A Fuzzy Logic Controller for a Hydrostatic Transmission for an Electric Hybrid Bus in Bogota

Presented as a partial requirement for a master degree in mechanical engineering by

Jorge Andres Leon Quiroga

Advised by

Andres Leonardo Gonzalez Mancera, PhD, main advisor
Universidad de los Andes

Jose Manuel Garcia, PhD, co – advisor
Purdue University

Universidad de los Andes
Faculty of Engineering
Department of Mechanical Engineering
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Agradecimientos.

Quiero agradecer a mis padres por su apoyo incondicional y porque desde siempre han sido los mejores maestros que he tenido. Quiero agradecer también a mis hermanas porque siempre han sido y serán las mejores amigas que tendré. Quiero agradecer también a todas las personas que me han acompañado durante mi camino y que siempre han estado dispuestos a apoyarme. En especial quiero agradecer a María Paula Mejía por su invaluable apoyo y compañía.

1. Content

2. Abstract

The control strategy in a hydraulic hybrid vehicle is very important for taking advantage of its full potential. In a hydraulic hybrid vehicle, the hydrostatic transmission can be controlled in order to optimize the entire system. A hydrostatic transmission is composed basically of a hydraulic motor and a hydraulic pump. For an electric hybrid vehicle an electric motor is the prime mover. In this study the control variables are the displacements of the hydraulic devices and the objective is to optimize the system efficiency. All possible combinations of displacements were tested through simulation and the most efficient combination was recorded. Around 9% of the energy can be saved using the most efficient configuration of the hydrostatic transmission. Fuzzy logic theory was used to simulate the driver input and proposed as a control strategy for the hydrostatic transmission.

3. Introduction

The continuous diminishing of oil reserves and the increasing demand of energy is a worldwide problem. Nowadays there are many alternatives for supplying energy such as renewable energy, nuclear energy, biofuel energy, etc. Another option for reducing the impact of gas emissions is improving the efficiency of the systems and devices that are already in use. Any improvement in the efficiency of devices can be significant and can have a global impact. In 2013 in the United States the transportation sector consumed 69% of all petroleum. So, improving the efficiency of the transportation systems can have a huge impact in the global energy consumption [1].

Due to recent concern about reducing emissions in traditional combustion engine vehicles, electric vehicles have been considered in the last years as an alternative for replacing the traditional vehicles. Electric vehicles use an electric motor as the prime mover and flywheels, fuel cells, ultracapacitors and batteries as energy sources. The advantages of electric vehicles over internal combustion engine vehicles are the high efficiency, smooth and quiet operation, no emissions and independence of petroleum based fuels [2].

Powertrain hybridization trough hydraulic transmission is one of the available options for improving efficiency in electric car [3]. On-road hybrid vehicle market is dominated by electric hybrids, however hydraulic hybrids have many benefits which make them a competing technology. It has been demonstrated that hydraulic hybrid transmissions increase 29% the fuel efficiency over the most efficient electric hybrid and 47% over an identical non-hybrid bus [4]. Moreover, the lifecycle cost of the hydraulic hybrid is 24% less than a conventional diesel bus and 36% less than an electric hybrid bus. Hydraulic hybrids use hydraulic accumulators as energy storage devices. Accumulators have a longer lifecycle than batteries and the batteries need to be replaced one or more times over their lifetime depending on application [5].

Hydraulic hybrid transmissions can be grouped into three main categories: parallel hybrids, series hybrids and hybrid power split transmissions. Each of them have benefits and drawbacks. These systems have been installed and studied in vehicles such as delivery trucks [6], refuse collection vehicles [7] and pickup trucks [3] with satisfactory results. The focus of this work is on series hybrids because of the simplicity of the system, ease of installation compared with other hydraulic systems, and because of its stiff nature that enables rapid changes in system pressure which is very important for satisfying the driver demands [8].

The control strategy of the system is very important in order to take advantage of its full potential. Some of the controllers developed until now has been designed for internal combustion engines and hybrid electric vehicles [9] [10]. Load leveling is a methodology in which the internal combustion engine is forced to operate around its most efficient point. This methodology has demonstrated great potential. However, in diesel engines the most efficient operating point is near to the peak torque curve, which is the region of maximum fuel consumption [11].

Fuzzy logic control is a convenient way to control hybrid vehicles. This has been demonstrated in previous research at Oakland University [12] [13] and Ohio State University [14] [15]. Some studies have focused on hybrid electric vehicles and their principal concern has been the power management strategy. That is the magnitude and rate at which the power needed by the car is delivered to the wheels [11]. Another study is focused on optimizing all components of the system in order to obtain the most efficient combination of variables [16]. Also, dynamic programming has been used to control hybrid hydraulic vehicles with good results [17]. An instantaneous optimization is made in dynamic programming. This method allows to compare in a fair way many options of hydrostatic transmissions [18].

In this work, a fuzzy logic control is proposed as a control strategy for a hydrostatic transmission in an electric bus for the Integrated Public System of Transportation (SITP) in Bogota, Colombia. A numerical model was developed in order to choose the elements of the system. These elements were selected in order to provide the vehicle requirements for a typical urban driving cycle. After that, a fuzzy logic control was used to model the human response and it was proposed as a control strategy for the vehicle.

4. Objectives

Main objective

Evaluate the performance of a fuzzy logic controller for a hydrostatic transmission for an electric hybrid vehicle in Bogota, Colombia.

Specific objectives

- Study and specify the operation conditions, requirements and constraints of the problem.
- Design the hydrostatic transmission.
- Design the control strategy.
- Evaluate the performance of the system and the control strategy through simulation.

5. Vehicle model

In this section, the vehicle model is discussed and explained. First, powertrain architecture is explained and modelled. All possible alternatives of use of the hydraulic system are discussed. After that, the vehicle dynamic model is explained.

5.1 Powertrain architecture

A non-conventional powertrain architecture is studied in this work. This powertrain architecture has been studied in a previous work [19]. The transmission and the driveshaft of a conventional vehicle is replaced for a hydraulic transmission. A schematic representation of the powertrain is shown in figure 5.1.

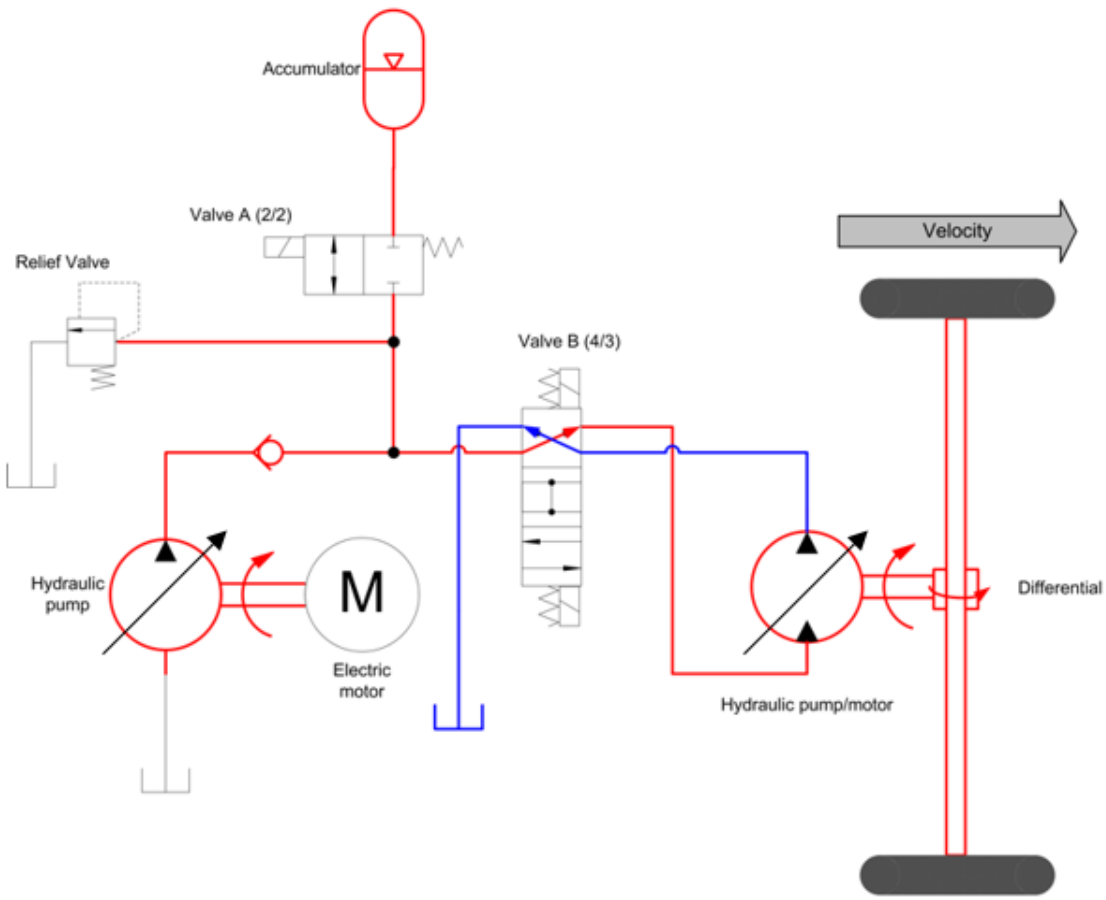

Figure 5.1. Schematic representation of the powertrain.

The system is composed by an electric motor, which is the prime mover, a variable displacement hydraulic pump, a variable displacement hydraulic pump/motor and one accumulator. Moreover, there are three valves which permit or restrict the flow in order to use the system in a convenient way. Also, a relief valve is coupled to the system for preventing pressure overload.

This system can work in three ways: as a hydrostatic transmission, as a recovery braking energy system and as a system propelled by the energy stored in the accumulator. Each of these operation modes are explained next.

5.1.1 Hydrostatic transmission

The hydraulic system can be used as a hydrostatic transmission in order to transmit power from the electric motor to the wheels. The electric motor is the prime mover and the pump moves the fluid from the reservoir to the hydraulic pump/motor which in this case is working as a motor. For this process valve A is closed while valve B is in the higher position (see figure 5.1). The flow through the hydraulic motor generates the power to move the vehicle. Equations 5.1 to 5.3 are the equilibrium equations for each component of the hydrostatic transmission.

Electric motor	$I_{EM}\dot{\omega}_{EM} = T_{EM} - T_p \qquad (5.1)$
Hydraulic pump	$I_p\dot{\omega}_p = T_p - M_p \qquad (5.2)$
Hydraulic motor	$I_m\dot{\omega}_m = M_m - T_m \qquad (5.3)$

Where I is the mass inertia of the component, T is the torque delivered by the component, $\dot{\omega}$ is the angular acceleration and M is the hydraulic torque. The subscripts EM, p and m indicate electric motor, hydraulic pump and hydraulic motor respectively. Hydraulic torque in the pump and the motor are calculated with equations 5.4 and 5.5.

$$M_P = \Delta P D_p / \eta_{m,p} \qquad (5.4)$$

$$M_m = \Delta P D_m \eta_{m,m} \qquad (5.5)$$

Where ΔP is the pressure drop in the system, D is the displacement and η_m is the mechanical efficiency. Again, subscripts p and m indicate hydraulic pump and hydraulic motor respectively. Moreover, the flow rate in the pump and the motor is given by equations 5.6 and 5.7 respectively.

$$Q_p = \omega_p D_p \eta_{v,p} \qquad (5.6)$$

$$Q_m = \omega_m D_m / \eta_{v,m} \qquad (5.7)$$

Where ω is the angular velocity and η_v is the volumetric efficiency.

5.1.2 Regenerative braking energy system

When the vehicle is braking the kinetic energy, that in a conventional vehicle is lost as a heat in the brakes, can be used to move flow through the hydraulic pump/motor to a hydraulic accumulator. This energy can be used later in order to propel the vehicle. In order to do this, valve A is open and valve B is placed in its lower position. At this time, the hydraulic motor is turned into a pump which moves fluid from the reservoir to the accumulator. This process is represented in figure 5.2.

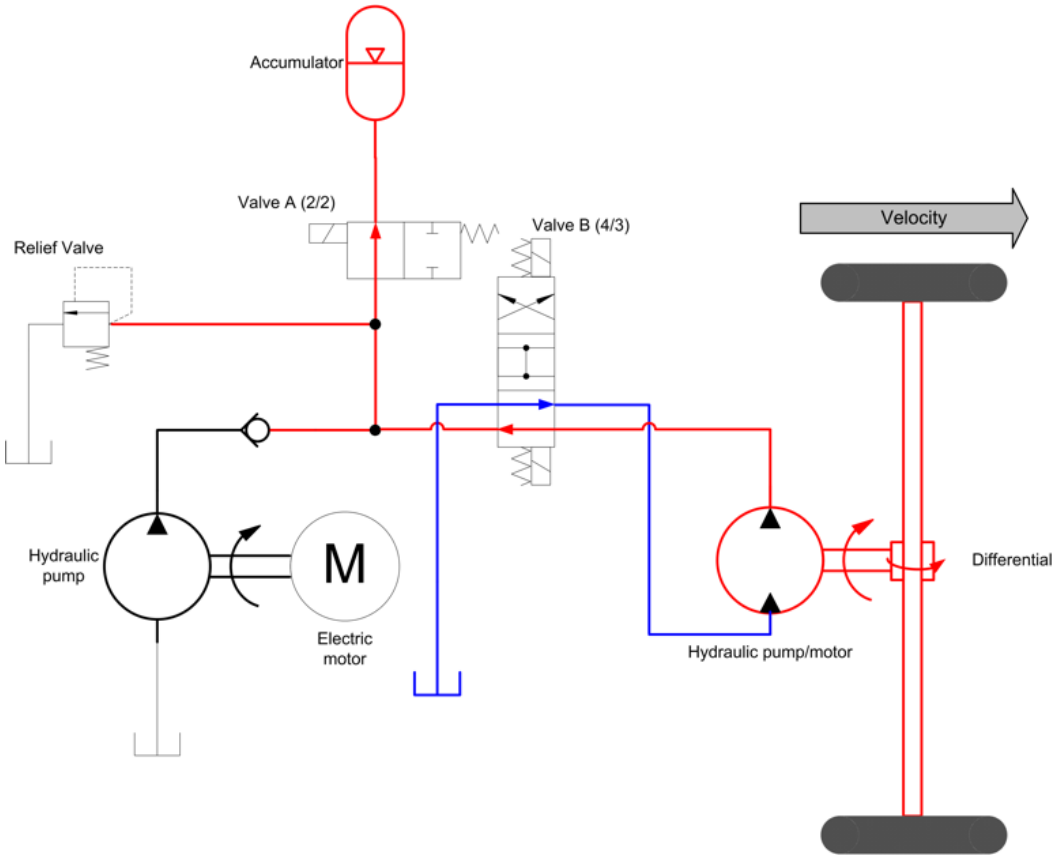

Figure 5.2. Hydraulic system in regenerative braking process.

