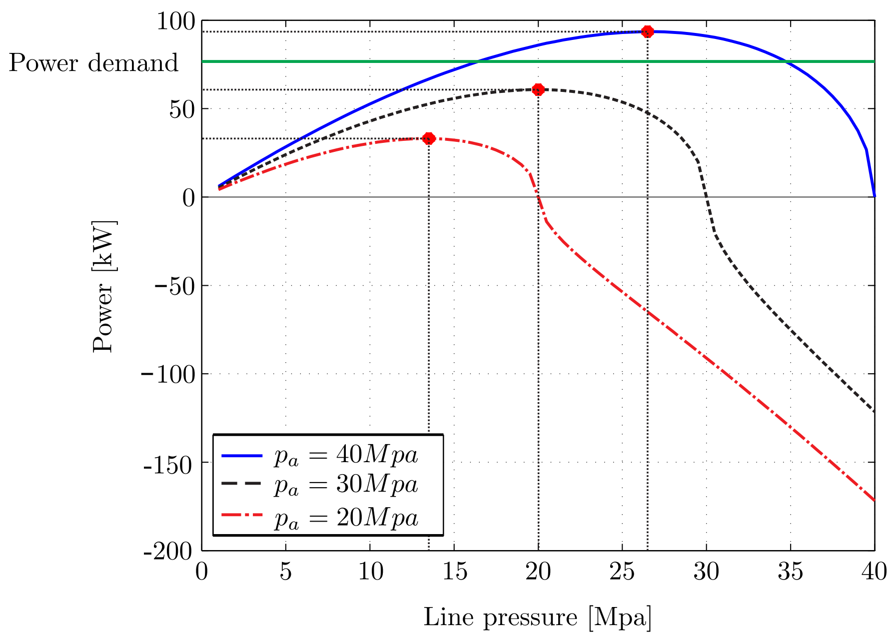

Figure 5.1: Accumulator power as a function of line pressure [KSeda]

5.1.1 Implementation of DP-based power management

The discrete model of the system is described by

$$x_{k+1} = f_k(x_k, u_k), \qquad k = 0, 1, ..., n-1 \tag{5.1}$$

where x_k denotes the state of the system, u_k control variables, and N time horizon. The cost function is defined by

$$J = min_{u \in U}[g_N(x_N) + \sum_{k=0}^{N-1} g_k(x_k, u_k(x_k))], \tag{5.2}$$

where U denotes a set of control variables in the feasible area of system operation. The term $g_N(x_N)$ corresponds to the penalization of system state final value from the initial value. In this case, only SoC is considered as the internal state of the system. The problem constraints strictly enforce the SoC final value to merge to the initial value. The second term in the cost function 5.2, $g_k(x_k, u_k(x_k))$ determines the cost due to the application of $u_k(x_k)$ to the system at each time step. The boundary conditions corresponding to the optimization problem are

$$\begin{aligned} SoC_{min} \leq SoC(k) \leq SoC_{max}, \\ 0 \leq T_e(k) \leq T_{e\,max}(w_e(k)), \\ 0 \leq w_e(k) \leq w_{e\,max}, \\ T_{m\,min} \leq T_m(k) \leq T_{m\,max}, \\ Q_{m\,min} \leq Q_m(k) \leq Q_{m\,max}, \end{aligned} \tag{5.3}$$

where SoC denotes accumulator SoC, Te engine torque, w_e engine speed, T_m motor torque, and Q_m motor flow rate. The cost function contains the terms related to engine fuel consumption and system overall efficiency. Based on equation 3.23, engine fuel consumtion is the function of engine torque and speed. The efficiency of the transmission system is the function of pump and motor efficiencies which are described by equations 3.14 to 3.18. In order to reduce the computational load, before proceeding the optimization, the power train feasible operation area for the specific driving cycle is specified. In order to determine the cost values of states closed to boundaries between feasible and infeasible areas, boundary-line method proposed in [Sun09] is used. In this method, before starting the optimization, boundary states and corresponding cost values are calculated in backward computation. The procedure starts from the last time step and proceed backward in time domain from time step $n-1$ to 0. where SoC denotes accumulator SoC, Te engine torque, w_e engine speed, T_m motor torque, and Q_m motor flow rate. The cost function contains the terms related to engine fuel consumption and system overall efficiency. Based on equation 3.23, engine fuel consumtion is the function of engine torque and speed. The efficiency of the transmission system is the function of pump and motor efficiencies which are described by equations 3.14 to 3.18. In order to reduce the computational load, before proceeding the optimization, the power train feasible operation area for the specific driving cycle is specified. In order to determine the cost values of states closed to boundaries between feasible and infeasible areas, boundary-line method proposed in [Sun09] is used. In this method, before starting the optimization, boundary states and corresponding cost values are calculated in backward computation. The procedure starts from the last time step and proceed backward in time domain from time step $n-1$ to 0.

5.1.2 Simulation results: DP-based power management

According to figure 2.2, performance and accuracy of DP optimization algorithm depend significantly on the grid size of the system state fluctuation plane. The size of the grids are set by setting the time step and stage sizes. Small grid size increases significantly the computational load while large grid size maybe cause non-reasonable dynamics of the system [Sun09]. The main reason for system dynamics non-reasonability for large grid size is the elimination of partial system dynamics in which the initial values of the states at each step does not match to the final values of the states at the last step. In this context, DP is accurate if the minimum cost function value for a specific optimization problem has the minimum value [Sun09]. In other words, for a specific DP optimization problem, the cost function value changes proportional to the grids size. In figure 5.2, the effects of the grid size on the cost function value minimization are shown. The cost function values are normalized by dividing to the maximum cost function values. Therefore the maximum cost function values for two different driving cycles have the same value. According to figure 5.2, it becomes clear that decrease of the grid size decreases the minimum cost function values. According to figure 2.2, in DP optimization problems the cost function values are calculated on the grid points. Because of the grid size decrement, the number of grids size as well as computation load increases. It becomes clear that computation load and DP accuracy conflict. Therefore, the optimal grid size value can be result from the minimization of the computation load and cost function values.

In figures 5.3 and 5.4, the optimization results for DP-based power management are shown. In this simulation, the grid size is set to 0.005. In the following sections, the results are analyzed and compared in order to compare the performance of the DP-based power management for two different driving cycles.

5.1.2.1 Accumulator SoC

In figures 5.3 (b) and 5.4 (b), the trends of the accumulator SoC are shown. Accordingly, the same SoC initial and final values for both driving cycles indicate that

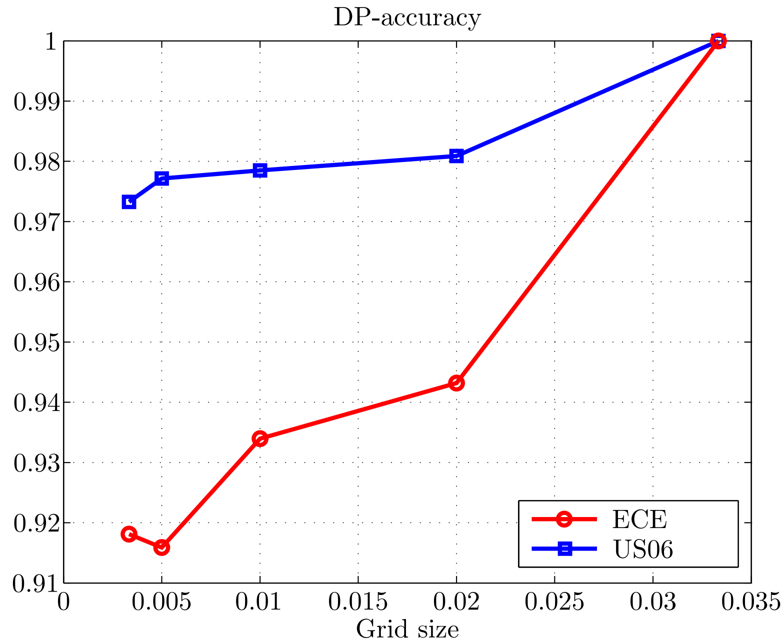

Figure 5.2: Effect of grid size on the accuracy of DP-based power management

DP-based power management considers the accumulator such a temporarily power source. In other words, the pure value of engine fuel consumption can be calculated only if the accumulator initial and final power values are the same. Therefore, comparison of the HHV fuel consumptions by applying different power management strategies and driving cycles are reasonable.

5.1.2.2 Flow rate distribution

In figures 5.3 (c) and 5.4 (c), the trends of the accumulator, pump, and motor oil flow rates are shown. The flow rate of the motor (shown by red line) represents vehicle power demand. The negative flow rate values of the motor show the operation of the motor as pump in braking phases. Moreover, negative flow rate values of the accumulator show accumulator charging in braking phases while positive values show accumulator discharging in acceleration phases. Pump and motor flow rates are proportional to the engine and vehicle velocities while accumulator flow rate is proportional to the accumulator and line pressure difference. Zero values of pump flow rate in braking phases show engine operation in on/off switching modes instead of idle mode operation.

5.1.2.3 Power distribution

In figures 5.3 (d) and 5.4 (d), the trends of power distribution between accumulator, pump, and motor are shown. Full charging and discharging of the accumulator specifically in aggressive acceleration and deceleration phases indicate accumulator-based vehicle power supply. The complete depletion of accumulator before upcoming deceleration phase, increases braking energy recapturing ratio. It becomes clear that the tendency of the SHHV for using accumulator power in aggressive acceleration and deceleration phases is more than the slight acceleration and deceleration phases. By analysing the power distribution graphs in figures 5.3 and 5.4, it becomes clear that, at each vehicle start up, utilizing of accumulator power is preferred. Due to low system efficiency, using accumulator power to supply the vehicle power demand in cruising phases is not affordable. It can be concluded that, by increasing of the SHHV efficiency in stop and go driving cycles, the utilization of accumulator power for start up the vehicle is preferred. The main reason is the high efficiency of the transmission system due to small velocity and large pump and motor displacement ratios in low power demands. In addition, the avoidance of engine operation at low power levels is an additional reason. As explained in section 4.1.2, engine BSFC values in low power levels are significantly larger than EOOP.

5.1.2.4 Pump/motor displacement ratios

In figures 5.3 (e) and 5.4 (e), the trends of pump and motor displacement ratios are shown. Whenever SoC hits its minimum value, system operates as hydrostatic transmission in PEM mode. In this operation mode, optimization of system efficiency has the first priority. For this reason, the displacement ratios of pump and motor are increased up to their maximum values. Moreover, the low displacement ratio of motor in cruising phase significantly decreases the system efficiency in cruising phase.

According to the results shown in figures 5.3 and 5.4 and in contrast to HEV, SHHV does not operate in CM mode. Although this degree of freedom is considered in the developed DP-based power management, it is not realized based on the simulation results. The main reason is the inefficient conversion of engine mechanical energy into hydraulic energy, because pump and accumulator do not always operate in the most efficient condition. On the other hand operation of the engine in idle mode is not efficient. For these reasons, power management refuses the engine operation in idle mode. Nevertheless, the increase of the engine switching number decreases the engine durability which is not discussed in the context of this contribution. The comparison of DP-based power management distribution graphs with one from accumulator depleting strategy shown in figure 4.9 indicates that PCS II and DP-based optimal control strategy have the same performance.

5.2 Predictive power management strategy

Model Predictive Control (MPC) represents a controller in which the output is calculated based on a local valid model, the given constraints or limitations, the

