DEVELOPMENT OF A FUZZY LOGIC GEAR SHIFTING IN AUTOMATIC TRANSMISSION FOR AUTOMOBILES

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ABSTRACT → chapter 1

DEVELOPMENT OF A FUZZY LOGIC GEAR SHIFTING IN AUTOMATIC TRANSMISSION FOR AUTOMOBILES

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This thesis is mainly concerned with the modeling and simulation of a Fuzzy Logic Controlled (FLC) gear shifting operation in an automatic transmission of an automobile by using enough number of inputs. During the design of the control system maximum fuel economy in normal driving conditions or maximum performance in aggressive driving conditions is considered for gear shifting operations.

A mathematical model of vehicle is constructed to realize the simulation of the Fuzzy Logic Controller (FLC) of the automatic transmission. The nonlinear vehicle model is prepared after the preparation of vehicle subsystems. Then, the FLC for automatic transmission is designed to cope for the requirements due to various road and driver inputs. In addition, the controller operates by eliminating failure risks that can occur on the engine, made by the driver. Finally, various case studies are performed in order to observe the reactions of the FLC under different driving conditions.

Keywords: Automatic Transmission, Fuzzy Logic Control (FLC), Torque Converter, Vehicle Model, Gear Shifting.

ÖZ

OTOMOBİL OTOMATİK VİTES KUTUSUNDA VİTES DEĞİŞİMİNİ SAĞLAYAN BULANIK MANTIK DENETLEYİCİNİN GELİŞTİRİLMESİ

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Bu çalışma, esas olarak otomatik vites kutusu olan bir otomobildeki vites değişimlerini sağlayacak olan bulanık mantık denetleyicisinin, yeterli sayıda girdi kullanarak tasarılanmasını incelemektedir. Bu tasarım yapılırken, vites değişimleri için, normal sürüş durumunda yakıt ekonomisi, sportif sürüş durumunda ise maksimum performans göz önünde bulundurulmuştur.

Otomatik vites kutusunu kontrol eden Bulanık Mantık Denetleyicisinin performansının bilgisayar ortamında incelenebilmesi için otomobilin bir matematik modeli hazırlanmıştır. Doğrusal olmayan araç modelini elde etmek için, otomobilin tüm alt sistemleri modellenmiştir. Daha sonra, otomatik vites kutusunu denetleyen Bulanık Mantık Denetleyicisi, değişik yol ve sürücü koşullarından kaynaklanacak çeşitli durumlar gözönüne alınarak tasarlanmıştır. Denetleyicinin çeşitli koşullar altında gösterdiği performans, değişik durumların gözönüne alındığı incelemelerle belirlenmiştir.

Anahtar Kelimeler: Otomatik Vites Kutusu, Bulanık Mantık Denetleyici, Hidrolik Kavrama, Araç Modeli, Vites Değişimi.

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a^*	Coefficient depending on Tire Inflation Pressure
A_f	Frontal Area of the Vehicle
b^*	Coefficient
C_D	Drag Coefficient
e_d	The Efficiency of Differential Gear
e_t	The Efficiency of Gears in Gearbox
f_r	Coefficient of Rolling Resistance
F_R	Total Resistances Acting over Vehicle
i_d	Differential Gear Ratio
i_t	The Gear Ratio of Selected Gear in Gearbox
J_e	Mass Moment of Inertia of Rotating Parts in the Engine
J_p	Mass Moment of Inertia of the Rotating Propeller Shaft
J_T	Total Mass Moment of Inertia of Rotating Masses
J_w	Mass Moment of Inertia of the Rotating Wheel
K_p	Pump Capacity Factor of the Torque Converter
m	Actual Vehicle Mass
m_{eq}	Equivalent Mass
$\acute{N}_e$	Angular Acceleration of the Engine
N_e	Angular Engine Speed
N_{out}	Angular Speed output from the Torque Converter
N_{tr}	Angular Speed of the Propeller Shaft in rpm
q	Dynamic Pressure
R_a	Air Resistance

R_{gr}	Gradient Resistance
R_i	Inertial Resistances
R_r	Rolling Resistances
r_w	Wheel Rolling Radius
SR	Speed Ratio of the Torque Converter
S_t	Slip Factor for Each Gear (t=1..5)
T_{BR}	Total Brake Torque acting on Wheels
T_e	Total Output Torque from the Engine
T_{out}	Torque Output from the Torque Converter
T_p	Total Torque coming in Torque Converter
TR	Torque Ratio of the Torque Converter
T_{tr}	Torque Output from the Transmission
T_w	Torque Acting on Wheel
V	Vehicle Speed of the Vehicle
W	Total Weight of the Vehicle
γ	Rotary Mass Factor
θ	Angle of the Slope
ρ	Density of the Ambient Air
ω_w	Angular Speed of the Wheel (1/s)

CHAPTER 1

INTRODUCTION

1.1. AUTOMATIC TRANSMISSION

Automatic transmission is widely used due to its simplicity and ease of use in driving. It gives the opportunity to improve and increase driving quality and safety. While driving a vehicle with a manual transmission, in case of vehicles with steering wheel on left, the driver should press on the clutch pedal by using the left foot and change the gear with the right hand, in the meantime the right hand leaving the steering wheel. Such a gearshift creates a physical burden on the driver, especially in heavy city traffic and lowers the level of drivers concentration. With an automatic transmission, the driver gains the freedom of his/her left leg and right hand. This freedom leads to improvement in concentration and eliminates the necessity of drivers right hand leaving the steering wheel, so the driver can hold it with two hands during entire driving. This also results in increased driving safety.

In conventionally controlled automatic transmissions, the controllers are designed to control a planetary gear type transmission with a torque converter. The shift schedule is controlled by two hydraulic pressures. While one hydraulic actuator controls the vehicle speed, the other controls engine output torque.

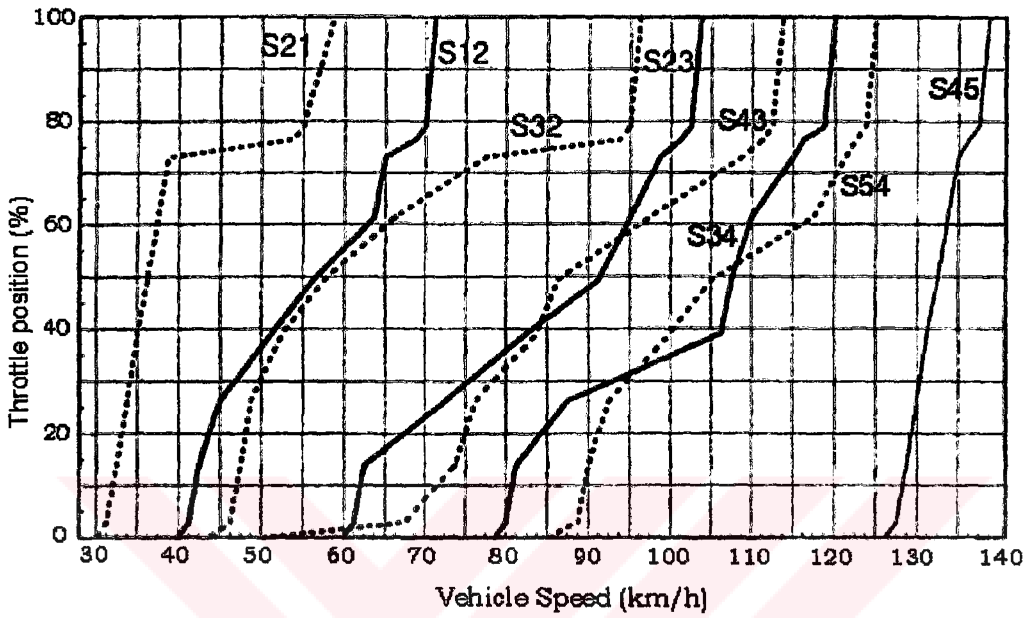

Figure 1.1: Conventional shift pattern [1]

This control system determines the required transmission gear ratio with a manually selected shift pattern that is previously determined. This shift pattern consists of curves, which define a shift operation according to the measured vehicle speed and throttle opening. Figure 1.1 shows a typical conventional shift pattern. In this figure, the solid lines show up-shifting operations for various output speeds and various throttles. The dashed lines show the downshifting operations occur for pre-specified output speeds and throttles. It makes possible to have better fuel economy that match the drivers intention and it allows the driver to select a shift schedule by means of a control switch. There are usually three shift patterns existing in today's automatic transmissions. However, with these patterns, it is very difficult to achieve optimum driving according to different driving conditions. The reason is that the shift patterns are not adapted to driving conditions automatically, but they should be selected manually. As it will be illustrated later, the vehicle speed and throttle opening are not sufficient to represent the effects of changing environment on driving conditions.

In various publications, different simulations about vehicles with automatic transmission are presented. Suzuki, Sakakiyama, Narumi [2] present the future trends of powertrain control that will improve fuel consumption, comfort and safety for the overall improvement of automobiles. In addition, they discuss various developments in powertrain control technologies for automatic transmissions, continuously variable transmission (CVT) and four-wheel drive. The discussion about automatic transmission subject deals with lock-up clutch control, the control of the shift quality that is provided by the use of hydraulic pressure, the integrated control and finally the future trends in the subject.

Danno, Togai, Fukui and Shimada [3] discuss the strategy and modeling of a powertrain control by drive by wire system, which is developed for Mitsubishi cars. The publication discusses the simulation steps to investigate the powertrain behaviors that are derived from a vehicle with automatic transmission. According to this simulation, the outline of a new powertrain model is presented. While dealing mainly with the control of intake air and throttle position, the publication also includes the torque converter and automatic transmission control strategy.

Hattori, Oshidari, Takatori, Iwanaga, Sugano, and Umebayashi [4] discuss a five speed automatic transmission for Nissan passenger cars use to be equipped with a torque converter. In this publication, the major features, structure and the performance of this transmission are described. It is also mentioned that this design with close ratio gearing is presented to provide excellent performance, smaller shift shock, and smoother drivability. It is stated that better standing start to high-speed acceleration performance and fuel consumption is achieved by wide range gearing.

In the publication [5], Fõrster discusses a few automatic transmission for Mercedes-Benz passenger car that will be widely used for all purposes and operating conditions. The main purpose of this is to create a basic statement for future progresses that will automate the production of gearboxes for these types of vehicles. Fõrster discusses automatic transmission by considering all the aspects and effects. Some general remarks on transmission design are made including torque converter properties; the usage of several automatic transmissions and their application according to vehicles classification are discussed. Their performances, fuel consumption rates, reliability, durability, and initial costs are all considered.

Bader [6] discusses the powertrain electronics applied for computer aided gearshift system used in Mercedes-Benz heavy-duty truck. The connection between the shift lever and transmission is provided by electronic control. The shift lever has only three directions that can be moved by the driver. These movements provide shifting up and down motions or neutral position. The gear selection is performed in full automation among

16 gears by not only controlling optimum shift sequences but also selecting individual shifts. Whenever it is necessary, the gears can be skipped for the right gear selection. In steady state condition, the gear shifting is performed aiming the fuel economy. In dynamic operation case, the gears are selected considering full power usage. Due to difficulty in gear selection within finely spaced gear ratios, a fixed relation of vehicle speed and gear step including the throttle position is considered. By using an engine torque versus engine speed diagram showing both engine power and fuel consumption rates, the operating points with minimum fuel consumption are primarily considered especially for steady state operation and shown as a curve. The region below this operating points curve is divided into several areas that show the operating points for 6 adjacent gears that can be chosen. One of these areas determines the optimum fuel consumption condition. The operating points area defined at the left of this area determines the shifting to lower gears. On the other hand, the area at the right determines up-shifting condition by indexing the number of gears showing how many gears that will be shifted up.

In vehicle powertrain simulation (VPS) discussed by Badgley, Phillips and Assanis [7], a powertrain of an army truck is developed considering low fuel consumption and to predict the performance. In this analysis, while the effect of all the components in powertrain is analyzed, the implementation of an automatic transmission alternative is also discussed. In this simulation, incremental time step simulation method is used. At the beginning of each time step, some calculations of coefficients is performed, which will link parameters like vehicle acceleration, engine acceleration, engine torque, torque converter input and output torques and brake torque. These coefficients vary depending on the resistances. In the transmission shift logic, two transmission output speeds for each transmission mode (torque converter and lockup clutch) are found for upshifting and two for downshifting cases. It is mentioned that the actual upshift/downshift point will be the lower value plus the latest throttle setting times the difference between the variable shift points for each mode. At each time step, the program checks whether the transmission output speed is within the variable limits or not. If not, then the new transmission mode is selected, and all the coefficients mentioned above are recalculated. It is stated that the program is applied several times to compare the effects on fuel economy and maximum performance of various operating conditions.

1.2. FUZZY LOGIC

Fuzzy logic has been gaining increasing acceptance during the past few years. There are more than three thousand commercially available products using fuzzy logic, ranging from washing machines to high-speed trains, temperature to satellite attitude control. Nearly in every application, some of the potential benefits of fuzzy logic can be realized, such as performance, simplicity, lower cost, and productivity. It provides a remarkably simple way to draw definite conclusions from vague, ambiguous, or imprecise information. In a sense, fuzzy logic resembles human decision making with its ability to work from approximate data and finding precise solutions [8].

Many engineering solutions are too precise leading to unnecessary time expenditure by the engineers and unnecessary cost in the product. For many engineering problems, on the other hand, high precision is not necessary. The difficult task of modeling and simulating complex real-world systems for control systems development is well known. Even if a relatively accurate model of a dynamic system is obtained, it is often too complex to use in controller development for many conventional control design procedures.

In classical logic, the system should deeply be understood, exact equations should be defined and precise numeric values have to be known. Unlike classical logic, fuzzy logic defines an alternative way of thinking that allows modeling complex systems using a higher level of abstraction that is originating from our knowledge and experience. In addition, it allows expressing this knowledge with subjective concepts such as very high, dark blue, and a long time that are mapped into exact numeric ranges.

1.3. REASONS TO USE FUZZY LOGIC CONTROL

Fuzzy logic has been found to be very suitable for embedded control applications. Several manufacturers in the automotive industry are using fuzzy technology to improve quality and reduce development time. In aerospace, fuzzy logic enables very complex real time problems to be tackled using a simple approach. In manufacturing, it is mostly used in scheduling of the production and in the deposition process control, but it is proven particularly invaluable in increasing equipment efficiency and diagnosing malfunctions.

In order to understand why a fuzzy based design methodology is very attractive in embedded control applications, it will be useful to examine a typical design flow with a figure comparing the conventional and fuzzy based design methodologies. Figure 1.2 illustrates the design step sequences required to develop two different controllers one using a conventional and the other using fuzzy approach.

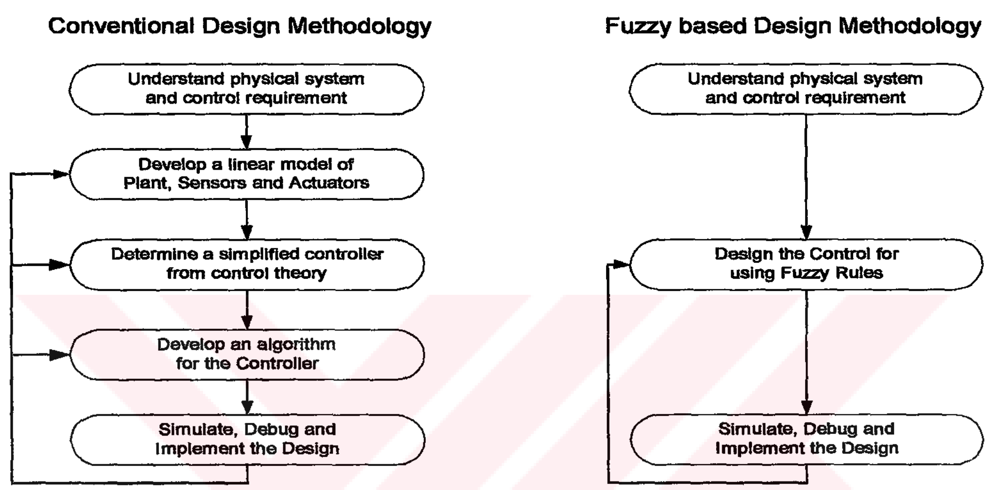

Figure 1.2:Conventional and Fuzzy Based Design

Using the conventional control system, the first step is to understand the physical system and its control requirements. Based on this understanding, the second step is to develop a model that includes the plant, sensors, and actuators. The third step is to use linear control theory in order to determine a simplified version of the controller, such as the parameters of a PID controller. The fourth step is to develop an algorithm for the simplified controller obtained in the previous step. The last step is to simulate the design including the effects of non-linearity, noise, and parameter variations. If the system does not show the wanted behavior or the performance is not satisfactory, repetition of the second, third, and fourth steps is in order. In other words, the system modeling is modified, the controller is re-designed, and the algorithm is re-written and retried.

With fuzzy logic, the first step is to understand and characterize the system behavior by using human's heuristic knowledge and experience. The second step is to design directly the control algorithm using fuzzy rules, which describe the principles of the controller's regulation in terms of the relationship between its inputs and outputs. The last step is to simulate and debug the design. If the performance is not satisfactory, modification of only some fuzzy rules is usually sufficient and then the system control is retried.

Although the two design methodologies are similar, the fuzzy-based methodology simplifies the design loop. This result in some significant benefits, such as reduced development time, simpler design, and faster time to market.

Most real life physical systems are actually non-linear systems. Conventional design approaches use different approximation methods to handle non-linearity. Some typical choices are, linear, piecewise linear, and lookup table approximations to trade off factors of complexity, cost, and system performance. A linear approximation technique is relatively simple, however it tends to limit control performance and may be costly to implement in certain applications. A piecewise linear technique works better, however it is tedious to implement because it often requires the design of several linear controllers. A lookup table technique may help improve control performance, but it is difficult to debug and tune. Furthermore, in complex systems where multiple inputs exist, a lookup table may be impractical or very costly to implement due to its large memory requirements.

Fuzzy logic provides an alternative solution to non-linear control because it is closer to the real world. Non-linearity is handled by rules, membership functions, and the inference process, which results in, improved performance, simpler implementation, and reduced design costs.

In many applications, fuzzy logic approach can result in better control performance than linear, piecewise linear, or lookup table techniques. With fuzzy logic, one can use rules and membership functions to approximate any continuous function in any degree of precision. Using four points (or four rules), one can approximate the desired control curve of a variable in controller. More rules can also be added to increase the accuracy of the approximation (similar to a Fourier transform) that yields an improved control performance. Rules are much simpler to implement and much easier to debug and tune than piecewise linear or lookup table techniques.

IF temperature IS *cold* **THEN** force IS *high*

IF temperature IS *cool* **THEN** force IS *medium*

IF temperature IS *warm* **THEN** force IS *low*

IF temperature IS *hot* **THEN** force IS *zero*

Rules are not like a lookup table because the fuzzy arithmetic interpolates the shape of the non-linear function. The combined memory required for the labels and fuzzy inference is substantially less than a lookup table, especially for multiple input systems. As a result, processing speed can be improved as well.

In the design of a fuzzy controller, one must first identify the main control variables. The second step is the determination of a term set of linguistic variables that is suitable for describing the values of each variable. For example, a term set including linguistic values such as {*Small, Medium, Large*} may be satisfactory in some cases; whereas other cases may instead require the use of a higher term set such as {*Very Small, Small, Medium, Large, and Very Large*}.

After the linguistic term sets for the main control variables are determined, a knowledge base is developed using these control variables and the values that they may take. The knowledge base used presently is a rule base and more than one rule may be defined simultaneously;

Figure 1.3 illustrates the basic architecture for a fuzzy logic controller. This architecture consists of four modules, which are fuzzification, fuzzy inference, fuzzy rules and defuzzification.

The fuzzy operations that are performed in fuzzy logic controller can be learned from [8] in detail. However, for the present case, it is better to mention briefly the procedures that are performed in FLC. In the *fuzzification module,* input parameters are transformed to grades of memberships of linguistic states with the help of defined membership functions. Grades of membership are the information that the inference mechanism can easily use and apply rules.

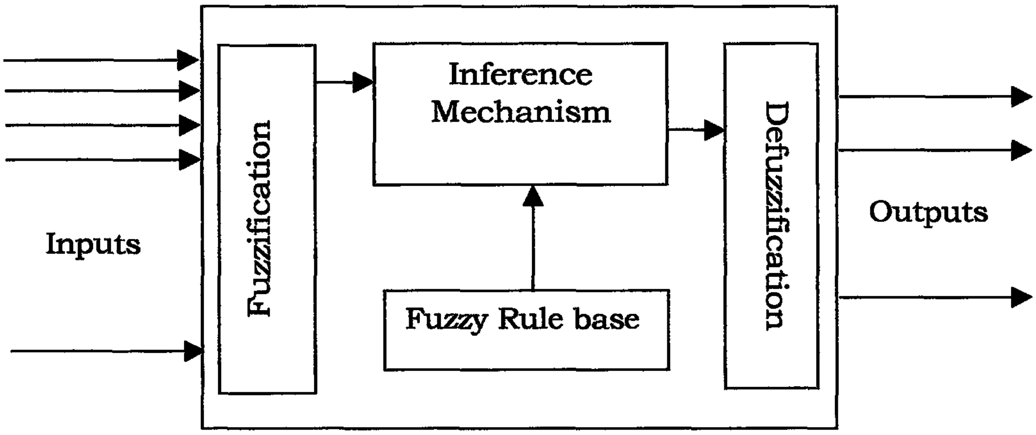

Figure 1.3: Architecture of Fuzzy Logic Controller

After transforming the crisp data to the fuzzy truth-values (membership grades), these values are processed by the inference mechanism.

An *inference mechanism,* working with predetermined rules, calculates each rule relevance and conclusion for the linguistic output variables. First, all the rules are compared to the controller inputs to determine which rules can be applied to the current situation. Secondly, the conclusions (what control actions to take) are determined using the rules that have been applied at the current time. These rules actually describe the behavior of the controller in the system. The conclusions are characterized with linguistic states of output variables.

The resulting data are brought into defuzzification module. The transformation is performed by *defuzzification module* from linguistic states to output signals that will be meaningful for the system. This phase of FLC performs an inverse procedure of fuzzification module.

The output of each rule reflects the relevance for at least one linguistic output variable. Each membership function belonging to a conclusion is weighted with the related rule relevance. The defuzzification uses all weighted membership functions of number of output to calculate the output signal by computation using several methods defined like center of gravity, center-average method or mean of maximum, etc.

1.4. APPLICATIONS OF FUZZY LOGIC

With the rapid development of electronics and the increasing demand for higher performance, fuzzy systems have been used widely in applications of engineering, science, business, medicine, psychology and other fields. In engineering applications, the main potential areas include the following:

- *Aircraft/Spacecraft:* Flight control, engine control, avionic systems, navigation, failure diagnosis and satellite attitude control, satellite position control

- *Automated highway systems:* Automatic steering, braking and throttle control for vehicles, and the control of the following distance

- *Automobiles:* Brake control, engine control, traction control, transmission control, suspension control, and cruise control

- *Autonomous Vehicles:* The control of ground and underwater vehicles

- *Manufacturing Systems:* Scheduling and deposition process control

- *Power Industry:* Motor control, power control and power distribution, and load estimation

- *Process Control:* Temperature control, pressure control, level control, distillation column control, desalination processes control, and failure diagnosis

- *Robotics:* Position control and path planning

Actually, these are only some examples that are applied among huge range of possible applications.

1.5. APPLICATIONS OF FUZZY LOGIC IN AUTOMATIC TRANSMISSION

There have been a relatively limited number of publications related to fuzzy control of automatic transmissions.

Tanaka and Wada, [9] presented a fuzzy control system for clutch engagement of an automated manual transmission. The fuzzy system estimates the driver's intention from the accelerator pedal operation. The system parameters are determined by bond graph simulation and performance tests are carried out. It is stated that the system can also be used for mechanical Continuously Variable Transmissions (CVT) in near future.

Bastian [10] presented some applications of fuzzy logic in automatic transmission control of automobiles and the usage in different automatic transmission control is summarized. The paper presents some particular cases of transmission control during normal driving before an approaching curve. The construction of a controller is performed for both cases of using and not using road information. It is also stated that the road information will be useful to simulate more human-friendly transmission and the importance of the adaptation of the controller to the driver individually. This transmission control unit is used to determine the optimal shift strategy that adapts the driver and some environmental conditions. This knowledge-based system is derived form the human expert knowledge of application engineers. Therefore, the implementation was made easier with the usage of fuzzy logic methodology and this system neglected the necessity of defining a corresponding mathematical model.

Yamaguchi, Narita, Takahashi, and Katou [11] presented in their paper an automatic transmission used in Nissan cars with some modifications since the model year of 1983. At the beginning, this electronically controlled automatic gearbox sets the shift schedule according to vehicle speed and throttle valve opening. Although it shows an improvement over hydraulically controlled ones, shift haunting occurs when driving uphill or towing an object. Therefore, a technique based on the use of fuzzy logic, is developed for estimating running resistance that is so called road gradient. The researchers aim to have a fuzzy control system capable of detecting driving conditions and the driver's intentions under a wider range of conditions based on the driving inputs. The research is mainly performed to approximate the driver intention in a winding road with uphill and downhill gradients. For this purpose, histograms of the throttle valve opening and its variance were prepared using data recorded for tens of drivers.

1.5.1. IMPLEMENTATION OF FUZZY CONTROL TO VEHICLE AUTOMATIC TRANSMISSIONS

As it can be found widely in industry and household applications, it is also usual to find widely the applications of fuzzy logic technology in automatic transmission control. This methodology was used in every type and class of cars in automotive industry. However, compared to others, Japanese automotive industry has been actively pursuing the application of fuzzy logic technology not only in automatic transmission control, but also in all the parts that requires control. The big Japanese automotive companies like Nissan, Honda, Mazda, and Mitsubishi have applied the fuzzy logic to their automatic transmission controller of their brand of car from the luxury class to the lower class vehicles.

1.5.1.1. INVECS

The fuzzy shift controller of INVECS that is developed for Mitsubishi cars is discussed in the research of Yamada, K., Yoshida, H., Hayafune, K., Hatta, K., Yoshida, S., at 1992 [10] and revised at 1996 by Usuki K., Fujita K., and Harada K. [12]. This provides easy driving ability on both ascending and descending roads. This controller uses eleven input variables to reflect the driver's intention, the road conditions and the vehicle state. The road condition is expressed as 'gradient resistance' used to determine the decline and incline of the road and 'winding rate' representing the degree of the road curvature. The two inputs and the others can be seen on the Table 1.1, below.

Table 1.1: Used Inputs in INVECS

Driver's intention	Road condition	Vehicle's state
• Throttle opening (TO) • Rate of change in TO • Speed reduction due braking	• Gradient resistance • Winding rate of the road	• Speed • Acceleration • Speed Change • Surplus engine torque • Lateral accel. • Steering wheel angle

By employing 11 fuzzy rules, the fuzzy controller selects one from five available shift modes. These shift modes are "level road", "winding road ascending", "high-speed ascending", "medium and low-speed ascending", and "descending" modes. Fuzzy logic is used to make a decision to select an appropriate shift mode by using a set of rules.

1.5.1.2. PROSMATEC Type 2

A decomposition of the shift control problem was performed in this example. The fuzzy logic in the automatic transmission chooses the suitable mode among four modes of drive. These are uphill, downhill, level road, and deceleration shift mode introduced in *"Introduction of new Domani"* discussed by Saito, M., Yoshida, Y., Watanabe, K., and Innami, Y. [13]. In this application, the usage of fuzzy theory is completely different. Fuzzy logic is directly applied to the uphill shift mode to enable a continuous change of the third and fourth gearshift pattern, because every driver has the experience to decide the shift timing according to the road inclination. So, the gearshift line is moved according to the inclination.

The inputs that are used for the controller are:

- Accelerator pedal displacement,
- Vehicle speed,
- Shift position,
- Engine speed,

- Brake signal, and
- Coolant water temperature

1.5.1.3. PROSMATEC Type F

Although the name of this example resembles the system name discussed above, this system is a completely different application of fuzzy theory to automatic transmission control. The controller only employs one shift map, and the gear selection based on this map is modified using fuzzy rules. [10]

To achieve human like behavior, the architecture for the automatic transmission controller is constituted by some modules. These modules are:

- Input module,
- Inference module,
- Knowledge base module, and
- Output module

In the input module, these following inputs are processed. These are accelerator pedal displacement, Vehicle speed, Shift position, Engine speed, Brake signal, and Intake manifold pressure.

The inference module consists of two blocks to present driver's intention and gear position respectively. The driver's intention, in this case a measurement of how much the driver wants to slow down, is determined using the input information and the knowledge base module. In the gear position block, the modification of the gear position determined by the shift map is determined using the information from the driver's intention block, the input module and the rules in the knowledge base module. The knowledge base module consists of two rule bases. These are rules for referring driver's intention and rules for inferring gear position. In addition, the road resistance parameter is calculated and it is utilized in the knowledge base module.

1.5.1.4 Statistical Approach

It is an alternative method of employing fuzzy logic for automatic transmission controller [10]. Fuzzy logic rules and membership functions are created by a statistical analysis of a large amount of data (driver's accelerator inputs, vehicle speed variation, braking inputs, steering inputs, and the road conditions). Based on histograms of the accelerator inputs and variances per unit of time, a distinction between inclined or level road can be made. As a result, trapezoidal membership functions of the environment are derived from these histograms.

The inputs of this automatic transmission control are

- Moving average of the accelerator pedal displacement
- Moving variance of the accelerator pedal displacement
- Vehicle speed

The result is compared with a predetermined value and a judgment about the environment is made using the basis of the fuzzy rules.

1.5.1.5 Toyota ECT-I

This is a transmission used in Toyota cars discussed by Kondo, Iwatsuki, Taga, Tanguchi and Taniguchi [14], which employs an engine and transmission integrated fuzzy control system. This system also detects the engine torque and clutch pressure during shifting. The design objectives of this control system are to provide a smooth and silent transmission that are in high priority. In addition, high efficiency, reduction of changes over system life and high reliability. The electronic control unit uses the data from the sensor and gives orders to the actuators for necessary operations. The input variables are engine speed, throttle position, pattern selection switch, neutral start switch, overdrive switch, vehicle speed, transmission speed that provides the detection of completion of each shift, kick-down switch that shows the driver's downshift intention and stop light switch. The outputs that controls the actuators are the orders controlling the clutch press, lock-up control for torque converter, engine ignition control and shift control. After the design of this system, a computer simulation is developed for the transmission control to observe the optimization and the performance of the system.

