Modelling of the Toyota Estima MPV 2.4 16v Hybrid

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Modelling of the Toyota Estima
MPV 2.4 16v Hybrid

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Traineeship report

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Abstract

The Toyota Motor Company developed the Toyota Estima Hybrid which is equipped with the Toyota Hybrid System-CVT (THS-C). In this report the THS-C will be investigated and a model is derived. This model is implemented in Matlab Simulink and some driving modes are simulated. This model is made to be implemented in ADVANCE, a coupled vehicle dynamics and power train simulations program developed by TNO, which will be done in future research.

Contents

1 Introduction

Toyota Motor Corporation developed the Toyota Hybrid System-CVT (THS-C) for mini-vans. In this report some dynamic aspects of the THS-C will be further investigated. The primary goal is to make a model of the Toyota Estima Hybrid in ADVANCE, a coupled vehicle dynamics and power train simulation program developed by TNO. Simulation of the power train using Simulink is performed and the results are evaluated.

First a study is preformed to make a model of the THS-C, which is in fact the front drive of the Toyota Estima. The THS-C is based on the Super CVT (K110) [7], but the conventional torque converter is replaced by a torsional damper and a new developed motor-generator. For smooth switching between the different driving modes it makes use of wet multiple disc clutches. The Toyota Estima is also equipped with a motor-generator which drives the rear wheels for optimal vehicle dynamics and further fuel saving. In fact, the powertrain is of a hybrid type. In the simulation study, the dynamic behavior of the front drive is evaluated and the functionality of the model for further research is discussed.

Chapter 2 explains the Toyota Estima Hybrid in more detail. In particular, the various driving modes and functions of the THS-C are explained. Chapter 3 explains how the mathematical model of this power train is derived. Chapter 4 explains how the Simulink model is composed, how the simulations were done and discusses the results. Finally the conclusions and some recommendations for further research are given in chapter 5.

2 Toyota Estima

In June 2001 Toyota introduced the Estima Hybrid in Japan, the worlds first hybrid minivan and the worlds first mass produced gasoline/electric hybrid four-wheel drive vehicle. In this section the Toyota Estima Hybrid will be further investigated.

2.1 Hybrid vehicles

The main idea of a hybrid vehicle is to improve the fuel-efficiency without loosing the main performance and comfort characteristics. Some functions of hybrid vehicles are, [1]:

1. Energy-loss reduction:
   Upon certain conditions, the system automatically stops the idling of the engine (idling stop), thus reducing the fuel consumption during vehicle stand-still.

2. Energy recovery and reuse:
   The energy that would normally be wasted as heat during deceleration and braking is recovered as electrical energy, which is used later to power the starter and the electric motor.

3. Motor assist:
   The electric motor assists the engine during acceleration.

4. High-efficiency operation control:
   The system maximizes the vehicle's overall efficiency by using the electric motor to run the vehicle under operating conditions in which the engine's efficiency is low and by generating electricity under operating conditions in which the engine's efficiency is high.

The parallel hybrid system has all of these characteristics and therefore provides both superior fuel efficiency and driving performance.
The Toyota Estima is a parallel hybrid, the Toyota hybrid system with CVT (THS-C) combines a newly developed high efficiency gasoline engine, front motor and CVT transmission. This system has two motors, and depending on the driving conditions, uses only the electric motor or the driving power from both the electric motor and the engine, in order to achieve the highest efficiency level. Furthermore, when necessary, the system drives the wheels while simultaneously generating electricity using a generator. In Figure 1, the hybrid system can be seen.
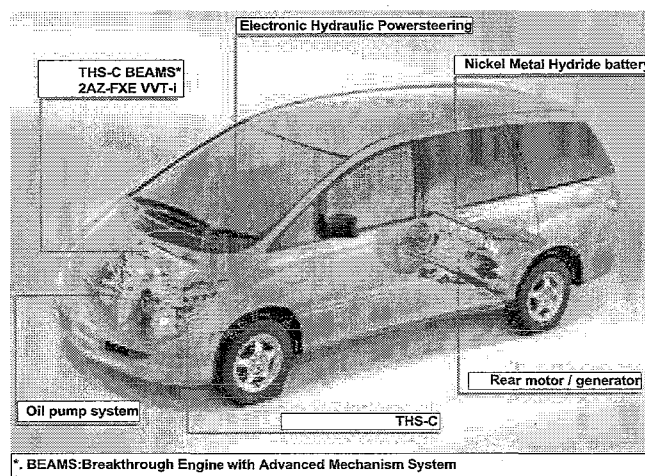

Figure 1: THS-C with engine, [9]

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For even better driving performance the Toyota Estima has also an electric motor to drive the rear axle. In this way the vehicle dynamics can be improved on slippery roads or in extreme situations e.g. j-turn, lane change, etcetera. This motor can also be used for regenerative braking. A short overview of the different driving modes in driving circumstances is given in appendix B.

For better fuel-efficiency Toyota developed a new way to drive the oil pump system of the CVT. It is an electric oil pump system, contrary to the oil pump directly driven by the engine or drive shaft in conventional CVT vehicles. This way the oil pump system can be run independent from the speed of the motor, engine or drive shaft.

In this paper the driven rear-axle and the fuel-efficiency gained by the oil pump system will not be considered; the main focus is the front drive. This means the dynamic model is completely based on the THS-C and high efficiency engine and how its functions.

A disadvantage of a hybrid vehicle is its weight which is substantially higher than the conventional counterpart. However in Table 1 it can be seen that still an incredible amount of fuel can be saved compared to the other models.

Table 1: Various types of the Toyota Estima, [4]

model	weight	top speed	acceleration	average consumption
Estima 2.4 16V	1630 kg	185 km/h	10.9 s	9.5 [l/100 km]
Estima 2.4 16V 4WD aut.	1690 kg	unknown	unknown	9.8 [l/100 km]
Estima 2.4 16V aut.	1630 kg	180 km/h	11.2 s	10.8 [l/100 km]
Estima 2.4 16V Hybrid	1850 kg	unknown	unknown	5.6 [l/100 km]

However missing in Table 1, it will be assumed that the Hybrid has a top speed of 190 km/h and shows a reasonable acceleration. The largest difference is the weight. This difference implies € 140,- a year more taxes then the lighter version in The Netherlands, [3]. But against this, for efficient vehicles one can get reductions on the import taxes in The Netherlands. In other words, the purchase of these kind of vehicles is promoted by the Dutch government, [10].

2.2 Driving modes

For optimal fuel-efficiency and driving performance there are several control strategies for the THS-C. The strategy developed by Toyota is discussed next.

In Figure 2 a scheme of the strategy is shown. Four different configurations can be seen:

1. EV power:
   The vehicle is only powered by the electric motor

2. Engine power:
   The vehicle is only powered by the engine

3. Electric torque converter power:
   The vehicle is accelerated by the engine and the electric motor is used as a torque converter for smooth pulling off

4. EV & Engine power:
   If the engine does not provide enough power the electric motor gives an extra boost
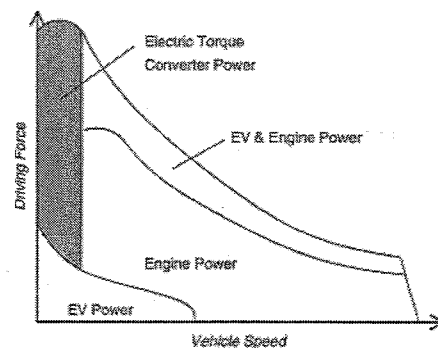
Figure 2: D-range driving mode and driving force, [4]

Besides the upper four, there are some more modes. In particular the THS-C has to charge the battery when it is on low capacity and has to drive the vehicle in reverse direction. Finally a neutral for situations like towing the vehicle is available. All these modes are realized with engaging and releasing

of two clutches and a brake. In Table 2 these modes are shown, with an overview which clutch or brake to open and close.

Table 2: Operation chart of friction elements, [4]

Driving mode	C1	C2	B1
EV power	◯		
Engine power	◯	◯	
Electric torque converter power		◯	
Reverse range^a	◯		△
P range charge			◯
N range			

◯: Engage, △: Slip.

^{a} C1: EV power, B1: Engine power (friction mode)

2.3 THS-C

The THS-C is based on the Super CVT (K110) which originally is developed by Toyota in 2000, [4], [7]. The conventional torque converter is replaced by a torsional damper and motor generator, a cross section can be seen in Figure 3.