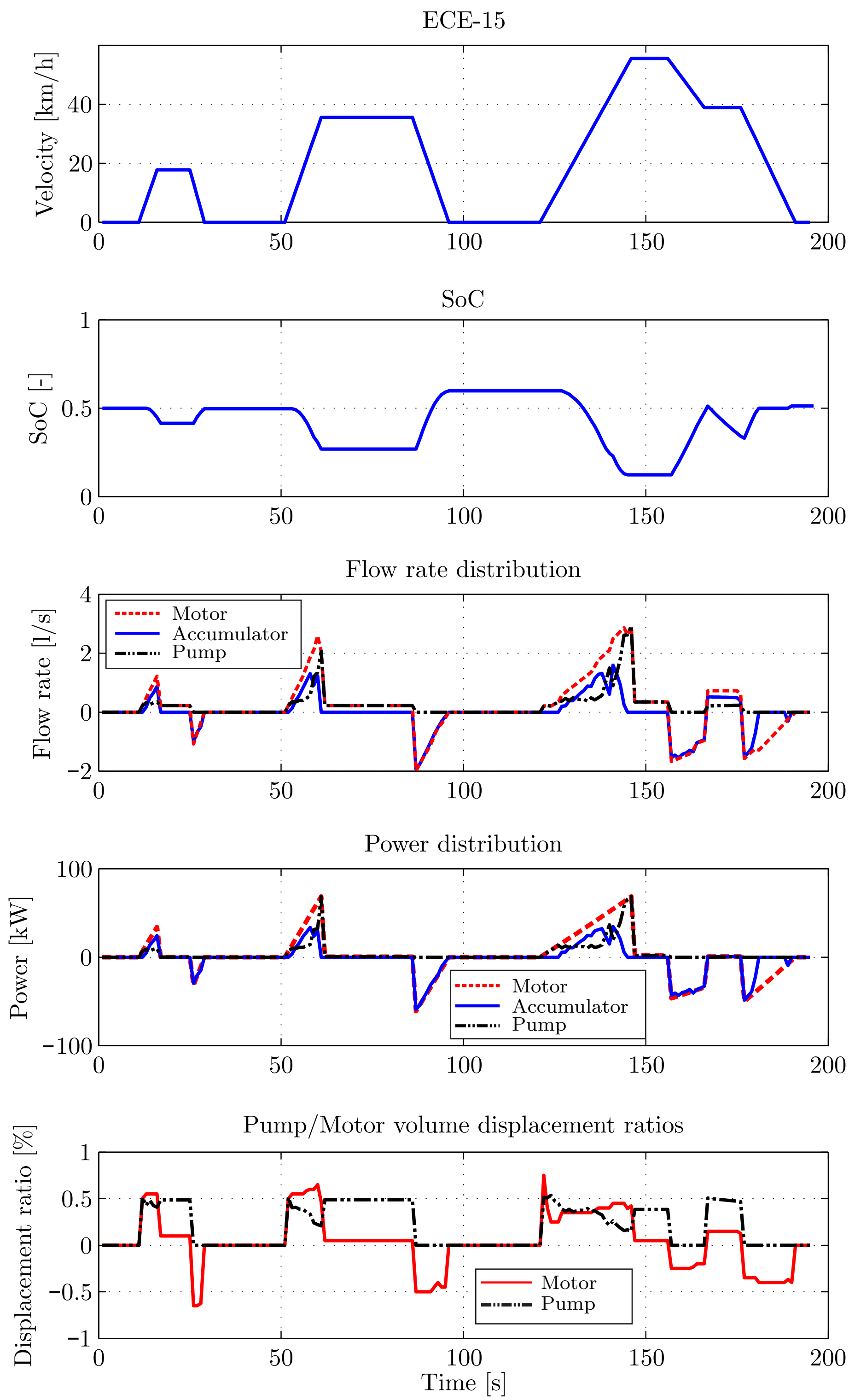

Figure 5.3: DP-based power management results for ECE-15 driving cycle

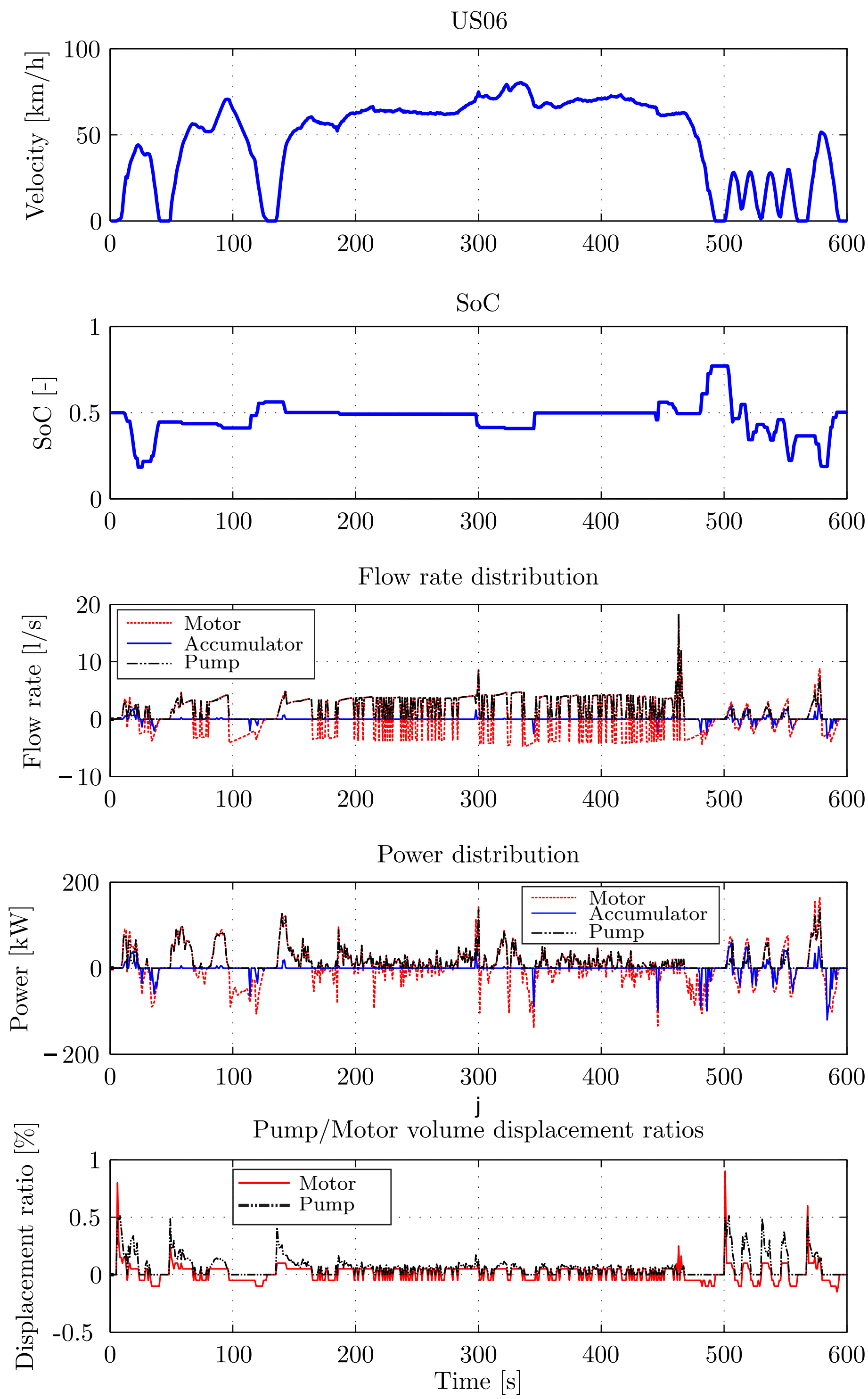

Figure 5.4: DP-based power management results for US06 driving cycle

given performance criteria, and the given information about suitable simulation horizons typical for the application context. The main idea of MPC is to vary the control variables so that the performance criteria are locally optimized. For this reason, local models, related assumed inputs, and time horizon are considered to be given. For the considered moment in time simulations are realized using modified and/or varied controller gains, results are compared with respect to the performance criteria. The best set of controller gains becomes active for a short period of time, the algorithm is repeated to adapt controller gains to update the local models and inputs.

In real-time, driver predicts the vehicle speed based on the road traffic, traffic rules, etc. for a limited prediction horizon. Accordingly, in this contribution driver model dictates the expected vehicle speed value to the controller in prediction horizon. In figure 5.5, the structure of MPC-based power management is shown. This controller uses driver model command as the velocity reference trajectory. According to table 4.1, SHHV may operate in up to six different modes. Each operation mode has its individual model. Therefore, internal model of MPC may contains up to six sub-models. In figure 5.5, the structure of the developed power management algorithm using MPC in the loop is shown. It consists of a driver model, MPC optimal control, and plant. The integration of driver model in a close loop form, is not only used for switching modes control of MPC, but also prepares partial information about the vehicle velocity in a limited time horizon.

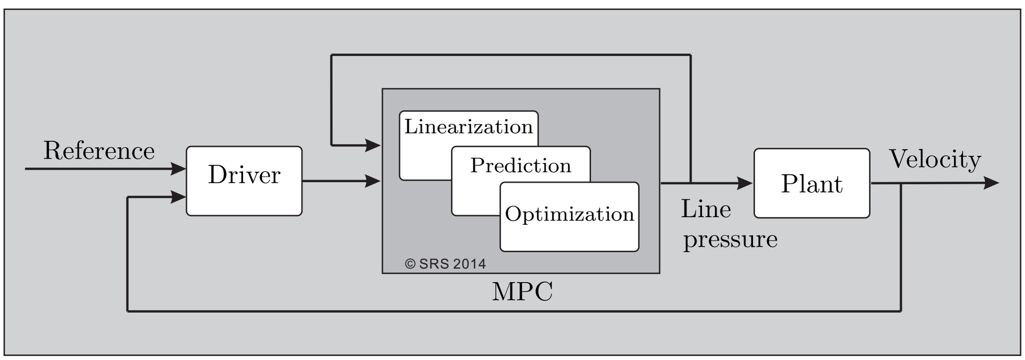

Figure 5.5: MPC-based power management structure [KSedb]

5.2.1 Implementation of MPC-based power management

The problem corresponding to application of optimal control methods in real time is computational load. In order to reduce the computational load of the MPC, model is simplified. Although the accuracy of the model and results may be scarified by model simplification, a trade-off between simplification and accuracy result an optimal model. In order to implement MPC, first of all problem must be changed to a standard form. For this reason, internal model is linearized using Taylor series.

At each operation point, system is linearized around the nearest equilibrium point. General formulation of the nonlinear system is described by

$$\begin{aligned} \dot{x}(t) &= f(x(t), u(t), t), \\ y(t) &= g(x(t), u(t), t). \end{aligned} \tag{5.4}$$

The model linearized form at the operation point $a = (x_p, y_p, u_p)$ is represented by

$$\delta\dot{x}(t) = (\frac{\partial f}{\partial x})_a \delta x(t) + (\frac{\partial f}{\partial u})_a \delta u(t), \tag{5.5}$$

$$\begin{aligned} \delta\dot{x}(t) &= A\delta x(t) + B\delta u(t), \\ \delta y(t) &= C\delta x(t) + D\delta u(t), \end{aligned} \tag{5.6}$$

where δx denotes perturbation state, δu perturbation input, and δy perturbation output. The predicted load cycle is the input of power management optimization. Typical optimization methods are based on quadratic programming. However formulation of such a constrained nonlinear optimal control problem in quadratic form is not easy. In this case, reformulation of objective functions presented in chapter 4 is undesirable. For this reason, instead of using engine torque and speed in objective functions, engine power is optimized directly. The optimization is done to shift engine operation points close to EOOP. The relevant objective function is described by

$$f_{Pe} = \int_0^T \sqrt{(P_e - P_{eoop})^2}. \tag{5.7}$$

The final formulation of the problem is solved using Pontryagin's minimum principle.

5.2.2 NSGA II optimization algorithm

Between numerical optimization methods, iterative-based optimization methods such as GA have large computational load. Nevertheless, different techniques are developed in order to reduce the computational load of global optimal power management approaches. In this section, a Non-dominated Sorting Genetic Algorithm II (NSGA II) which is based on GA is applied for optimization of partially known optimization problem. The main advantage of this method is its applicability for optimization problems in which, detail information about the model is not available.

For the implementation of NSGA-II optimization method to HHV, nonlinear dynamical model of the power train as well as nonlinear objective functions are directly used in the optimization loop. In figure 5.6, schematic of the optimization loop is depicted. The forward-facing model of the HHV is used in the optimization loop. The model contains vehicle, accumulator, and line pressure dynamics. To reduce computational load, other components such as valves are model using quasi-static models. Other controller which adjust motor/pump displacement ratios determines the power distribution ratio between two power sources. During the iterative optimization loop, NSGA-II generates control parameters for the controlled system and the important properties of the system, which are formulated as objective functions are evaluated at each iteration. Both system performance and efficiency are optimized simultaneously. Therefore, a possible trade-off between both system performance and efficiency can be achieved. The optimization time is adapted to prediction horizon. The manipulating parameters are the individual control parameters of the pump and motor. The iteration continues till the minimum values of the objective functions in a deterministic variation domain are achieved.

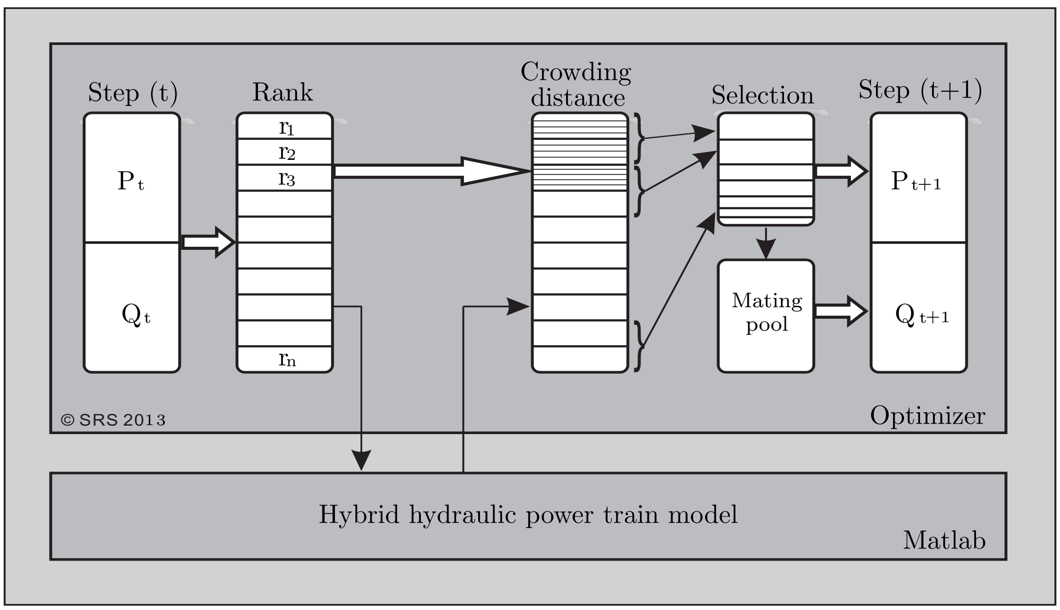

Figure 5.6: NSGA II-based hybrid power train optimization [KMMS13]

5.3 On-line instantaneous optimized power management

The performance of both off-line and predictive optimized power management approaches discussed in previous sections significantly depend on driving cycle. In real-time, without using telematics information, there is not any information available

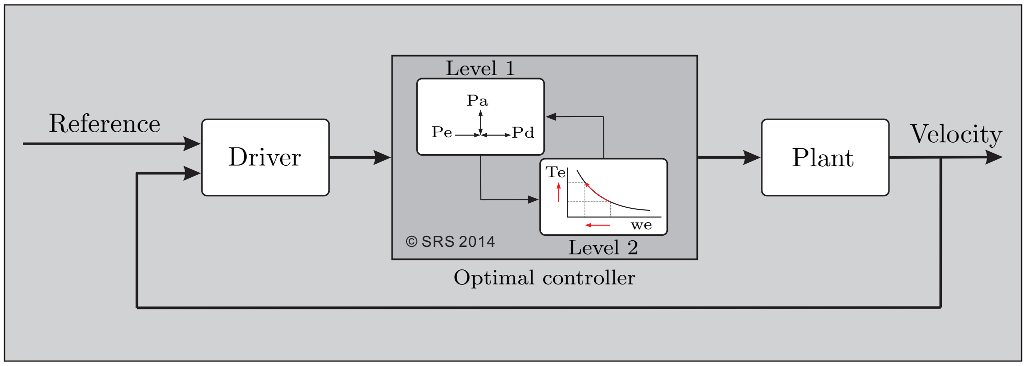

Figure 5.7: Instantaneously Optimized Power Management structure (IOPM)[KSedb]

about the whole or part of the upcoming driving cycle. On the other hand, rule-based power management methods discussed in chapter 4 are sub-optimal. Therefore, application of power management optimization approaches based on the instantaneous information about the vehicle velocity can overcome the problems relating to previously discussed power management approaches.

In this section an instantaneous optimized power management (IOPM) is applied to a SHHV. In figure 5.7, the structure of the IOPM power management is shown. The closed loop control system consist of a driver model to realize demanded power by tracking the driving cycle which is assumed as given. In real-time, demanded power is defined by driver according to road situation and human behavior. For this reason, a PI controller is applied to realize driver inputs, here brake and gas pedal. Time delay between power demand signal taken from driver model and real vehicle power demand is the main disadvantage of this power management strategy. Although the effect of this disadvantage can be reduced by reducing of time step size, it increases the computational load. In switching between two modes, the effect of this disadvantage is more critical. For simplification of the optimal control and computational load reduction, the problem is solved in three levels. In the first level, driver model feeds vehicle demanded torque. Power distribution between power sources is optimized in the second level while the sub-system, here pump displacement ratio is controlled in the third level. The boundary conditions are fed from level three to two. Therefore, feasibility of the solution is controlled iteratively between two levels. Hereby the power demand of the vehicle is considered to be completely supplied because the maximum engine power is more than vehicle power demand.