1.5.2 IMPLEMENTATIONS OF IMPROVED FUZZY AND ADAPTIVE SYSTEMS

1.5.2.1 A Neuro-Fuzzy System

In this approach, a back-propagation neural network is employed to determine the optimal gear position of an Isuzu NAVi-5 transmission system. [10]

The inputs of this neural network are;

- Automobile load
- Driver's intention
- Accelerator position displacement
- Vehicle speed

Among those inputs, only the accelerator position displacement and the vehicle speed can be measured. The other inputs are determined using fuzzy logic rules. The fuzzy rules for calculating the automobile load uses the accelerator displacement, the input shaft speed and its rate as inputs. The rules for obtaining the driver's intention are using the accelerator position displacement and its rate of change as inputs.

1.5.2.2 An Expert System

The controller of this system consists of a rule base system in which 29 rules are obtained either from the human operator intention, or from the vehicle dynamic model. This approach uses seven inputs: Throttle opening, change of throttle opening, gear position, brake, engine acceleration, drive shaft revolution, and speed. [10]

The controller has three outputs:

- Load detection
- Driver classification
- Gear shift authorization

The first two outputs are used in selecting a suitable gear-shifting pattern, and the last output issues a final decision in this gear selection.

The load detection is performed to recognize the environment. Different vehicle load states like mountain, valley, and plain are possible. These load states are obtained using fuzzy logic rules.

1.5.2.3 A Hierarchical Expert System using Road Information

Considering the obtained above solutions, to improve automatic transmission control performance, it is better to include the environmental effects as an additional input. [10]

Initially, the vehicle position on the road and the road map are known parameters. The inputs for this system are the distance to the next curve, vehicle speed, actual gear position, acceleration of the vehicle, accelerator pedal depression, and curve radius. For the design of this control system, the human knowledge is included. For this purpose, the driving behavior of two different drivers at the same road with some curves are observed and they are classified according to their driving styles. Therefore, considering these observed data, the hierarchical expert system is constructed. The construction is performed layer by layer to clarify the analysis. The first layer describes the vehicle position on the road, the second layer includes the driver's type and driving styles and it is based on the vehicle position. In third layer, the membership functions of the rules referring the driver's intention are selected and adjusted. With the analysis of this information, the shift authorization output and the driver classification are obtained as outputs.

1.6. CONCLUSION

Although, there are many applications of fuzzy logic controllers in automotive transmissions in literature with various names, the basic structure of all these systems is about the same. While some uses many inputs to obtain a more accurate and efficient system, some uses fewer inputs, which may also provide reasonably good reliability and efficiency. By analyzing all these systems, one can reach the conclusion that the use of a high number of inputs gives better results in transmission functionality and accuracy. However, the input variables that are commonly used almost in all systems are vehicle speed, engine speed and the throttle input. Further, the road information is very useful to realize more human based transmission controller, so it is better to include this data into the controller.

An intelligent control module should take the environmental effects and the drivers intention into consideration.

Thus, it has been decided to design a fuzzy logic controller for use in automobile transmissions, which will limit the input variables to vehicle speed, engine speed, and the throttle input and will take the road effects into account.

CHAPTER 2

VEHICLE MODEL WITH FLC AUTOMATIC TRANSMISSION

2.1. INTRODUCTION

The block diagram of a car equipped with a basic automatic transmission with fuzzy logic control is illustrated in figure 2.1. Generally, the fuzzy logic controller plays the role of an observer that selects the shifting strategy after evaluating the inputs. For this study, the fuzzy controller receives four input signals:

a) Vehicle speed,

b) Engine speed,

c) Engine load (road information), and

d) Throttle opening

and produces one output signal, which is the selected gear number.

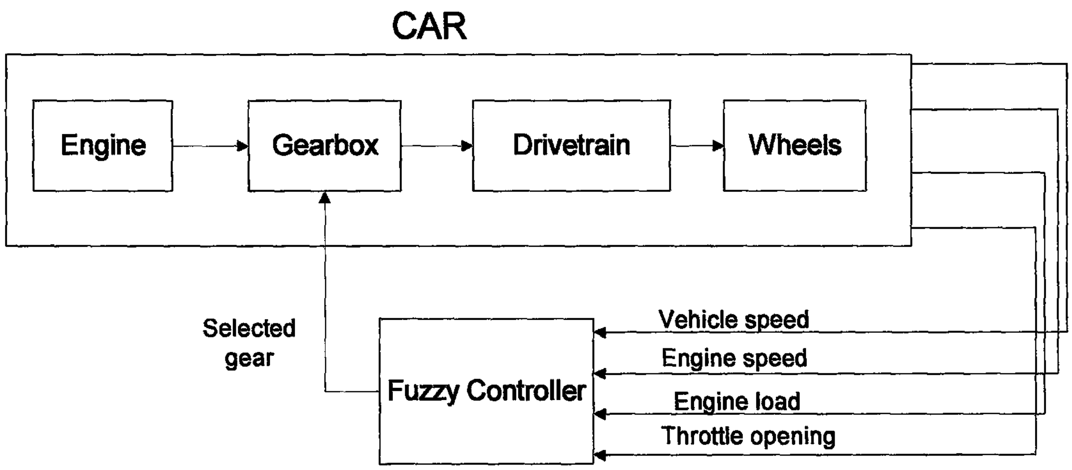

Figure 2.1: Automatic transmission system with FLC

The details of working process of a fuzzy logic controller shown in figure 2.2 were briefly discussed in previous chapter. Apart from the operation procedure, in present case, it is seen that there are four inputs that will be used and one output that will be get. Therefore, by knowing the procedures inside the controller, the further information about fuzzy logic controller will be discussed for automatic transmission case.

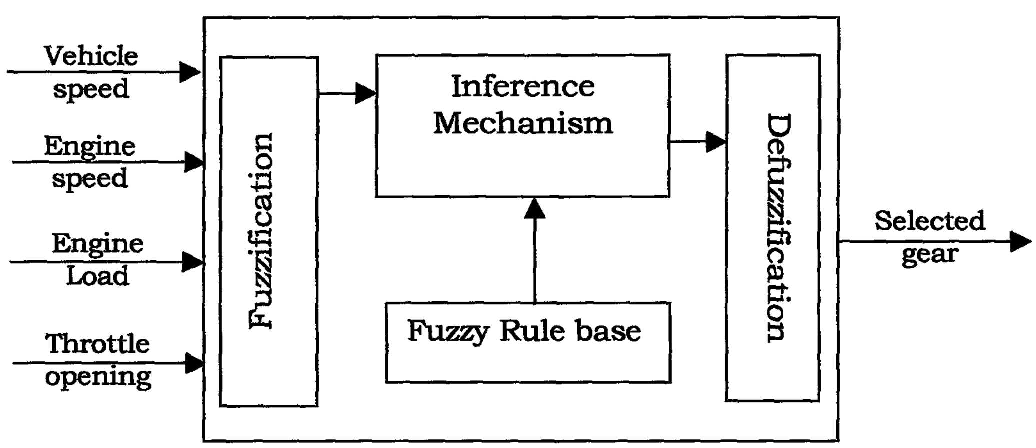

Figure 2.2 Block diagram of Fuzzy Logic Controller

Therefore, to use a fuzzy logic controller, one first needs to define membership functions for input and output variables. For this purpose, first, some referential fuzzy sets for each membership function of input are formed within pre-assumed input ranges. For each linguistic variable, it is assumed to have five referential fuzzy sets and it is arranged that they divide the input range of each input into equal pieces. This determination provides initial states for membership functions, which will surely be modified later by performing trial and error procedures.

While performing a trial and error procedure for a fuzzy system, some significant features of the rule base should be proved that the fuzzy control functions effectively. These features are completeness, consistency and interaction of the rule bases. To analyze these features, it will be better to state their definition.

A fuzzy model is said to be complete if a certain state of the output can be assigned to any state of the inputs. In other words, basically if each of the numerical inputs' states activates at least one rule, then the rule base is said to be complete. The consistency of a fuzzy model is another feature that should be handled. If there are at least two rules with same premises but with different conclusions, then the rule base is said to be inconsistent. In literature [15], a basic relationship is given to check completeness and consistency in a rule base simultaneously. For this purpose, the sum of membership degrees of an input state is checked. If the sum of the degrees for the premises of all satisfying rules for any input is equal to 1, then the rule base is both complete and consistent. If any sum for any input is greater than 1, than the rule base is incomplete. If it is less than 1, it is inconsistent. Therefore, this check is made for all inputs' membership functions.

It is not necessary that a fuzzy controller is complete. It can also be incomplete when the states of a model not mentioned in the rules do not occur in practice. However, it is risky to have an incomplete fuzzy controller. For this purpose, it has to be examined if non-defined states occur under real conditions or not.

The third feature is different in basis. It defines the continuity relation between two rules, by determining whether the rule base is continuous or not. This feature is checked by using the 3-D plots observable in SIMULINK [24, 25].

For the first input of vehicle speed, it is initially assumed to have five referential fuzzy sets (as stated above). The membership function's shape is selected as triangular, which is thought initially to be suitable. After many trials, it is concluded that it will be better to have more than five fuzzy sets (see figure 2.4). To emphasize the membership functions, it is observed that at low vehicle speeds, the fuzzy sets cover smaller number of inputs so the functions are closer and as the vehicle speed increases, the functions becomes wider. Therefore, at high speeds, fuzzy sets cover larger interval of inputs. For instance, there are three fuzzy sets covering an input interval of [20,60], while one fuzzy set can cover the interval [110,170]. The reason for that is actually to catch a better gear shifting selection at low speeds. For high speeds, only two gear numbers (the fourth gear and the fifth gear) can be selected. However, all of five gear numbers can be selected at low speed to increase the performance or comfort and fuel economy during the drive. Especially, for the vehicle speeds between 30 and 50 km/h, and around is important and for specified range the smallest interval that is covered by fuzzy subset is obtained. While using high number of fuzzy sets, this leads to the requirement of more fuzzy rules.

For the second input of engine speed, it is decided to have four fuzzy sets, which will cover the entire engine speed range. It is observed that after many trials, instead of having five referential fuzzy sets the number of four is sufficient that leads to elimination of extra rules. The membership functions are selected in trapezoidal shape, which enable to have full certainty (when the membership function is equal to one) for some intervals of each fuzzy set of input. Figure 2.5 shows the membership function of engine speed variable. This trapezoidal shape of membership suits better for the input of engine speed. With this shape, 100 % certainty is set for a range of engine speed and that provides easier selection for controller. After the setting of linguistic values 'high' and 'low' that determine lowest and highest engine speeds where the engine efficiency is lower, the other two membership functions are placed according to the obtained result of many trials and researches in literature.

The research in literature [1][11] shows that drivers who desire a comfortable drive and want economical ride, mostly prefer a slow driving and press to throttle pedal carefully. Therefore, the gear shifting operations are decided to occur at low engine speeds for low throttle values by referring to these statistical results. In addition, in following figure, the fuel consumption rate is given with output torque curves with corresponding throttle inputs. Although, the figure obtained [11] is represented differently from widely used specific fuel consumption maps, it shows the relation of consumption with various throttle inputs and corresponding engine speed and torque.

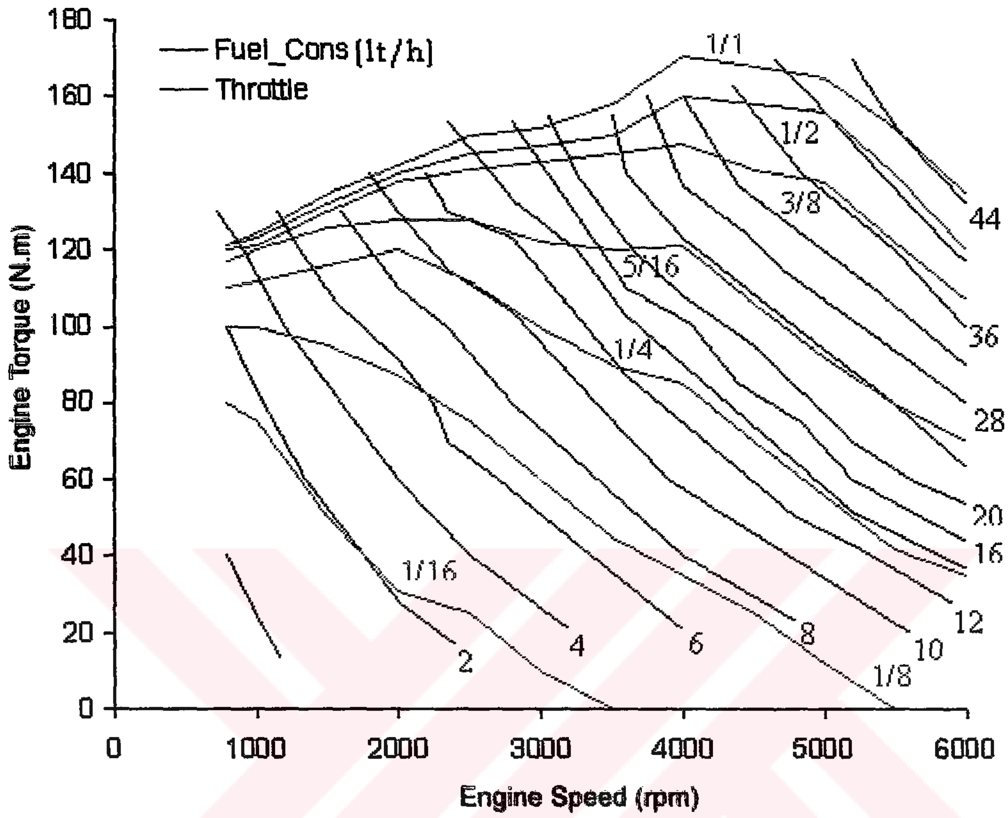

Figure 2.3: Representation of fuel consumption with various throttle [11]

At a specific engine speed, the fuel consumption differs for various throttle inputs. From the 1/16-opened throttle to full throttle input, the fuel consumption rate increases. For instance, while at 4000 rpm the fuel consumption for 1/8 throttle input is around 7 liter/h, this rate is 28 liter/h for half opened throttle valve. Moreover, while at low throttle inputs, the increase in engine speed does not lead to increase in fuel consumption. However, at high throttle inputs, the fuel consumption shows drastic increase while increasing engine speed.

Therefore, the driver that wants economic driving or comfort driving drives the vehicle with low throttle input, which is described within fuzzy logic controller. For instance, as the throttle input is 1/16 the vehicle consumes a maximum of 3 liter/h, however, as the engine speed increases, obtained torque output decreases. While the throttle input is kept under quarterly opened throttle, the critical speed is observed as 3000 rpm. Although the fuel consumption for these throttle inputs does not increase highly, but the engine torque output decreases. Then this result shows poor engine torque while driving, which will result with wider throttle opening by driver. With ¼ throttle opening, the vehicle fuel consumption is approximately 10 liter/h at that engine speed. Therefore, the gear shifting strategy for economic driving is the shifting the gear mostly at 3000 rpm.

The decision of gear shifting at high engine speeds is taken for who want to drive at high performance. With general intentions, an aggressive driver wants to use the vehicle at its limiting cases and depresses to throttle pedal totally. Moreover, its desire is to use high engine speeds by hearing high-speed engine sound. For high performance results, it is known that the driver should drive in an engine speed interval. The lowest speed of this interval is the engine speed that maximum torque is obtained and the highest is that maximum power is released. It is seen in figure that for full throttle, the maximum torque is obtained at 4000 rpm. For highest speed, power graph is required. It is projected from known torque graph and engine properties, maximum power can be predicted as 5500 rpm approximately. In addition, this equilibrium is formed at starting from 4000 rpm to highest 5500 rpm for used engine model. Therefore, for high throttle input, fuzzy controller decides to shift the gears at high engine speeds that will provide better acceleration results.

Therefore, It is observed that important engine speed interval is 4000 rpm to 4200 rpm that represent the boundary points of two fuzzy sets.

For the third input of engine load, it is initially set to have five referential fuzzy sets on a selected range of engine load [0 1]. However, these fuzzy sets undergo some changes on the mapped interval, and for some of them it is better to use trapezoidal membership function. In addition, it is also observed that the predetermined universe of discourse cannot fulfill the desired working conditions; the input range is enlarged from [0 1] to [0 1.5]. According to the definition of engine load, the input values can be greater than one, which means that the total resistance torque is greater than the provided engine torque. While determining the input range, main observation is obtained from the trials and so, it is decided to increases the highest number of the selected range from 1 to 1.5 that will be sufficient (see figure 2.6).

Another remark about this figure is the presence of differently shaped membership functions. While the trapezoidal shape is used for some linguistic values (fuzzy sets), for other membership functions of fuzzy sets, the triangular shape is preferred. For some fuzzy sets, it is better to use triangular one when to have a full certainty at one point is sufficient.

For the throttle input, four referential fuzzy sets are initially selected. However, it is observed that three fuzzy sets are sufficient to describe the system sufficiently. Therefore, these three fuzzy sets are adjusted according to the system behavior and of course according to the relation between other inputs after performing many trials with the system. Figure 2.7 shows the membership function of input variable of throttle.

The output variable *Shift* planned to be presented with singleton membership function. This means that only for one input value, the gear number is considered. This is presented with triangular membership functions having very narrow bases. Therefore, it is aimed that the shifting operation has to be kept around constant gear numbers during defuzzification. (Figure 2.8)

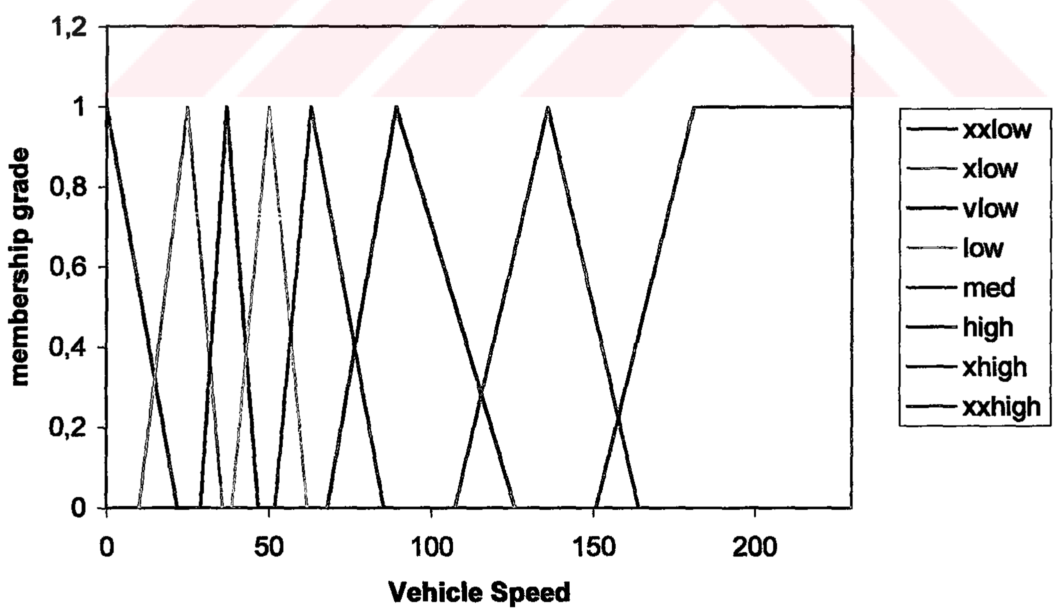

Figure 2.4: Membership function of Vehicle Speed

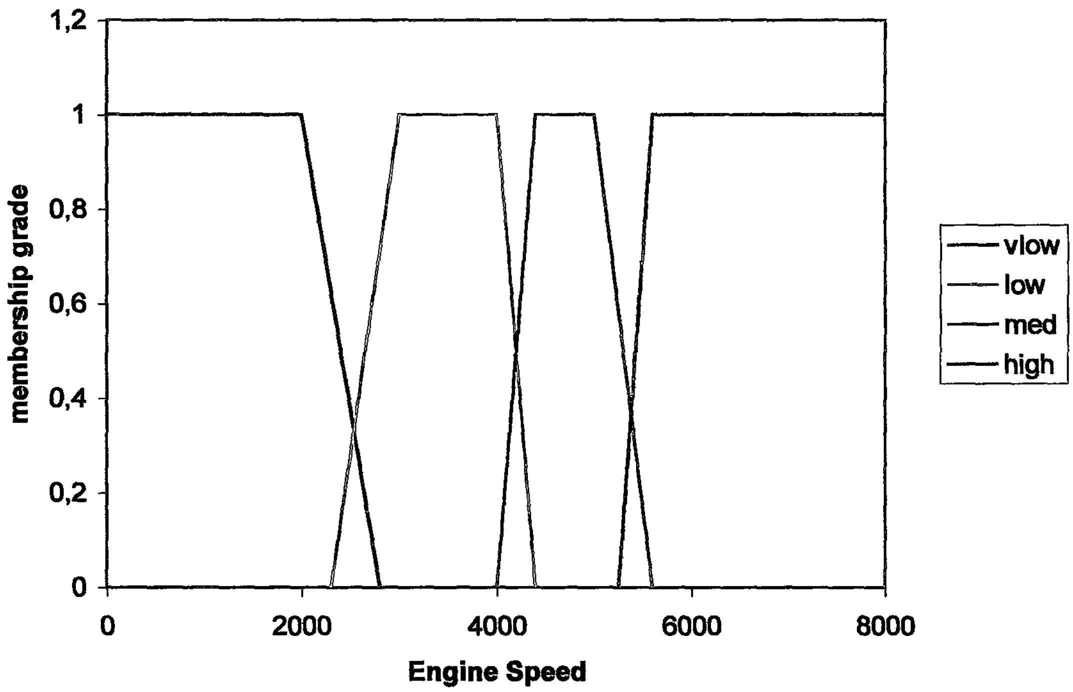

Figure 2.5: Membership function of Engine Speed

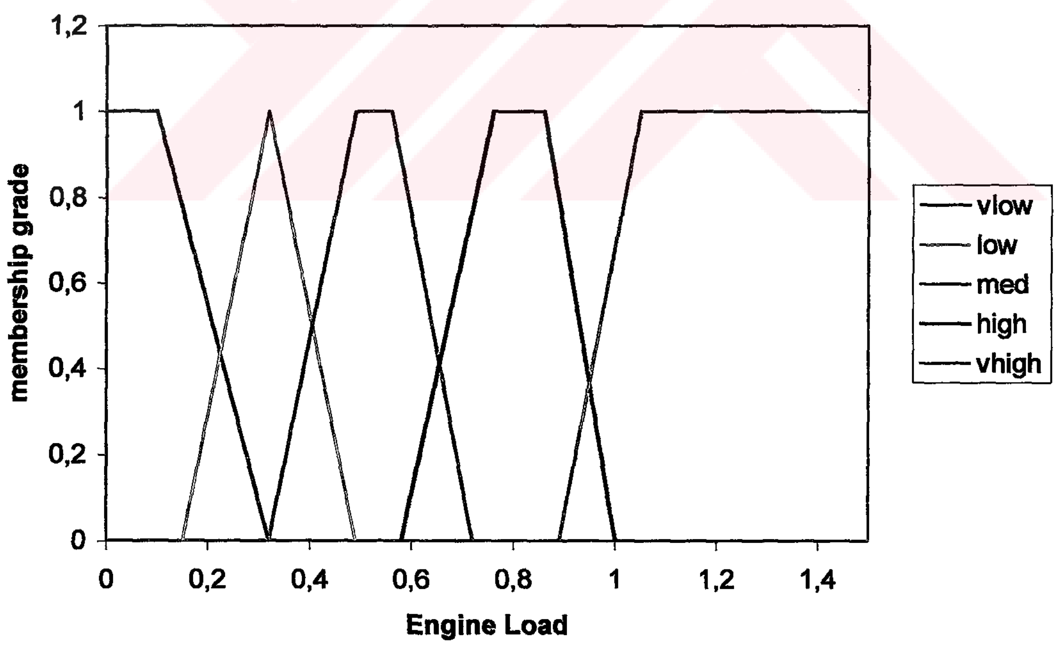

Figure 2.6: Membership function of Engine Load

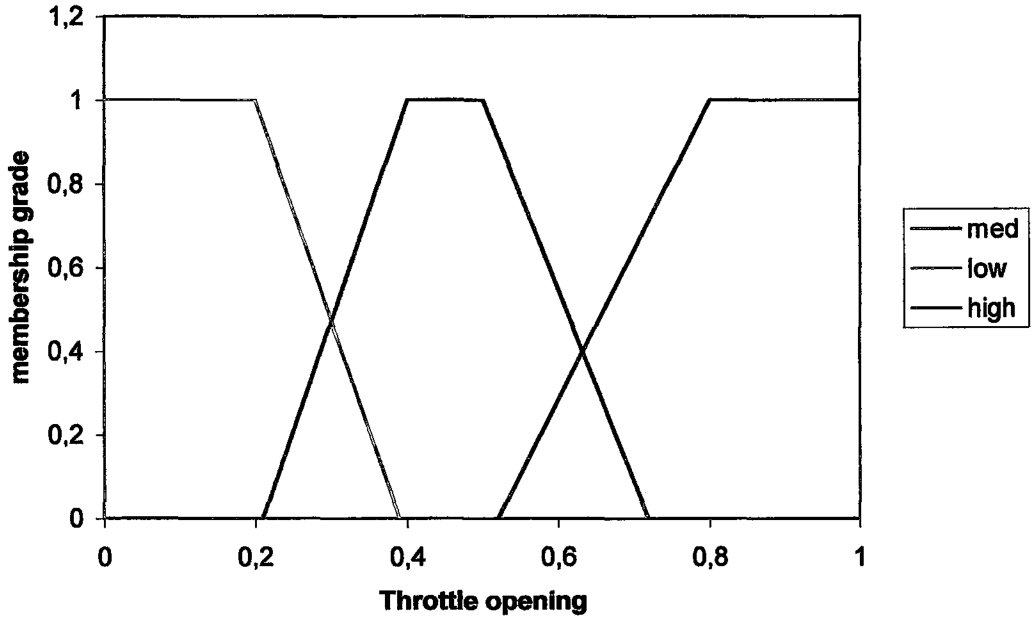

Figure 2.7: Membership function of Throttle Opening

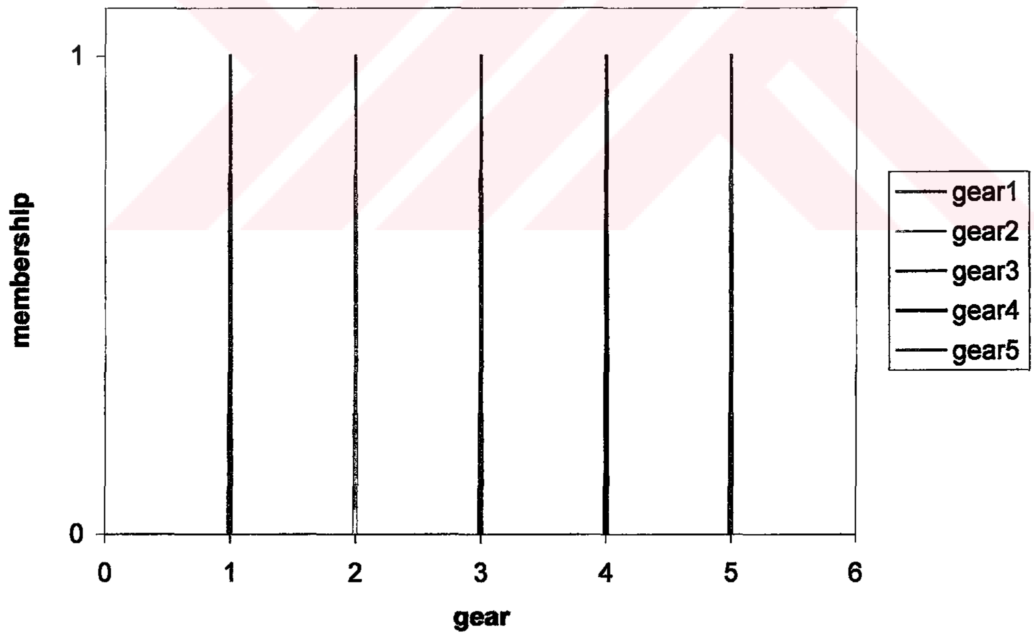

Figure 2.8: Membership function of Selected Gear Output

To obtain the value for the output variable using fuzzy logic controller, according to the input membership functions, the input variables must first be obtained. To be able to produce the input variables to feed to FLC, one has to model the plant, i.e., the vehicle. By describing the plant by mathematical expressions, the required inputs can be generated and with the selected gear as the output from FLC input into plant gearbox, new values are calculated. In order that the FLC takes the inputs, analyze them and perform decision-making logic, it will need the fuzzy logic rules developed to cope with all the possible driving scenarios.

By considering above statements, the rules are generated. All rules that are used in FLC are given in Appendix

For the realization of this control system and adaptation to an automatic transmission, sensors are needed to detect reference criteria that are vehicle speed, engine speed, throttle input, and the engine load.

There is no difficulty on detecting the first three criteria with the sensors. [16] The vehicle speed is monitored and measured by vehicle speed sensor. This sensor is located on the transmission, differential, transaxle or speedometer head. The throttle input is measured with throttle position sensor called TPS sensor. It is mounted on the throttle shaft or the carburetor or throttle body, and this sensor changes resistance as the throttle opens and closes. The engine speed values can directly be used from the mechanical sensor of the vehicle. The main problem encountered is the engine load sensing difficulty. Actually, it should examine the contents of the resistances. These cannot be measured directly during the movement of the vehicle. It is known that the inertial resistance does not depend on a variable, so the contribution of it can be found easily. The rolling resistance and air resistance depend on the vehicle speed, so no other sensor is needed for the calculation of them. The last resistance is the gradient resistance and this depends on the weight of the vehicle and the grade of the road. As the weight is a known parameter, the other parameter should be predicted. For this purpose, the grade of the road should be sensed with a level sensor. However, the calculation of all the parameters requires a lot of work and hardware power. Another method of obtaining the engine load is to use Manifold Absolute Pressure (MAP) sensor. This sensor monitors intake vacuum and it is mounted on to the intake manifold. The information obtained from this sensor can be used to measure engine load. When the engine load increases, manifold pressure decreases and the intake vacuum is proportionally increased. So, this value can be used to refer for engine load.

The output of the system is the gear change order as known. The realization of this order is performed by actuators provided. This actuator performs its duty by hydraulic pressures with a control valve design. This is mostly a linear solenoid valve that provides quick response and reduction of hysteresis.