The accumulator can be modelled as a system in adiabatic expansion or compression. For nitrogen accumulators the n constant is assumed to be 1.4. The pressure and volume in the accumulator can be related by equation 5.8 assuming an isothermal process.

$$P_i V_i^n = P_f V_f^n \qquad (5.8)$$

5.1.3 Accumulator discharge

The energy stored in the accumulator can be used for launching the vehicle. In a vehicle without energy recovery system, the energy used for this purpose is drawn from the battery and given to the pump by the electric motor. In order to use the energy stored in accumulator,

valve A is opened and valve B is placed in the higher position. The hydraulic pump/motor works as a motor in this process, which is represented in figure 5.3

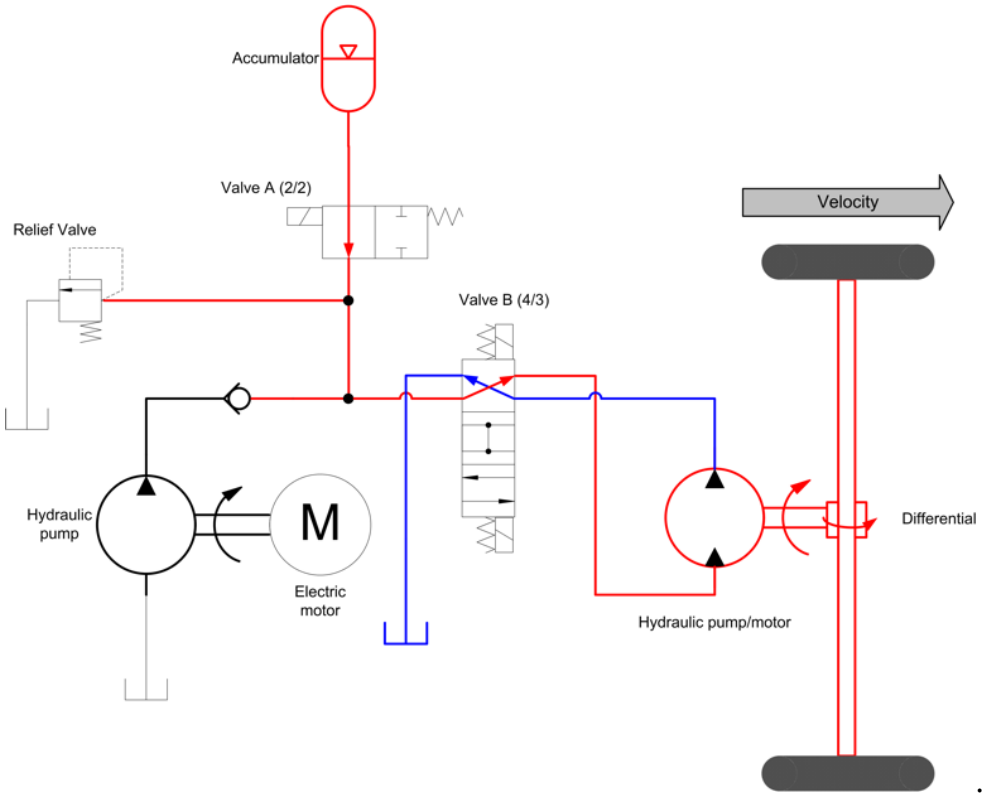

Figure 5.3. Hydraulic system in discharge process.

5.2 Dynamic model

A one degree of freedom vehicle dynamic model was developed in order to study vehicle dynamic. If the car is taken as a control volume, the forces acting over it are shown in figure 5.4.

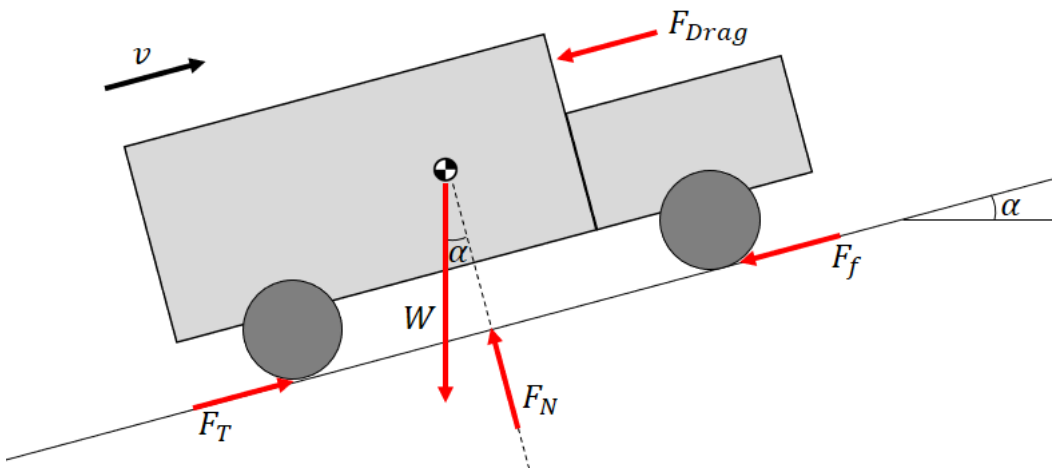

Figure 5.4. General free body diagram for the car.

Where F_{Drag} is the drag force, F_f is the friction force, F_N is the normal forces on the wheels, α is the inclination angle of the road, vector v shows the direction of movement, W is the vehicle weight and F_T is the tractive force. All the forces mentioned act on the car for any kind of powertrain. The equilibrium equations are presented in equations 5.9 and 5.10.

$$ma = F_T - F_f - W \sin\alpha - F_{Drag} \qquad (5.9)$$

$$F_N - W \cos\alpha = 0 \qquad (5.10)$$

Equation 5.9 can be solved in two ways. The first way is solving the equation for calculate acceleration and integrate it in order to calculate velocity. The second way is using a driving cycle for calculate acceleration and after that, traction force can be calculated in order to solve dynamics of the powertrain. The second way is the used in this work. Acceleration and velocity are known for every instant in a driving cycle, so traction force can be calculated from equation 5.9. The traction force is applied on the wheels and it is the final result of the power transmission. A free body diagram of the wheel is shown in figure 5.5 and the equation 5.11 is the dynamic equation for the wheel.

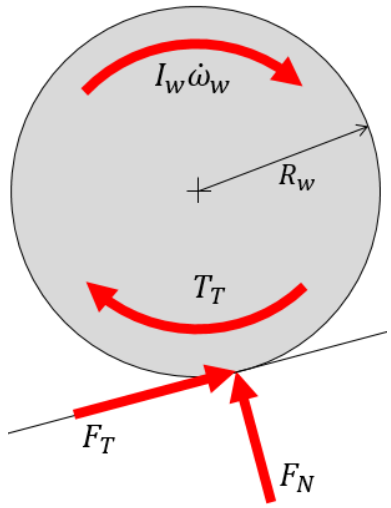

Figure 5.5. Free body diagram for the wheel.

$$I_w\dot{\omega}_w = T_T - F_T R_w \qquad (5.11)$$

Where I_w is the mass inertia of the wheel, $\dot{\omega}_w$ is the angular acceleration, T_T is the traction torque and R_w is the wheel radius. With equations 5.9 and 5.11 and the driving cycle, the acceleration, velocity, tractive force and tractive torque can be calculated.

The equations presented here are the same for a common commercial vehicle with an internal combustion engine and mechanical powertrain. The principal differences between a common vehicle and the one studied in this work is in the powertrain. In a common commercial vehicle, the powertrain is composed by mechanical gears that transmit the power from the engine to the wheels in a convenient manner, while in a hybrid hydraulic vehicle the power transmission is through the components of the hydrostatic transmission. The relationships between displacements of the hydraulic components allow to transmit power as the mechanical transmission.

5.3 Coupled model

Dynamic and hydraulic models must be solved in a coupled way. The equations that model the entire system are presented again here:

Electric motor	$I_{EM}\dot{\omega}_{EM} = T_{EM} - T_p \qquad (5.1)$
Hydraulic pump	$I_p\dot{\omega}_p = T_p - M_p \qquad (5.2)$
Hydraulic motor	$I_m\dot{\omega}_m = M_m - T_m \qquad (5.3)$

Hydraulic torque in the pump	$M_P = \Delta P D_p / \eta_{m,p}$ (5.4)
Hydraulic torque in the hydraulic motor	$M_m = \Delta P D_m \eta_{m,m}$ (5.5)
Flow rate in the pump	$Q_p = \omega_p D_p \eta_{v,p}$ (5.6)
Flow rate in the hydraulic motor	$Q_m = \omega_m D_m / \eta_{v,m}$ (5.7)
Dynamic equation for the vehicle	$ma = F_T - F_f - W \sin\alpha - F_{Drag}$ (5.9)
Dynamic equation for the wheel	$I_w \dot{\omega}_w = T_T - F_T R_w$ (5.11)

In order to solve these equations, some assumptions must be taken into account:

1. Flow rate out of the pump is the same entering to the hydraulic motor, so $Q_p = Q_m$
2. Pressure drop in the pump and the hydraulic motor is the same.
3. There is no slip in the tires.

With the first assumption ($Q_p = Q_m$), a relationship between velocity in the hydraulic motor and velocity in the pump can be made.

$$\omega_p = \omega_m \frac{D_m}{D_p} \frac{1}{n_{v,m} n_{v,p}} \qquad (5.12)$$

With the third assumption a relationship between car velocity and wheel velocity can be made. A similar relationship can be made for acceleration.

$$\omega_w = V / R_w \qquad (5.13)$$

$$\dot{\omega}_w = a / R_w \qquad (5.14)$$

With equations 5.13 and 5.14, motor angular velocity and angular acceleration can be calculated.

$$\omega_m = \omega_w N_{dif} \qquad (5.15)$$

$$\dot{\omega}_m = \dot{\omega}_w N_{dif} \qquad (5.16)$$

With a similar equation, torque in the hydraulic motor can be calculated as a function of the traction torque in the wheel.

$$T_m = T_T / N_{dif} \qquad (5.17)$$

Pump angular acceleration is given by equation 5.18: $\dot{\omega}_p = d\omega_p / dt$ (5.18)

Between the electric motor and the hydraulic pump, it is necessary to have a gear. The relationship between hydraulic pump velocity and electric motor velocity is given by equation 5.19. In a similar way, with equation 5.20 the electric motor angular acceleration can be calculated.

$$\omega_{EM} = \omega_p N_{dif,p} \qquad (5.19)$$

$$\dot{\omega}_{EM} = \dot{\omega}_p N_{dif,p} \qquad (5.20)$$

The method used for solving the system of equations is described next.

1. From the driving cycle, acceleration and velocity are determined.
2. From equation 5.9 tractive force is calculated.
3. From equations 5.13 to 5.16 angular velocity and acceleration of the wheels and the hydraulic motor are determined.
4. Tractive torque in the wheels are determined by equation 5.11.
5. Torque in the hydraulic motor can be solved from equation 5.17.
6. From equation 5.3, hydraulic torque (M_m) in the hydraulic motor is calculated.
7. Pressure drop in the system is calculated from equation 5.5.
8. From equation 5.4, hydraulic torque (M_p) in the hydraulic pump is calculated.
9. Pump angular velocity is calculated from equation 5.12.
10. Pump angular acceleration is calculated with a discretized approximation of equation 5.18. $\dot{\omega}_p = (\omega_i - \omega_{i-1}) / \Delta t$
11. From equation 5.2, torque in the hydraulic pump is calculated.
12. Electric motor angular acceleration is calculated with equation 5.20.
13. From equation 5.1, torque in the electric motor is determined.

The method described is used to select the hydraulic components of the hydrostatic transmission. With this method it is possible to determine the energy consumption of the hydrostatic transmission for a determined driving cycle. Moreover, it is possible to determine the influence of some variables related with the hydraulic components as their mass inertia and displacements.

6. Case of study

The case of study of this work is a commercial passenger bus Chevrolet NKR Reward Euro IV which is a typical bus of the Integrated Public System of Transportation (SITP) in Bogota. This vehicle uses an internal combustion engine as a prime mover. An example of this bus is presented in figure 6.1.

Figure 6.1. Urban route of SITP

The principal characteristics of this bus are described in table 6.1.

Power (hp @ rpm)	122 @ 2600
Torque (Nm @ rpm)	353 @ 1500
Differential ratio	5.571
Traction	4x2
Wheels	205/75 R17.5
Gross weight (kg)	5600
Load capacity (kg)	3672

Table 6.1 Vehicle specifications.

The driving cycles are widely used for estimation of fuel consumption and other performance characteristics in vehicles. As a reference, an urban driving cycle of an articulated bus of Transmilenio was used [20]. The driving cycle is shown in figure 6.2.

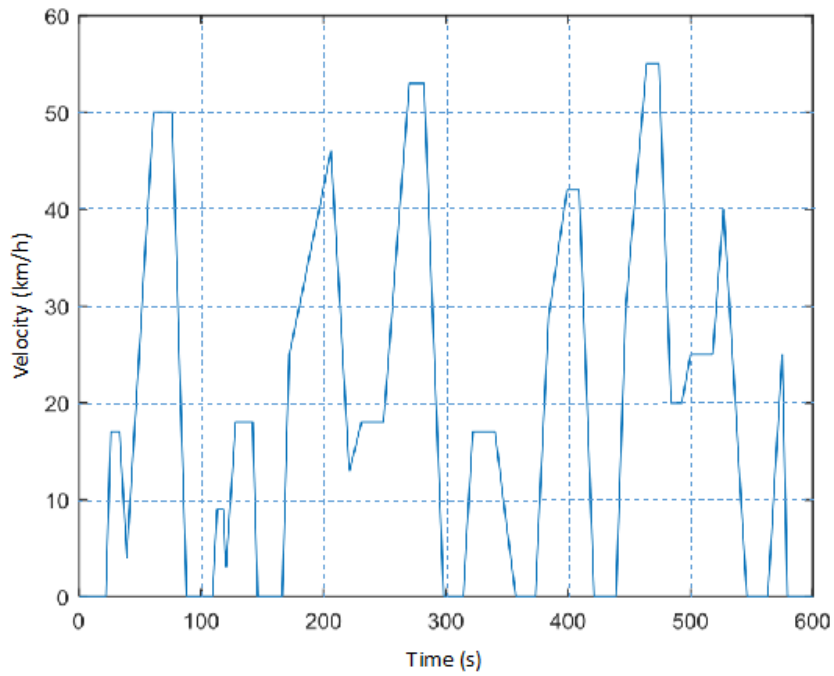

Figure 6.2. Urban driving cycle [20].

The equations written in chapter 5 were used to select the components of the system. First, hydraulic motor was selected in order to satisfy the torque and angular velocity requirements of the driving cycle presented in figure 6.2. With the driving cycle it is possible to have the velocity and acceleration, with equations 5.9 and 5.11 the tractive force and the tractive torque can be calculated. The hydraulic motor must be able to satisfy this requirement of tractive torque and angular velocity because it is directly attached to the wheels through the differential. Many hydraulic motors were compared. The comparison between limits of the hydraulic motors and requirements of the vehicle for the driving cycle chosen as a reference are presented in figure 6.3.

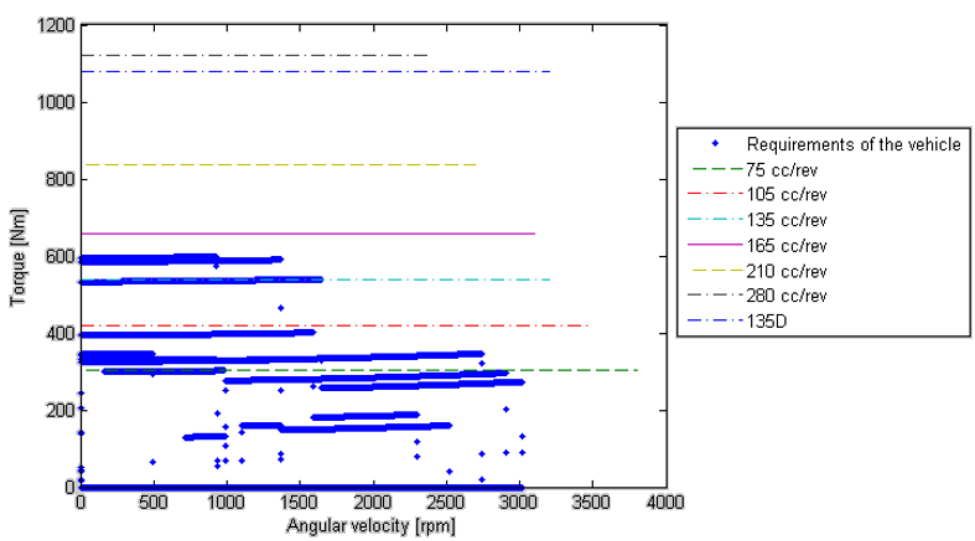

Figure 6.3. Comparison of hydraulic motors.

From the figure it is possible to determine that the motor of 165 cc/rev is capable of satisfying the requirements. The principal characteristics of this hydraulic motor are described in table 6.2.

DuraForce HVM 165	
Max. Continuous speed (rpm)	3100
Nominal pressure (bar)	420
Torque (Nm)	659
Power (kW)	214
Weight (kg)	75
Displacement (cc/rev)	165
Mass of inertia ($kg \cdot m^2$)	0.0306

Table 6.2. Hydraulic motor specifications.

In a similar way, requirements of the vehicle were compared with capabilities of many hydraulic pumps. In figure 6.4, this comparison is shown for the pump chosen.

