

Institutionen för systemteknik

Department of Electrical Engineering

Examensarbete

Modeling and control of a Parallel HEV Powertrain with focus on the clutch

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan vid Linköpings universitet
av

Mahdi Morsali

LiTH-ISY-EX-15/4869-SE

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TEKNISKA HÖGSKOLAN

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Titel Title	Undersökning av ett problem Modeling and control of a Parallel HEV Powertrain with focus on the clutch
Författare Author	Mahdi Morsali

Sammanfattning Abstract
<p>Nowadays, the increasing amount of greenhouse gases and diminishing of the existing petroleum minerals for future generations, has led the automotive companies to think of producing vehicles with less emissions and fuel consumption. For this purpose, Hybrid Electric Vehicles (HEVs) have emerged in the recent decades. HEVs with different configurations have been introduced by engineers.</p> <p>The simulation platform aim for a parallel HEV, where the intention is to reduce the emissions and fuel consumption. The simulation platform includes an Electric Motor (EM) in addition to an Internal Combustion Engine (ICE). A new transmission system is modeled which is compatible with parallel configuration for the HEV, where the inertial effects of the gearbox, clutch and driveline is formulated. The transmission system includes a gearbox which is equipped with synchronizers for smooth change of gears.</p> <p>The HEV is controlled by a rule based controller together with an optimization algorithm as power management strategy in order to have optimal fuel consumption. Using the rule based controller, the HEV is planned to be launched by EM in order to have a downsized clutch and ICE. The clutch modeling is the main focus of this study, where the slipping mechanism is considered in the simulation. In the driveline model, the flexibility effects of the propeller shaft and drive shaft is simulated, so that the model can capture the torsional vibrations of the driveline. The objective of modeling such a system is to reduce emissions and fuel consumption with the same performance of the conventional vehicle. To achieve this goal first a conventional vehicle is modeled and subsequently, a hybrid vehicle is modeled and finally the characteristics of the two simulated models are studied and compared with each other.</p> <p>Using the simulation platform, the state of charge (SOC) of the battery, oscillations of propeller shaft and drive shaft, clutch actuations and couplings, energy dissipated by the clutch, torques provided by EM and ICE, fuel consumptions, emissions and calculation time are calculated and investigated. The hybridization results in a reduction in fuel consumption and emissions, moreover, the energy dissipated by the clutch and clutch couplings are decreased.</p>

Nyckelord Keywords	parallel hybrid electric vehicles, clutch modeling, driveline modeling, torsional vibrations, slipping clutch, energy management, optimization
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Abstract

Nowadays, the increasing amount of greenhouse gases and diminishing of the existing petroleum minerals for future generations, has led the automotive companies to think of producing vehicles with less emissions and fuel consumption. For this purpose, Hybrid Electric Vehicles (HEVs) have emerged in the recent decades. HEVs with different configurations have been introduced by engineers.

The simulation platform aim for a parallel HEV, where the intention is to reduce the emissions and fuel consumption. The simulation platform includes an Electric Motor (EM) in addition to an Internal Combustion Engine (ICE). A new transmission system is modeled which is compatible with parallel configuration for the HEV, where the inertial effects of the gearbox, clutch and driveline is formulated. The transmission system includes a gearbox which is equipped with synchronizers for smooth change of gears.

The HEV is controlled by a rule based controller together with an optimization algorithm as power management strategy in order to have optimal fuel consumption. Using the rule based controller, the HEV is planned to be launched by EM in order to have a downsized clutch and ICE. The clutch modeling is the main focus of this study, where the slipping mechanism is considered in the simulation. In the driveline model, the flexibility effects of the propeller shaft and drive shaft is simulated, so that the model can capture the torsional vibrations of the driveline. The objective of modeling such a system is to reduce emissions and fuel consumption with the same performance of the conventional vehicle. To achieve this goal first a conventional vehicle is modeled and subsequently, a hybrid vehicle is modeled and finally the characteristics of the two simulated models are studied and compared with each other.

Using the simulation platform, the state of charge (SOC) of the battery, oscillations of propeller shaft and drive shaft, clutch actuations and couplings, energy dissipated by the clutch, torques provided by EM and ICE, fuel consumptions, emissions and calculation time are calculated and investigated. The hybridization results in a reduction in fuel consumption and emissions, moreover, the energy dissipated by the clutch and clutch couplings are decreased.

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Notation

Notation	Description	unit
H	Hamiltonian equivalent	J/s
P_f	Fuel power	J/s
P_{ech}	Electrochemical power	J/s
λ	The constant of Hamiltonian equation	[-]
m_v	Vehicle mass	Kg
v	Vehicle speed	m/s
F_t	Traction force	N
F_a	Aerodynamic force	N
F_r	Rear resistance	N
F_g	Gravity forces	N
F_d	All other disturbing forces	N
ω_t	Gearbox angular speed	rad/s
i_t	Gear ratio	[-]
T_t	Gearbox torque	$N.m$
b_t	Gearbox damping coefficient	$N.m.s/rad$
T_p	Propeller shaft torque	$N.m$
J_t	Inertia of the gearbox	$Kg.m^2$
J_{cl}	Inertia of the clutch	$Kg.m^2$
J_{gear}	Inertia of each gear	$Kg.m^2$
J_p	Inertia of propeller shaft	$Kg.m^2$
J	Clutch total inertia	$Kg.m^2$
ω_{cl}	Clutch angular speed	rad/s
ω_{ICE}	ICE angular speed	rad/s
T_{ICE}	ICE torque	$N.m$
J_{ICE}	Inertia of engine	$Kg.m^2$
T_{cl}	Clutch torque	$N.m$

Notation	Description	unit
P_{cl}	Power dissipated by clutch	J/s
E_{cl}	Energy dissipated by clutch	J
k_p	Propeller shaft stiffness	$N.m/rad$
c_p	Damping of propeller shaft	$N.m.s/rad$
k_d	Drive shaft stiffness	$N.m/rad$
c_d	Damping of drive shaft	$N.m.s/rad$
θ_t	Gearbox angular rotation	rad
θ_f	Final drive angular rotation	rad
ω_f	Final drive angular speed	rad/s
i_f	Final drive ratio	[-]
b_f	Final drive damping coefficient	$N.m.s/rad$
J_f	Final drive inertia	$Kg.m^2$
ω_w	Wheel angular speed	rad/s
T_d	Drive shaft torque	$N.m$
T_{em}	EM torque	$N.m$
i_{em}	EM current	A
L_{em}	EM inductance	H
R_{em}	EM resistance	Ω
ω_{em}	EM angular speed	rad/s
k_t	Torque constant of EM	$N.m/A$
k_{em}	Back EMF constant	$N.m/A$
U_{em}	EM voltage	V
U_{max}	Maximum voltage of battery	V
SOC	State of charge	[-]
Q_{max}	Maximum capacity of the battery	$A.s$
SOC_{init}	Initial state of charge	[-]
$M_{consumed}$	Total amount of fuel consumed in drive cycle	Kg
\dot{m}	Fuel and air flow rate	Kg/s
L_{100Km}	Fuel consumption	$L/100km$
λ_d	air/fuel equivalence ratio	[-]
A/F	air/fuel ratio	[-]
m_a	Air mass	Kg
m_f	Fuel mass	Kg

Introduction

The thesis is a joint project with Linköping University¹, Kongsberg Automotive² and Vicura Engineering Academy³ and is about modeling a powertrain for a parallel Hybrid Electric Vehicle (HEV), which aims at developing a simulation platform that enables studies of the functionality and the performance of transmission concepts for HEV vehicles. Specifically, the functionality of the clutch, synchronizers, its actuation, and the possibilities with different gear coupling configurations in combination with the control of the transmission, the Electric Motor (EM), and the Internal Combustion Engine (ICE). The objective with the studies is to verify by simulation, a concept solution which leads to low fuel consumption, low emissions, less but sufficient hardware contents (downsizing clutch, minimizes size and number of mechanical synchronizers or preferably manages without them) that decreases space, weight and cost.

In this chapter, similar research subjects related to the thesis are presented. Also, the methods and tools which will be used to simulate the problem are discussed and explained; additionally, the general goals of the thesis are introduced. The available simulation platform and the blocks which shall be modeled will be discussed.

1.1 Purpose and goal

The objective of this thesis is to design a model to study the functionality of the transmission system and reduce the emissions and fuel consumption compared to a conventional vehicle.

¹<http://www.vehicular.isy.liu.se/>

²<http://www.kongsbergautomotive.com/>

³<http://www.vicura.se/>

Forward modeling will be used to simulate the system in MATLAB Simulink⁴, where the model will follow a drive cycle and based on the speed demand from a drive cycle, the controller will decide to either use the acceleration or brake pedal. Generally, there are two methods used for modeling the vehicles namely reverse modeling (quasi-static modeling) and forward modeling (dynamic modeling). In reverse or quasi-static method the speed and acceleration of the vehicle and the slope of the road is known, hence, the dynamic of the wheel is known and the required torque which shall be supplied by power source can be calculated, however, in dynamic modeling or forward modeling the model is based on mathematical description of the system (Guzzella and Sciarretta [2007]). Using forward modeling the system is defined with a set of differential equations and gives further insight to the driveline and vehicle dynamics (Koprubasi [2008]).

A conventional vehicle model including driver model, Mean Value Engine Model (MVEM), Engine Control Unit (ECU) and clutch and gearbox models, is available which is the same as the model used in project 1 of the course TSFS09 of the Vehicular System division⁵.

In order to capture the torsional vibrations in the driveline while starting or shifting a gear, a flexible propeller shaft and drive shaft will be included in the driveline model.

In order to connect the EM to the gearbox a synchronizer together with dog clutch will be used. The vehicle is supposed to be launched by EM and there is no need to modulate the clutch during launch, therefore the clutch can be downsized. The dry operating clutches will be used for this purpose.

The overall structure of the conventional model is illustrated in Figure 1.1. The given model will be first used as baseline for simulations. Then it will be transformed into a HEV model aiming for fuel consumption and emission reduction. The following components will be added to conventional vehicle model to transform it to HEV:

- EM
- Driveline (propeller shaft, drive shaft, final drive)
- Clutch and gearbox
- Transmission control unit
- Clutch actuation
- Energy management strategy
- Drive cycle selection

As shown in Figure 1.1, the model includes driver model, ECU, engine, clutch and gearbox, driveline and vehicle dynamics. The blocks shown with green color will remain unchanged and the blocks shown in light brown will be changed. The light brown blocks are clutch-gearbox and driver model. In addition, an EM and energy management blocks will be added to the model.

⁴<http://www.mathworks.se/>

⁵<http://www.vehicular.isy.liu.se/en/Edu/Courses/TSFS09/>

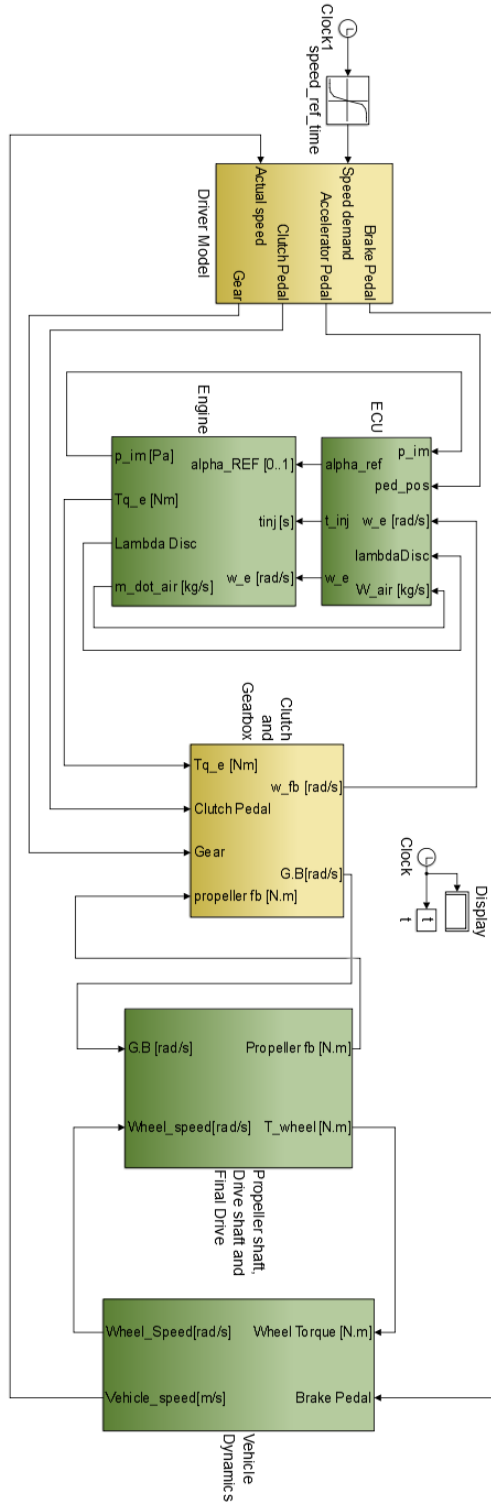


Figure 1.1: A baseline model of a conventional vehicle.

1.2 Problem Formulation

The thesis aims at developing a simulation platform that is suitable for investigating clutch characteristics and its control in a parallel hybrid electric powertrain.

Figure 1.2 shows a schematic of the transmissions for parallel HEV which is suggested by Vicura. The suggested transmission system is intended for a conventional vehicle where it can be used as a HEV transmission system by applying changes in the gearbox. The transmission system includes five speeds (five gear ratios), four synchronizers, and a clutch to connect the engine to the gearbox. In the gearbox model the selection of the gears are done using the synchronizer together with dog clutches. A synchronizer simply synchronizes the speed of the input shaft to the output shaft and after equalizing the speeds the two disks of the cone clutches move towards each other to make contact between dog clutches. For the gear transmission system three synchronizers shall be used and each synchronizer can be actuated by a hydraulic, pneumatic or electromechanical actuator.

As illustrated in Figure 1.2, synchronizer 1 can connect the EM to the input shaft of the gearbox. The other synchronizers (2, 3 or 4) can be actuated to transmit the torque from input shaft to output shaft. As shown in the figure synchronizers 3 and 4 are double-sided and can move to left and right to select the desired gear. The details on synchronizer actuations are mentioned in Section 2.6

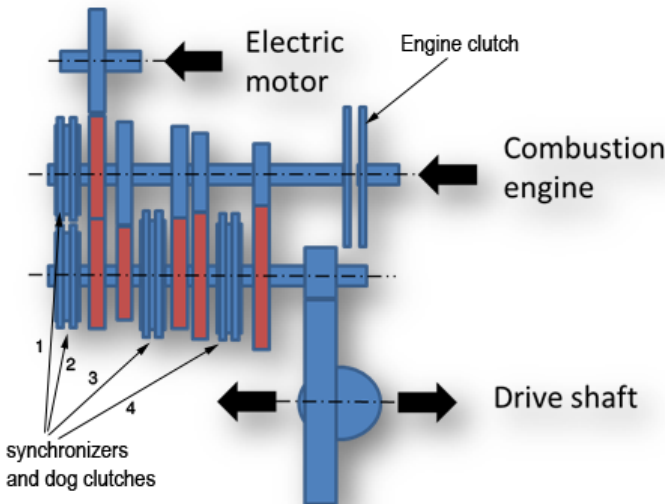


Figure 1.2: A schematic of the parallel HEV transmission including gearbox and driveshaft.

HEV's can be controlled using different configurations, where most typical configurations are series and parallel. In Figures 1.3 and 1.4 powertrain configuration

in parallel and series hybrid vehicles are shown:

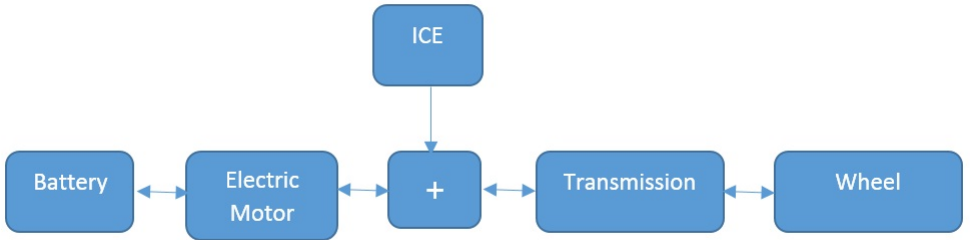


Figure 1.3: Parallel HEV configuration.