A schematic view of the THS-C is shown in Figure 4. From this latter view the dynamic model is derived, which is done in the next section.
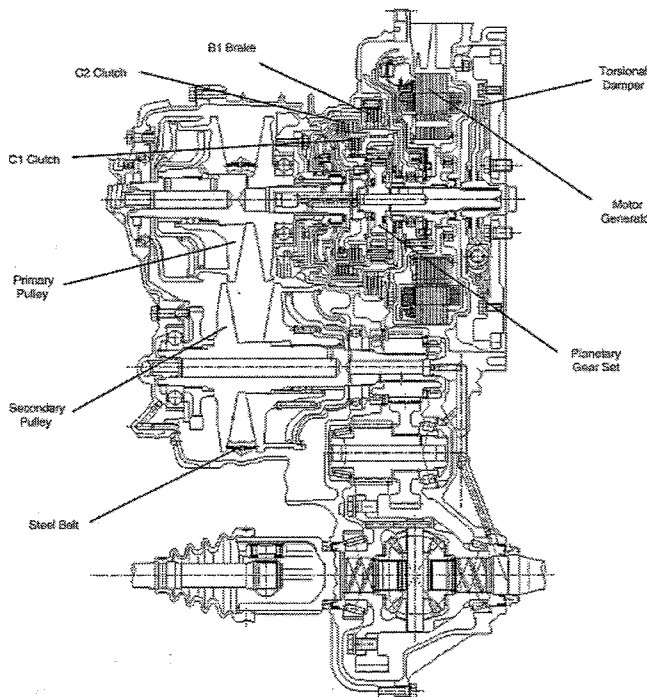
Figure 3: Cross section of transaxle, [4]
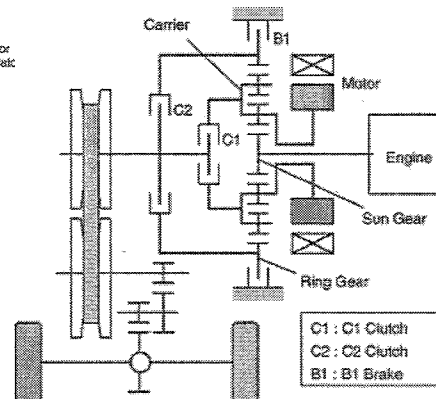
Figure 4: Schematic of THS-C, [4]

3 Modelling

In this section all the relevant equations will be derived. The explanations of the symbols can be found in appendix A.

3.1 Equations of motion

To derive the equations of motion, first a model of the front drivetrain is composed. Figure 5 shows the model of the front drive.
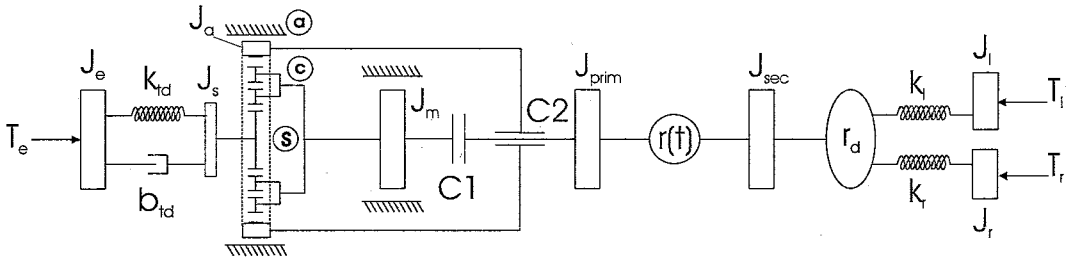
Figure 5: Model of the front drive

In Figure 5 the different parts of the front drivetrain can be seen. From the left to the right, the engine torque drives the inertia of the engine; a torsional damper is modelled with a linear spring and damper and is directly connected to the planetary gear via the sun; the ring of the planetary gear can be connected to the world through brake 1 ($B1$), but is also connected to the primary pulley via clutch 2 ($C2$); the electric motor having inertia J_m and torque T_m is directly connected to the carrier but can also be connected with the primary pulley by closing clutch 1 ($C1$); the CVT is presented by two inertias and a time dependant ratio (J_{prim}, J_{sec} and $r(t)$); the secondary pulley is connected to the final reduction gear, which is further connected to the differential; the stiffness of the axles to the wheels is presented by two linear springs; the wheels are presented as inertias so there is no tyre model included.

The relevant torques and sign conventions are defined in Figure 6, with the positive direction to the right.
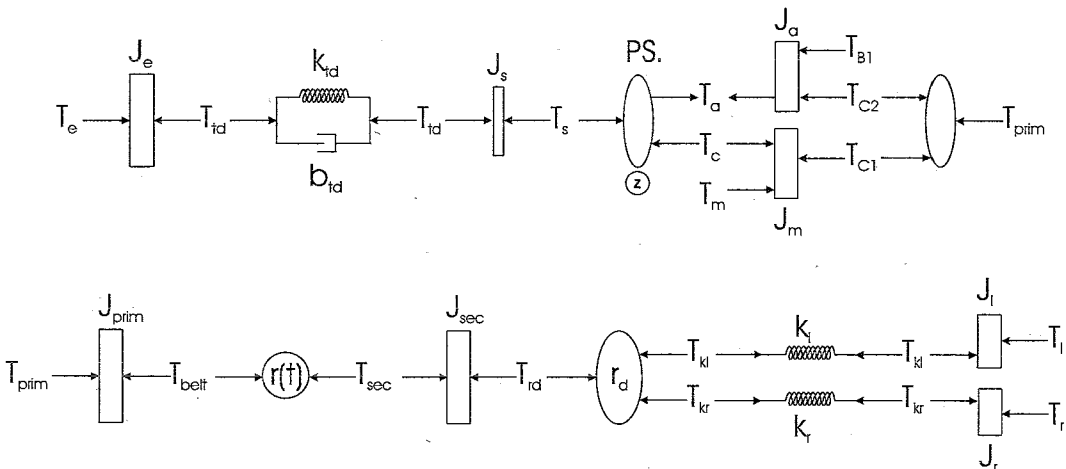
Figure 6: Definitions and sign conventions of the front drive

The relevant equations now can be derived. However, first a closer look at the planetary gearset is taken.

3.1.1 Planetary gear

In this paragraph the planetary gearset and its relevant equations are derived. The THS-C consists of a double pinion planetary gear. If we assume that all gear meshes in the planetary gear are mass-less we know that conservation of power must hold:

$$P_c = P_s + P_a \tag{1}$$
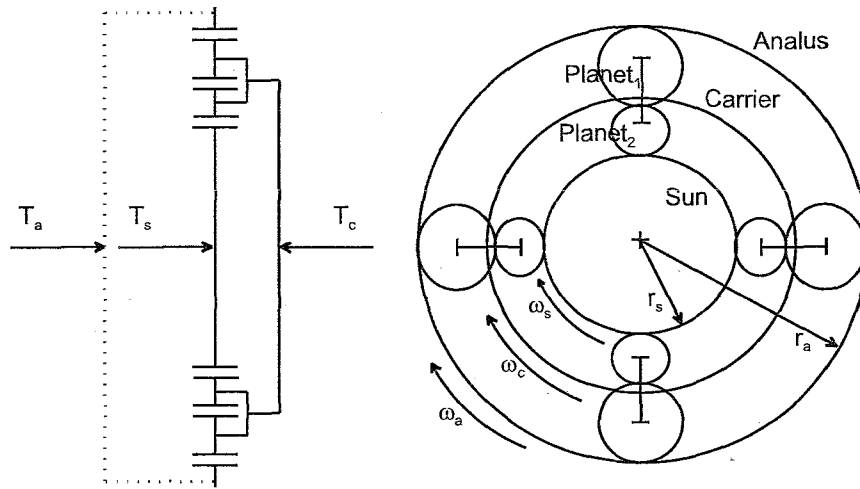
Figure 7: Planetary gear

In this equation we follow the sign conventions as in Figure 7. Furthermore,

$$P = T \cdot \omega \tag{2}$$

Because we assume there are no losses in the planetary gear we use equation (2) to solve equation (1), yielding:

$$T_c \cdot \omega_c = T_s \cdot \omega_s + T_a \cdot \omega_a \tag{3}$$

And for the rotational speeds we know the following must hold, [6]:

$$\omega_s = (1 + z) \cdot \omega_c - z \cdot \omega_a \qquad ; z = -\frac{r_a}{r_s} \tag{4}$$

Because the planetary gear is a double pinion gear the characteristic geometric ratio z is defined to be negative. With equation (4), equation (3) can be rewritten into:

$$T_c \cdot \omega_c = T_s \cdot ((1 + z) \cdot \omega_c - z \cdot \omega_a) + T_a \cdot \omega_a \tag{5}$$

And:

$$\omega_c \cdot (T_c - T_s \cdot (1 + z)) = \omega_a \cdot (T_a - z \cdot T_s) \qquad ; \forall \{\omega_c, \omega_a\} \tag{6}$$

This must hold for all ω's so the torques are related to each other according equation (7):

$$\begin{aligned} T_c &= (1 + z) \cdot T_s \\ T_a &= z \cdot T_s \qquad ; z = -\frac{r_a}{r_s} \end{aligned} \tag{7}$$

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3.1.2 Dynamic equations

Now the equations of motion will be derived for the model of the front drive line. To start with the engine and the torsional damper:

$$J_e \cdot \dot{\omega}_e = T_e - T_{td} \tag{8}$$

$$\dot{T}_{td} = k_{td} \cdot (\omega_e - \omega_s) + b_{td} \cdot (\dot{\omega}_e - \dot{\omega}_s) \tag{9}$$