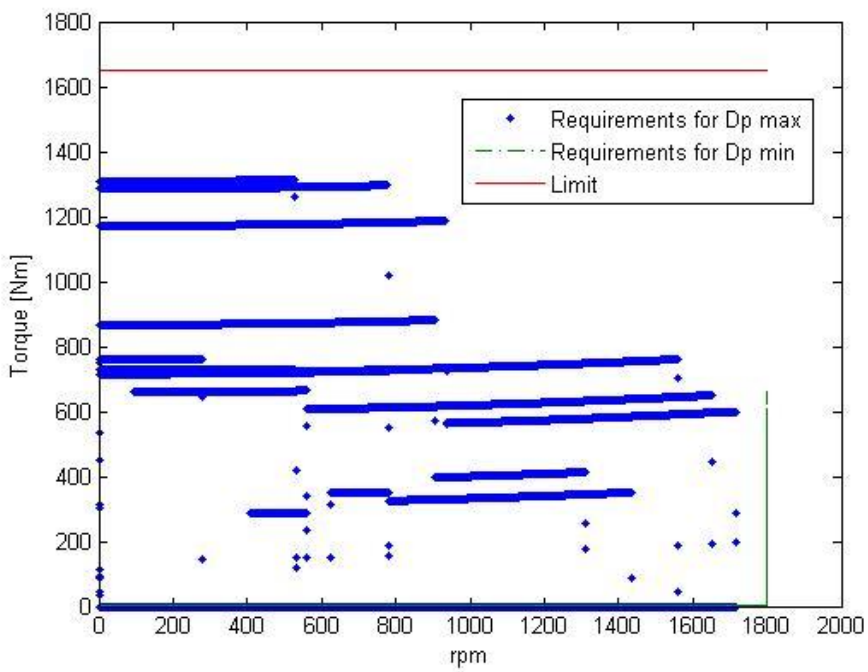

Figure 6.4. Comparison of hydraulic pump limits.

From this figure it is possible to see that the pump is able to satisfy the requirements of the vehicle even when it works with either its maximum or minimum displacement. So, this pump can be used as a variable displacement pump. Using the pump as a variable displacement device, allows to have a continuous variable transmission, which can be used in order to adjust the torque and angular velocity in the prime mover for doing it to work more efficiently. The principal characteristics of the hydraulic pump are described in table 6.3.

PVW 250	
Max. Continuous speed (rpm)	1800
Nominal pressure (bar)	350
Torque (Nm)	1670
Power (kW)	315
Weight (kg)	212
Displacement (cc/rev)	250
Mass of inertia ($kg \cdot m^2$)	0.146

Table 6.3. Hydraulic pump specifications.

Now, an electric motor must be selected in order to be the prime mover of the system. The motor selected was a REMY HVH 250, which is widely used for electric cars. The principal characteristics of this motor are shown in table 6.4.

REMY HVH 250	
Max. speed (rpm)	10500
Max. torque (Nm)	422
Max. power (kW)	151
Weight (kg)	57.2
Mass of inertia ($kg \cdot m^2$)	0.086

Table 6.4. Electric motor specifications.

This electric motor is capable of satisfying the requirements of the system for the driving cycle chosen as a reference (figure 6.2). A comparison between requirements of the system and motor limits are shown in figures 6.5 and 6.6. These figures show the requirements when the pump is working with either its maximum or minimum displacements.

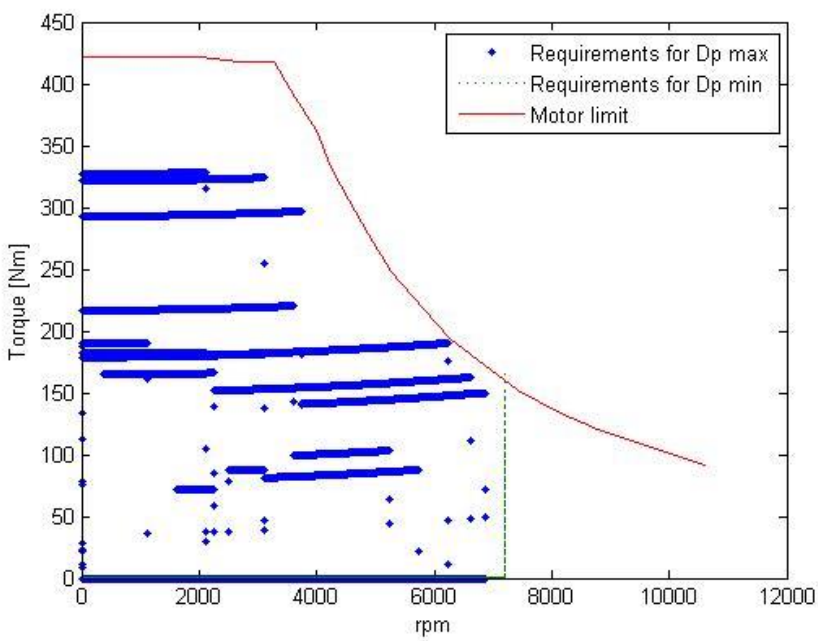

Figure 6.5. Comparison between system requirements and motor torque limit.

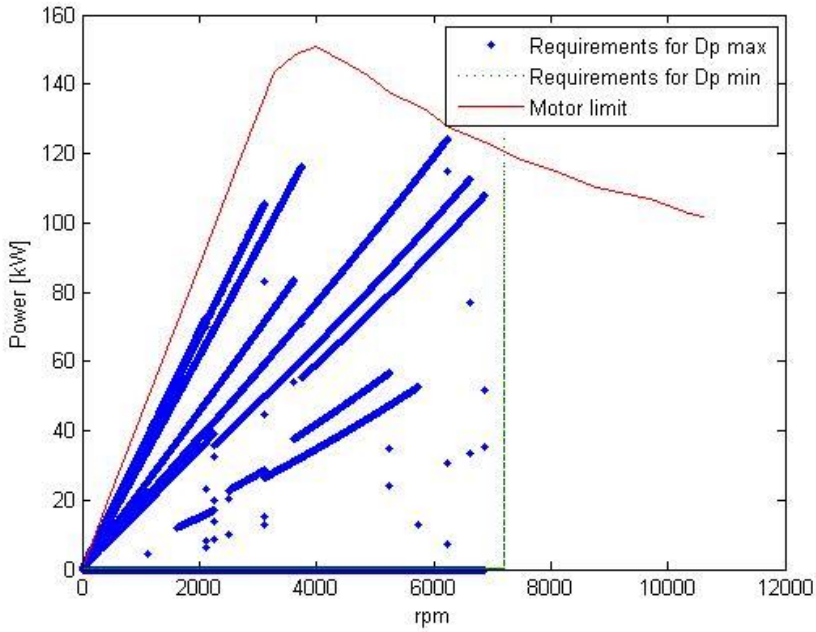

Figure 6.5. Comparison between system requirements and motor power limit.

The devices selected are capable of satisfying the requirements of the vehicle for a typical driving cycle. Moreover, because of the possibility of a continuous variable transmission, the system can be optimized at every time in order to rise the efficiency and diminish the spent of energy.

7. Control strategy

The control strategy of the system is very important for taking advantage of its full potential. It is important to optimize all the system and not a particular device of the system. Also, it is important that the control system has as an input signal something that can be measured easily and related with the driver intention as the percentage of compression of the pedals. In a conventional internal combustion engine vehicle, the driver adjusts the torque of the motor with the pedals in order to reach a desired velocity and acceleration. The objective of the control system is to optimize the system every time, whatever can be the driver desire. In other words, the only concern of the driver must be maintaining the desired velocity and the control system must optimize the entire system.

7.1 Study of the system efficiency

The final purpose of any powertrain is to transfer the power from the engine to the wheels and move the vehicle. The power delivered by the motor is reflected in the vehicle velocity and in the tractive torque at the wheels. For any kind of powertrain, if the driver wants to move to any specific velocity, the car needs a specific tractive torque at the wheels. For this reason, the powertrain performance is analyzed in terms of tractive torque and vehicle velocity. There are many possible configurations of the hydraulic system that allow the vehicle to move with those tractive torque and velocity, but there is one optimal configuration.

An instantaneous quasi-static optimization is performed for a discrete mesh of tractive torque versus velocity. At each combination of tractive torque and velocity the efficiency of the entire powertrain is optimized. For each combination of tractive torque and velocity, all the possible combinations of hydraulic motor displacement and hydraulic pump displacement are tested. For a specific combination of hydraulic motor and hydraulic pump displacement, hydrostatic transmission variables such as pressure system, hydraulic motor and hydraulic pump velocities, flow rate and efficiency are calculated and the constraints are verified. The most efficient combination of displacements for every tractive torque and vehicle velocity is recorded as well as all other variables associated with the hydrostatic transmission. The process is described schematically in figure 7.1.

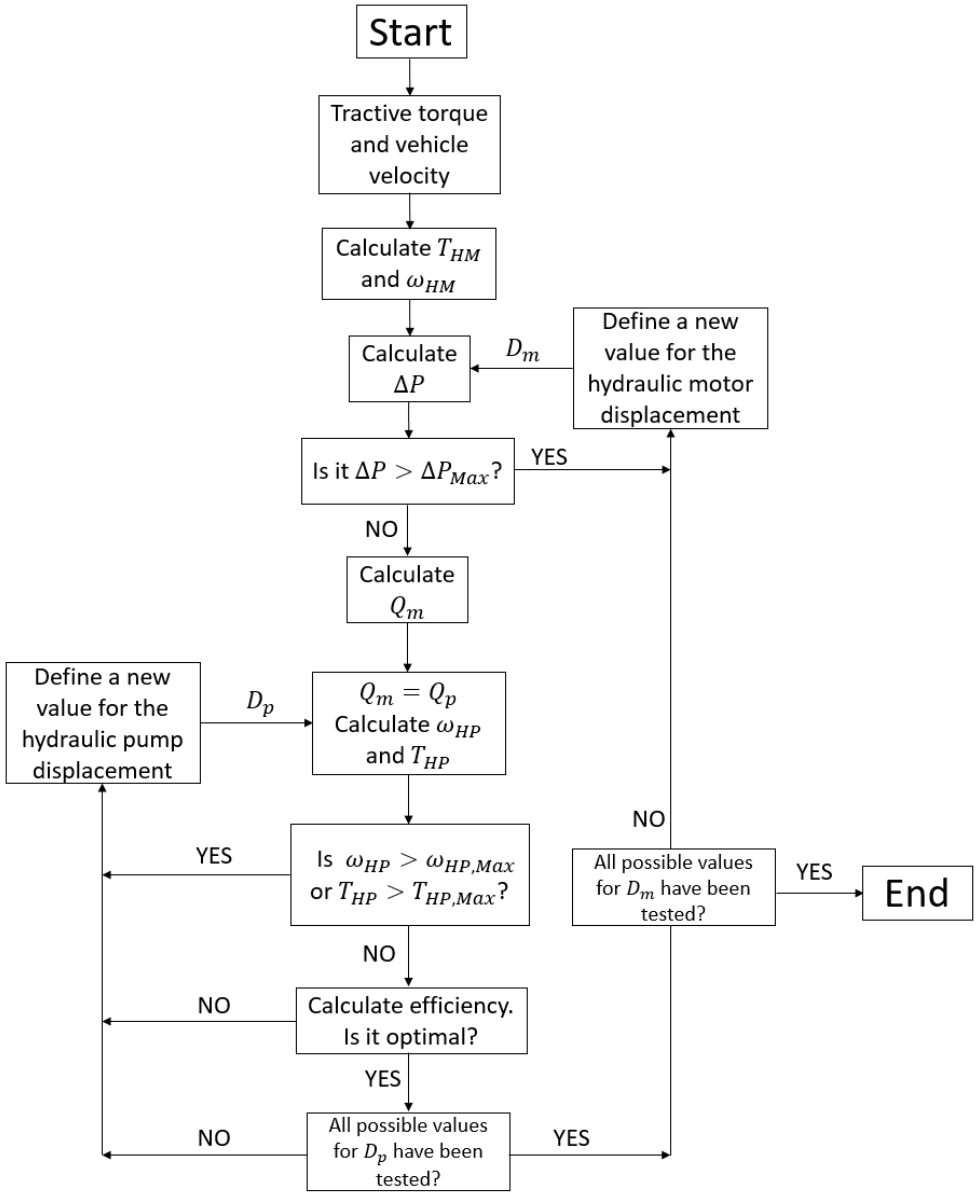

Figure 7.1. Schematic representation of the process.

Optimal hydraulic motor displacement and hydraulic pump displacement are shown in figures 7.2 and 7.3.

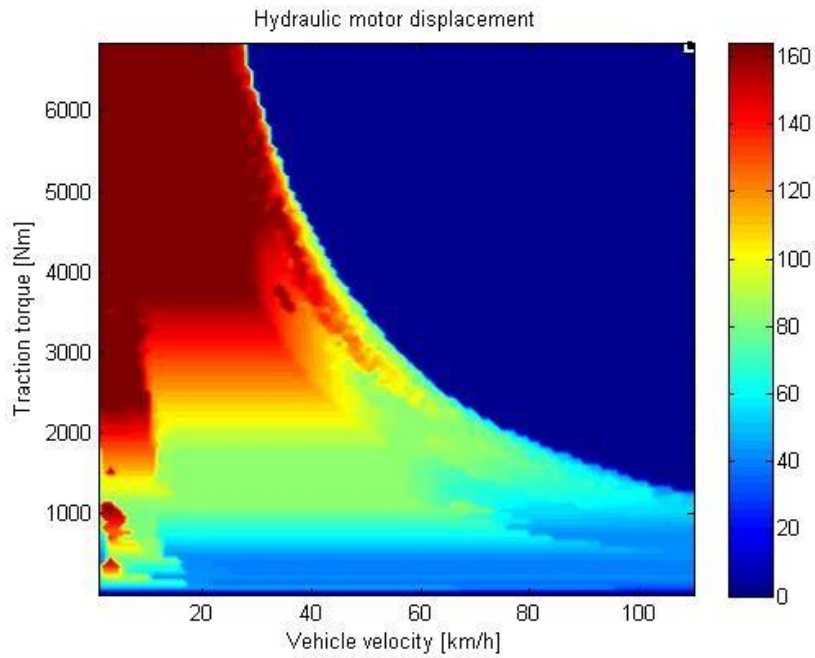

Figure 7.2. Optimal hydraulic motor displacement.

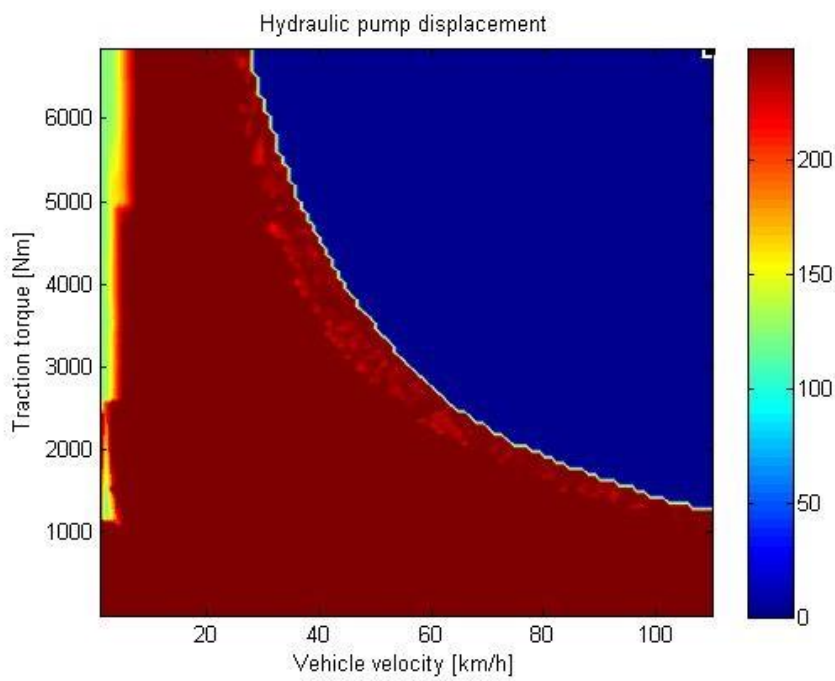

Figure 7.3. Optimal hydraulic pump displacement.

From figure 7.2 it is possible to see that the hydraulic motor displacement is low for low tractive torque and high for high tractive torque. Hydraulic pump displacement, on the other hand, is almost the same for the most of possible combinations of tractive torque and velocity. So, from these figures it is possible to conclude that a good hydraulic transmission can be

composed of a variable displacement hydraulic motor and a fixed displacement hydraulic pump.