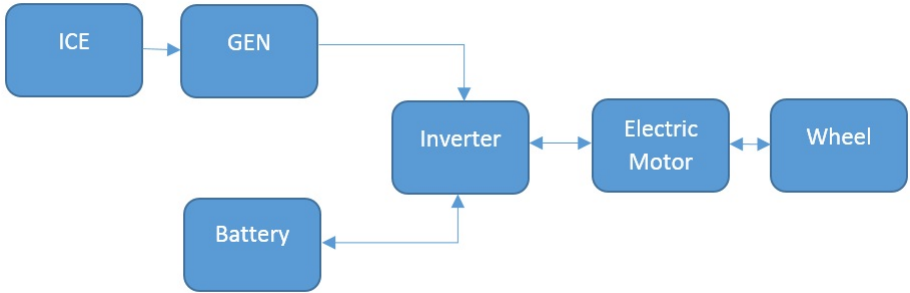


Figure 1.4: Series HEV configuration.

As shown in Figure 1.3, the transmission system in parallel configuration will work by both ICE and EM. In series configuration the EM will provide propelling power to the vehicle and the power for EM will be provided by both battery and generator.

In this thesis the conventional vehicle model will be used as baseline to model a parallel HEV. The complete model will have sub-models covering driveline, EM, energy storage, clutch, engine and HEV controller. In addition to these vehicle components there will be a drive cycle selector to upload different drive cycles and test the simulated model. The driveline model will have torsional vibration effects from drive shafts and propeller shaft. In the complete system it will be possible to simulate and study vehicle launch, regenerative braking, ICE or EM start and stop, vehicle acceleration with gearshifts, as well as run complete driving cycles. The thesis will also develop an energy management strategy for the hybrid configuration and test it in the simulation platform. Finally, using a drive cycle the proposed control strategy for the parallel HEV will be tested. Fuel consumption

by the powertrain or optimal SOC for the battery or the speed of engine can be plotted and compared to other control strategies

1.3 Related research

This section covers the related research investigations in the literature, relevant to the thesis. The most important part is the transmission system and its control. The related research is about the clutch control, synchronization and its mechanism, driveline torsional vibrations and energy management.

1.3.1 Clutch Control

Clutches can be classified into wet and dry while considering the operating condition. Wet clutches are those which operate in a fluid because of large amount of energy that might be dissipated (Orthwein [2004]). In dry clutches less energy will be dissipated and therefore the fuel consumption is less compared to the wet clutches. On the other hand wet clutches can damp vibration easier and smoother motion is expected. Wet clutches are mostly used for off-highway applications, agricultural applications (Dutta et al. [2014]) and also in automatic transmission.

In order to have a perfect clutch control first the quality of the gear shifting and all the affecting parameters should be investigated. The quality of shifting depends on shift shock and shifting time (Lee et al. [2000]). When the discs are about to meet each other, unstable vibrations because of speed difference might occur which is called resonant vibrations (Lee et al. [2000], Kugimiya et al. [1995]). A clutch consists of two plates, the plates can have two behaviors while they meet each other, they can either slip or lock-up to each other. When the clutch is in engaged mode, two plates will have same angular velocities ($\omega_v = \omega_e = \omega$), the clutch will remain in locked mode when the friction torque is still less than the maximum friction torque of the two plates of clutch. In Minh and Rashid [2012], the slipping and engaged behavior of the clutch has been formulated and simulated.

In order to control driveline vibrations, different strategies might be used. T. Kugimiya et al have developed an automatic transmission fluid for slip-controlled lock up clutch systems. In this system the lock-up clutches slip continuously, while they engage at low speeds (Kugimiya et al. [1995]).

Another method to reduce the resonant vibration is to control the speed of the connecting shafts via the clutch. H.D Lee et al have used the speed control method to reduce the speed difference while connecting two shafts; this research paper focuses on the clutch control of a parallel HEV. While the clutch is dis-engaged the speed controller will check the speed of the induction machine, next the speed of the ICE will be checked and the controller will make the speed difference minimum to make the two halves of the clutches ready to connect (Lee et al. [2000]).

L. Glielmo and F. Vasca have designed a controller for a dry clutch using feedback control. In the mentioned research, the authors have an approach to minimize the control time. The control strategy is possible by speed control of the crank

shaft rotor and clutch disk rotor. The objective is to achieve smooth shifting by minimizing the slipping effect (Glielmo and Vasca [2000]). In (Ercole et al. [1999]), the authors have considered a fuzzy controller for a servo-actuated transmission system for standing starts and gear shifts. The objective of the controller is to minimize the dissipated power by driveline and prevent engine stall during standing starts. V. T. Minh and A. A. Rashid have formulated the clutch and designed a controller for a parallel HEV; fuzzy controller have also been used to control the slipping and lock-up modes of the clutch. The controller input is supposed to be the throttle openings and the output is slip gain of the clutch (Minh and Rashid [2012]). W. Lhomme et al (Lhomme et al. [2008]) have also used the same torque equations for slipping and lock-up mode in (Minh and Rashid [2012]), they have proposed switched causal modeling for transmission of parallel HEVs. Nonlinear behavior of the clutch and its governing equations make it difficult to control.

Different transmission systems can be applied for different applications. For a heavy duty vehicle, an automated transmission (AT) is not appropriate because of huge amount of losses, on the other hand, continuous variable transmission (CVT) can only be used for small passenger cars (Lee et al. [2000]). Automated manual transmissions (AMT) are appropriate choice for heavy duty vehicles since they are fuel efficient and can be easily implemented using a manual gearbox and a pneumatic actuator (Lee et al. [2000]). Depending on different applications and required demands from a vehicle, different transmission strategy and hence different clutch control will be used. A. Schmid et al have designed a CVT equipped hybrid vehicle (Schmid et al. [1995]), which is more fuel efficient than the normal transmission system. Using CVT the operating points of the Spark Ignited (SI) engine will move towards higher efficiency areas and consequently the total amount of fuel consumption will decrease (Schmid et al. [1995]).

1.3.2 Synchronization

In transmission gearbox, in order to have a smooth connection between drive shaft and driven shaft synchronizer can be used. In Figure 1.5 different parts of a synchronizer are illustrated. A synchronizer typically locks the input shaft to the output shaft; using a synchronizer the speed of the input shaft and output shaft will become equal before the engagement of dog clutches. Dog clutches are used to transfer the torque and speed without slipping. The sleeve part of the synchronizer can move in axial direction to grab engagement gear. In (Tseng and Yu [2015]) a controller for synchronizer mechanism can be found, which is designed for rapid and accurate gear shift.

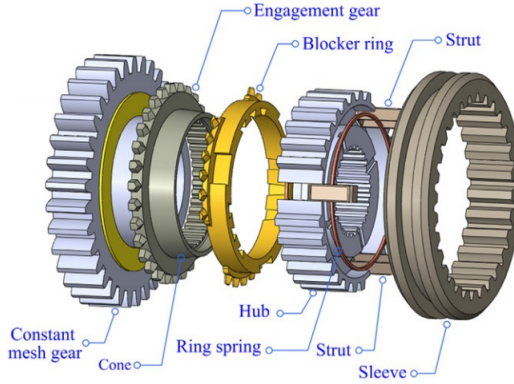


Figure 1.5: Different parts of a synchronizer (Tseng and Yu [2015]).

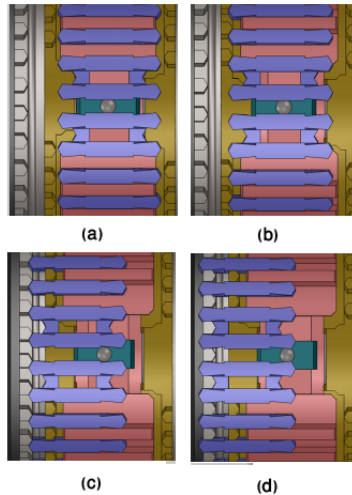


Figure 1.6: Different steps regarding the engagement mechanism of the synchronizer (Berbyuk et al. [2012]).

As mentioned before synchronizer is used to synchronize speed in order to have a smooth connection between input shaft and output shaft. A synchronizer has three main parts including cone clutch, translational detent and a dog clutch. According to Figure 1.5 when the sleeve moves towards the cone, the frictional torque equalizes the speed between cone clutch and hub. When the force acting on the sleeve exceeds the detent force the dog clutches can engage. In order to have a better understanding of the synchronizer mechanism Figure 1.6 has been provided. As shown in the figure the sleeve approaches the cone clutch and after providing enough force from actuator side the sleeve will move to it's final position

to fully engage and transfer the torque.

For more detailed analysis of synchronizer it is recommended to refer to Berbyuk et al. [2012].

1.3.3 Driveline formulation

Figure 1.7 shows different subsystems engaged in driveline.

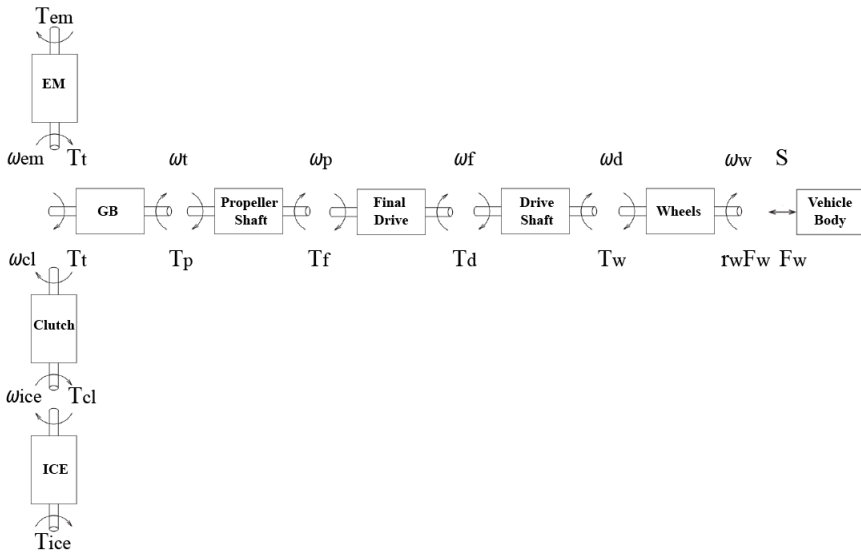


Figure 1.7: Subsystems included in vehicle driveline.

The easiest way to model the behavior of driveline is to consider an infinite stiffness for different parts of driveline. However, a flexible driveline can be more realistic and more accurate. A flexible driveline has been modeled in Eriksson and Nielsen [2014] using Newton's equations, where in this project same procedure will be used to model the driveline. In Figure 1.8 a schematic driveline with flexible propeller shaft, drive shaft and final drive can be seen.

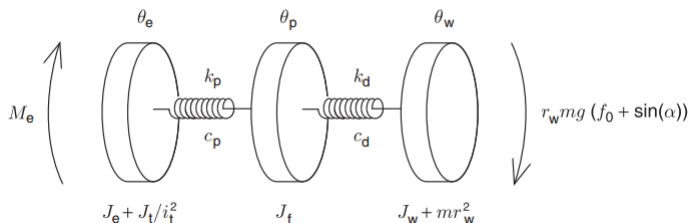


Figure 1.8: A schematic of flexible driveline (Eriksson and Nielsen [2014]).

For more details on driveline modeling it is recommended to see Eriksson and Nielsen [2014].

1.3.4 Energy management

In order to control a parallel HEV different methods can be used. The common approaches to control a HEV are optimal control, sub-optimal control and heuristic algorithms (Pérez et al. [2006], Thounthong et al. [2015], Chasse and Sciarretta [2011]). Heuristic approaches can be divided into rule-based controllers and fuzzy logic controllers. The rule-based controller or fuzzy logic controller can be designed based on the optimal results obtained from optimal control analysis. In the rule based methods it is not required to have information about the entire cycle, however, the results may not be perfect because fuzzy logic does not give the solution based on an equation, however, this method is easy to implement and computationally inexpensive. On the other hand, optimal control finds the optimal solution for a certain criteria by using methods such as deterministic dynamic programming. Phatiphat Thounthong (Thounthong et al. [2015]) and his colleagues have designed a fuzzy logic controller for a vehicle with short term storage system where the short term storage is a supercapacitor. Deterministic Dynamic Programming (DDP) is an example of optimal control method. To solve a problem using DDP the entire drive cycle should be known in advance. Pérez et al. [2006] have used dynamic programming to find an optimal power management in a series HEV. Equivalent mass minimization is another optimal control which can be used to minimize overall fuel consumption. In a hybrid vehicle in order to calculate the optimal fuel consumption, the power used by combustion engine and equivalent electric power used by electric motor should be minimized. For this purpose the Hamiltonian minimization can be used (Chasse and Sciarretta [2011]):

$$H = P_f + \lambda \cdot P_{ech} \quad (1.1)$$

Where H is the Hamiltonian equivalent, P_f is the power consumption by fuel, P_{ech} is the electrochemical power consumed by the battery and λ is the variable which makes the electrochemical power equivalent to power of fuel. In a parallel HEV finding optimum SOC is of interest, therefore the derivative of H with respect to SOC should be calculated:

$$\frac{\partial H}{\partial SOC} = \frac{\partial P_f}{\partial SOC} + \frac{\partial P_{ech}}{\partial SOC} \cdot \lambda + \dot{\lambda} \cdot P_{ech} = 0 \quad (1.2)$$

Solving these equations, the minimum amount of energy which can be consumed by vehicle can be calculated in each state. Dongsuk Kum and his colleagues have done a research on supervisory control of a parallel electric vehicle by minimizing the total mass consumption (Kum et al. [2011]).

The related research on the clutch modeling, synchronization, driveline formulation and energy management form a background for further investigation and simulation. In the simulated model some of the concepts mentioned in related research are directly used. For clutch modeling the slipping effects are considered in

the simulated model (Minh and Rashid [2012]). The synchronizers are simulated using a simple concept and does not contain the physical relations reviewed in the literature, however, without having a deep understanding of synchronizers mechanism it is not possible to model it. In order to consider the flexibility effects of the driveline the model suggested in (Eriksson and Nielsen [2014]) consisting of flexible propeller shaft and drive shaft is used. And finally, the energy management strategy is a combination of a rule-based controller and Hamiltonian minimization (Chasse and Sciarretta [2011]) which is discussed in related research section. The HEV is simulated by using the concepts which are discussed in the literature review and the suggested transmission system by Vicura. The clutch model is improved where detailed information such as dissipated energy, clutch couplings and duration of clutch operations can be calculated.

1.4 Expected results

The objective of the thesis is to reduce the emissions and fuel consumption of a vehicle by designing a new transmission model which enables efficient integration of the existing ICE and an EM. To have a suitable model for the conventional vehicle, the baseline model will be integrated with driveline blocks such as propeller shaft, drive shaft and final drive. The gear changing strategy and clutch actuation blocks will be added to the baseline model. The gearbox and clutch model of the baseline model will be changed to have the characteristics requested by Vicura. The completed conventional vehicle will be simulated for a given drive cycle, consequently, the emissions and fuel consumption for the vehicle will be calculated. Subsequently, the transmission blocks including clutch and gearbox will be simulated to be compatible with the electric motor. Afterwards, the synchronizer actuation and gear selection will be modeled based on SOC, vehicle speed and torque demand. After modeling and testing transmission and gear selection block it will be integrated by the conventional model.

Next, an electric motor will be added to the model together with an energy management strategy and the effect of hybridization will be inspected for both emissions and fuel consumption with same drive cycle as in conventional vehicle. The HEV is intended to have the same efficiency as the conventional vehicle in terms of driving torque, power and acceleration.