The only state of the model used in the second level is accumulator SoC. Line pressure is proportional to torque demand fed from driver model and the manipulating variable is motor displacement ratio. Motor power demand is described by

$$P_m = P_d/\eta_m, \tag{5.8}$$

in which η_m is the function of motor displacement ratio. In order to simplify the problem, accumulator power and demanded power are formulated as functions of the motor displacement ratio. Accumulator power is also the function of line pressure. Finally engine power as the function of motor displacement ratio is described by

$$P_e = (P_m - P_a)/\eta_P. \tag{5.9}$$

Furthermore, using backward optimization process in DP, boundary conditions corresponding to line pressure and accumulator flow rate are converted to boundary conditions for pump displacement ratio. The single state single variable optimal control problem is solved using Pontryagin's minimum principal. The low level controller consists of a decoupled model of the accumulator, line pressure, and engine. At each step the state, here SoC, as well as boundary conditions are updated for the next step.

5.4 Simulation results and discussion

Both MPC-based and IOPM power management strategies are applied to SHHV. The setup of the model for simulation of both power management strategies are same as DP-based power management. Accumulator size is 55 liter, while the maximum volume displacement of pump and motor are set to 54 and 72 liter per minute. Maximum line pressure is adapted to the maximum pressure of the accumulator, here 40 Mpa. Set pressure value of the PRV is adjusted to the maximum line pressure. For reduction of power losses through PRV, maximum value of line pressure for both power management approaches are set to the maximum line pressure. According to the technical data of the accumulator, maximum allowable accumulator pressure is 20 Mpa [BR14c]. In both acceleration and deceleration phases, maximum flow rate of the accumulator does not exceed from the maximum value of the motor.

5.4.1 Effect of prediction horizon

Performance of the MPC depends on parameters of the controller. The main parameter of MPC is prediction horizon. In this context, accuracy of the controller is defined as the integration of tracking error of the vehicle velocity respecting to the reference driving cycle. The increase of prediction horizon increases the accuracy as well as computational load of MPC. In figure 5.8, normalized value of computational load and tracking error for three prediction horizons are shown. Accordingly, computational load for prediction horizon 15 is 70% larger than prediction horizon 10, while its accuracy is improved only 3%. On the other hand, tracking error for the prediction horizon 5 is not comparable with one from prediction horizon 10, while its computational load is only 20% smaller. The corresponding results for prediction

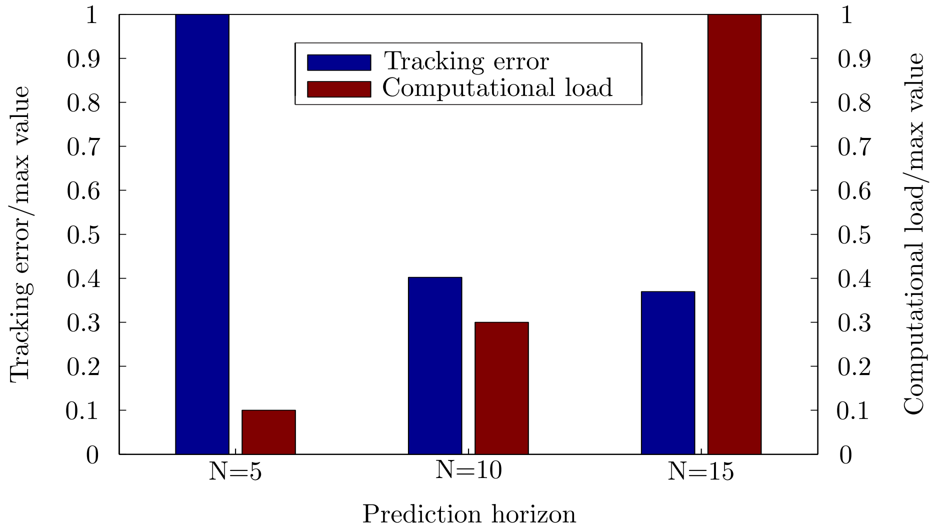

Figure 5.8: Comparison of tracking error and compuation load for different prediction horizon

horizon 10 shows better trade-off between accuracy and computational load of the MPC rather than two other prediction horizons. Therefore, in this contribution, prediction horizon 10 is used for simulation.

5.4.2 Performance

In deterministic DP-based power management, driving cycle is considered as known. Moreover, power demand is calculated using driver model in the loop. Therefore, the only objective function for DP-based power management is fuel consumption minimization while vehicle velocity tracking error is considered to be zero. In figure 5.9, normalized velocity tracking errors for MPC and IOPM strategies based on simulation results are shown. The results are normalized by dividing to the maximum values.

In section 4.2, the convectional vehicle is explained in detail. Therefore, smaller value represents better vehicle performance. Accordingly, IOPM shows better performance than MPC. In MPC-based power management, both fuel consumption and velocity tracking error are minimized simultaneously while in IOPM these are minimized in two stages. Simultaneous minimization of fuel consumption as well as vehicle velocity tracking error in hybrid hydraulic power management leads to a conflicting optimization problem. In such problems, simultaneous optimization of two objective functions can lead to a trade-off between optimal values of them. On the other hand, this trade-off depends on the weighting factors. For these reasons and based on the simulation results, performance of MPC is worse than those resulting using IOPM. Moreover, larger velocity tracking error causes inaccuracy of load

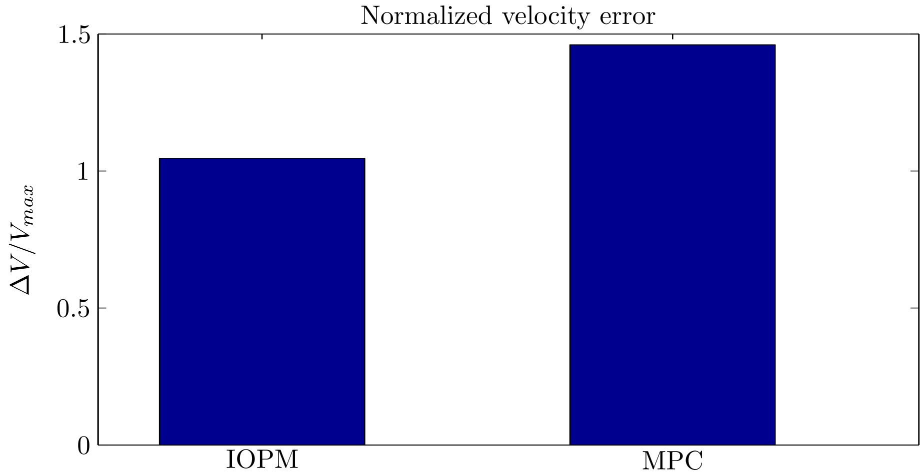

Figure 5.9: Comparison of scaled velocity tracking error

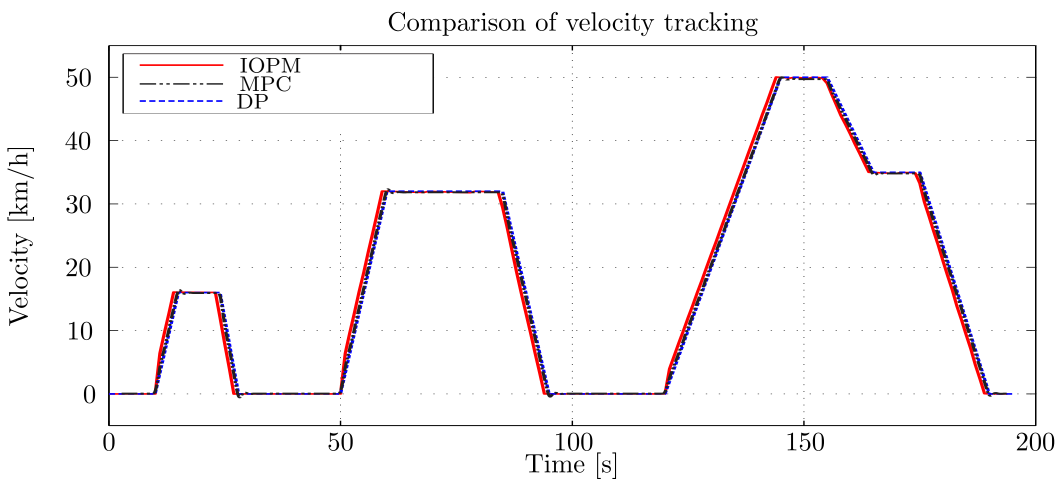

Figure 5.10: Reference velocity tracking

profile prediction which effects the efficiency of the power management algorithm. In figure 5.10, velocity tracking error is shown. For MPC-based power management, there is a time delay between predicted load profile and reference load profile. It maybe the main disadvantage of the MPC- based power management algorithm. Performance of both MPC and IOPM significantly depend on the control parameters which are considered as constant. Therefore, efficiency of the developed power management approaches are discussed in the next section.

5.4.3 Efficiency

In figure 5.11, the trends of the accumulator SoC for three power management strategies, namely DP, MPC, and IOPM and for ECE driving cycle are shown. The initial and final SoC values for all power management strategies are the same. For

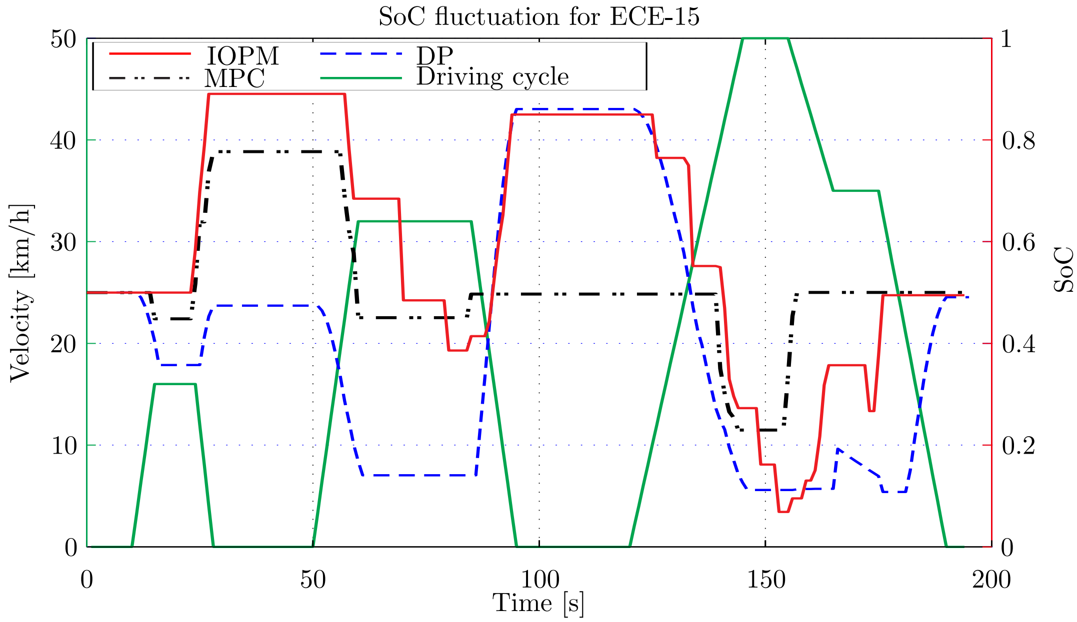

Figure 5.11: Fluctuation of SoC for ECE-15 driving cycle

this reason, the initial and final accumulator power values are the same. Therefore, the comparison of the SHHV fuel consumption values by applying the mentioned power management strategies is reasonable. All three power management strategies optimize the accumulator depleting pressure control strategy. However, their trends particularly for the first 100 second are different. The benchmark, here the DP-based power management shows proportional trends with driving cycle. The trends of SoC for the DP-based power management shows accumulator charge and discharge in all vehicle velocity levels. Due to the difference between the availability of load cycle for the DP, MPC and IOPM power management strategies similar SoC trends for them is not expected. Both IOPM and MPC power management strategies try to keep accumulator charge at high level. The maximum and minimum accumulator SoC for MPC shown by black dashed line are about 0.78 and 0.22 respectively, while these two values for two other power management strategies are about 0.85 and 0.1. It becomes clear that IOPM which optimizes the power distribution based on instantaneous vehicle load cycle does not use the whole accumulator operation range. Due to the lack of information about the upcoming vehicle load cycle, IOPM tries to keep accumulator charge moderately. The trends of accumulator SoC for the MPC which optimizes the power distribution based on the partial information about the future of vehicle load cycle, is close to the DP-based power management. Comparison of the accumulator SoC for IOPM and MPC shows the effect of load cycle information on the power management performance.