The optimal total efficiency of the hydrostatic transmission is shown in figure 7.4. From this figure it is possible to see that the hydraulic transmission can reach efficient values of 80%, but the efficiency is low for low values of tractive torque. In this region of the figure the efficiency is between 3% and 65%. An alternative for improving the efficiency can be the use of a power split transmission, which can be activated for low torque values [3].

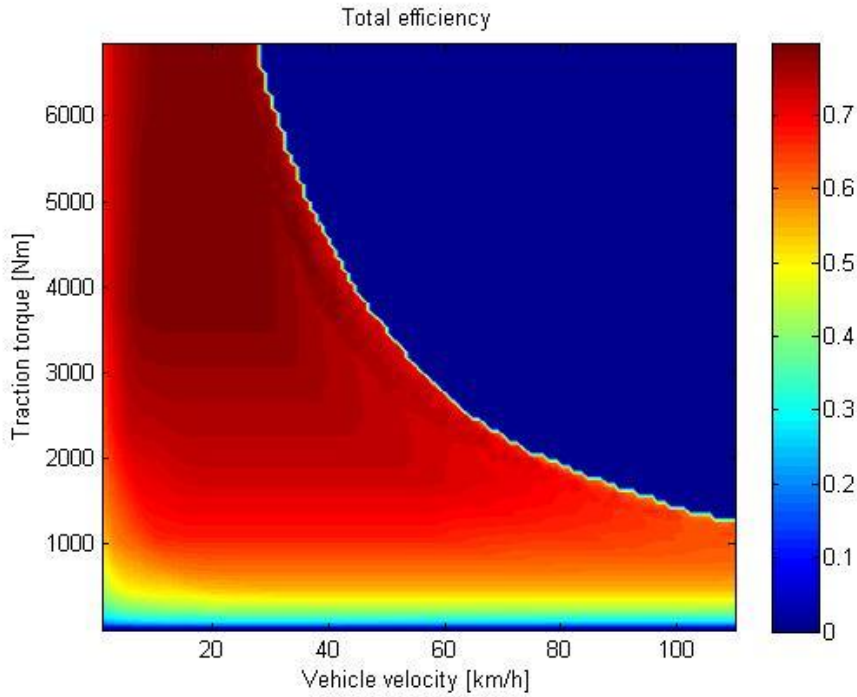

Figure 7.4. Optimal total efficiency of the hydrostatic transmission.

7.2 Fuzzy logic control

The objective of the control system is to try to maintain the hydraulic system as close as possible of its most efficient point. This can be achieved using a convenient control strategy. It has been demonstrated that fuzzy logic control could be applied to the design of electric hybrid vehicles [21] [22] and hydrostatic transmissions [23].

Fuzzy logic control is able to handle with numerical data and with linguistic variables. For example, the driver intention can be measured with the percentage of compression of the pedals. If the acceleration pedal is compressed to 100%, the driver wants to accelerate as much as the motor allow it. This case can be associated with the linguistic variable "High acceleration". In this work, fuzzy logic was used to model the driver input and also it was used as a control strategy for the hydrostatic transmission.

7.3 Driver response model

In the real life the driver controls velocity and acceleration with braking and acceleration pedals. If he wants to go faster, he pushes the accelerator. If he wants to go slower, he stops depressing the accelerator if the desired deceleration is not so big, and begin to push the brake pedal if he wants a bigger deceleration. So, the velocity and acceleration must be the inputs to the driver model. The outputs must be the percentages of compression of the brake and acceleration pedals.

The theoretical percentage of compression for both, brake and accelerator pedals, can be calculated analytically. Traction force can be calculated from equation 9 for any value of acceleration and vehicle velocity. The product of traction force and velocity is equal to the power delivered to the vehicle. This power is a percentage of the power that the motor can deliver. This percentage is directly related with the percentage of compression of the pedals. If the driver wants all the power, he will push the accelerator in a 100%. So, the theoretical percentage of compression is defined as the ratio between the product of traction force and vehicle velocity and the power that the motor can delivered. This theoretical value can be used in order to estimate the driver intention for any possible combination of acceleration and vehicle velocity. A map of theoretical percentage of pedal compression is shown in figure 7.5. For instance, if the vehicle is going to 40 km/h and its acceleration is 0.6 m/s^2, the percentage of compression for the acceleration pedal must be equal or very close to 50 %.

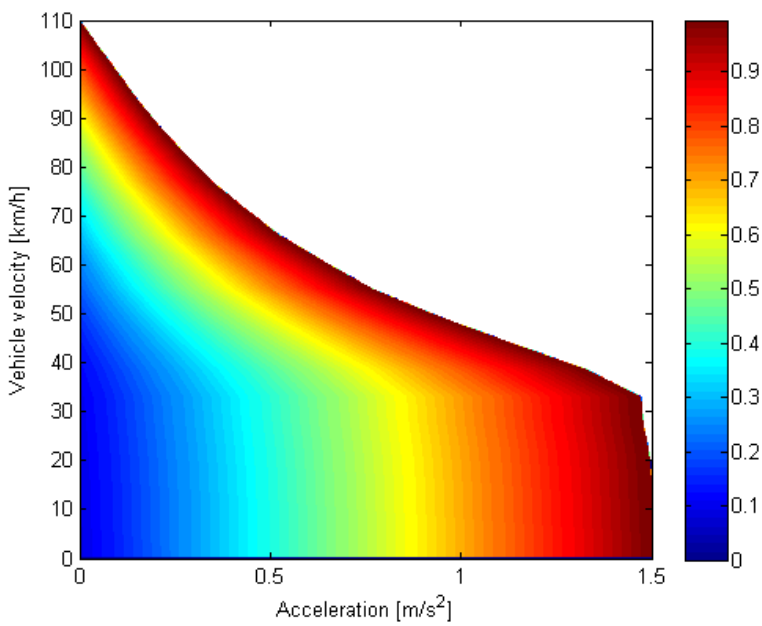

Figure 7.5. Theoretical percentage of accelerator pedal.

In a similar way, there is a limit for the maximum braking force. This limit depends of many factors such as slip ratio and temperature. In this work, as a first approach, a constant coefficient of friction will be assumed [24]. Traction force can be calculated again from equation 9 and compared with the maximum braking force. The theoretical percentage of

compression is defined as the ratio between the traction force and the maximum braking force. A map of theoretical percentage of compression of brake pedal is shown in figure 7.6. In this figure there is a region in which the percentage of compression is zero. This is because the rolling resistance and the drag force is enough to decelerate the vehicle.

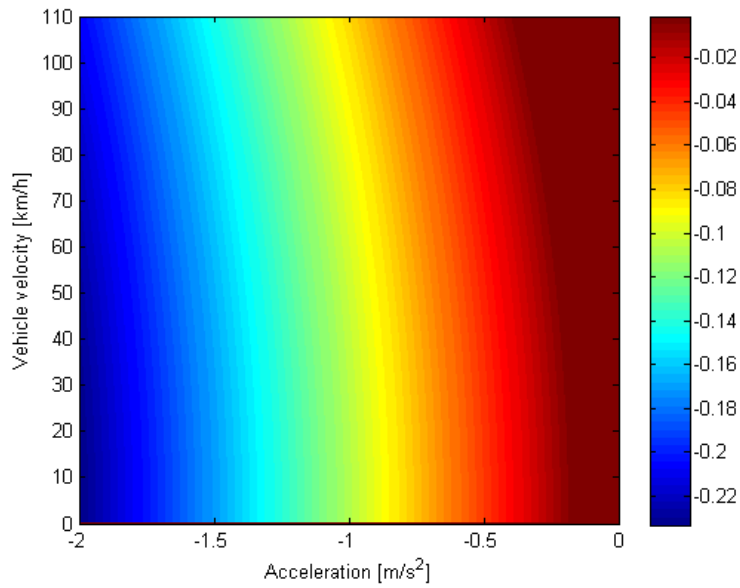

Figure 7.6. Theoretical percentage of brake pedal.

With figures 7.5 and 7.6 it is possible to define the membership functions, fuzzy sets and fuzzy rules. This must be related with linguistic variables associated with the information of the figures. For example, when the acceleration is positive and very high and the velocity is very low, the percentage of compression of the accelerator is very high. When the acceleration is negative and very low and the velocity is very low, the percentage of compression of the brake pedal is zero. The driver response model has three inputs: vehicle velocity, vehicle acceleration and error. The error is calculated with equation 7.1.

$$error = \frac{v_{actual} - v_{reference}}{v_{actual}} \qquad (7.1)$$

The membership functions for all the inputs are shown in figures 7.7, 7.8 and 7.9. The output of the driver response model is the percentage of compression of the pedals. The membership functions for the outputs are shown in figure 7.10.

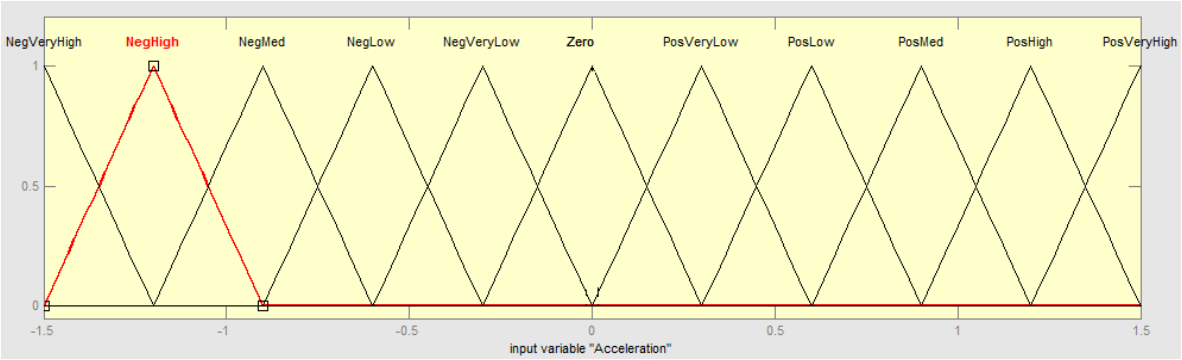

Figure 7.7. Membership functions for the vehicle acceleration.

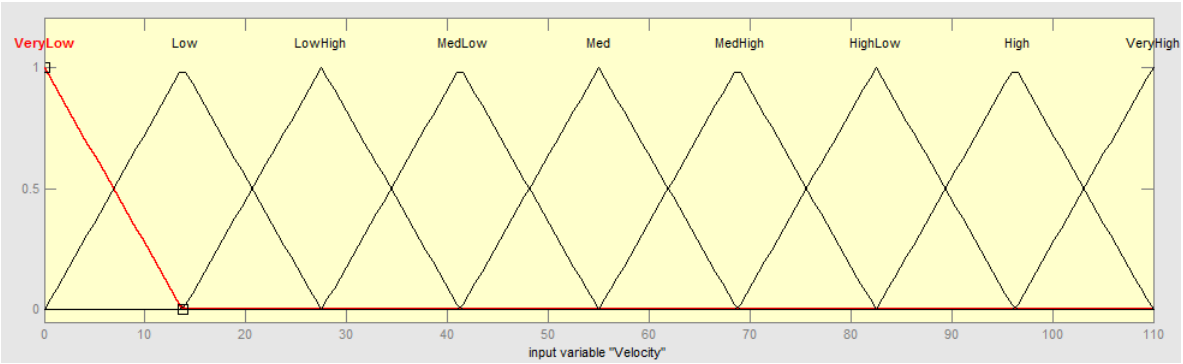

Figure 7.8. Membership functions for the vehicle velocity.

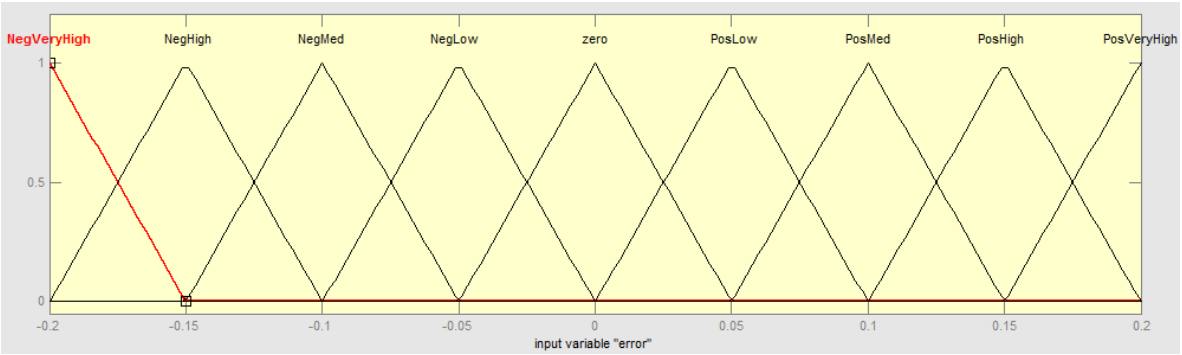

Figure 7.9. Membership functions for the error.

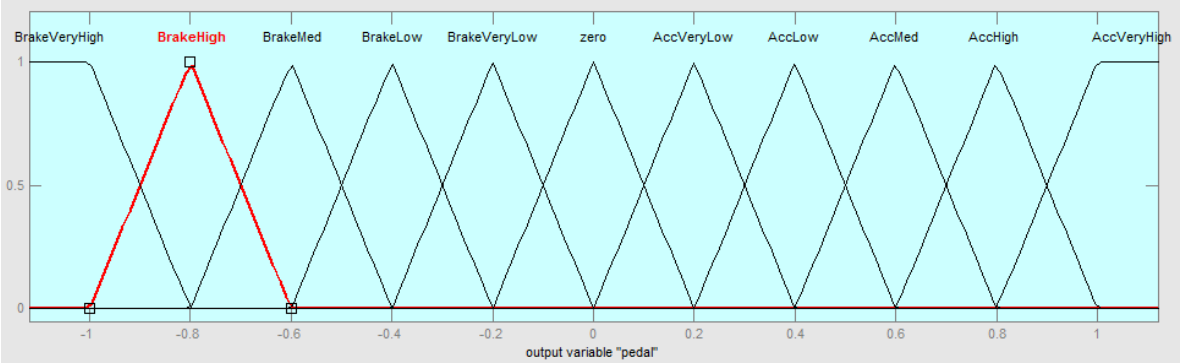

Figure 7.10. Membership functions for the compression of the pedals.

With these functions it is possible to define the fuzzy rules. The fuzzy rules are resumed in figure 7.11. The fuzzy logic controller takes velocity and acceleration and estimate the driver response (it is percentage of compression of the pedals). After that, vehicle velocity and acceleration are calculated and compared with the desired velocity. The output of the fuzzy logic controller also depends of the error, which is related with the difference between actual and desired velocity. A schematic representation of the driver response model is shown in figure 7.12.

Acc\Vel	VeryLow	Low	LowHigh	MedLow	Med	MedHigh	HighLow	High	VeryHigh
NegVeryHigh	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow
NegHigh	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow
NegMed	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow	BrakeVeryLow
NegLow	Zero	Zero	Zero	Zero	Zero	Zero	Zero	Zero	Zero
NegVeryLow	Zero	Zero	Zero	Zero	Zero	Zero	Zero	Zero	Zero
Zero	Zero	Zero	Zero	AccVeryLow	AccLow	AccLow	AccMed	AccHigh	AccVeryHigh
PosVeryLow	AccVeryLow	AccVeryLow	AccVeryLow	AccVeryLow	AccLow	AccMed	AccHigh	AccVeryHigh	AccVeryHigh
PosLow	AccLow	AccLow	AccLow	AccMed	AccHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh
PosMed	AccMed	AccMed	AccMed	AccHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh
PosHigh	AccHigh	AccHigh	AccHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh
PosVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh	AccVeryHigh

Figure 7.11. Fuzzy rules.