In other words, it can be said that the intention of the research is not to have a HEV model with better performance but same performance as the conventional vehicle and with less emissions and fuel consumption.

Methodology

In this chapter the details of the physics of parallel HEV will be discussed. The simulated models will be elaborated and the equations describing the behavior of each component will be explained. The model consists of driver model, HEV controller, ICE, ECU, clutch and gearbox, electric unit, driveline and vehicle dynamics. The EM and ICE are two sources of energy, where both components have been modeled in a separate sub model. The HEV controller manages the optimal usage of the energy between two sources of energy. A rule-based controller and an optimization method form a power management strategy for HEV controller. In driveline model, there are sub models including propeller shaft, final drive and drive shaft, which can capture the torsional vibrations of the driveline. In Figure 2.1, an overall structure of the HEV model with all consisting sub-models can be seen.

2.1 Vehicle Dynamics

By considering forces acting on vehicle, the vehicle actual speed (v) and wheel angular speed can be identified. Traction force ($F_t(t)$) generated by EM or ICE, aerodynamic forces ($F_a(t)$), rolling resistance ($F_r(t)$), gravity forces ($F_g(t)$), force dissipated ($F_d(t)$) by the brakes and inertia of the vehicle are the main forces which are affecting the vehicle dynamics.

The longitudinal dynamics of a vehicle can be described by (2.1) (Guzzella and Sciarretta [2007]).

$$m_v \frac{d}{dt} v(t) = F_t(t) - F_a(t) - F_r(t) - F_g(t) - F_d(t) \quad (2.1)$$

Where, m_v is the vehicle mass.

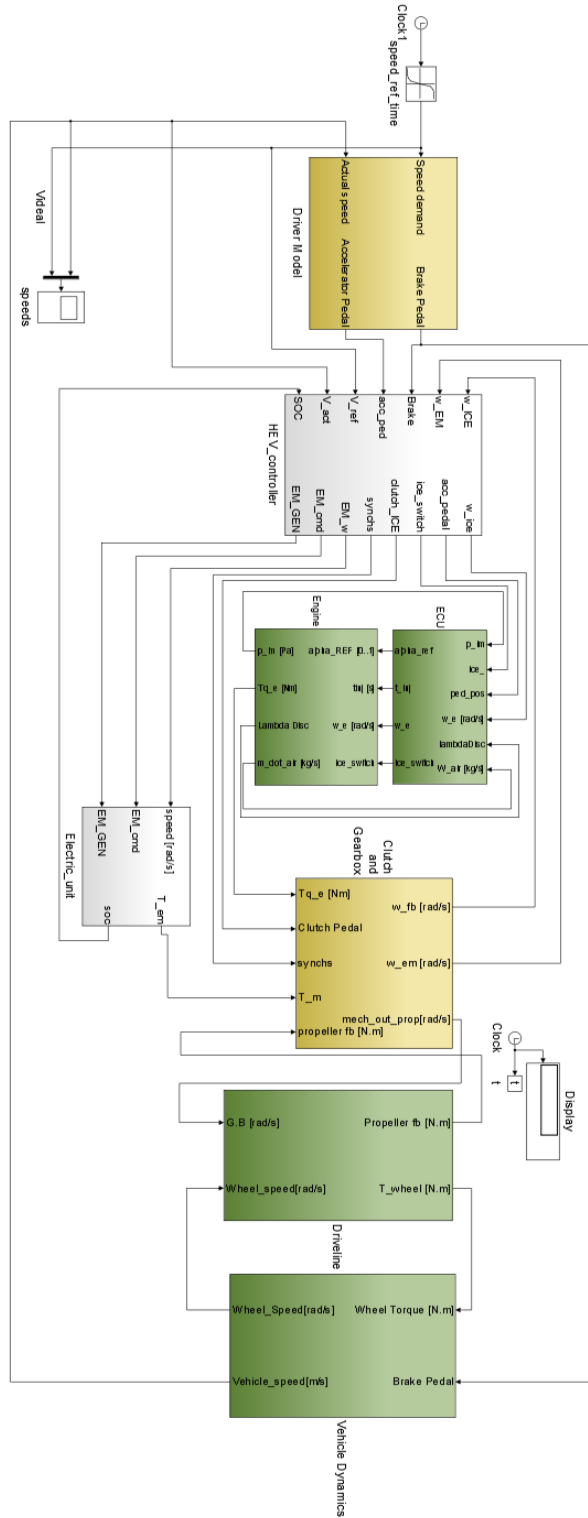


Figure 2.1: The parallel HEV model.

2.2 Transmission System

The transmission system was previously discussed in Chapter 1. Here, more details of the transmissions system will be explained.

In the suggested model for transmission system, 8 different cases may happen during the operation of the gearbox. In the following, the details of the transmissions system modes are discussed with figures, where red boxes represent the active components in each state and orange arrows show the direction of the transmission power.

In standstill mode, Figure 2.2, none of the components (clutch and synchronizers) are connected. The throttle opening and the voltage of the EM is zero.

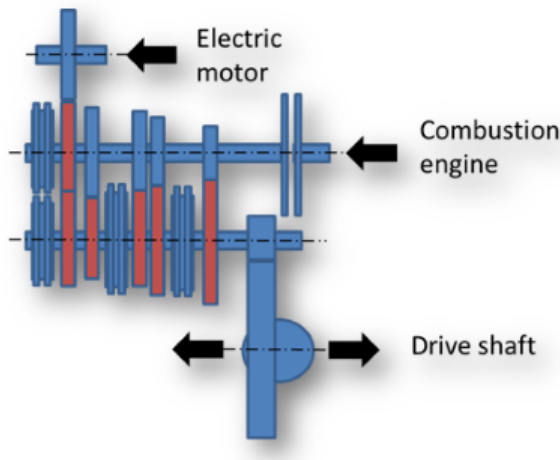


Figure 2.2: Standstill mode.

Figure 2.3 shows the standstill charging mode. In this case the ICE is charging the battery in standstill mode and ICE clutch together with EM synchronizer is connected.

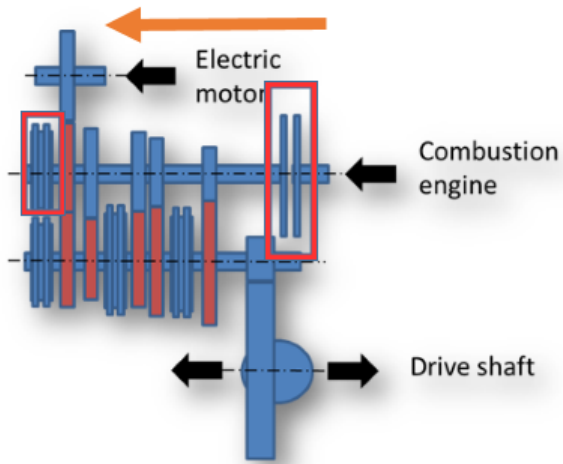


Figure 2.3: Standstill charging mode.

Figure 2.4 shows the ICE mode, where the ICE clutch is connected and ICE is providing torque to the vehicle.

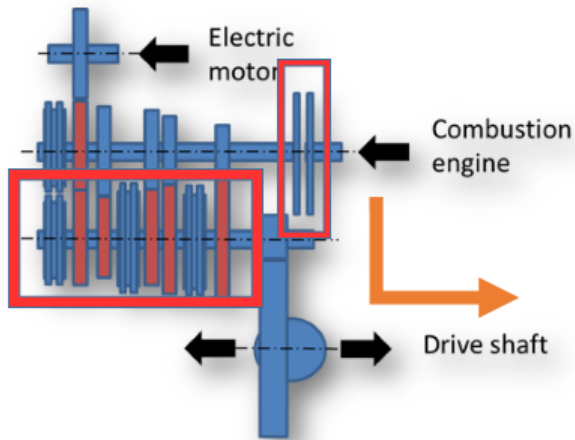


Figure 2.4: ICE mode.

Figure 2.5 shows the EM mode, where only EM is active and providing torque.

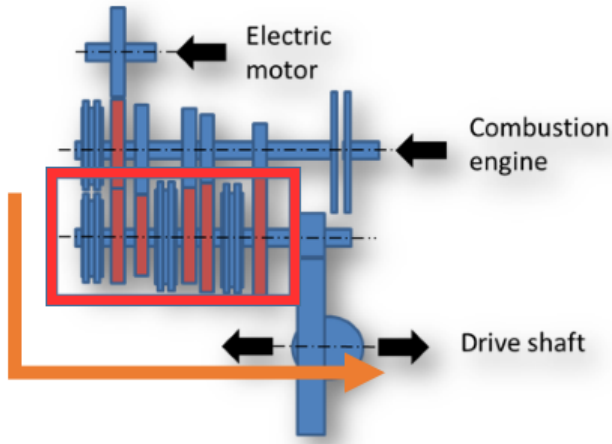


Figure 2.5: EM mode.

Figure 2.6 shows the hybrid mode, where both ICE and EM are providing torque to the vehicle. The ICE clutch is connected.

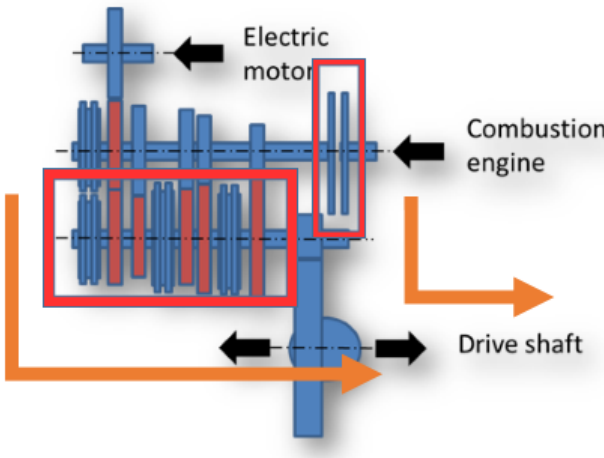


Figure 2.6: Hybrid mode.

Figure 2.7 shows the charging mode, where ICE is providing torque to propel the vehicle and also recharging the battery at the same time. In this case the EM works as a generator and EM synchronizer and ICE clutch are connected.

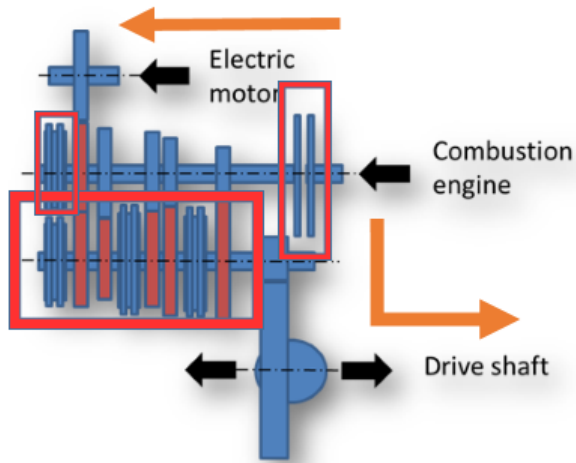


Figure 2.7: Charging mode.

Figure 2.8 shows the vehicle in regenerative braking mode, where the brake energy is stored in the battery. ICE and ICE clutch are disengaged and EM works as generator.

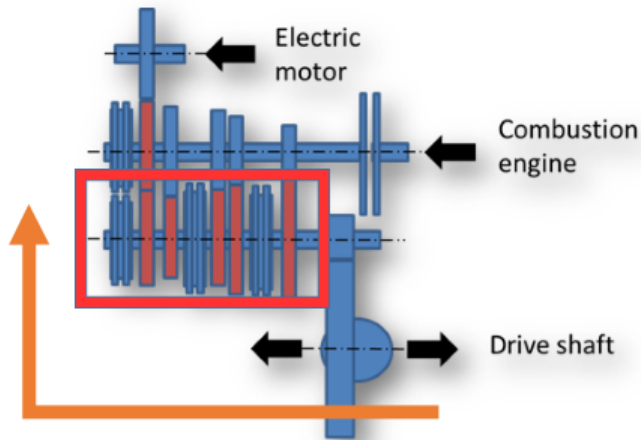


Figure 2.8: Regenerative braking mode.

Figure 2.9 shows the vehicle in coasting mode, where the all the energy on propeller shaft is stored in the battery. ICE and ICE clutch are disengaged and EM works as generator.

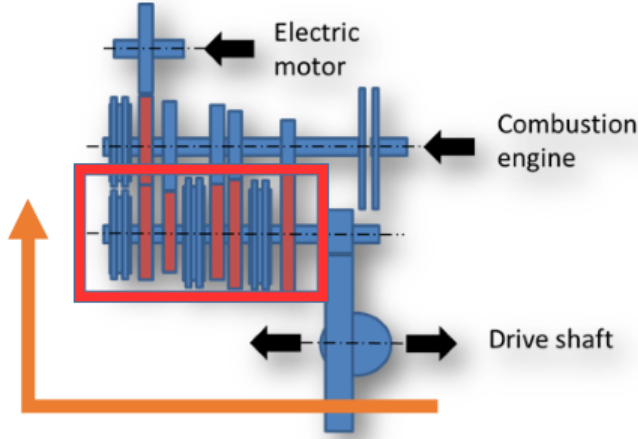


Figure 2.9: Coasting mode.

2.2.1 Gearbox

The gearbox model has 5 gears and each gear has a gear ratio and inertia. The gearbox is engaged or disengaged using synchronizers and the clutch. The synchronizers on the output shaft of the gearbox have a switching signal, where only one synchronizer can be active at a time. The rules of activating and deactivating synchronizers are mentioned in section 2.6.2. In appendix the Simulink implementation of the gearbox is shown.

The clutch model and inertial effects of the gearbox is also modeled in the illustrated appendix. The details of clutch model will be discussed in section 2.2.2.

To consider the inertial effects of the gearbox (2.2) has been implemented in gearbox model.

$$\omega_t = \int \frac{i_t T_t - b_t \omega_t - T_p}{J_t + i_t^2 J_{cl} + J_{gear} + J_p} dt \quad (2.2)$$

Where ω_t is the gearbox angular speed, i_t is the gear ratio, T_t is the gearbox torque, b_t is damping coefficient of the gearbox, T_p is the propeller shaft torque, J_t is the transmission inertia, J_{cl} is the clutch inertia, J_{gear} is the inertia of each gear and J_p is propeller shaft inertia.

2.2.2 Clutch and flywheel model

In this section the details of clutch model will be discussed. The parallel HEV is intended to be launched by EM, for this reason a downsized clutch will be used (smaller inertia and maximum friction torque).

In Table 2.1 the characteristics of the clutch for the conventional vehicle and HEV which has been used for the simulation has been mentioned.

Table 2.1: Clutch specifications

	Maximum friction torque [N.m]	Inertia [$kg.m^2$]
HEV	200	0.05
Conventional vehicle	500	0.1

The slipping effect has been considered in the clutch model; slipping will occur if the clutch is switched on and the speed difference between two plates of clutch is higher than 1 rad/s.

In the following the assumptions made to model the clutch will be discussed.

When the clutch is locked the overall inertia is:

$$J = J_{cl} + J_{ICE} \quad (2.3)$$

Where in this equation J_{ICE} is the ICE inertia.

When the clutch is unlocked or decoupled the overall inertia is equal to the inertia of the clutch:

$$J = J_{cl} \quad (2.4)$$

When the clutch is locked the angular speed of ICE (ω_{ICE}) is equal to the angular speed of the clutch (ω_{cl}).

$$\omega_{cl} = \omega_{ICE} \quad (2.5)$$

Otherwise the clutch angular speed will follow (2.6):

$$\omega_{cl} = \int \frac{T_{ICE} - T_{cl}}{J_{ICE}} dt \quad (2.6)$$

Where, T_{ICE} and T_{cl} are the torques by ICE and the clutch. Depending on clutch position (locked, unlocked or slipping), the transferred torque can be different. When the clutch is unlocked the transferred torque is zero and while the clutch is locked clutch delivers all the torque produced by ICE:

$$T_{cl} = T_{ICE} \quad (2.7)$$

When the clutch is slipping, the torque is calculated as follows:

$$T_{cl} = \text{sgn}(\omega_{ICE} - \omega_t) \times T_{f,max} \quad (2.8)$$

Where, $T_{f,max}$ is the maximum friction torque which can be transferred by the clutch.