In figures 5.12 to 5.14, the power distribution between engine and accumulator respecting to the vehicle power demand are presented. From figure 5.12, it becomes

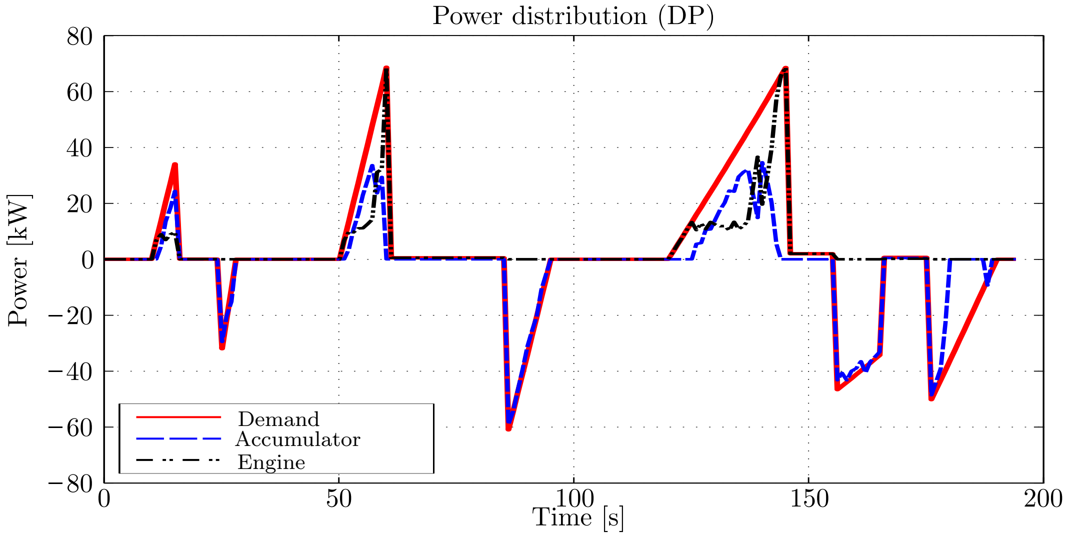

Figure 5.12: Power distribution in DP-based power management for ECE-15

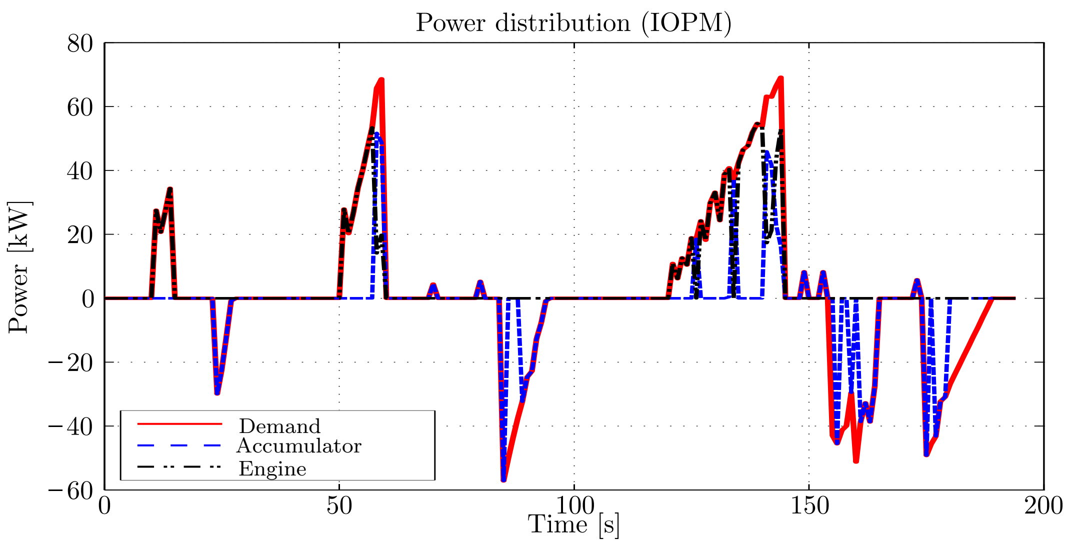

Figure 5.13: Power distribution in IOPM power management for ECE-15

clear that applying DP-based power management, SHHV is mainly supplied using accumulator power. In comparison to PCS II, it becomes clear that DP-based power management performance is almost same as PCS II. Power distributions corresponding to the application of IOPM and MPC to SHHV for ECE driving cycle are shown in figures 5.13 and 5.14 respectively. Both IOPM and MPC power management strategies try to shift the engine operation points close to the EOOP with the power value of 50 kW. However MPC shows better performance than IOPM. Comparison of the accumulator power contribution for both IOPM and MPC power management strategies shows that the performance of the IOPM for recapturing of braking power is more than MPC. Moreover, MPC uses more accumulator power than EOOP to

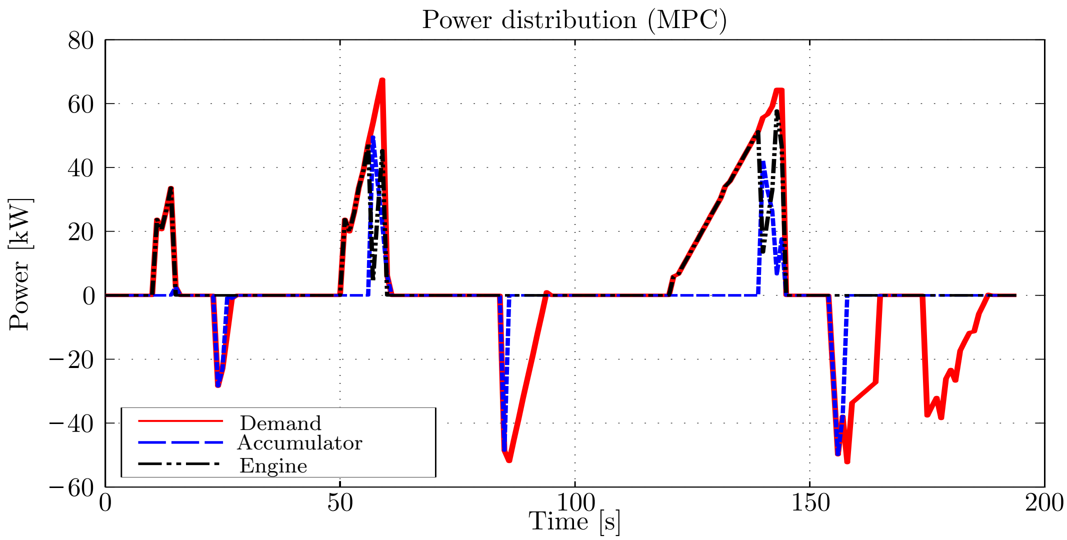

Figure 5.14: Power distribution in MPC power management for ECE-15

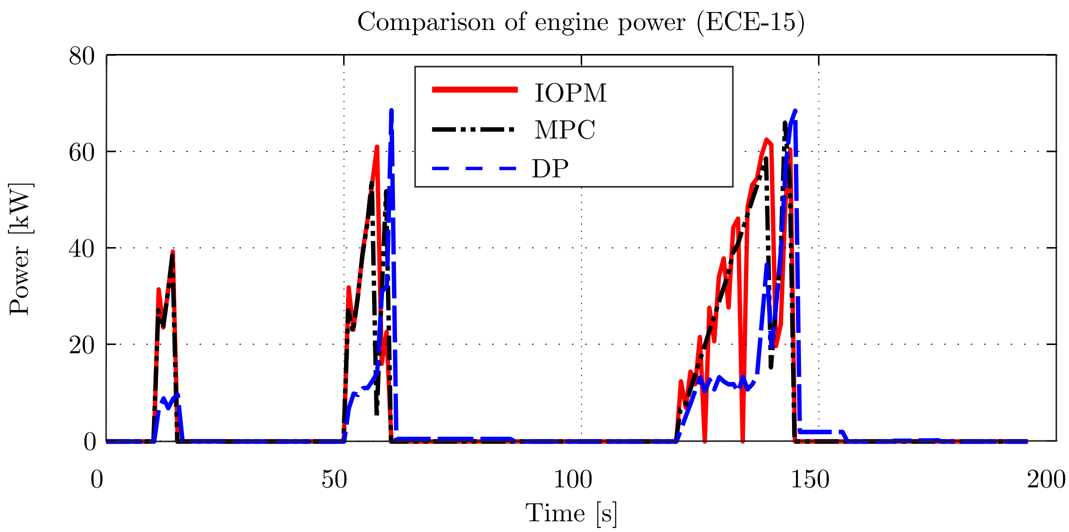

Figure 5.15: Comparison of engine power for ECE-15 driving cycle

supply vehicle power demands. Therefore performance of the MPC is comparable to the performance of the PCS III.

In figure 5.15, fluctuations of the engine power for three optimized power management algorithms are shown. The comparison of the maximum power values shows that the maximum engine power relates to the DP-based power management. Nevertheless the contribution of the engine power for DP-based power management is less than two other power management strategies, particularly at the beginning of acceleration phases. From the engine on/off switching graph related to the IOPM it becomes clear that IOPM performance is same as PCS I. Moreover, maximum engine power for the MPC is smaller than two other power management strategies.

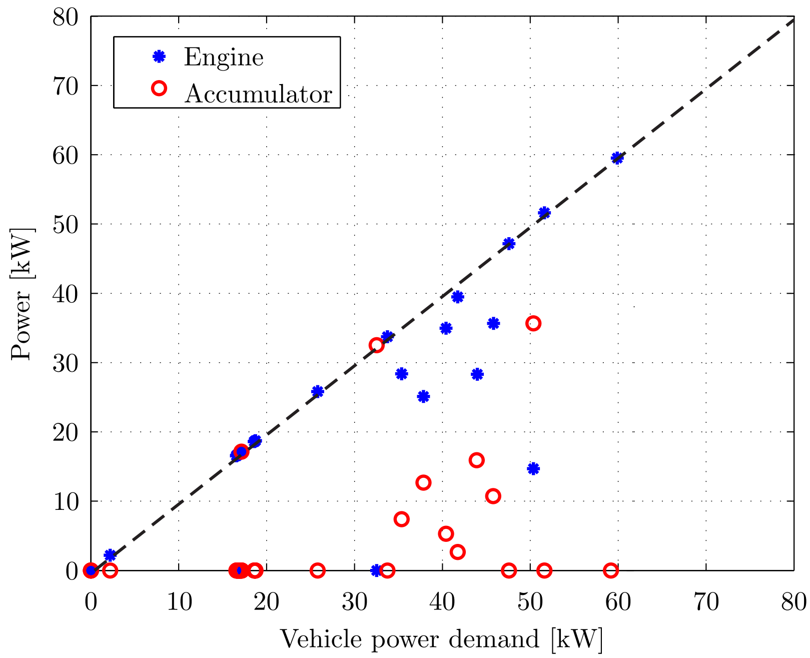

Figure 5.16: MPC power distribution rate for ECE driving cycle

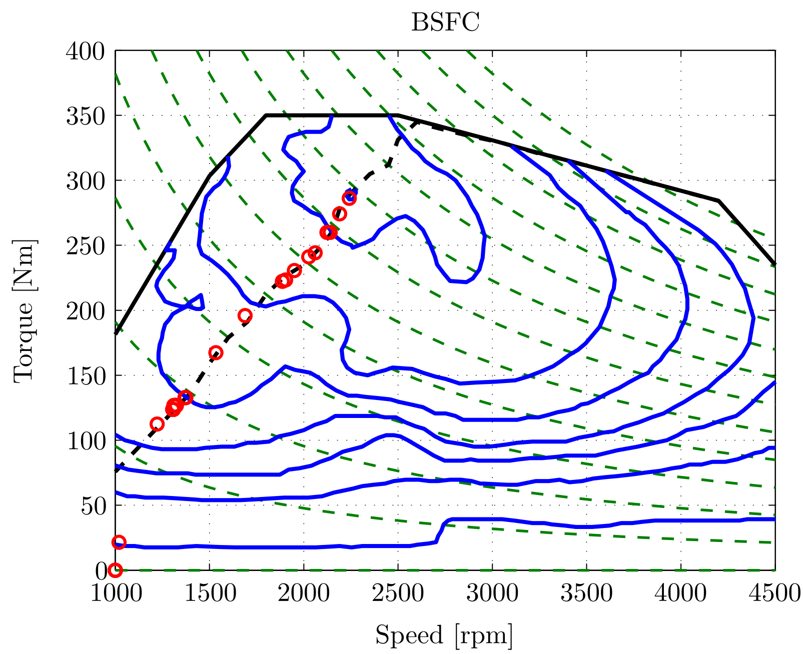

Figure 5.17: MPC Engine operation points for ECE driving cycle

5.4.4 Engine operation points

After presentation of the power distribution trends for three optimized power management methods, in this section the effects of the power management strategies

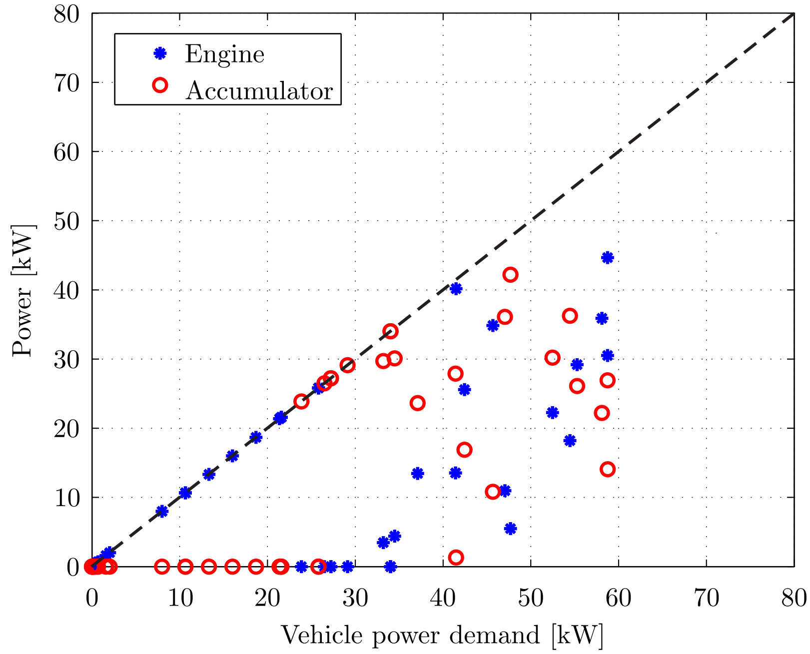

Figure 5.18: IOPM power distribution rate for ECE driving cycle

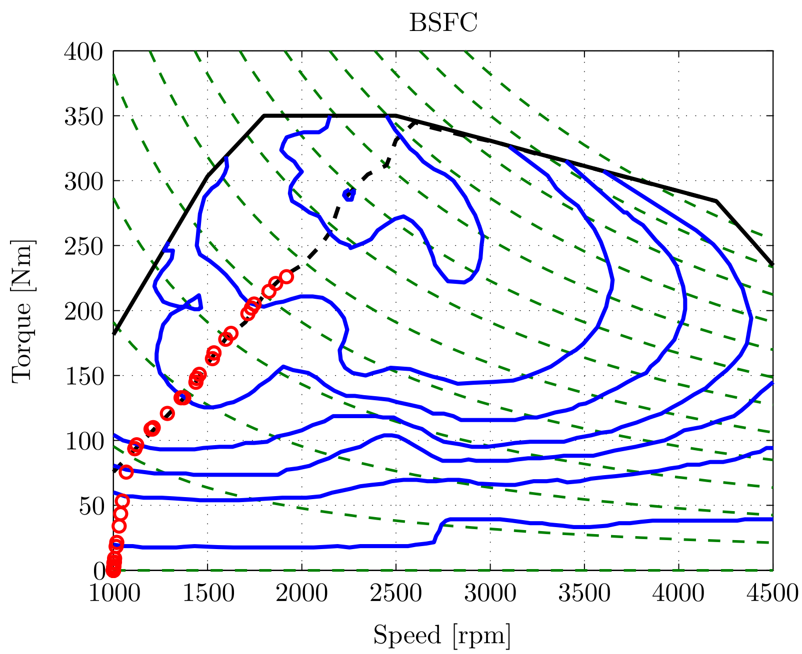

Figure 5.19: IOPM Engine operation points for ECE driving cycle

on the engine operation are discussed. For a reasonable comparison, the operation points of the engine on the engine BSFC map as well as point to point power distribution between power sources are presented in figures 5.16 and 5.17. The engine operation points as well as power distribution points related to the MPC are shown

