İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

MODELLING AND SIMULATION OF LONGITUDINAL DYNAMICS OF ELECTRIC VEHICLES

M.Sc. Thesis by
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Programme : **Automotive**

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<u>**İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENTİTÜSÜ**</u>

ELEKTRİKLİ TAŞITLARIN DOĞRUSAL DİNAMİĞİNİN MODELLENMESİ VE SİMÜLASYONU

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FOREWORD

In recent years, improvements at electric vehicle technology, which are in parallel with the developments at elektric motors, batteries and power electronic technologies, promise preventive solutions and to eleminate wholly against threats such as increasing costs depending on liquid fuel decreased, greatness of economic effects and emissions. Generally, Electric Vehicles use, which consisting of one or multiple electric motors to take care of propulsion and a battery as energy source. Power and torque of driving motor need to be transformed to meet vehicle requirements and in the meantime, motor and vehicle characteristics need to be guarded. As a matter of fact, high handling performance accompanied with energy consumption, efficiency and emissions is requested by an electric vehicle according to conventional internal combustion engines vehicles.

In the thesis, creating a handling vehicle dynamics simulation model which can work in real time, modelling longitudinal dynamics of handling dynamics simulation software of Surtak driving simulator for an electric vehicle and programming this software in the Matlab environment and development were planned. The software of surtak handling dynamic simulation which is taken as the base in the construction of the project is represented in this work. The creating of Electric motors model instead of internal combustion engine that surtak driving simulator use; battery and control unit models are expressed and the programming of software in Matlab environment for electric and internal combustion engine vehicles are represented in the thesis.

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ABBREVIATIONS

E-VS	**:** Electric Vehicle Simulation Programme
ABC	**:** Active Body Control
SIL	**:** Software In The Loop
HIL	**:** Hardware In The Loop
SURTAK	**:** Sürüş Taklitçisi
EV	**:** Electric Vehicle
ICE	**:** Internal Combustion Engine
PLIB	**:** Polymer Lithium-ion Battery
SOC	**:** State of Charge
ECE-15	**:** City Driving Cycle
ECE	**:** Economic Commission for Europe
EUDC	**:** Extra Urban Driving Cycle
NEDC	**:** New European Driving Cycle
EC	**:** Elemental Carbon
FTP	**:** Federal Test Procedure
ODE	**:** Ordinary Differential Equation
IVP	**:** Initial Value Problems
BDP	**:** Boundary Value Problems

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MODELLING AND SIMULATION OF LONGITUDINAL DYNAMICS OF ELECTRIC VEHICLES

SUMMARY

In the thesis, creating a handling vehicle dynamics simulation model that can work in real time, modelling longitudinal dynamics of handling dynamics simulation software of Surtak driving simulator for an electric vehicle and programming this software in the Matlab environment and development were planned.

In the second chapter, generally handling, vehicle dynamics and simulation, modelling in handling, concepts and levels of vehicle dynamics simulation, advantages and difficulties of vehicle dynamics simulation, the software of Surtak handling, vehicle dynamics simulation and the structure of driving simulator are expressed.

History and structure of electric vehicles and electric vehicle technologies are represented in the third chapter. In the thesis electric vehicle modelling, forces and moments that are affected to the car while driving and which parameters need to be considered at modelling are mentioned in detail. Electric motors are used in electric vehicle instead of internal combustion engine that Surtak driving simulator uses. Modelling of an electric motor and electric motor map are represented in electric motor model section. Then, batteries that electric vehicle use as energy source is generally described. The short information is given about what battery is acceptable for an electric vehicle and the modelling of battery is expressed. New European Driving Cycle is followed by the vehicle to make cruise control. The vehicle is driven by speeding up and down and braking on the new European driving cycle along 10,76km. In the next years, some information is lastly given about what the position of electric vehicle is going to be in the next years.

In the forth chapter of longitudinal dynamics modelling of an electric vehicle and simulation, a tire model was handled, which plays the most important role at the motion of a vehicle consequence of forces and moments affects on it. And accordingly a tire model called Pacejka was modelled and programmed in Matlab environment in the thesis. Pacejka tire model is extensively mentioned in this chapter and some information is given about how longitudinal force is obtained.

Electric vehicle model is consists of two electric motors packed in wheels, battery and control unit models. The model utilizes an electric motor map apart from the equations of tire, battery and electric motors. And this map is represented with internal combustion engine and fuel consumption maps.

The vehicle model is driven in a loop of Matlab programme that was constituted by using the method of Euler. The information is given about how the vehicle is driven in the loop that was constituted and Euler method is shortly expressed in the section.

In the last chapter, calculations are done by Matlab programmes relating to the longitudinal dynamics software of electric and internal combustion engine vehicle models. And the results are evaluated by energy consumption and range. We financially result the energy consumed by electric and internal combustion engine vehicles and their range in conclusion.

ELEKTRİKLİ TAŞITLARIN DOĞRUSAL DİNAMİĞİNİN MODELLENMESİ VE SİMÜLASYONU

ÖZET

Bu çalışmada alternatif tahrik sistemlerini de içerecek şekilde gerçek zamanda çalışabilen özgün bir taşıt doğrusal dinamik simülasyon modeli kurulmuştur. Oluşturulan model bileşenleri arasında klasik tahrik makinası olan içten yanmalı motor olduğu gibi, elektrik motoru ve batarya modelleri de bulunmaktadır. ITU otomotiv laboratuarında bulunan SURTAK sürüş simülatörü seyir dinamiği simülasyon yazılımının doğrusal dinamiği ele alınmış ve simülasyon yazılım ortamı olarak MATLAB seçilmiş, tahrik konseptleri arasında yakıt ve enerji tüketimi, işletme maliyeti ve menzil gibi temel unsurlar karşılaştırmalı olarak hesaplanıp sunulmuştur.

Çalışmada öncelikle gerçek zamanda çalışabilen özgün bir taşıt seyir dinamiği simülasyonu, elektrikli taşıtların yapısı ve elektrikli araç teknolojileri, genel olarak elektrikli taşıtların modellenmesi, elektrikli tüm taşıt modelleri, elektrik motorlarının modellenmesi, bataryaların yapısı ve hangi batarya tiplerinin elektrikli taşıtlar için uygun olduğu ile ilgili bilgi verilmekte ve bataryaların basitçe nasıl modellendiği açıklanmaktadır.

Modelleme ve simülasyon kısmında ise, öncelikle Matlab ortamında programlanmış olan Pacejka tekerlek modeli açıklanmaktadır. Bilgisayardaki taşıt doğrusal dinamik modeli ile çeşitli tahrik tipleri ve konfigürasyonlar için NEDC Avrupa cevrimi takip edilerek taşıt durum değişkenleri ve temel karşılaştırma büyüklükleri incelenmiştir. Günümüz piyasa enerji fiyatları kullanılarak klasik içten yanmalı motorla tahrik (dizel motoru) ve elektrikle tahrik konseptleri karşılaştırılmıştır.

İlk bölümde, genel olarak taşıt seyir dinamiği, simülasyon, taşıt dinamiği simülasyon kavramları ve aşamalarından, taşıt dinamiği simülasyonunun avantajları ve zorlukları ile projenin yapımında taban olarak aldığımız Surtak seyir dinamiği simülasyon yazılımından ve sürüş simülatörünün yapısından bahsedilmektedir.

Taşıtlarının Dinamiği; Doğrusal Dinamik, Yanal Dinamik ve Düşey Dinamik olmak üzere üç ana bölüme ayrılır. Doğrusal Dinamik: taşitin seyir yünündeki ivmelerini, Yanal Dinamik: seçilen doğrusal seyir özellikleri eşliğinde direksiyon sapması cevabını, (taşıt gövdesi ve akslarının yan yöndeki ivmelerini, gövde yalpasını, düşey eksen etrafında dönmesini ve yüzmesini) ve Düşey Dinamik: karayolu zemininin düşey yöndeki profilinin etkisi altında cadde normali yönündeki gövde ve aks ivmelerini, dinamik tekerlek yüklerini ve kafa vurma hareketlerini incelemektedir. Taşıta ait bu üç dinamiğin beraberce, gerçek zamanda tüm çalışma bölgesi için çözülmesi zorunluluğu vardır.

Taşıt Seyir dinamiği kavramından da anlaşılan, sürücü-taşıt-çevre kapalı kontrol çevriminin beraberce davranışıdır. Bu çevrimde frekansı en fazla 5Hz'e kadar olan değişimler rol oynar. Bu frekansların üzerinde kabaca 20 Hz'e kadar, direksiyon sistemi darbe davranışı, Shimmy hareketleri gibi efektler ortaya çıkar. Yaptığımız Tez çalışmasında 5Hz'e kadar olan dinamik olaylar incelenmektedir. "Seyir dinamiği" adı verilen bu araştırma bölgesinde taşıt açısından 4 etken grubu rol oynamaktadır:

1- Temel Konsept
2- Lastik Tekerlek
3- Tekerlek Asılışları
4- Gövde Kontrol Sistemleri

Taşıt dinamiği simülasyon kavramlarına bakılırsa; Gerçek bir sistemin modeli üzerinde deneyler yapmak Simülasyon olarak tanımlanmaktadır. Gerçek bir sistemin Matematik Modeli üzerinde deneyler yapmak Bilgisayar Destekli Simülasyon, Hareket denklemlerinin zaman boyutunda nümerik integrasyonu ise Zaman Simülasyonu olarak adlandırılmaktadır.

Gerçek karayolu deneylerinin zorluklari ve Taşit Dinamiği simülasyonunun avantajlari sunlardir:

Gerçek karayolu deney ve test çalışmalarının zorlukları; zaman alıcı olması, maliyetlerin yüksek olması, taşit ve varyasyonlarının gerçek olarak elde edilmesi zorunluluğunun zaman ve maliyet bakımından pek çok dezavantaj getirmesi, deney parametrelerinin, özellikle de dış etkenlerin sürekli değişkenlik göstermesinden dolayı dış etkilere ve aşınma etkilerine maruz kalması, test sürücülerinin karar kriterlerinin zamanla değişmesi, kayması ve unutulması, dolayisiyla genelleştirilemeyen bir durumun söz konusu olması, müşteri istek ve kabiliyetlerinin test sürücüsü olarak kullanılan uzman pilotlarınkilerle örtüşmemesi, fiziksel seyir sınırlarının incelenmesinin can ve mal emniyeti kriterleri açısından olumsuzluklar taşımasıdir.

Taşıt dinamiği simülasyonlarındaki avantajlar; geliştirme sürecinin erken safhalarında (elde fazla parametre yok iken) testlerin mümkün olması, maliyetin az, ucuz (hesaplı), tekrarlanabilir, tam protokollenebilir (kayit altina alinabilir), izole parametre etütleri mümkün, bozucu etkilerden arınmış, tahribatsız, aşınmasız ve can ile mal emniyeti açısından tehlikesiz olmasidir.

Ozetle projenin yapımında taban olarak aldığımız Surtak seyir dinamiği simülasyon yazılımına deginirsek; sürüş simülatörü temelde çok sayıda yazılım ve birkaç donanım parçasından oluşmaktadır. Pek çok avantajı ve aynı zamanda kendi mevcut teknik özelliklerinin de getirdiği sınırlamalar dahilinde, tasarlanmış bulunan sürüş simülatörü "SürTak", taşıt dinamiği ve sürücü davranışları ile ilgili neredeyse her alanda araştırma yapmaya imkan tanımaktadır. Belli hareket bölgeleri için belli oranlarda doğrulanmış gövdesi 5 serbestlik dereceli (3 birincil hareket + 2 ikincil hareket) ve tamamı 11 serbestlik dereceli bir binek taşıt modeli üzerine kurulmuş olan sistem, Visual C++® programlama dilinde oluşturulmuş bir bilgisayar koduna dayanmaktadır. Sürüş simülatörünün temel öğeleri olan görüntü, ses ve sürücü ara yüzü için Microsoft DirectX® kütüphanelerinden destek alınmaktadır. Serbestlik derecelerini içeren denklemler dışında çoğu alt-sistem için içten yanmalı motor momenti haritası, yakıt tüketimi haritası, trilok dönüştürücü pompa çarkı katsayısı eğrisi ve moment dönüştürme eğrisi, otomatik vites kutusu planet grubu verim haritası ve vites değiştirme haritası gibi bir takım haritalar kullanmaktadır.