2.2.2.1 Dissipated Energy by Clutch

To calculate the dissipated energy by the clutch first the dissipated power (P_{cl}) by the clutch should be calculated:

$$P_{cl} = T_{cl} \cdot \Delta\omega_{cl} \quad (2.9)$$

The total dissipated energy (E_{cl}) can be calculated using (2.10).

$$E_{cl} = \int P_{cl} dt \quad (2.10)$$

2.3 Driveline

The driveline of the vehicle consists of propeller shaft, final drive and drive shaft. In Sections 2.3.1, 2.3.2 and 2.3.3 the details of the driveline model will be discussed. The flexibility of propeller shaft and drive shaft have been considered in calculations. In final drive model, the inertial effects from final drive have been modeled and integrated with the driveline model. The spring and damping coefficients of the propeller shaft and drive shafts are specified in Table 2.2

Table 2.2: Specifications of propeller shaft and drive shaft

	Damping coefficient [N.m.s/rad]	Spring coefficient [N.m/rad]
Propeller shaft	15	5000
Drive shaft	1	1000

2.3.1 Propeller shaft

To capture the torsional vibrations of the propeller shaft and oscillations resulting from that, flexibility of the propeller shaft will be modeled in this section. (2.11) describes the torsional vibrations of the propeller shaft using a linear first order differential equation.

$$T_p = c_p(\omega_t - \omega_f) + k_p(\theta_t - \theta_f) \quad (2.11)$$

Where k_p and c_p are the spring and damping coefficients of the propeller shaft.

2.3.2 Final drive

The inertial effects and damping of the final drive has been modeled in driveline model. The governing equations regarding the behavior of final drive are as below:

$$\omega_f = \int \frac{T_p i_f - b_f \omega_d}{J_f + i_f^2 J_p} dt \quad (2.12)$$

Where ω_f is the angular speed of final drive, i_f is the final drive ratio, b_f is the damping coefficient of final drive and J_f is the final drive inertia.

2.3.3 Drive shaft

Similar to the propeller shaft the torsional vibrations of drive shaft can be described by (2.13):

$$T_d = c_d(\omega_f - \omega_w) + k_d(\theta_f - \theta_w) \quad (2.13)$$

Where k_d and c_d are the spring and damping coefficients of the drive shaft.

2.4 EM and Battery

The EM model consists of an inductance and a resistance. The input signals to EM block are, rotational speed and voltage and output signals are torque available by EM and current. The governing equations to formulate the EM model are as below ((2.14) and (2.15)):

$$T_{em} = i_{em} \cdot k_t \quad (2.14)$$

Where in (2.14), T_{em} is the EM torque, i_{em} is the EM current and k_t is the torque constant of EM.

$$L_{em} \frac{d}{dt} i_{em} = U_{em} - R_{em} i_{em} - \omega_{em} k_{em} \quad (2.15)$$

In (2.15), L_{em} is the inductance of EM, U_{em} is the voltage of EM, R_{em} is the resistance of EM, ω_{em} is the angular speed of EM and k_{em} is the back EMF constant of EM.

The battery is defined using (2.16) and (2.17). Using (2.16) the SOC can be calculated at each moment.

$$SOC = SOC_{init} + \frac{1}{Q_{max}} \cdot \int i(t) dt \quad (2.16)$$

Where SOC_{init} is the initial SOC and Q_{max} is the maximum capacity of the battery.

The battery characteristic has also been applied to the formulation. The voltage of the battery changes by SOC. $f(SOC)$ is the battery characteristic and indicates the dependency of available voltage to SOC.

$$U_{em} = U_{max} \cdot EM_{cmd} \cdot f(SOC) \quad (2.17)$$

Where U_{max} is the maximum available voltage by battery and EM_{cmd} is the signal from PI controller. The PI controllers for EM and ICE are explained in section 2.5.

2.5 Driver Model

The driver model consists of a PI controller, where reference signal is reference speed and feedback signal is actual speed. The PI controller decides to either use the accelerator pedal (throttle position) or brake pedal. Similarly the EM command (to increase or decrease voltage) is controlled by a PI controller, which is based on actual speed and reference speed. The Simulink implementation of PI controller for ICE and EM are shown in the appendix.

2.6 HEV controller

The HEV controller block enables the user to run the vehicle in three different cases including pure electric mode, pure ICE mode or hybrid mode. The HEV controller block, considers the situation of the vehicle and all the components engaged, and decides the best solution to run the vehicle. The HEV controller gets some input signals and feedback signals and based on the value of signals decides output signals. The input signals to HEV controller are:

- ICE speed
- EM speed
- Brake position
- Acceleration pedal position
- Reference speed
- Actual speed of the vehicle
- SOC

And the output signals are:

- ICE speed
- EM speed
- ICE switch on or off
- clutch activation and deactivation
- Throttle opening
- Voltage of EM
- EM on or off or working as generator
- Synchronizers activation or deactivation

ICE and EM speeds are both as input and output signals. This is because the controller sets the speed to 0 while ICE or EM is switched off. In all the other cases the input signal and output signal (speed of ICE or EM) are equal. In the following sub sections the details of HEV block will be elaborated.

2.6.1 Energy management

The energy management of the parallel HEV includes a rule based controller and an optimization method (equivalent mass minimization). The rule based controller decides what action to take at each moment of the drive cycle, and equivalent mass minimization is to optimize the fuel consumption while the vehicle is working in hybrid mode.

2.6.1.1 Rule Based Controller

The rule based controller is some sets of rules which decides what action to take at each moment of the drive cycle. The controller makes the decision based on reference speed, SOC, brake pedal position, acceleration pedal position, and approximate required torque. The output of the rule based controller is a number, which is representative of a state to run the vehicle. In Table 2.3 the role of each state has been specified. The Simulink implementation of the rule based controller is illustrated in appendix.

Table 2.3: States and Specifications of Each State

State	Role	Clutch	Throttle opening	EM voltage	EM	ICE
1	Standstill	off	0	0	0	0
2	Standstill charging	on	> 0	> 0	-1	1
3	ICE mode	on	> 0	0	0	1
4	Electric mode	off	0	> 0	1	0
5	Hybrid mode	on	> 0	> 0	1	1
6	Charging mode	on	> 0	> 0	-1	1
7	Braking mode	off	0	> 0	-1	0
8	Coasting mode	off	0	> 0	-1	0

In Table 2.3 it is specified that, which component is active at each state. In EM column "1" is representative of a switched on EM, "0" as switched off and "-1" as generator. Similarly in ICE column, "0" is representative of a switched off ICE and "1" is as switched on ICE.

In electric mode the only source of energy is EM and in ICE mode ICE is the only source of energy.

In Hybrid mode both ICE and EM are providing torque to the vehicle with respect to an optimization method (Optimization method will be discussed in section 2.6.1.2).

While the vehicle is braking or coasting the energy on the propeller shaft will be transferred to the generator to be stored in the battery.

In charging mode all the torque to propel the vehicle is provided by ICE and the battery will be recharged with a constant torque of 20 N.m/s.

The rule based controller is some set of if conditions:

- When the vehicle has a speed of $V = 0$ and $SOC > 0.3$ it is in standstill mode
- When the vehicle has a speed of $V = 0$, and $SOC < 0.3$ the standstill charging mode is active. The ICE will start charging Battery until a $SOC > 0.5$ is achieved.
- By increasing the vehicle speed ($V > 0$), EM will start working. The vehicle always will be launched by EM.
- If $SOC > 0.5$ and $V < 130$ km/h, the vehicle will remain in EM mode.

- If $0.3 < SOC < 0.5$ and $V < 130$ km/h, the vehicle will switch to hybrid mode.
- If the brake pedal is activated ($B > 0$) the vehicle will switch to regenerative braking mode.
- If $SOC < 0.3$ the charging mode will be activated and will remain in this mode until a $SOC > 0.5$ is achieved.
- If $V > 130$ km/h only ICE will work and EM will be switched off.
- If $V > 0$ and both accelerator pedal and brake pedal are zero coasting mode activates.

2.6.1.2 Equivalent mass minimization

In order to optimize fuel consumption, an optimization method has been used. Hamiltonian minimization or equivalent mass minimization has been used to make an optimization while both EM and ICE are providing torque. For this purpose throttle opening and voltage signal both are interpreted as power. The Hamiltonian equivalent (H) of the electrochemical power (P_{ech}) and fuel power (P_f) is:

$$H = P_f + \lambda \cdot P_{ech} \quad (2.18)$$

$$\frac{\partial H}{\partial SOC} = \frac{\partial P_f}{\partial SOC} + \frac{\partial P_{ech}}{\partial SOC} \cdot \lambda + \dot{\lambda} \cdot P_{ech} = 0 \quad (2.19)$$

After finding the H value, using (2.19) the Hamiltonian equivalent can be minimized. Fuel power and electrochemical power does not change with SOC, therefore in order to satisfy (2.19), the $\dot{\lambda}$ value should be zero.

$$\dot{\lambda} = 0 \quad (2.20)$$

According to (2.20), λ is constant where it is calculated iteratively for the drive cycles. In appendix the Simulink implementation of the optimization algorithm has been shown.

2.6.2 Synchronizer and clutch actuation

As described in Chapter 1 the gearbox has four synchronizers; where synchronizer 1 connects EM to the gearbox input shaft and the other synchronizers connect input shaft to the output shaft. The synchronizer of the EM is activated if the vehicle is in hybrid mode and the battery needs to be recharged by ICE. The battery is charged by ICE in standstill charging mode and ICE charging mode. Other three synchronizers are activated and deactivated with respect to the actual and reference speed of the vehicle. Each synchronizer corresponds to a specific gear. Table 2.4 specifies which synchronizer corresponds to which gear. Synchronizers 3 and 4 can move to left and right to select the desired gear. In the word "Synch4R" Synch means synchronizer and 4 is the number of synchronizer and R means synchronizer on the right side; this also applies to the other synchronizers.

The activation and deactivation of each gear is controlled by speed control. Each gear has an up-shift speed and down-shift speed value. If the actual speed increases beyond up-shift speed the gear gets activated and the gear will not change until the actual speed drops below down-shift speed. In Table 2.5 the up-shift and down-shift value of each gear is mentioned.

Table 2.4: Synchronizers and it's corresponding gear

Synchronizer	Gear
Synch4R	1
Synch4L	2
Synch3R	3
Synch3L	4
Synch2	5

Table 2.5: Up-shift and down-shift speed of each gear

Gear	Up-shift speed [km/h]	Down-shift speed [km/h]
1	eps	0
2	10	2
3	40	25
4	70	50
5	100	80

In appendix the Simulink implementation of synchronizers control is illustrated.

2.7 Fuel Consumption and Emissions

By integrating the injected fuel ($\dot{m}_{injected}$) over the cycle duration the total fuel consumption ($M_{consumed}$) can be calculated:

$$M_{consumed} = \int \dot{m}_{injected} dt \quad (2.21)$$

The total fuel mass should then be divided by total traveled distance ($X_{traveled}$) and density of fuel (ρ) to calculate the liter fuel consumed per traveled distance (L_{100Km}):

$$L_{100Km} = M_{consumed} \cdot /(\rho \cdot X_{traveled}) * 100 \quad (2.22)$$

The emissions can be calculated considering, the mass flow rate of air (\dot{m}_a) and fuel (\dot{m}_f), traveled distance and air/fuel equivalence ratio (λ_d), where λ_d is

calculated using (2.23) and (2.24).

$$\lambda_d = \frac{(A/F)}{(A/F)_s} \quad (2.23)$$

In (2.23), $(A/F)_s$ is stoichiometric air/fuel ratio.

$$(A/F) = \frac{\dot{m}_a}{\dot{m}_f} \quad (2.24)$$

The amount of produced emissions are calculated using a stoichiometric relation for the fuel. The emissions include CO, HC and NO_x and the emission model is appropriate for a conventional vehicle. However, in this project it is also used for the HEV model. For a HEV, other criteria can be applied in order to accurately calculate the emissions.

In the Appendix, the details of calculations for emissions and fuel consumption can be found.

In this section the important results of the simulated model will be discussed. For testing the vehicle two drive cycles namely EUDC and FTP75 has been selected, where EUDC is a highway drive cycle and FTP75 is an urban drive cycle. Each drive cycle has been tested in two different conditions:

- ICE mode
- Hybrid mode

In ICE mode the EM is supposed to be switched off and not working. In hybrid mode the vehicle is running with respect to the rule base algorithm which was discussed previously. In order to have a charge sustaining strategy after running the drive cycle a proper λ (the constant in (2.18)) value has been calculated for each drive cycle.

The important outcomes of the simulation which will be discussed here are SOC, torques, torsional vibrations caused by propeller shaft and drive shaft, ICE clutch operation, fuel consumption, emissions and the calculation time for the drive cycles.

The results are shown in terms of figures and tables. Each figure has two sub-figures, where the first (a) represents the calculated results in hybrid mode and second (b) represent the results in ICE mode. In all test cases for simulated model the actual speed can follow the reference speed and this is an illustration of good performance of the Simulink model.

3.1 State of Charge

In this section the SOC for EUDC and FTP75 drive cycles have been calculated and plotted in Figures 3.1 and 3.2. As it is illustrated in Figures 3.1a and 3.2a the battery is charge sustaining in hybrid mode(initial and final SOC is equal to 0.5 in all simulations).

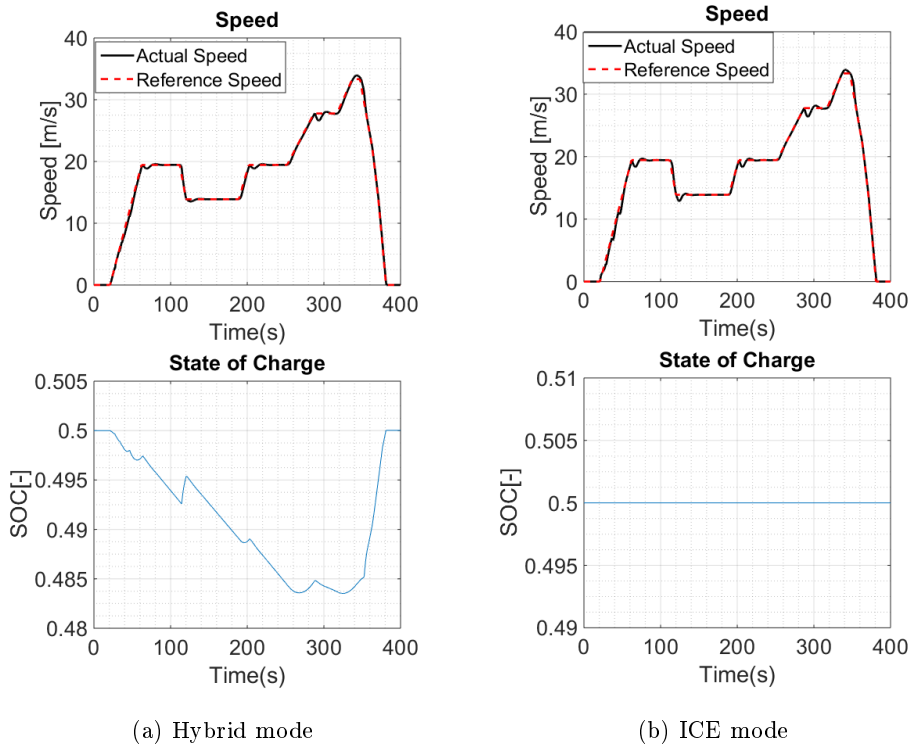


Figure 3.1: State of Charge for EUDC.