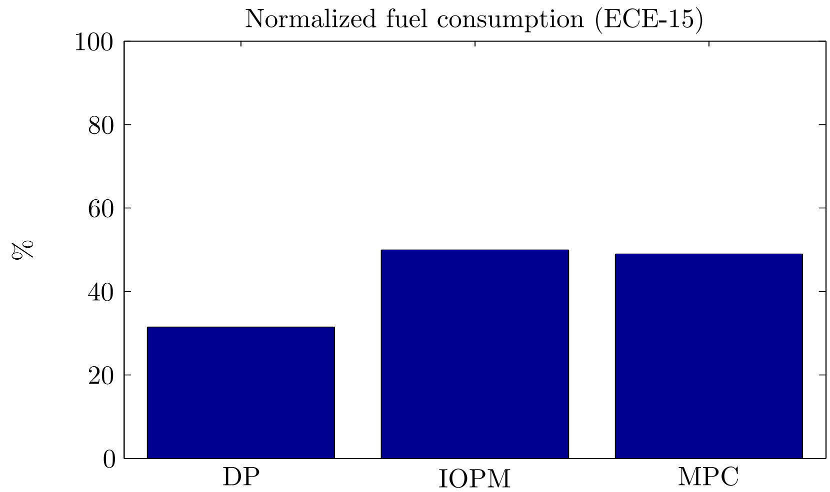

Figure 5.20: Comparison of fuel consumption for ECE-15 driving cycle

in figures 5.16 and 5.16. In figure 5.16, engine and accumulator powers versus vehicle power demand are shown. The dashed line represents the limit for the vehicle demanded power. Distributions of the engine and accumulator powers in different level of vehicle power demands are precisely shown. The blue points shown in figure 5.16 represent the engine operation points shown in figure 5.17. Limitation of the accumulator and engine power summation to the dashed line indicates that using MPC, accumulator is never charged using engine power. Accordingly the centralization of the accumulator operation points is in the middle powers between 30 to 50 kW. It can be concluded that the MPC shifts the engine operation points close to EOOP with assistance of the accumulator. This effect is shown precisely in figure 5.17. Centralization of engine operation points close to the EOOP indicate the ability of the MPC to realize EOOP engine control strategy discussed in section 4.1.2. The results corresponding to the IOPM are shown in figures 5.18 and 5.19. Accordingly accumulator operation points are centralized in power values more than 35 kW. In other words, IOPM limits the maximum engine operation points. This effect is precisely shown in figure 5.19. Engine operation points are centralized along the EOOL and the maximum value is limited to the 45 kW while the maximum power demand is 60 kW. It becomes clear that IOPM performance is same as the PCS III.

In order to compare the performance of the optimized power management methods to reduce fuel consumption, the value of fuel consumption are compared in figure 5.20. The values are normalized according to fuel consumption of conventional vehicle. The conventional vehicle is precisely explained in section 4.2. In figure 5.20, fuel consumption of conventional vehicle is considered to be 100 %. Accordingly, the efficiency of IOPM is same as MPC.

6 System analysis and optimal design of hybrid hydraulic vehicles

In the context of hybrid hydraulic power trains, besides power management optimization and vehicle load cycle, system optimal design is the other aspect influencing vehicle efficiency and performance. Optimal design of HHV consists of system topology determination, components size, and parameter adjustment. System optimal design is prior to power management. Nevertheless, simultaneous optimization of both power management and system design may significantly increase the system efficiency. However, solution of such a complex optimization problems containing multi-parameters and multi-objective functions is not straightforward. In the context of HHV, sizing relates to the components size determination in order to increase system performance. Depending on the HHV topology, accumulator volume and pump/motor volume displacement ratios are the main system components sizing parameters. The pump/motor displacement ratios are recognizable in relation to the vehicle and engine performance characteristics. However, they can be adjusted in the specific allowable range. In order to assist the engine, accumulator volume is the main parameter effects the HHV desired characteristics. The amount of oil volume can be captured within the accumulator is the function of accumulator volume. In addition to the sizing parameters, accumulator initial pressure effects the characteristics of the HHV. It determines the minimum accumulator operating pressure. According to the HHV power distribution concepts explained in the last chapters, line and accumulator pressures are the main control variables effecting the HHV power distribution. It becomes clear that accumulator volume and initial pressure are the main system design parameters [RHS12]. However, pump/motor volume displacement ratios effect the characteristics like regenerative braking torque value, braking time, and braking energy regeneration ratio [CV12]. Although, pump/motor volume displacement ratios have conflicting effects on the mentioned system characteristics. Therefore, system characteristics optimization by adjusting the main parameters can improve desired characteristics of the HHV [CQJ08].

In this chapter, the architectures of typical HHV topologies are investigated. By applying DP-based power management for optimization of PCS II control strategy developed in chapter 5, fuel economy of the typical HHV topologies are compared based on the simulation results. Sensitivity of the HHV performance to the most important system design parameters such as accumulator size and initial pressure are analyzed. Accordingly, a concept for SHHV design parameters optimization based on the criteria such as fuel consumption, velocity tracking error, Degree of Hybridization (DoH), and efficiency recovery rate is presented. Some parts of this chapter are published [KS12, MKS13].

6.1 Typical topologies of hybrid hydraulic power train

The architectures of the hybrid vehicles significantly effect system characteristics and power transmissions. Typical power transmissions used in HHV are mechanical and hydraulic transmission systems. Depending on the HHV topology, engine power can be transmitted only through a hydraulic system, a mechanical system, or combination of both hydraulic and mechanical systems. Due to difference between efficiency and performance of mechanical and hydraulic transmission systems, different HHV topologies have their individual characteristics and control approaches. The main disadvantages of hydraulic power transmissions are low efficiency and leakage while their advantages are smooth power transmission and infinite transmission ratios. Limited transmission ratios and stepwise power transmission are the main disadvantages of the mechanical transmission systems while high efficiency and high speed operation are the main advantages.

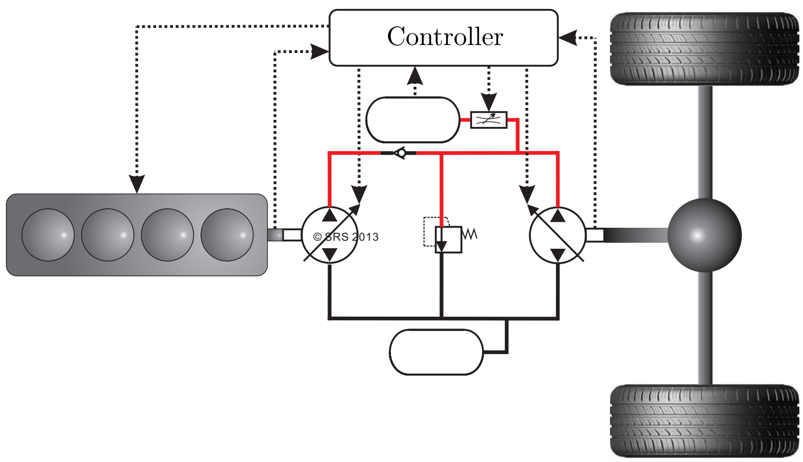

Figure 6.1: Structure of series hybrid hydraulic vehicle (SHHV) [KSedb]

6.1.1 Series hybrid hydraulic power train

By connecting two accumulators to the high and low pressure lines of a hydrostatic transmission system, series hybrid hydraulic vehicle (SHHV) which is also known as pure hydrostatic power train can be configured. In figure 6.1, the structure of SHHV is shown. This configuration consists of a pump directly coupled to the engine, a drive motor/pump directly coupled to the vehicle, and two bladder type gas charged accumulators connected to the hydraulic system houses. In order to capture braking energy, drive motor/pump switches to pump mode in braking phases. A directional valve in high pressure line placed between pump and accumulator, prevents flow

Table 6.1: Typical SHHV topologies

Topology	**Pump displacement**	**Motor/pump displacement**	**Accumulator connection**
A	Fixed	Variable	Direct
B	Variable	Variable	Direct
C	Fixed	Variable	Flow control valve
D	Variable	Variable	Flow control valve

regression to the pump in braking phases. In order to protect the system against failures due to high pressure, a PRV is integrated in the system. Within this configuration, accumulator can be charged either using engine or braking energies. Power sources operations in SHHV depend on the line pressure. In other words, the key parameter to control power distribution between accumulator and engine is the line pressure.

The advantage of this configuration is engine optimal operation due to mechanical decoupling of vehicle and engine. In addition, continuous variation of transmission ratio improves the vehicle performance. However, SHHV suffers from inefficient operation of the hydraulic components. For this reason, system optimal control in order to improve system efficiency is unavoidable. Vehicle performance depends significantly on the design characteristics as well as size of the hydraulic components [DAIS10]. Therefore, besides power management optimization, customization, and optimal design of the hydraulic system in order to improve system efficiency and performance are necessary.

Using typical control valves in SHHV, different HHV topologies can be realized. These topologies have their particular characteristics from performance, efficiency, and control points of views. The characteristics of HHV not only depend on the topology but also depend on the driving cycle. In the next section, typical configurations of SHHV are presented and discussed. In table 6.1, hydraulic components integrated in the four typical SHHV topologies are mentioned. Accordingly, all topologies are equipped with variable displacement motor/pump in order to supply vehicle demanded torque.

6.1.1.1 Topology A: Motor-controlled series hybrid hydraulic power train

The motor-controlled topology consists of a variable displacement motor and a fix displacement pump. In other words, the only control variable of the system is pump displacement ratio. The only valve used in this topology is a pressure relief valve for restriction of maximum line pressure. The first power source, here the engine, is used to compensate accumulator pressure drop. The second power source, here the accumulator, can be charged using either engine or braking energies. Due to direct connection of accumulator and line pressure, accumulator can be depleted till reaching the equilibrium line pressure. Due to the use of fixed-displacement pump, engine operation points depend on the line pressure and pump flow rate. Therefore, engine optimal operation and efficient power transmission through the hydraulic system cannot be guaranteed for this topology. However, simple architecture and control algorithms are the most advantages of this topology.

6.1.1.2 Topology B: Pump and motor-controlled series hybrid hydraulic power train

Pump and motor-controlled SHHV consists of a variable displacement motor and pump. Same as topology A, the only valve used in this topology is a PRV. Variable displacement pump increases the range of transmission ratio. Also engine and vehicle are mechanically decoupled. Therefore, by control of the pump and motor displacement ratios, fuel efficiency for this topology can be guaranteed. Due to direct connection of the accumulator and line pressure, line and accumulator pressures are in balance. The power flow is controlled by control of drive motor/pump while variable displacement pump controls the engine optimal operation respecting to the vehicle power demand and power management strategy.

The advantages of this topology are optimal operation of the engine, smooth power transmission, larger efficiency relative to other series topologies due to the use of hydraulic components and braking energy regeneration. However, dependency of the line and accumulator pressure limits the system pressure range to the accumulator maximum and minimum pressure. Moreover, accumulator power can not be controlled independent from the line pressure.

6.1.1.3 Topology C: Motor and accumulator-controlled series hybrid hydraulic power train

The structure of motor and accumulator-controlled topology is the same as topology A. However, it consists of different valve control systems. In this topology, the accumulator is connected to the line pressure using a flow control valve. Although, accumulator power is proportional to the accumulator and line pressure difference, it can be controlled using flow control valve. Within this topology, accumulator can be charged using both engine and braking energies. The main advantage of this topology is control of the accumulator power using flow control valve. On the other hand, power losses due to valves leakages and inefficient engine operation due to direct coupling of accumulator to the line pressure are the advantages of this topology.

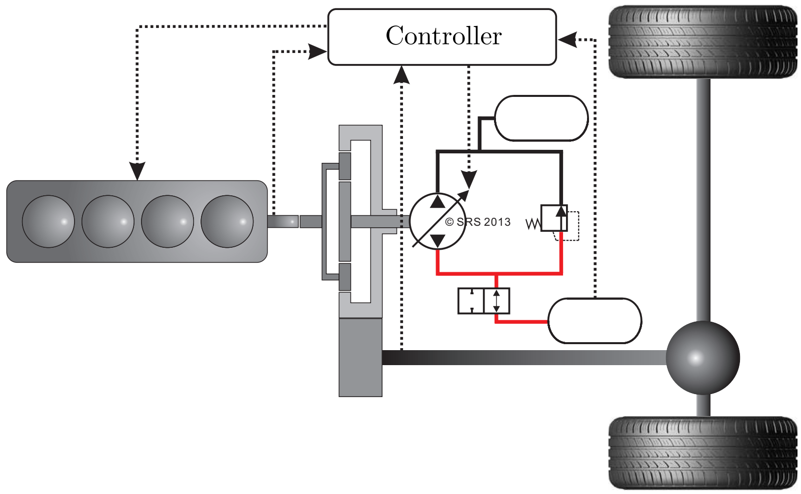

Figure 6.2: Structure of parallel hybrid hydraulic vehicle (PHHV)

6.1.1.4 Topology D: Pump, motor and accumulator-controlled series hybrid hydraulic power train

The structure of pump, motor and accumulator-controlled series hybrid hydraulic power train is depicted in figure 6.1. In this topology both pump and motor/pump have variable volume displacement ratios and accumulator flow rate is controlled using a flow control valve. This topology has the advantages of both topologies B and C. In contrast to topologies B and C, this topology has three control variables. Therefore development of power management strategy for this topology is more complicated than other topologies. Integration of more hydraulic elements in the structure of this topology increases the power losses due to leakages and frictions. Optimal control of this power train to improve its efficiency is one of the objective functions corresponding to the power management optimization. However, this task increases the complexity of the power management strategy. In addition, larger weight and high price due to the increase of the number of hydraulic elements are other disadvantages of this topology.