Now the inertias of the planetary gear will be taken into account:

$$J_s \cdot \dot{\omega}_s = T_{td} - T_s \tag{10}$$

$$J_a \cdot \dot{\omega}_a = -T_a - T_{B1} - T_{C2} \tag{11}$$

For the front motor/generator holds:

$$J_m \cdot \dot{\omega}_m = T_c + T_m - T_{C1} \tag{12}$$

The inertias of the CVT are modelled by:

$$J_{prim} \cdot \dot{\omega}_{prim} = T_{prim} - T_{belt} \tag{13}$$

$$J_{sec} \cdot \dot{\omega}_{sec} = T_{sec} - T_{rd} \tag{14}$$

The stiffness of the front axle can be modelled as:

$$\dot{T}_{k_l} = k_l \cdot (\omega_{r_{d_l}} - \omega_l) \tag{15}$$

$$\dot{T}_{k_r} = k_r \cdot (\omega_{r_{d_r}} - \omega_r) \tag{16}$$

There is no model for the tyres, only the inertias of the wheels are modelled:

$$J_l \cdot \dot{\omega}_l = T_{k_l} - T_l \tag{17}$$

$$J_r \cdot \dot{\omega}_r = T_{k_r} - T_r \tag{18}$$

3.1.3 Kinematic equations

In section 3.1.1 we already mentioned equation 4, it is repeated here, [6]:

$$\omega_s = (1 + z) \cdot \omega_c - z \cdot \omega_a \qquad ; z = -\frac{r_a}{r_s} \tag{19}$$

For the differential holds:

$$\omega_{sec} \cdot r_d = \frac{\omega_l + \omega_r}{2} \tag{20}$$

$$\omega_{sec} = \frac{\omega_{prim}}{r_{CVT}} \tag{21}$$

3.1.4 Static equations

In section 3.1.1 the static equations for the planetary gear are derived, the relevant equations are:

$$T_a = z \cdot T_s \tag{22}$$

$$T_c = (1 + z) \cdot T_s \tag{23}$$

For the torques working on the primary pulley the following holds:

$$T_{prim} = T_{C1} + T_{C2} \tag{24}$$

For the torques through the CVT holds:

$$T_{sec} = T_{belt} \cdot r_{CVT} \tag{25}$$

And for the differential holds:

$$T_{rd} = T_{k_l} + T_{k_r} \tag{26}$$

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3.1.5 Stick / slip conditions

In the Toyota Estima two clutches and one brake are used to switch into the appropriate driving mode. To detect whether a clutch sticks or slips the 'Karnopp' approach is adopted, [8], [11]. Next for every clutch the equations are derived.

Clutch 1: The torque which can be transmitted through a clutch depends on the clamping pressure. For $C1$ the following holds:

$$T_{C1} = \begin{cases} \dfrac{(T_c + T_m) \cdot J_{prim} + (T_{belt} - T_{C2}) \cdot J_m}{J_m + J_{prim}} & \text{if } |\omega_m - \omega_{prim}| < \varepsilon \\[2mm] & \text{and } |T_{C1}| \le |T_{C1_{stick}}| \\[2mm] n \cdot p \cdot A \cdot R \cdot \mu_{C1} \cdot sign(\omega_m - \omega_{prim}) & \text{else} \end{cases} \tag{27}$$

With:

$$T_{C1_{stick}} = n \cdot p \cdot A \cdot R \cdot \mu_{C1_{stick}} \cdot sign(T_{C1}) \tag{28}$$

This is the torque which can be maximally transmitted if the clutch sticks. $\mu_{C1_{stick}}$ is always greater then μ_{C1}, otherwise the clutch will never close smoothly. Equation (27) is clarified with Figure 8.
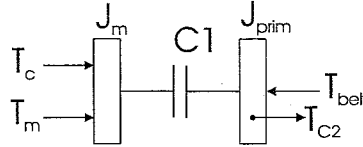
Figure 8: Stick/slip system for clutch 1

Clutch 2: The principal for $C2$ is the same as $C1$ only other torques apply for this subsystem, as can be seen in Figure 9:

$$T_{C2} = \begin{cases} \dfrac{(-T_a - T_{B1}) \cdot J_{prim} + (T_{belt} - T_{C1}) \cdot J_a}{J_{prim} + J_a} & \text{if } |\omega_a - \omega_{prim}| < \varepsilon \\[2mm] & \text{and } |T_{C2}| \le |T_{C2_{stick}}| \\[2mm] n \cdot p \cdot A \cdot R \cdot \mu_{C2} \cdot sign(\omega_a - \omega_{prim}) & \text{else} \end{cases} \tag{29}$$

With:

$$T_{C2_{stick}} = n \cdot p \cdot A \cdot R \cdot \mu_{C2_{stick}} \cdot sign(T_{C2}) \tag{30}$$

Figure 9: Stick/slip system for clutch 2

Brake 1: For $B1$ holds:

$$T_{B1} = \begin{cases} -T_a - T_{C2} & \text{if } |\omega_a| < \varepsilon \wedge |T_{B1}| \le |T_{B1_{stick}}| \\ n \cdot p \cdot A \cdot R \cdot \mu_{B1} \cdot sign(\omega_a) & \text{else} \end{cases} \tag{31}$$

with:

$$T_{B1_{stick}} = n \cdot p \cdot A \cdot R \cdot \mu_{B1_{stick}} \cdot sign(T_{B1}) \tag{32}$$

This is clarified with Figure 10:
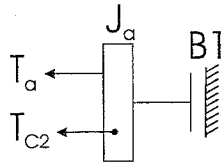
Figure 10: Stick/slip system for brake 1

The next step is to implement these equations into the Simulink model.

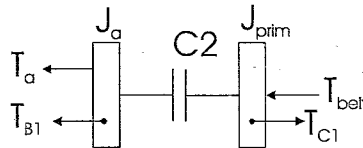## 3.2 Simulink

In Simulink the front drive is split up in different parts which interact with each other. The engine and torsional damper are two separate blocks. The planetary gear, the two clutches and the brake are grouped in one block, but this block has three subsystems. In the first subsystem clutch 1, in the second subsystem clutch 2, and in the third subsystem brake 1 are implemented. The CVT, final drive and axle are three separate blocks. The vehicle inertia and road resistance are modelled in one block. Finally, the drive modes and torque controller are also modelled in one block. In the next chapter the Simulink model and simulation results will be discussed in more detail.

4 Simulation

To validate the model we only implement the model for the THS-C and a simple vehicle model. Prior to implementing the model in Advance we first simulate the model in basic Simulink. Advance is an advanced simulation tool with extended data-handling within the Simulink environment. Coupled power train and vehicle dynamics modelling can get quite complex, therefor the handling of the parameters and data is important.

4.1 Simulink

A model is implemented in Simulink to look how the THS-C operates. With the equations from Chapter 3 this model can be realized. First, we observe how parameters can be organized in 'Matlab-structures'.

4.1.1 Structures

For the handling of the vehicle parameters we choose to use structures, in order to divide the different parts of the front drive into substructures. This enables easy adjustment of the parameters for each part. The input structure is called 'd'. This structure contains all the parameters which are used in the model. The used structure for the drivetrain data can be found in appendix E.

4.1.2 Models

Now the different parts of the model will be discussed. The different blocks are:

1. Engine

2. Torsional damper

3. Planetary gear, the two clutches and the brake

4. CVT

5. Final drive

6. Axle

7. Drive modes & torque controller

These blocks can also be seen in appendix C. The engine, torsional damper, CVT, Final drive and axle are relatively simple models, these are given in appendix D. In the next sections the model of the Planetary gear, the two clutches and the brake and the model of the driving modes and torque controller are further evaluated.

4.1.3 Planetary gear, the two clutches and the brake

For the clutches and the brake the same model is used. This to keep the model open and easy to adjust for different configurations, i.e. locations of clutches and brakes. First the basic clutch model will be discussed. To model a clutch a Matlab-function is written, this because slip and stick modes are easier to describe in Matlab than in Simulink. If the clutch slips, a controllable torque is transmitted and the in- and output ω's are not the same. If the clutch sticks the torque transmitted has a fixed relation with the in- and output torques and the in- and output ω's are the same, i.e. see equation (27). Because the clutch switches between these modes and we want this to be a smooth transition, a smooth slip control function must be designed. The m-file used is given in appendix F. The design of the slip control function lies outside the scope of this report. Some conceptual ideas for this function are discussed in Serrarens, et al [8].