The battery is recharged when the vehicle is braking or coasting. When the vehicle is in standstill mode the battery is unused and while accelerating it gets discharged.

Figures 3.1b and 3.2b show the SOC while the vehicle is running in ICE mode. The SOC is constant in both figures.

FTP75 drive cycle has too many accelerations and decelerations, but EUDC drive cycle has limited decelerations. For this reason in FTP75 the battery is charged more frequently.

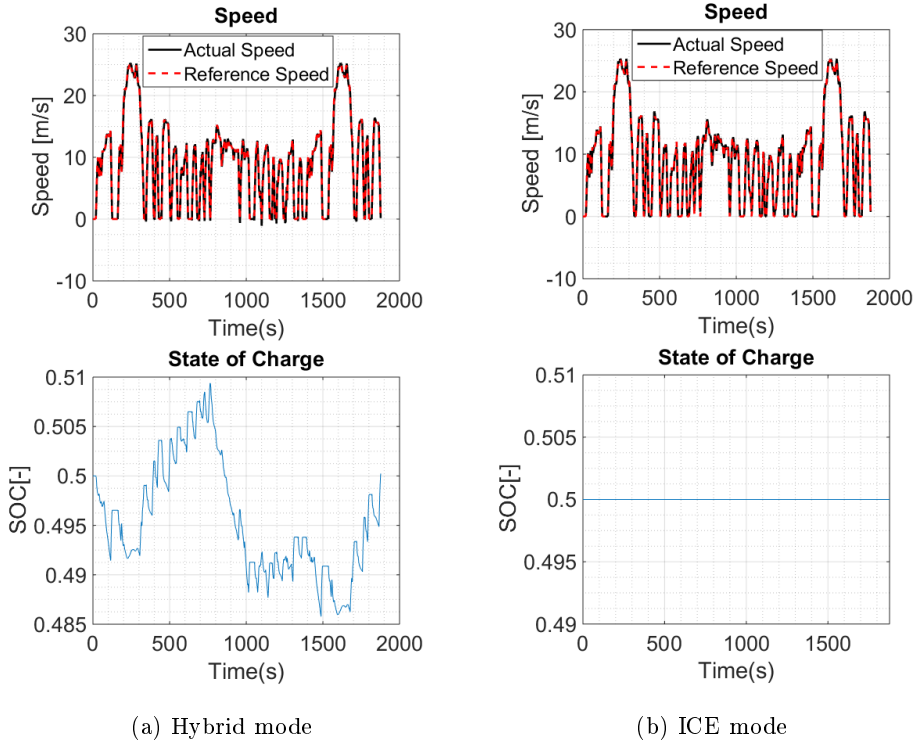


Figure 3.2: State of Charge for FTP75.

3.2 Torques

In this section the torque which can be delivered by EM or ICE are calculated and plotted in Figures 3.3 and 3.4. Figure 3.3 shows the available torque from EM and ICE for EUDC drive cycle and Figure 3.4 shows the available torques for FTP75 drive cycle.

By comparing Figure 3.3a and Figures 3.3b it can be seen that in ICE mode the EM is switched off and all the torque is provided by ICE. Also it can be seen that, in hybrid mode the torque is provided by both ICE and EM.

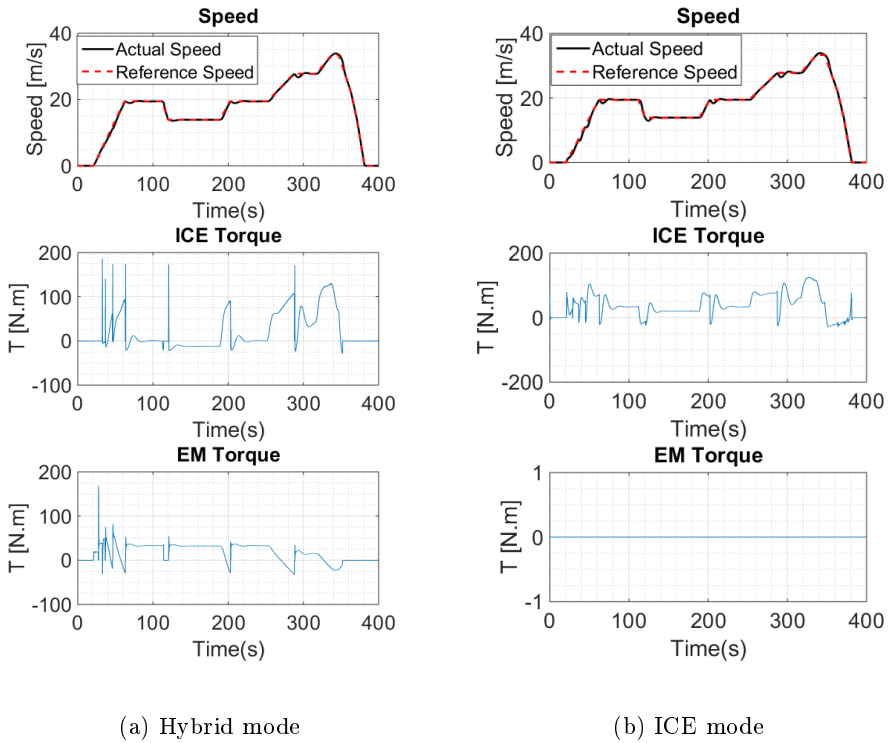


Figure 3.3: Available torque from EM and ICE for EUDC.

Also in Figure 3.4b it can be seen that all the torque is provided by ICE and EM is switched off and Figure 3.4a shows that in hybrid mode the torque is provided by EM and ICE. In Figure 3.4a it can be seen that the ICE is switched off for a period of time; as shown in Figure 3.2a the SOC is greater than 0.5 for this period of time. The rule based algorithm decides to run the vehicle in EM mode (ICE switched off) while the $SOC > 0.5$, therefore, the ICE is disengaged.

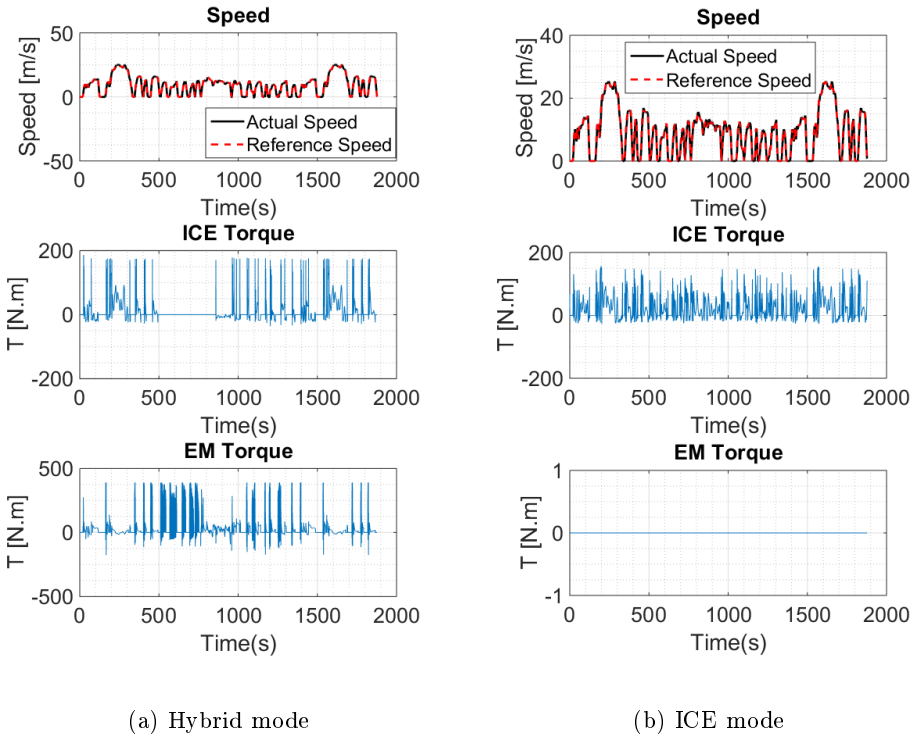


Figure 3.4: Available torque from EM and ICE for FTP75.

3.3 Torsional Vibrations

In this section the torsional vibrations caused by propeller shaft and drive shaft has been calculated and plotted. The torsional vibrations are due to the damping and spring coefficients of propeller shaft and drive shaft.

3.3.1 Propeller Shaft

In this section the torsional vibrations caused by propeller shaft will be discussed. In Figure 3.5 the oscillations of gearbox and propeller shaft are shown.

As shown in Figures 3.5a and 3.5b for both hybrid and ICE mode there exist some oscillations while shifting the gear. In Figure 3.5b it is shown that while decelerating the propeller shaft oscillates by gear shifting but in hybrid mode, there is no oscillation. This is because, in hybrid mode the ICE clutch is disengaged while decelerating. The speed difference between gearbox and propeller shaft is obvious which is a result of propeller shaft flexibility.

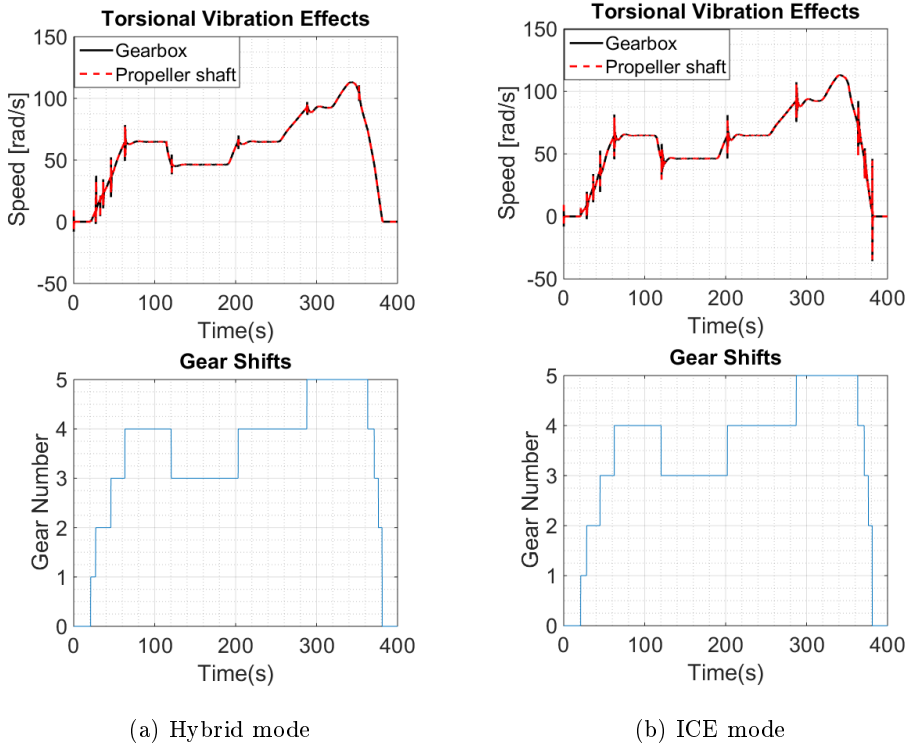


Figure 3.5: Torsional vibrations caused by propeller shaft for EUDC.

As shown in Figure 3.6 the oscillations are harsher in FTP75 drive cycle. because of repeating accelerations and decelerations and too many gear shifting.

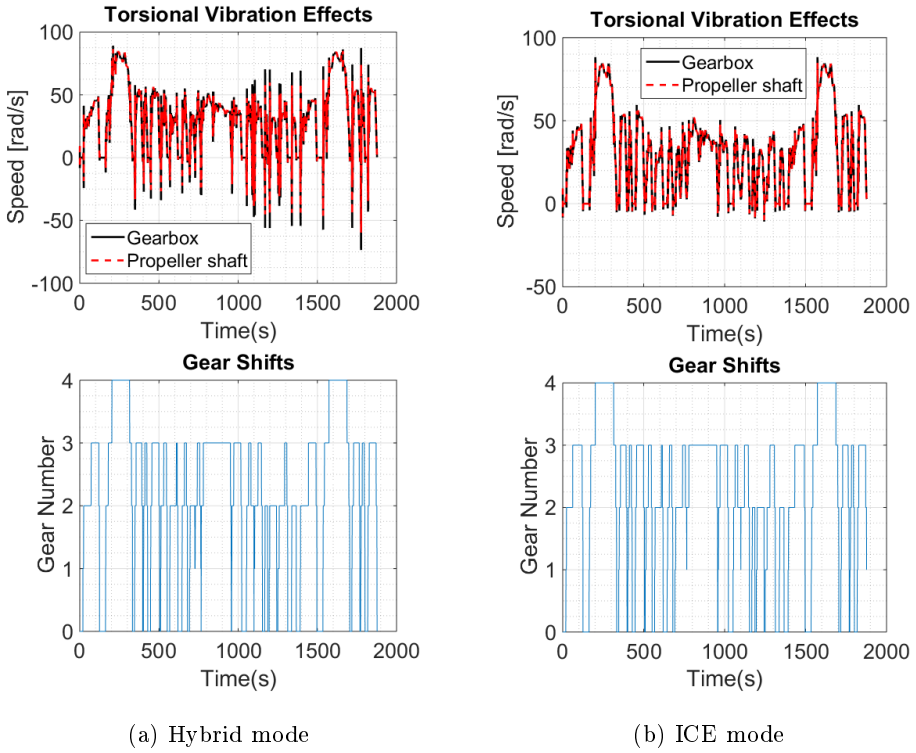


Figure 3.6: Torsional vibrations caused by propeller shaft for FTP75.

3.3.2 Drive Shaft

In section 3.3.1 the oscillations caused by propeller shaft was discussed. Similarly, in this section the vibrations resulted from drive shaft will be discussed.

In Figures 3.7 and 3.8 the oscillations caused by drive shaft flexibility has been calculated and plotted. The oscillations between propeller shaft and drive shaft is more obvious compared to oscillations from gearbox and propeller shaft. The reason is because the drive shaft is closer to the wheels compared to the propeller shaft, furthermore, the drive shaft has a smaller damping coefficient and stiffness, while the propeller shaft is stiffer.

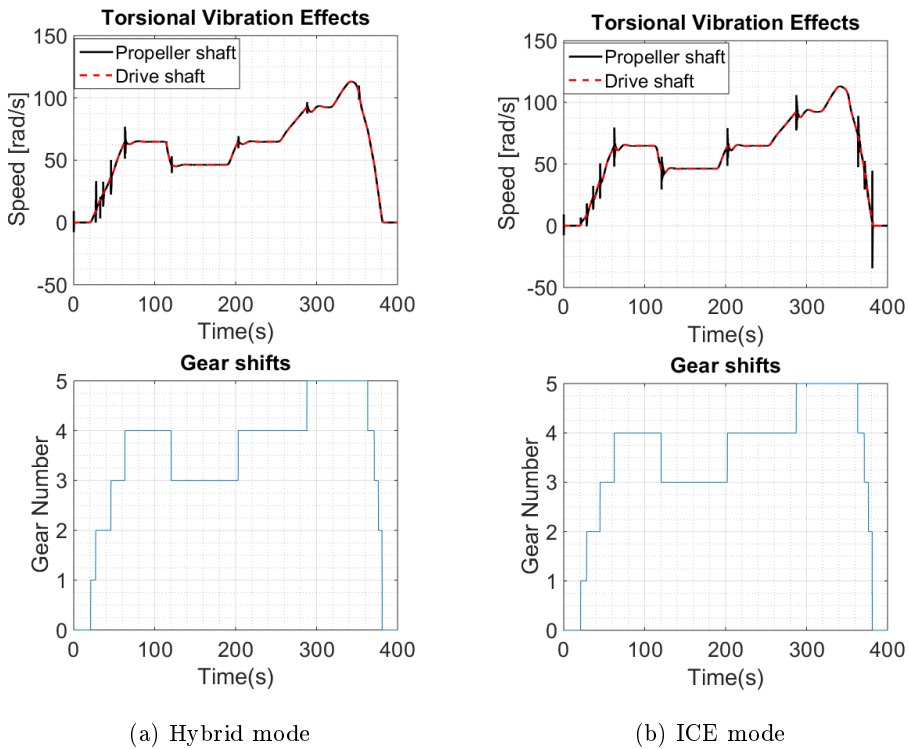


Figure 3.7: Torsional vibrations caused by drive shaft for EUDC.