6.1.2 Parallel hybrid hydraulic power train

The parallel hybrid hydraulic vehicle (PHHV) which is also known as power assist topology [BRV$^+$10] is illustrated in figure 6.2. This configuration consists of a variable displacement motor/pump coupled to a conventional mechanical transmission system using a mechanical coupling such as planetary gear. In this topology, pump/motor volume displacement ratio and gear ratio of the mechanical gear box are the control variables. Maximum line and accumulator pressures are controlled

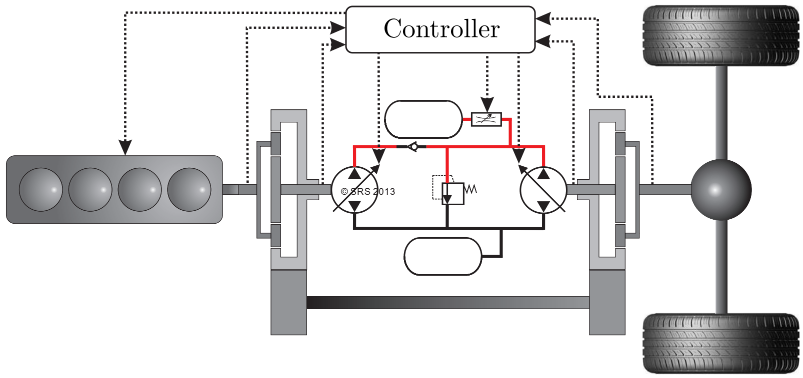

Figure 6.3: Structure of power-split hybrid hydraulic vehicle

Table 6.2: Typical topologies of power-split hybrid hydraulic vehicle [CLC11]

Type	**Engine to transmission**	**Transmission to vehicle**	**Number of modes**
Input-coupled	Simple gear	Planetary gear	6
Output-coupled	Planetary gear	Simple gear	6
Compound-coupled	Planetary gear	Planetary gear	11

using a pressure relief valve connected between high and low pressure lines. The advantages of this configuration are efficient engine power transmission through the mechanical transmission, easy implementation to convectional vehicles, and system operation reliability in case of breaking-dawn the hydraulic system. However, mechanical connection between vehicle and engine causes inefficient engine operation.

6.1.3 Power-split hybrid hydraulic power train

The power-split topology is the combination of series and parallel topologies. It utilizes the benefits of both series and parallel power trains in different operation conditions. In figure 6.3, the structure of power-split hybrid hydraulic vehicle is shown. Accordingly, engine power can be transmitted through two different paths, namely mechanical and hydraulic transmissions. Both transmission systems are coupled using two mechanical coupling (planetary gears) in each side. Using different mechanical couplings for connection of transmission system to vehicle and engine, different architecture for power-split hybrid hydraulic vehicle can be realized. In table 6.2, couplings which are used in typical power-split hybrid hydraulic vehicles are mentioned [CLC11]. More operation modes can be realized using compound-coupled power-split. Moreover efficiencies of both compound-coupled and input-coupled topologies are significantly more than output-coupled topology.

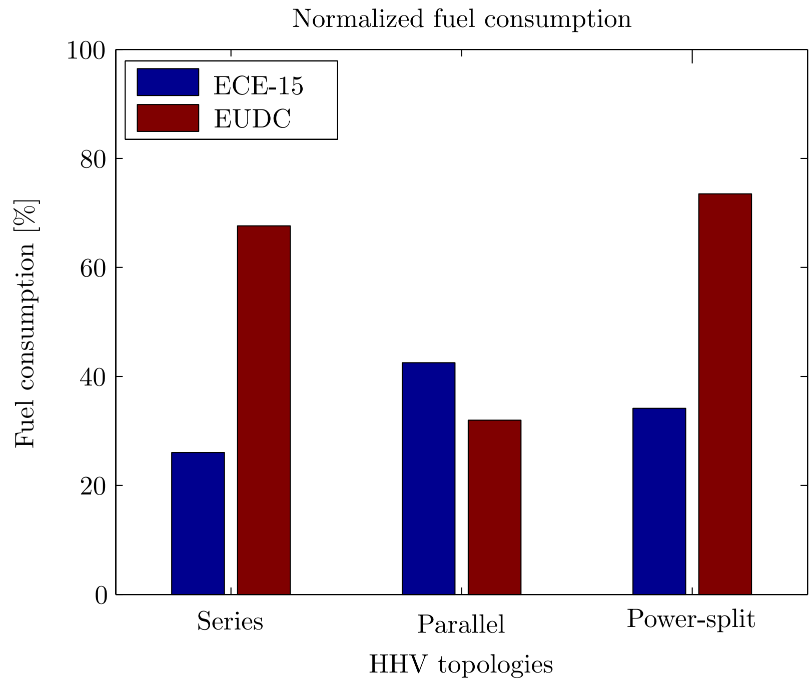

Figure 6.4: Comparison of fuel consumption of typical HHV topologies

6.2 Comparison of hybrid hydraulic power train topologies

Typical researches in the field of hybrid power trains deal with topologies design, comparison of topologies, and the evaluation of different topologies effects on the vehicle performance. An approach for optimization of the components size and HHV topologies is presented in [CLC11]. For this reason, using a general model, three different configurations of the power-split topologies are investigated. The comparison of the optimization results show that the input coupled power-split HHV is the most efficient topology. Despite the components size optimization, here the size of the accumulator as the key parameter is not optimized. The same method is applied to compare the performance and fuel consumption of the typical HEV topologies, namely series, parallel, and power-split [DCLC13]. According to the results the capability of power-split topology to reduce the fuel consumption is higher than two other topologies discussed. The characteristics of a hybrid mechanical power train (HMP) consisting of a continuous variable transmission (CVT) and flywheel, are investigated in [BRV$^+$10]. The size of the key components are optimized for the given constraints, in order to optimize several functionalities of the hybrid power train e.g. braking energy regeneration and engine efficient operation. The results demonstrate the potential of HMP to reduce fuel consumption. It can be concluded that the power management approaches and the performance and efficiency of the power trains significantly depend on the the power train topology.

In this section, typical HHV topologies are compared based on their fuel economy.

The DP-based power management is implemented for optimization of power distribution in three typical HHV topologies. For a reasonable comparison, size and parameters of the components are considered the same for all topologies. In figure 6.4, fuel consumption corresponding to typical HHV topologies for two different driving cycles are shown. Comparable fuel consumption of series and power-split topologies in EUDC indicates the tendency of power-split topology for power transmission through the hydraulic system. Moreover, high efficiency of parallel HHV in EUDC driving cycle indicates high efficiency of mechanical transmission in highway driving cycles. On the other hand, high efficiency of series topology in ECE driving cycle indicates its efficiency in stop and go driving cycles. In contrast to high fuel economy of parallel topology in EUDC driving cycle, its efficiency in ECE is the worst. The main reason is low recapturing rate of the system due to power transmission through mechanical power transmission. The worse efficiency of power-split topology in ECE driving cycle in comparison to series topology is due to system braking energy recapturing rate decrement. It can be concluded that with the same system parameters, series and parallel topologies have the best fuel economy respectively in city and highway driving cycles. However from the application point of view, power-split topology shows better performance in different driving cycles by adaptive operation.

6.3 Optimal design of hybrid hydraulic power train

Methodologies to optimize components size in order to improve HHV performance and efficiency are given in [PMGE13, MKS13, ML11]. Besides application of new power management optimization methods to HEV, the effect of operational costs such as vehicle acceleration requirements, on the components size and system cost are studied in [PMGE13]. Application of this approach for optimal design of the HEV is claimed. The same approach is applied to HHV [ML11]. A multi-objective multi-parametric global optimization strategy is developed in [MKS13] for optimization of HHV components size regarding globally optimized power management. The results show the most optimal components size considering the optimal power management. In [KS12], the effects of key parameters such as accumulator initial pressure on HHV are investigated. The results show the components size effect on both power train performance and efficiency. In [KI11], a method is proposed to select the size of the accumulator. Based on the average velocity of a given driving cycle, the maximum energy which can be captured during a single braking phase is described by

$$E_{accumulator} \geq E_{braking}, \tag{6.1}$$

in which $E_{braking}$ denotes the braking energy and $E_{accumulator}$ braking recaptured energy. Based on the vehicle kinetic energy and required work for compression of the accumulator's gas, the above relation becomes

$$\int_{V_1}^{V_2} pdV \geq \frac{1}{2}mv^2, \tag{6.2}$$

where p denotes the accumulator pressure, V the gas volume, m vehicle mass, v vehicle velocity, V_1 and V_2 the accumulator initial and final volumes respectively. Accumulator charge and discharge process are considered to be adiabatic [KI11]. Therefore, accumulator volume V_1 results from

$$V_1 \geq \frac{(1-n)(\frac{1}{2}mv^2)}{p_2(\frac{p_1}{p_2})^{\frac{1}{k}} - p_1}. \tag{6.3}$$

The histogram of the braking energy distribution in different vehicle velocity represents available energies for regeneration. The amount of vehicle kinetic energy in velocities less than average velocity of the driving cycle is considered for determination of accumulator volume. This method is very general and depends on the driving cycle. As a theoretical approach, here the effects of temperature loss in the accumulator are neglected.

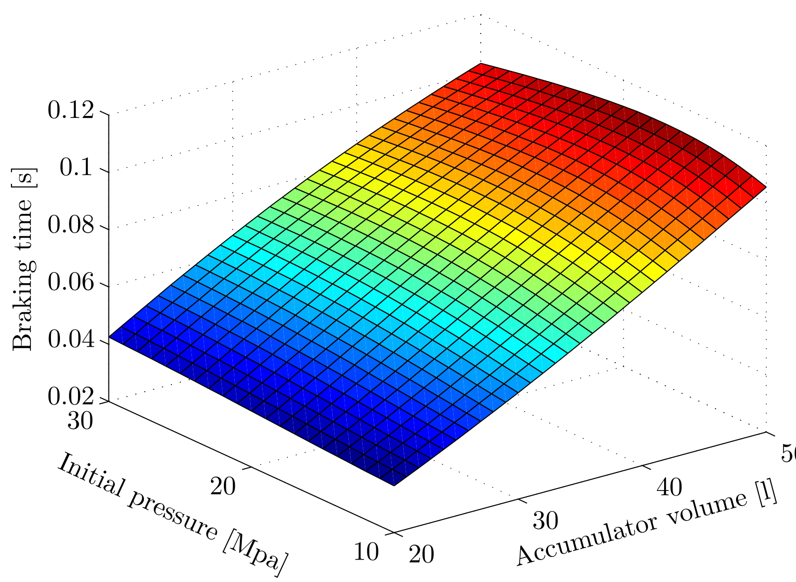

Figure 6.5: Effect of accumulator parameters on the vehicle braking time [KS12]

6.3.1 Accumulator parameters

Accumulator as the key component of the HHV influences the system performance. Two key parameters of the accumulator are volume and initial pressure which effect the static and dynamical characteristics of the accumulator. The main criteria for selection of the accumulator size are maximization of braking energy recapturing rate during braking phases and accumulator power supply during vehicle acceleration phases. In addition, vehicle acceleration in both braking and acceleration phases depend on the accumulator dynamic response. In brake phase, instead of conventional braking system, accumulator supplies power for deceleration. In order to improve vehicle safety, optimization of regenerative braking system is unavoidable. For this reason, the effects of accumulator size and initial pressure on the vehicle acceleration and deceleration are illustrated in figures 6.5 and 6.6 respectively. The setup of the model for this simulation is same as PHHV shown in figure 6.2. For the simulation of acceleration phase, vehicle is accelerated using pre-charged accumulator till the accumulator becomes fully depleted. In deceleration phase, the brake force is realized only using accumulator resistance force. For this reason, motor/pump volume displacements are considered as constant.

According to figure 6.5, braking time is used as the criteria for the evaluation of regenerative braking performance. Braking force realized by regenerative braking system is the function of accumulator pressure. The increase of motor/pump volume displacement ratios increase the braking force. On the other hand, the increase of the accumulator initial pressure increases the braking force. In figure 6.6, it is shown that more energy can be captured if the accumulator size is large enough. Also the accumulator energy level can be improved by the increase of the accumulator initial pressure. The ideal value or the initial pressure is the pressure in which the vehicle stops without using conventional braking system. The intensity of the braking as well as the stop time are additional factors affected by the initial pressure of the accumulator. It becomes clear that, due to reverse effects of the accumulator parameters on the system performance and efficiency, simultaneous adjustment of the accumulator parameters is undesirable.

6.3.2 Pump/motor parameters

Motor and pump sizes are other parameters influencing the HHV characteristics. The criteria to select pump and motor are their maximum power that they can transfer. In other words, the pump coupled to the engine, should be able to transfer engine maximum power, considering maximum torque and speed of the engine. The motor/pump coupled to the vehicle must be able to transfer maximum vehicle power demand in acceleration and braking phases. In acceleration phase, motor must be able to transfer the maximum vehicle demanded power considering maximum wheel torque and velocity of the vehicle. In braking phase, hydraulic motor must be

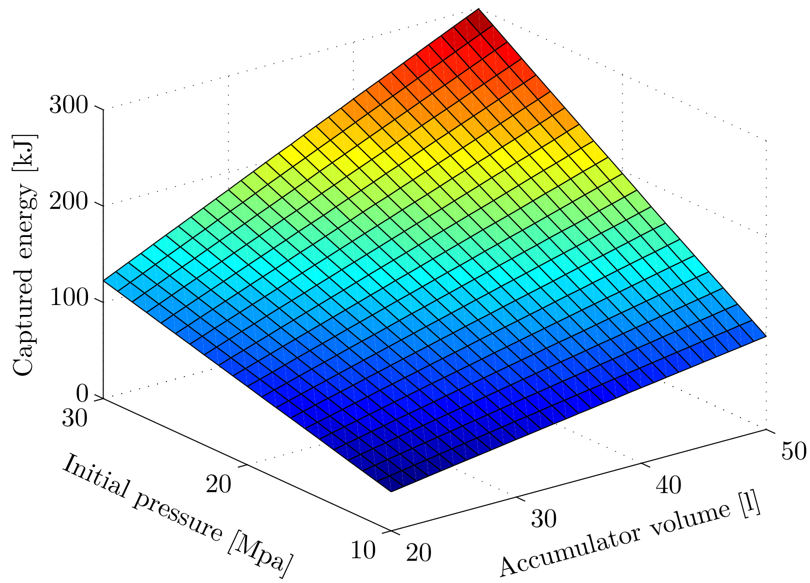

Figure 6.6: Effect of accumulator parameters on the captured energy [KS12]

able to capture as much as braking energy by the accumulator. In addition to the explained criteria, pump and motor efficiencies depend on the components size. According to the figure 3.3, pump/motor efficiencies for large volume displacement ratio and line pressure are large. Considering a constant torque value, by decreasing the pump/motor size its operation is shifted to high pressure level with low flow rates. Moreover, the internal parameters of the hydraulic system, line pressure, and oil flow rate are other criteria effecting the sizing. System maximum pressure is selected corresponding to the accumulator maximum pressure. Although the minimum line pressure can be less than accumulator minimum pressure, because accumulator pressure valve controls its low pressure. For the pump and motor size optimization, system is simplified to a hydrostatic transmission. According to the relation 4.6, maximum transmission ratio of the hydrostatic transmission is the function of pump and motor displacement ratios. Considering the boundary conditions corresponding to the engine and vehicle operations, size of pump/motor are determined by

$$\frac{\omega_{e-max}}{\omega_{v-max}} D_p \leq D_m \leq \frac{T_{v-max}}{T_{e-max}} D_p, \tag{6.4}$$

where ω_{v-max} denotes maximum wheel speed, ω_{e-max} maximum engine speed, T_{v-max} maximum wheel torque, and ω_{e-max} maximum engine torque. Considering maximum line pressures and pump and motor flow rates, additional boundary conditions for determination of the pump and motor volume displacement are determined by

$$\begin{aligned} \frac{T_{v-max}}{p_{m-max}} \leq D_m \leq \frac{Q_{m-max}}{\omega_{v-max}}, \\ \frac{T_{e-max}}{p_{p-max}} \leq D_p \leq \frac{Q_{p-max}}{\omega_{e-max}}. \end{aligned} \tag{6.5}$$

Based on the explained criteria and simulation results, engine, and vehicle predefined operational conditions such as maximum torque, power, and velocity, optimal size for the pump and motor are $54\frac{l}{min}$ and $72\frac{l}{min}$ respectively.