Bir sonraki bolum olan elektrikli araçlar kısmında ise, elektrikli araçların tarihi, yapısı ve elektrikli araç teknolojileri anlatılmaktadır. Elektrikli araçların modellenmesi, taşıta seyir esnasında etki eden kuvvet ve momentler ve hangi parametrelerin dikkate alınması gerektiği detaylı olarak aciklanmaktadir. Projenin yapımında taban olarak alınan sürüş simülatörünün, kullanmış olduğu içten yanmalı motor yerine, bir elektrik motoru kullanılmakta ve bu elektrik motorunun modellenmesi ile birlikte motor modelinin kullanmış olduğu elektrik motoru haritalarina yer verilmektedir. Elektrikli taşıtların enerji kaynağı olarak kullandığı bataryaların yapısal olarak tarifi ve hangi bataryanın elektrikli taşıtlar için uygun olduğu ve bir bataryanın nasıl modellendiği aciklanmaktadir.

Bilgisayardaki taşıt doğrusal dinamik modeli ile çeşitli tahrik tipleri ve konfigürasyonlar için NEDC Avrupa cevrimi takip edilmekte ve taşıt durum değişkenleri ile temel karşılaştırma büyüklükleri incelenmektedir. Taşıt, bu cevrimi takip ederek (gaza basarak, ayağını gazdan çekerek yada fren yaparak) hareketini sürdürmektedir.

Bir elektrikli taşit modeli tasarlanırken güç organlarının dizilisi, birbirleri ile bağlantı şekilleri, batarya ve elektrik motoru seçiminde çeşitli optimizasyon çalışmalarının yapılması gerekli olup; maliyet, emisyon, verim, bakim masraflari, sürüş konforu, güvenilirlik, performans ve menzil gibi bir takim hedefler göz önünde bulundurularak, elektrikli taşıt tasarımları gerçekleştirilmelidir.

Olusturmus oldugumuz elektriği taşit modelimiz; arkadan tahrikli bir taşıt olup, modelimizde aracı tahrik etmek için kullandığımız tekerlekler içerisine yerleştirilmiş 2 adet elektrik motoru ve enerji depolamak için kullandığımız bataryalar yer almaktadır. Tüm hareket gücünü sağlayan ve araç üzerindeki diğer enerji ihtiyacını karşılayan bataryalardaki enerji; batarya sisteminin güvenli şarj/deşarj işlevi ile bataryanın ömrünü belirleyen, aracın geri kalanıyla haberleşmeyi sağlayan bir elektronik kontrol ünitesi vasıtasıyla tekerleklerdeki elektrik motorlarına iletilmektedir. Bunu yaparken aktarma organları ve diferansiyel gibi sistemleri kullanmıyor olmamız verim, güvenilirlilik ve performansı arttırırken bakım ihtiyaçlarını azaltmaktadır.

Ayrica, elektrikli taşitlarda kullanılacak bataryalarda; yüksek enerji yoğunluğu, düşük ağırlık, uzun servis ömrü, güvenilirlik, geri dönüşüm kolaylığı, yüksek şarj kabul kapasitesi, şarj derinliği, düşük ısı üretimi, batarya ve ortam sıcaklığı, belirli sıcaklık değerleri arasındaki performans isteklerinin karşılanması ve kendi kendine düşük deşarj gibi özellikler aranmaktadır.

Elektrikli bir taşıtın doğrusal dinamiğinin modellenmesi ve simülasyonu kısmında da, gerek üzerine etkiyen momentler gerekse kuvvetler neticesinde taşıtın hareketindeki en önemli rolü oynayan, belki de bir taşıt dinamiği simülasyonun en önemli parçası olan etkin bir lastik tekerlek modeli ele alınmış ve Pacejka Tekerlek modeli modellenmiş ve Matlab ortamında programlanmıştır. Bu bölümde Pacejka tekerlek modelinden geniş olarak bahsedilmekte ve doğrusal kuvvetin nasıl elde edildiği ile ilgili bilgiler verilmektedir. Elektrikli taşıt modeli, tekerlekler içerisine yerleştirilmiş iki adet elektrik motoru, batarya ve kontrol ünitesi modellerini içermektedir. Model tekerleklere, bataryalara ve elektrik motoruna ait serbestlik derecelerini içeren denklemlerin dışında bir elektrik motoru haritası kullanmakta olup ve bu harita içten yanmalı motora ait moment ve yakıt tüketimi haritaları ile birlikte bu bolum içerisinde verilmektedir. Taşıt, Matlab yazılımında, Euler metodu kullanılarak oluşturulmuş bir döngü içerisinde hareketini sağlamaktadır. Oluşturulan bu döngü içerisinde taşıt nasıl hareket ediyor ile bilgi verilmekte ve Euler metodundan kısaca bahsedilmektedir.

Sonuç olarak, içten yanmalı motora sahip taşıt modeli ve elektrikli taşıt modeli doğrusal dinamiği yazılımına ait programlar ile ilgili hesaplamalar yapılmakta ve sonuçlar, günümüz piyasa enerji fiyatları kullanılarak klasik içten yanmalı motorla tahrik (dizel motoru) ve elektrikle tahrik konseptleri icin, yakıt ve enerji tüketimi ile maliyet ve menzil gibi temel unsurlar karşılaştırmalı olarak hesaplanip sunulmakta ve yorumlanmaktadir.

1. INTRODUCTION

Electric vehicle concept had completed its journey from Arge departments of automobile sector through the prototype factory plant in the last ten years. Journey through the multiple production lines had partially been finished. Modern day vehicles are driven by internal combustion engines commonly. Electric vehicles will be preferred predominantly in the next decade by the improvements at electric vehicle technology which are in parallel with the developments at technologies of electric motors, batteries and power electronic, lowest rumble levels, zero emission aspects and high energy efficiency that electric vehicles have.

If we considered in the terms of our country, it’s seen that all sources of oil are stick to foreign countries and transportation is provided by public highways substantially besides of the deep effects are expected in the long-term. Accordingly, even the slightest proceeds of increase in productivity that will be provided at transportation will be very high.

Innovations that electric vehicles bring were shortly summarized in shorthand below:

- Electric motor was developed by using modern electric drive technology is consist of power converter and energy source.
- Electric energy is stored in battery.
- It’s a fundamental transformation that the transportation service will be provided by zero emission and higher efficiency beyond a new vehicle concept.
- Electric vehicles will provide to create smart systems which are appropriate to modern transportation network.
- Operating conditions and working loops will be re-defined.
- End user, each maintenance and production levels, education, standardisation and infrastructure in relevant sectors will occur.

Structure and the software of a driving simulator are firstly mentioned in this thesis. The history and general description of electric vehicles, electric vehicle modelling, structural description of battery and battery modelling, drive cycles, Pacejka tire model definition and modelling, the programme calculations relative to longitudinal dynamics software of electric and internal combustion engine vehicle models are investigated. The models, which are programmed by Matlab, follow New European Driving Cycle to make cruise control. And the results are investigated and evaluated by energy consumption and range. Conclusions are indicated that how far the vehicle could go and how many kW of energy consumed financially. Similar results are obtained for the internal combustion engine vehicle.

2. VEHICLE DYNAMICS SIMULATION

2.1 Handling, Vehicle Dynamics and Simulation

The main purpose of industrialized and developing countries is to protect their competitiveness by putting successive products into the international market and to design developed products, produce and develop them. The communication between financial record, design and test departments which are the main phases of product development has gained importance. Putting more developed and economical products into the market in a short time is possible in the case of providing this communication and coordination efficiently. Automotive industry takes an interest from all developments in technique. A motor vehicle is an extremely complicated technique system and thus, each step of development is achieved via huge efforts.

Handling, is the behaviour of driver-vehicle-environment closed control loop together (Atabay 2004). The frequency that play a role in the loop are till 5Hz. Some kind of effects like shimmy motions appear over these frequencies, nearly till 20Hz.

Four factor groups play a role in the field of vehicle dynamics:

1- Fundamental concept: Wheelbase, track width, the location of centre of gravity, aerodynamic parameters and etc.

2- Pneumatic tire is most important factor at investigations till 5 Hz. Obtaining even the Stationary characters has been a problem up till now. Moreover, Handling, is a transient chain of events which goes over statonary events.

3- Suspension system elements have connective task between vehicle body and wheels. Kinematic and elastokinematics characteristics affect motion characters. Kinematic characteristics are rigid body kinematics of the connection of suspension arms. Elastokinematic characteristics try to explain the positional behaviour of rim under forces and moments. Life (fatigue), strength, assembly volume and cost are boundary conditions of suspension systems.

4- Control systems like ABS, ASR and ESP were improved for extreme cases of Handling, in nowadays. These systems are put into use rarely in the linear vehicle dynamics where the accelerations occur under 4 m/s^2. And they could be seen as a useful additions although they are basic. However, recent investigations are about systems, which interfere to the motion of the body continuously, like ABC (Active Body Control) systems.

2.2 Modelling in Handling, Vehicle Dynamics and Simulation Concepts

Vehicle Dynamics can be separated to three main chapters: (Atabay 2004)

1- Longitudinal Dynamics: Investigates the longitudinal acceleration of vehicle in the moving direction as a result of engine throttle position, transmission ratio, clutch control or hydrodynamic convertor and brake pedal force.

2- Lateral dynamics: Investigates steering response (Lateral acceleration of vehicle body and axles), body rolling, yaw and side slip, accompanied by selected longitudinal motions characteristics (velocity, acceleration).

3- Vertical dynamics: Investigates vertical accelerations of vehicle body and axle, dynamic wheel loads and the pitch motion of the vehicle body.

The basis of vehicle dynamics simulation is the response of equationsof vehicle components and subsystems, which are affected by external and internal forces and moments.

A simulation model must be sufficiently in detail depending on the aim of application. However, it mustn’t be so complicated. Basic models can be seen very useful to eliminate practicable ideas, especially at concept phase of vehicle design in the case of whole detailed vehicle data set does not exist.

Tire forces which originated from torque, engine speed, brake system and steering system play the most important role at the motion of a vehicle. In this regard, an active tire model is probably the most important part of vehicle dynamics simulation. Tire models must generate reliable results even while operating parameters is approaching to boundary values. Tire models should be able to response against to longitudinal and lateral slip, camber angle and quick variations of wheel load. Nonetheless, a model of handling (vehicle dynamics) simulation is a tough problem to generate (Atabay 2004).

A good correlation are generally required between the real test measurements and simulation results depending on the area where simulations are performed. Many simulation models are calibrated to represent only some special maneuvers as indicated in Atabay (2004). However, the main task naturally is, accurate correlation of results must be generated by the model at each evolution combination of vehicle dynamics manuevers.

Movements, which have low-frequency and high amplitude but not have linear characteristics, are the point in vehicle dynamics. Investigating the system analytically is nearly impossible because of nonlinear system behaviours and many degrees of freedom.

Time simulation, which is the numerical integration of equations in time, is the only method that can be used for many important applications. Digital calculators can only able to execute discrete operations.