4.1.4 Driving modes & torque controller

The Toyota Estima is equipped with a new developed control strategy. This strategy was already shown in Figure 2. To imitate this strategy in the model, a subsystem is used which checks in what mode the drivetrain is operating and controls the clutches, see appendix F. In Figure 11 the CVT ratio and a percentage of the maximum throttle opening can be given as an input.
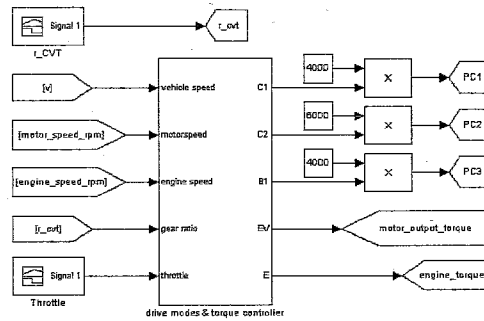

Figure 11: Simulink model of the driving modes & torque controller

In the block called "drive modes & torque controller" the m-file "drivemode" is used. This is given in appendix G. When the throttle opening is set to 50% and the CVT ratio is set to 2.396 we get Figure 12. In the background, Figure 2 from section 2.2 can be seen. Here we can see the model starts in electric torque converter power mode and switches into engine power mode. From Figure 2 the points are determined where the appropriate clutches and brake are (de-)activated, e.g. the vehicle top speed is set to 190 km/h, so with this the maximum vehicle speed in EV-power can be evaluated, here this is around 77 km/h. The maximum speed in electric torque converter power mode is 21 km/h.
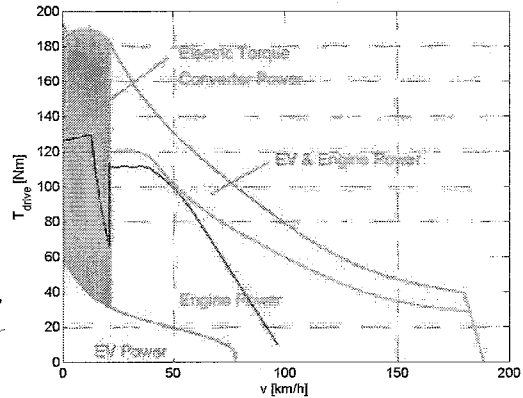
Figure 12: Driving force versus the vehicle speed

4.2 Results

For the simulations it is decided to investigate three different mode transitions. These are:

1. Clutch 2 closed from the beginning and clutch 1 closing during the simulation

2. Clutch 2 closed from the beginning, during the simulation clutch 2 opens while clutch 1 closes, after some time, clutch 2 closes again

3. Clutch 2 closed from the beginning, clutch 1 closes during the simulation. Then clutch 2 opens and after some time clutch 2 closes again

Some plots will be shown illustrating the above described mode transitions.

The first simulation is performed with constant throttle opening and CVT ratio. This simulation shows how the clutches close and how the speeds of the planetary gear gradually synchronize.

(a) Planetary gear

(b) Clutch 1 and clutch 2
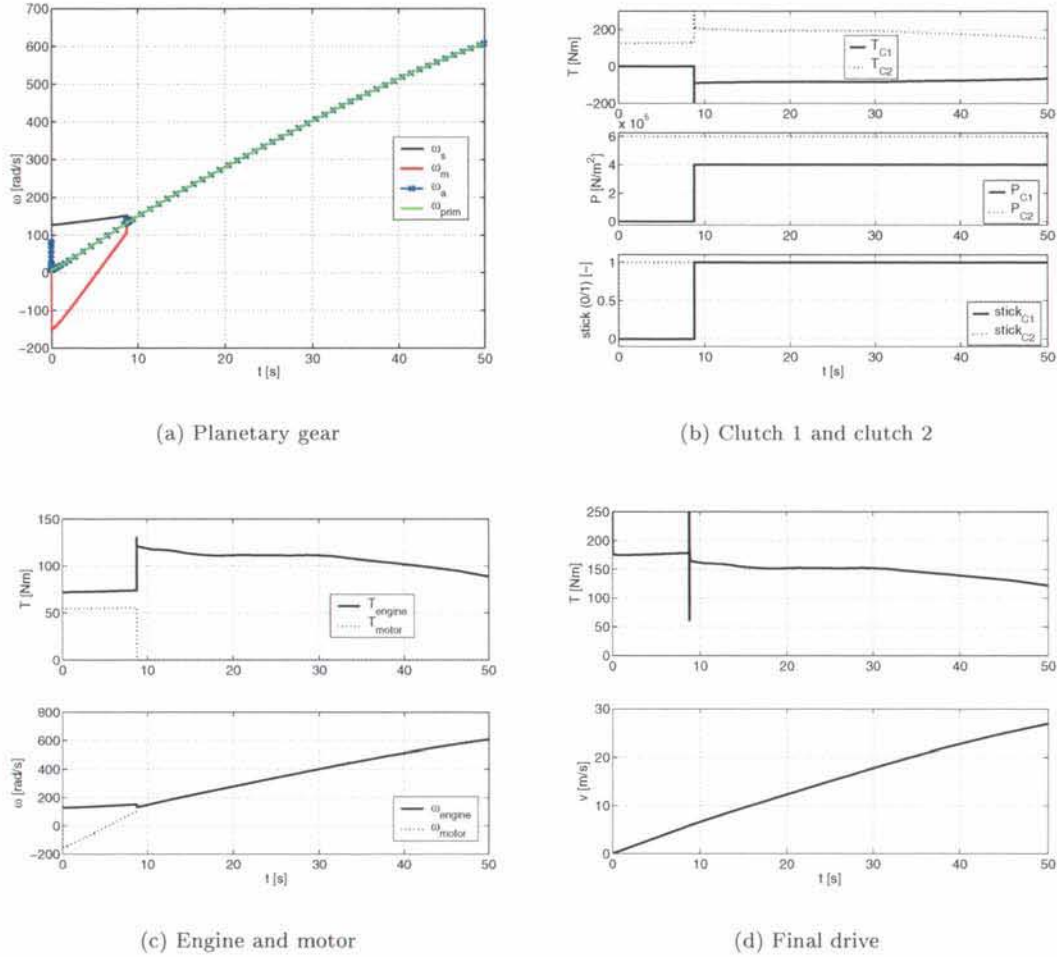
(c) Engine and motor

(d) Final drive

Figure 13: Simulation with $r_{CVT} = 1.412$ and the throttle is 50% open

The start of the simulation is when the vehicle is standing still, then a step on the throttle is given and the model starts in the electric torque converter mode. Both the engine and motor deliver power to drive the vehicle. Here can also be seen that the annulus and the primary CVT pulley have the same velocities. Since the annulus and primary CVT pulley are direct connected with each other because clutch 2 is closed. After a while the vehicle is launched to about $10km/h$ and the electric-motor is shut down, so the vehicle is only powered by the engine. This is realized by engaging clutch 1 and keeping clutch 2 closed, which is called the engine power mode. In engine power mode the electric motor does not give any power anymore, and the speeds of the engine and electric motor are the same. This was also expected because they are coupled with each other through the planetary gear, which members are now linked to each other as clutch 1 and 2 are both closed and thus rotate at the same velocities. The vehicles acceleration is not very high but this is caused by the constant CVT ratio, normally the CVT will begin in 'low gear' and gradually shift up to 'overdrive', i.e. here from 2.396 to 0.428. Controlling the CVT, however, is left for future or may be similar to the control applied in other CVT vehicles

Here the results are shown with almost the same starting conditions as the first simulation, the CVT ratio is the only difference. Now the CVT is set to overdrive, which means the vehicle uses the CVT ratio which is normally used when the vehicle is already at a reasonable speed. This can be compared like the sixth gear in a manual transmission.