2.2. MODEL OF DRIVETRAIN

To implement the control unit and to realize the simulation, the model of the drivetrain of a vehicle should be examined and each equation that describes the vehicle parameters should be outlined. [17], [18]

For simplicity, the drivetrain of a vehicle will be examined in two parts. The first part deals with the components between the engine and the torque converter (front half of drivetrain) and the second part includes the components connecting the torque converter to the wheels (rear half of drivetrain). This two-piece examination is applied because at each part, the quantities at the endpoints can be related with linear equations. The torque converter has non-linear properties, which will be examined later. In the first part, the equations relate the engine torque is related the torque converter input torque and engine acceleration, and at the second half the equations relate the torque converter output torque, the brake torque and vehicle accelerations.

During these examinations, some assumptions are made. First, the change in weight distribution, both statically and dynamically, between axles and wheels, is assumed to have an insignificant effect in determining the overall rolling resistance. So, this is based on two restrictions. First, all tires must be identical in size and are inflated to their normal pressure, and they have always the same speed. Secondly, the rolling resistance at all wheels is proportional to the load on each wheel, with the proportionality constant that should be the same for all wheels. Vertical forces are assumed constant and they can be ignored except for their effect on rolling resistance.

For the drivetrain model, the resistances that occur during the motion of the vehicle have to be calculated. The free-body diagram of the vehicle body is shown in the below figure.

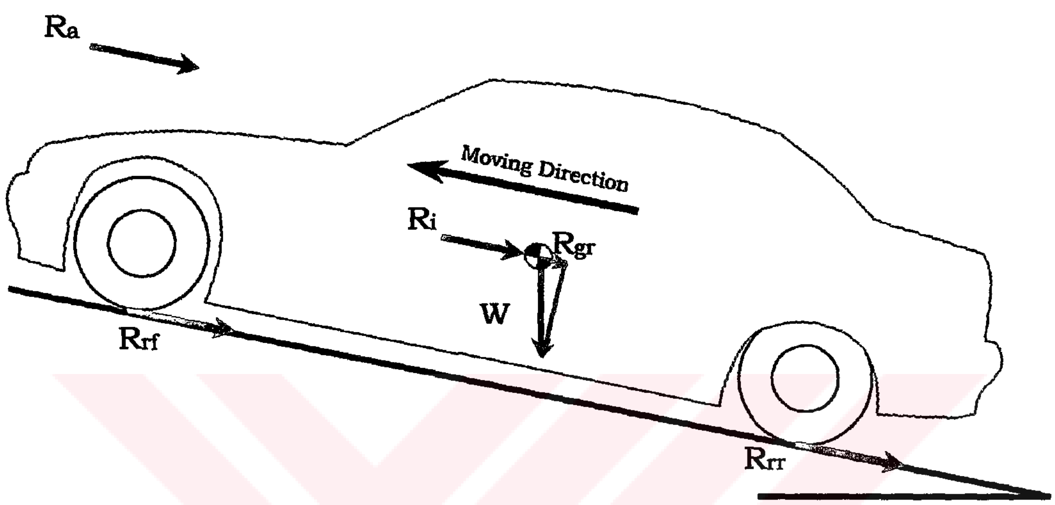

Figure 2.9: Resistances existing on the Vehicle

2.2.1. ROLLING RESISTANCE

Rolling resistance occurs mainly because of the deformation of tires and road surface, and energy dissipation through impact. It depends on vehicle speed, tire inflation pressure, road surface, vertical load on tire, and tire diameter. Rolling resistance increases with the square of vehicle speed, and decreases with increasing tire inflation pressure. This resistance exists in all cases during the movement of vehicle.

The rolling resistance expression is in the form:

$$R_r = f_r \cdot W = \left(a^* + b^* \cdot V\right) \cdot W \tag{2.1}$$

a^* is calculated by including the reference tire inflation pressure and normal tire load. Also a^* is found separately for textile and steel belted radials. The values are calculated for flat and dry concrete or asphalt roads. For other road surfaces, a multiplication factor is applied to coefficient of rolling resistance (f_r). The constant b^* is given for radial tires as $1,5 \times 10^{-5}$. The constant a^* is normally calculated accounting the factors above [17]. However, for simplicity, a^* can be taken as an average value of 0,01.

2.2.2. GRADIENT RESISTANCE

Gradient resistance is applicable for the case of a vehicle climbing a gradient. When the vehicle goes down a gradient then this value is added to the tractive effort. The value of gradient resistance can be formulated as:

$$R_{gr} = W \cdot \sin\theta \tag{2.2}$$

where θ is the angle of the slope [17]. The grade ratio (GR) can be described as the tangential value of the gradient and the gradient in percentage called the grade (G). Then:

$$GR = \tan\theta = \frac{1}{x} = \frac{G}{100} \tag{2.3}$$

2.2.3. AIR RESISTANCE

The force exerted on the vehicle, by the air through which it passes, is called air resistance. This resistance is directly related with the frontal area of the vehicle and the square of the momentary vehicle speed.

$$R_a = q \cdot C_D \cdot A_f \tag{2.4}$$

Dynamic pressure is given by:

$$q = 1/2 \cdot \rho \cdot V^2 \tag{2.5}$$

ρ shows the air density and V is the vehicle speed relative to air. By taking the air density as 1.227 kg/m^3, the two above equations can be given as [17]:

$$R_a = 0.047 \cdot C_D \cdot A_f \cdot V^2 \tag{2.6}$$

Here A_f is in [m^2] and velocity V is taken in [km/h]. In general, when the frontal area is not known, the area is found by the expression:

$$A_f = 0.8 \cdot \textit{(Max height) (Max width)} \tag{2.7}$$

Air resistance depends only on the vehicle speed with square power. So, one can say that the air resistance becomes more important on high-speed cruise of vehicles. During low speed cruises, the air resistance contribution to total resistance stays comparative to others.

2.2.4. INERTIAL RESISTANCES

The inertial resistance exists when the vehicle has linear motion or during acceleration and braking. This resistance occurs due to the rotation parts of vehicle like the engine parts, transmission parts and the wheels. The angular inertia of the rotating parts contributes to the linear inertia of the vehicle and so the effective mass of the vehicle increases respectively. Therefore, knowing that tractive force of a vehicle does not change, the resulting acceleration decreases with increasing effective mass of the vehicle. Firstly, the effective mass is calculated and each inertial force is calculated during kinematic analysis of the vehicle when calculating the inertial resistance. The inertial resistance changes directly with gear ratio so; the inertial resistance will be different for each gear. One can write the equivalent mass as [17]:

$$m_{eq} = m + \frac{J_T}{r_w^{\,2}} \tag{2.8}$$

or an expression by a multiplication factor γ which is called "rotary mass factor" and the product is the "effective mass" can be written.

$$m_{eq} = m \cdot \gamma \tag{2.9}$$

$$\gamma = 1 + \frac{J_T}{m \cdot r_w^{\,2}} \tag{2.10}$$

Total mass moment of inertia can be written as

$$J_T = J_w + J_p \cdot i_d^{\,2} + J_e \cdot i_t^{\,2} \cdot i_d^{\,2} \tag{2.11}$$

An approximate expression of the rotary mass factor can be given as:

$$\gamma = a + (b \cdot i_t \cdot i_d)^2 \tag{2.12}$$

Where i_t is the transmission gear ratio (t=1...5) and i_d is the differential gear ratio respectively. Where a and b are determined by using available data and applying in the above equations. On the other hand, for an approximate calculation, for passenger cars, one can obtain the rotary mass factor [17] with approximate coefficients as,

$$\gamma = 1.03 + 0.0016 \cdot (i_t \cdot i_d)^2 \tag{2.13}$$

In the above formula, the only variable is the gear ratio, counting that the differential gear ratio is constant for every automobile. Therefore, the effective mass is included separately for each gear in vehicle dynamics.

2.3. ANALYSIS OF DRIVETRAIN

In the previous title, the drivetrain was analyzed into two pieces as front half and rear half and it is stated that both parts consists two parts. For convenience, it is better to examine the drivetrain in four parts will be suitable. Each part will be examined with detail in following paragraphs and all variables that are used for the analysis are marked over the figure 2.10 below.

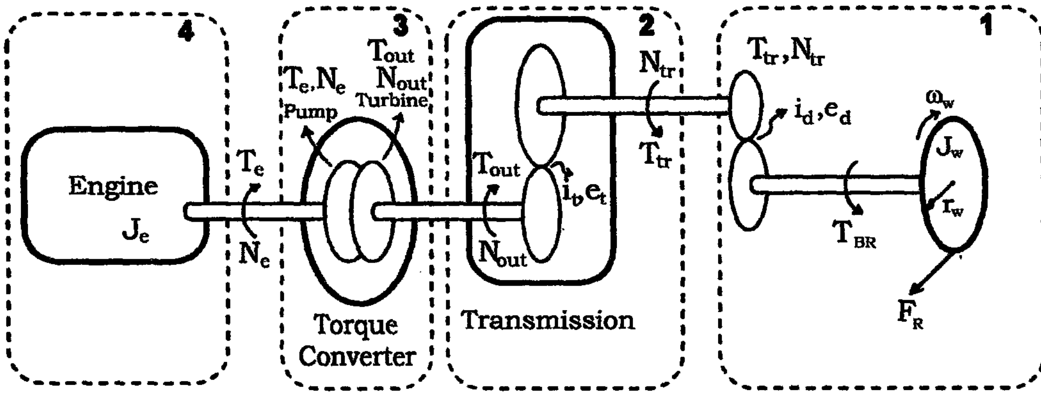

Figure 2.10: Drivetrain Analysis of the vehicle

2.3.1 FIRST PART OF DRIVETRAIN:

First part of the driveline is composed of the driving parts from the wheel of the vehicle to the input of the rear differential including the rear differential and rear axle. Figure 2.10 shows schematic representation of first part of drivetrain. For convenience, it is marked with number '1'.

The resistance effects are counted on the wheel opposing the movement direction. So total resistance force is given

$$F_R = R_a + R_{gr} + R_r \tag{2.14}$$

The road is assumed flat, smooth and dry concrete or asphalt so the multiplication factor for road conditions is taken as unity.

By using the moment equilibrium, the torque required at wheels is found as,

$$T_W = T_{tr} \cdot i_d \cdot e_d - T_{BR} - F_R \cdot r_w \quad (2.15)$$

Where T_{tr} is tractive torque, i_d and e_d are rear differential ratio and its efficiency, T_{BR} is brake torque and r_w is the wheel radius. As known, the resistances and the brake torque are forces opposite to the movement direction, so these are included to the equilibrium with negative sign in front. On the other side, by using Newton's first law applied to rotational masses, the equilibrium is given:

$$T_W = J_T \cdot \alpha \quad (2.16)$$

and $$\alpha = \frac{T_W}{J_T} \quad (2.17)$$

Where J_T is the total mass moment of inertia of rotating masses, which is calculated by equation (2.11), where α is the angular acceleration. From the above equation, the angular velocity is found by integrating the angular acceleration with respect to time. By importing the moment equilibrium, the equation can be written as:

$$\omega_w = \int \frac{T_{tr} \cdot i_d \cdot e_d - T_{BR} - F_R \cdot r_w}{J_T} \cdot dt \quad (2.18)$$

Using the above equation, the linear speed of vehicle is found by:

$$V = 2 \cdot \pi \cdot r_w \cdot \omega_w \quad (2.19)$$

This value of momentary velocity will be used for the calculation of new total resistances and as an input to the FLC.

2.3.2. SECOND PART OF DRIVETRAIN:

The second part of drivetrain is composed of propeller shaft, the automatic transmission assembly and output shaft of torque converter. Figure 2.10 gives the schematic view of these given components where it is marked with number ‘2’.

The torque over propeller shaft T_{tr} is the output torque of the transmission. It found directly by multiplying the input torque with corresponding gear ratio. Therefore, the transmission output speed is

$$T_{tr} = T_{out} \cdot i_t \cdot e_t \tag{2.20}$$

The transmission input speed (N_{out}) is found with the equation:

$$N_{out} = N_{tr} \cdot i_t \tag{2.21}$$

Where N_{tr} is propeller shaft speed and this is found by multiplying the wheel angular speed with the rear differential ratio.

2.3.3. THIRD PART OF DRIVETRAIN:

The third part of drivetrain consists of the torque converter. Detailed explanation of torque converter is given in the next part, but the existing equations used during the calculations can be given below. Mainly three ratios are used to describe the characteristic of the torque converter. These three ratios are speed ratio (*SR*), torque ratio (*TR*) that is a function of speed ratio and finally, the torque coefficient (K_p) that is also a function of speed ratio. The equations are given:

$$SR = \frac{N_{out}}{N_e} \tag{2.22}$$

$$TR\left(N_{out}/N_e\right) = \frac{T_{out}}{T_e} \tag{2.23}$$

$$K_p\left(N_{out}/N_e\right) = \frac{N_e}{\sqrt{T_e}} \tag{2.24}$$

The torque coefficient versus speed ratio and torque ratio versus speed ratio graphs are specifically given for each different torque converter. So, by using these graphs and interpolating the data, the input and output values can be found relatively.

2.3.4. FOURTH PART OF DRIVETRAIN:

The fourth part of drivetrain consists of the engine, the driveshaft, any driveshaft-connected accessories, a bevel gear if used and the torque converter input section. In present case, no accessories and no gear mechanism is used on the driveshaft. Figure 2.10 shows schematic model of these components. This part is marked on the figure with number '4'. Therefore, the driveshaft will not have an extra torque contribution because of lack of accessories and won't have a mechanical torque change due to lack of gear arrangement. Therefore, the engine output torque and the torque converter input torque would be equal in equilibrium condition. However, in the presence of acceleration, this equilibrium won't be true. The engine speed is accelerated by the difference between the torque generated in the engine and the torque absorbed by torque converter.

$$J_e \cdot \dot{N}_e = T_e(N_e, throt) - T_p(N_e, N_{out}) \qquad (2.25)$$

Where N_e shows engine angular velocity, 'N_{out}' is turbine angular velocity, '*throt*' is the throttle intake desired. Engine torque is functions of engine speed and throttle. The graph showing the relation of these three variables defines the engine characteristic of the vehicle. This graph will be specific for a type of engine.

2.4. ENGINE MODEL

The engine model consists of the engine characteristic and the engine speed dynamics that is described above under the title of the fourth part of the driveline in addition to the descriptions stated in the previous paragraph. The static characteristics of the engine is pre-calculated and provided in the model. Engine torque is defined as a non-linear function of throttle and engine speed. The figure 2.11 represents the operating characteristics of the engine used for the simulation [11]. It is a 4-cylinder engine with a maximum torque of 171 N-m at 4000 rpm.

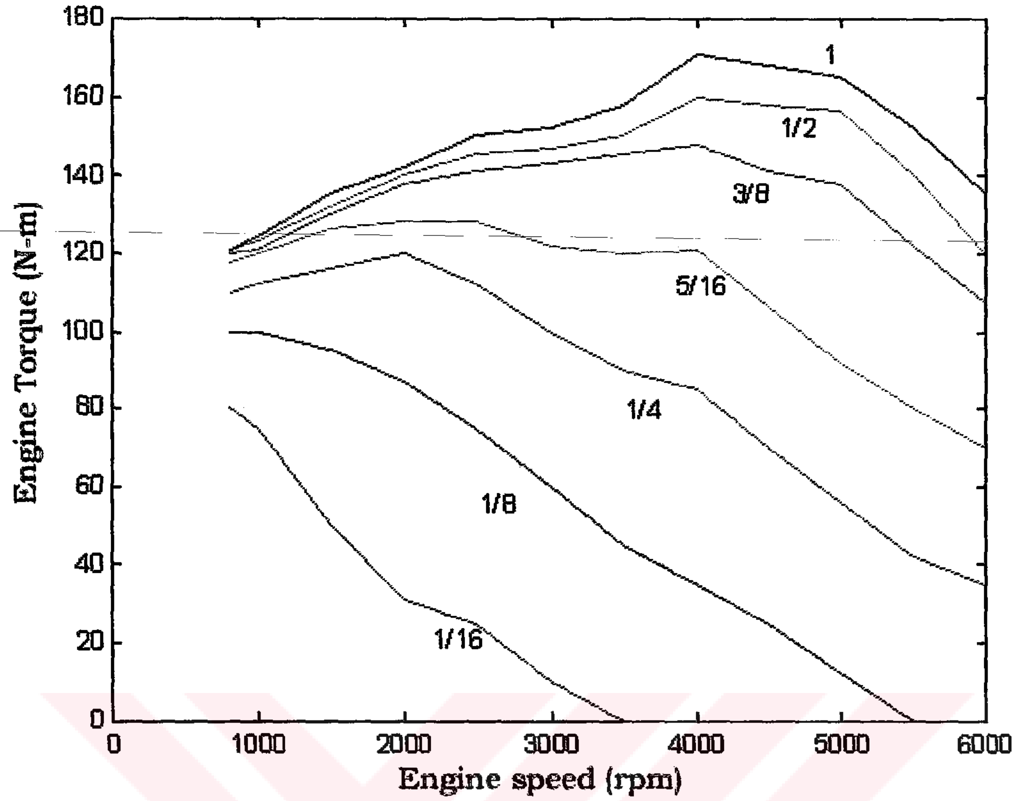

Figure 2.11: Engine Characteristics of the Vehicle [11]

The engine model utilizes multiple torque curves versus engine speed operating points. The engine characteristic is provided into the simulation program as a two-dimensional look-up table. The engine model interpolates between two closest known points in two directions to determine specific torque values both for desired rpm and for desired throttle opening. The program assumes that the engine is electronically revolutionary limited at 6000 rpm, so the torque is temporarily halted if the engine achieves a speed greater than the maximum speed.

T.C. YÜKSEKÖĞRETİM KURULU
DOKÜMANTASYON MERKEZİ

2.5. TORQUE CONVERTER

Torque converter is a device that converts torque by hydrodynamic means, which is the conversion of kinetic energy of hydraulic fluid to rotary motion. Many types of hydrodynamic converters from the basic type of fluid clutches to the sophisticated multi-stage torque converters are used in real applications. However, mostly used one is the single stage three member torque converter that is composed of three vaned wheels; An impeller (pump) connected to the input shaft, a runner (turbine) connected to the output shaft and a reaction member (stator) fixed or rotating according to existing state. All these members are enclosed in a housing filled with full of hydraulic fluid. The figure 2.12 shows one of schematic configuration of torque converter used commercially.

The main advantage of torque converter is that its torque ratio changes automatically like an infinitely varying transmission according to the load conditions. In a vehicle, the torque ratio of the converter is a maximum when the vehicle is to be moved from the rest that the torque requirement for the acceleration is the greatest. While the speed increases and the necessity for further acceleration diminishes, the ratio decreases automatically. Other advantage is the silent and smooth power transmission. In addition, because the moving parts operates in a bath of a lubricating oil without a mechanically connection, the wear and the stall of engine practically does not exist.

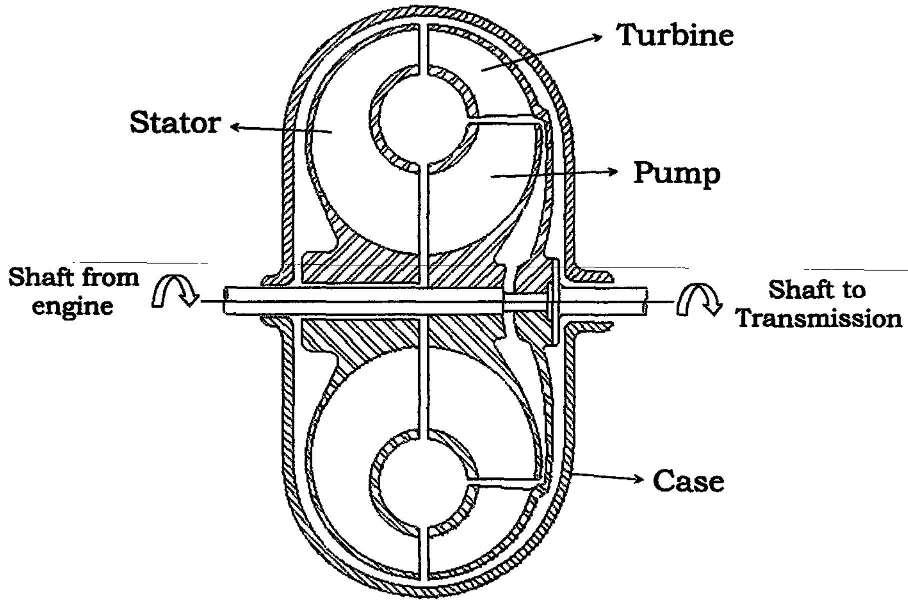

Figure 2.12: Schematic representation of torque converter

The impeller (pump) is directly connected to the engine crankshaft and it operates as a centrifugal pump and while rotating, it pumps the fluid from the point near to axis of rotation to the farther point. The fluid leaves the impeller at this point and starts to pass through the point farther from axis of rotation of runner vanes. When the fluid leaves the impeller, it hits to the blades of the turbine, produces force on these vanes of turbine and because the mechanism is circular, this force creates a torque on the turbine output shaft. Then the fluid comes back to the initial position on the impeller by passing through the vanes of the turbine and the vanes of the stator. [19]

When the turbine (runner) rotates with/without the effect of the fluid, it also creates centrifugal pumping action. However, the result of this state shows different behaviors according to the difference between the speeds of impeller and turbine. When the impeller speed is greater than that of the turbine, the centrifugal pumping action of the impeller overcomes that of the runner that shows an opposite pumping effect. Therefore, the cycle described above is performed repeatedly. When the speeds of both elements are the same, fluid flow does not exist within the converter and they rotate together.

The stator is a member that arranges and directs the flow of fluid between the impeller and the turbine and helps to drive the impeller rather than oppose it.

2.5.1. PROPERTIES OF THE TORQUE CONVERTERS

Some properties of a torque converter are given in figure 2.13. As observed, the output torque reaches to a maximum value when the vehicle is at rest and it is proportional to the square of impeller speed.

The stall torque ratio (the torque ration when the output shaft is not rotating) of torque converters in passenger cars is usually kept in a range of 2.0 to 2.6. Higher ratios are not useful because when the converter operates at high torque ratio range, its efficiency diminishes and the hydraulic fluid is overheated easily, which shorten the life of the fluid. The torque ratio also determines how much the torque multiplication will be.

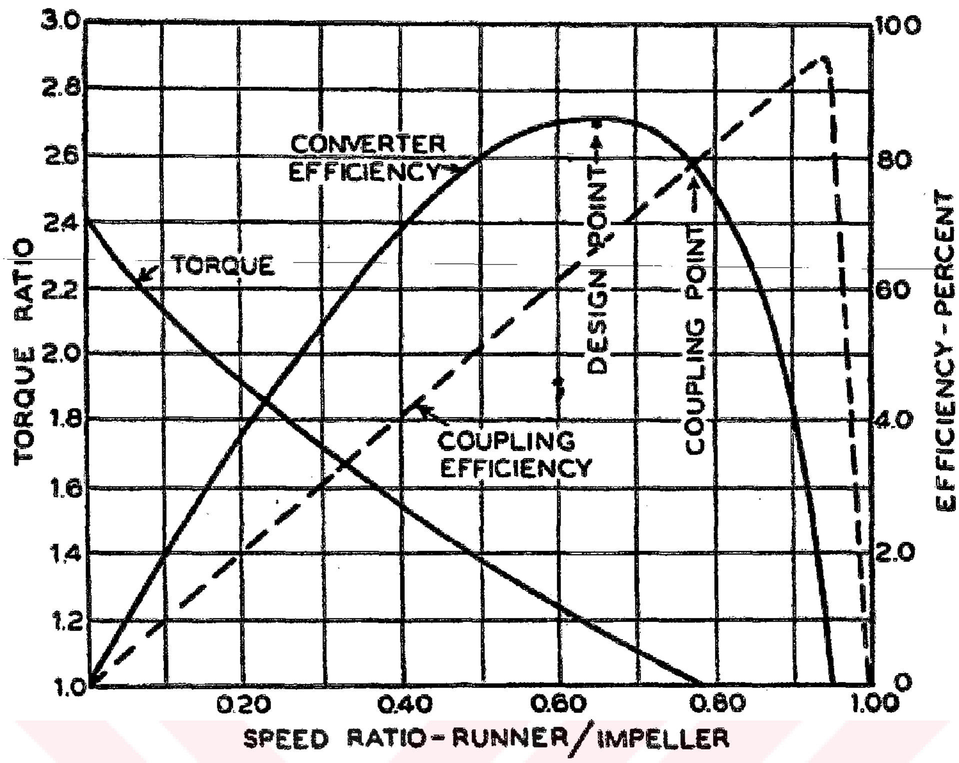

Figure 2.13: Torque Converter characteristics -1[20]

Therefore, the stall torque ratio is the maximum torque multiplication that can be obtained. The torque multiplication is actually a function of a speed relationship between the impeller and the turbine. The heavier the load on the output shaft, the slower will be the turbine speed and the more the input torque will be multiplied. During the operation of the torque converter while the vehicle is moving, the torque multiplication takes different values in a range of 1.0 (the point when the stall speed is reached) and the stall torque ratio. As the difference between the speeds of the turbine and the impeller gets high, the torque is multiplied with greater values and as the difference is lower, the torque multiplication is reduced proportionally.

When starting from the rest, in other words, the output shaft speed is zero; the efficiency of the torque converter is also zero. As the turbine speeds up, the efficiency of the converter increases and reaches to the maximum value at the design point (determined by the designer while designing). The maximum efficiency that can be obtained will not be higher than around 90% depending on the number of stages (number of set of impeller and turbine combination), number of blades, angles of blades, etc. At a certain point, the output torque becomes equal with the input torque value. This point is called "coupling point" when the torque ratio is equal to 1.0. The output speed that the hydraulic lock-up occurs is called the "stall speed" of the converter, which is a specific value for each torque converter. The efficiency at this point is not as high as the obtained efficiency at design point, but it is close to it. However, the efficiency increases linearly towards 100% from this point on when the lock up occurs. The efficiency of a converter can also be defined as the multiplication of torque ratio (TR) with speed ratio (SR). The figure 2.14 [19] shows the efficiency as well as these characteristics of torque ratio and input speed against the output shaft speed. As can be seen, when the coupling occurs, the torque ratio is equal to 1.0 at the stall speed of around 2000 rpm. In addition, the pump coefficient factor (Kp) is also shown. This ratio is another characteristic variable the represent the relationship of torque and speed of a converter and it is defined as the ratio of the impeller speed and the square root of the input torque.

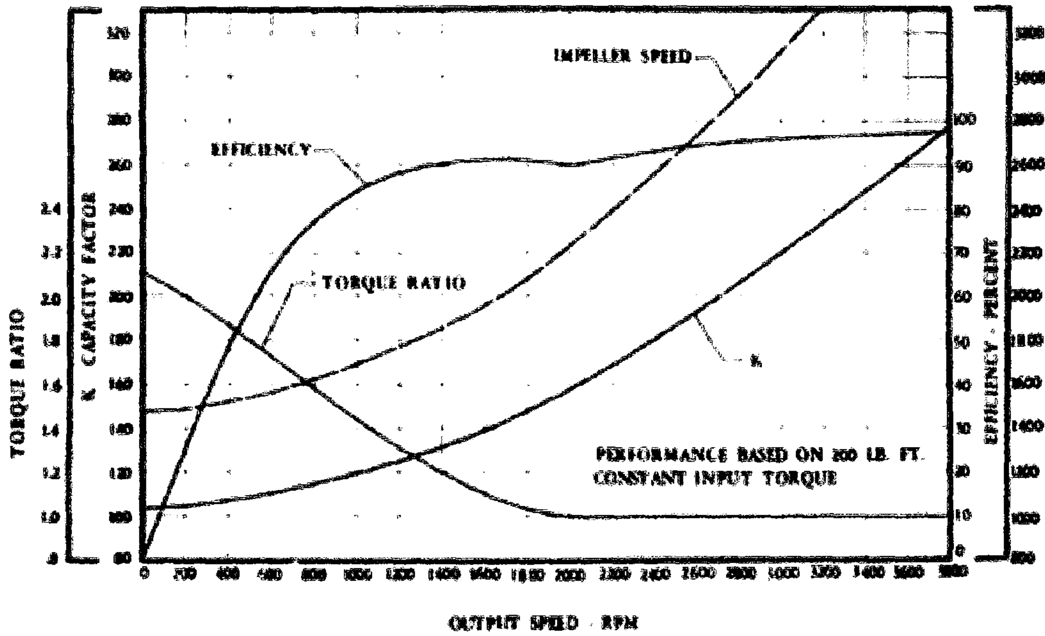

Figure 2.14: Torque Converter characteristics -2[19]

While after hydraulic coupling occurs, it is best to have freely rotating stator around its base shaft with the fluid by taking some amount of energy. However, before reaching this event, the direction of vortex created on fluid at the turbine should be changed and directed to the impeller blades without taking energy from it and so it must remain stationary to fulfill this duty and to provide torque multiplication. Therefore, the stator is mounted stationary but the hub of the stator contains a clutch unit that permits the stator to rotate in the same direction with the input shaft and prevents it from rotating in counter direction. Therefore, until coupling point, the stator remains stationary and after the hydraulic coupling, it rotates freely together with the impeller and the turbine.

When the vehicle is at rest, maximum slip of 100% within the converter is observed because the output speed is zero. As the turbine speed increases, and so the torque ratio decreases, the hydraulic slip between the impeller and the turbine decreases until the coupling is attained. In figure 2.15 [21], the decrease in hydraulic slip is shown as linearly dependent to torque ratio, but for some design features, this relation shows hyperbolic variation [20]. At coupling point, the slip occurs at around 10% and as the speed increases, the slip converges to a value that is different from zero, unless the hydraulic coupling is assisted with mechanical coupling arrangement.