Behaviours of driver-vehicle-environment closed control loop are tried to be explained by handling characteristics. Reactions which are occurred by driving that depend on design parameters of vehicle are followed by the effects of driver to vehicle at the time of driving. Driver controls the vehicle by the steering, engine and brake system. In the meantime; position, velocity and acceleration of the vehicle are kept in required values and thus the vehicle behaves by desired route of the road and map.

Moving characteristics, traffic rules and actual traffic situation are considered by the driver. Driver remains in effect of alcohol, exhaustion, medicine and psychologically situation. Inertia and control forces occur at the time of driving effects to driver attitudes. Environment conditions (as ground features, lateral winds etc.) play a role at the behaviour of system. The most significant aim of automotive industries is to increase the safety of vehicles. Active safety of vehicles is referred here to avoid the accidents. However, those efforts are full of difficulty in consequence of the increasing of traffic flow.

Simulation subroutines which are designed for automobiles are corresponding real hardware parts.

In this case, 4 phases are referred as a combination of mechanics, hydraulics, pneumatic, electric and electronic systems for acceptable vehicle dynamics simulation.

- "Off-line" Simulation: Subsystems of vehicle like mechanics, hydraulics and pneumatic etc. are constituted as virtual at simulation. Basis algorithms and rules at control units, which were attached to whole vehicle model, are developed and tested together. All parameters based on mechanic simulation of vehicle are obtained at this stage, without thinking of electronic control units.

- Control Prototyping: Control unit algorithms which are wanted to be constituted (vehicles are now adays consist of many control units) are tested by real automobiles nevertheless the investigations remain on computers. Complex impacts which are not considered at off-line simulations are monitored and fixed. Controllers on vehicles, which don't provide a freedom to person who develops the software, are basic controllers optimized by considering the cost of it.

- SIL (Software-in-the-Loop): In this phase, Algorithms which were prepared for control units are inserted into a control unit assumed as a real by constituting virtual control unit hardware. This virtual control unit is tested by simulation or software developed privately.

- HIL (Hardware-in-the-Loop): System, which is controlled, is constituted virtually and implicated into the simulation loop as a real element in this kind of simulations (Atabay 2004). Whole vehicle actuators and sensors are simulated. Simulation must be fed by all signals at exact time, which the control unit needs. The frequency of data gathering of control unit is indicated and generally fixed. And that can be possible to be provided an association between computation time and validity at modelling of whole system that was simulated out of control unit. And so, simulation has no right to make the control unit wait.

An off-line model may consist of many algorithms that need to computation time. However, if a real component will be implicated to virtual world of simulation, this component must be fed with signals in real time by simulation and the simulation should be able to collect reactions of it. A user interface that generally works on monitor is constituted on basic HIL simulators work in real time. And there are cumulative virtual buttons and switches on this monitor to simulate the system by mouse. However, here the basis disadvantages is only a function can be controlled by a mouse at the same time.

2.3 Advantages and Difficulties of Vehicle Dynamics Simulation

The main difficulties of modelling vehicle dynamics for a full vehicle are:

1- Vehicle characteristics are formed by many nonlinear impacts. Tires and elastic bushings in the suspensions are the most important ones among the others.

2- The expectations of vehicle behaviours are much variety and generally incompatible. For instance, typical incompatibles occur between comfort and safety for designing of many subsystems. One of the reasons to cause critical uncertainties result from the driver because of being a human and the other reason result from environment where the vehicle is moving in.

3- Vehicles are becoming more complicated and vehicle behaviours are dominated nowadays by equipments of electronic, electric, hydraulic and pneumatic systems.

4- Many parameters belong to vehicle are exposed to critical variations while driving (The state of loading, tire and damper etc.). And this situation blocks to reveal the behaviour of real system at simulations.

All under these conditions, vehicle behaviours naturally have to be kept independent away from little variations at vehicle parameters. However, it will not to be easy to test this process. Knowing how the vehicle behaves in an operation area is not enough to know how it behaves in case of little variation of operation area because of the non-linear properties. In this case, the only solution is to model whole subsystems including all parameters and bring them together to solve the problem. Vehicle manufacturers need to set up a test procedure which is extremely wide and expensive to evaluate the behaviour of a vehicle in whole operation area and vehicles need to be tested for many times. The only solution to avoid this method is to test truly by increasing test points and set up the connection with hyperlinks called vehicle dynamics simulation programme between points.

Creating computer aided driver-vehicle-environment closed control loop virtually is rather complicated. There are some problems by transferring 3 elements of the loop to virtual environment and these are given below:

1- Vehicles, which include mechanics, hydraulics, and pneumatics, electric and electronic systems, are complex technique systems and tough to model.

2- Community of drivers are formed by billions of controllers that show many different characteristics. Furthermore; their detection, deciding and implementation mechanisms are not cleared up in consequence of characters being human.

3- Geometry, dry and wet friction features of ground, vision characteristics, interaction to other people at the traffic and road roughness are some parts of real world. And however, this can be modelled for limited state selected.

Using of modelling and simulation techniques at vehicle development operations are shorten the time of vehicle development programme, also reduce numbers and total working periods of workers (engineers and others). Thus, engineers can be nominated to other department. Numbers of vehicle prototypes which are expensive could be reduced in this way.

All hardware, software and manpower which are necessary for simulation cost a lot of money to automotive industry. And basis factors of these costs are:

1- Hardware: Computers consist of main computers (as servers etc.) and network configuring systems or hardware. The cost of hardware is approximately fixed for the computers used at development department of a selected company (Atabay 2004). Whereas the performance of hardware is increased exponentially according as same reference.

2- Software: Expectations of software are increased by increasing capabilities of hardware. Manpower that's necessary shows an advance for maintenance and accordance of all system, too.

3- Private systems as hardware and software need to be developed for requirements of automotive industry. This especially costs too expensive to automotive industry. The reason is to be needed of experts due to complexity of problems. Verifying simulations required very high-priced tests. Automotive companies prefer to set up associated working groups intercompany for this type of developments

4- Money and time are notable which is used to educate engineers, technicians and experts that will be capable to use the systems in the companies. Time spent on computers must be calculated to prepare run and evaluate simulations and this time needs to be taken into consideration.

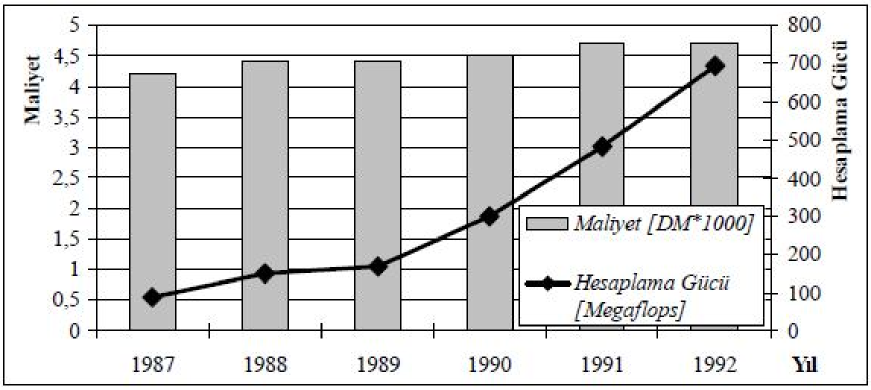

Figure 2.1 : The cost of computers and computational power at Audi Company according as years (Atabay 2004)

2.4 Vehicle Dynamics Simulation Levels

There are three methods at mathematical explanations of a mechanical system and technical dynamics systems.

1- Method of multibody dynamics
2- Finite elements method
3- Continuous system method (continuum mechanics)

Here is a general and explicit phase to implement the simulation:

1- Definition of problem: The point should be considered here is to see the effort and cost that will occur. Moreover, this work needs to be done even though it won't end by clear results.

2- Simulation implementation: This phase is mainly belongs to theorists.

3- Presenting results to constructors as suggestions: constructors are not ready to hear results coming from the simulation. Besides, they don't tend to apply these results. However, good senses of constructors can be seen to give better results occasionally.

2.5 Points Should Be Noticed At Vehicle Dynamics Simulation

Some points which need to be noticed about vehicle dynamics simulation are given below. One part of points consists of general phrases and other part of it consists of special cases.

- Many scientific or computational problem occur while using and solving the equations and programming them directly at simulation.

- If functional value of variable is frequently needed while programming (for instance tangent of some variable) this can be calculated at the start of integration step and kept at another variable. Thus, same variable can be used in many ways and a significant saving is provided of computation time.

- Speed difference of virtual and real vehicle need to be less than 1 km/h to be expected the same result of them according as Atabay (2004). And this is extremely tough to actualize at model verification works.

- The simulation of vehicle subsystems stand alone is not enough according as many academics. Basic tests with some boundary conditions are not enough because all subsystems are in interaction. For that reason all systems must be simulated as a whole.

- Model verification can’t be investigated completely without being sure that all simulation sub-models work properly. This situation can be observed, especially at tire models, the simulation can supply critical uncorrect states in the handling simulation. Each simulation model generates exact results if each sub-model of them is correct.

- It’s represented that the integration, which is centred or not, can be used for differential equations belongs to state variables in Atabay 2004. In the centred integration, instant derivatives belong to all state variables are found at single stage and integration is done in the following step for all state variables. In the integration that’s not centred,   derivative of each variable is found then integration is done and the next variable is passed after this process.

4 points needs to be noticed at solving differential equations in computer aided simulations according as Atabay 2004 by the selection of integration algorithm it must be considered.

1- Step controlled methods are not appropriate.
2- Implicit methods are not appropriate.
3- Multi step methods are not appropriate.
4- Low order methods are possibly appropriate.

Classified points are added if this application is an HIL application or it’s planned to be an HIL application in the future.

1- There are partly input and output signals in the system. In this case, input parameters are not identified contrary to the type “offline simulations”. Integration methods, which need initial parameters of next time steps, can’t be used.

2- Phase shifts between input and output signals must be as little as possible at HIL simulation works as a closed loop.

3- Integration steps have the longest period are chosen to have the best stability and have error as little as possible at offline simulations. Time parameters of the system are not considered at HIL applications. For instance, if the system is a control unit, the frequency of output parameters must be higher than alert frequency of control unit (Atabay 2004).

The soul of digital simulation is constituted by digital integration. Automatic step controlled integration algorithms became so popular at the type of off-line simulations. However, computation of variable step are not used in a programme which is a type of HIL (Hardware-in-the-Loop) simulation. How long the simulation will take must be known by the agency of constant time steps

Different integration steps for vehicle sub systems can be chosen that are expressed and examples about it are given in Atabay 2004. For instance, Atabay chose a 10ms integration step for body, 5ms for axles and 1ms for wheels in the investigation. Fast output signals of subsystems are applied to input of subsystem by taking the average of output signals. Fast inputs of subsystem can be found by the extrapolation of slow outputs of subsystem. Atabay 2004 used Euler integration method at real-time handling, vehicle dynamics simulation. Model equations consist of some approaches at vehicle dynamics simulation and they are used to obtain an opinion with errors as little as possible about behaviours of real vehicle. Choosing an integration method should be suggested according as the degree of approximation in the model (Atabay 1994).

The dynamics of state variables show belongs to vehicle resort huge variability. Body can be moved around a few Hz and the wheel can be blocked in a few milliseconds. If elasticity of powertrains and steering system are considered, this comes with high-frequency movements. There are a few characters consisting of impact or sudden variance in many events like clutching, braking and differential lock. Vehicle system must be seen as a hard dynamics system due to all this points. Hard differential equation systems can be solved by implicit methods. Nevertheless, the most usable one of them is the Euler method.

2.6 Surtak Vehicle Dynamıcs Sımulatıon Software

SURTAK Driving Simulator was programmed and constructed at the Istanbul Technical University Automotive Laboratory. In this thesis the main vehicle dynamics simulation mainframe of SURTAK has been used and integrated to an electric road vehicle structure. The programming language was transferred from C++ to MATLAB environment to achieve user friendly programming and profit from the visiualization properties of MATLAB.