(a) Planetary gear

(b) Clutch 1 and clutch 2

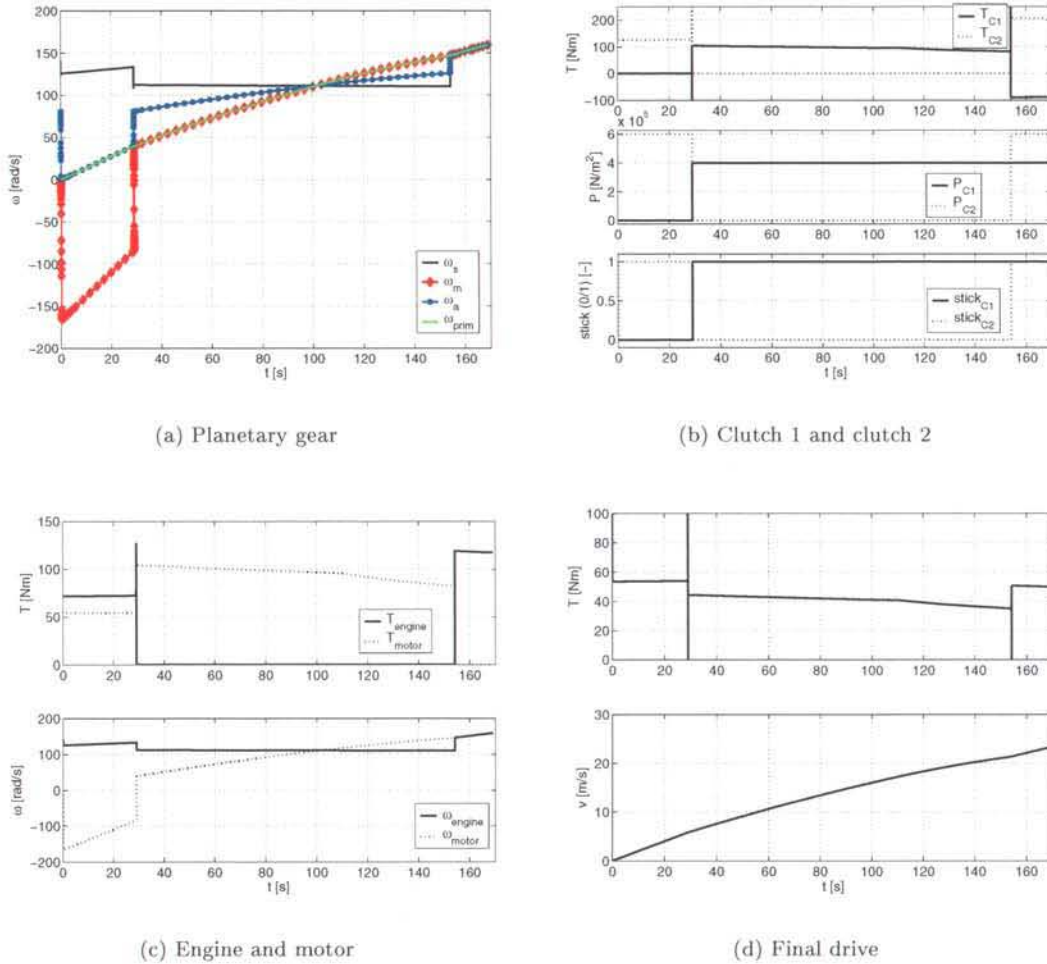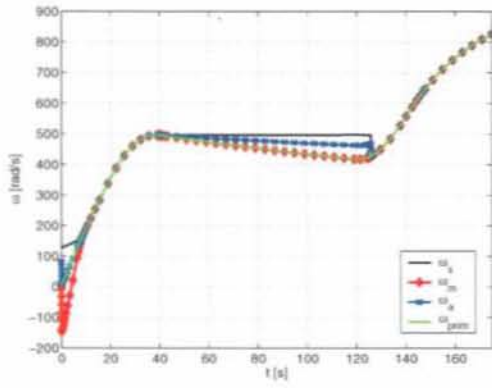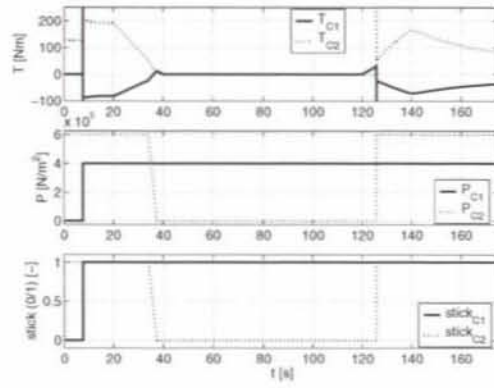(c) Engine and motor

(d) Final drive

Figure 14: Simulation with $r_{CVT} = 0.428$ and the throttle is 50% open

Here the vehicle is also standing still at the beginning of the simulation and a step on the throttle is given. The model starts in the electric torque converter mode, this mode is when a large driving force is needed and the vehicle speed is still low. Here again the annulus and primary CVT pulley are directly connected, so they have the same velocity. After a while the driving force is getting less and the model switches to EV power mode. Here the velocities of the electric motor and the primary CVT pulley are the same, this because clutch 1 is closed. In this mode the engine delivers no power. The speed is still increasing because the throttle is still open, after some time clutch 2 is engaged and the model is in engine power mode, here the electric motor delivers no power. Now the planetary gear has a ratio of 1, because both clutch 1 and 2 are closed. The annulus, carrier and sun have the same velocity. In this simulation the acceleration is worse then before, this is like expected because the CVT ratio is set to 'overdrive'.
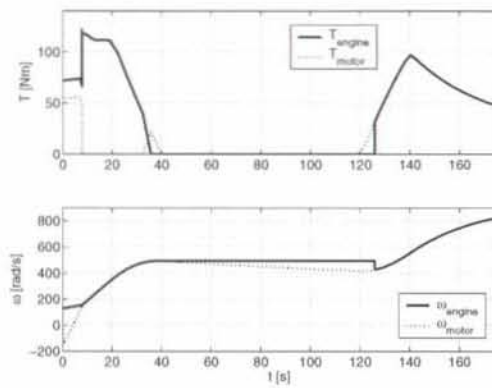
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The next simulation is initially the same as the simulation shown before, but the throttle opening is varied. The CVT ratio is set to 1.6088, which would provide the vehicle a higher acceleration from standstill. The simulation also shows how the model reacts on a period of less throttle during driving.
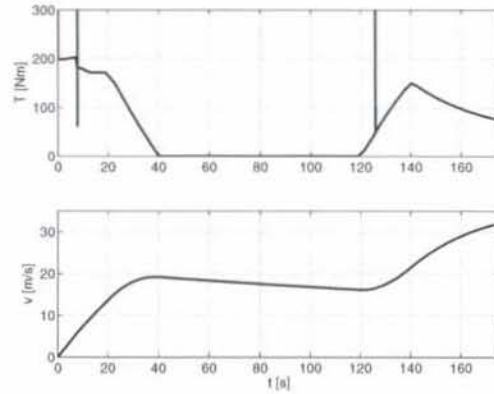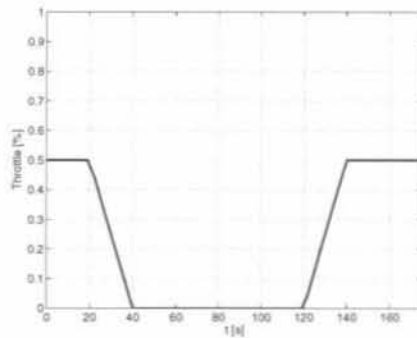

(a) Planetary gear

(b) Clutch 1 and clutch 2

(c) Engine and motor

(d) Final drive

(e) Throttle opening

Figure 15: Simulation with $r_{CVT} = 1.6088$ and variable throttle opening

Again the vehicle starts with clutch 2 closed, electric torque converter power mode. After a while clutch 1 closes, with clutch 2 still closed, engine power mode. Then there is no throttle given and the vehicle slows down and the driving force decreases quickly, clutch 2 opens and the vehicle gets in EV power mode. When throttle opening is resumed the vehicle starts to accelerate again and after a short time clutch 2 closes again to get in engine power mode. Now, the velocities of the engine and electric motor become quite high so normally the CVT would be shifted up to restrict the engine revolutions to its operational maximum. In this simulation it can be clearly seen that the vehicle is slowing down when the throttle is closed, this is due to air- and rolling-resistance.

4.3 ADVANCE

ADVANCE is a toolbox developed by TNO, this toolbox is made to help the development of intelligent powertrain and chassis systems. Because we want to implement the model into ADVANCE it is important to evaluate how Advance is built, and how it works. The global layout of ADVANCE is divided in six modules, i.e.:

1. Body

2. Chassis

3. Control

4. Driver

5. Powertrain

6. Test

This can also be seen in Figure 16 with the model tree on the left and the top level on the right of the model.
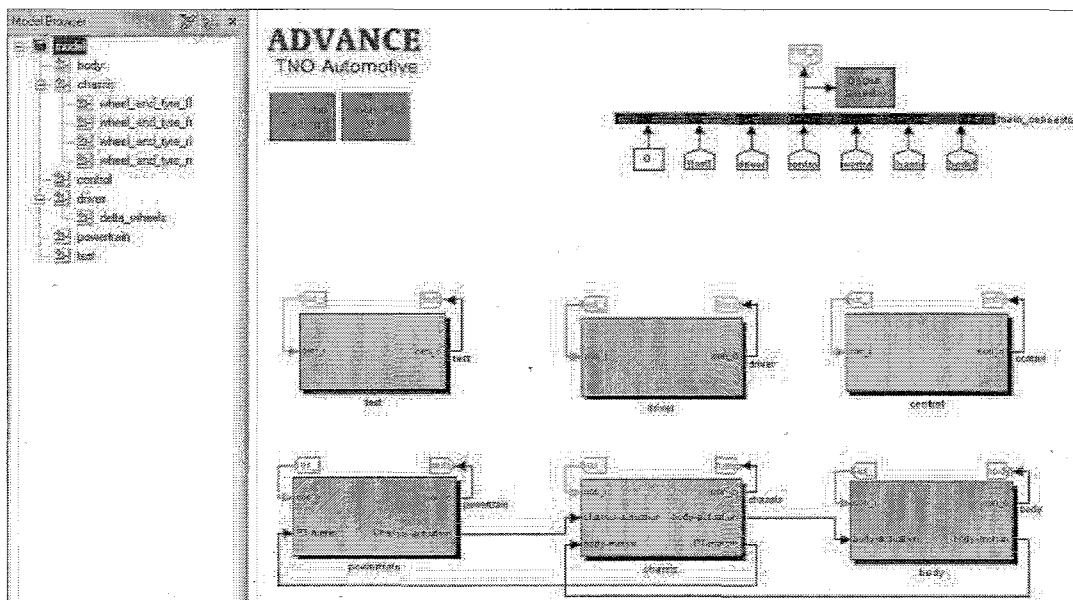

Figure 16: Top level in the ADVANCE model. [2]

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For each module TNO has made some models. The powertrain and control module have to be refurnished with the THS-C model. The standard parts can be used from ADVANCE, like the CVT, differential, axle and engine model. Some standard parts TNO has developed, like the tyre model and the driver, chassis, test and body module can also be used. But the parameters have to be adjusted to make these models resemble the Toyota Estima Hybrid. However the Toyota Estima Hybrid is only available on the Japanese market its hard to find all the parameters. The parameters of the 'normal' Estima or Previa, which is available on the European market, can be used for the major part. In appendix H an overview of all the parameters found can be seen.