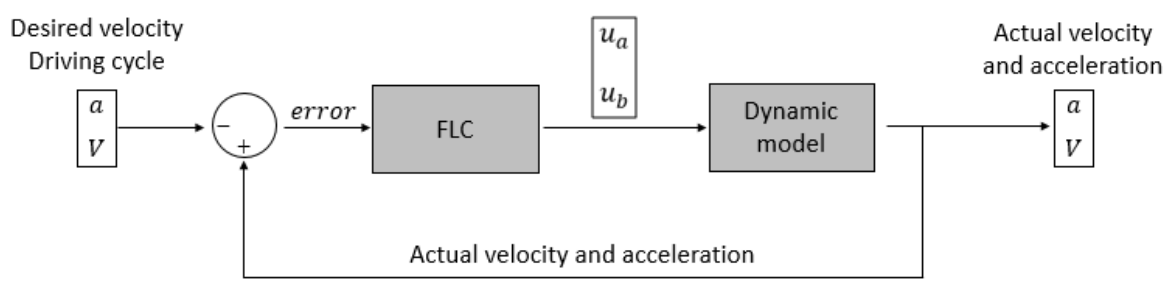

Figure 7.12. Schematic representation of the driver response model.

This model was implemented in Matlab/Simulink environment with the function fuzzy. The Simulink model is shown in figure 7.13.

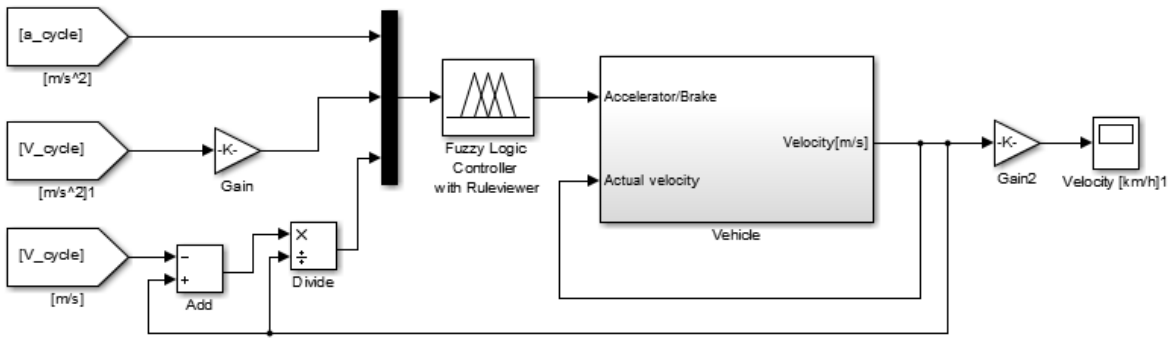

Figure 7.13. Simulink model of the driver response model.

This model has three inputs: acceleration, velocity and error. The output is the percentage of compression of the pedals. The output is one of the inputs of the plant call vehicle. The model of the plant is shown in figure 7.14.

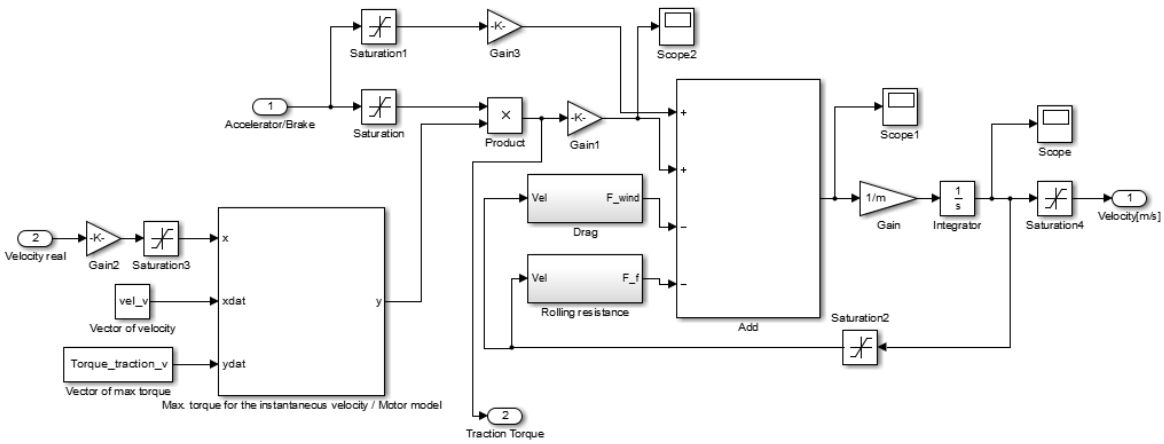

Figure 7.14. Detail of the model of the vehicle.

In the block "Add" the sum of the terms of the equation 5.9 is done. The inputs 1 and 2 of this block are the traction force. The input 1 is the traction force for braking and the input 2 is the traction force for moving forward. In the input 2, a linear model of the accelerator is made. In the "Lookup Table Dynamic" the maximum tractive torque is interpolated for each value of vehicle velocity. The data input of this "Lookup Table Dynamic" is the maximum tractive torque as a function of the vehicle velocity. This function was made with the assumption of a 100% efficient transmission. Once the maximum tractive torque is calculated, this value is multiplied for the percentage of compression of the pedal. The output of the model is the actual vehicle velocity, which is compared with the reference velocity.

7.4 Hydraulic system control

The control system for the hydrostatic transmission must be able to maintain the transmission in its most efficient point of operation. The most efficient point was determined in a previous section with the instantaneous steady optimization.

The inputs of the control system must be the actual vehicle velocity, the tractive torque and the error between the actual electric motor velocity and the ideal electric motor velocity. With vehicle velocity and tractive torque, the optimal electric motor velocity is calculated with the instantaneous steady optimization. The optimal value is compared with the actual value and the fuzzy logic control adjust the system in order to be as near as possible of the optimal value. A schematic representation of this process is shown in figure 7.15.

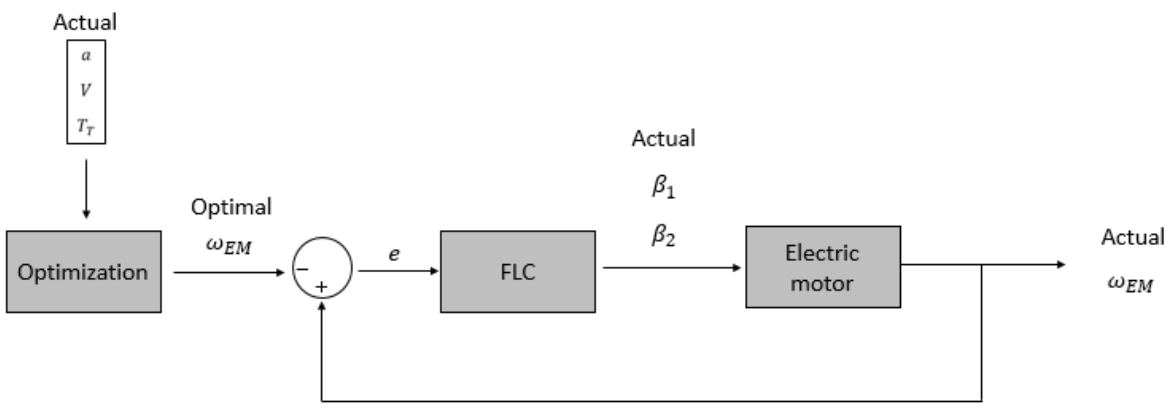

Figure 7.15. Schematic representation of the hydrostatic transmission control strategy.

The membership functions can be defined with the results of the instantaneous steady optimization. The membership functions for all the inputs are shown in figures 7.16, 7.17 and 7.18. The membership functions for the output are shown in figure 7.19.

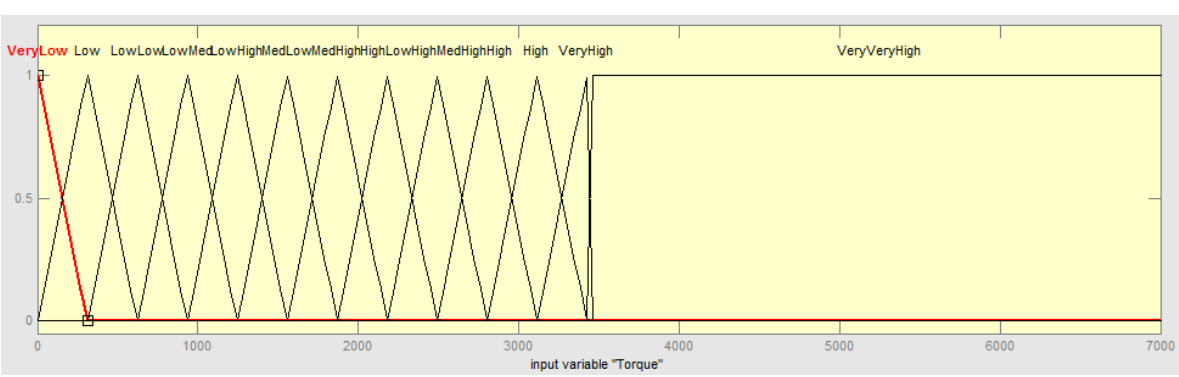

Figure 7.16. Membership functions for the tractive torque.

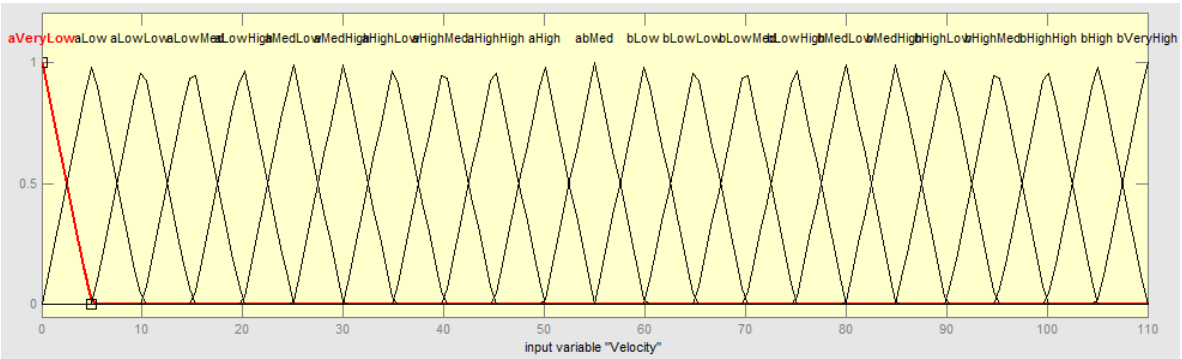

Figure 7.17. Membership functions for the vehicle velocity.

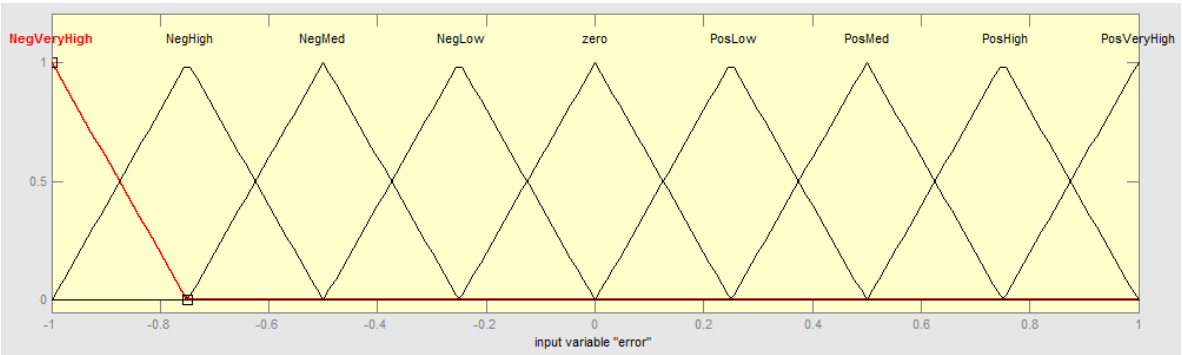

Figure 7.18. Membership functions for the error.

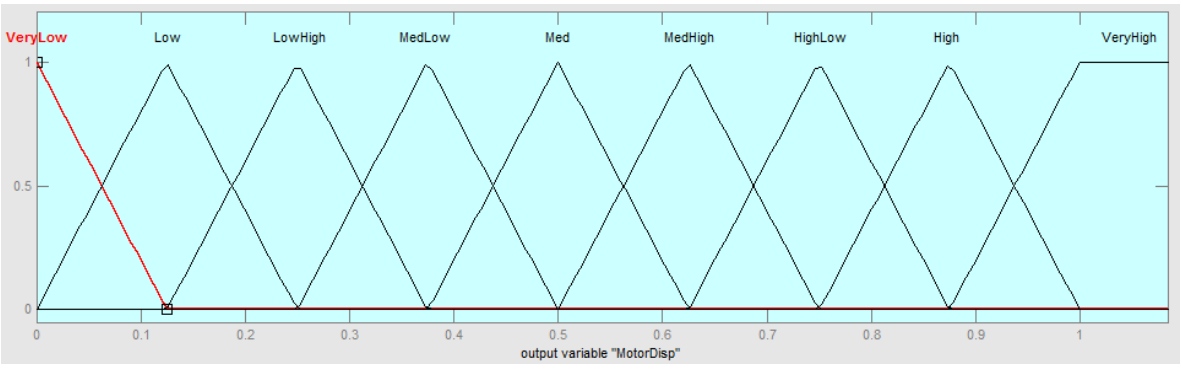

Figure 7.19. Membership functions for the hydraulic motor displacement.

This model was implemented in Matlab/simulink environment with the function fuzzy. The Simulink model is shown in figure 7.20.

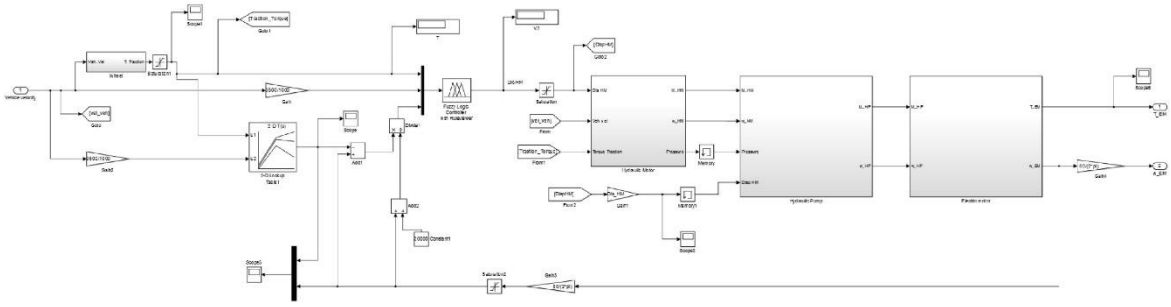

Figure 7.20. Simulink model for the hydrostatic transmission control.

As was mentioned before, the fuzzy logic control has three inputs. A detail of this part of the model is shown in figure 7.21.

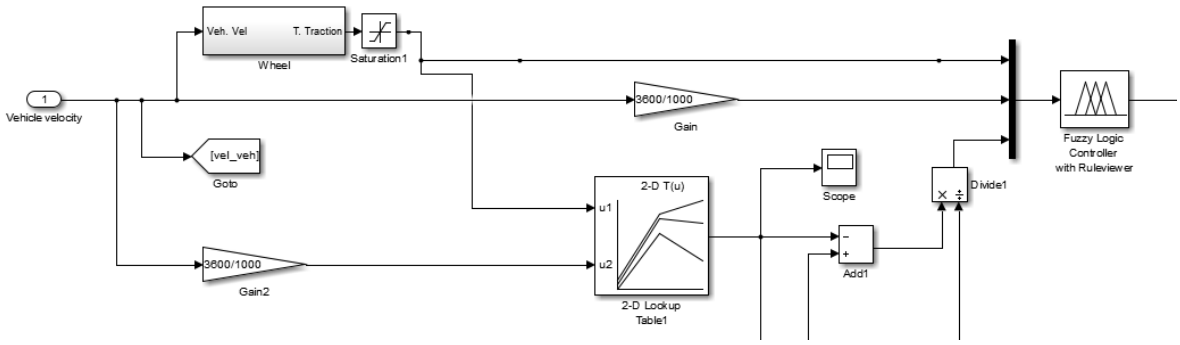

Figure 7.21. Detail of the hydrostatic transmission model.

The first input is the tractive torque which is calculated again with equations 5.9 and 5.11. the second input is the vehicle velocity and the third input is the error between actual electric motor velocity and reference electric motor velocity which was calculated in the study of the efficiency. With the block "2-D Lookup Table" a 2D interpolation can be made. In this case, a map of electric motor velocity as a function of tractive torque and velocity is read and for every time the reference electric motor velocity is determined for a specific value of tractive torque and vehicle velocity. This reference is compared with the actual value and the error is the input of the fuzzy logic controller.