Surtak Driving Simulator

Figure 2.2 : Screen Image of SURTAK Phase I

Figure 2.3 : Real Road Image

SURTAK Driving Simulator, which was designed and manufactured to control vehicle parameters and human factors, mainly consists of many types of software and a few hardware parts. SURTAK driving simulator has a lot of advantages. Furthermore, SURTAK driving simulator provides investigations in all areas about vehicle dynamics and human factors. The system programmed in C++® was created onto a vehicle model which has 11 degree of freedom and was verified for specific maneuvers. The simulator uses for image, sound and driver interface data processing the DirectX® SDK Libraries of Microsoft®. To achieve a high quality and real-like image objects like trees, traffic signs and road textures can be also added to the vision. The widely used CAD software AutoCAD® is used for the generation of tracks and the virtual environment structure. Virtual tracks are designed which agree with road design engineering aspects. Real tracks are also imported into the system such as “Istanbul Park” the new Formula 1 race track in Turkey. Engine noise is transmitted to the driver as a function of engine speed and throttle position.

2.6.1 Vision, Sound and Driver Interface Data Processing

A screen image of the earlier versions of SURTAK can be seen in Figure 2.2 and 2.3 which inspired from a real photo. The driver sees the simulated scenario due to a video projector at a resolution of 1024x768. A commercial 5+1 PC speaker system is fed with an engine noise signal proportional to engine speed and throttle position. The engine noise signal for the simulator is generated due to modification of the playing frequency of a real engine noise wave file.

Driver inputs for the virtual vehicle get into PC through well-known commercial man-machine USB interfaces. Some screen captures from the latest image generation algorithms including the Istanbul Park race track and a Formula 1 race car are in the Figure 2.4 and 2.5. A virtual proving ground is under construction which will be similar to real ones.

Figure 2.4 : Screen Image of SURTAK Phase II

Figure 2.5 : Screen Image of SURTAK Phase II

A simple simulator platform is designed and constructed as a first step to carry all the hardware elements of the simulator. Next step should be a real car cabin (also fixed based) which would probably encourage a developed version of the simulator software and attached to real driver interface elements like steering wheel and pedals of a real vehicle. A view of the current platform is in Figure 2.6. A flow chart of the whole software of SURTAK can be seen in Figure 2.7.

Figure 2.6 : SURTAK (1.0) Fixed-Based Driving Simulator Platform

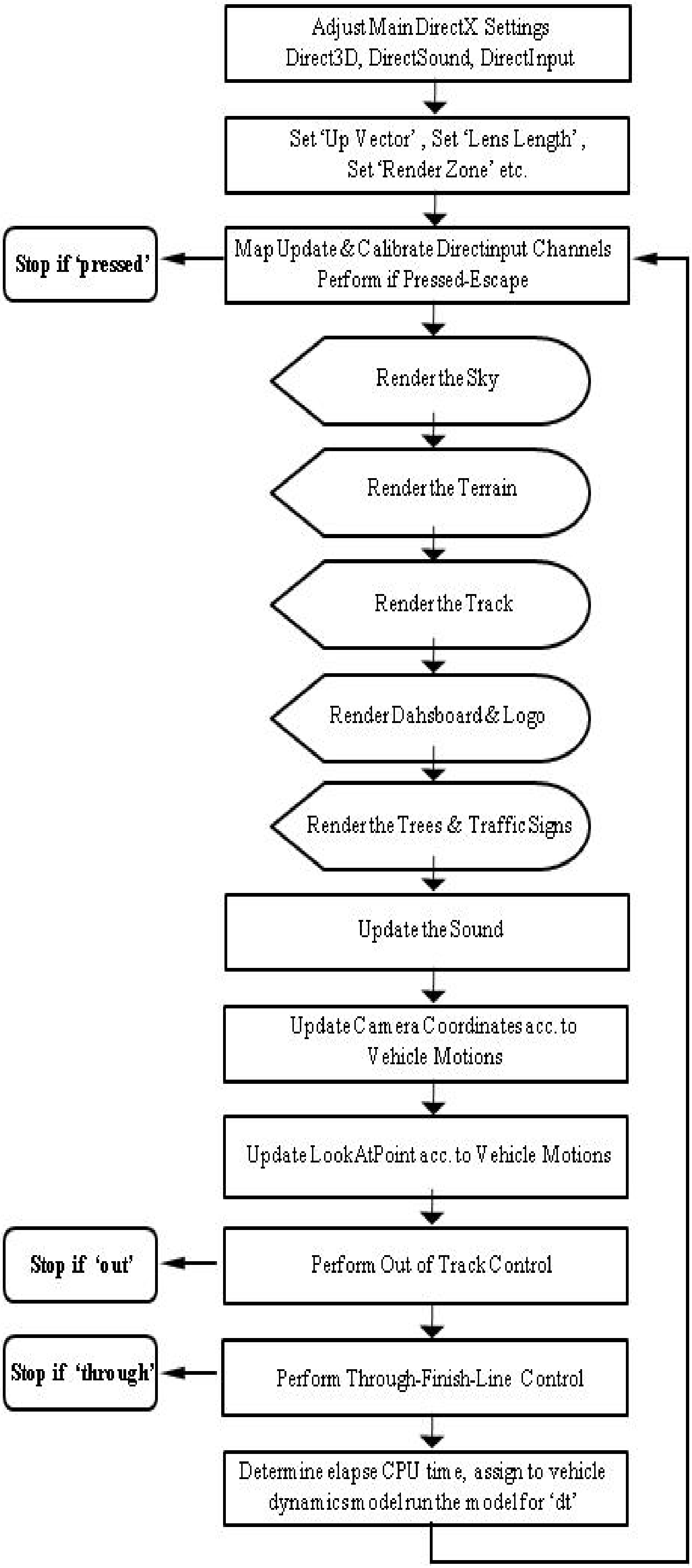

Figure 2.7 : Flow Chart of SurTak Phase II

2.6.2 Vehicle Dynamics Model

Vehicle Dynamics model, where HSRI tire model was used, utilizes equations includes 11 degree of freedom and maps for many subsystems (body of the model has 5 degrees of freedom). In this thesis the use tire model is the Pacejka tire model and not the HSRI tire model like in Surtak. This is also an enhancement of the simulator software (See Section 4).

These are:

- Internal combustion engine torque map,
- Internal combustion engine fuel consumption map,
- Trilok convertor pump rotor coefficient curve,
- Trilok convertor torque conversion curve,
- Automatic transmission planet group efficiency map,
- Automatic transmission gear shifting map. (Computation of gear shifting contains tranmission ratio, rotating mass factors and time lag of gear shifting, too.

The other subsystems of SURTAK are:

- Suspension System
- Steering System
- Brake System
- Road Load System
- And The Body.

All kinds of computation should be able to think so widely due to wide operating range which contain handling situations that a driver can constitute in a driving simulator. That’s the tough side of this kind of investigation. The scenarios of driving simulators are full transient. Stationary cases are almost never the point in this case. Here concrete computation is utilized for extremely minimal computation steps by the help of the actual speed computers and so that each case can be thought as if it’s static.

3. ELECTRIC VEHICLES

3.1 History of Electric Vehicles

The first electric vehicle was built by Frenchman Gustave Trouvé in 1881. It was a tricycle powered by a 0.1 hp DC motor fed by lead–acid batteries. The whole vehicle and its driver weighed approximately 160 kg. A vehicle similar to this was built in 1883 by two British professors. These early realizations did not attract much attention from the public because the technology was not mature enough to compete with horse carriages. Speeds of 15 km/h and a range of 16 km were not exciting for potential customers. The 1864 Paris to Rouen race changed it all: the 1135 km were run in 48 h and 53 min at an average speed of 23.3 km/h. This speed was by far superior to that possible with horse-drawn carriages. The general public became interested in horseless carriages or automobiles, as these vehicles were now called. [Modern Electric, Hybrid Electric, Fuel Cell Vehicles]

The following 20 years were an era during which electric vehicles competed with their gasoline counterparts. This was particularly true in America, where there were not many paved roads outside a few cities. The limited range of electric vehicles was not a problem. However, in Europe, the rapidly increasing number of paved roads called for extended ranges, thus favouring gasoline vehicles. The first commercial electric vehicle was Morris and Salom's Electroboat. This vehicle was operated as a taxi in New York City by a company created by its inventors. The Electroboat proved to be more profitable than horse cabs despite a higher purchase price (around $3000 vs. $1200). It could be used for three shifts of 4 h with 90-min recharging periods in between. It was powered by two 1.5 hp motors that allowed a maximum speed of 32 km/h and a 40-km range.

The most significant technical advance of that era was the invention of regenerative braking by Frenchman M.A. Darracq on his 1897 coupe. This method allows recuperating the vehicle's kinetic energy while braking and recharging the batteries, which greatly enhances the driving range. It is one of the most significant contributions to electric and hybrid electric vehicle technology as it contributes to energy efficiency more than anything else in urban driving. In addition, among the most significant electric vehicles of that era was the first vehicle ever to reach 100 km/h. It was “La Jamais Contente” built by Frenchman Camille Jenatzy. Note that Studebaker and Oldsmobile first started in business by building electric vehicles.

As gasoline automobiles became more powerful, more flexible, and, above all, easier to handle, electric vehicles started to disappear. Their high cost did not help, but it is their limited driving range and performance that really impaired them vs. their gasoline counterparts. The last commercially significant electric vehicles were released around 1905. During nearly 60 years, the only electric vehicles sold were common golf carts and delivery vehicles.

In 1945, three researchers at Bell Laboratories invented a device that was meant to revolutionize the world of electronics and electricity: the transistor. It quickly replaced vacuum tubes for signal electronics and soon the thyristor was invented, which allowed switching high currents at high voltages. This made it possible to regulate the power fed to an electric motor without the very inefficient rheostats, and allowed the running of AC motors at variable frequency. In 1966, General Motors (GM) built the Electro van, which was propelled by induction motors that were fed by inverters built with thyristors.

The most significant electric vehicle of that era was the Lunar Roving Vehicle, which the Apollo astronauts used on the Moon. The vehicle itself weighed 209 kg and could carry a payload of 490 kg. The range was around 65 km. The design of this extraterrestrial vehicle, however, has very little significance down on Earth. The absence of air and the lower gravity on the Moon, and the low speed made it easier for engineers to reach an extended range with limited technology. During the 1960s and 1970s, concerns about the environment triggered some research on electric vehicles. However, despite advances in battery technology and power electronics, their range and performance were still obstacles.

The Modern Age of EV

The modern electric vehicle era culminated during the 1980s and early 1990s with the release of a few realistic vehicles by firms such as GM with the EV1 and PSA with the 106 Electric. Although these vehicles represented a real achievement, especially when compared with early realizations, it became clear during the early 1990s that electric automobiles could never compete with gasoline automobiles for range and performance. The reason is that in batteries the energy is stored in the metal of electrodes, which weigh far more than gasoline for the same energy content. The automotive industry abandoned the electric vehicle to conduct research on hybrid electric vehicles. After a few years of development, these are far closer to the assembly line for mass production than electric vehicles have ever been.