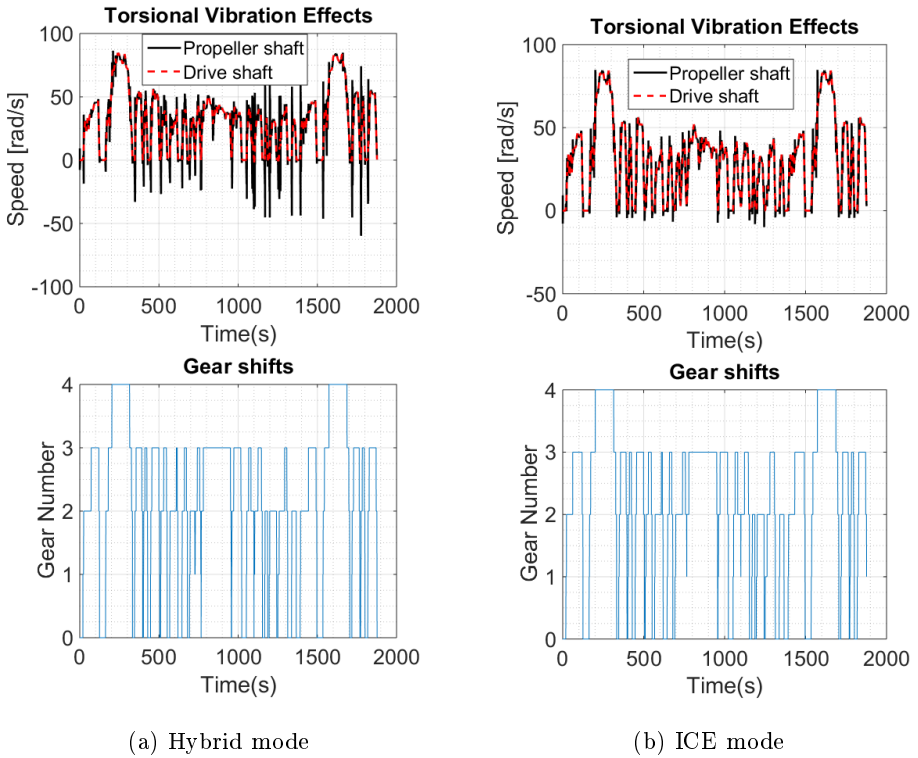


Figure 3.8: Torsional vibrations caused by drive shaft for FTP75.

3.4 Clutch Operation

The clutch operation in the drive cycles for hybrid mode and ICE mode will be discussed in this section. The clutch gets activated and deactivated with respect to the rule based algorithm. The operation of clutch for the EUDC and FTP75 drive cycles have been calculated and plotted in Figures 3.9 and 3.10. When the clutch is open it is deactivated and does not transfer the torque produced by the ICE, on the other hand, when the clutch is closed it is activated and transfers all the torque produced by the ICE. The transient state between open and closed clutch is slipping clutch.

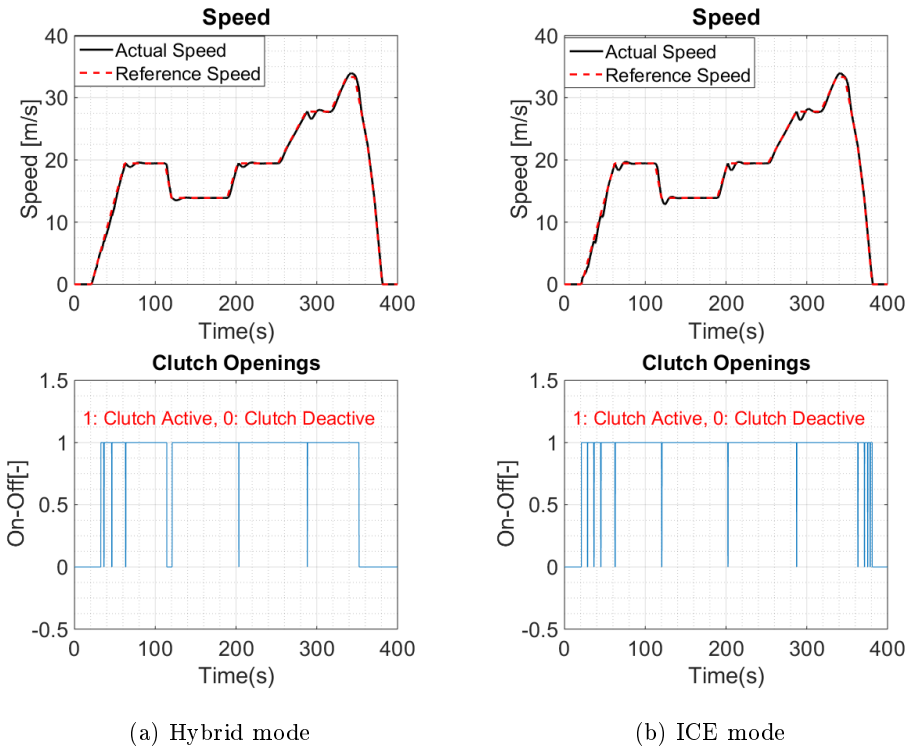


Figure 3.9: Clutch openings in EUDC (Activating and deactivating).

The FTP75 drive cycle is an urban drive cycle and there are many decelerations in this drive cycle and the battery is charged more frequently. For this reason for a long period of time the SOC is greater than 0.5. As it was defined in rule based conditions, while the SOC is greater than 0.5, the ICE and ICE clutch are switched off. Resultantly, the FTP75 drive cycle has much less openings while it is in hybrid mode (HEV mode). On the other side, the EUDC is a highway drive cycle, where it has limited decelerations, therefore the clutch has more operating

time. In EUDC, the vehicle is running mostly in hybrid mode, but in FTP75 drive cycle the vehicle is running mostly in EM mode (ICE is switched off).

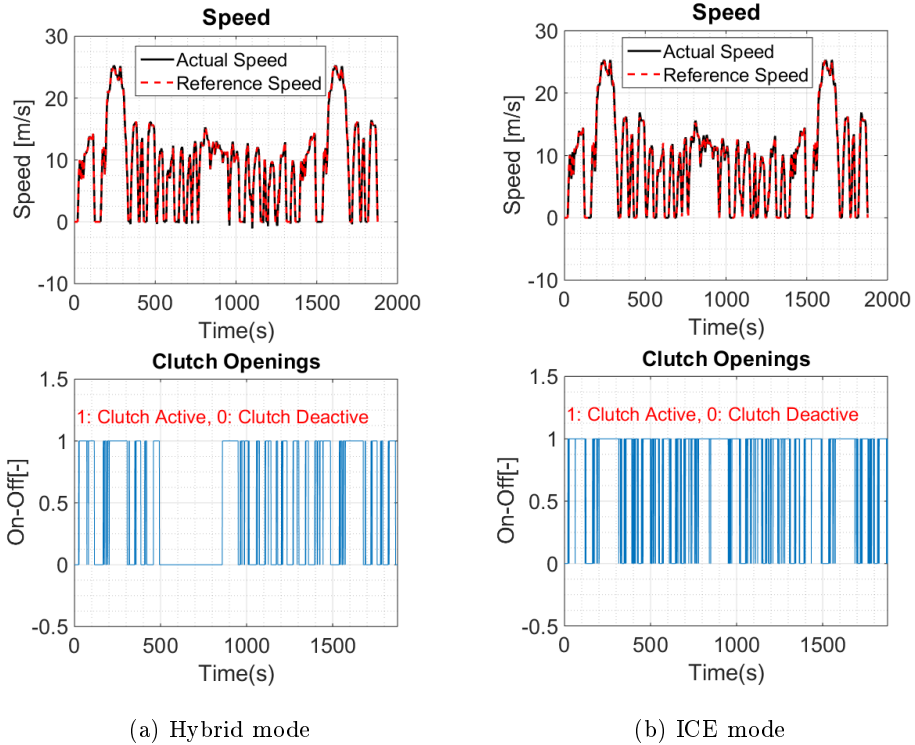


Figure 3.10: Clutch openings in FTP75 (activating and deactivating).

In Table 3.1 the percentage of clutch operating time to the total drive cycle duration has been calculated and summarized.

Table 3.1: Clutch openings [% of cycle duration]

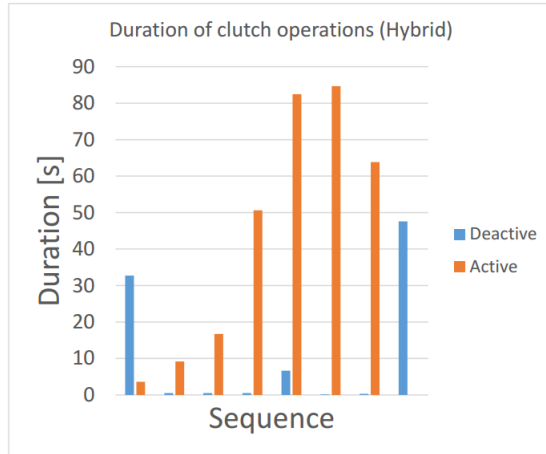
	Hybrid mode	ICE mode
EUDC	77.76	88.62
FTP75	47.71	78.95

Table 3.2 shows the number of clutch couplings during drive cycles in both hybrid and ICE mode. The number of clutch couplings are less in hybrid mode. As mentioned in rule based controller, in many cases the ICE is not used or used together with EM, therefore there is less clutch usage in hybrid mode..

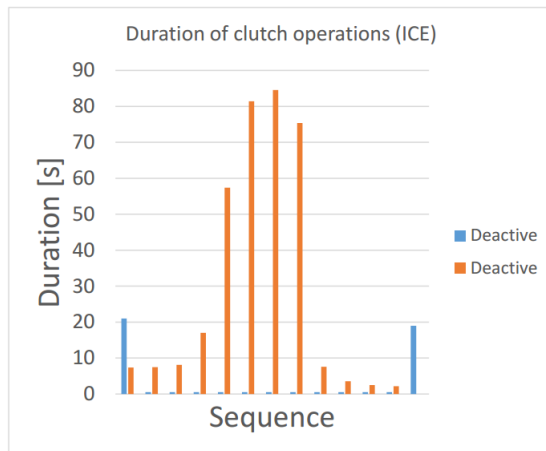
Table 3.2: Number of clutch couplings

	Hybrid mode	ICE mode
EUDC	7	14
FTP75	74	136

In the Figures 3.11 and 3.12, the coupled and decoupled clutch duration is shown; the blue bars show the clutch when it is decoupled and orange show the clutch when it is coupled. The blue bars and orange bars happen with sequence. Figures 3.11a and 3.12a show the drive cycles in hybrid mode and Figures 3.11b and 3.12b show the drive cycles in ICE mode. The duration of decouplings are clearly longer in hybrid cases, specially when the vehicle is launched. By comparing Figures 3.11a and 3.11b it can be seen that first decoupling for hybrid case is 33 seconds but for the ICE mode is 20 seconds. Also after the last deceleration, the clutch in hybrid mode is decoupled for 47 seconds and for ICE mode 19 seconds. The controller is designed in such a way that the vehicle is launched and stopped without using the clutch.

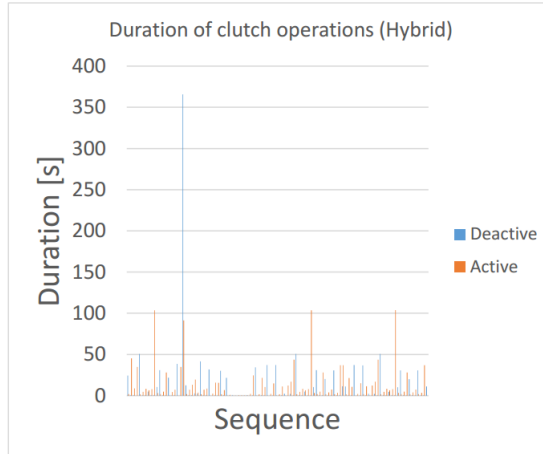


(a) Hybrid mode

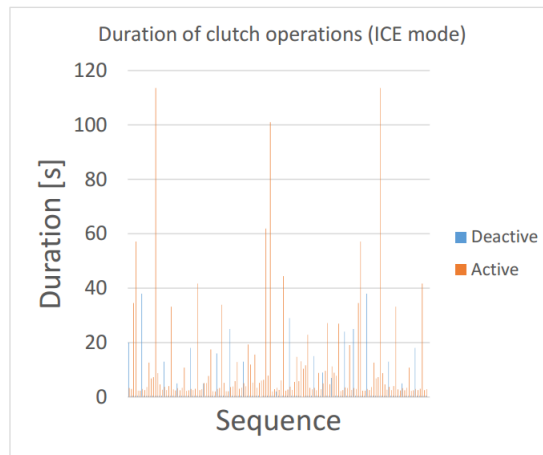


(b) ICE mode

Figure 3.11: Duration of clutch operations in EUDC (activating and deactivating).



(a) Hybrid mode



(b) ICE mode

Figure 3.12: Duration of clutch operations in FTP75 (activating and deactivating).

3.4.1 Dissipated Energy by Clutch

In section 2.2.2.1, the details regarding calculations of dissipated energy is discussed. The dissipated energy is calculated by integrating dissipated power during the drive cycle.

In Figures 3.13 and 3.14 the dissipated power during the drive cycles for hybrid and ICE modes have been plotted. In Table 3.3 the total dissipated energy by the clutch during two drive cycles are summarized. As shown in the table the dissipated energy by the clutch in hybrid mode is less for both drive cycles.

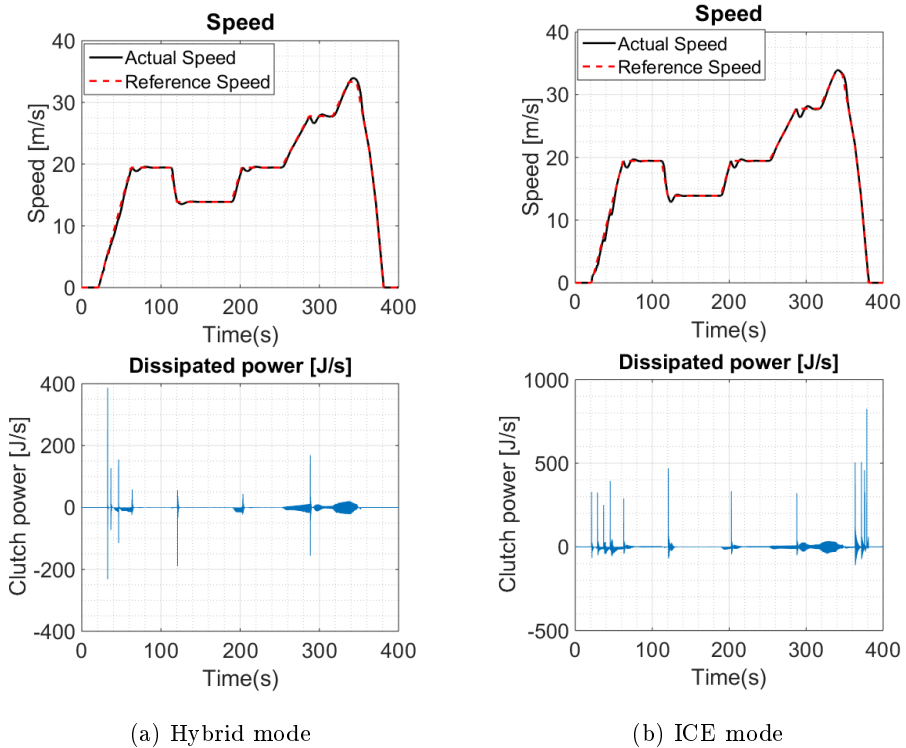


Figure 3.13: Dissipated power in EUDC.