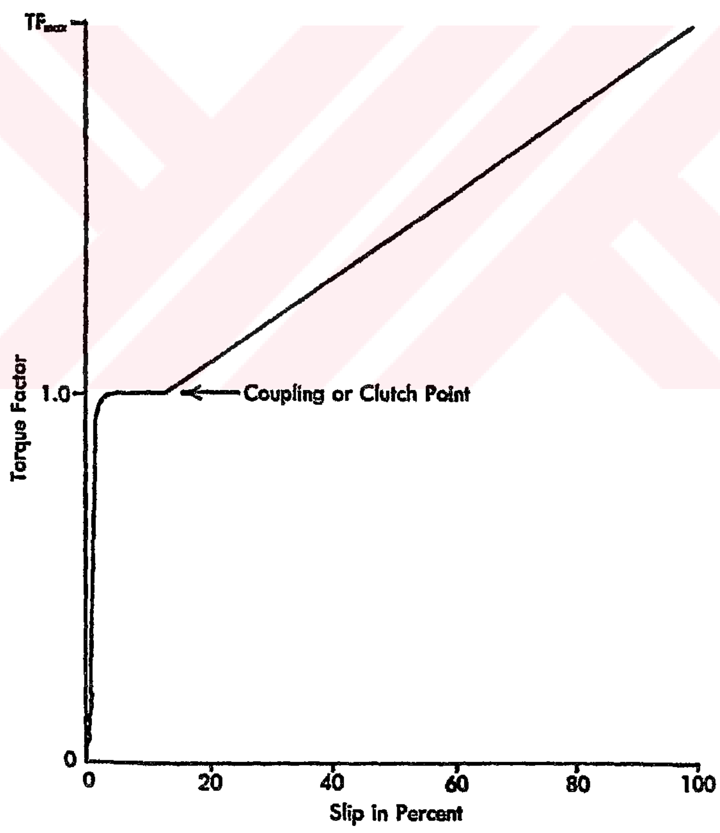

Figure 2.15: Slip in a Torque Converter [21]

Although torque converter provides many advantages, it also has some drawbacks. The main drawback is the loss of power because 100% efficiency cannot be caught even after the coupling occurs. In addition, when the vehicle is at rest, the produced power is absorbed by the hydraulic fluid and transformed to heat energy. Therefore, this leads to a bit higher fuel consumption for the vehicle and a bit lower efficiency compared to mechanical clutch in manual transmissions.

2.5.2. TORQUE CONVERTER APPLIED TO THE VEHICLE MODEL

For the study, the characteristics of torque ratio (TR), speed ratio (SR) that is the ratio of output speed to input speed and Kp will be useful for the implementation of the torque converter to the model. A torque converter that is suitable for the engine will be used and the figures 2.16 and 2.17 show these performance curves. The dot line in figure 2.16 shows 100% efficiency curve for the torque converter according to the definition of efficiency.

By using the data provided in figures, the relation between input torque and output torque and corresponding relation between input speed and output speed of the torque converter.

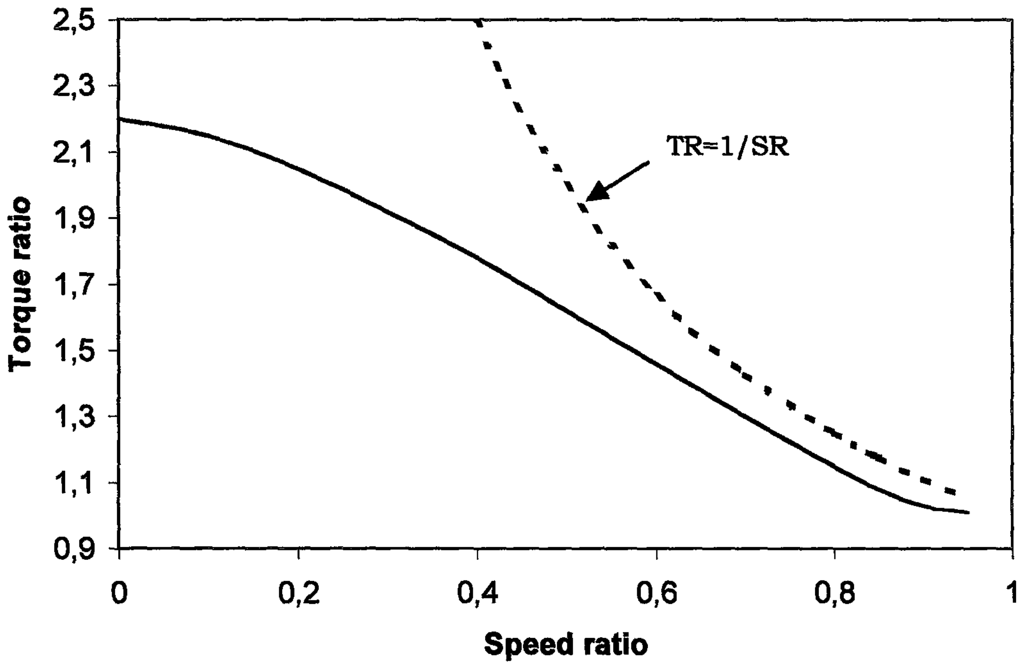

Figure 2.16: Torque ratio versus speed ratio

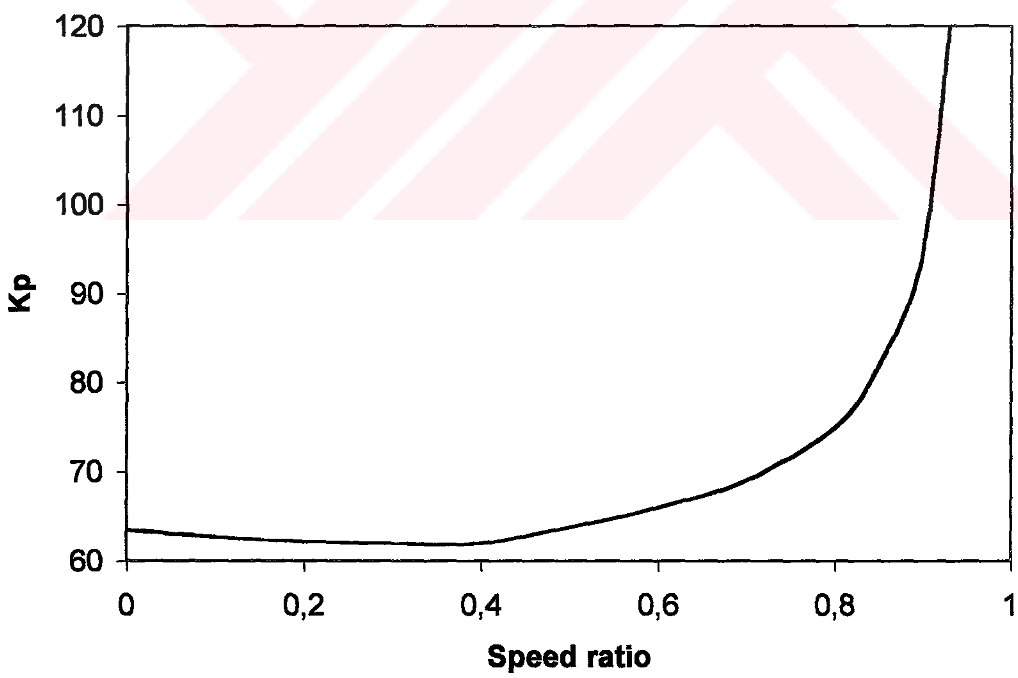

Figure 2.17: Pump capacity factor versus speed ratio [16]

Throughout the study, the fuzzy toolbox of MATLAB© version 5.3 is used. The design and the kinematics of the vehicle are created in SIMULINK a simulation program of the MATLAB©. The Mamdani fuzzy inference method is selected for the present study. For the fuzzy logic operators "and" and "or", which are used during inference mechanism of controller, are performed by the method of *"min"* function, and by *"max"* function respectively. For the implication part, the operation is performed by *"min (minimum)"* function. The last step before the final operation is aggregation and the aggregation method is selected as *"max (maximum)"*. The *"centroid method"* is selected for the final step, defuzzification.

CHAPTER 3

SIMULATION

3.1 INTRODUCTION

Figure 3.1 shows the diagram prepared in SIMULINK. The only input that should be provided outside the simulation is the throttle opening and so, a previously created throttle schedule is entered into the program by the user. Other inputs to the fuzzy controller are found and fed within the simulation. For each small increment of time, the inputs are recalculated and fed again into the controller.

To make the start of the simulation easier, and to produce user-friendly running, the initialization of the variables are performed within the SIMULINK by double clicking the related box. This will introduce the vehicle data and the fuzzy rules and membership functions to the system.

Mainly, the simulation schema is composed of five subsystems. These are the engine, torque converter, transmission, vehicle, and the fuzzy logic controller. Each subsystem is fed with inner inputs and, outputs are obtained after related calculations are performed in them.

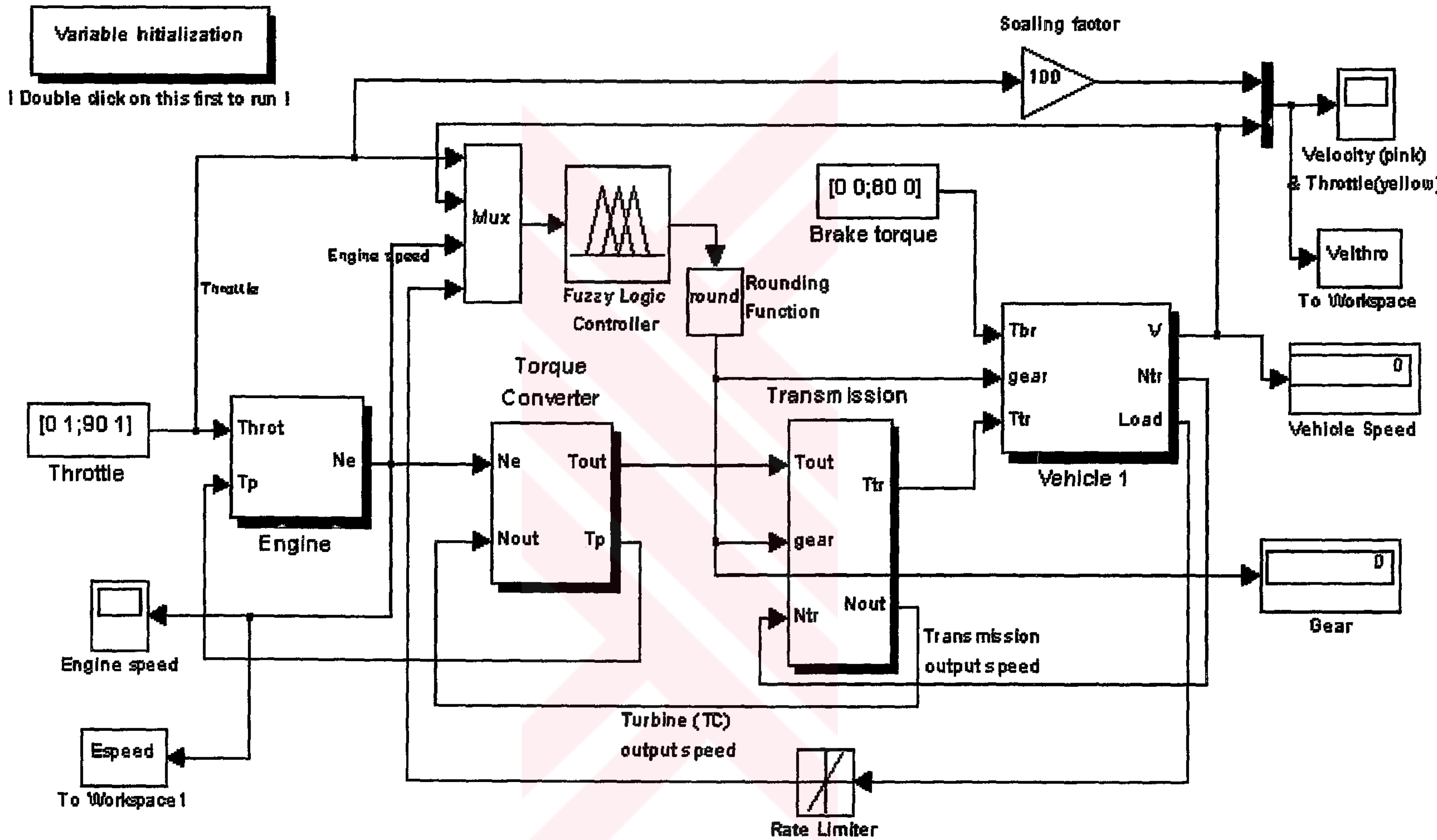

Figure 3.1: Simulation schema of the Vehicle

Notice that excluding FLC, the other four subsystems are linked to each other by feeding back the output of next subsystem to previous one. To understand the working procedure of the simulation, it is important to carry out an examination of each subsystem. Because previous studies about FLC are performed and determined in previous chapters, the examination about this controller system will not be carried out.

3.1.1. ENGINE SUBSYSTEM

As observed in figure 3.1, this subsystem has two inputs and one output. First input is the throttle input that is also input for FLC. Second input is the torque at torque converter's pump side, which is the output of the next subsystem. The output is the engine speed. For a better understanding, it will be useful to refer to schematic representation of this subsystem from figure 3.2.

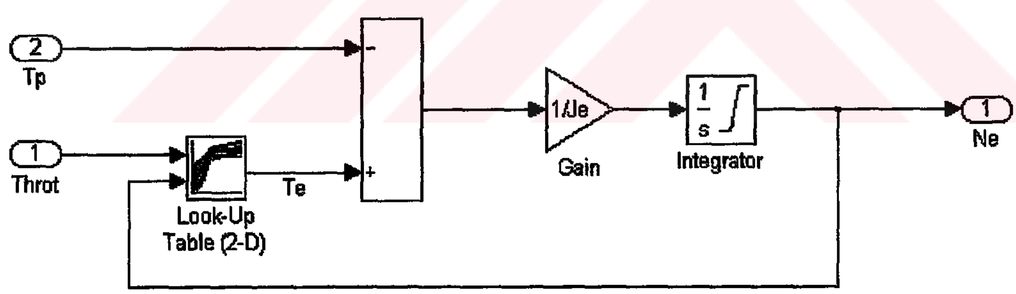

Figure 3.2: Schematic representation of Engine Subsystem

The simulation schema is prepared referring to the mathematical engine model. The 2-D look-up table is used to obtain the engine torque corresponding to the instantaneous throttle and the engine speed values. This look-up table is actually prepared by using the engine characteristics given in figure 2.11. After the subtraction of pump torque from the engine torque, the value is divided by engine parts moment of rotating inertia. This operation gives the angular acceleration of the engine; so to obtain the engine speed, the integration is performed. The desired properties for integration can be arranged by preparing the block parameters. In this integrator property box, the initial condition value is specified as 1000 rpm, which is the roughly the engine speed for most of the vehicles at rest. The engine speed output limitation is enabled to prevent the output from exceeding specifiable levels. The integrator is bounded by the upper and lower saturation limits to prevent out-of-range operations. These boundaries are 800 rpm for lower and 6000 rpm for higher. When the output is outside the boundaries, the integrator is turned off. Therefore, somehow the engine speed passes upper saturation limit, it is not allowed by the integrator and the engine speed is kept at 6000 rpm.

3.1.2. TORQUE CONVERTER SUBSYSTEM

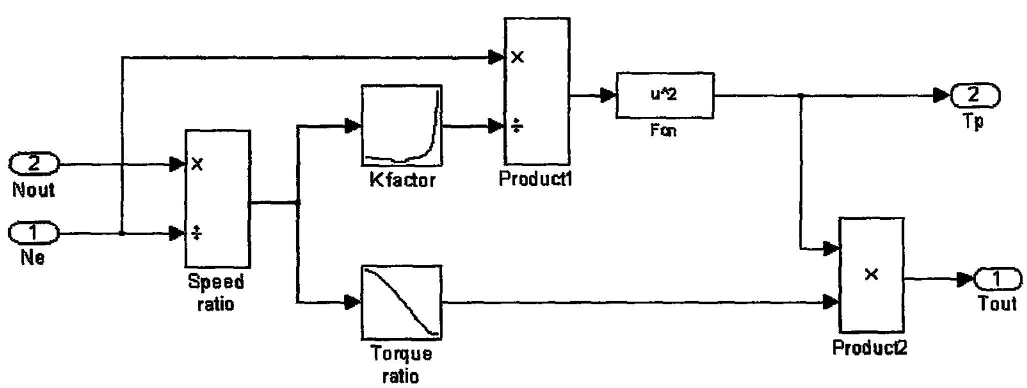

Figure 3.3: Representation of the Torque Converter

Second main block of the model includes the Torque Converter parameters. The content of this block is shown in figure 3.3 as taken from SIMULINK.

This block uses two inputs from internal sources, one is the output from engine block and other is from transmission block. These inputs are engine speed and output angular speed of torque converter. To obtain the outputs, mathematical representations of Torque ratio and K-factor are included in subsystem by using 1-D look-up table. The ratio of two inputs that is the engine speed ratio is obtained and used as a reference value for K-factor and torque ratio blocks. By using this ratio, Kp value and the torque ratio are obtained. The engine torque and the output torque from torque converter are obtained as two outputs by performing defined calculations observed in the schema. So briefly, with this subsystem, using input and output angular speeds of torque converter, the input and output torques are calculated.

As explained, the obtained engine torque is fed back as input to the engine block and the output torque of torque converter is used as an input for the transmission subsystem block.

3.1.3. TRANSMISSION SUBSYSTEM

In this subsystem block, the transmission data is represented to the system. This subsystem uses three inputs from the simulation workspace and they are processed to obtain two outputs. One of the inputs is the gear number that is received from the FLC. The gear number is initially determined as first gear. By referring to look up table seen in figure 3.4, the gear numbers are converted to the mathematically meaningful corresponding gear ratios. The data are provided within the block properties.

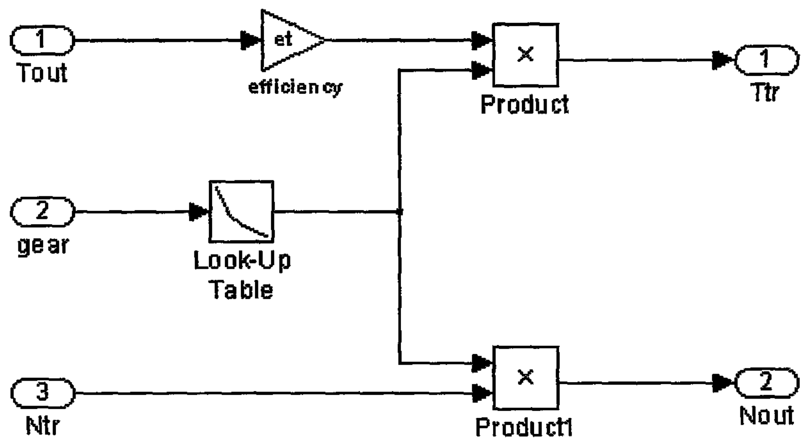

Figure 3.4: Schema of Transmission Subsystem

So other two inputs, the angular speed of the shaft incoming to the transmission and the torque at the shaft outgoing from the transmission are processed by using these ratios by including gear efficiency for both. Efficiency for all gears is considered the same and constant, so the efficiency is included with two gain blocks. After performing production of gear ratio with two inputs, two outputs are obtained. These are the angular speed of the shaft outgoing from transmission and the torque at the shaft coming from torque converter. As the torque output is used as an input for vehicle subsystem, the angular speed output is processed by torque converter subsystem that provides a feedback loop between two blocks.

3.1.4. VEHICLE SUBSYSTEM

This subsystem block represents mathematical expressions of the vehicle parts from transmission to the wheel stated in previous chapter. In addition, the calculations of the resistance forces are performed in this subsystem block. This block needs to receive three inputs to perform provided calculations. These inputs are the gear, the brake torque that is provided if applied by the user and the torque on the propeller shaft. Gear input is used to select the slip factor that depends on gear directly, which reduces the efficiency of transmitting the engine torque to road. Slip factors for each gear is stated in table 3.1. Moreover, gear input is used for the calculation of the inertial resistance. The brake torque input is directly included to the sum of the resistance torque derived from resisting forces. Finally, the torque on propeller shaft, which is the third input, is first carried on the wheel axis by using rear differential ratio. The loss on the differential gear is also counted by including its efficiency.

After transmitting the torque on propeller shaft over the wheel axis, total resistance torque unless inertial resistance torque is subtracted to obtain net tractive torque. Then net tractive force is calculated and the slip factor and inertial resistance are included. Then the instantaneous acceleration of the vehicle is obtained. By integrating this value, the vehicle speed is obtained. Detailed representation can be seen in figure 3.5. This is one of the desired outputs. The found value of vehicle speed is used for the calculation of resistances. The other output that is the angular propeller shaft speed is then obtained from this value by including wheel radius, differential gear and unit conversions.

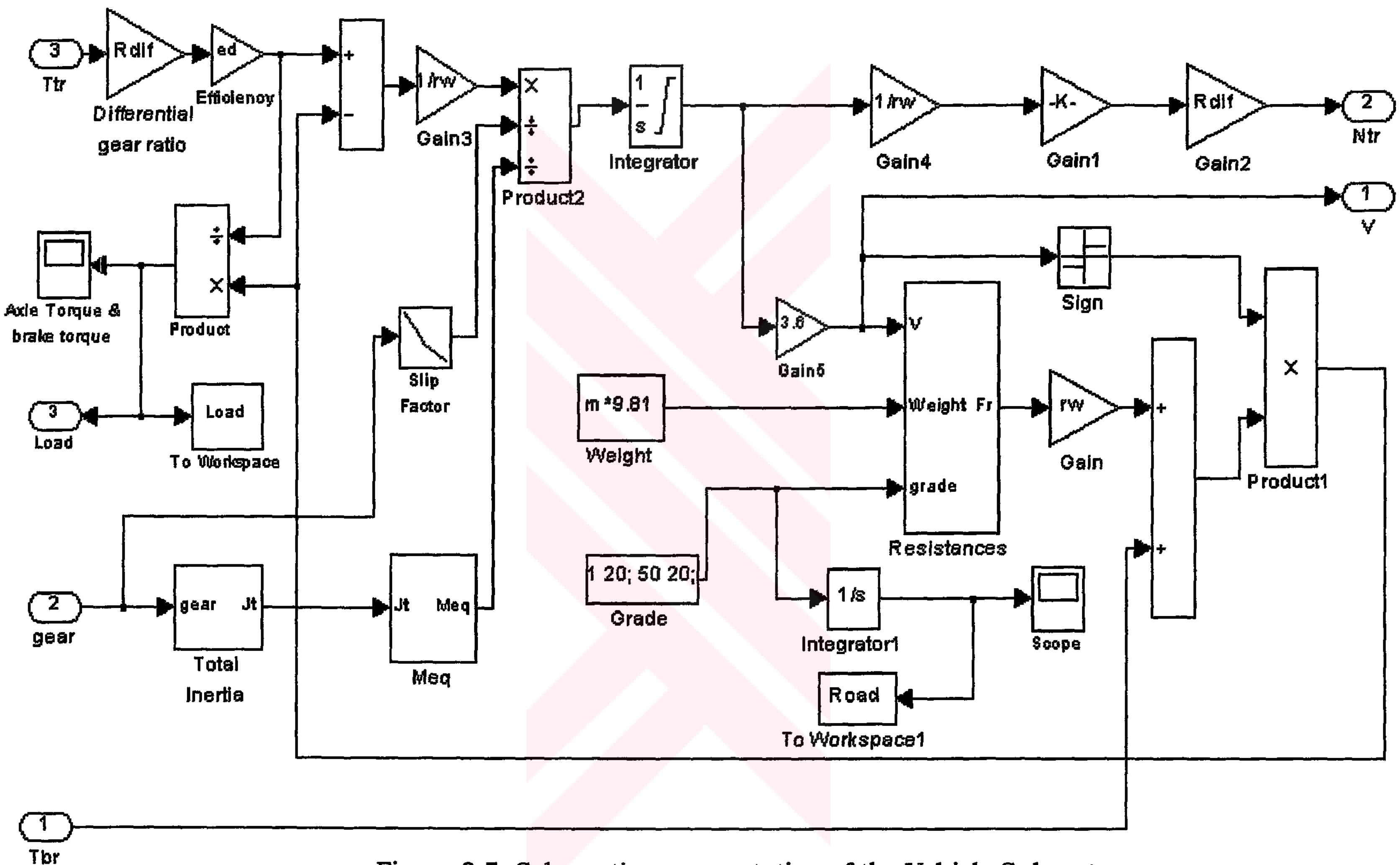

Figure 3.5: Schematic representation of the Vehicle Subsystem

The final output is obtained by comparing the resistance torque and the tractive torque on the wheel axis. The third output is the ratio of these values and called engine load. The inertial resistance of the vehicle is included by calculating effective mass within the subsystem box called 'Meq' and using this value to find the vehicle acceleration.

Different from the three inputs, there are different variables that are stated and entered by the user, which are grade and other vehicle variables like weight of the car. These are entered by input box of constant to the simulation.

While this subsystem is designed, it should notice the calculation of resistances. Because the engine speed is initially set to 1000 rpm, the other inputs and outputs are calculated according to this condition and so the resistances too. However, the vehicle waits at rest and its speed is zero when the resistances calculations are carried. This state does not reflect the point for a real case. As the vehicle does not move, the resistances occurring over the vehicle should be zero except gradient resistance. Therefore, to reflect this point to the simulation, a small controller should be adapted to the simulation. For this purpose, the usage of a signum function will be very suitable. Notice that the definition of signum function provides that for zero input, the output value is zero and for positive input, the output value is one. Therefore, by using the vehicle speed as variable, signum function overcomes this state of maladjustment. While the simulation runs, the calculated value of the total resistances is multiplied with the output of signum function for each time increment. Actually, the only difference on the resistance contribution to system is observed when the vehicle speed is zero, which makes zero contribution of resistance.

3.2. NUMERICAL SOLUTION METHODS

As explained in the previous chapter, there are systems of ordinary differential equations and integrations to be solved. In literature, there are some techniques -all based on Taylor series- used to solve the differential equations, like Euler's, Heun's, Runge-Kutta 3rd, 4th, and high order methods. Euler's method is a straightforward and the easiest method to solve an ordinary differential equation. The techniques stated above are in the order of having less to more calculation steps and have more to less relative error. The figure 3.6 shows the percent relative error obtained from each technique [22].

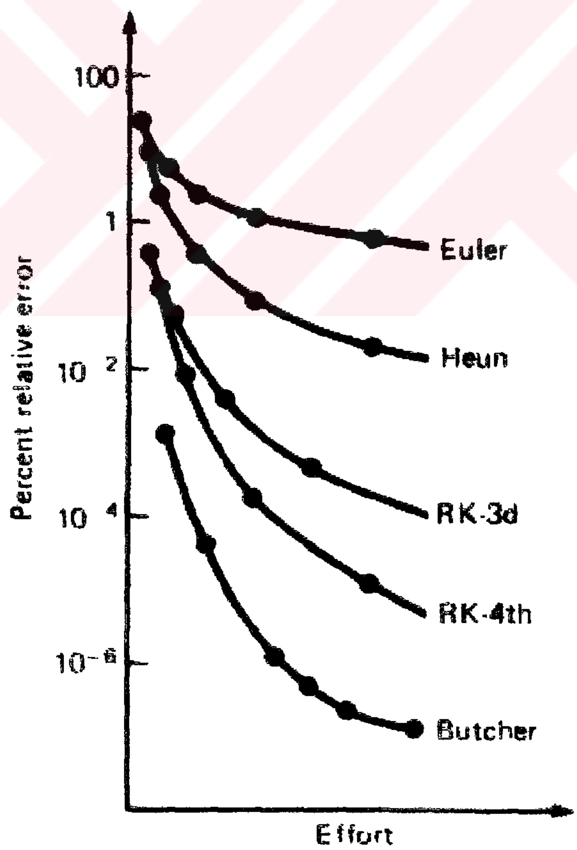

Figure 3.6: Percent relative error graph

In order to solve the differential equations and run the simulation, the Runge-Kutta method order four and five is selected from the simulation parameters menu of the SIMULINK in order to achieve numerical results as a function of time.

The program shows different behavior according to the selected time increment value. With a proper delta time step, the maximum performance of the vehicle can be obtained. This smallest time step selection (dt) is directly related with the method selected. For relatively smaller values in the range of 0.1 seconds to 0.01 seconds, the shifting simulation does not work correctly and the procedure goes to out of range. From the time increment value of 1e-3 up to 1.5e-6, the simulation becomes more stable as getting lower values, although some instabilities occurs during the gear shifting. By a trial and error procedure, it is found that the simulation should be run within 1e-6 ≥ dt ≥ 1e-7. All simulations and analysis run with a time step of 1e-6 seconds to have an optimized simulation. Although this value causes slow run of the simulation, it provides accurate results for the present system. In addition, the step size is kept as automatic selection in order to provide better step size selection that gives lower error values in simulation outputs. That selection turns the Runge-Kutta method to the adaptive step size control version of Runge-Kutta method with the same order.

Another instability creating problem occurs due to the torque converter parameters. Because these parameters are taken from the graph analysis with an approximation, this leads to an error at the output obtained. However, this state does not cause seriously affecting problem.

3.3. VEHICLE VARIABLES

For the simulation, there is a need of having some parameters that describe the vehicle. The list of the specific vehicle variables and values included in the model are given in table 3.1. [11][23]

Table 3.1: Vehicle Variables used in simulation

Vehicle mass (kg)	1080	i_d	3,563
Vehicle width (m)	1,735	e_d	0,96
Vehicle height (m)	1,423	e_t (t = 1..5)	0,98
J_e (kg.m^2)	0,036	C_d	0,31
J_w (kg.m^2)	2,8	r_w (m)	0,3
J_p (kg.m^2)	0,06	a*	0,01
i_1	3,875	b*	$1,5.10^{-5}$
i_2	2,14	S_1	1,06
i_3	1,384	S_2	1,04
i_4	1,00	S_3	1,02
i_5	0,721	S_4	1,01
		S_5	1

In the table, the efficiency of the gears in gearbox is assumed the same for convenience. Moreover, a* is taken 0,01 with an approximation, although it depends on some equations including vehicle parameters. The terms that are stated with the letter 'S', describes the slip factor for a driven tire and it is stated for each gear. Therefore, during the simulation, the tire slips occurring during the movement of the car, was taken into consideration.

When feeding back the engine load from the vehicle, a rate limiter is used in order to decrease the rate of the instantaneous changes occurring during shifting operation. Rate limiter limits high falls in order to feed it back properly to the fuzzy logic controller. It limits the decrement when the change in engine load falls under ten percent change of the time interval.

The engine characteristic data are taken from figure 1.12 and for each throttle opening at engine speeds, the torque values are obtained. This shows an approximation on torque values, but it is tried to minimize. At the throttle opening of 1/16, the engine torque gets negative values when the engine speed is higher than 3500 rpm. However, the values are taken to be zero. Because, at this throttle opening, it is not allowed the engine speed to pass 3000 rpm, this problem due to the lack of data does not have any effect on the simulation run.