The output of the fuzzy logic controller is the input of the plant. The plant is composed for three devices: the hydraulic motor, the hydraulic pump and the electric motor. A detailed view of the plant is shown in figure 7.22.

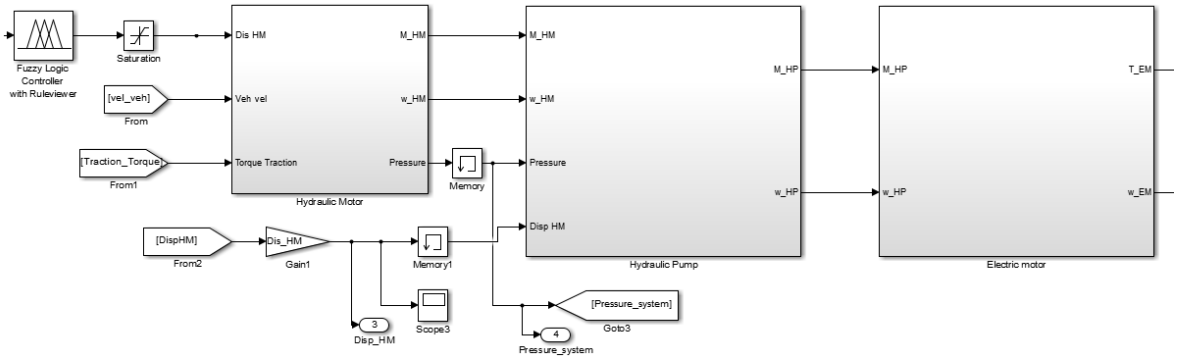

Figure 7.22. Detailed view of the plant.

The output of the plant is the electric motor velocity. The fuzzy logic control is designed in order to maintain the electric motor velocity as close as possible of its reference value, which is obtained from the study of the efficiency. The first part of the plant is the hydraulic motor. The hydraulic motor model is shown in figure 7.23.

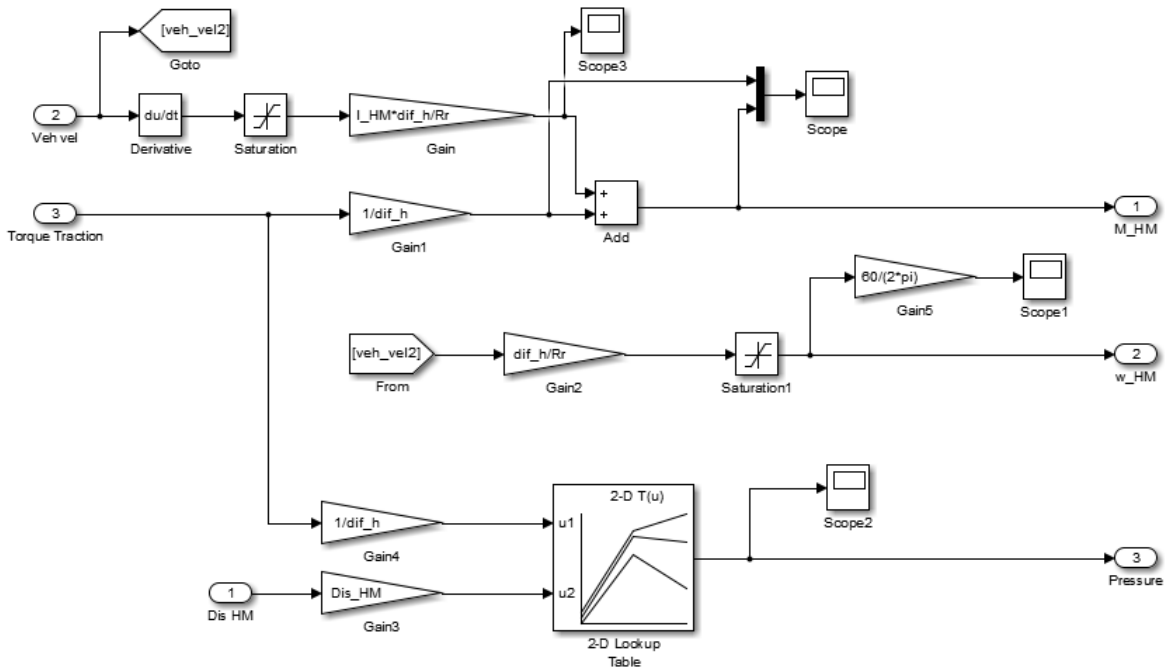

Figure 7.23. Hydraulic motor model.

The inputs of the hydraulic motor model are three: the displacement of the hydraulic motor (which is the output signal of the fuzzy logic controller), the vehicle velocity and the tractive torque. The outputs of this model are three: the hydraulic torque in the motor, the angular velocity of motor and the pressure of the system. The first and second inputs are calculated with the equations of the hydraulic devices presented in a previous section. The pressure of the system is calculated with equation 5.5.

$$M_m = \Delta P D_m \eta_{m,m} \qquad (5.5)$$

This equation was solved for a mesh of many values of hydraulic motor torque and displacement. In the Matlab file `sol_P_HM`, the value of pressure which is the solution of the equation for any combination of torque and displacement is saved. The solution of this equation is used in the hydraulic motor model. With the "2-D Lookup Table" an interpolation of the solution is made. The inputs of this table are the torque and the displacement and the output is the pressure, which is the solution of the equation 5.5.

Before the hydraulic motor model, a subsystem with the hydraulic pump model is placed. A detailed view of this model is shown in figure 7.24.

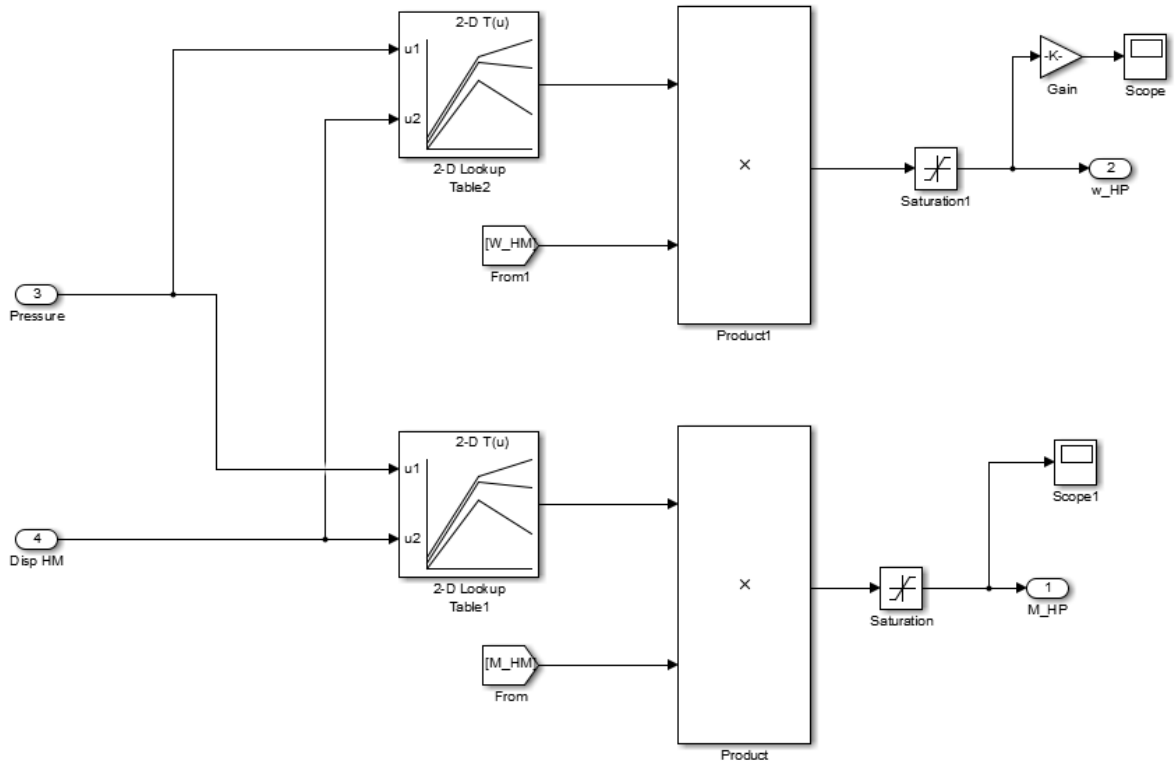

Figure 7.24. Hydraulic pump model.

This model has four inputs: hydraulic motor torque, hydraulic motor angular velocity, system pressure and hydraulic motor displacement. The model has two outputs: hydraulic pump torque and hydraulic pump angular velocity. The hydraulic pump angular velocity is calculated from equation 5.12.

$$\omega_p = \omega_m \frac{\boldsymbol{D_m}}{\boldsymbol{D_p}} \frac{\boldsymbol{1}}{\boldsymbol{n_{v,m} n_{v,p}}} \qquad (5.12)$$

The factor shown in bold is calculated with the file `eff_hydrostatic_transmission` for a mesh of hydraulic motor displacement and system pressure. With the "2-D Lookup Table2" the value of the factor in equation 5.12 is calculated through interpolation for any value of pressure and hydraulic motor displacement. After that, the factor and the hydraulic motor velocity are multiplied. In the model, there is a second "2-D Lookup Table". With this block, the hydraulic pump torque is calculated. The torque is calculated from the next equation.

$$M_p = M_m \frac{\boldsymbol{D_p}}{\boldsymbol{D_m}} \frac{\boldsymbol{1}}{\boldsymbol{\eta_{m,m} \eta_{m,p}}}$$

The factor shown in bold is calculated with the file `eff_hydrostatic_transmission` for a mesh of hydraulic motor displacement and system pressure. This factor is calculated

through interpolation for any value of pressure and hydraulic motor displacement. After that, the factor and the hydraulic motor torque are multiplied.

The last part of the plant is the electric motor. The Simulink model for the electric motor is shown in figure 7.25.

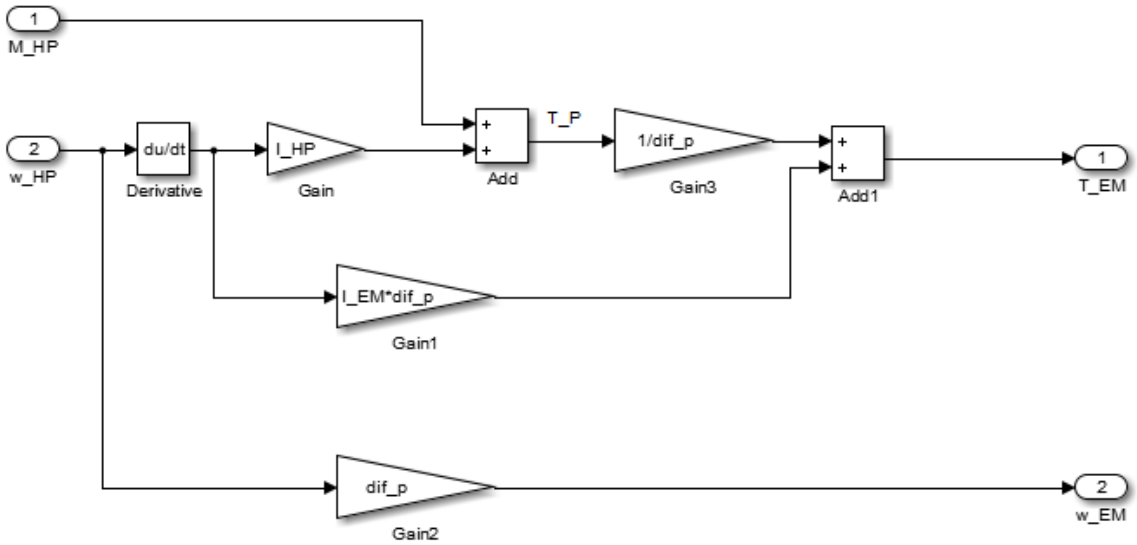

Figure 7.25. Electric motor model.

In this model, the equations of the electric motor are solved. The inputs are the variables of the hydraulic pump: torque and angular velocity. The outputs are angular velocity and torque in the electric motor. This velocity is compared with the reference and the error between them is one of the three inputs of the fuzzy logic controller.

8. Results

The results obtained in this work are presented in this section.

8.1 Driver response model

The driver response model implemented in Simulink was tested for some conditions. The first one is the driving cycle presented in figure 6.2. The comparison between desired velocity and actual velocity is presented in figure 8.1. The percentage of compression for the acceleration pedal is shown in figure 8.2.

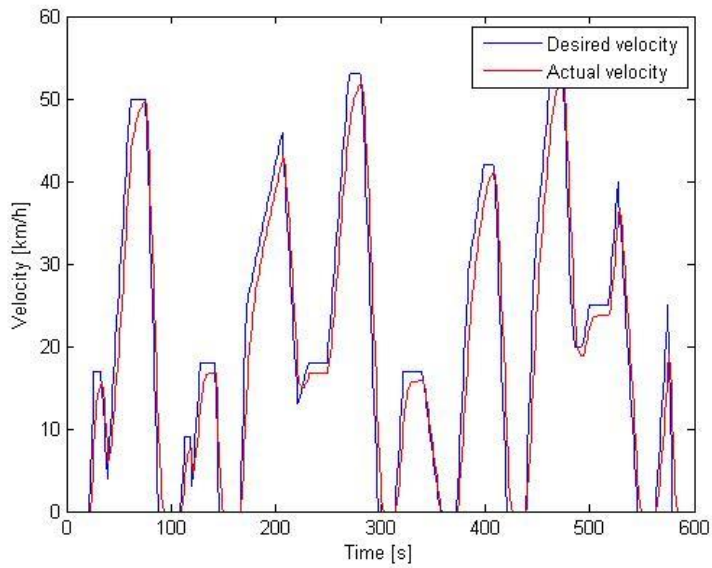

Figure 8.1. Comparison between desired velocity and actual velocity.

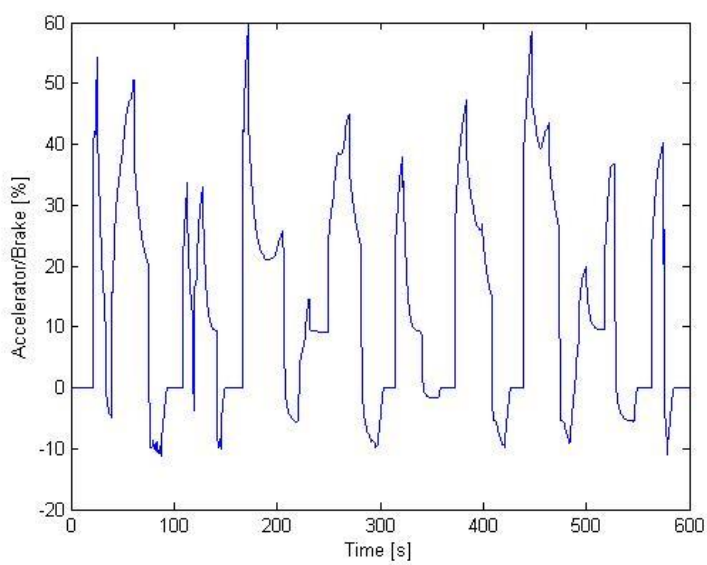

Figure 8.2. Percentage of compression for the acceleration pedal.

From figure 8.1 it is possible to see that the results for the driver response model are good. The error for velocity is around 7%. Also, this model was tested for a square signal in order to see its response characteristics. A comparison between reference velocity and actual velocity is shown in figure 8.3.