6.4 Optimal design approach

In the last chapters, rule-based and optimized power management algorithms are developed and implemented to the SHHV in order to improve fuel economy and driveability. In the following section, these characteristics are improved with respect to the optimization of system design parameters and power management strategies. According to the results presented in the last chapters, fuel consumption and driveability are two conflicting system characteristics. Power management strategy and components size are the main aspects effecting the trade-off between these two conflicting characteristics. Therefore, simultaneous optimization of them is one of the important goals in the context of HHV. In the following section, first corresponding objective functions applied for the components size optimization are introduced. The concept of the optimal design process is introduced in detail. The effects of design parameters on the cost functions are shown using simulation results. Optimal SHHV design parameters accordance to different driving cycles are presented based on the simulation results.

In figure 6.7, the process of HHV optimal design is shown. It consists of two optimization loops, one for power management optimization and the other for size and design parameters optimization. For optimization of both power management and design parameters, off-line global optimization methods are used. As realized for power management optimization, optimal design parameters are searched for a given driving cycle. Deterministic DP is used for optimization of power management strategies. For optimization of design parameters, NSGA II algorithm is used. Both DP-based power management and NSGA II algorithm are explained in chapter 5. Optimization process consists of design parameters initialization according to the feasible area of the system operation. Although the variation domain of design parameters must be predefined for the NSGA II optimization algorithm, with interdependency of the sub-systems as well as system complex dynamics, feasibility check is not a straightforward problem. In the backward-facing model for realization of DP-based power management, first the feasible area of the system operation is

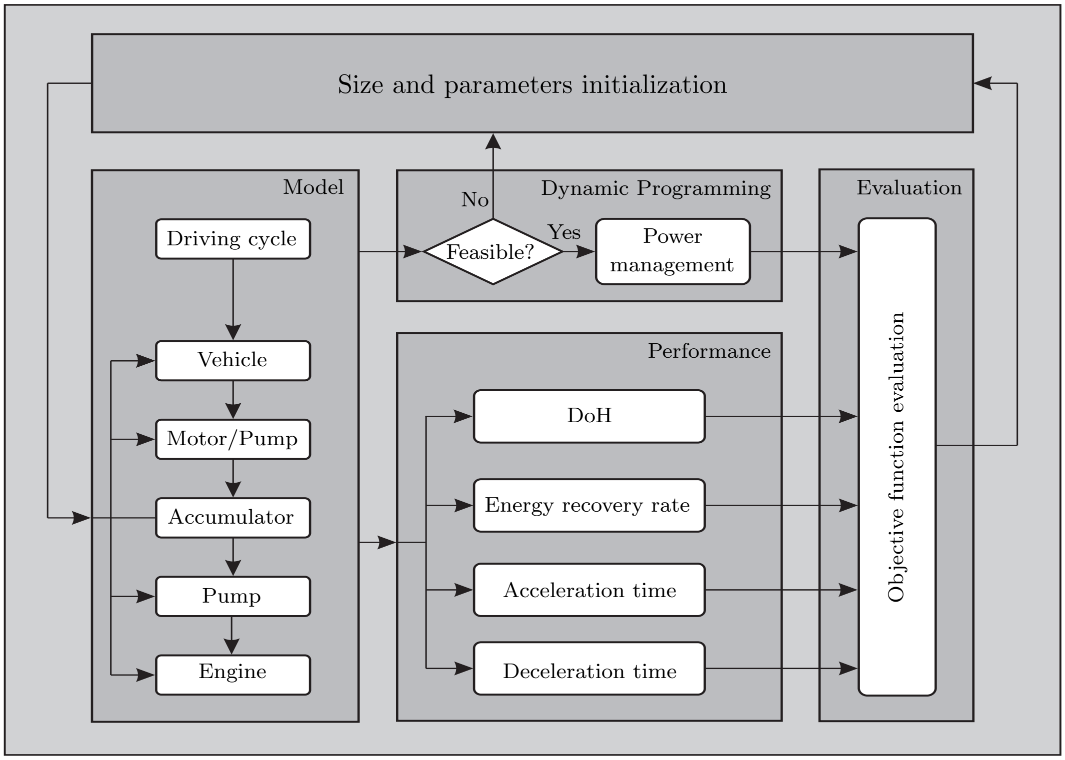

Figure 6.7: Concept of HHV optimal design process

checked. In the second step, performance of the system is evaluated in accordance to the corresponding evaluation criteria.

For the evaluation of the power management, several criteria are explained in chapter 4. Due to optimization of only vehicle performance using the explained criteria, the explained criteria are not satisfactory for the evaluation of design parameters. For simplification and satisfaction of both power management and design parameters optimization, different criteria are explained in this section. These criteria evaluate efficiency and driveability of the HHV. In order to optimize the efficiency and driveability simultaneously, both design and control parameters have to be manipulated in the optimization loop. For the evaluation of fuel performance, the criteria explained in section 3.3 is used. In order to evaluate the ability of the system to regenerate braking energy, energy recovery rate and DoH are used. The evaluation of the driveability is performed on the base of 0-100 km/h vehicle acceleration time. In the following section, these criteria are explained in detail.

6.4.1 Efficiency recovery rate

In order to evaluate system efficiency, system capability for braking energy regeneration is evaluated and optimized. By definition, energy recovery rate is the ratio of braking energy captured by the accumulator to the available kinetic energy [Ryd09]. This objective function is mathematically described by

$$f_{\eta_{rec}} = \frac{E_{accumulator}}{E_{braking}} = \frac{p_0 V_0^k \int_{V_1}^{V_2} pdV}{\frac{1}{2}mv^2}, \tag{6.6}$$

where $E_{accumulator}$ denotes recovered energy in the accumulator and $E_{braking}$ available braking energy.

6.4.2 Degree-of-Hybridization

In order to evaluate the maximum power supply of the hydraulic system, the ratio of the maximum engine power to the maximum vehicle power demand is applied as other objective function. This ratio is defined as the DoH and described by

$$f_{DoH} = \frac{P_{eng,max}}{P_{dem,max}}, \tag{6.7}$$

in which $P_{eng,max}$ denotes maximum power supplied by the accumulator and $P_{dem,max}$ maximum vehicle power demand respecting to the driving cycle. In other words, DoH evaluates the ability of HHV to overcome the maximum vehicle power demand. Small value of DoH indicates the ability of the accumulator to overcome maximum vehicle power demand. Therefore, DoH represents the ability of engine downsizing.

6.4.3 Acceleration time from 0-100 km/h

In order to evaluate the driveability of HHV, acceleration time for vehicle velocity change from 0 to 100 km/h is used. This is a usual criteria for driveability evaluation of conventional vehicles. Accordingly the objective function is defined by

$$f_{acc} = \frac{100}{a}, \tag{6.8}$$

in which a denotes vehicle acceleration value.

By summation of the objective functions, the multi-objective optimization problem is written in the form of a single-objective optimization problem defined by

$$f = k_1 f_{\eta_{rec}} + k_2 f_{DoH} + k_3 f_{acc} + k_4 f_{fuel}, \tag{6.9}$$

where k_i represents the weighting factors and f_i the objective functions. Due to the difference between the order of the objective function values, weighting factors have to be adjusted somehow that all of them have same contribution in the final

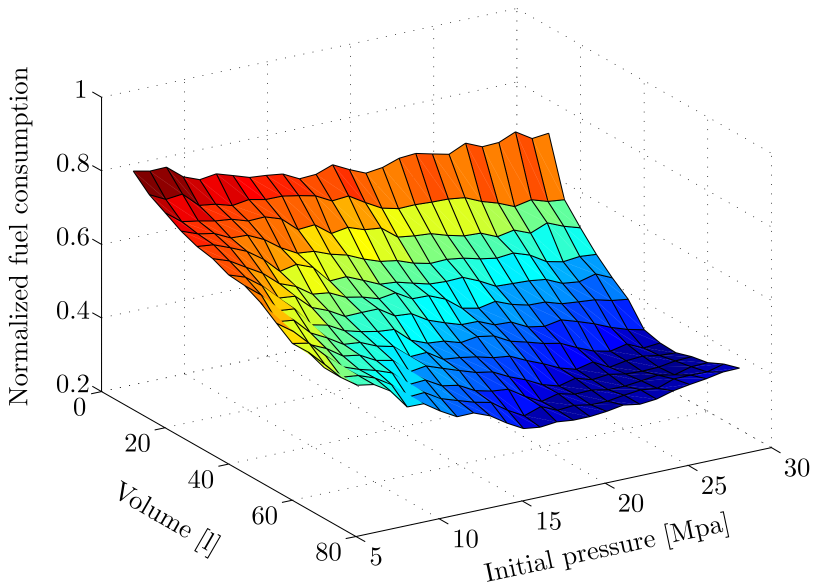

Figure 6.8: Vehicle fuel consumption corresponding to different accumulator parameters

weighting factor. Several criteria for the adjustment of objective functions weighting factors are developed such as adaptive adjustment of the weighting factors [MS12]. The method used in this contribution is based on normalization of the objective functions values. In order to unitize the effect of the objective functions in the main objective function, the weighting factors are selected somehow that the multiplication of each pair of weighting factor and objective function has the value between 0 and 1.

6.5 Simulation results and discussion

In figures 6.8 to 6.10, the relation of system parameters and objective functions are shown. It is shown that larger accumulator size with large value of initial pressure results the best fuel economy. The main reason is that a large accumulator can capture more power in the form of oil volume than a small accumulator. Moreover, large accumulator size with small value of initial pressure has small power density and may not fully supply the vehicle power demand. The reason is the value of accumulator pressure at the end of each braking modes. The increase of the accumulator initial pressure on one hand improves the accumulator power density, on the other hand declines the accumulator energy density. The domination of accumulator power density to the energy density is determined by power management.

In figure 6.9, the relation of DoH with variation of system parameters are shown. In contrast to fuel consumption, by increasing of the accumulator size and initial pressure, the value of DoH increases. For the engine downsizing, small value of DoH

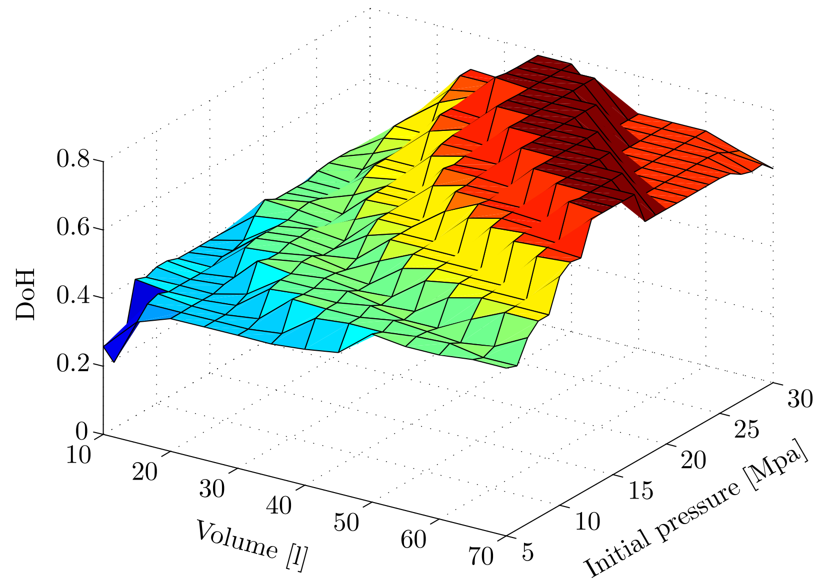

Figure 6.9: Vehicle DoH corresponding to different accumulator parameters

is desirable. Large DoH value indicates the maximum power of the engine while for the purpose of engine downsizing maximum engine power must be decreased by assistance of the accumulator. Therefore, DoH is in conflict with fuel consumption of the vehicle. It becomes clear that domination of energy density to power density of the accumulator or vice versa has conflicting effects on different aspects of the system.

Within deterministic DP, vehicle performance cannot be evaluated. For the evaluation of vehicle performance, vehicle acceleration time from 0 to 100 km/h is considered. The relation between system parameters and vehicle performance are depicted in figure 6.10. Sensitivity of acceleration time to the system parameters is equivalent to the sensitivity of DoH to the same parameters. In other words, larger accumulator has the large acceleration time while small accumulator size with large initial pressure value decreases the vehicle acceleration time. Accumulator energy density is proportional to accumulator volume while accumulator power density is proportional to accumulator pressure. In other words, by increasing accumulator volume its energy density increases while its power density decreases. Moreover, by increasing accumulator initial pressure, its power density increases while its energy density decreases. Energy density and power density are two properties which effect respectively efficiency and driveability of HHV. It becomes clear that not only the quantity of captured braking energy is important from the fuel economy point of view, but also quality of the accumulator power or maximum accumulator pressure is important from the vehicle performance point of view. According to figure 6.10, the accumulator size has a larger effect on the acceleration time than the accumulator initial pressure. The main reason is the reverse effect of accumulator size on the dynamical characteristics of the accumulator.