In the context of the development of the electric vehicle, it is battery technology that is the weakest, blocking the way of electric vehicles to market. Great effort and investment have been put into battery research, with the intention of improving performance to meet the electric vehicle's requirement. Unfortunately, progress has been very limited. Performance is far behind the requirement, especially energy storage capacity per unit weight and volume. This poor energy storage capability of batteries limits electric vehicles only to some specific applications, such as at airports and railroad stations, on mail delivery routes, and on golf courses, etc. In fact, basic study shows that electric vehicles will never be able to challenge liquid fuelled vehicles even with the optimistic value of battery energy capacity. [Modern Electric, Hybrid Electric, and Fuel Cell Vehicles]

61 million vehicles were totally manufactured in the world in 2009. In the same year, 870 thousand vehicles were manufactured in the automotive industry of Turkey. Turkey is 16th at automotive manufacturing in the world in 2009. Furthermore, Turkey is the first at light commercial vehicle manufacturing, the second at bus manufacturing and the 9th at heavy truck manufacturing. And generally, Turkey is at the 7th position of total manufacturing in Europe.

Utilization of many alternative drive systems are come into question to reduce greenhouse gas emissions arising from fossil fuel used in vehicles has internal combustion engine. Electric vehicles are one of the most significant candidates to provide CO_2emissions rates projected in 2012 and after the year 2012. Electric vehicles, which have only batteries as energy source, are convenient for the utilization of city driving because of limited ranges they have like 150 – 200 km.

An affiliation is come up between car park and electric energy infrastructure systems to charge electric vehicles by connecting to the electric energy grid directly. Electric distribution grid is an inseparable part of automotive technology owing to this affiliation. Whole electric vehicle system source to wheels needs to be optimized by calculating all energy conversion cycle orientated to reduce CO_2 emissions.

Electric vehicle subsystems and other parts of it need to be recuperated to get some additional technologic gains. The main of these additional technologic gains are like battery, ultra condenser, electric energy storage system, electric machines and drivers for electrically driving, electronic control systems like vehicle and battery control systems, power electronic systems, electrically driving concepts and electrically auxiliary driving system. Another important point is the lack of technical regulation on this issue.

Motor vehicles regulation in Turkey has to be reviewed on the issue of electric vehicles. Especially additional security terms that arising from the use of high voltage must be updated by considering of European Commission.

3.2 Selected Configuration of Electric Drive The Basis Electric Vehicle Structure and Technologies

Electric vehicles have different driving layouts. These are the systems which are driven by electric energy and the mixed systems which electric energy and internal combustion engines are used together.

Electric vehicles are one of the most significant candidates to provide CO_2emissions rates projected in 2012 and after the year 2012. Electric vehicles use an electric motor to drive and battery to store energy. Energy at batteries provides all moving power and energy requirements on vehicle. Batteries are charged by mains electricity. The advantages of electric vehicles with respect to conventional vehicles have internal combustion engines are high efficiency, very low rumbling noise levels and zero emission sights. The values of battery energy density are lower than liquids fuels and so today's battery technology is the most important disadvantages of it. The formation of electric vehicle concepts is not the advantages only for four wheels vehicles. This is an advantage for two and three wheels vehicles as well.

The drive train for an electric vehicle features less energy conversion steps than a hybrid drive train and contains drive train components which all work at high efficiency, the drive train for an electric vehicle will not be introduced, only non-differential system with battery and two electric motor will be handled in this thesis.

The Electric Vehicle (EV)

Generally, an Electric Vehicle uses, which consisting of one or multiple electric motors to take care of propulsion and a battery as energy source. The transport of electric energy is handled by the power electronics. As an EV does not have the ability to "refill" the energy supply in a quick way, the amount of energy that is carried on board must be enough to keep the vehicle driving for the rest of the day so the batteries may be recharged at home during the night. The possibility to charge an EV's battery by means of connecting it to the electricity grid is called "plug-in". In Figure 3.1 the topology layout is presented.

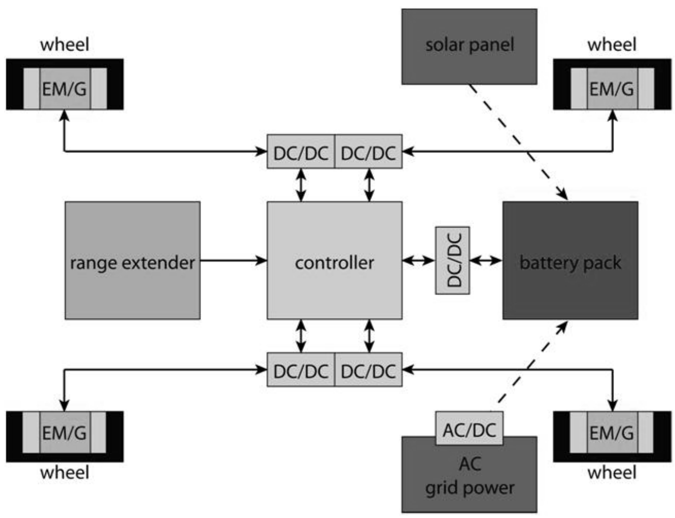

Figure 3.1 : Illustration of EV Topology

A Special Configuration : The Non-Differential System uses Battery and Two Electric Motor

The system has two wheels drive (Rear or Front Wheels). Those wheels are driven by two free electric motors. Using of electronic control unit provides optimum traction at different road conditions. The system in this thesis does not utilize mechanic drive train. Therefore this increase the efficiency and reliability, and besides decrease maintenance demands. The system is as demonstrated by Figure 3.2.

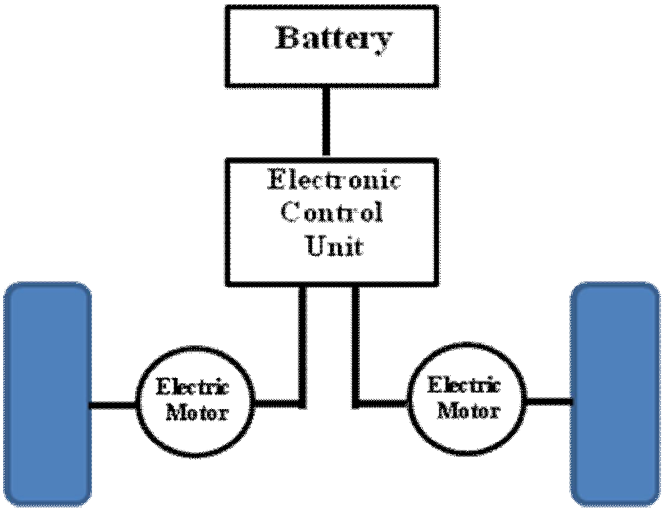

Figure 3.2 : The Non-Differential System use Battery and Two Electric Motors

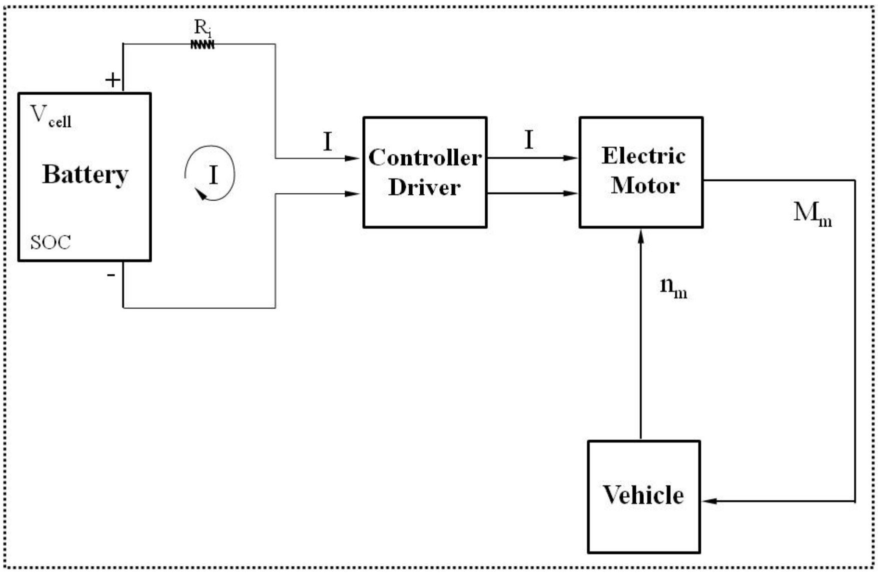

Figure 3.3 : The Non-Differential EV System Model

3.3 Electric Vehicle Modelling

With all vehicles the prediction of performance and range is important. Computers allow us to do this reasonably easily. Above all, computer based methods allow us to quickly experiment with aspects of the vehicle, such as motor power, battery type and size, weight and so on, and see how the changes affect the performance and range. Here how the equations have been developed will be shown to perform quite accurate and useful simulations. Furthermore, it will be shown that how this can be done without using any special knowledge of programming techniques, as standard mathematics and spreadsheet programs such as MATLAB and EXCEL make an excellent basis for these simulations. And, there are some features of electric vehicles that make the mathematical modelling of performance easier than for other vehicles will be designated.

The first parameter we will model is vehicle *performance*. By performance we mean acceleration and top speed, an area where electric vehicles have a reputation of being very poor. It is necessary that any electric vehicle has a performance that allows it, at the very least, to blend safely with ordinary city traffic. Many would argue that the performance should be at least as good as current IC engine vehicles if large scale sales are to be achieved.

Another vitally important feature of electric vehicles that we must be able to predict is their *range*. This can also be mathematically modelled, and computer programs make this quite straightforward. The mathematics we will develop will allow us to see the effects of changing things like battery type and capacity, as well as all other aspects of vehicle design, on range. This is an essential tool for the vehicle designer.

Going on to show how the data produced by the simulations can also have other uses in addition to predicting performance and range. For example; how data about the motor torque and speed can be used to optimise the compromises involved in the design of the motor and other subsystems.

3.3.1 Tractive Effort

The first step in vehicle performance modelling is to produce an equation for the tractive effort. This is the force propelling the vehicle forward, transmitted to the ground through the drive wheels.

Consider a vehicle of mass m, proceeding at a velocity v, up a slope of angle α , as in Figure 3.3. The force propelling the vehicle forward, the tractive effort, has to accomplish the following:

- overcome the rolling resistance;
- overcome the aerodynamic drag;
- provide the force needed to overcome the component of the vehicle's weight acting down the slope;
- accelerate the vehicle, if the velocity is not constant.
- each of these will be considered in turn.

3.3.1.1 Rolling Resistance Force

The rolling resistance is primarily due to the friction of the vehicle tyre on the road. Friction in bearings and the gearing system also play their part. The rolling resistance is approximately constant, and hardly depends on vehicle speed. It is proportional to vehicle weight. The equation is:

$$F_R = f_R F_Z \tag{3.1}$$

where f_R is the coefficient of rolling resistance. The main factors controlling f_R are the type of tyre and the tyre pressure. Any cyclist will know this very well; the free-wheeling performance of a bicycle becomes much better if the tyres are pumped up to a high pressure, though the ride may be less comfortable.

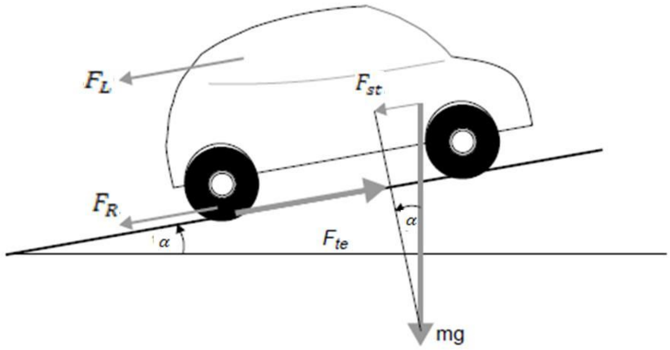

Figure 3.4 : The forces acting on a vehicle moving along a slope [EV Technology Explained, 2003]

The value of f_R can reasonably readily be found by pulling a vehicle at a steady very low speed, and measuring the force required. Typical values of f_R are 0,015 for a radial ply tyre, down to about 0,005 for tyres developed especially for electric vehicles.