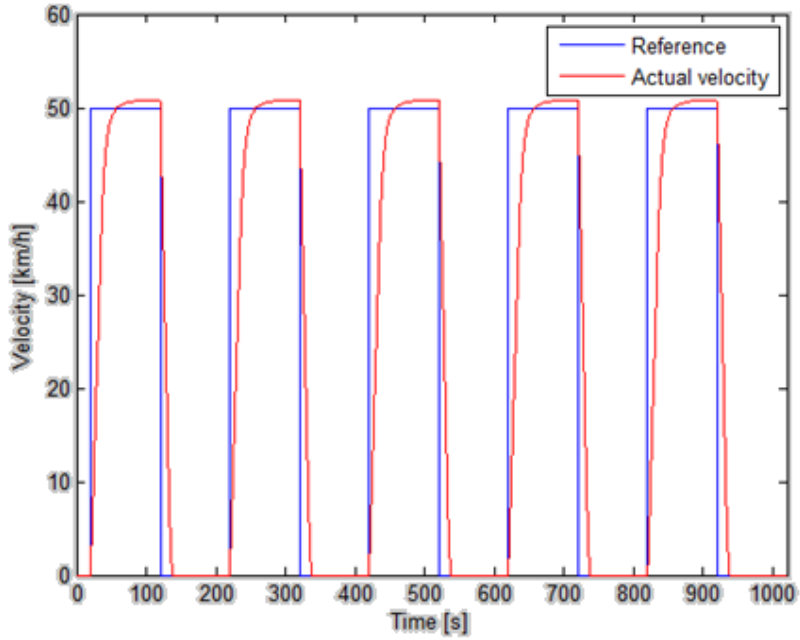

Figure 8.3. Comparison between reference velocity and actual velocity.

From this figure it is possible to see that the model is able to follow the proposed driving cycle. The percentage of compression for the acceleration pedal is shown in figure 8.4.

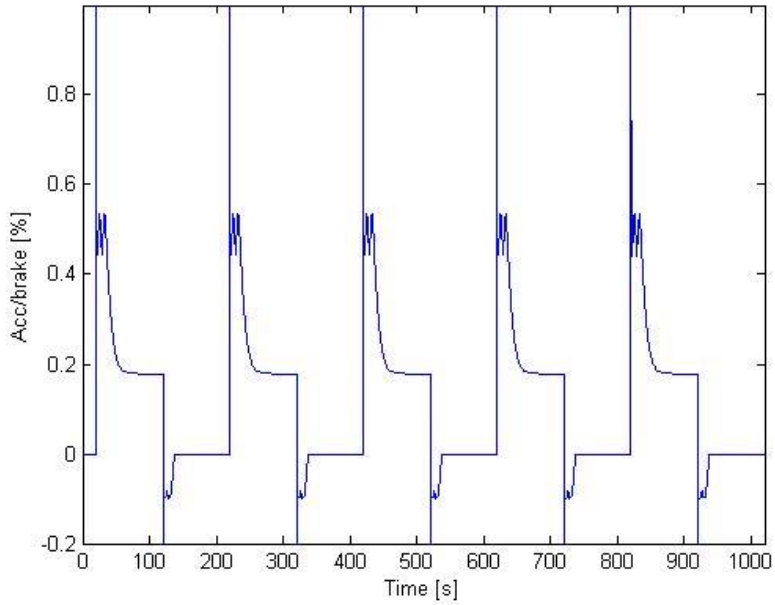

Figure 8.4. Percentage of compression for the acceleration pedal.

8.2 Hydraulic transmission control

The principal objective of the control system for the hydrostatic transmission is to maintain the system as close as possible of the most efficient point. This can be achieved controlling the displacement of the hydraulic motor in order to change the transmission ratio for maintaining the system in its most efficient point, which was determined with the instantaneous steady optimization described before.

The control system was tested for the driving cycle shown in figure 6.2. A comparison between ideal electric motor velocity and actual electric motor velocity is shown in figure 8.5.

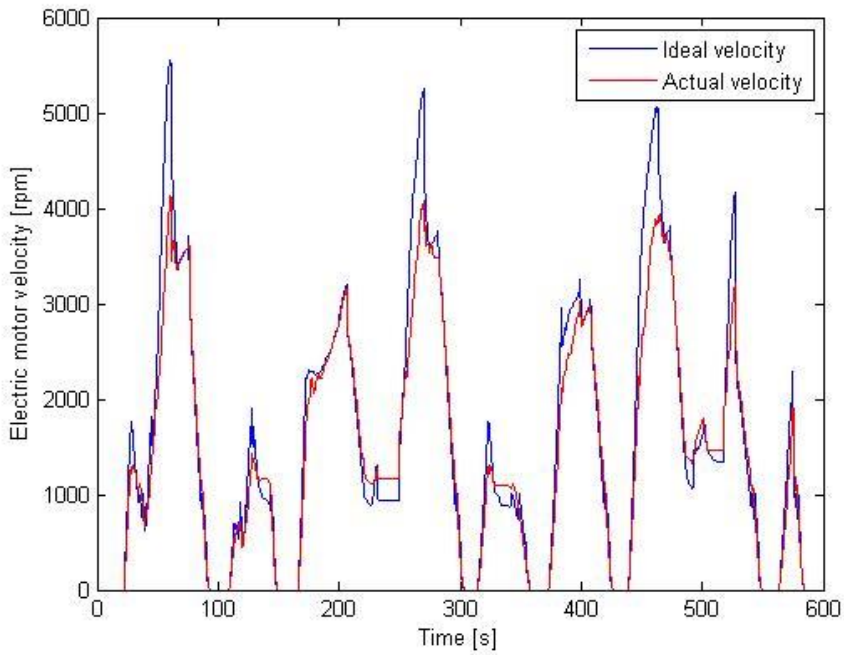

Figure 8.5. Comparison between desired velocity and actual velocity.

From figure 8.5 it is possible to see that the control system is able to maintain the electric motor velocity close to its most efficient point. The error is around 11%. The hydraulic motor displacement, which is the output of the control system, is shown in figure 8.6.

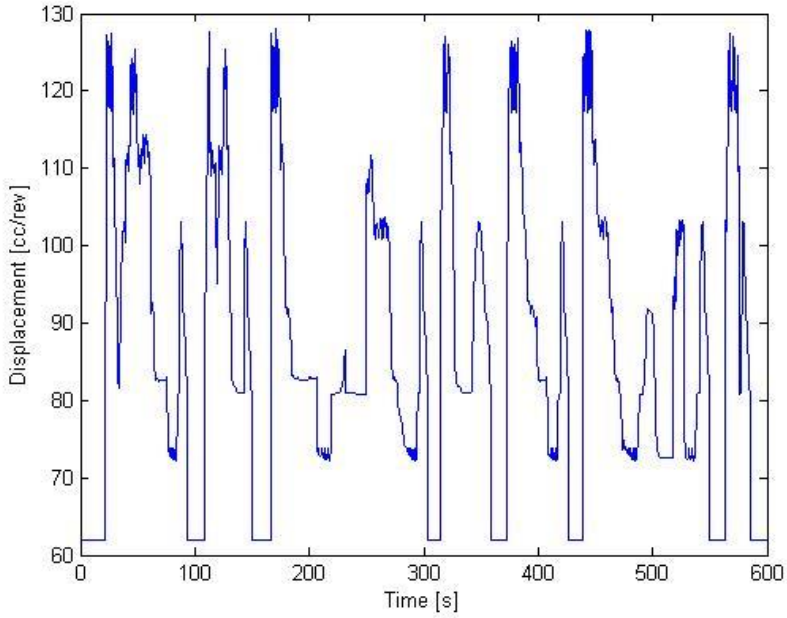

Figure 8.6. Hydraulic motor displacement.

The control for the hydrostatic transmission was tested for a square signal. A comparison between reference angular velocity and actual angular velocity of the motor is shown in figure 8.7.

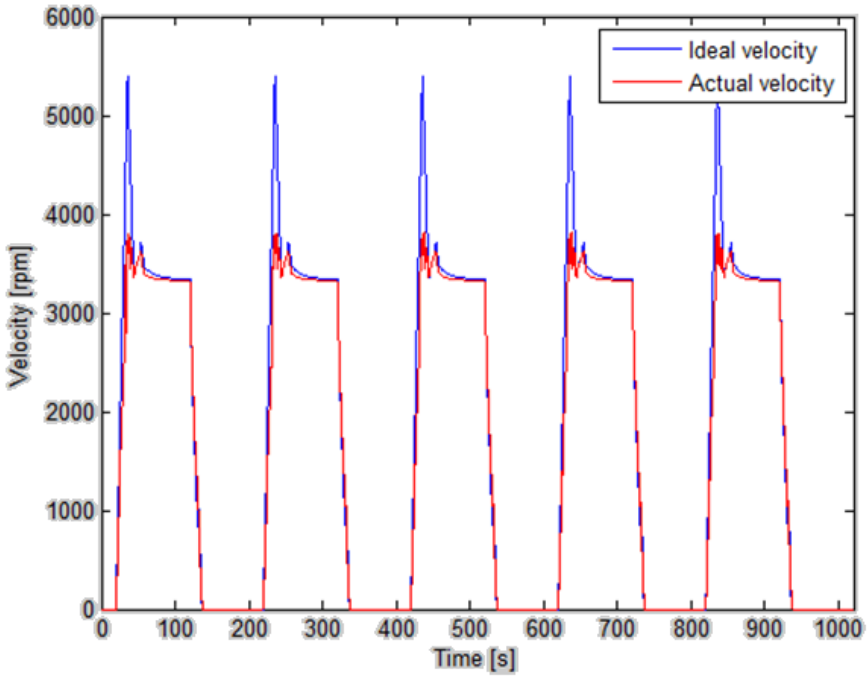

Figure 8.7. Comparison between desired velocity and actual velocity.

From figure 8.7 it is possible to see that the hydraulic system control is able to maintain the electric motor velocity very close of its reference value. The displacement of the hydraulic motor, which is the output of the control system, is shown in figure 8.8.

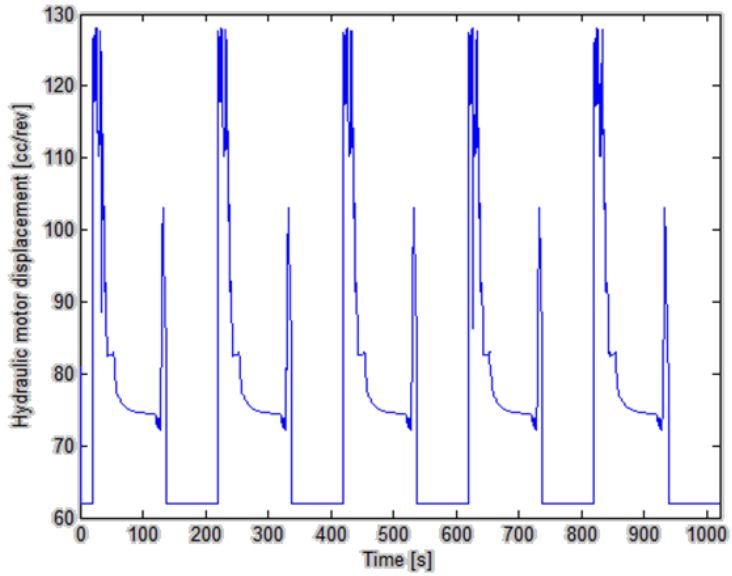

Figure 8.8. Hydraulic motor displacement.

8.3 Energetic analysis

A significant increase in energy savings was achieved using the instantaneous steady optimization. For the driving cycle shown in figure 6.2 the energy saved can be around 6% over a hydrostatic transmission without optimization. This energy saving is due to the raise in efficiency of all the system. With the control system, the energy saving is around 4.5%, which is very close to the ideal energy saving. A comparison between the power consumption of the system is shown in figure 8.9.

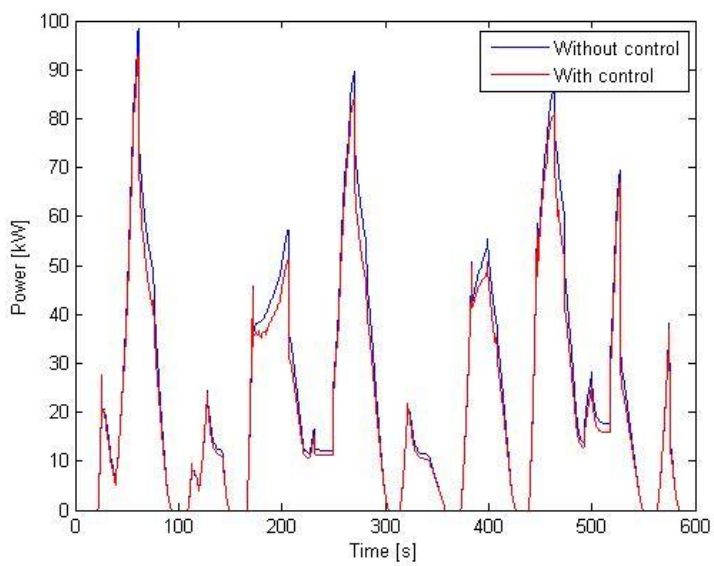

Figure 8.9. Power consumption.

From figure 8.9 it is possible to see that the efficiency is improved with the proposed control system. The power consumption is reduced when the proposed control strategy is implemented. As was mentioned before, the energy saving is around 4.5%.

For the square signal, the power consumption is shown in figure 8.10.

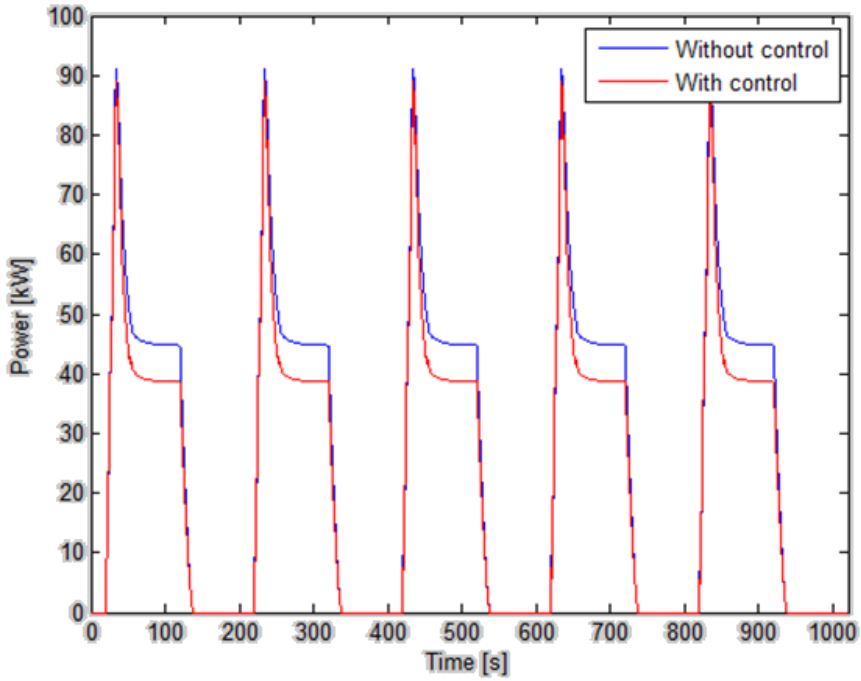

Figure 8.10. Power consumption for the square signal.

9. Conclusions

A numerical model was developed in order to select the components of a hydrostatic transmission for a terrestrial commercial vehicle. The model allows to compare at the same time many options of hydraulic components and select the component which can supply the requirements of the vehicle for a specific driving cycle. This model takes into account relevant characteristics of the devices such as its mass inertia and its limit capabilities.

One of the most important advantages of a hydrostatic transmission is the possibility of adjusting the displacement of the hydraulic devices for changing the transmission ratio. This property allows to have a continuous variable transmission. The efficiency of the hydraulic devices depends highly of its displacements, so it is important to study their influence in the efficiency of all the system. For this purpose, a quasi-static model was used to determine the most efficient possible configuration of the hydrostatic transmission. For a specific value of vehicle velocity and traction torque, all the possible configurations were evaluated through simulation and the most efficient configuration was recorded. In figures 7.2 and 7.3 optimal hydraulic motor displacement and optimal hydraulic pump displacement are shown. From figure 7.3 it is possible to see that the hydraulic pump displacement is almost the same in all regions, but the optimal hydraulic motor displacement takes many values. So, a good hydrostatic transmission can be made with a variable displacement hydraulic motor and a fixed displacement hydraulic pump. The optimal total efficiency of the system can reach values of 80% for high values of tractive torque (figure 7.4), but for low values of tractive torque the optimal efficiency is between 3% and 65% which is very low compared with a traditional mechanical transmission. For low tractive torque values, an alternative as power split transmission can be useful.