The torque converter's characteristic graphs are taken from [16]. Both graphs are examined, divided into small interval and the values for each increment is noted down with a certain error.

3.4. VEHICLE TESTING AND SIMULATION RESULTS

3.4.1. INTRODUCTION

In this part of the chapter, some pre-assumed studies are carried on for the designed vehicle. First, the accuracy of the simulation will be tested. The obtained results will be shown with the help of the graphs and the results will be discussed and compared with an existing car performance. Second, various case studies will be performed in the simulation. The results will be discussed with the assistance of the graphs.

3.4.2. SIMULATION OF THE VEHICLE ACCELERATION ON VARIOUS THROTTLE INPUT

As stated in previous chapter, the simulation is performed in dry, asphalt or concrete road conditions. Initially, the vehicle speed is zero, at rest. The road gradient is accepted as 0% that means that the road is level and no bend or curve exists. It is assumed that the car does not have high slip on the start and the torque from the wheel is transmitted efficiently.

The test will be carried out for different throttle openings like wide-open throttle (100%), half opened (50%) and quarterly (25%) opened throttle valves are tested. For the simulation, the first aim is to inquire the shifting operations whether it works or not. Therefore, the simulation time is taken as 30 seconds, because it is sufficient to view all the shifting operation for all throttle inputs.

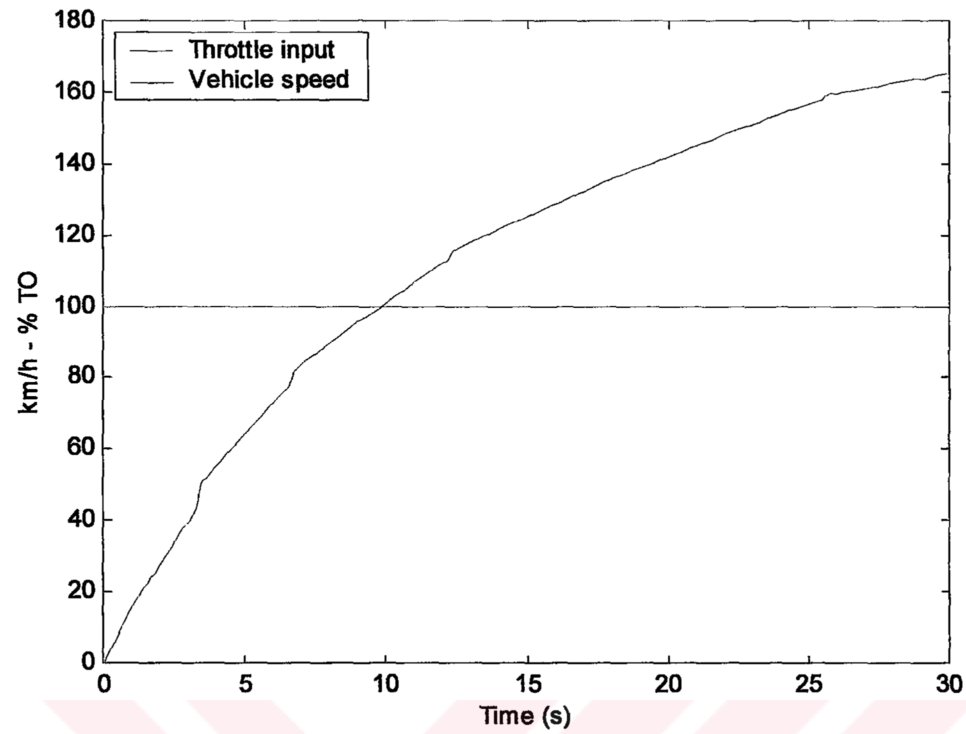

Figure 3.7: Vehicle Speed and throttle versus time

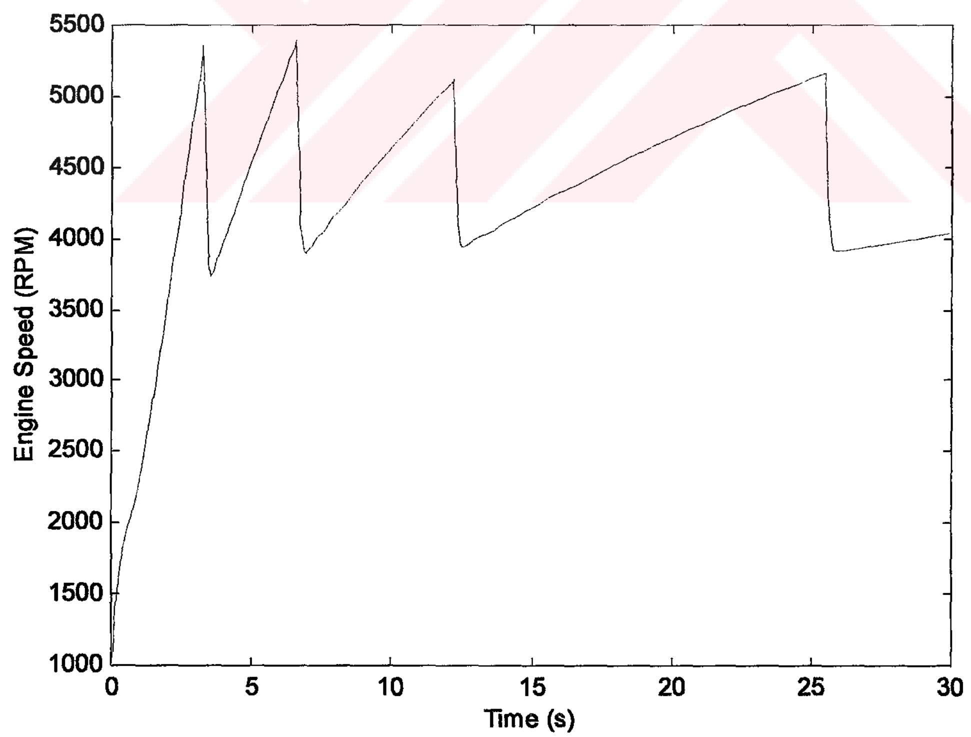

Figure 3.8: Engine speed versus time

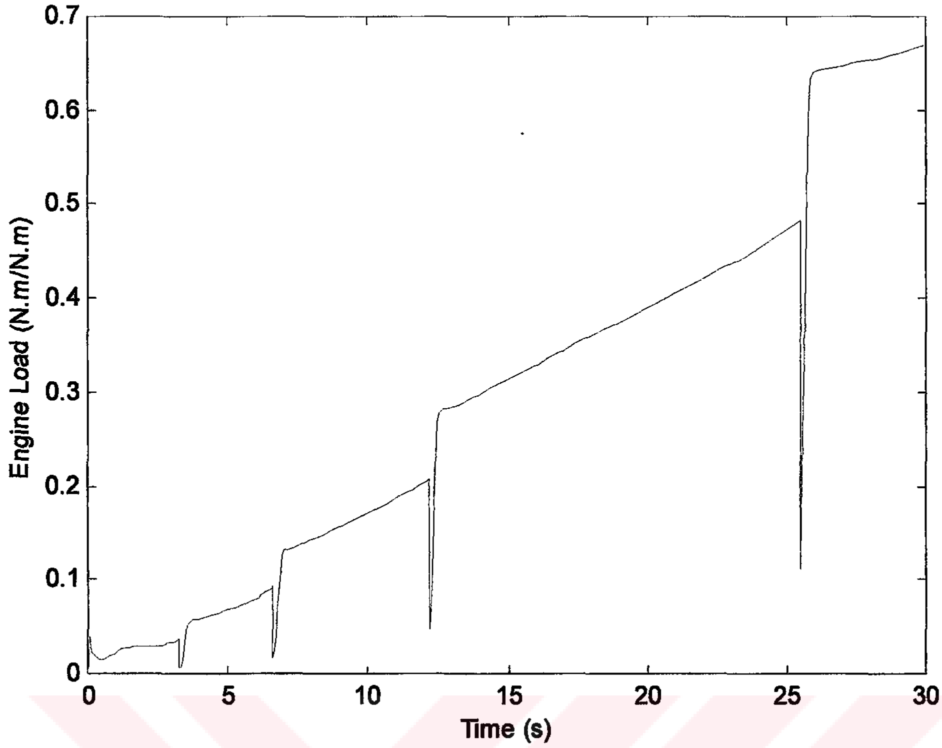

Figure 3.9: Engine Load versus time graph

The vehicle speed during this time interval and wide-open throttle input are shown in figure 3.7. The gear changing times and corresponding engine speeds can be obtained from the figure 3.8. In this figure, engine speed reaches to a maximum speed then faces with a sudden drop and recovers its trend back. This procedure shows gear-shifting operation. Engine load during this simulation can be viewed at the figure 3.9. This figure is based on the proportion of the resistances to the engine torque. That means when the value of 1.0 is attained, the total resistance is equal to the engine torque over the wheel axles. Similarly, this simulation is applied for the other throttle valve opening cases. Related figures are shown respectively.

At full throttle, the gear is shifted up from first gear to second gear at the vehicle speed of 43 km/h and at the engine speed of 5350 rpm. This gear change occurs at time t=3.29 sec and finishes at t=3.54 sec. Similarly, shifting from the second to third gear occurs at 76.5 km/h and 5300 rpm at t= 6.42 sec. The gear shifting finishes at t=6.7 sec. When the vehicle speed reaches to 115.6 km/h at t=12.75 seconds and at the engine speed of 5205 rpm, the gear is shifted from third gear to fourth gear. At time t=13.03 seconds, the shifting is completed. The gear is shifted to fifth gear at the vehicle speed of 157.7 km/h and engine speed of 5150 rpm at 25.38 seconds of the simulation and shifting is completed at time t=25.72 sec.

Therefore, the gear is shifted at high engine speed for full throttle, as desired in FLC. Another comment is the relation between the vehicle acceleration and the engine load. The instantaneous incremental slop of the vehicle speed decreases with time as the engine load increases. At each gear shifting, it is observed that the engine load drops and increases sharply. Actually, this occurs because the torque output of the torque converter increases and decreases sharply during this operation. This is due to the torque characteristic. In addition, it is seen that the vehicle speed increases highly during the shift operation, which also occurs in real systems but with a smaller acceleration than observed in simulation.

The shift time duration is also an important criterion for shifting test. The values given in above paragraph, but for clarity, these will be shown in table 3.2. The duration of each shifting operation is calculated in table. It is known that the shifting duration for a full throttle driving condition should be within the range of 0.2 to

0.3 seconds per shift. So, the shift durations during the simulation are found within the above range stated in [15]. Moreover, It should be mentioned that the shift durations increases as the gear selection increases from the small to high gear. The shift duration with 1-2 shifts should typically be the quickest among the others.

Table 3.2: Shifting start and end time for full throttle

Shifting	Start time (s)	End time (s)	Duration (s)
1 to 2	3.29	3.54	0.25
2 to 3	6.42	6.7	0.28
3 to 4	12.75	13.03	0.28
4 to 5	25.38	25.72	0.34

At half opened throttle, the gear is shifted up at lower engine speed that is described as medium engine speed. After the vehicle starts moving, first to second gear shifting occurs at 32 km/h and at an engine speed of 4100 rpm at time t=2.49 sec. The engine speed falls to 3010 rpm at time t=2.8 sec during shifting. Next shifting operation occurs at the engine speed of 4100 rpm and at the forward speed of 57 km/h at t=5.0 seconds. Then the engine speed becomes to equilibrium at 3040 rpm at time t=5.4 sec for third gear then the vehicle accelerates again up to 3660 rpm that is 78 km/h of vehicle speed. At these values, the fourth gear is selected when time is 8.28 seconds. The engine speed falls to 2900 rpm and when the time index shows t=8.75 sec. At the fourth gear, the vehicle is driven up to 113 km/h and to the corresponding engine speed of 3830 rpm at t=17.87 seconds. During shifting operation, the engine speed falls to 2970 rpm at time t=18.38 sec and the vehicle continues to accelerate at fifth gear. The graphical representation of vehicle speed and throttle, engine speed and engine load are given in figures 3.10, 3.11, and 3.12.

As mentioned, it is preferred that the gear shifting operation is performed in the range of medium engine speed, which is between 3500 rpm and 4500 rpm.

The shift time durations are given in table 3.3. The starting times for each gear shifting are earlier and the durations for each gear shifting are longer than the corresponding gear shifting in full throttle case, as expected. In addition, the difference between each gear shifting duration for half throttle input is longer in increasing order of gear shifting.

Table 3.3:Shifting start and end time for ½ throttle

Shifting	Start time (s)	End time (s)	Duration (s)
1 to 2	2.49	2.80	0.31
2 to 3	5.0	5.4	0.4
3 to 4	8.28	8.75	0.47
4 to 5	17.87	18.38	0.51

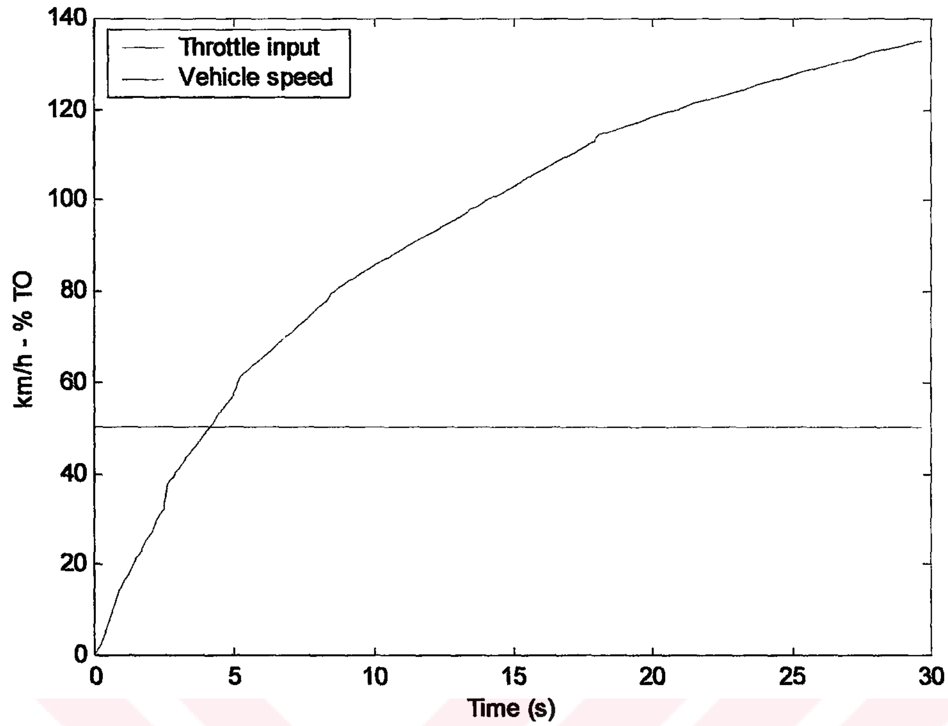

Figure 3.10: Vehicle Speed and throttle versus time

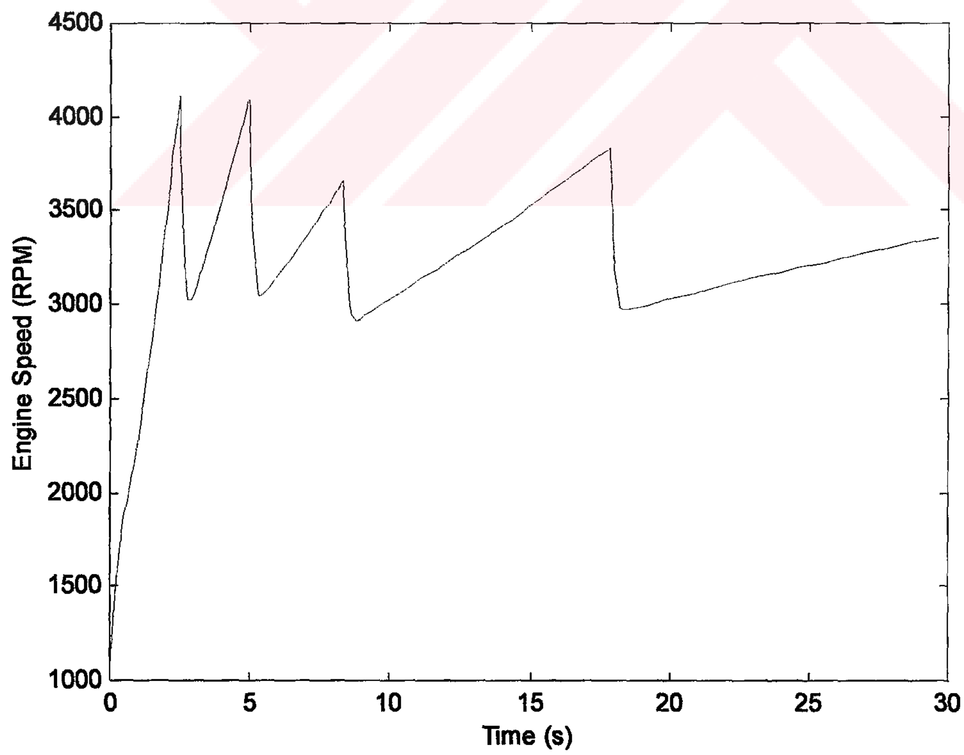

Figure 3.11: Engine speed versus time

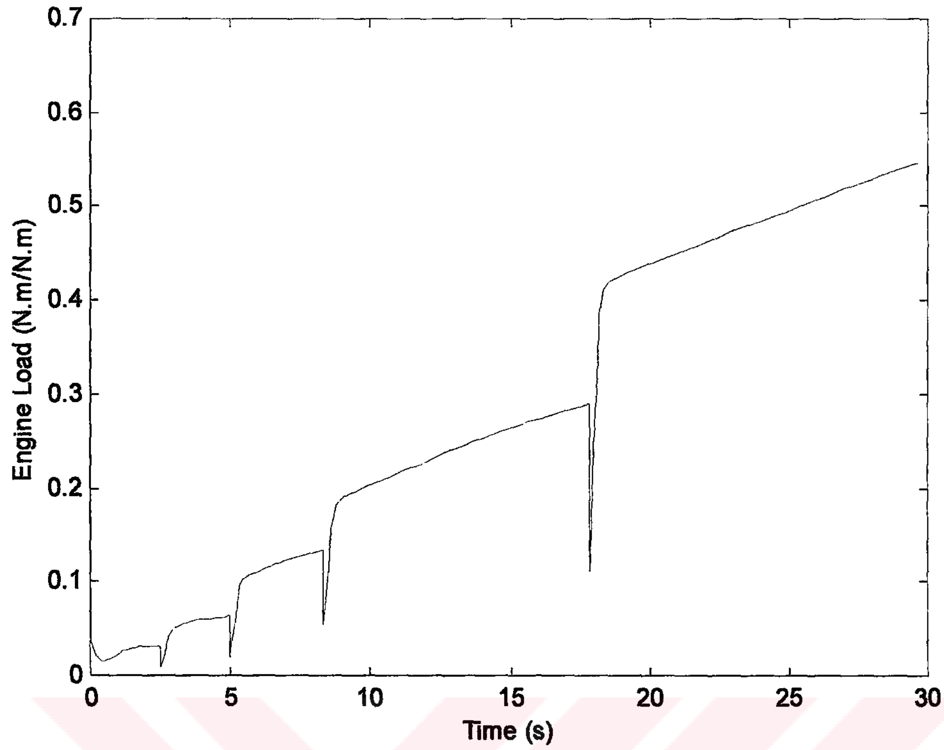

Figure 3.12: Engine Load versus time graph

With quarterly pressed throttle, the test is repeated under the same conditions. The first gear shifting occurs at t=1.1 s. The gear is shifted from one to two at a vehicle speed of 15 km/h and at an engine speed of 2140 rpm. The engine speed falls to engine speed of 2035 rpm and restarts to increase up to next shifting. Then the next gear shifting occurs at the speeds of 33.5 km/h and 2545 rpm at t=3.38 s and finishes at t=3.88 s. The next third to fourth gear shifting occurs at 2840 rpm at time t=8.98 s. The vehicle speed is 60 km/h. The decision for gear changing from fourth to fifth gear is made at 2840 rpm and at time t=17.06 sec. The current vehicle speed is measured as 83 km/h.

For this level of throttle opening, it is decided to change gears at low speeds of engine. This creates smoother and more economical driving of the vehicle. It is known that at first gear, the fuel consumption is higher than all the other gears. Because, the fuel consumption is directly related with the engine torque and the engine torque obtained in first gear is higher than other gears. Therefore, the gear is shifted from first to second gear early. In most cases, the least fuel consumption is obtained in the engine speed range of 2500 to 3500 rpm. As given above, all the gear shifting procedures are performed in this range in order to reduce the fuel consumption with low throttle case. The minimum engine speed during this shifting procedure is kept higher than 2000 rpm, because at lower speeds, the gear shifting can cause unnecessary forcing of engine.

The shift time durations for quarterly opened throttle valve are given in table 3.4. It is expected that compared to previous cases, the gear shifting starts earlier and the durations for each gear shifting are longer than the corresponding gear shifting. Moreover, the time difference between two successive gear-shifting durations are longer than the corresponding ones in the previous cases.

Table 3.4: Shifting start and end time for ¼ throttle

Shifting	Start time (s)	End time (s)	Duration (s)
1 to 2	1.1	1.43	0.34
2 to 3	3.38	3.88	0.5
3 to 4	8.98	9.55	0.57
4 to 5	16.35	17.0	0.65

The vehicle speed and the throttle are viewed in figure 3.13. It should be noted that the vehicle speed variation is smoother than the previous cases. This can also be obtained form the engine load representation in figure 3.14. The variation in engine load shows more linearity according to the previous cases. The engine speed of the process can be obtained from the figure 3.15.

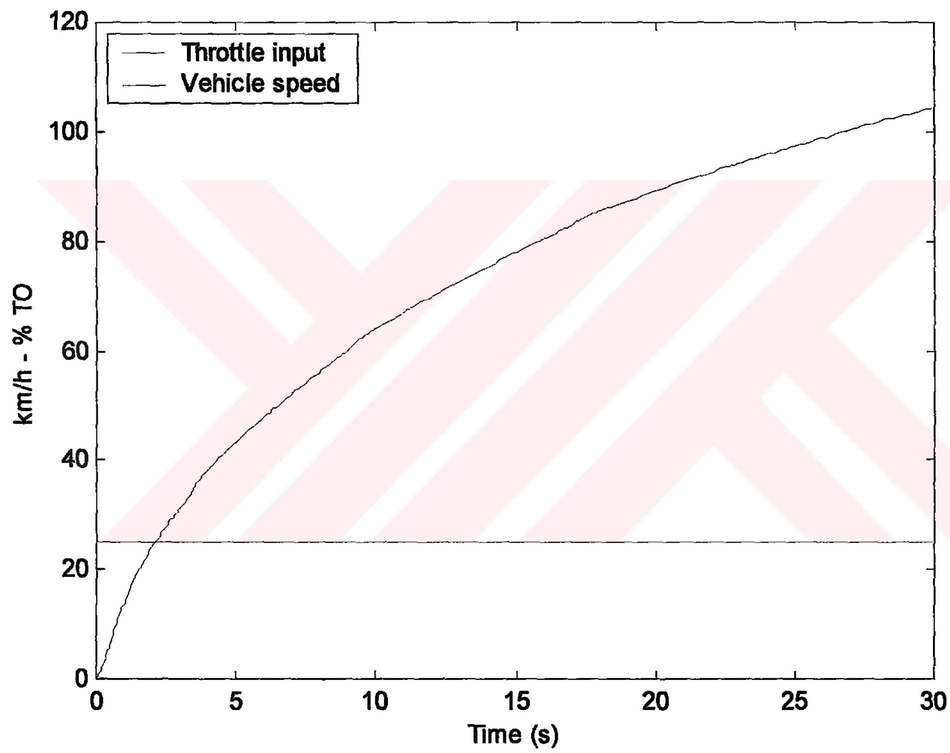

Figure 3.13: Vehicle Speed and throttle versus time

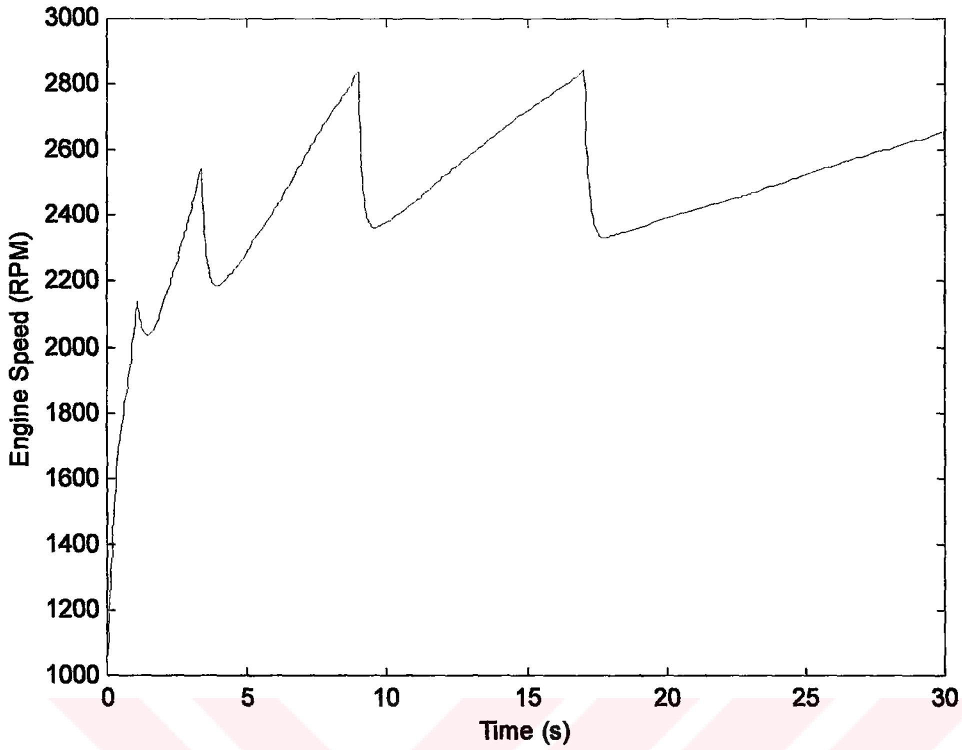

Figure 3.14: Engine speed versus time

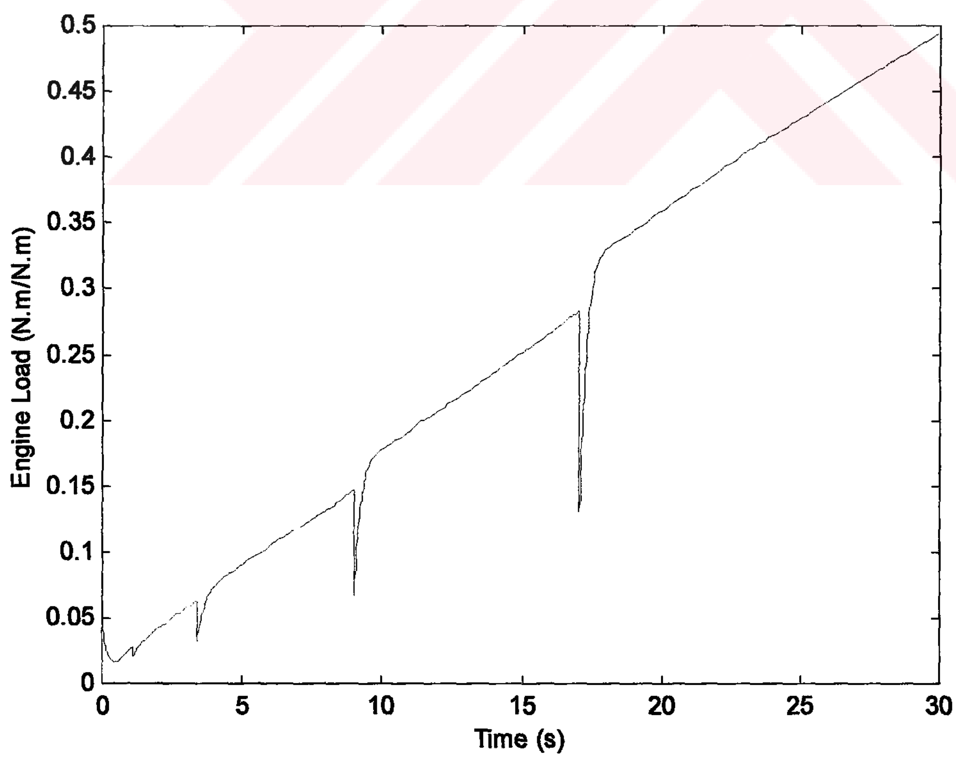

Figure 3.15: Engine Load versus time graph

To compare all, it can be seen that when selecting the desired throttle, the purposes are different in three cases. In full throttle, the top performance is expected from the engine, so the gears are shifted at high engine speeds. This will create harsher driving, by considering the engine load graph and vehicle speed. In quarter throttle, the desired driving style is smother and more economical. Therefore, the engine is not forced and the gears are shifted at low engine speeds.

In addition, the engine load values show very close values at the end of the thirty seconds test. In the order, the values are 0.65, 0.55, and 0.5. This situation occurs because, at the end of test, with high throttle input, the engine torque reaches to high values and so the resistances. At low throttle inputs, the engine torque is comparably low and so the resistances. Then the ratio in all cases becomes closer. However, as expected, the values are obtained in decreasing order.

3.4.3. SIMULATION OF THE VEHICLE PERFORMANCE:

As stated in previous chapter, the simulation is performed in dry, asphalt or concrete road conditions. The vehicle is initially at rest. The road does not have any gradient change and any curvature during the test process. It is assumed that the vehicle does not show a bad traction over the surface for all gears. The weather conditions are counted as standard atmospheric conditions (the temperature is 22^0 C and the humidity is 63%) and the weather is not windy. The throttle of the vehicle will be widely open, which means that the throttle opening is 100%.

In literature, for testing the performance of a vehicle, there are various criteria that are considered and that answer the questions about the performance. Mainly used criterion in metric scale is the time to speed up from rest to 100 km/h. This value is generally used as an index for all test applications.

For the acceleration test, the time to reach to 100 km/h will be kept. For this purpose, the simulation time is taken as 20 seconds, which is a sufficient time for most cars. The throttle input and the vehicle speed are shown in Figure 3.16.