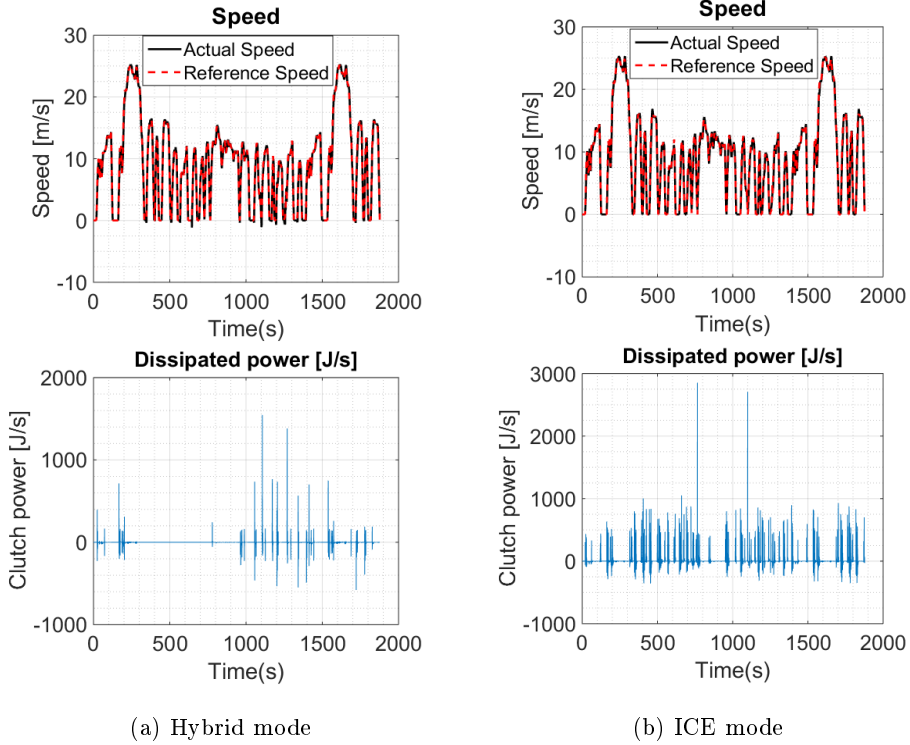


Figure 3.14: Dissipated power in FTP75.

Table 3.3: Dissipated energy [J]

	Hybrid mode	ICE mode
EUDC	-384.6661	-558.1863
FTP75	-1.9765e+03	-8.7659e+03

3.5 Fuel Consumption and Emissions

In this section, the fuel consumption and emissions of the vehicle for EUDC and FTP75 drive cycles have been calculated. As shown in Table 3.4 for both drive cycles the fuel consumption has decreased in hybrid mode. In FTP75 drive cycle, because of too many braking occasions, the battery is charged more frequently; therefore, the fuel consumption has decreased more in this drive cycle.

Table 3.4: Fuel Consumptions

	Hybrid mode [L/100km]	ICE mode [L/100km]
EUDC	5.27	6.61
FTP75	4.73	8.12

In Tables 3.5 and 3.6, the emissions for two drive cycles have been calculated. The emissions for both drive cycles have been reduced for parallel HEV.

Table 3.5: Emissions of EUDC

	Hybrid mode [g/km]	ICE mode [g/km]
CO	0.04	0.37
HC	0.05	0.09
NOx	0.01	0.07

Table 3.6: Emissions of FTP75

	Hybrid mode [g/km]	ICE mode [g/km]
CO	0.1	0.22
HC	0.05	0.09
NOx	0.02	0.04

3.6 Calculation Time

The calculation time for the tested drive cycles have been determined and summarized in Table 3.7. The cycle duration is also shown in the table. By comparing the real time value and calculation time it can be concluded that the simulated model is approximately 10 times faster than real time.

Table 3.7: Calculation Time

	Drive Cycle Duration [s]	Calculation Time [s]
FTP75	1877	210
EUDC	400	31

Conclusions

In this chapter the main conclusions resulted from the thesis are discussed. In the simulated model, different drive cycles can be uploaded to the model and detailed physics of the system can be studied, moreover, it is possible to calculate the emissions and fuel consumption of the vehicle in hybrid mode and also conventional mode. The model is capable of running the vehicle in pure electric mode, hybrid mode or ICE mode. The status of battery, EM, ICE, gearbox, synchronizers, clutch, propeller shaft, final drive and drive shaft can be studied at each moment. In hybrid mode, the Hamiltonian minimization can be used to optimize the fuel consumption (optimal usage of electrochemical and fuel power (section 2.6.1.2)). The model is also flexible to changes in all parts, since the HEV problem is broken down to small parts, and each part of the vehicle is defined in a different subsystem. The goals which were aimed to be fulfilled in section 1.1 and 1.4 are achieved and satisfied.

After having a complete model for the current purpose, it is tested for two different drive cycles where one is a highway drive cycle and the other is urban drive cycle with too many accelerations and decelerations. The following conclusions can be made according to the simulation results::

- The actual speed can follow the reference speed of the drive cycle in both hybrid mode and ICE mode.
- The parallel HEV has less fuel consumption and less emissions compared to a conventional vehicle.
- The battery is charge sustaining by defining an optimal λ value where λ is calculated iteratively.
- For FTP75 drive cycle the reduction in fuel consumption is more obvious since there are many decelerations in this drive cycle.
- The new transmission system is compatible with the new hybrid configuration.

- The clutch used for this simulation is downsized because the vehicle is launched by EM, and found to be working compatible with the new gear-box design.
- The driveline model has torsional vibration effects of the propeller shaft and drive shaft; the oscillations due to flexibility of propeller shaft and drive shaft are illustrated.
- The suggested model can respond to the disturbances caused by the gear shifts or the changes in the transmission system conditions.

Besides the main conclusions and strong points of the thesis, the following possible improvements are suggested for to the future models:

- A more detailed synchronizer model.
- A better PI controller to prevent spikes in ICE and EM torque and tuning the PI controller to reduce torsional vibrations in hybrid mode.
- A model to update λ value to have charge sustaining model (For current model λ should be adjusted manually).
- A model which considers the efficiency map of the EM and efficiency of the battery.
- Since the suggested emission model is suitable for a conventional vehicle, a more detailed model to calculate the emissions for a HEV can be developed.
- While $SOC < 0.3$ the battery can be charged based on some rules in order to have a variable rate while charging instead of charging with a constant rate ($20N.m/s$).

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Appendix

4.1 Matlab Functions for Emissions Calculation

4.1.1 Emissions Calculation

```
1 function [EmissionmassBeforeCat ,EmissionmassAfterCat] = ...
2     calcemissions(tp, lambda3, dist, mairc, mfc, lightOffTime)
3 % [EmissionmassBeforeCat ,EmissionmassAfterCat] =
4 % calcemissions(t, lambda, dist, mairc, mfc, lightOffTime)
5 %
6 % Funktionen calcEmissions anvander foljande indata for att
7 % bestamma emissionsnivan fore och efter cat.
8 % Indata:
9 % t      tillhorande tidsvektor med tidpunkter for varje sampel [s]
10 % lambda en vektor med labmdavarden for varje tidpunkt i en korcykeln
11 % [s]
12 % dist   den korda distansen i m vid varje sampel.
13 % mairc  vektor med luftflode in i cylinder for varje tidpunkt [kg/s]
14 % mfc    "-"      branslefode      "-" [kg/s]
15 % lightOffTime Tiden i sekunder tills katalysatorn startar [s]
16 % Utdata:
17 % emissionmassBC emissionsniva for [CO NOx HC] i [g/km] fore cat
18 % emissionmassAC emissionsniva for [CO NOx HC] i [g/km] efter cat
19 %
20 % Av : Ingemar Andersson
21 %     Per Andersson
22 %
23 % $Date: 2004/07/06 09:37:50 $
24 % $Revision: 1.1.1.1 $
25 %
26 % Konvertera fran SI-enheter
27 dist = dist/1e3;
```

```

28 mairc = mairc * 1e3;
29 mfc   = mfc * 1e3;
30
31 tic;
32
33 %disp('Beraknar emissioner')
34 result = emissions(lambda3);
35 %disp('Utvarderar ')
36
37 % Utvarderar data
38
39 loff = tp > lightOffTime; % Ger alla tidpunkter storre an light off
40
41 MM = [28 14+16 16]; % Molmassor. Kompenserar for att emissions
42 % returnerar tal i procent for CO samt promille for NOx och HC.
43 % Observera att O2 och H2 inte anvands
44
45 MCO = 12+16;
46 MNOx = 14+16;
47 MHC = 12+4;
48 MO2 = 32;
49
50 % Total massa emissioner
51 EMF = mairc + mfc;
52
53 Mtot = 28.5; % Ur Heywood.
54
55 len = length(result.emCO);
56
57 resultCO = EMF .* (result.emCO*MCO)/(Mtot)/100;
58 resultNOx = EMF .* (result.emNOx*MNOx)/(Mtot)/1000;
59 resultHC = EMF .* (result.emHC*MHC)/(Mtot)/1000;
60 resultO2 = EMF .* (result.emO2*MO2/Mtot);
61
62 EmissionmassBeforeCat = [resultCO resultNOx resultHC]'*...
63 [diff(tp); 0]/dist(length(dist));
64
65 disp('Emissioner fore katalysator:')
66 disp(sprintf('\t CO : %1.2f [g/km]', EmissionmassBeforeCat(1)))
67 disp(sprintf('\t HC : %1.2f [g/km]', EmissionmassBeforeCat(3)))
68 disp(sprintf('\t NOx : %1.2f [g/km]', EmissionmassBeforeCat(2)))
69
70
71 %result2(i,:) = EMF(i) * (XvolCat(i,:).*MM)/(Mtot);
72 resultCO = EMF .* (result.emCO*MCO)/(Mtot)/100.*(1-result.catEffCO.*loff);
73 resultNOx = EMF .* (result.emNOx*MNOx)/(Mtot).*...
74 (1-result.catEffNOx.*loff)/1000;
75 resultHC = EMF .* (result.emHC*MHC)/...
76 (Mtot).*(1-result.catEffHC.*loff)/1000;
77 resultO2 = EMF .* (result.emO2*MO2/Mtot);
78
79 EmissionmassAfterCat = [resultCO resultNOx resultHC]'*[diff(tp); ...
80 0]/dist(length(dist));
81 disp(' ')
82
83 disp('Emissioner efter katalysator:')
84 disp(sprintf('\t CO : %1.2f [g/km]\tGransvarde EURO 3 : 2.3 [g/km]
85 EURO 4 : 1.0 [g/km]', EmissionmassAfterCat(1)))

```



```

85 disp(sprintf('\t HC : %1.2f [g/km]\tGransvarde EURO 3 : 0.20 [g/km]
EURO 4 : 0.10 [g/km]', EmissionmassAfterCat(3)))
86 disp(sprintf('\t NOx : %1.2f [g/km]\tGransvarde EURO 3 : 0.15 [g/km]
EURO 4 : 0.08 [g/km]', EmissionmassAfterCat(2)))
87 %t oc

```

4.1.2 Emissions

```

1 function r = emissions(lambda)
2 % EMISSION, emission calculation as a function of lambda.
3 % result = emission(lambda);
4 % where 0.7 <= lambda <= 1.4
5 %
6 % Result is a structure with following fieles
7 %
8 % emissionName This contains a list of the order of the next field , emission.
9 % emH2 Emission in mole fractions of H2
10 % catEffH2 Contains a value between 0.0 and 1.0 describing the
11 % efficiency of the catalytic coverter for H2.
12 % Is in this version always 0.
13 % emCO Emission in volume percent
14 % catEffCO Contains a value between 0.0 and 1.0 describing the
15 % efficiency of the catalytic coverter for CO.
16 % emNOx Emission in volume promille
17 % catEffNOx Contains a value between 0.0 and 1.0 describing the
18 % efficiency of the catalytic coverter for NOx.
19 % emHC Emission in volume promille
20 % catEffHC Contains a value between 0.0 and 1.0 describing the
21 % efficiency of the catalytic coverter for HC.
22 % emO2 Emission in mole fractions of O2
23 % catEffO2 Contains a value between 0.0 and 1.0 describing the
24 % efficiency of the catalytic coverter for O2.
25 % Is in this version always 0.
26 %
27
28 % Av: Per Andersson
29 % $Revision: 1.1.1.1 $
30
31 % Set cat eff window. offset = 0.01 ==> 0.99 <=window <= 1.01
32 offset = 0.02;
33
34 % Remove error prone data
35 lambda = max(0.7+ offset ,lambda);
36 lambda = min(1.4- offset ,lambda);
37
38
39 % Set emission names
40 r.emissionNames = {'H2', 'CO', 'NOx', 'HC', 'O2'};
41
42 % Initiate result to zero.
43
44 % Set up local variables
45 H2 = 1; % Index in matrix for H2
46 CO = 2; % Index in matrix for CO
47 NOx = 3; % Index in matrix for NOx
48 HC = 4; % Index in matrix for HC

```

```

49 O2 = 5; % Index in matrix for O2
50
51
52 % Locals for indexing in structure emission.
53 beforeCat = 1; % First row
54 utilization = 2; % Second row
55
56 % Set up tables for interpolation
57 % First column : lambda
58 % 2nd column : Value in percent
59
60 % Tables for gas AfterCat is not currently used. They was used to
61 % calculate cat efficiency. But it did not give a satisfieying result.
62
63 H2TableBeforeCat = [
64     0.67, 6.2e-2;
65     0.77, 3.3e-2;
66     0.83, 0.02;
67     0.91, 8.8e-3;
68     1.0 , 2.2e-3;
69     1.11, 0;
70     1.25, 0;
71     1.4 , 0];
72
73 COTableBeforeCat = [
74     0.7   16;
75 % 0.88, 7;
76     0.9 , 6;
77     0.96, 3;
78     1.0 , 1.59;
79     1.1 , 0.47;
80     1.15, 0.45;
81     1.2 , 0.47;
82     1.3 , 0.50;
83     1.4 , 0.48];
84
85 COTableAfterCat = [
86     0.88 , 6;
87     0.9 , 4.78;
88     0.976, 0.5;
89     0.99 , 0.375;
90     1.004, 0.33;
91     1.15 , 0.28];
92
93 COCatEff = [
94     0.7 - offset , 0.0;
95     0.89 - offset , 0.0;
96     0.92 - offset , 0.03;
97     0.95 - offset , 0.08;
98     0.97 - offset , 0.123;
99     0.98 - offset , 0.36;
100    0.99 - offset , 0.62;
101    1.0 - offset , 0.92;
102    1.003 - offset , 0.985;
103    1.005 - offset , 0.995;
104    1.01 - offset , 1.0;
105    1.025 - offset ,1.0;
106    1.05 - offset , 1.0;