With an appropriate control strategy, the efficiency of the hydrostatic transmission can be improved considerably. Through simulation, it was determined that the energy consumption can be reduced until 6% with respect to a hydrostatic transmission without optimal control. Using a fuzzy logic controller, the efficiency was improved 4.5%. The efficiency of the system can be improved, as can be seen in figures 8.9 and 8.10.

Fuzzy logic control was selected as a control method because it has used successfully in hybrid vehicles [12] [13] [14] [15]. In this work, fuzzy logic was used to model the driver input and also it was used as a control strategy for the hydrostatic transmission.

In order to model the driver input, a computational fuzzy logic model was developed and implemented in Simulink. The membership functions and fuzzy rules were defined according to the figures 7.5 and 7.6, which were obtained with the dynamic model of the vehicle. The results of this model are good. The error between desired velocity and actual velocity is around 7%. The model can be improved adding membership functions in order to raise the resolution.

The results for the hydrostatic transmission control are good. The fuzzy logic controller is able to maintain the hydrostatic transmission very close to its most efficient point. In figures 8.5 and 8.7 it is possible to see that the ideal and actual electric motor velocity are very close. The error between them is around 11%. Moreover, the efficiency of the system is improved through the time, as can be seen in figures 8.9 and 8.10.

10. Future work

1. It is necessary to review the acceleration/braking model used in the driver response model. In the present work a linear model was used, but no other models were reviewed. The difference between reference and actual velocity in figure 8.1 can be improved using a more realistic model.
2. In the study of the efficiency of the system in chapter 7.1, the influence of vehicle acceleration must be studied. In the present work, a mesh of tractive torque and velocity was made and for every possible combination of tractive torque and velocity, the efficiency of the system was studied. As a future work, it is necessary to add the acceleration of the vehicle as a variable. So, the efficiency will be studied for every combination of tractive torque, vehicle velocity and acceleration.
3. In a future work it is necessary to simulate the performance of the system for an actual driving cycle in order to have more realistic results.
4. It is necessary to compare the control strategy proposed in this work with other control strategies in order to identify pros and cons of every possible alternative.
5. In the present work, the power consumption of the proposed system was compared with the power consumption of a vehicle with a hydrostatic transmission without control. In a future work, it is necessary to compare the power consumption of the system proposed in this work with a fully electric hybrid vehicle.
6. In a future work it is necessary to add the model of the regenerative braking and the accumulator. Also, it is necessary to review the braking model proposed in the present work.
7. Alternatives as power split transmission must be studied and evaluated in order to improve the efficiency of the system for low values of tractive torque.

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Appendix

In this appendix, an explanation about using the Matlab/Simulink simulations is made. Next, all the Matlab files are explained.

File name	Function
`Torque_traction.m`	The maximum traction torque that the electric motor can provide for a specific vehicle velocity. This traction torque is calculated with the assumption of a 100% efficiency of the transmission. This file need the electric motor data. The data file has three columns. The first one is the angular velocity in rpm, the second one is the maximum torque and the last one is the maximum power.
`Driving_cycle.m`	The driving cycle used as a reference is calculated in this file. Also, this file is used for reading the results of the study of the efficiency of the system.

`eff_hydrostatic_transmission.m`	With this file the hydrostatic transmission efficiency is calculated for all possible values of pressure and displacement. Also, the next numbers are calculated:
	`Factor_HP_Torque`
	$$\mathrm{M_p} = \mathrm{M_m}\frac{\boldsymbol{D_p}}{\boldsymbol{D_m}}\frac{\mathbf{1}}{\boldsymbol{\eta_{m,m}\eta_{m,p}}}$$
	`Factor_HP_Vel`
	$$\omega_\mathrm{p} = \omega_\mathrm{m}\frac{\boldsymbol{D_m}}{\boldsymbol{D_p}}\frac{\mathbf{1}}{\boldsymbol{\eta_{v,m}\eta_{v,p}}}$$

`EM_eff.m`	The efficiency map for the electric motor is generated in this filed.
`sol_P_HM.m`	The pressure is solved for every possible value of displacement and hydraulic motor torque. The equation solved is the next:

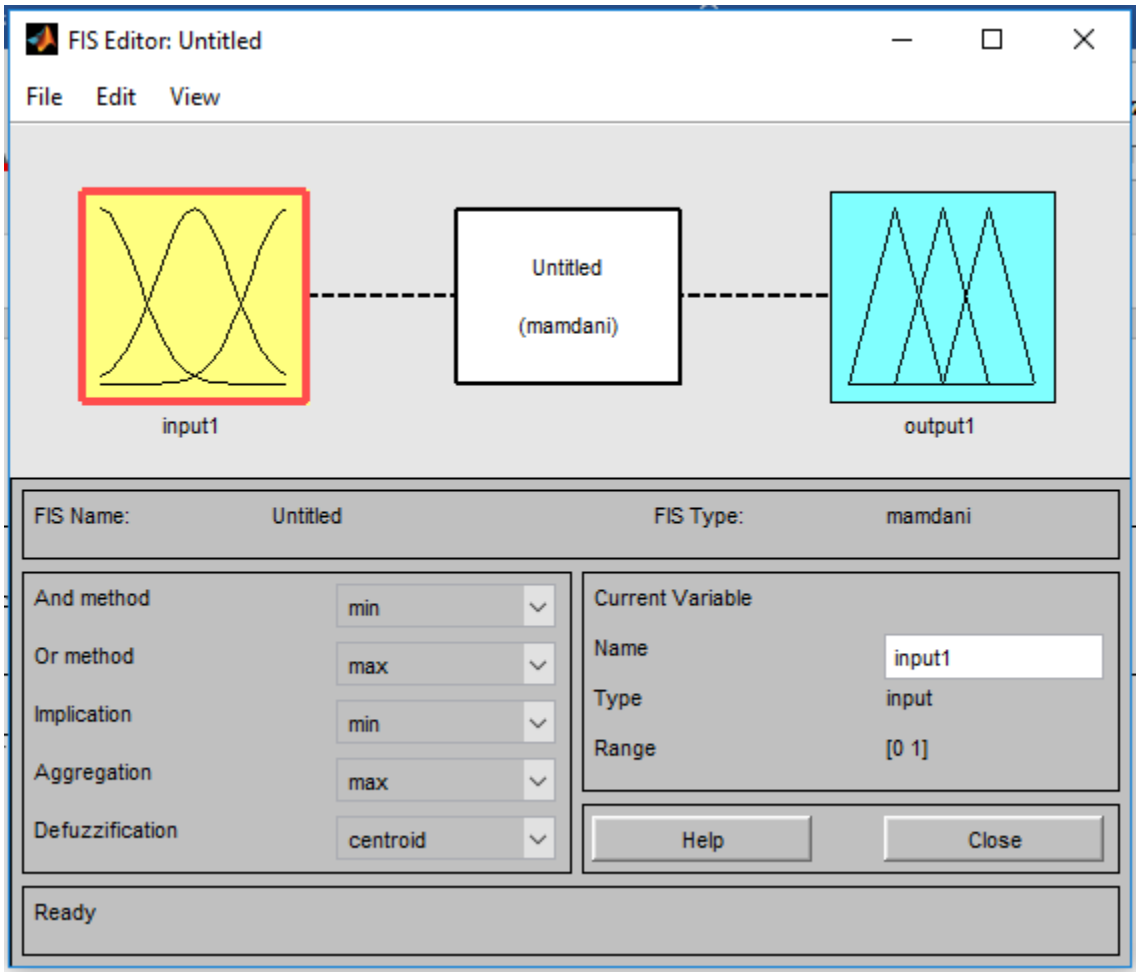
	$M_m = \Delta P D_m \eta_{m,m}$
`vehicle.m`	The vehicle variables are described in this file. All the files are written for reading the variables from this file.
`hydrostatic_transmission.m`	The hydrostatic transmission variables are described in this file. All the files are written for reading the variables from this file.
`efficiency_HP.m`	This function has as inputs the pressure (bar) and displacement (0 - 1) and the output is a vector with three positions: the first position is the total efficiency, the second position is the volumetric efficiency and the last position is the mechanical efficiency.
`efficiency_HM.m`	This function has as inputs the pressure (bar) and displacement (0 - 1) and the output is a vector with three positions: the first position is the total efficiency, the second position is the volumetric efficiency and the last position is the mechanical efficiency.
`efficiency_EM.m`	This function has as inputs the torque (Nm) and the angular velocity (rpm) and the output is the electric motor efficiency (0 - 100).
`optimization.m`	The study of the efficiency of the system is made with this file. The computational cost associated with this file is big.
`Controller_v4`	The Simulink models for the human response and the hydrostatic transmission are in this file.

For running the simulation, it is necessary to run the files in the next order:

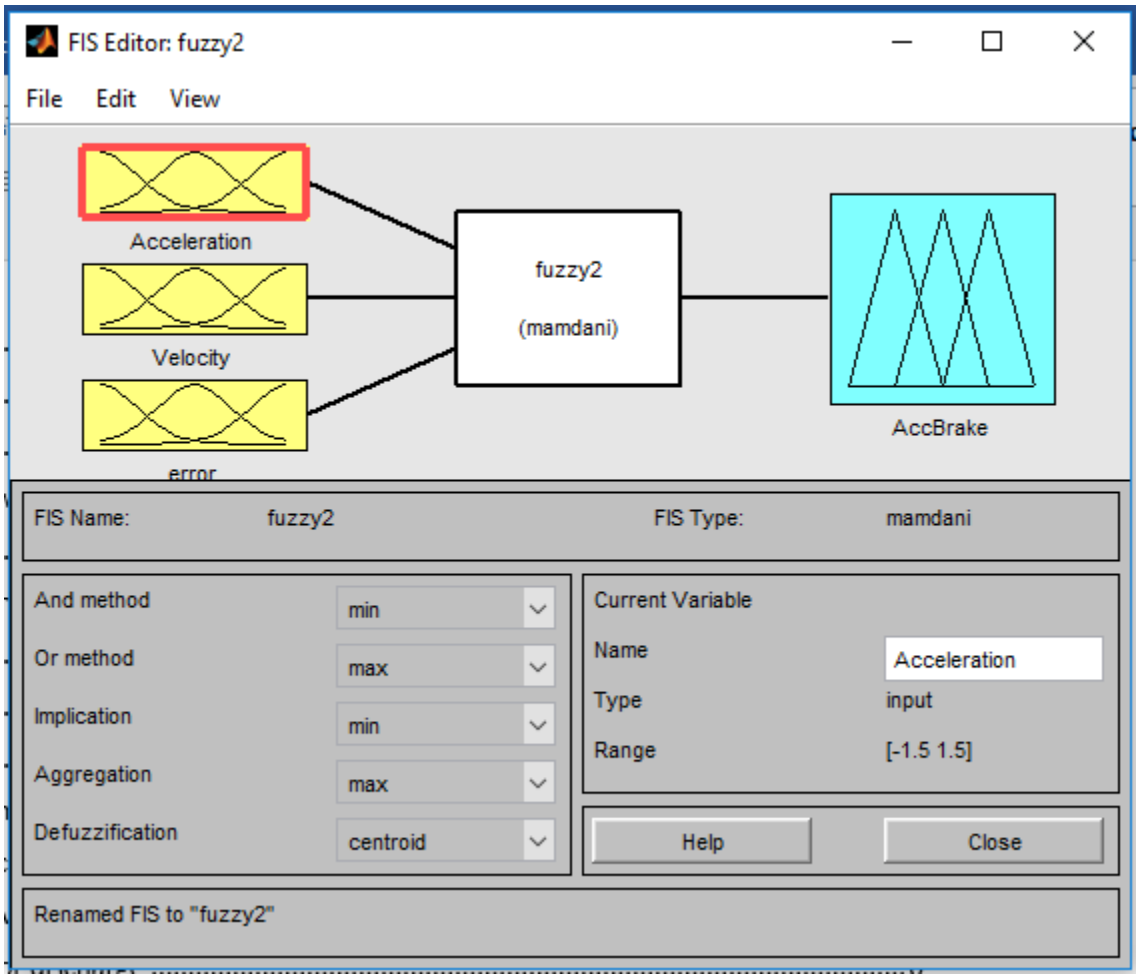
1. `Torque_traction.m`

2. `Driving_cycle.m`
3. `eff_hydrostatic_transmission.m`
4. `EM_eff.m`

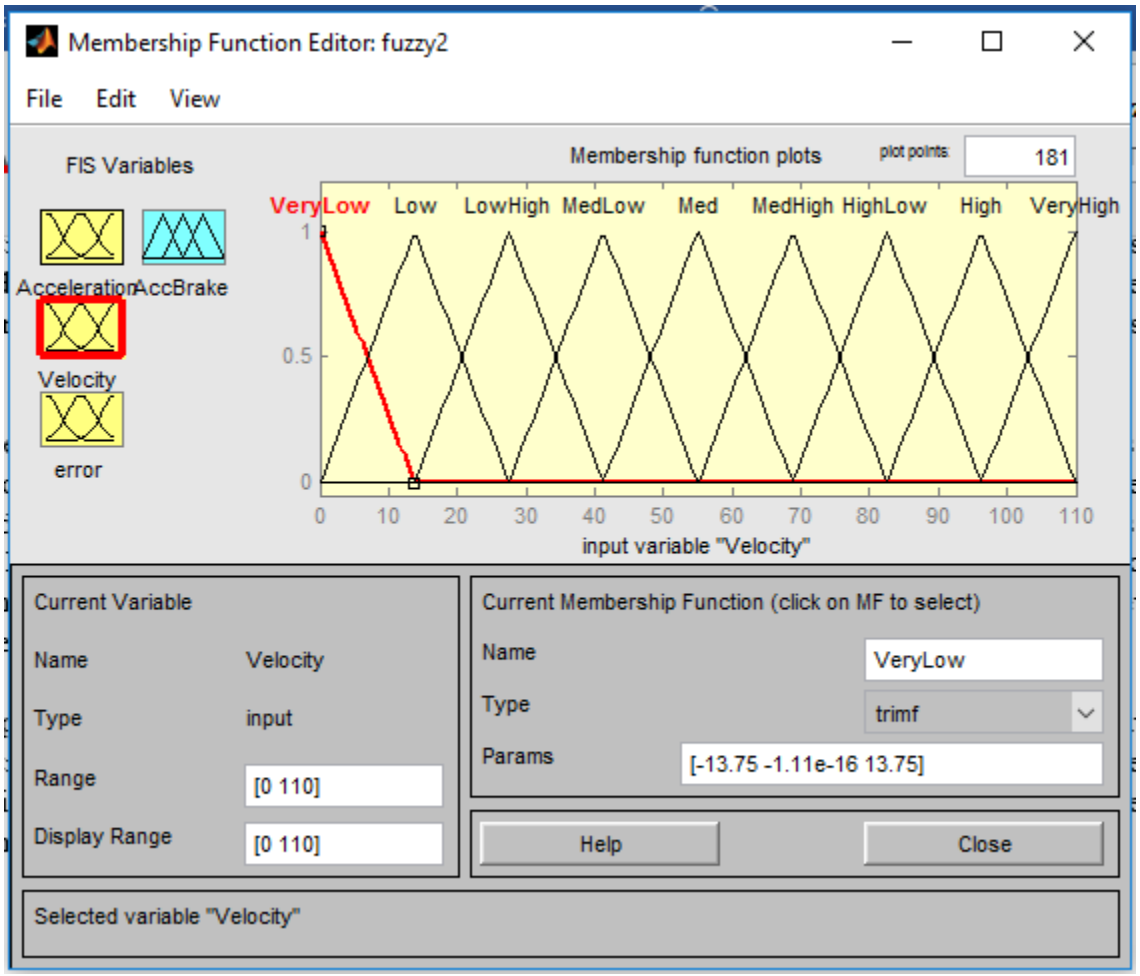
After that, it is necessary to open the fuzzy logic dialog box with the function `fuzzy` in the command window. The fuzzy logic dialog box has the next view.



In this dialog box, it is necessary to charge the files of the membership functions and the fuzzy rules. For doing that, click on File/Import/From File… and the file must be selected. When the file has been selected, the dialog box has the next view.



The inputs are on the left and the output is on the right. For modifying the membership functions, double click on one of the inputs. A new dialog box is open.



Once the fuzzy functions have been charged, the simulation can be run in Simulink. Before running these files, it is necessary to have the results for the study of the efficiency obtained with the file `optimization.m` and also it is necessary to have the solution of the pressure from file `sol_P_HM.m`.