3.3.1.2 Aerodynamic Drag

This part of the force is due to the friction of the vehicle body moving through the air. It is a function of the frontal area, shape, protrusions such as side mirrors, ducts and air passages, spoilers, and many other factors. The formula for this component is:

$$F_L = \frac{1}{2}\rho A C_d v^2 \qquad \textbf{(3.2)}$$

where ρ is the density of the air, A is the frontal area, and v is the velocity. C_d is a constant called the drag coefficient. The drag coefficient C_d can be reduced by good vehicle design. A typical value for a saloon car is 0.3, but some electric vehicle designs have achieved values as low as 0.19. There is greater opportunity for reducing C_d in electric vehicle design because there is more flexibility in the location of the major components, and there is less need for cooling air ducting and under-vehicle pipe work. However, some vehicles, such as motorcycles and buses will inevitably have much larger values, and C_d figures of around 0.7 are more typical in such cases. The density of air does of course vary with temperature, altitude and humidity. However a value of $1.25\ kg.m^{-3}$ is a reasonable value to use in most cases. Provided that SI units are used (m^2 for A, $m.s^{-1}$ for v) then the value of F_L will be given in Newtons.

3.3.1.3 Hill Climbing Force

The force needed to drive the vehicle up a slope is the most straightforward to find. It is simply the component of the vehicle weight that acts along the slope. By simple resolution of forces we see that:

$$F_{st} = mg \sin \alpha \tag{3.3}$$

3.3.1.4 Acceleration Force

If the velocity of the vehicle is changing, then clearly a force will need to be applied in addition to the forces shown in Figure 3.4. This force will provide the linear acceleration of the vehicle, and is given by the well-known equation derived from Newton’s second law,

$$F_B = \lambda m a \tag{3.4}$$

However, for a more accurate picture of the force needed to accelerate the vehicle we should also consider the force needed to make the rotating parts turn faster. In other words, we need to consider rotational acceleration as well as linear acceleration. The main issue here is the electric motor, not necessarily because of its particularly high moment of inertia, but because of its higher angular speeds.

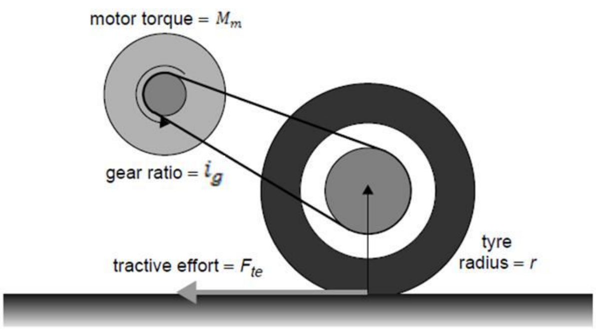

Figure 3.5 : A Simple Arrangement for Connecting a Motor to a Drive Wheel [Electric Vehicle Technology Explained]

3.3.1.5 Total Tractive Effort

The total tractive effort is the sum of all these forces:

$$F_{te} = F_R + F_L + F_{st} + F_B \qquad \textbf{(3.5)}$$

where:

- F_R is the rolling resistance force, given by equation **(3.1);**
- F_L is the aerodynamic drag, given by equation **(3.2)**;
- F_{st} is the hill climbing force, given by equation **(3.3)**;
- F_B is the force required to give linear acceleration given by equation **(3.4)**;

We should note that F_B will be negative if the vehicle is slowing down, and that F_{st} will be negative if it is going downhill.

3.3.2 Electric Motor Model

The vehicle will be propelled by two independent in-wheel electric motors. The torque and power combined for these motors should be enough to sufficiently accelerate and to reach a top speed. Motors are chosen with a maximum torque and a maximum power for these reasons. The benefit of these in-wheel motors is that maximum torque is instantly accessible from standstill, which results in a good acceleration.

An in-wheel motor is more efficient than an ICE and a central electric motor. This is because the in-wheel motor is directly driven. No losses in the gearbox and differential occur. These losses are caused by the design of a gearbox, because it is built to withstand the maximum power of an engine at maximum speeds. All bearings, cogwheels and axes are bigger and heavier than would be necessary to run at nominal load only. Therefore, at nominal load it takes large amounts of energy just to move the gearbox.

The use of in-wheel motors is also beneficial for the design of the exterior, because the space that is normally occupied by axles, the transmission and the engine can now be used in any way a designer wishes.

In figure 3.5 the motor characteristic of the in-wheel motors is shown. Also the road load is shown in this figure to identify the torque reserve and theoretical top speed of 190 km/h.

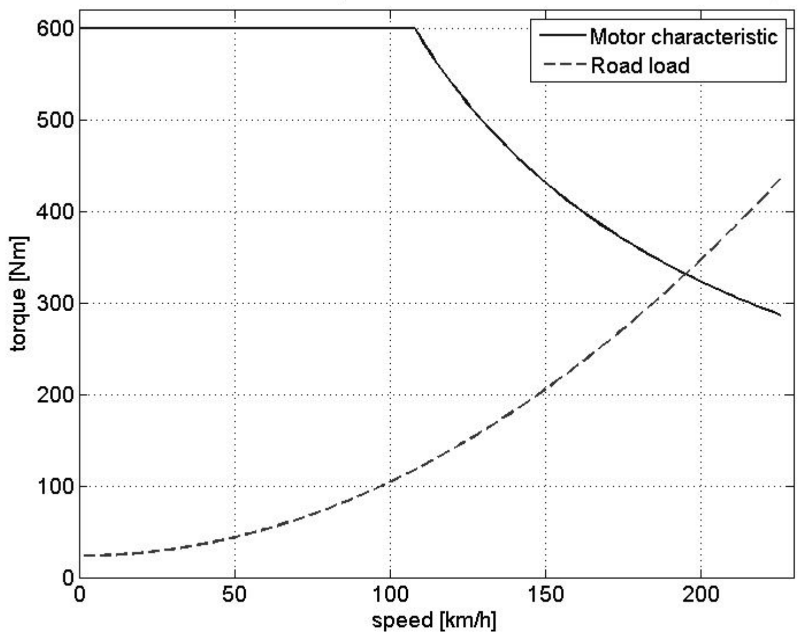

Figure 3.6 : The Non-Differential System use Battery and Two Electric Motor

Performance of Electric Vehicles

A vehicle's driving performance is usually evaluated by its *acceleration time, maximum speed, and grade ability*. In EV drive train design, proper motor power rating and transmission parameters are the primary considerations to meet the performance specification. The design of all these parameters depends mostly on the speed–power (torque) characteristics of the motor.

Traction Motor Characteristics

Variable-speed electric motor drives usually have the characteristics shown in Figure 3.6. At the low-speed region (less than the base speed as marked in Figure 3.6), the motor has a constant torque. In the high-speed region (higher than the base speed), the motor has a constant power. In low-speed operations, voltage supply to the motor increases with the increase of the speed through the electronic converter while the flux is kept constant. At the point of base speed, the voltage of the motor reaches the source voltage. After the base speed, the motor voltage is kept constant and the flux is weakened, dropping hyperbolically with increasing speed. Hence, its torque also drops hyperbolically with increasing speed. It is clear that with a long constant power region, the maximum torque of the motor can be significantly increased, and hence vehicle acceleration and grade ability performance can be improved and the transmission can be simplified.

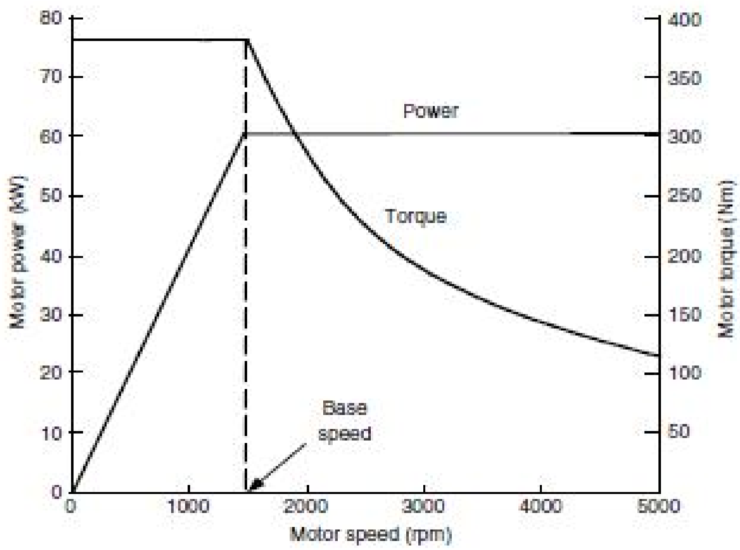

Figure 3.7 : Typical variable-speed electric motor characteristics [Electric Vehicle Technology Explained]

Efficiency

Efficiency is the proprotion of mechanic power required by electrik motor to the power taken out of the battery of electric motor.

$$\eta_{EV} = \frac{M_m * \omega_m}{V_{cell} * I} \quad \textbf{(3.6)}$$

3.3.3 Battery Model

Battery in electric vehicles is one of the most significant components to designate condition and characteristic of the vehicle. Some of specifications are required from batteries which are going to be used in electric vehicles. And these are:

- High energy density, low weight,
- Long service life,
- Reliability,
- Ease of recycling,
- High charge capacity,
- Low heat generation,
- Fulfilling performance demands between temperatures of -20 and +50,
- Low self-discharge,
- Ease of maintenance and preparation.

A polymer lithium-ion battery (PLIB) is an example of a developing battery technology, which is chosen as a energy source of EV in the thesis. It is of the lithium-ion type and features higher energy density, lower weight and lower costs than other battery technologies such as NiMH or lead-acid batteries. The latest addition to the PLIB family is the LiFePO4 type. The advantage of this type is that it is cheaper, safer when used in large battery packs, cycle has a longer life and can be charged in less time than the widely used LiCoO2 type. The LiFePO4 PLIB has a remarkable cycle efficiency of 99.8%. All the above mentioned advantages of this LiFePO4 PLIB make this the current "best choice". The chosen battery nominal capasity is 250 Amph, internal resistance is 320 miliohm and the battery voltage is 120 volt. The battery used in the thesis is 30 kWh battery.

A good battery management are necessary in electric vehicles owing to difficulty of battery recycling, expensiveness of materials used in battery manufacturing and shortness of battery service life. Battery equations used are obtained by using equivalent circuit diagram in Figure 3.7.

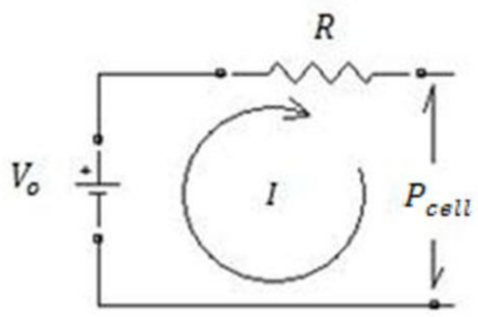

Figure 3.8 : Equivalent Circuit Diagram [Electric Vehicles]

Battery internal resistance R is obtained by tables as a function of battery temperature and battery state of charge (SOC) in the model. In the case of discharge the magnitude of current draw I is computed by battery internal resistance and open circuit voltage of the battery V_o.

Accordingly, P_s and P_{cell} are in turn battery internal power and battery terminal power.

$$P_s(t) = V_o\big(SOC(t)\big)I(t) \qquad \textbf{(3.7)}$$

$$P_{cell}(t) = V_o\big(SOC(t)\big)I(t) - I(t)^2R\big(SOC(t), sign(Pcell)\big) \qquad \textbf{(3.8)}$$

3.3.3.1 Battery Modelling

State of charge of the battery(SOC) is the ratio of remaining charge $Q(t)$ to battery nominal capacity Q_0. The state of charge changes between the values of 0 and 1. And, differential equations used in state of charge computation are;

$$SOC(t) = \frac{Q\,(t)}{Q_0} \qquad \textbf{(3.9)}$$

$$Q(t) = I(t) \qquad \textbf{(3.10)}$$

$$\frac{dSOC}{dt} = \frac{I(t)}{Q_0} \quad \Rightarrow \quad SOC = \int \frac{I(t)}{Q_0}dt \qquad \textbf{(3.11)}$$

In the case of charge and discharge, battery power and battery current will change signal and internal resistance will be different from initial case. Furthermore, discharge current will be negative for all time.