```

```
107     1.1 - offset , 1.0;
108     1.15 - offset , 1.0;
109     1.4 - offset , 1.0];
110
111 NOxTableBeforeCat = [
112     0.7 , 0.5;
113 % 0.87 , 1.5;
114     1.0 , 2.8;
115     1.03 , 2.9;
116     1.1 , 2.75;
117     1.4 , 0.94;
118 ];
119
120 NOxTableAfterCat = [
121     0.9 , 0.047;
122     0.99 , 0.047;
123     1.0 , 0.094;
124     1.004 , 0.19;
125     1.018 , 5;
126     1.032 , 2.85;
127     1.05 , 2.85;
128     1.1 , 2.5];
129
130 NOxCatEff = [
131     0.7 + offset , 1.0;
132     0.8 + offset , 1.0;
133     0.9 + offset , 1.0;
134     0.97 + offset , 1.0;
135     0.98 + offset , 1.0;
136     0.99 + offset , 1.0;
137     1.0 + offset , 0.95;
138     1.01 + offset , 0.2;
139     1.015 + offset , 0.8*mean([0.2 0.043]);
140     1.02 + offset , 0.043;
141     1.03 + offset , 0.029;
142     1.08 + offset , 0.02;
143     1.15 + offset , 0.0;
144     1.2 + offset , 0.0;
145     1.4 + offset , 0.0];
146
147 HCTableBeforeCat = [
148     0.7 , 5;
149 % 0.88 , 3.5 ;
150     0.9 , 3.18;
151     1.0 , 2.06;
152     1.1 , 1.17;
153     1.14 , 1;
154     1.2 , 1.17;
155     1.3 , 2.2;
156     1.35 , 3.18;
157     1.4 , 5.4];
158
159 HCTableAfterCat = [
160     0.9 , 0.56;
161     1.0 , 0.05;
162     1.1 , 0.08;
163     1.2 , 0.08];
164
```

```

165 HCCatEff = [
166     0.7 - offset , 0.05;
167 % 0.88 - offset , 0.17;
168     0.9 - offset , 0.18
169     0.97 - offset , 0.58;
170     0.98 - offset , 0.72;
171     0.99 - offset , 0.84;
172     1.00 - offset , 0.90;
173     1.01 - offset , 0.93;
174     1.02 - offset , 0.93;
175     1.03 - offset , 0.93;
176     1.15 - offset , 0.93;
177     1.4 - offset , 0.93];
178
179 O2TableBeforeCat = [
180     0.7 , 2.1e-3;
181     0.77, 2.2e-3;
182     0.83, 2.67e-3;
183     0.91, 4.4e-3;
184     1.0 , 6.67e-3;
185     1.11, 2.67e-2;
186     1.25, 4.8e-2;
187     1.4 , 7e-2];
188
189
190
191
192
193 % Check limits of lambda:
194 lambda(find(lambda < 0.7)) = 0.7;
195 lambda(find(lambda > 1.4)) = 1.4;
196
197 % Assign to structure
198
199 r.emH2      = interp1(H2TableBeforeCat(:,1) , H2TableBeforeCat(:,2) , lambda);
200 r.catEffH2  = 0;
201
202 r.emCO      = spline(COTableBeforeCat(:,1) , ...
203     COTableBeforeCat(:,2) , lambda);
204 r.catEffCO  = interp1(COCatEff(:,1) , COCatEff(:,2) , lambda);
205
206 r.emNOx     = spline(NOxTableBeforeCat(:,1) , ...
207     NOxTableBeforeCat(:,2) , lambda);
208 r.catEffNOx = interp1(NOxCatEff(:,1) , NOxCatEff(:,2) , lambda);
209
210 r.emHC      = spline(HCTableBeforeCat(:,1) , HCTableBeforeCat(:,2) , lambda);
211 r.catEffHC  = interp1(HCCatEff(:,1) , HCCatEff(:,2) , lambda);
212
213 r.emO2      = spline(O2TableBeforeCat(:,1) , O2TableBeforeCat(:,2) , lambda);
214 r.catEffO2  = 0;

```


4.2 Simulink Implementation

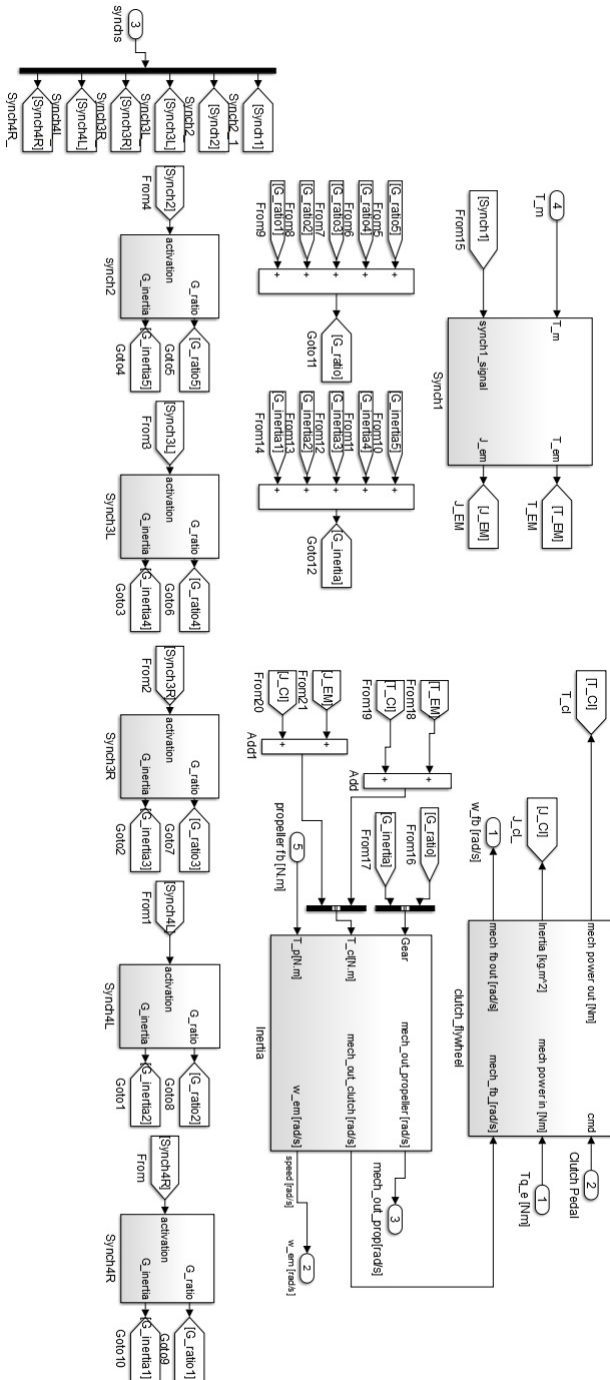


Figure 4.1: Simulink model of gearbox.

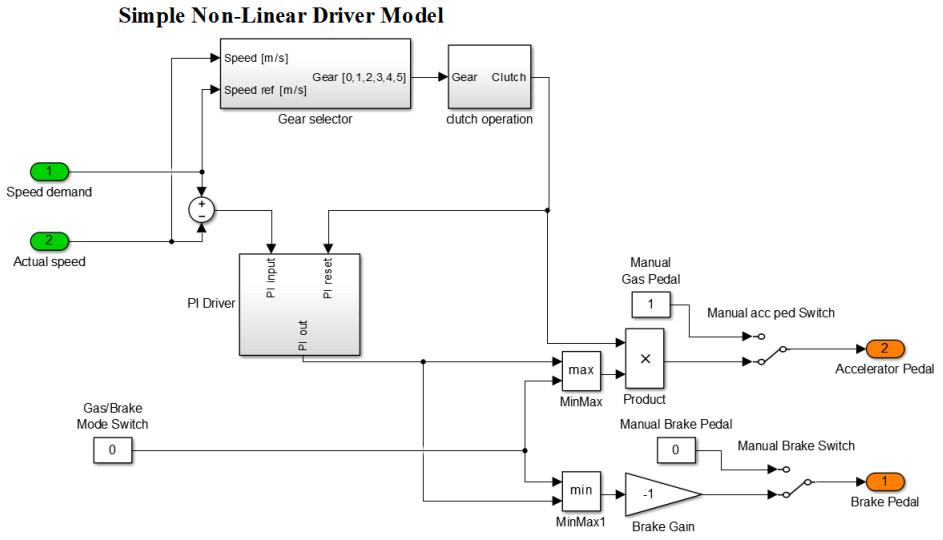


Figure 4.2: Brake and throttle control.

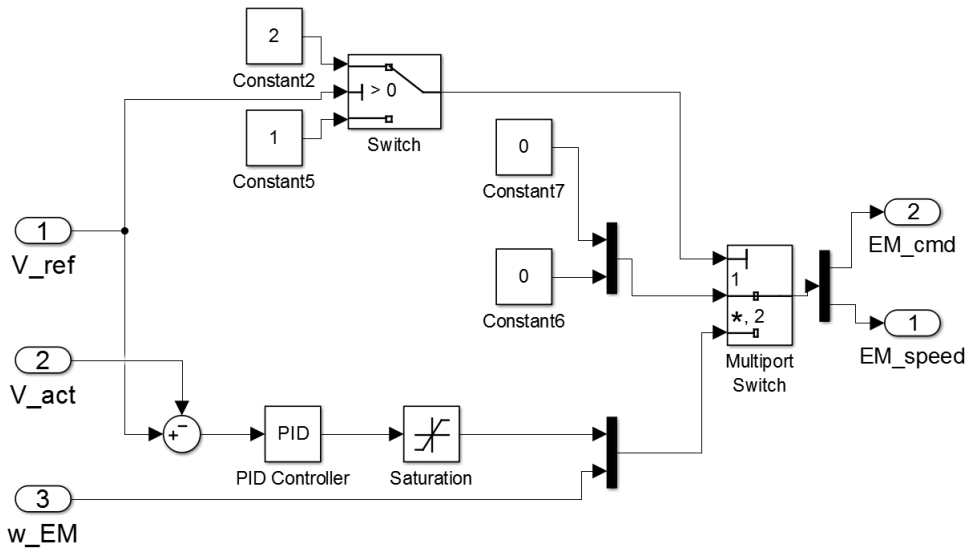


Figure 4.3: EM command and angular speed control of EM.

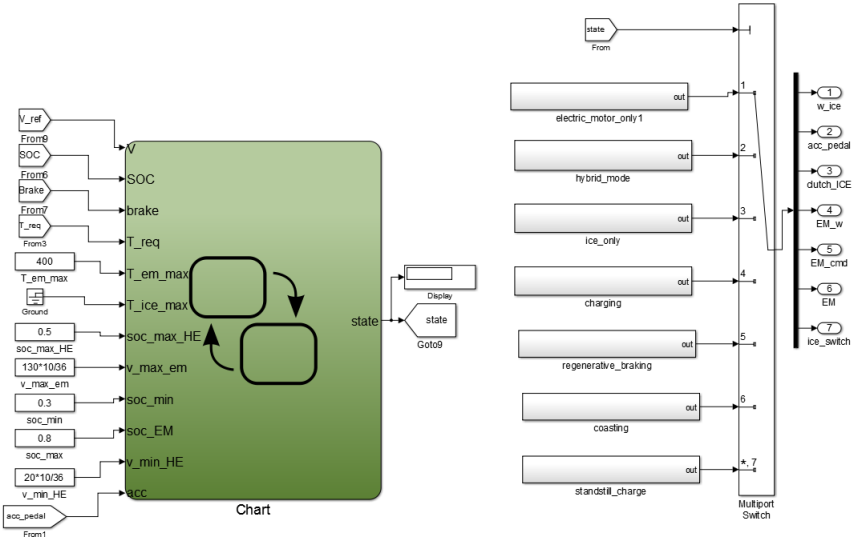


Figure 4.4: Rule based controller.

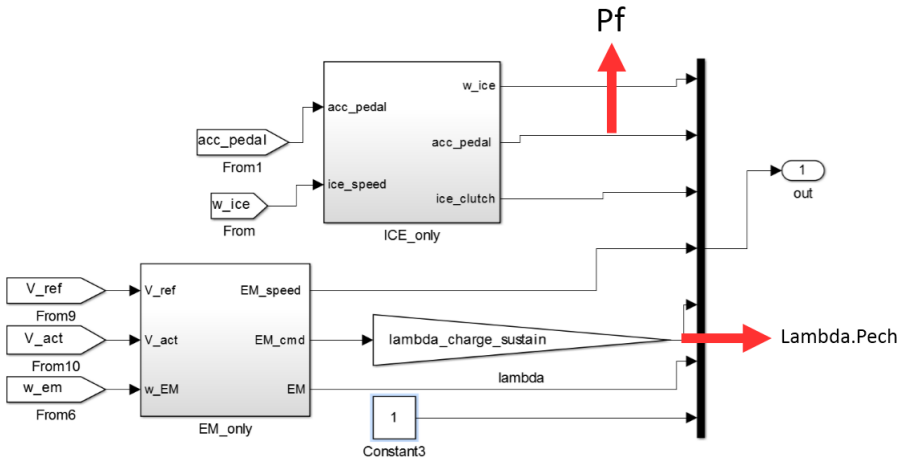


Figure 4.5: Equivalent mass minimization for hybrid mode.

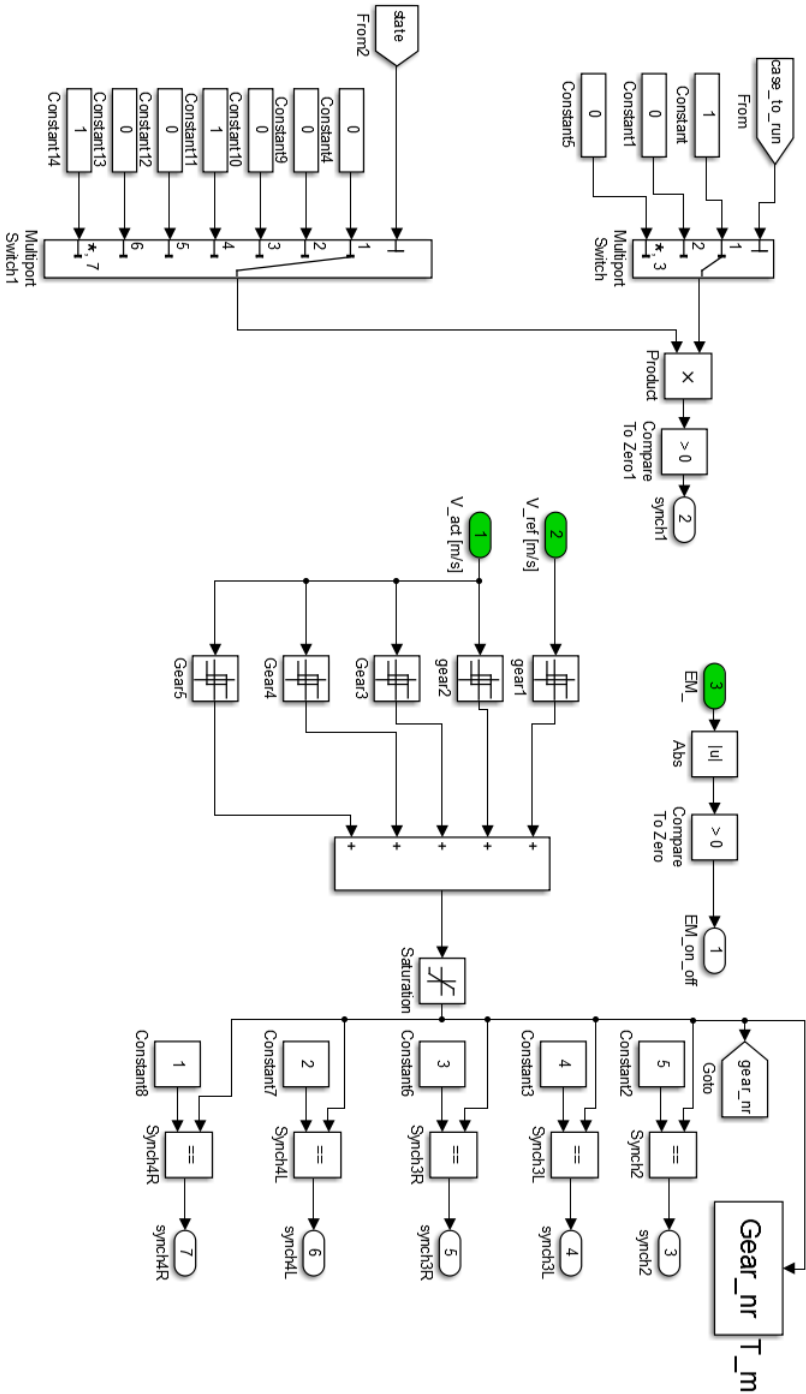


Figure 4.6: Simulink implementation of activating and deactivating each synchronizer.



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