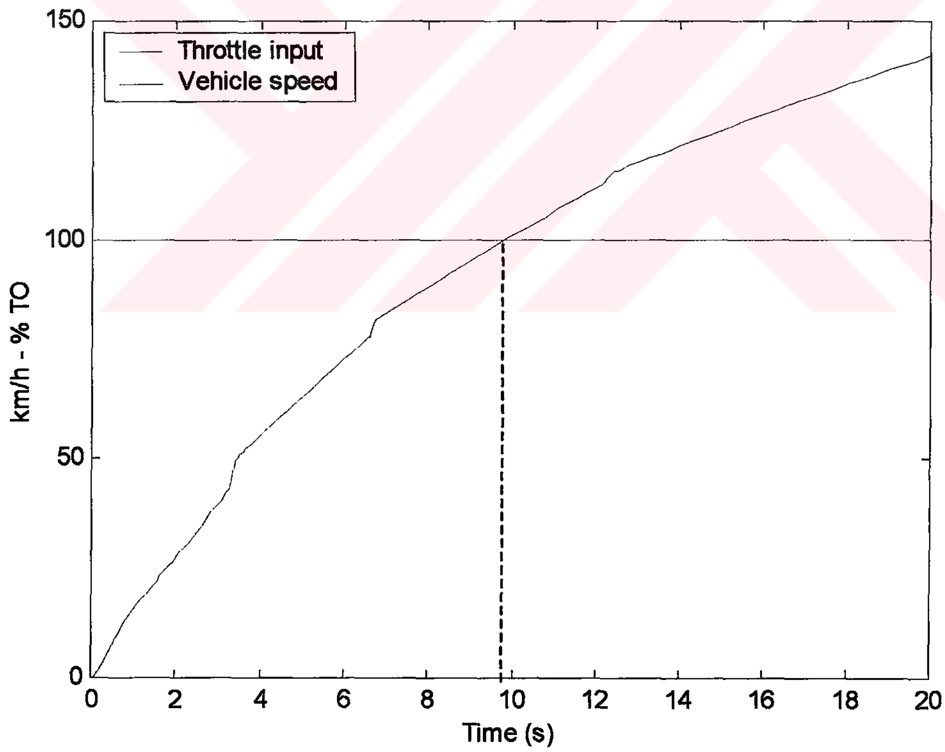

Figure 3.16: Vehicle Speed and throttle versus time

The time value to reach to 100 km/h for existing vehicle is 9,83 s. If the duration is compared with the various vehicles in [17], this value is reasonable for a vehicle with a highest engine torque of 171 N-m and with a mass of 1140 kg. The vehicle reaches to 100 km/h at the third gear.

Another criterion about the vehicle performance is the top speed of the vehicle. To determine the top speed, the throttle valve is fully opened and the above conditions are fully considered. The simulation duration is limitless and the simulation will be stopped when the vehicle speed does not change with time. The test results can be viewed from the following figures. Figure 3.17 shows the vehicle speed variation and the throttle input in percentage during the simulation. Figure 3.18 shows the engine speed variation and the figure 3.19 shows the engine load variation during simulation.

From the figures, it can be observed that the maximum speed for each gear is approximately determined by the maximum allowable speed, that is, it is mechanically limited for each gear except for the fifth gear. Top speed for each gear can be calculated approximately with the following equation [17].

$$V = \frac{r_w . n_{e,\max}}{i_t . i_d . S} \tag{3.1}$$

The results found with above equation will be very close to the simulation results for full throttle input, obtained in previous section. However, the top velocity found according to the equation shows that if the vehicle speed were mechanically limited at fifth gear, the top speed would be 230 km/h approximately. However, the vehicle top speed found from simulation shows that the speed is power limited. This means that the engine power can just balance the resistances acting on the vehicle at this speed. This is easily observed in engine load figure where the ratio of the resistances acting on the vehicle to the engine torque on the wheel axle converges the value of 1.0.

The vehicle's top speed is observed as 203.8 km/h after a long simulation run. This speed is obtained at engine speed of 4852 rpm.

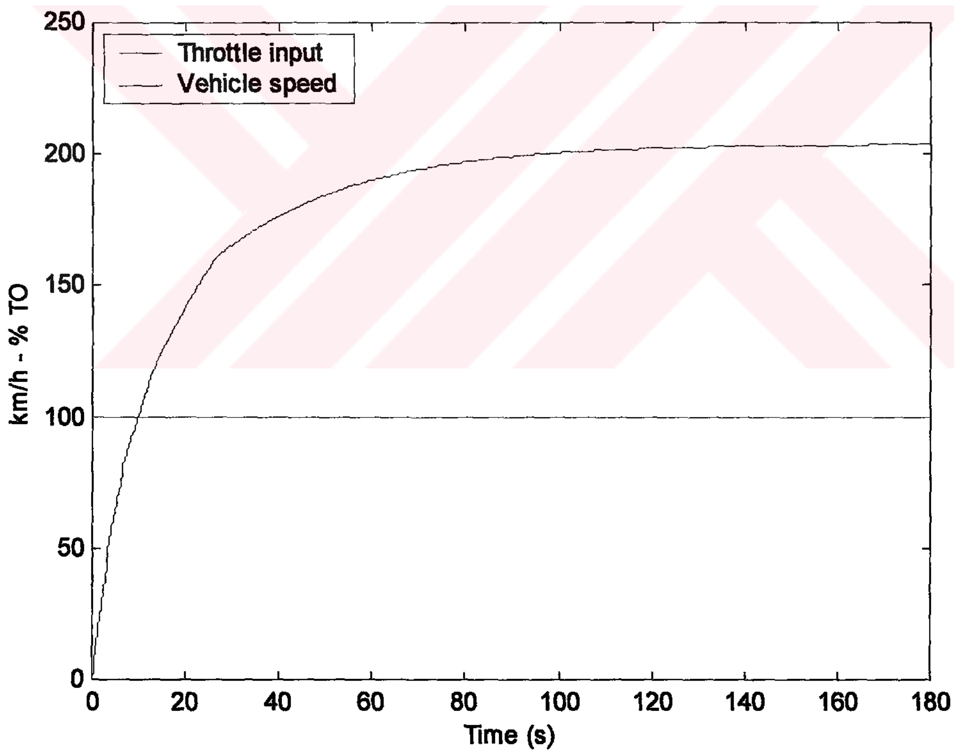

Figure 3.17: Vehicle Speed and throttle versus time

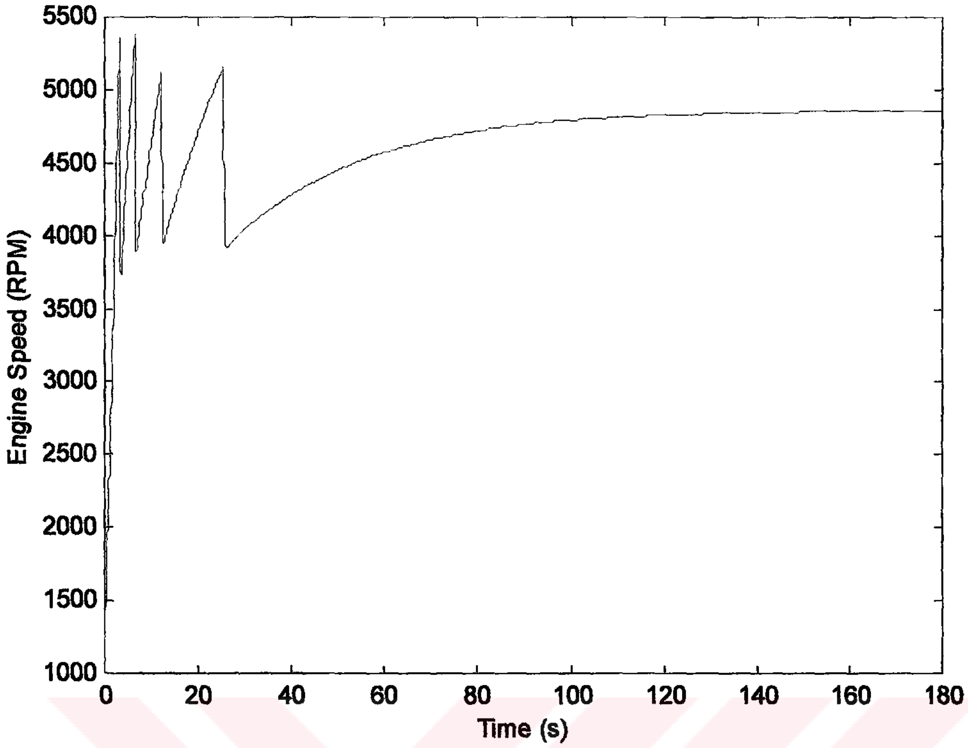

Figure 3.18: Engine speed versus time

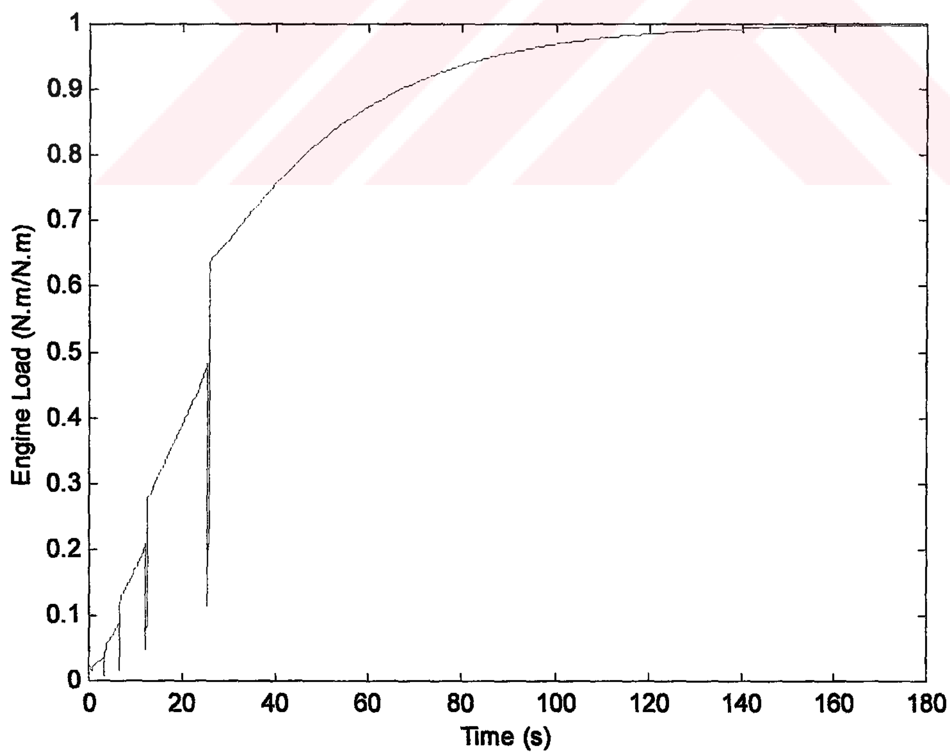

Figure 3.19: Engine Load versus time graph

3.4.4. SIMULATION OF OVERTAKING SLOW VEHICLE ON A FLAT ROAD

On a flat asphalt or concrete road with dry road conditions, the vehicle starts the simulation from rest. The atmospheric conditions are the same as in previous cases. It is assumed that the vehicle does not slip on the road when it starts to overtake.

First, the vehicle starts from rest by applying a throttle of 60% at time equal to zero and when the target speed is reached, the 10% throttle is sufficient to keep the vehicle at this speed. The overtaking maneuver starts at t=50 s that is characterized in simulation by fast sudden opening of the throttle. Throttle valve is highly opened in half of a second and the vehicle speed is increased. The vehicle speed reaches to 130 km/h at sixtieth second. The complete maneuver is completed in ten seconds (between 50th and 60th seconds). Then, the driver starts to decrease its speed by decreasing throttle input within seven seconds and it holds the throttle pedal pressed at 10% of full throttle.

The vehicles speed variation and applied throttle profile during simulation can be observed from figure 3.20. The engine speed variation, so the gear shifting operations, and the engine load variation can be observed from figure 3.21 and from figure 3.22 respectively.

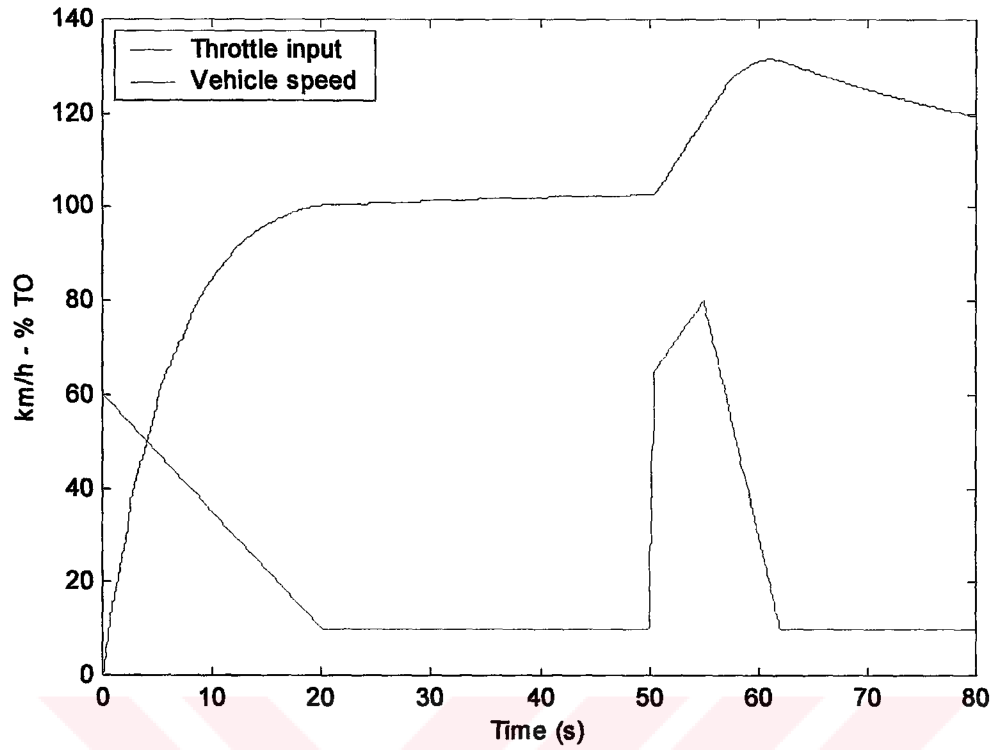

Figure 3.20: Vehicle Speed and throttle versus time

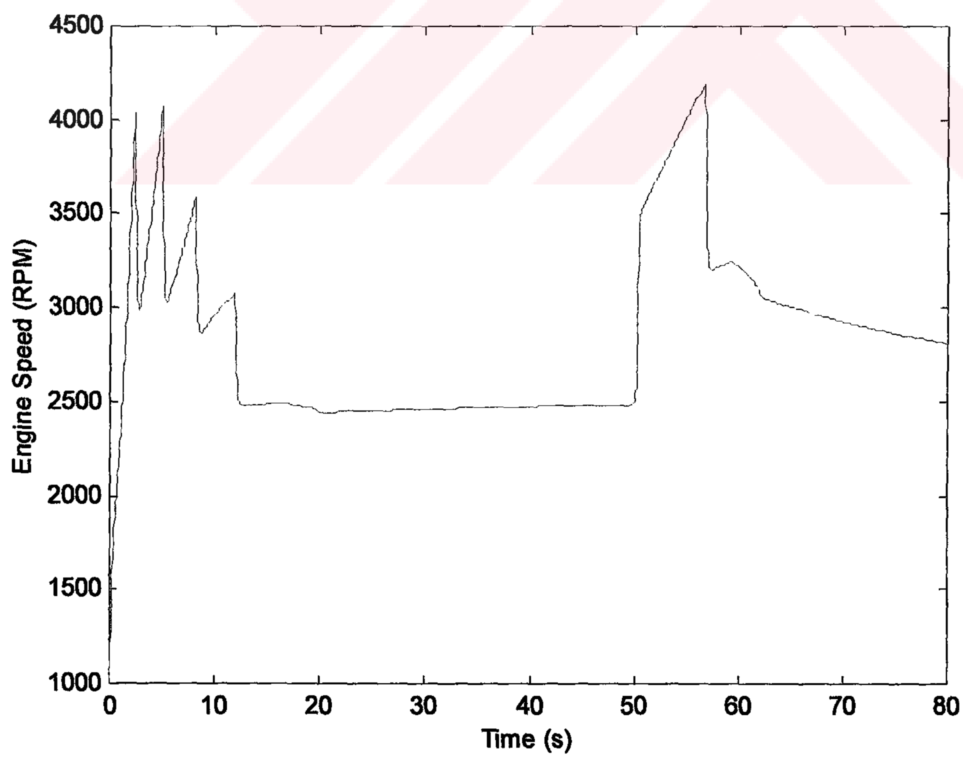

Figure 3.21: Engine speed versus time

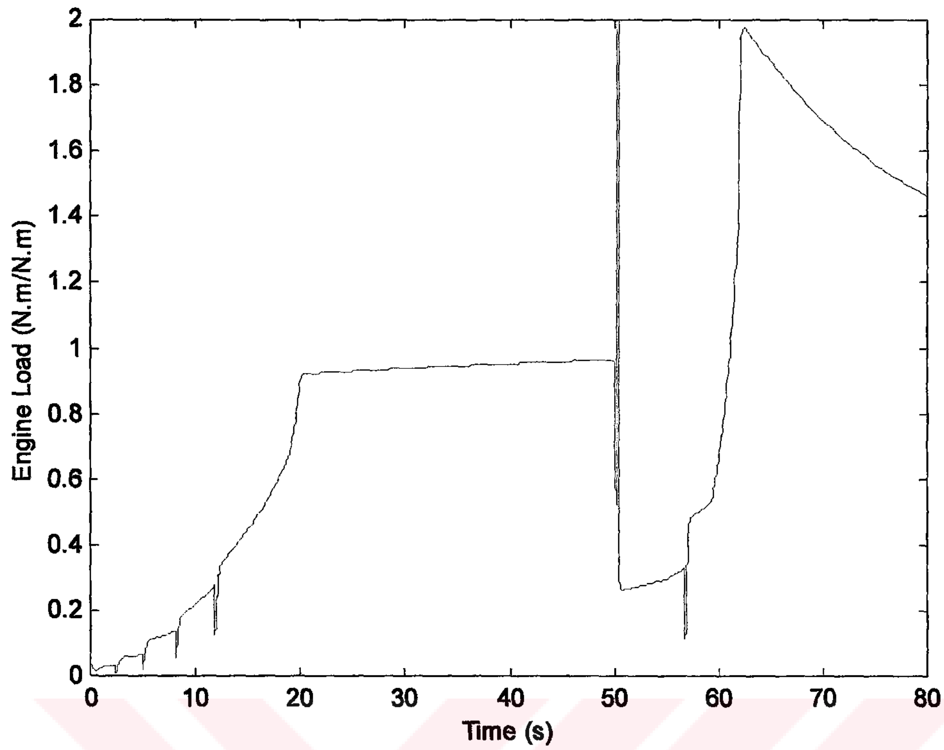

Figure 3.22: Engine Load versus time graph

As can be observed from figure 3.21, when the simulation time shows 15 seconds, the fifth gear is selected and at this gear, the vehicle catches the speed of 100 km/h. When the driver decides to pass slow vehicle in front, it presses down the throttle pedal to increase the vehicle speed. The gear is shifted down to fourth gear to fulfill the driver's intention of acceleration. This shifting operation occurs because as the gear gets smaller, obtained torque so the acceleration is improved. Therefore, the vehicle gains acceleration required to pass slow vehicle within a smaller time length by minimizing the possibility of having danger. During passing, after using the acceleration superiority of the fourth gear, at high engine speed the gear is shifted up to fifth gear again.

In figure 3.22, at fiftieth second, there is a huge engine load increment when the gear is shifted down. This high deflection shows that this high variation on engine load occurs in automatic transmission, and engine torque instantaneously approaches to zero, and present resistances exists during this operation. Moreover, it can be observed that at around sixty-fifth second, the engine load ratio shows the value of 2.0. This shows that the resistances are greater than the provided engine torque, which leads to a deceleration of test vehicle.

This simulation shows the ability that the FLC automatic transmission responds finely for different and immediate inputs by the driver. Moreover, as the driver initially accelerates to desired speed by decreasing throttle input, the gear is shifted at lower engine speed to provide economical driving.

3.4.5. SIMULATION OF DRIVING ON A FLAT ROAD WITH VARIATION OF THROTTLE

This simulation is also performed in similar conditions like road conditions, atmospheric conditions stated previously.

To create a simulation that reflects the actual driving conditions to the simulation, it is better to give a varying throttle input to the simulation. As can be estimated, in real case a driver cannot drive a vehicle with a constantly pressed throttle pedal. Therefore, in order to diminish this unreal case, varying throttle input pattern is prepared and applied, which is stated below in detail.

The vehicle is initially at rest. During simulation, the wheels do not show high slip over the road, and the front wheels are not steered at all. In addition, the braking torque is not applied during any time interval of simulation. At the beginning of the simulation, a sudden throttle input of 10% is applied at time equal to zero. The throttle input is increased to 40% at time equal to 10 seconds. The throttle variation can be observed from the figure 3.19. After the tenth seconds of the simulation, the vehicle starts to decelerate by decreasing the throttle input gradually from 40% to 5% until time equal to 25 seconds. Until 25th second, the vehicle reaches to 95 km/h and at this time, the gear is shifted to fifth gear. When simulation time shows 7.35 seconds, because the driver's intention of acceleration is not fulfilled, the gear is shifted down from third to second gear. At 8.1 seconds, after the desired acceleration is obtained, the gear is shifted up again to third gear. From 25th seconds, during 5 seconds of time interval, the vehicle throttle is kept constant at 5%. For this constant throttle input, the vehicle speed starts to decrease, because obtained engine torque for this amount of throttle cannot overcome existing resistances as can be seen from figure 3.23. From thirtieth seconds, the throttle applied by driver is increased gradually to 20% in ten seconds. The vehicle speed also shows a gradual increase at this interval, further the gear is not changed, and it is kept at fifth gear. From fortieth second, the throttle input provided diminishes gradually to 5% at 55th second. However, the vehicle speed increases from the beginning because provided throttle first overcome all resistances. However, at some throttle input, which is around 10% at time equal to 50 seconds, acceleration of the vehicle stops, vehicle speed reaches to a maxima of 104 km/h and starts to decelerate. So, for this vehicle speed and for this gear, with the throttle input of 10%, the engine torque and opposing resistances becomes equal. This can also be viewed from figure 3.25 showing the ratio of axle load. The throttle pedal is pressed gradually beginning from 55 seconds and the throttle is increased to 50% at time shows 65 seconds. In addition, the vehicle speed starts to increases with a high acceleration value at fifth gear. However, because the acceleration desired by the driver is not enough, the gear is shifted down to fourth gear at time equal to 60.5 seconds. The vehicle is driven in fourth gear during 4 seconds and by attaining an engine speed of 3910 rpm, the gear is shifted up to fifth gear again.

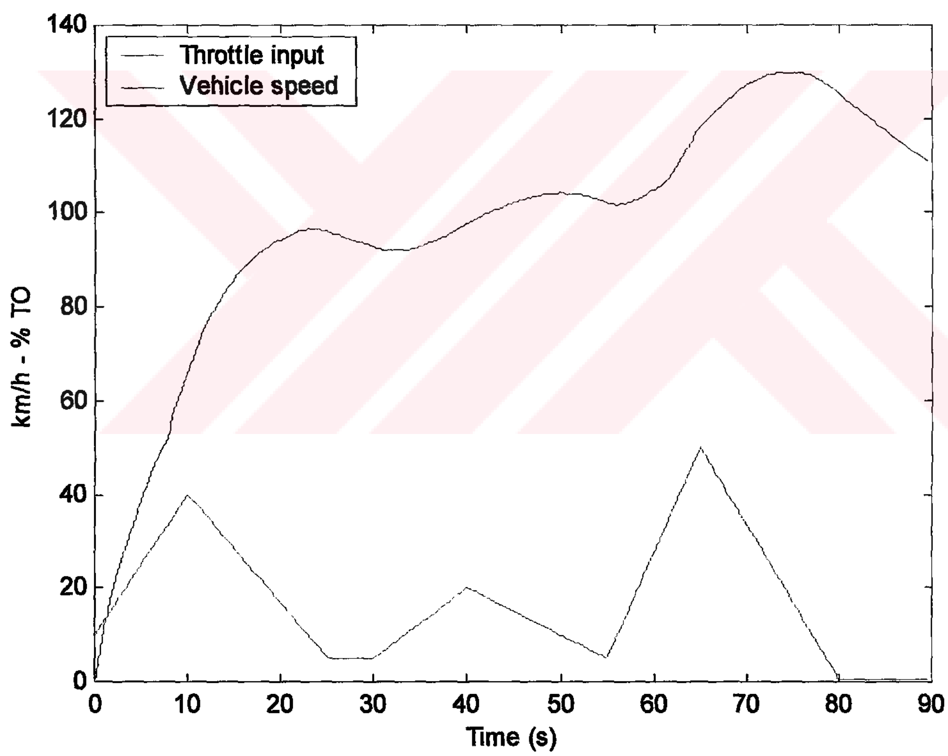

Figure 3.23: Vehicle Speed and throttle versus time

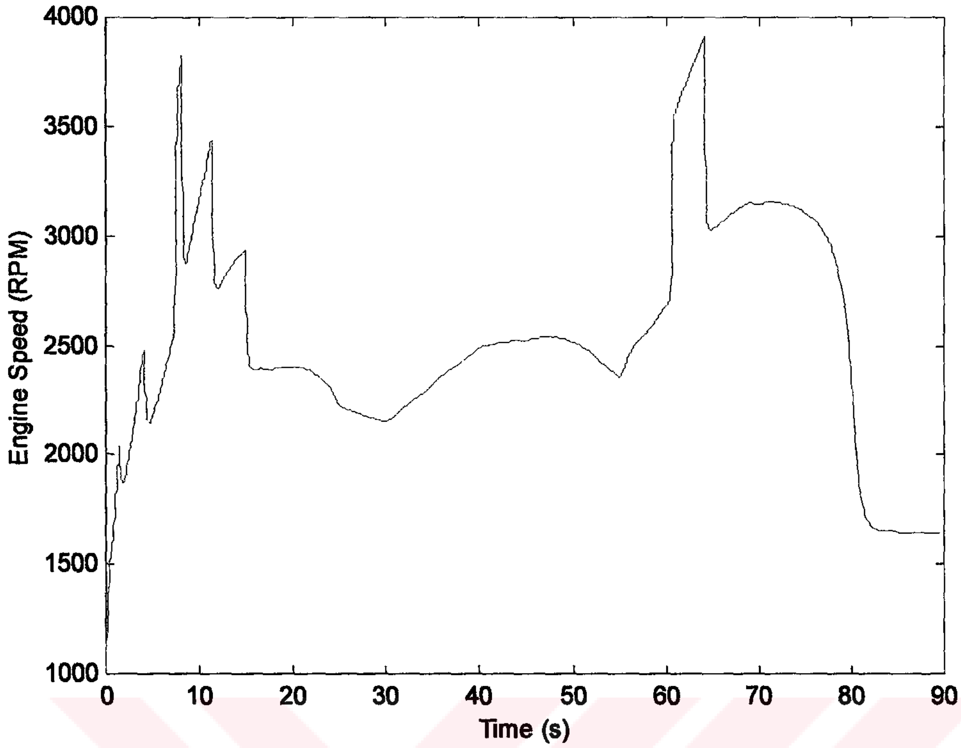

Figure 3.24: Engine speed versus time

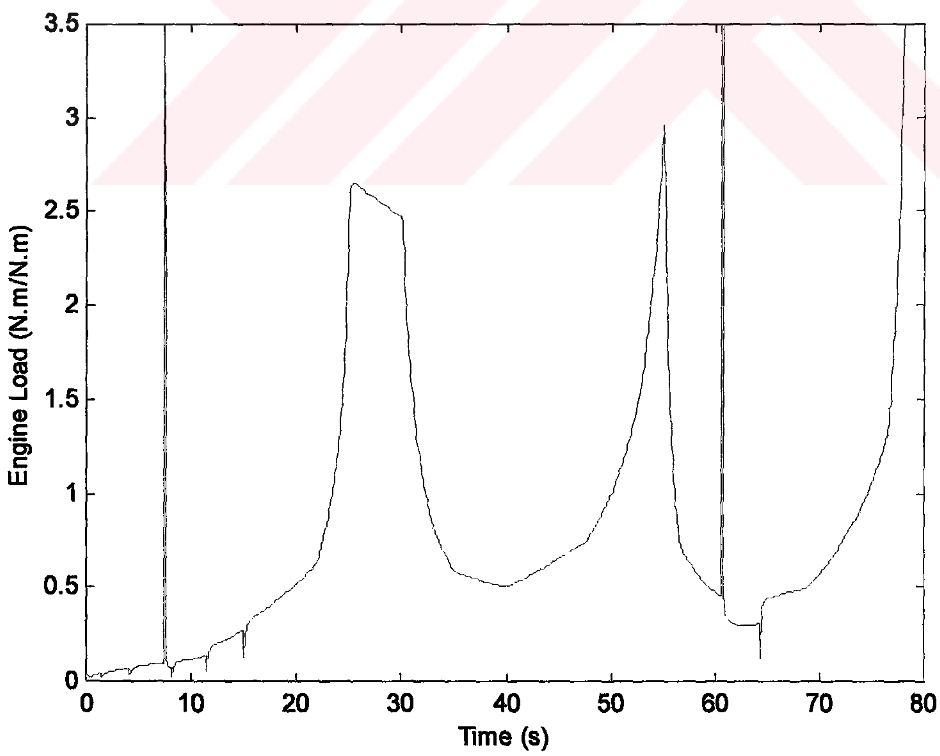

Figure 3.25: Engine Load versus time graph

After 65^{th} second, the throttle pedal is released gradually until getting 0.5 % of throttle at 80^{th} seconds. It should be noted that this amount of throttle couldn't be obtained in real case. However, to simulate the released throttle pedal case, the throttle input is represented by 0.5 % of throttle. This value of throttle is equal to driving the vehicle on first gear with an engine speed of 1500 rpm. The engine speed variation can be observed from figure 3.24.