5 Conclusions and recommendations

The main goal of this traineeship is to model the drive train of the Toyota Estima Hybrid. This is first done in Matlab Simulink, and after this it should be implemented into ADVANCE. But because of the time restrictions it was not possible to make the model fully operational in ADVANCE, and the goal was to model the front drive of the Toyota Estima Hybrid in Matlab Simulink. In chapter 4 the results of some simulations can be seen, from which we can conclude that the model displays a comprehensive representation of the real drive train. The clutches switch on at the desired time and the torques are representative. But some aspects are not modelled yet. For instance, the control strategy of the CVT is not modelled so there is chosen to set the CVT on a constant ratio.

Some recommendations are, to implement the rear motor generator together with regenerative braking, this will make the model having a closer resemblance to the real vehicle.
Another recommendation is to implement the model in ADVANCE and to use the already developed vehicle and chassis modules of ADVANCE. This because TNO build these modules and they are well tested with experiments. The hardest part may be to determine the right parameters which are needed in the ADVANCE modules. It is hard to find all the parameters of the Toyota Estima Hybrid, this because this vehicle is only sold in Japan and most of the information is in Japanese.
Also in Matlab Simulink there is new environment called 'SimDriveline', this has blocks to model a drive line, like inertias and torsional springs. With this toolbox it is simpler to make a model of a power train. It would also be nice to compare the simulations of the Toyota Estima Hybrid with simulations of the conventional Toyota Estima, so the difference between the hybrid configuration and the 'normal' configuration can be seen.

On the 74th Geneva international Motor Show [5] Lexus, which is a part of Toyota, presented a new model called the RX400h. This model makes use of the configuration as the Toyota Prius Hybrid. This is another hybrid configuration and is also designed to gain more fuel efficiency. The difference with the Estima is that this car will be sold in Europe, so it would be easier to gain more information about this car. This would make it easier to create a close matching total model. And if there are reasonable results from the complete vehicle model, some tests could be conducted to verify the simulations.

A Symbols

The symbols used stand for:

symbol	description	unit
P	power	$[W]$
T	torque	$[Nm]$
ω	rotational speed	$[rad/s]$
z	planetary gear ratio	$[-]$
r	radius or ratio	$[m]$
J	inertia	$[kgm^2]$
k	spring stiffness	$[N/m]$
b	damping stiffness	$[Ns/m]$ or $[Nsm]$
n	number of friction surfaces	$[-]$
p	control pressure	$[N/m^2]$
A	pressure surface	$[m^2]$
R	effective ratio	$[m]$
μ	tangential friction coëfficiënt	$[-]$

Table 3: Symbols

And the subscripts stand for:

symbol	description
s	sun
c	carrier
a	annulus
e	engine
td	torsional damper
m	motor/generator
$C1$	clutch 1
$C2$	clutch 2
$B1$	brake 1
$stick$	holds if the clutch/brake is closed
$prim$	primary pulley
$belt$	CVT belt
sec	secondary pulley
d	differential
CVT	CVT
kl	stiffness of the left rearaxle
kr	stiffness of the right rearaxle
l	left
r	right

Table 4: Explanation of subscripts

B Driving modes

(a) EV power

(b) Engine power and recharging battery
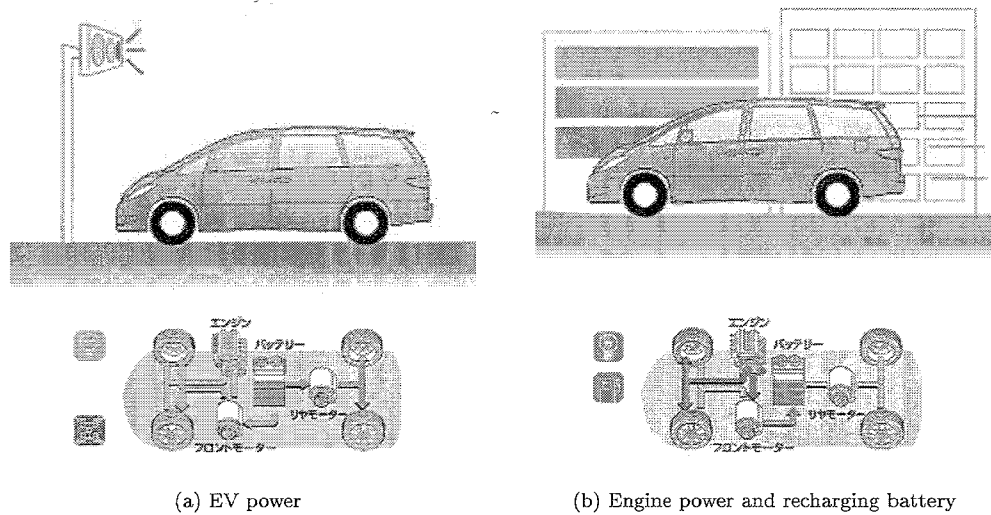
Figure 17: EV power & engine power and recharging battery mode, [9]

Figure 17(a) EV power, the vehicle is driven by the front and rear motors, 4WD. Figure 17(b) engine power and recharging battery mode, the vehicle is driven by the engine and the engine drives the front motor simultaneously which recharges the battery.

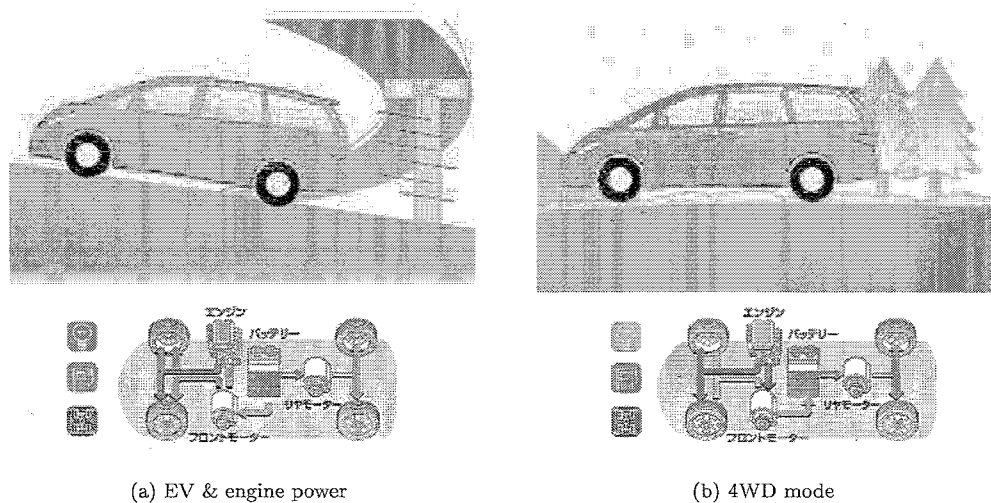(a) EV & engine power

(b) 4WD mode

Figure 18: EV power & engine power and 4WD mode, [9]

Figure 18(a) EV & engine power, the vehicle is driven by the engine and the front and rear motor, 4WD. Figure 18(b) 4WD mode, the vehicle is driven by the engine and the engine drives the front

motor simultaneously which recharges the battery. And the rear motor is used to drive the rear wheels, 4WD.

(a) Regenerative braking

(b) Electric torque converter power
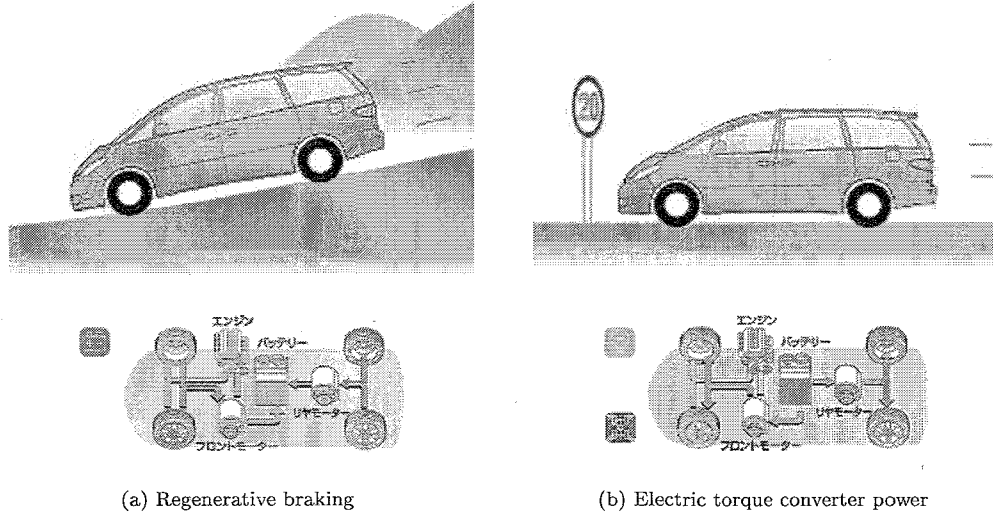
Figure 19: Regenerative braking & electric torque converter power, [9]

Figure 19(a) regenerative braking, the vehicle drives both the front and rear motor to recharge the battery. Figure 19(b) electric torque converter power, the vehicle is driven by the front and rear motor at low speeds when extra power is needed, 4WD.