Specifically, voltage and internal resistance at any time are a function of the state of charge, while the total cell output is a function of the voltage, internal resistance, and current. The cells in the battery are connected in series. Voltages are in volts and resistances in milliohms.

$$V_0(SOC) = 108.51 + 21.645 * SOC - 10.159 * SOC^2 \tag{3.12}$$

$$R(SOC) = 382.34 - 234.16 * SOC + 195.75 * SOC^2 \ if\ SOC < 0.4 \tag{3.13}$$

$$R(SOC) = 1.75 \ if\ SOC \geq 0.4 \tag{3.14}$$

$$V_{cell} = V_0(SOC) - R(SOC)I(t) \quad \text{(for discharge)} \tag{3.15}$$

The power for discharge case of the battery is;

$$P_{dischg} = V_o I(t) - R(SOC)I(t)^2 \quad \text{(for discharge)} \tag{3.16}$$

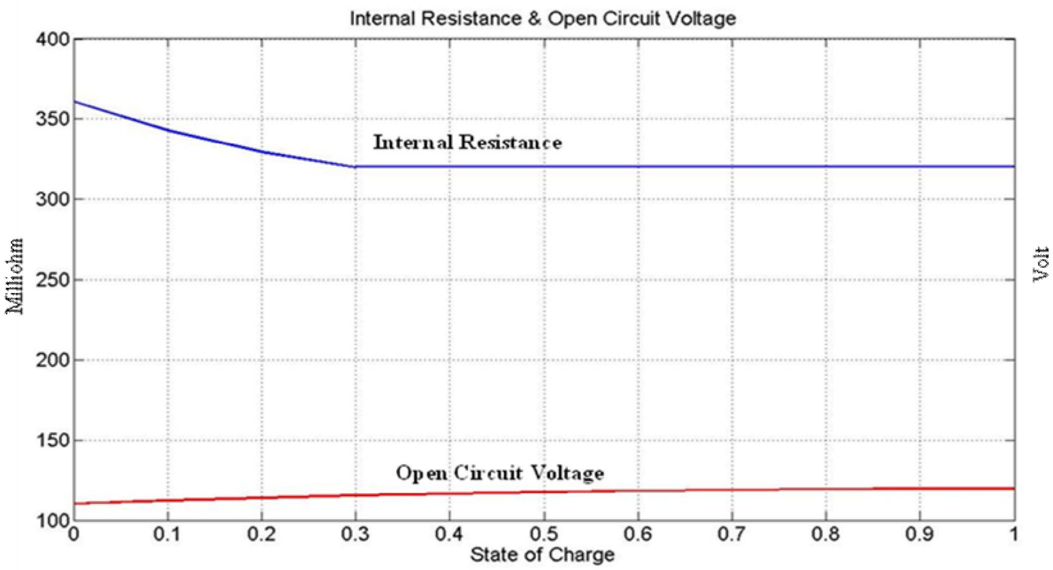

Figure 3.9 : Voltage and internal resistance of Lithium-ion battery

So the voltage output is a function of the state of charge, as is the internal resistance, as illustrated graphically:

The power at battery terminal and internal battery power are various due to the internal resistance of the battery, so the efficiency of battery can be computed by the equation given below. The efficiency is a function of state of charge and internal resistance.

$$\eta_{cell}(t) = \frac{P_s(t)}{P_{cell}(t)} = \frac{V_o I(t)}{V_o I(t) - R(SOC) I(t)^2} \quad \textbf{(3.17)}$$

State of charge *SOC* and internal resistance of the battery change according to battery temperature. Battery internal resistance is mostly higher at lower temperature. Air conditions that battery works have to be taken in account therefore. Variation of the battery internal resistance according to battery temperature is ruled out in this thesis.

All the systems associated with electric vehicles need to be worked where they are the most efficient in areas. Therefore, algorithms can be generated to work them in the most efficient areas, if requested efficiencies of battery and electric motor obtains. Service life of batteries is not at requested levels. So; battery parameters kind of temperature, charge time and current depth of battery are managed very well to keep battery life longer.

3.3.4 Modelling Electric Vehicle Range

3.3.4.1 Driving Cycles

It is well known that the range of electric vehicles is a major problem. In the main this is because it is so hard to efficiently store electrical energy. In any case, this problem is certainly a critical issue in the design of any electric vehicle. There are two types of calculation or test that can be performed with regard to the range of a vehicle.

The first, and much the simplest, is the constant velocity test. Of course no vehicle is really driven at constant velocity, especially not on level ground, and in still air, which are almost universal further simplifications for these tests. However, at least

the rules for the test are clear and unambiguous, even if the test is unrealistic. It can be argued that they do at least give useful comparative figures.
The second type of test, more useful and complex, is where the vehicle is driven, in reality or in simulation, through a profile of ever changing speeds. These test cycles have been developed with some care, and there are (unfortunately) a large number of them.

The cycles are intended to correspond to realistic driving patterns in different conditions. During these tests the vehicle speed is almost constantly changing, and thus the performance of all the other parts of the system is also highly variable, which makes the computations more complex. However, modern computer programs make even these more complex situations reasonably straightforward. These driving cycles (or schedules) have primarily been developed in order to provide a realistic and practical test for the emissions of vehicles.

In the European scene, the cycles tend to be rather simpler, with periods of constant acceleration and constant velocity. Of particular note is the ECE-15 drive cycle, which is useful for testing the performance of small vehicles such as battery electric cars. In EC emission tests this has to be combined the extra-urban driving cycles (EUDC), which has a maximum speed of 120 $km.h^{-1}$.

New European Driving Cycle

European Driving Cycles belong to the modal driving type, which simply consists of constant speed and constant accelerations, not like transient driving cycles such as that in the United States. This is modelled by FTP (Federal Test Procedure) which has variations in speed and acceleration on a second by second basis. The European driving cycles were prepared by the Economic Commission for Europe (ECE) and the European Economic Community. The early European driving cycles were a simple representation of the driving pattern in the city only and were composed of fifteen driving conditions, and thus were called the ECE15. Since 1992, the Extra Urban Driving Cycle (EUDC) was added on to the previous cycle to represent rural driving conditions. The two cycles were then combined (ECE+EUDC) and referred to as the New European Driving Cycle (NEDC) (Tzirakis, et al., 2006) as shown in Figure 3.9 and 3.10.

$$NEDC = 4 \times ECE - 15 city\ cycle + 1 \times EUDC\ Freeway\ drving\ Cycle$$

$$1 \times ECE - 15 = ca.\ 0{,}97 km$$

$$1 \times EUDC = ca.\ 6{,}88\ km$$

$$Total\ NEDC = ca.\ 4 \times 0{,}97 km + 6{,}88 km = 10{,}76 km$$

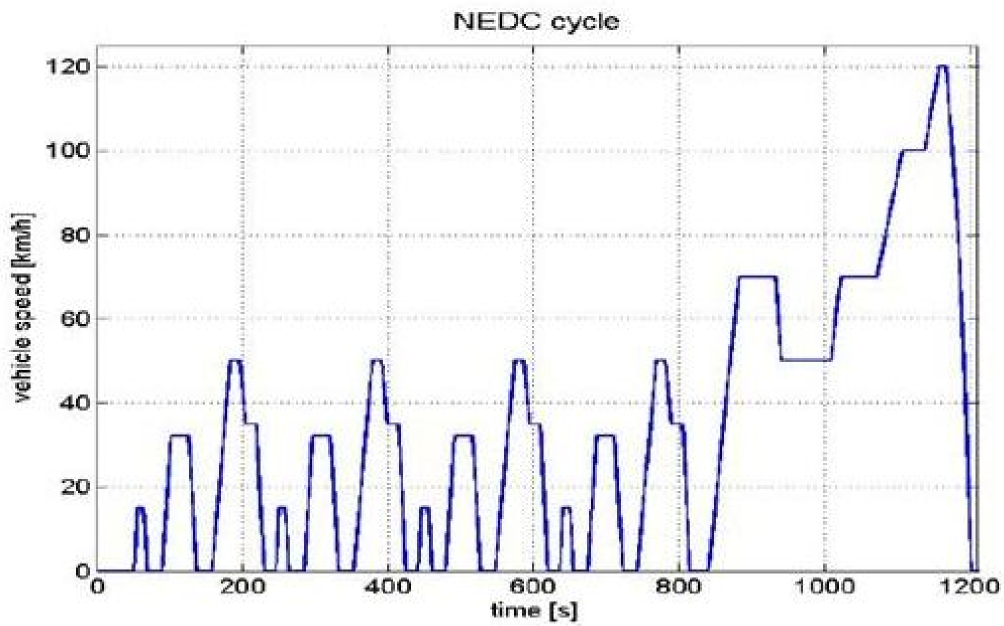

Figure 3.10 : New European Driving Cycle used by e-VS

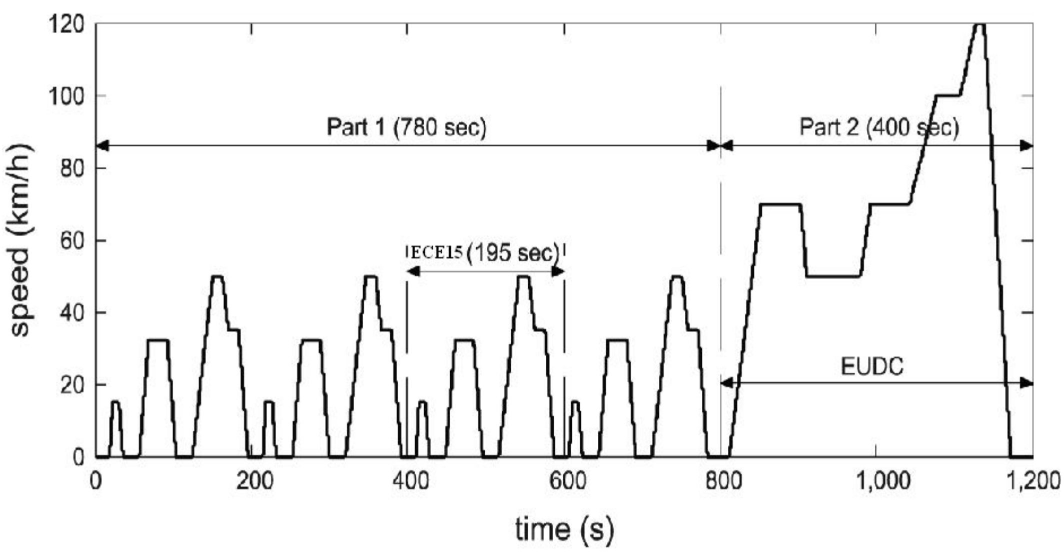

Figure 3.11 : Illustration of NEDC Driving Cycle

3.3.5 Future of Electric Vehicle

Market share of electric vehicles are expected to be among 3% and 10% between the years of 2020 and 2025 according to statistics published by ACEA. In 2015; in Europe, 480 thousand electric vehicles.