At the end part of the simulation, as mentioned above, the driver releases the throttle pedal. As obtained in actual cases, it is expected to obtain a gradual decrease in engine speed and vehicle speed. Although in figure 3.21, the vehicle speed shows a gradual decrease due to the superiority of resistances to the engine output torque, in figure 3.20, the engine speed graph shows a sharp drop that does not represent the real case and that does not fit with the vehicle speed variation. The vehicle variation suits the actual case. In addition, in figure 3.21, the engine load during this part of simulation, the ratio of resistances to the engine torque on wheel axle reaches to a value of 900 that can be reasonable because the engine torque gets very low according to the resistances. These deviations occur because the simulation variables like engine speed and torques are related to throttle with equations. Therefore as the throttle input decreases, the engine speed decreases relationally, but in real case, the engine speed does not show similar drop because the engine speed is also related with the vehicle speed, and the engine speed should drop relatively with vehicle speed due to compression of the engine. At this case, the engine compression is not represented.

In figure 3.21, some instantaneous pulses can be observed. These are representing the shift down operations during simulation.

Actually, at this moment, it can be observed that overloading on gearbox occurs although in actual case, this overloading does not occur as much as these values. From first peak, it is seen that the ratio of resistances to engine torque is 15 times and at second peak, this value is 35 times. The rest of the figure operates as expected. When the vehicle accelerates, the engine load ratio remains under unity and when it decelerates, the ratio is above unity.

3.4.6. SIMULATION OF THE VEHICLE ON A HILLY ROAD

For this simulation, the atmospheric conditions stated in previous pages are also valid. This simulation is carried on an asphalt and dry road. However, in this case, the road is not flat and it has a constant gradient. First, it is assumed that the simulation is performed on a 5% continuously inclined road with full throttle input and this inclination is constant during whole operation. As remembered, the only difference from the performed simulation under the title 3.4.2 is the presence of a gradient. Therefore, the results will compare with the results of previous simulation.

The throttle will be kept at 100% opened during whole simulation. The duration is decided 50 seconds to observe all gear shifting operation. As remembered on flat road simulation, 30 seconds time interval was sufficient to observe it.

Initially, the vehicle is at rest and it starts to move with first gear at full throttle. The vehicle accelerates at first gear until the engine speed reaches to 5375 rpm at time equal to 3.55 seconds. At this moment, the vehicle speed is equal to about 43 km/h. The gear is shifted up to second gear. After recovery of the engine speed at 3730 rpm, the engine speed starts to increase. On second gear, the engine speed increases until time shows 7.22 seconds. At this time, the engine speed is 5303 rpm and the vehicle velocity is equal to 76.4 km/h, then the automatic transmission shifts up from second gear to third gear. At third gear, the vehicle reaches to 100 km/h as observed in previous case, but differently, the vehicle acceleration is slower. At this speed, the simulation clock shows 11.8 seconds as can be observed from figure 3.26.

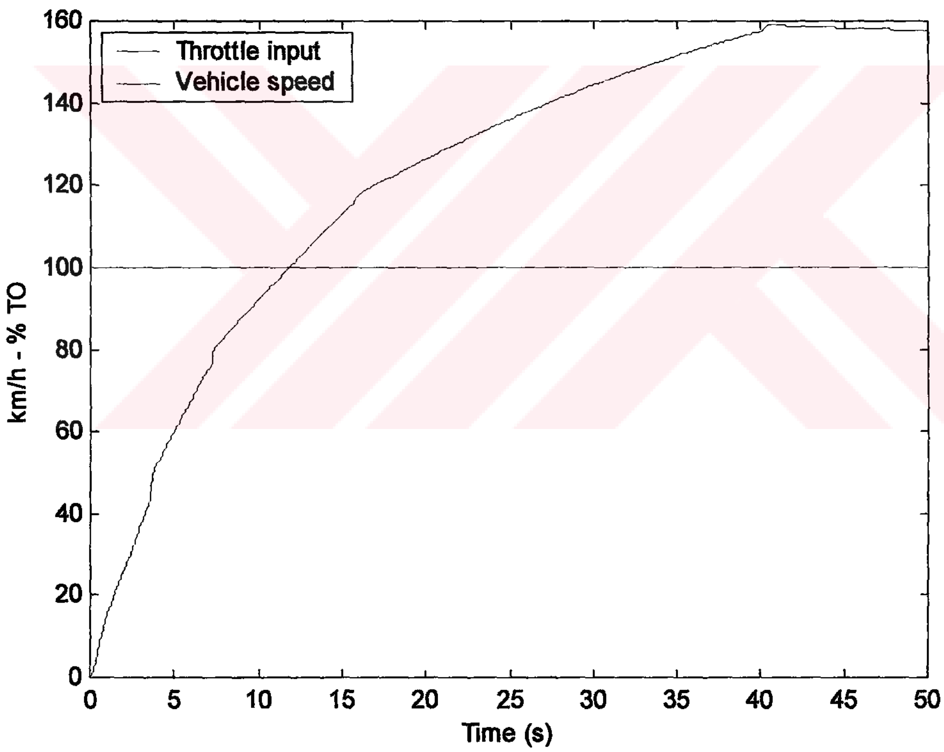

Figure 3.26: Vehicle Speed and throttle versus time

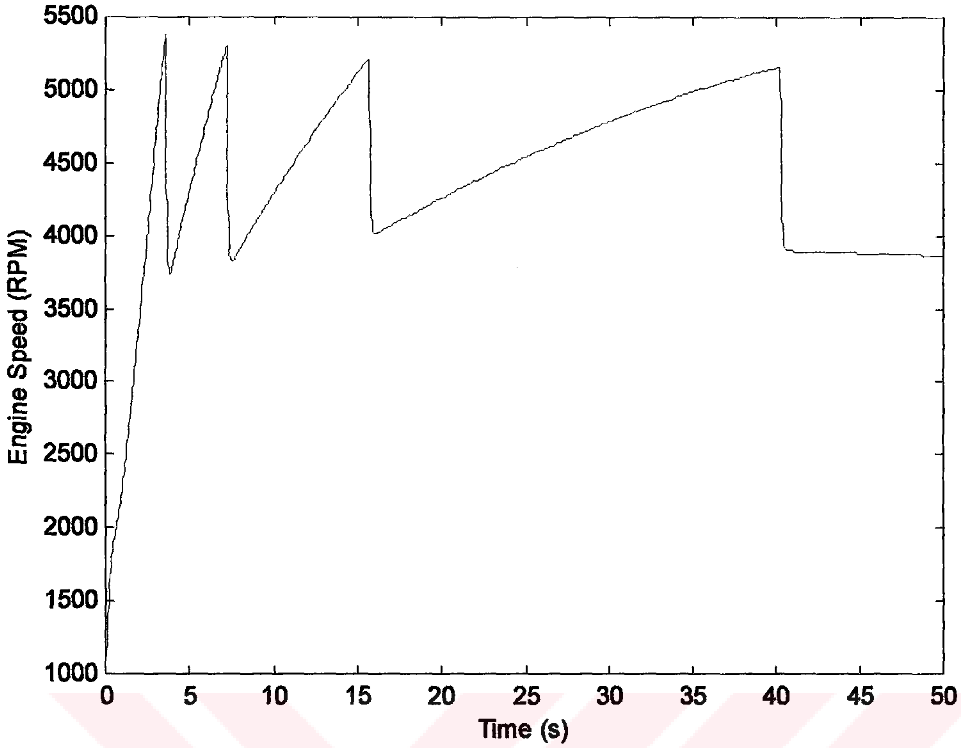

Figure 3.27: Engine speed versus time

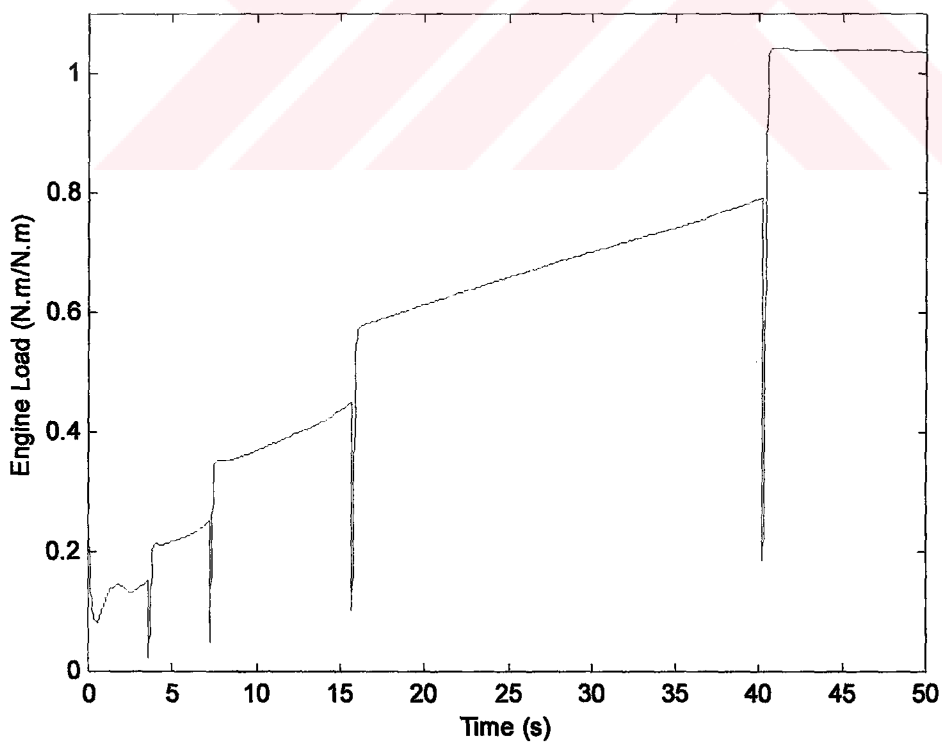

Figure 3.28: Engine Load versus time graph

When the gear is shifted up to third gear, engine speed falls to 3820 rpm and it starts to increase after this speed and at third gear the engine speed reaches to 5210 rpm. After the engine speed reaches to this value at 15.73 seconds, the gear is shifted up to third gear. The velocity of vehicle is 115.5 km/h at this moment. From this velocity, the vehicle is at fourth gear and the engine speed starts to accelerate from 4000 rpm. At 40.2 seconds, the engine speed reaches to 5160 rpm and the vehicle has a velocity of 159 km/h. It is time to shift the gear up to fifth gear. After gear-shifting operation is completed, the vehicle velocity starts to decrease, so the engine speed as can be viewed from figure 3.27. At this gear, the resistances are greater than the engine torque transported to the wheels. The graphical representation can be viewed from figure 3.28. The engine load ratio is slightly higher than unity, and as time passes, it converges slowly to equilibrium position.

From this experience, it can be observed that maximum vehicle speed is obtained by fourth gear on an inclined road with a grade of 5%. At fifth gear, this highest speed (159 km/h) cannot be caught. As observed in simulation on a flat road, the gear changing took place earlier than present case. The gear shifting occurs at 3.29 s, 6.42 s, 12.75 s, and 25.38 seconds of simulation in order from lowest to highest. For an inclined road condition, these times are 3.55 s, 7.22 s, 15.73 s and finally 40.2 seconds. Although at low gears, the gear shifting time is very close, at high gears, the time differences between similar gear-changes gets larger. For instance, for the first gear shifting, while the time difference between two states is only 0.26 second, for last gear shifting, the difference between occurring times is about 15 seconds. The reason is that the ratio of resistances to the engine

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torque transmitted to wheels at first gear is very low, however this ratio at fifth gear is very high compared to first gear. Because, the engine torque transmitted to wheels at fifth gear is smaller than at first gear and in addition, the resistances at first gear are lower because of low velocity that nullifies the air resistance. Therefore, at high gears, to overcome opposing forces becomes more difficult and this results later gear shifting.

In figure 3.28, the engine load variation with respect to time is a bit different from obtained at flat road condition. In present case, the augmentation at each gear occurs slightly as vehicle speed increases. However, in previous case, this increment is steeper. This state proves that the gradient resistances have equal effect with the air resistance while the air resistance is more dominant than the other resistances in first simulation.

The second part includes the simulation procedure made on an inclined road with 10% grade. This amount of grade is maximum inclination that can be observed on state highways. With a full throttle input; the simulation duration is again 50 seconds.

As can be observed from figure 3.30, the automatic transmission doesn't use the fifth gear; the last gear selected for this track is the fourth gear. Because, at this gear, maximum engine torque, which overcomes the resistances, is observed, so the vehicle reaches to maximal speed at this gear. This observation cannot be obtained from the simulation in projected time. However, from figure 3.29 showing the vehicle speed profile and figure 3.30, the state can be predicted easily. This prudence is also improved by inquiring engine load ratios in figure 3.31. If the duration is determined sufficiently long, the top speed at this track will be approximately 140 km/h at fourth gear.

Second gear is selected after driving on first gear until the vehicle speed reaches to 43.2 km/h at 3.85 seconds. When gear shifting occurs, the engine speed has reached to 5380 rpm. The next gear shifting occurs at an engine speed of 5306 rpm at time equal to 8.3 seconds. At this moment, the vehicle forward velocity is 76.4 km/h. During the third gear, the vehicle is driven until the engine speed reaches to 5215 rpm. When gear-shifting operation will take place at this speed at 21.32 seconds, the vehicle speed diagram shows 115.5 km/h.

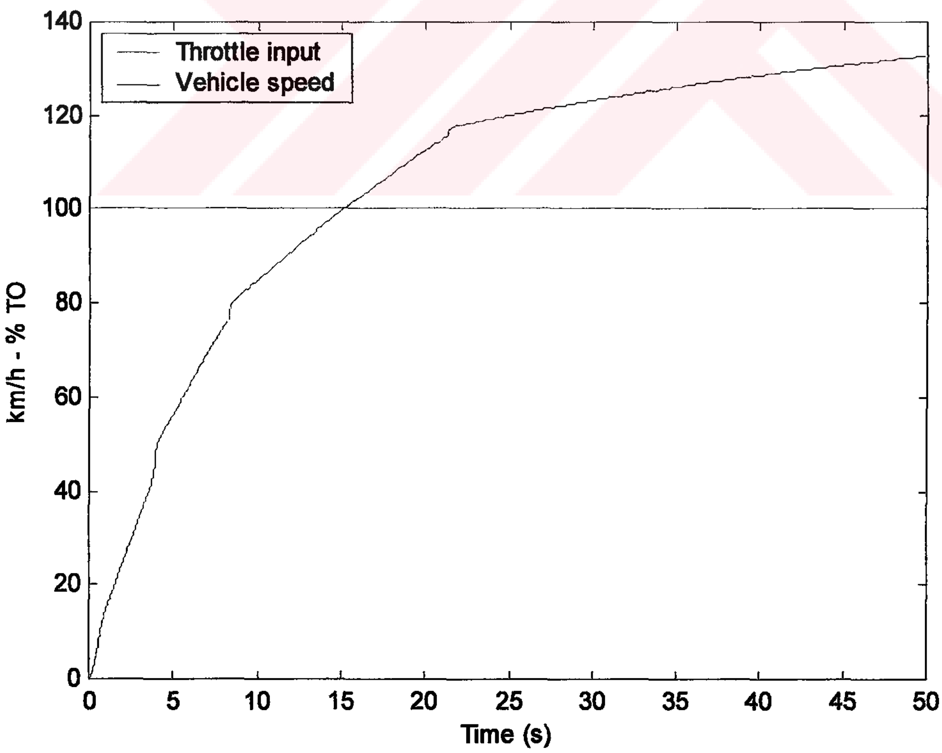

Figure 3.29: Vehicle Speed and throttle versus time

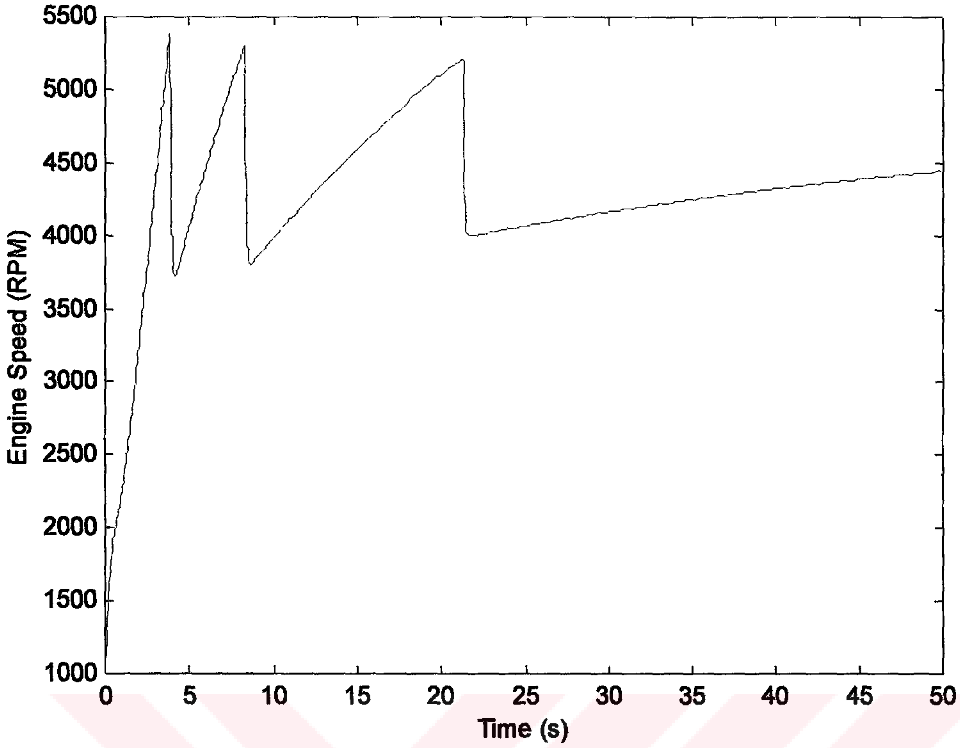

Figure 3.30: Engine speed versus time

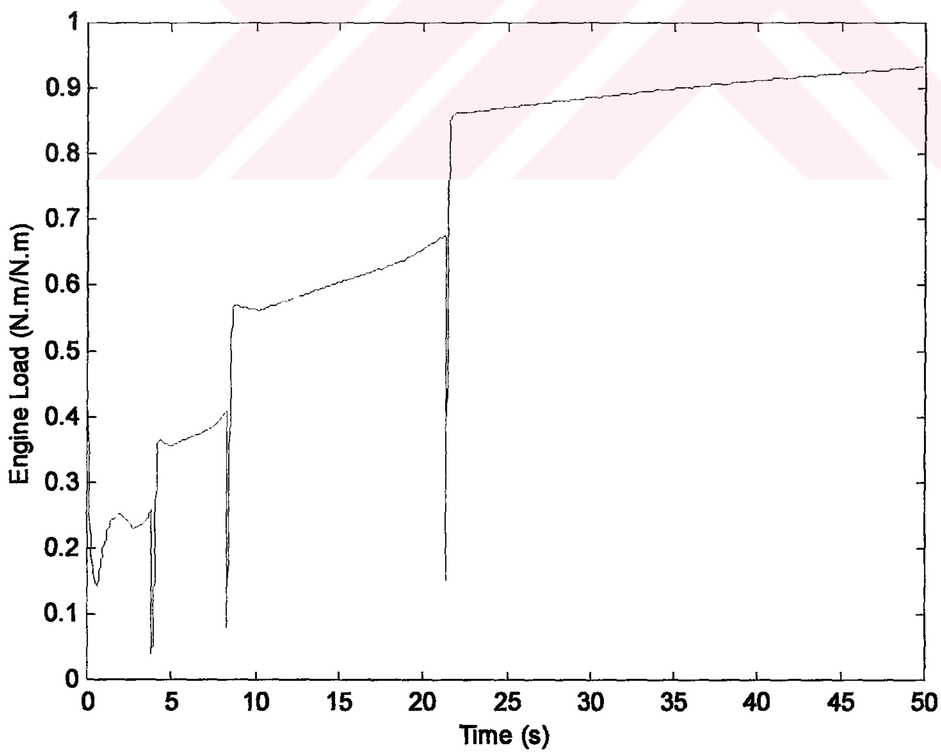

Figure 3.31: Engine Load versus time graph

As mentioned above, the gear shifting operation are delayed as the inclination of roads increase, which is surely expected. To recall the shifting times, let's state them again. On a road with 5% of grade, the shifting operations occur at 3.55, 7.22, 15.73 and 40.2 seconds. On a road inclined with a grade of 10%, the operations take place at 3.85, 8.3 and 21.32 seconds. The gear changing from fourth to fifth gear does not occur within determined simulation time. When comparing these data including the flat road values, the gear changes times at low gears, are very close, although there is a delay at these times. At high gears, the time differences between shifting operations become higher while the road gets inclined because of the reason that is stated in above paragraphs.

Although the timing of shift operations varies for each situation, the engine speed and so the vehicle speed, at which gear shifting occurs, don't differ that much. Because, when the throttle gets very high, the FLC will decide to shift the gears at very high engine speed.

At this part of the simulation, different throttle input is applied to the vehicle running on an inclined road. The grades of the road will be 5% and 10% again. The throttle will be quarterly opened and be kept constantly at this rate.

First, the simulation will be performed for an inclined road having 5% of grade. The engine torque obtained by the input of quarterly opened throttle will be examined by the performance output from the related figures. Following figures of velocity, engine speed and engine load include the simulation results both for a flat road and for inclined road with 5% grade.

First figure, 3.32, shows the vehicle speed variation during both simulations. As expected, at the end of 50 seconds simulations, the achieved vehicle speed on a flat road is higher. In five seconds, the vehicle acceleration in both cases is very close, however as the vehicle speed increases, the acceleration obtained differs from each other. At tenth second of both simulations, the vehicle speed reached on a flat road is 64 km/h, but on an inclined road, this value is 55 km/h. The speeds are actually close when the vehicle moves at low gears. At fiftieth second, the vehicle achieves 123 km/h of vehicle speed on the flat road. On inclined road with 5% grade, 94 km/h of velocity is obtained.

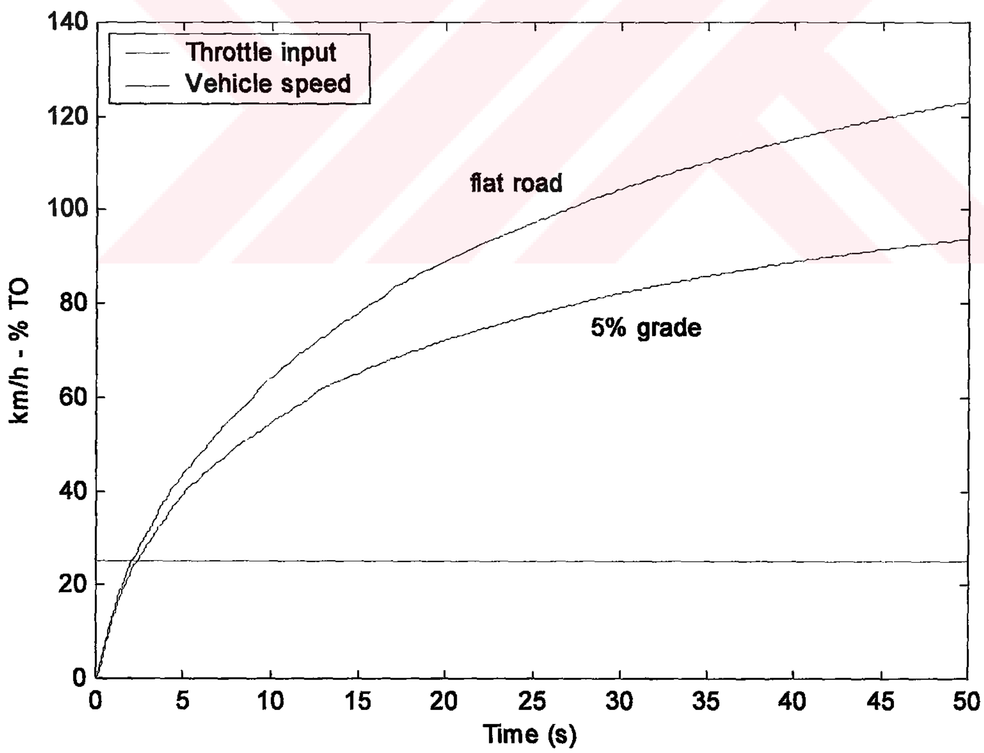

Figure 3.32: Vehicle Speed and throttle versus time

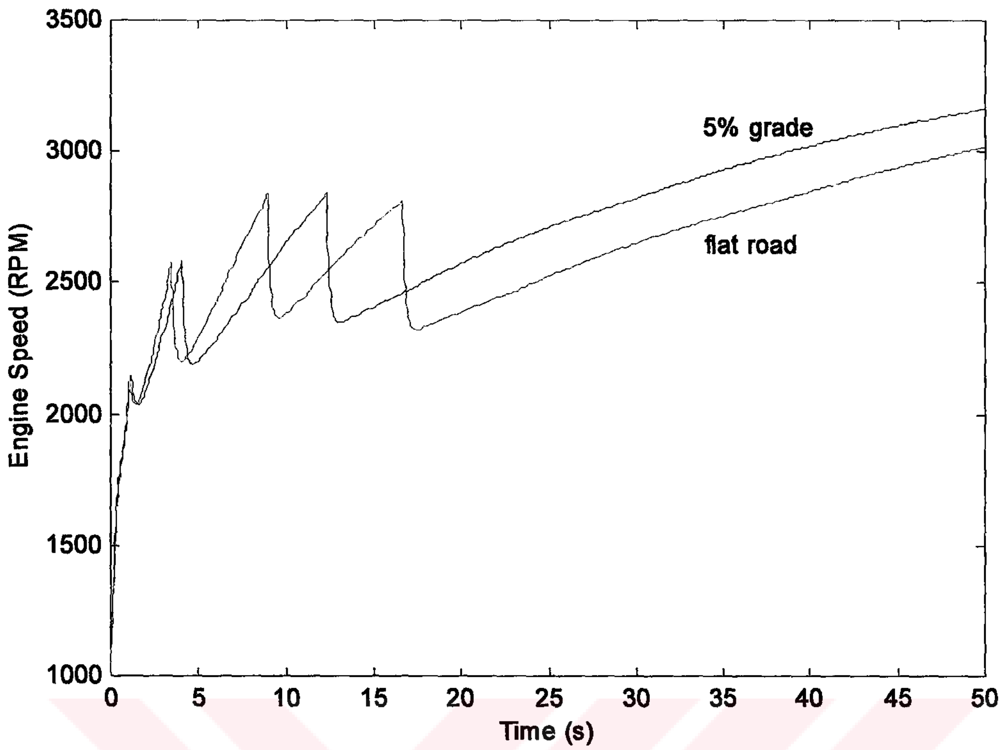

Figure 3.33: Engine speed versus time

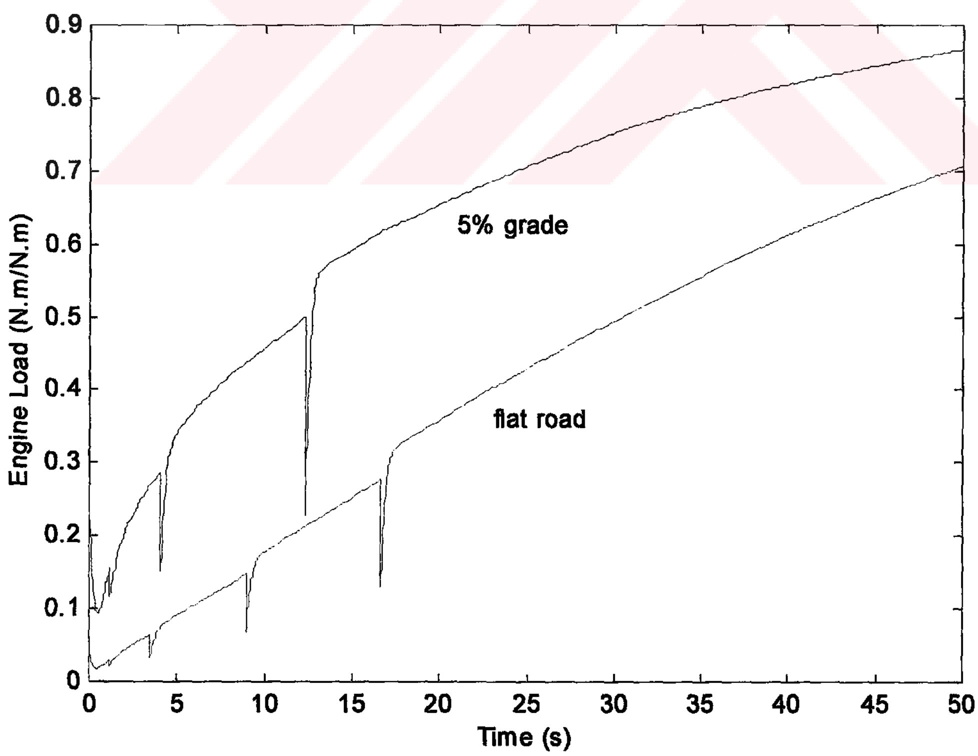

Figure 3.34: Engine Load versus time graph

In figure 3.33, the vehicle's engine speed variation, so the gear shifting operations are observed. The gear shifting operation of first gear to second gear occurs at 1.11 second for the flat road case. For inclined road case, the gear shifting operation occurs approximately at 1.17 second of simulation, which is about the same of the first case. The changes are performed at engine speeds of 2135 and 2147 rpm in the order of previous statement. As time passes, a delay on time occurs on shifting. Especially, from the third gear, this delay gets bigger. In addition, it should be mentioned that on inclined road, the FLC does not use the fifth gear. Therefore, after shifting operation from third to forth gear, the FLC decides to continue the road test at fourth gear by accounting engine load ratio observed in figure 3.34.

If this figure is examined, it is seen that when driving on flat road, the engine load ratio shows a linear increase while the vehicle accelerates by quarterly opened throttle input. As it gets close to equilibrium value, the ratio shows hyperbolic behavior. Actually, it means that the total of all resistances increases proportionally with the provided engine torque for this quantity of input. Therefore, the ratio of them that is represented by engine load ratio increases linearly during the simulation. On the other hand, the engine load ratio for inclined road shows a hyperbolic increase during the entire simulation. During first 20^{th} second part of the simulation, the increase in total resistances is higher than the engine torque. For instance, at fifth second of simulation, the engine load ratio for inclined road is 3.5 times greater than the ratio for flat road. Knowing that the gradient resistance is constant, the total of the resistances is very high with respect to flat road case. At the early stages, the gradient resistance is dominant compared to others. As the vehicle speed increases with time, the dominance of gradient resistance becomes less with respect to other resistance because the air resistance increases proportionally with the square of the velocity. However, if the values are examined, it can be seen that at the end of simulation, the gradient resistance is already higher than the total of other resistances. The gradient resistance value is 560 N approximately, however the total of resistances at the end of simulation is about 400 N. At the last part of the simulation, the ratio observed in flat road gets closer to the ratio on inclined road and as known they will saturate to unity.