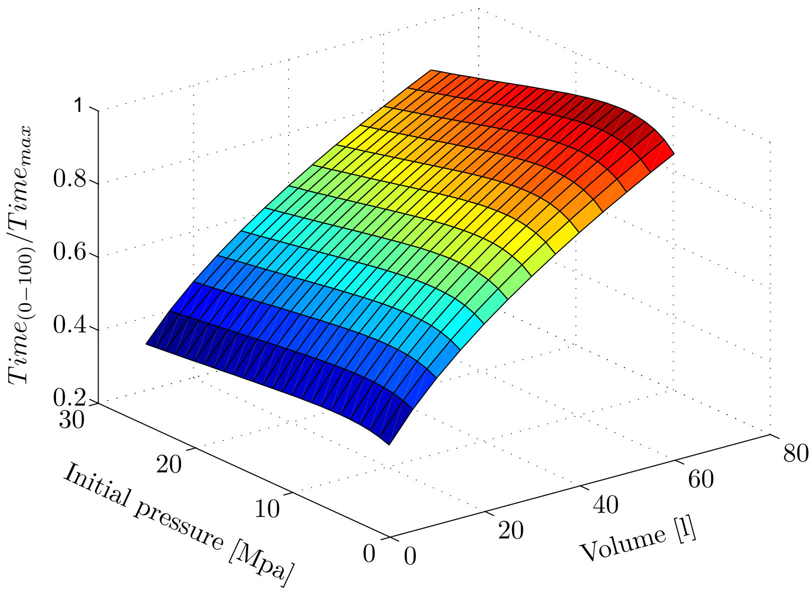

Figure 6.10: Acceleration time from 0-100 km/h corresponding to different accumulator parameters

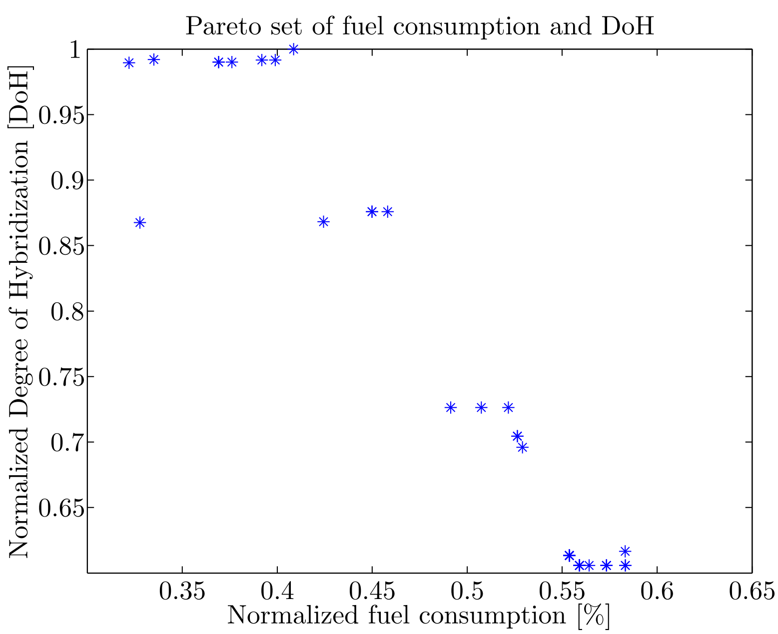

Figure 6.11: Pareto set of fuel consumption and DoH for SHHV design optimization for ECE driving cycle

As mentioned, optimal parameters are determined using NSGA II optimization algorithm. Optimization results are shown in figures 6.11 to 6.12. Two Pareto graphs in figures 6.11 and 6.12 demonstrate the conflict between objective functions. From the results shown in figure 6.11, it becomes clear that fuel consumption and DoH

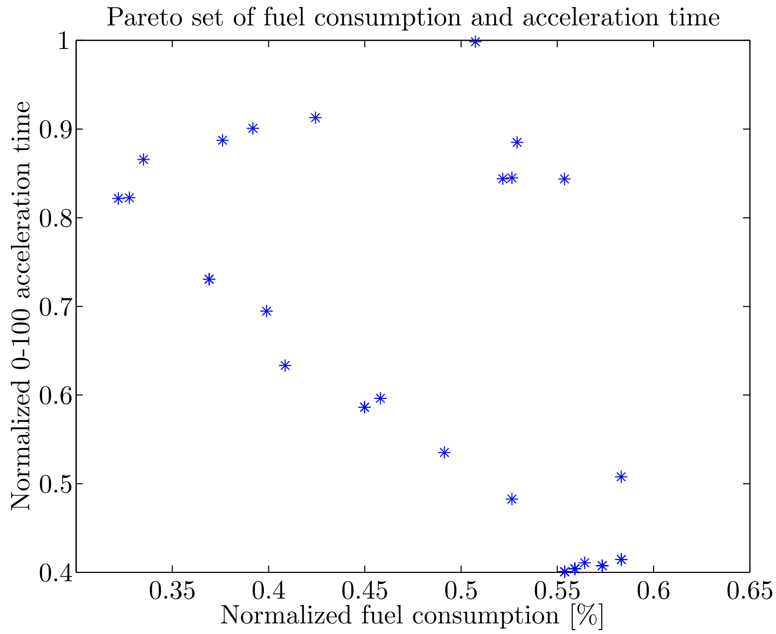

Figure 6.12: Pareto set of fuel consumption and acceleration time for SHHV design optimization for ECE driving cycle

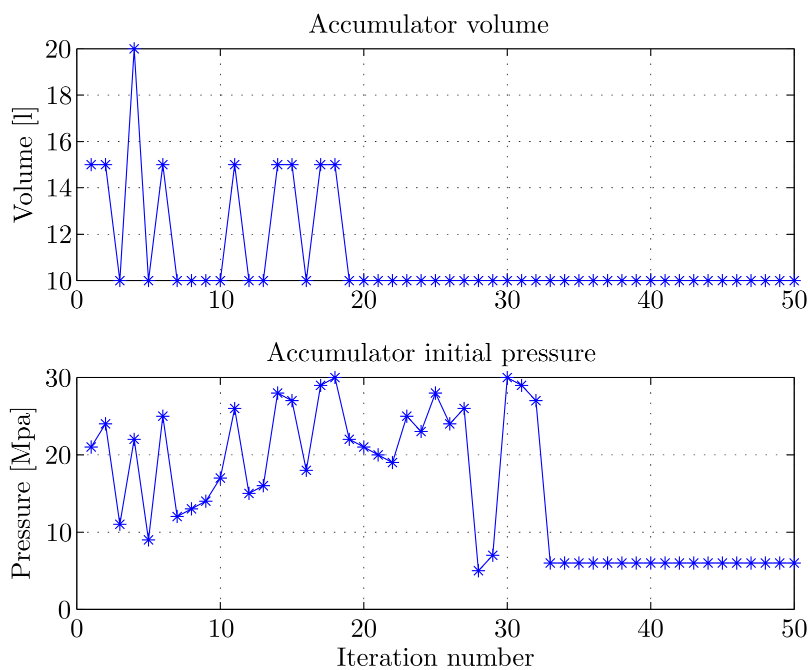

Figure 6.13: Trends of manipulation parameters

conflict because their values do not merge to their minimum values simultaneously. In other words, increase of the DoH decreases the fuel consumption value and vice versa. The same conflict between fuel consumption and acceleration time is shown

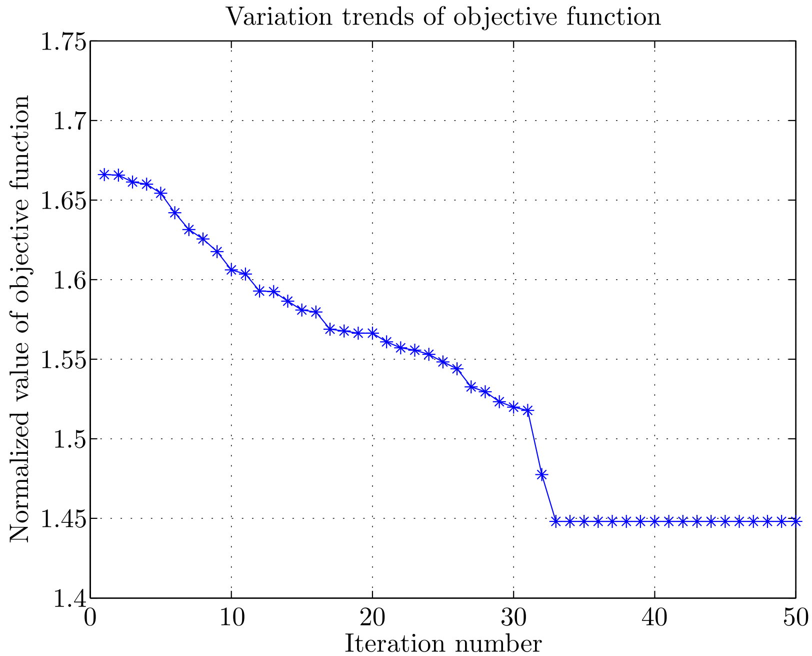

Figure 6.14: Variation trends of objective function

in the figure 6.12.

In figure 6.13, trends of the manipulating parameters, namely accumulator volume and initial pressure are shown. The optimal values of the accumulator volume and the initial pressure for SHHV topology and ECE driving cycles are 10 l and 6 MPa respectively. The results are based on simultaneous optimization of SHHV efficiency and driveability. The trends of the global cost function is shown in figure 6.14. For the given iteration number, the value of the objective function merges to its minimum value of about 1.448. Although these are individual results for the given driving cycle, this approach can be used for HHV design optimization by applying different driving cycles. Based on the results interpolation, appropriate design of the system for a combined driving cycles can be result. Although the results may differ, the application of same size accumulator to different topologies and different driving cycles without sacrificing the desired characteristics of the HHV is only possible with adaptive adjustment of the accumulator initial pressure. In other words, accumulator size is a fix design parameter while initial pressure as a control variable can be adaptively adjusted in order to improve the characteristics of HHV for different driving cycles [LVS07].

7 Summary and Outlook

This work focuses on the development of power management and control strategies as well as optimal design methodology for improvement of the desired characteristics of hybrid hydraulic vehicles. The main goal of this thesis is development of new pressure control strategies in order to realize optimal power management strategies for the hybrid hydraulic vehicles. Optimization of power management strategies and system structure using existing numerical optimization approaches are other goals achieved in this thesis.

7.1 Conclusion

The implementation of deterministic Dynamic Programming (DP)-based power management for a given driving cycle leads to an optimized control trajectory which serves a benchmark. The Benchmark can be utilized to improve the developed power management strategies. Due to large computation effort and previous knowledge requirement about the load cycle, DP and Genetic Algorithm are usually used for the realization of off-line power management approaches. Rule-based pressure control strategies developed in this work can be used as on-line and real-time power management because their computational loads are relatively small. Nevertheless, rule-based pressure control strategies are sub-optimal power management approaches. However their optimality can be improved using optimization algorithms. For this reason, the control parameters can be optimized using Non-dominated Sorting Genetic Algorithm II (NSGA II). Using globally optimal control trajectories taken from DP-based power management, the optimality of rule-based control strategies can be refined.

In order to develop on-line and real-time applicable power management strategy, vehicle load cycle can be predicted using Model Predictive Controller (MPC) approach. However, the accuracy of MPC significantly depends on the prediction horizon and or the system model. Due to operation of Instantaneous Optimal Power Management (IOPM) based on instantaneous vehicle load cycle information, it can be used for realization of real-time power management strategies. However, the main challenge for the application of IOPM is computational effort reduction. For this reason, system model is simplified using quasi-static sub-models. Considering the accuracy sacrifice limits, IOPM leads to a simple strategy.

Direct optimization methods (DP) and searching algorithms (NSGA II) are appropriate optimization approaches for optimization of both design and control parameters of HHV. In order to optimize components size and system parameters, NSGA II can search for the optimal parameters within the given boundaries for the parametrized topology while power management and fuel consumption are optimized using DP-based power management strategy.

7.2 Contributions

The main contribution of this thesis is the development of a comprehensive methodology for optimal design and control of the HHV with regard to driving cycle for future research in this field. The comprehensive literature research about hybrid vehicles power management strategies within this thesis gives the opportunity to select the appropriate algorithm for the optimal control of an arbitrary hybrid vehicle. Additionally, parametric modeling of the hydraulic elements and engine developed and verified here gives the opportunity to develop typical and arbitrary hybrid hydraulic power train topologies. The methodology used for the verification of simulation results here can be used for system parameters identification and evaluation of different HHV topologies.

Typical topologies are modeled using both dynamic and quasi-static sub-models in order to develop different control strategies and power management approaches. Based on the series hybrid hydraulic vehicle (SHHV) operation analysis, three types of power and pressure control strategies are developed and implemented to SHHV. Hence, this thesis has a contribution to a conceptual development, implementation, and comparison of rule-based, off-line, and on-line power management strategies. The ideas of rule-based pressure control strategies focus on power distribution control between engine and accumulator. For this reason, main line pressure is considered as the control input of the SHHV. The challenges of the problem focus on simultaneous optimization of vehicle power demand and reference velocity tracking error which are two conflicting objective functions. The implementations of typical driving cycles within simulation are realized to evaluate the performance of the developed rule-based pressure control strategies for different load cycles.

The implementation of optimization algorithms into the pressure control strategies are realized in order to optimize power management strategies. Power management strategy optimization problems are multi-objective multi-parametric problems containing distinguished object functions. Development and implementation of direct search algorithms, MPC algorithms, and instantaneous optimization algorithms in order to optimize the power management algorithms realize the real-time applicable optimized power management strategies with focus on problem simplification and computation load reduction. The implementation of the MPC algorithm allows the prediction of load profile and driver behavior in advance, for compensation of time delay due to significant large computational load of the iterative-based optimization algorithms. The implementation of heuristic direct search algorithms are realized to optimize system parameters, components size, and power management strategies simultaneously.

7.3 Outlook

Several aspects for the future work in the field of HHV can be proposed due to its novelty. Additional sub-models can be integrated to the HHV models in order to improve the model accuracy. For this reason, application of the engine model containing subsystems and sub-controllers can improve the model accuracy. Due to the low efficiency of typical hydraulic components, research topics in the field of high efficient hydraulic components development like pumps and valves are interesting. The main disadvantage of hydraulic accumulator is its small energy density. Application of open accumulator concept with adjustable gas initial pressure is an appropriate solution for accumulator low energy density. The possibility of parallel application of several accumulators is proposed as an alternative solution for the increment of accumulator energy density. The implementation of other optimal control approaches may lead to improve HHV desired characteristics. The implementation of improved estimation and prediction algorithms in order to improve performance and computational load of the power management for application into real-time control can be included as well.

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The thesis is based on the results and development steps presented in the following previous publications:

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In the context of the research projects at the Chair of Dynamics and Control, the following student theses have been supervised by Mohammad Ali Karbaschian and Univ.-Prof. Dr.-Ing. Dirk Söffker. Development steps and results of the research projects and the student theses are integrated to each other and hence are also part of this thesis.

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