C   Model
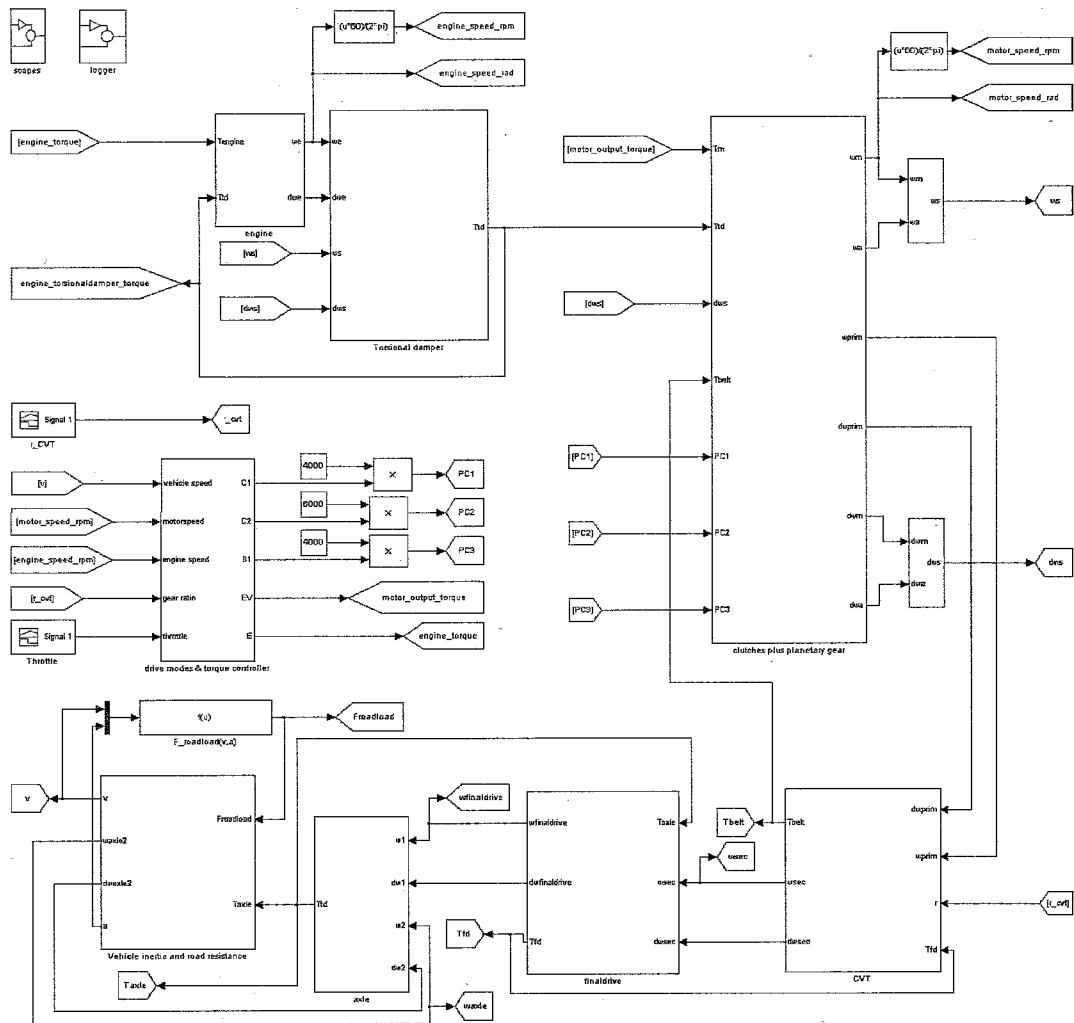
Figure 20: Model in Simulink

D Sub-models
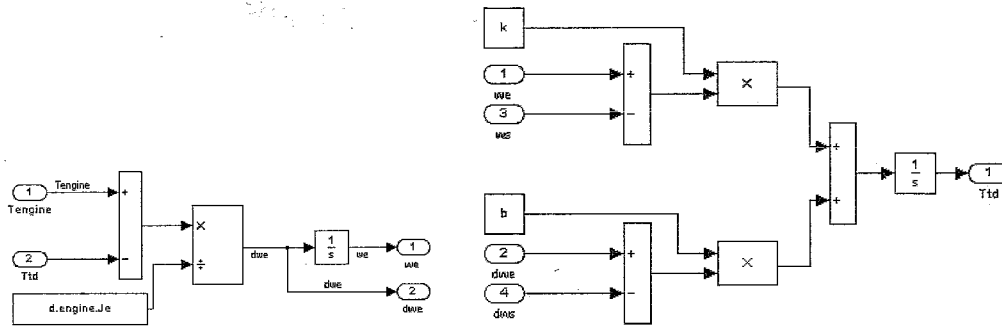
(a) Engine model

(b) Torsional damper model
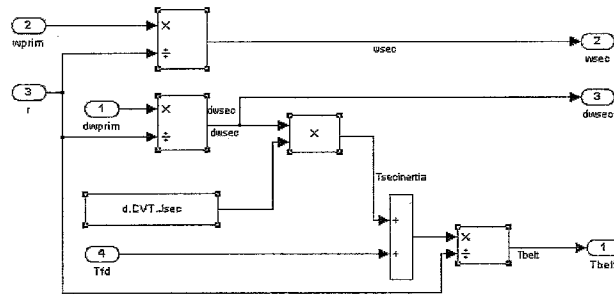
(c) CVT model
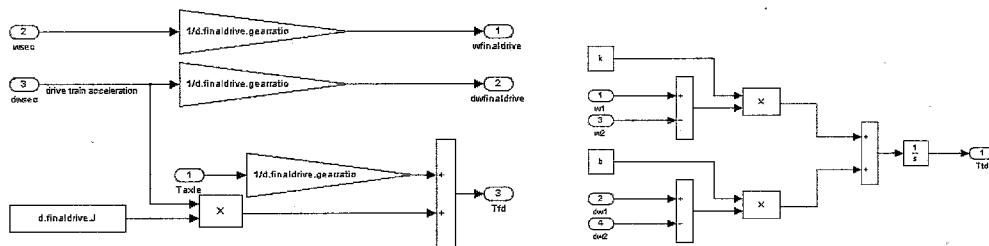
(d) Final drive model