For an inclined road with 10% grade, the above simulation is repeated with same variables. The throttle will be kept constantly at quarterly opened position. In the following figures, for the comparison of two cases, the data obtained on inclined roads with 5% and 10% grades will be given. However, at this part, a detailed examination will not be provided, only figures are presented and a short description will be made.

In figure 3.35, the vehicle speed variation is provided with throttle input in percentage. As observed, the vehicle speed increment so the acceleration at fourth gear occurs slowly on a 10% inclined road. That is the result of the engine load ratio in figure 3.37, where it is seen that the engine load ratio reaches to value of 0.95 after shifting to fourth gear. However, for a 5% inclined road, this ratio after shifting to fourth gear is only 0.65.

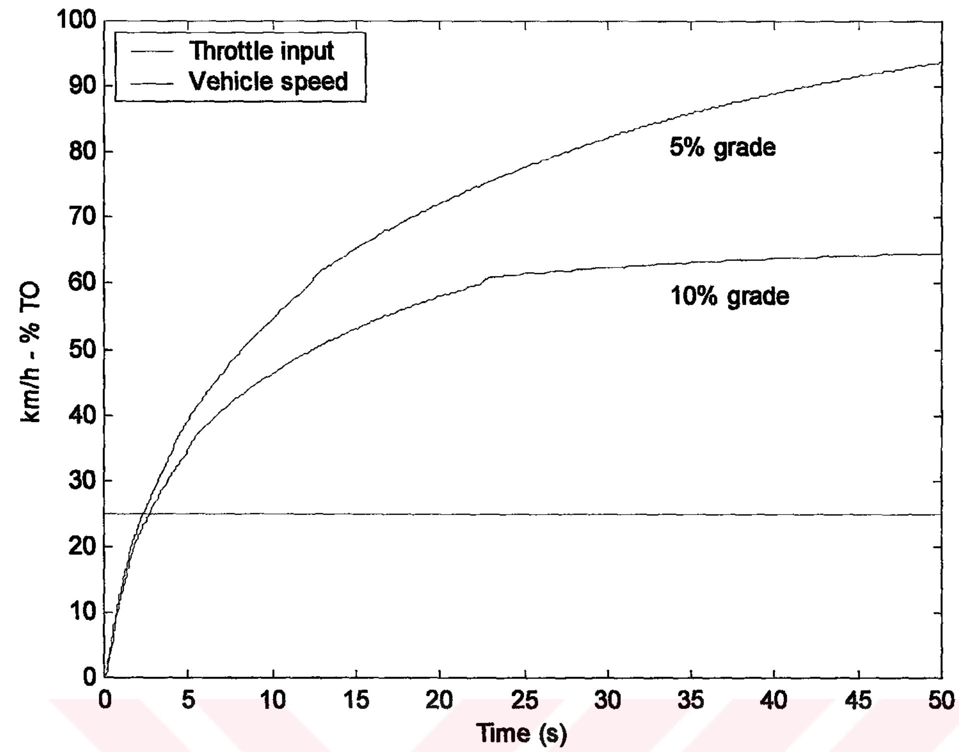

Figure 3.35: Vehicle Speed and throttle versus time

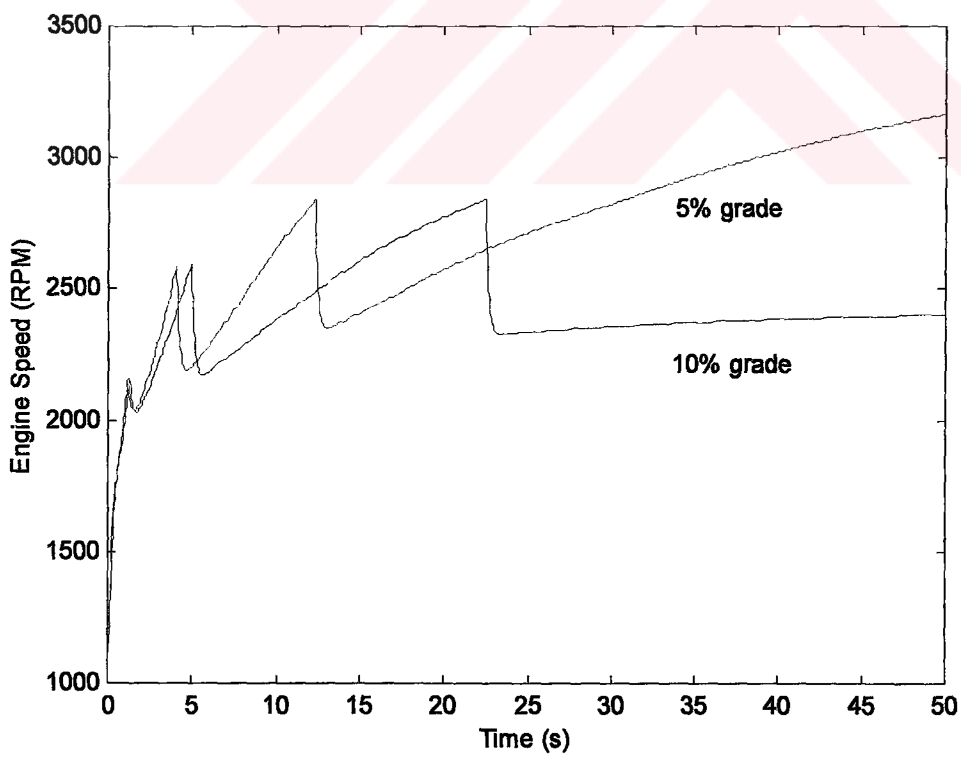

Figure 3.36: Engine speed versus time

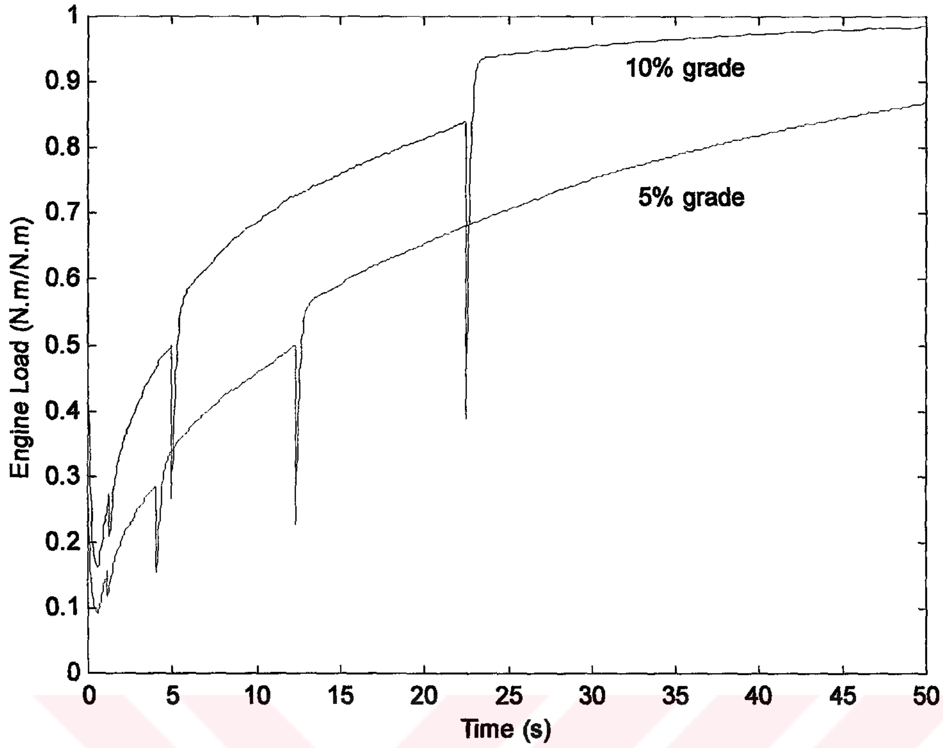

Figure 3.37: Engine Load versus time graph

In figure 3.36 that shows engine speed variation, the gear shifting timings differs increasingly from lowest to highest gear. While the gear shifting from first to second gear occurs about at the same time, there is about one second of delay between both simulations for shifting from second to third gear. At shifting to fourth gear, there is a 10 seconds of delay between two simulations. Moreover, it is also observed that the fifth gear is not selected during both simulations.

3.4.7. SIMULATION OF DRIVING ON A HILLY ROAD WITH VARIED THROTTLE

For this simulation, as suggested for all the simulations performed and described in previous titles, the current ambient conditions are taken into consideration. The road is assumed asphalt and dry. The road will not be constantly flat and it has a road profile with various grades. The throttle is also variable with respect to time and it has the same profile with performed and described simulation in the subtitle of 3.4.5.

The vehicle is initially at rest. During simulation, the front wheels are not steered and the vehicle follows a straight path while climbing up the hill.

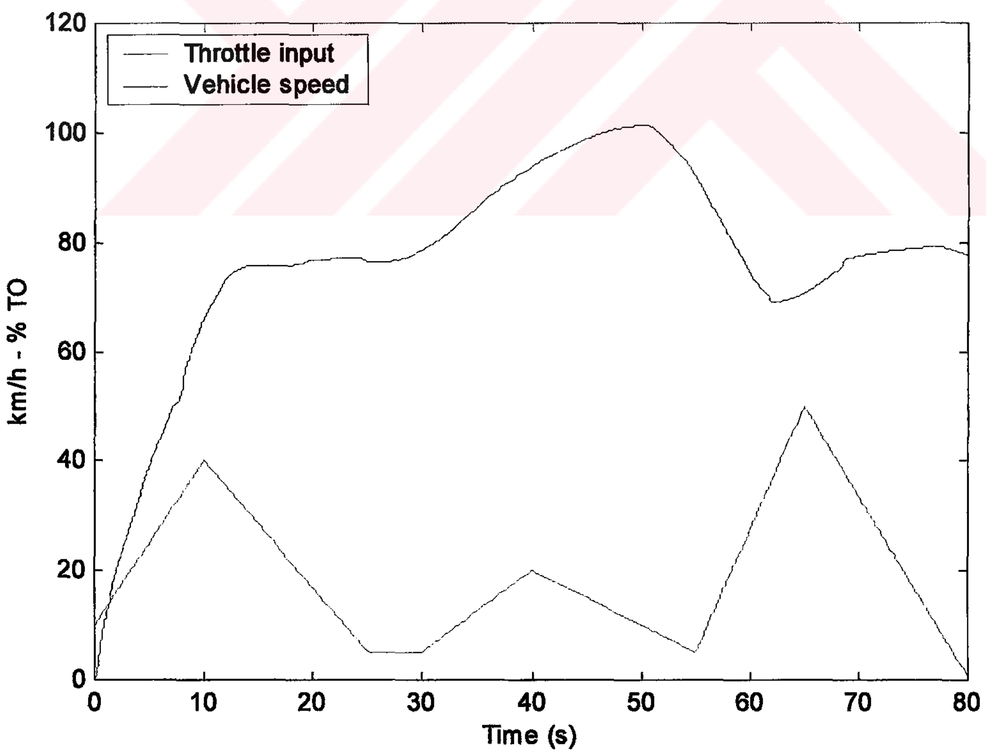

Figure 3.38: Vehicle Speed and throttle versus time

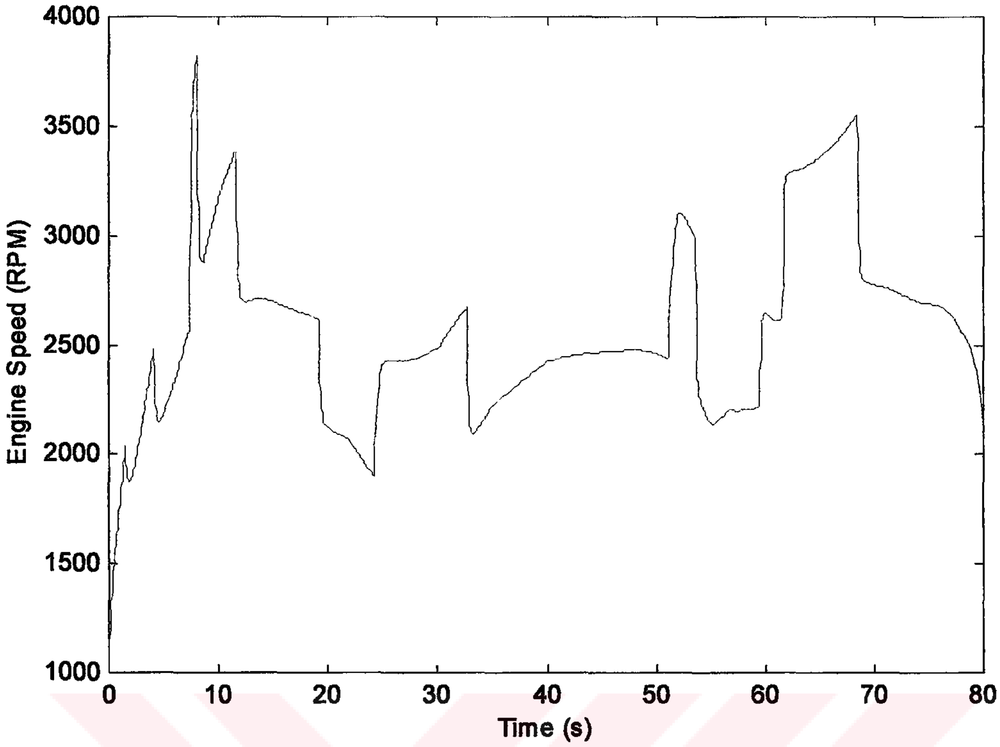

Figure 3.39: Engine speed versus time

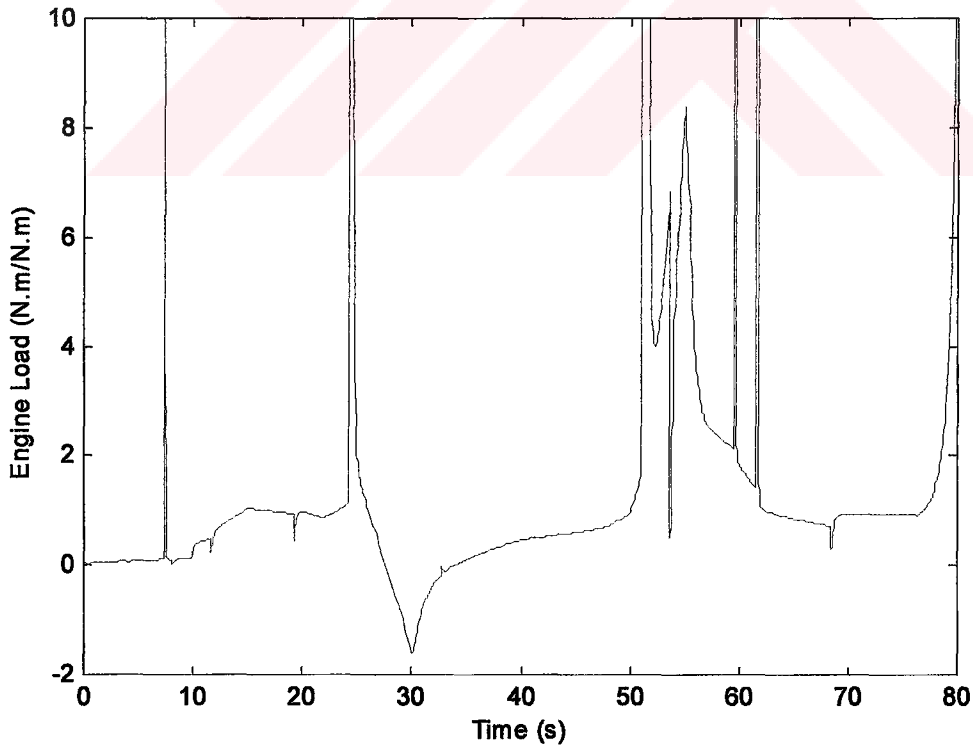

Figure 3.40: Engine Load versus time graph

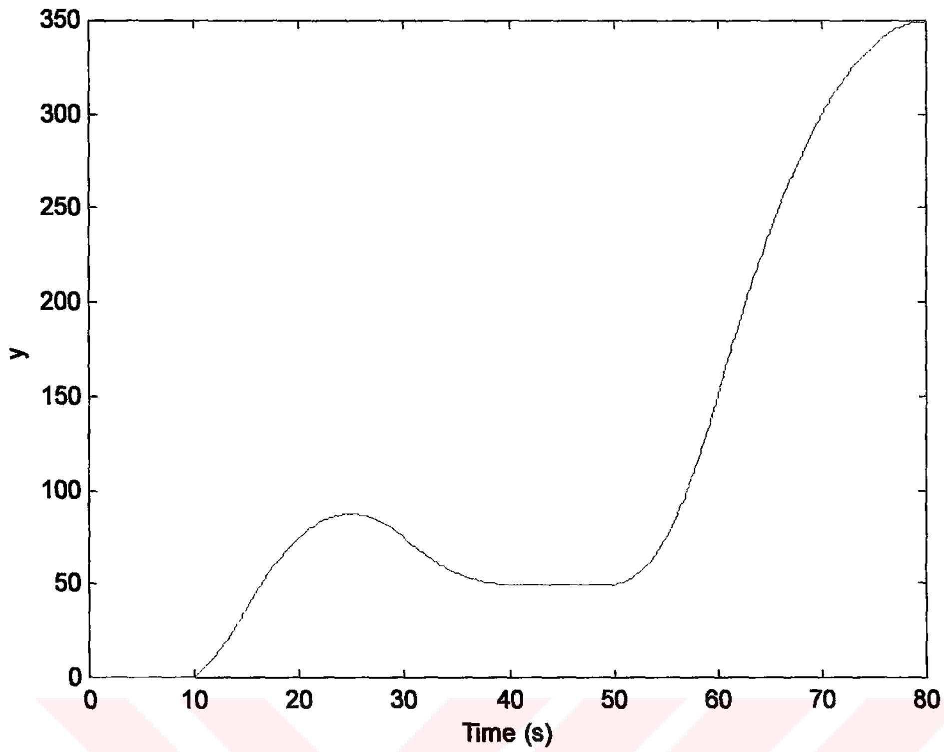

Figure 3.41: Figure representing road profile

The simulation starts with the application of a 10% throttle input at time equal to zero. The entire throttle profile and velocity variation can be observed from figure 3.38. The throttle input is increased linearly to 40% opened value at tenth second of the simulation. During this time interval, the vehicle runs on flat part of the road as seen in figure 3.41 that shows approximate road profile and the gears are shifted up to increase the vehicle speed. The gear shifting operation and so the engine speed variation can be observed from figure 3.39. At this time interval, the gear is shifted up to third gear until time shows the seventh second. At this moment, the gear is shifted down to second gear to acquire the desired acceleration and after one minute, the gear is shifted to third gear again. From this moment, the throttle input is decreased linearly with time until 25th second of the simulation. At 25th second, the throttle input is 5% opened and until this point, the small hill profile is passed and the road becomes flat at this time. While the throttle input is decreased, the vehicle speed is kept approximately constant. At tenth second, the gear was at the third gear. At 12th second, the gear is shifted up to fourth gear and at 19th second, the gear is shifted up again to fifth gear because the throttle input is decreased. However, the engine speed falls steeply, which shows that the total resistance is very high compared to the torque transmitted to the road. So, from a point, at time is equal to 24 seconds, the gear is shifted down to get a higher torque value. After 25th second of simulation, the road becomes downhill until fortieth second. During this time interval, the throttle profile shows different path. For a 5 second of time, the throttle is kept at 5% and it is increased from this value at thirtieth second to 20% opened value at time equal to 40 seconds. During this 15 seconds interval, the gear changing procedure is like that. The gear was four at time equal to 25th second. As the vehicle goes downhill, the vehicle shows higher acceleration and the gear is shifted up to fifth gear at time equal to 32.5 seconds. Accordingly, the vehicle velocity shows increasing intention from a speed of 76 km/h to 94 km/h with time. For 15 seconds of period from forty to fifty five seconds, the throttle is decreased linearly with a slow rate from 20% to 5%. While the throttle is decreased, the road shows flatness during 10 seconds and from the fiftieth second, the road slowly gets hilly. As the road is flat, the gear is kept at number five, but after the road shows a gradient, the gear is shifted down to fourth gear at time is equal to 51 seconds to obtain desired acceleration. After 55th second of simulation, the driver decides to increase the throttle input linearly with time from 5% to 50% in ten seconds. For this period, the road is hilly with a constant grade. The vehicles reaction to these values is driving vehicle at fifth gear until time is equal to 59 seconds. At this point, the throttle input is higher than 22% and the controller decides to shift the gear down to fourth gear. After two seconds of driving, because the acceleration observed does not suit with the desired one, the gear is again shifted down to third gear. This shift schedule is observed from figure 3.39 with two stepwise engine speed variations. The vehicle accelerates in third gear at around 3000 rpm. The acceleration seen from engine speed variation figure shows a hyperbolic increase. This represents that while the vehicle is accelerating, the road gradient is decreasing slightly with time and the throttle is already increasing. The vehicle speed shows continuous deceleration during this time interval until the third gear is selected at time equal to 61.5 seconds. The vehicle speed decreases from 101 km/h to 69 km/h during this period. After this point, the vehicle speed starts to increase at third gear. From the 65th second of the simulation, the throttle pedal is again released and the throttle input is decreased linearly to slow throttle at the end of the simulation. During this time interval, the vehicle increases although the throttle is decreased, because the grade of the road is becoming flat towards the end. After engine speed reaches to 3550 rpm, the controller shifts up to fourth gear when the simulation timer is showing 68.5 seconds. Therefore, during this ride at fourth gear the engine speed shows decrease while the vehicle speed shows slow acceleration. After the 77th second, the vehicle speed starts to decrease and the vehicle shows deceleration while the engine speed starts to decrease steeply. This fact was explained in the previous part of the chapter under the title 3.4.5. The engine load variation can also be viewed from figure 3.40. It is seen that at time interval of 27 to 34 seconds, the engine load ratio decreases under the zero value and falls to minus two values at 30^{th} second. This case occurs because from 25^{th} to 40^{th} second, the road has a downhill profile and the maximum downhill grade is obtained at time is equal to 30 seconds. In addition, according to the road profile, the engine load variation deflects highly because at hilly road, the main resistance is the road gradient.

When compared with the simulation performed on flat road with similar throttle variation, the gear-shifting schedule follows a similar path. At the beginning of both simulations, the schedule occurs similarly. However, at the rest of simulation, the gear selection is slightly different as expectedly. On the other hand, the velocity profiles are very different from each other. When on flat road, the vehicle speed reaches to a high speed of about 130 km/h at 75^{th} second, the vehicle's highest speed at this simulation is 101 km/h reached at 50^{th} second. When comparing the engine load profiles, these two profiles are completely different. At the present case, because the road shows some downhill and uphill profiles, it is observed that the change in engine load occurs steeply compared to the flat road case. These differences are expected because of hilly road profile.

CHAPTER 4

CONCLUSION

4.1 BRIEF OBJECTIVE

The main objective of this research is to design a fuzzy controlled automatic transmission of an automobile. For this purpose, this fuzzy controller is applied to a passenger car in a range of medium size.

During this research, it is aimed to obtain a controller system that shifts the gears to obtain a controller with optimum inputs and rules, in the sense that a minimum number of sensors and hardware are needed. When doing this, the main duties of the controller are not sacrificed.

4.2 DISCUSSION OF FUZZY CONTROLLED AUTOMATIC TRANSMISSION

For this research, a medium 1.8-liter engine car is selected. In most small engine car, the automatic transmission is generally produced with four gears to limit the size and the cost of the transmission. However, in most present cases, five gear automatic transmissions are used in small cars. Moreover, for a car at high-class range, a five speed automatic transmission is preferred. Therefore, a five-gear automatic transmission is selected by considering the class, weight and the engine torque of the vehicle. The fifth gear gives the opportunity of more economical driving rather than a drive with high performance.

In literature, fuzzy controllers using many inputs that can reach up to 12 inputs are observed. However, as concluded before, the design made with four stated inputs also gives successful results and this is performed without highly sacrificing the sophistication, reliability and efficiency of the system. The increase in the input variable number provides more quicker response and higher efficiency. However, for a medium class car, it is shown that the prepared design with four input variables adequately fulfills the requirements.

One of known drawback of traditional transmission control system is that the gear shifting operations are not performed properly under certain driving conditions. However, this simulation proved that a fuzzy logic based control system could overcome some of these problems and shows satisfactory performance.

After covering necessary steps for the design of the fuzzy logic controller, a computer simulation, which represents the dynamic performance of the transmission for an existing vehicle and engine parameters, is prepared. While testing the design, it is important to represent the environmental factors to the simulation to obtain close results in real application. Therefore, various conditions like gear shifting performance of the vehicle on a flat road or on a gradient are tested for the control unit and its reactions for these conditions are noted down. The gear shifting operations are performed efficiently at desired conditions and the gear shifting times are modified according to the road and environmental conditions.

Finally, with this thesis, it is proved that using four inputs, one of which includes the environment effects, is sufficient to have a fuzzy logic controlled automatic transmission fulfilling the desired duties successfully. While performing this, it is also observed that the expectations set at the beginning are not omitted and most of the goals are acquired.

4.3 FUTURE STUDIES ON THE SUBJECT

About this subject, many studies and improvement about the subject can be performed. Some of these improvements are listed as follows.

- Some transmission control schedules like winter or sporty modes may be evaluated based on this fuzzy controller.

- Self-learning algorithms may be applied to fuzzy controller to create a gear shifting operation that is self-adapting to driver's intention.

- The fuzzy controlling on automatic transmission may be widened to whole vehicle, like a drivetrain control including traction and clutch control.

- This research may be applied to scaled or full-scale vehicle to prove the theoretical observations in real state.

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APPENDIX

RULES USED IN FUZZY LOGIC CONTROLLER

All the rules (47 rules) that are used in fuzzy logic controller are given as follow.

IF Vspeed is *xxhigh* **then** Shift is *no5*

IF Throttle is *low* **and** Vspeed is *xxlow* **then** Shift is *no1*

IF Throttle is *low* **and** Vspeed is *xlow* **then** Shift is *no2*

IF Throttle is *low* **and** Vspeed is *vlow* **then** Shift is *no3*

IF Throttle is *low* **and** Vspeed is *med* **then** Shift is *no4*

IF Throttle is *low* **and** Vspeed is *high* **and** Eload is not *vhigh* **then** Shift is *no5*

IF Throttle is *low* **and** Vspeed is *xhigh* **then** Shift is *no5*

IF Throttle is *low* **and** Vspeed is *low* **then** Shift is *no3*

IF Throttle is *low* **and** Vspeed is *high* **and** Eload is *vhigh* **then** Shift is *no4*

IF Throttle is *medium* **and** Vspeed is *xxlow* **then** Shift is *no1*

IF Throttle is *medium* **and** Vspeed is *xlow* **and** Espeed is *low* **then** Shift is *no1*

IF Throttle is *medium* **and** Vspeed is *xlow* **and** Espeed is *medium* **then** Shift is *no2*

IF Throttle is *medium* **and** Vspeed is *med* **and** Espeed is *medium* **then** Shift is *no3*

IF Throttle is *medium* **and** Vspeed is *xhigh* **and** Eload is *high* **then** Shift is *no5*

IF Throttle is *medium* **and** Vspeed is *high* **and** Espeed is *low* **and** Eload is *not vhigh* **then** Shift is *no4*

IF Throttle is *medium* **and** Vspeed is *high* **and** Eload is *vhigh* **then** Shift is *no3*

IF Throttle is *medium* **and** Vspeed is *xhigh* **and** Eload is *low* **then** Shift is *no5*

IF Throttle is *medium* **and** Vspeed is *low* **and** Espeed is *medium* **then** Shift is *no3*

IF Throttle is *medium* **and** Vspeed is *vlow* **then** Shift is *no2*

IF Throttle is *medium* **and** Vspeed is *low* **and** Espeed is *low* **then** Shift is *no2*

IF Throttle is *medium* **and** Vspeed is *high* **and** Espeed is *med* **and** Eload is *not vhigh* **then** Shift is *no4*

IF Throttle is *medium* **and** Vspeed is *high* **and** Espeed is *vlow* **then** Shift is *no4*

IF Throttle is *medium* **and** Vspeed is *xhigh* **and** Eload is *med* **then** Shift is *no5*

IF Throttle is *medium* **and** Vspeed is *xhigh* **and** Espeed is *vhigh* **then** Shift is *no4*

IF Throttle is *high* **and** Vspeed is *med* **and** Espeed is *high* **and** Eload is *med* **then** Shift is *no3*

IF Throttle is *high* **and** Vspeed is *xxlow* **and** Espeed is *high* **and** Eload is *vlow* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *xxlow* **and** Espeed is *high* **and** Eload is *low* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *low* **and** Espeed is *not medium* **and** Eload is *low* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *high* **and** Espeed is *medium* **and** Eload is *vhigh* **then** Shift is *no3*

IF Throttle is *high* **and** Vspeed is *high* **and** Espeed is *not high* **and** Eload is *vlow* **then** Shift is *no3*

IF Throttle is *high* **and** Vspeed is *xlow* **and** Eload is *not low* **then** Shift is *no1*

IF Throttle is *high* **and** Vspeed is *vlow* **and** Eload is *not low* **then** Shift is *no1*

IF Throttle is *high* **and** Vspeed is *xlow* **and** Espeed is *high* **and** Eload is *low* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *vlow* **and** Espeed is *high* **and** Eload is *low* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *med* **and** Espeed is *medium* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *low* **and** Eload is *high* **then** Shift is *no1*

IF Throttle is *high* **and** Vspeed is *med* **and** Eload is *vhigh* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *high* **and** Espeed is *high* **and** Eload is *vhigh* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *high* **and** Espeed is *low* **and** Eload is *high* **then** Shift is *no3*

IF Throttle is *high* **and** Vspeed is *high* **and** Espeed is *not high* **and** Eload is *not low* **then** Shift is *no3*

IF Throttle is *high* **and** Vspeed is *xxlow* **then** Shift is *no1*

IF Throttle is *high* **and** Vspeed is *xhigh* **then** Shift is *no4*

IF Throttle is *high* **and** Vspeed is *low* **and** Eload is *med* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *low* **and** Eload is *vhigh* **then** Shift is *no1*

IF Throttle is *high* **and** Vspeed is *low* **and** Eload is *low* **then** Shift is *no2*

IF Throttle is *high* **and** Vspeed is *low* **and** Eload is *vlow* **then** Shift is *no2*

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