(e) Axle model

Figure 21: Sub-models of the total model seen in appendix C

E   Drivetrain_data.m

```
% Drivetrain data:
% This file contains all the parameters for the drive train
% This file has to be run before running the model

d = [ ];

% Engine (2AZ-FXE):
d.engine.Je =   0.13;   % [kg(m^2)]
% Engine torque characteristics:
d.engine.w  = [ 0 1000 1100 1700 1900 2300 2800 3600 3700 4000 4800 5100 ...
              ... 5500 5700 ]; % [rpm]
d.engine.T  = [ 0    0  137  160  164  168  179  188  190  190  180  175 ...
              ... 167  160 ]; % [Nm]

% Motor torque characteristics:
d.motor.w   = [  0 1130 1240   1500 2000 2500 3000 3500 4000 4500 ...
              ... 5000  5600 ];  % [rpm]
d.motor.T   = [ 110  110 100.11 82.76 62.07 49.65 41.38 35.46 31.03 27.58 ...
              ... 24.82 22.16 ]; % [Nm]

% SUPERCVT (K110):
d.CVT.ktd   =   6e3;    % [N/m]
d.CVT.btd   =   9300;   % [Ns/m]
d.CVT.Jprim =   0.1;    % [kg(m^2)]
d.CVT.Jsec  =   0.1;    % [kg(m^2)]

% Clutch1:
d.clutch1.n        = 2 ;      % [-], number of friction surfaces
d.clutch1.A        = 0.5 ;    % [m^2], pressure surface
d.clutch1.R        = 0.127 ;  % [m], effective ratio
d.clutch1.mju      = 1 ;      % [-], tangential friction coefficient
d.clutch1.mjustick = 1.2 ;    % [-]
d.clutch1.epsilon  = 0.1 ;    % [-], small coefficient ...
                              ... to detect wether the clutch sticks or slips
d.clutch1.cst      = d.clutch1.n*d.clutch1.A*d.clutch1.R* ...
                     ... d.clutch1.mju;       % [-]
d.clutch1.cstick   = d.clutch1.n*d.clutch1.A*d.clutch1.R* ...
                     ... d.clutch1.mjustick;  % [-]

% Clutch2:
d.clutch2.n        = 2 ;      % [-]
d.clutch2.A        = 0.5 ;    % [m^2]
d.clutch2.R        = 0.127 ;  % [m]
d.clutch2.mju      = 1 ;      % [-]
d.clutch2.mjustick = 1.2 ;    % [-]
d.clutch2.epsilon  = 0.1 ;    % [-]
d.clutch2.cst      = d.clutch2.n*d.clutch2.A*d.clutch2.R* ...
                     ... d.clutch2.mju;       % [-]
d.clutch2.cstick   = d.clutch2.n*d.clutch2.A*d.clutch2.R* ...
                     ... d.clutch2.mjustick;  % [-]
```

```
% Brake1:
d.brake1.n          =   2;      % [-]
d.brake1.A          =   0.5;    % [m^2]
d.brake1.R          =   0.127;  % [m]
d.brake1.mju        =   1;      % [-]
d.brake1.mjustick   =   1.2;    % [-]
d.brake1.epsilon    =   0.1;    % [-]
d.brake1.cst        =   d.brake1.n*d.brake1.A*d.brake1.R* ...
                        ... d.brake1.mju;       % [-]
d.brake1.cstick     =   d.brake1.n*d.brake1.A*d.brake1.R* ...
                        ... d.brake1.mjustick; % [-]


% Planetary gear:
d.planetarygear.Js  =   0.01;       % [kg(m^2)]
d.planetarygear.Ja  =   0.01;       % [kg(m^2)]
d.planetarygear.Jm  =   0.01;       % [kg(m^2)]
d.planetarygear.z   =   -1.754 ;    % [-]


% Final drive:
d.finaldrive.gearratio  =   5.182;  % [-]
d.finaldrive.J          =   0.7;    % [kg(m^2)]
d.finaldrive.broad      =   1e3;    % [Ns/m]
d.finaldrive.kaxle      =   1e3;    % [N/m]


% Tyre diameter 205/65 R15:
d.tyre.diameter        =   2*0.65*0.205+0.0254*15; % [m]
d.tyre.radius          =   d.tyre.diameter/2;       % [m]
d.tyre.contourlength   =   pi*d.tyre.diameter;      % [m]
d.tyre.fr              =   d.vehicle.mass*d.roadload.g*d.tyre.froll; % [N]...
                           ... rolling resistance


% Vehicle parameters:
d.vehicle.mass      =   1850;   % [kg]
d.vehicle.Jeq       =   d.vehicle.mass*(d.tyre.diameter/2)^2;
d.vehicle.Cd        =   0.3;    % [-]
d.vehicle.wheelbase =   2.9;    % [m]
d.vehicle.frtrckwd  =   1.545;  % [m]
d.vehicle.rrtrckwd  =   1.530;  % [m]
d.vehicle.h         =   1.780;  % [m]
d.vehicle.A         =   d.vehicle.h*d.vehicle.frtrckwd; % [m^2]

d.drivemode.vETC    =   21/3.6; %[m/s]
d.drivemode.vEV     =   77/3.6; %[m/s]


% Air coefficients:
d.air.rho           =   1.293;  % [kg/m^3]
d.air.resist        =   0.5*d.air.rho*0.3*d.vehicle.A; % [N], air resistance
```

F Switchfunction.m

```
function [output] = switchfunctotappart(u,J1,J2,cst,cstick,eps);

% This function detects wether the clutch/brake sticks or slips.
% First it detects in what state the clutch/brake is and out of the input it
% calculates/decides if the clutch/brake switches or stays in the same state.

% Function inputs:
pressure    = u(1); % [N/m^2]
win         = u(2); % [rad/s]
wout        = u(3); % [rad/s]
Tin         = u(4); % [Nm], input torque
Tout        = u(5); % [Nm], output torque
TC          = u(6); % [Nm], transferred torque
stick       = u(7); % [-], wether the clutch/brake sticks or slips; 1/0

% Speed difference between in- and output:
delta_w     =   win-wout;   % [rad/s]

% If clutches are slipping the following interface torques apply:
% TC_slip   = cst*pressure*sign(delta_w);    % [Nm]

% If clutches are sticking the following interface torques apply:
% TC1_stick = (J2*(Tc+Tm)+J1*(Tbelt-TC2))/(J1+J2);  % [Nm]
% TC2_stick = (J2*(Ta-TB1)+Ja*(Tbelt-TC1))/(J2+Ja); % [Nm]
% TB1_stick = Ta; % [Nm]

Tclutchmax  = cstick*pressure*sign(TC);     % [Nm]
% Tslip      = cst*pressure*sign(delta_w);    % [Nm]
% if J2 > 0

% First detect if the clutch/brake is already closed:
if stick == 1, % if this is true, the transferred torque is calculated
    Tclutch3 = (J2*(Tin)+J1*(-Tout))/(J1+J2); % [Nm], transferred torque
else           % if the clutch/brake slips the transferred torque is different
    Tclutch3 = cst*pressure*sign(delta_w); % [Nm], transferred torque
end

% Now its examined if the clutch/brake switches or not:
if abs(delta_w) <= eps & abs(TC) < abs(Tclutchmax)
    TC  = Tclutch3;       % [Nm]
    dw1 = 1/J1*(Tin-TC); % [rad/s^2], input acceleration
    dw2 = dw1;            % [rad/s^2], output acceleration
    stick = 1;           % [-], the clutch/brake is closed
else
    TC  = Tclutch3;          % [Nm]
    dw1 = 1/J1*(Tin-TC);     % [rad/s^2]
    dw2 = 1/J2*(Tout+TC)*(J2>0); % [rad/s^2]
    stick = 0;               % [-], the clutch/brake is opened
end

% Function outputs:
output = [TC,stick,dw1,dw2];
```

G Drivemode.m

```matlab
function [output] = drivemode(u,vETC,vEV,tyrediameter,gearratio)

% This function detects in what kind of driving mode the vehicle is and which
% clutch/brake should be closed/opened.

% Function inputs:
speed       =   u(1);   % [m/s], vehicle speed
Tmotor      =   u(2);   % [Nm], torque delivered by the electric motor
Tengine     =   u(3);   % [Nm], torque delivered by the IC engine
R           =   u(4);   % total gear ratio
throttle    =   u(5);   % [-], throttle opening on a range of 0 to 1


Tmax    =   Tmotor + Tengine; % maximum possible torque
Tused   =   throttle*Tmax;    % used torque


if      speed <= vEV & Tused <= Tmotor; % EV power
    C1 = 1;
    C2 = 0;
    B1 = 0;
    EV = Tused;
    E  = 0;
elseif  speed <= vETC & Tused > Tmotor; % Electric torque converter power
    C1 = 0;
    C2 = 1;
    B1 = 0;
    EV = (Tused - Tmotor) / (Tmax - Tmotor)* ...
        ... (Tmotor - Tmotor / (gearratio / (1 + gearratio))) + ...
        ... Tmotor / (gearratio / (1 + gearratio));
    E  = -1 / (1 + gearratio)*EV;
elseif  speed > vETC & speed <= vEV & Tused > Tmotor & Tused <= Tengine;...
        % ... Engine power
    C1 = 1;
    C2 = 1;
    B1 = 0;
    EV = 0;
    E  = Tused;
elseif  speed > vETC & Tused >= Tengine; % EV & engine power
    C1 = 1;
    C2 = 1;
    B1 = 0;
    EV = (Tused-Tengine);
    E  = Tengine;
elseif speed > vEV & Tused < Tengine; % Engine power
    C1 = 1;
    C2 = 1;
    B1 = 0;
    EV = 0;
    E  = Tused;
end

% Function outputs:
output = [ C1,C2,B1,EV,E ];
```

H Toyota Estima Hybrid specifications

Car model number	ZA–AHR10W
Number of seats	7–8
Engine code	2AZ–FXE
Engine type	Otto, front transversal
Number of cylinders	4, in-line
Engine capacity	$2362cm^3$
Fuel	Hybrid, injected by cylinder
Supercharging	None
Bore	$88.5mm$
Stroke	$96mm$
Compression ratio	12.5
Maximum power	$96kW(JIS)5600rpm$
Maximum torque	$190Nm(rpm)4000rpm$
Drive	front-wheel drive
Gearbox	4 auto
Steering type	Hydraulic power
Wheels	$6.55JJ \times 15$
Front tyres	$205/65R15$
Rear tyres	$205/65R15$
Front suspension	MacPherson strut, coil spring, stabilizer bar
Rear suspension	Coil spring, torsion beam, stabilizer bar
Wheelbase	$2900mm$
Length	$4770mm$
Width	$1790mm$
Height	$1780mm$
Track (front)	$1545mm$
Track (rear)	$1530mm$
Curb weight	$1850-860kg$
Trunk	$495l-2370l$
Fuel tank	$75l$
Average consumption	$5.6l/100km(work)$
Front motor/generator	1EM
voltage	$216V$
Maximum power	$13kW(1130-3000rpm)$
Maximum torque	$110Nm(0-1130rpm)$
Rear motor/generator	1FM
voltage	$216V$
Maximum power	$18kW(1910-2500rpm)$
Maximum torque	$108Nm(0-400rpm)$

Table 5: Specifications of the Toyota Estima MPV 2.4 16v Hybrid

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