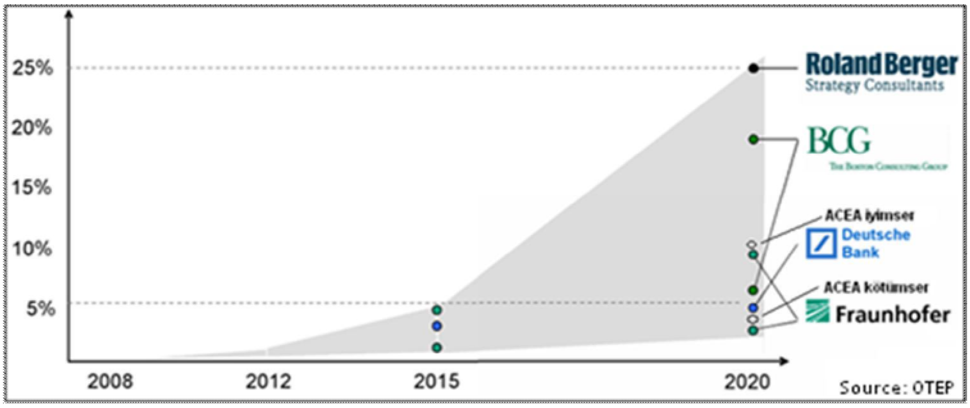

Figure 3.12 : Market Share Prospects of Electric Vehicles [Otep]

In 1996, there were just 3,280 electric vehicles in the USA. By 2007, that number had surged to over 55,000 as demonstrated by Figure 3.12. In the USA, 1 million electric vehicles and in the world, 1.7 million electric vehicle circulations are planned in accordance with incentives allocated to purchase of electric vehicles and city projects were generated upon infrastructure investments.

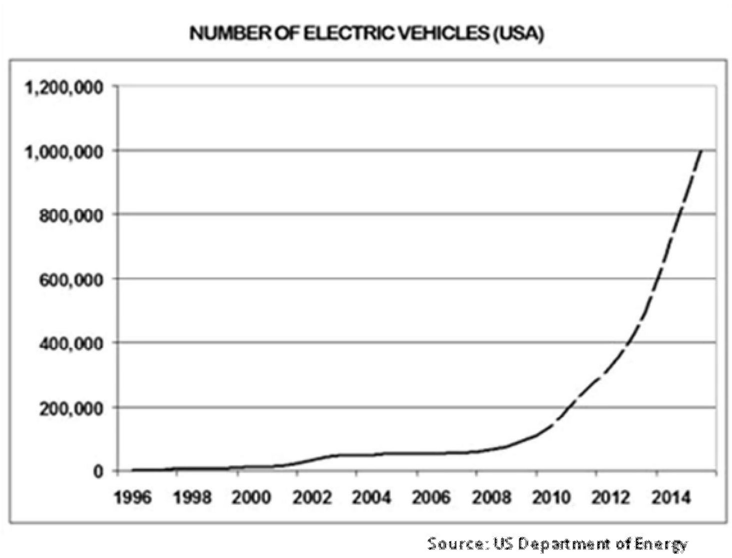

Figure 3.13 : Future of Electric Vehicles

4. MODELING AND SIMULATION OF LONGITUDINAL DYNAMICS OF ELECTRIC VEHICLES

4.1 Pacejka's Magic Formula

The most widely used nonlinear tire model is the Pacejka Model. This model and its variants are also called the Magic Tire Formulas. Typical Pacejka curves consist of longitudinal (forward) force (**F_X**), lateral (sideways) force (**F_y**) and aligning torque (**M_Z**), which you feel at the steering wheel are used in as a big part of the tire models. They model forces that are generated by the tire as a result of this not following the road precisely. Steering the tire a little and getting a slip angle are input into the Pacejka **F_y** formula giving a sideways force. Press the throttle makes wheel start to spin a bit; this gives a different ratio of wheel spin speed called ground speed and gives a forward (longitudinal) force.

Pacejka's Magic Formula is a standard in a lot of today's racing simulations applications. The development of the model was started in the mid-eighties. In a cooperative effort TU-Delft and Volvo developed several versions (1987, 1989, and 1991). Multiple versions mostly use a 1989 version of Pacejka it seems; the formula was originally taken from Giancarlo Genta's book 'Motor Vehicle Dynamics', which is from 1997. In these models the combined slip situation was modelled from a physical view point. In 1993 Michelin introduced a purely empirical method using Magic Formula based functions to describe the tyre horizontal force generation at combined slip. Magic Formula uses newly named variables that are the base fitting default Pacejka formula coefficients, which are a<n> for lateral forces, b<n> for longitudinal and c<n> for aligning torque (**M_z**), to get the mathematical curves to match the empirical tire data. It stands for Magic Formula that is used more often these days presented below. Here we focus on the default model of Pacejka and speak of the newest version of which is expressed in text.

The magic tire formula (Pacejka Model) is based on an empirical curve fit of tire force data using equations of the form. It's refer to for a detailed treatment of the pure slip part of this model (that is: at either lateral slip α or longitudinal slip s). For the side force $\mathbf{F_y}$ and the longitudinal force $\mathbf{F_X}$ that part of the model remained unchanged.

The formula reads:

$$y(x) = Dsin\left[Carctan\{Bx - E\left(Bx - actan(Bx)\right)\}\right] \qquad \textbf{(4.1)}$$

$$Y(X) = y(x) + Sv \quad ; \quad x = X + Sh \qquad \textbf{(4.2)}$$

Where;

Y**:** output variables Fx, Fy or Mz

X: input variables *s* or α.

The magic formula y(x) typically produces a curve that passes through the origin, reaches a maximum and subsequently tends to a horizontal asymptote.

4.1.1 Pacejka Tire Formulation

The basic form of each of characteristics of the tire suggests the use of the sine function as a first step in developing the final formula.

$$Y = Dsin(BX) \qquad \textbf{(4.3)}$$

with Y standing for either side force, self aligning torque or longitudinal force and X denoting slip angle (α) or longitudinal slip (s). The longitudinal slip is defined as the ratio of the difference between the speed of rotation of the driven or braked tire and of the straight free rolling tire, and the speed of rotation of the straight free rolling tire expressed as a percentage. A negative value results from a barking torque. D is the peak value and the product DB equals the slip stiffness at zero slip.

Equation (3.3) doesn’t give a good representation for larger values of X. A gradually increasing extension of the X axis appears to be necessary. To accomplish this, the arctan function has been used. The formula (3.3) now changed into;

$$Y = D\sin\left(C\arctan(BX)\right) \quad \textbf{(4.4)}$$

In Equation (3.4), D is still peak value; the slip stiffness at zero slip is now equal to the product BCD (from now on called the stiffness). The coefficient C governs the shape of the curve. For large values of X, Eq. 3.4 reduces to;

$$Y = D\sin\left(\tfrac{1}{2}\pi C\right) \quad \textbf{(4.5)}$$

Consequently, C defines the extent of the sine that will be used and therefore determines the shape of the curve. The value of C makes the curve look like a side force, longitudinal force or self aligning torque characteristic. With C determined by the shape and D determined by the peak value, only B is left to control the stiffness.

Still the equation (3.4) is not good enough to describe every possible measured characteristic. There may be a need for an additional coefficient which makes it possible to accomplish a local extra stretch or compression of the curve. The coefficient E has been introduced into the formula in such a way that stiffness and peak value remain unaffected.

$$y(x) = D\sin\left[C\arctan\left\{B(X) - E\left(B(X) - \mathrm{actan}\left(B(X)\right)\right)\right\}\right] \quad \textbf{(4.6)}$$

The influence of E on the side force characteristic has been shown in Fig. 3.1. Similar effects occur with the self aligning torque and longitudinal force characteristics. The result is an equation with four coefficients, which is able to describe all the measured characteristics. The four coefficients are B, C, D, E.

So far, the characteristics are assumed to pass through the origin. In reality, however, this will not always be the case. Due to ply steer and rolling resistance, the characteristics will be shifted in the horizontal or vertical directions. In order to fit the measured characteristics, these shifts should be included in the equation. We obtain;

$$y(x) = D\sin\left[C\arctan\left\{B(X+Sh) - E\left(B(X+Sh) - \mathrm{actan}\left(B(X+Sh)\right)\right)\right\}\right] + Sv$$

(4.7)

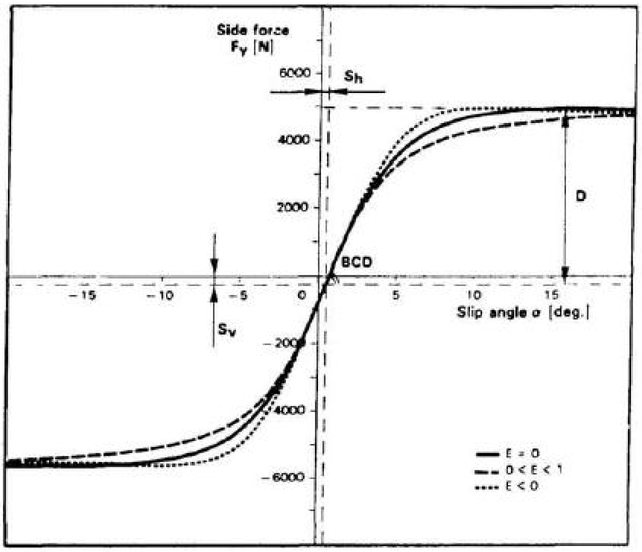

Figure 4.1 : Coefficients Appearing in Tire Formula

4.1.1.1 Using Correct Coordinate System and Units at Pacejka

The test data and resulting coefficients that come from the Pacejka tire model conform to a modified SAE tire coordinate system. The standard SAE tire coordinate system is shown next and the modified sign conventions for Pacejka are described in the tables below.

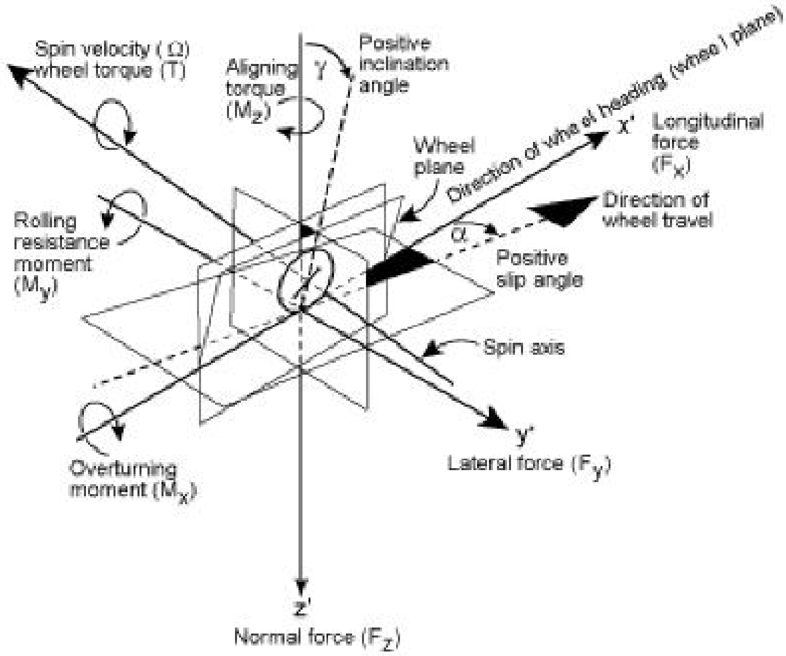

Figure 4.2 : SAE Tire Coordinate System

4.1.2 Longitudinal Force

The static input is the b<n> coefficients, and the dynamic input is load (Fz) and slip Ratio (Percentage Slip). The official Pacejka Longitudinal formula goes like this;

Longitudinal direction;

$$F_x(s) = D\sin\left[C\arctan\left\{B(s + Sh) - E\left(B(s + Sh) - actan\left(B(s + Sh)\right)\right)\right\}\right] + Sv$$

(4.8)

Where B, C, D, E, Sv, Sh are six coefficients which depend on the vertical tire force Fz and the camber angle γ. These six coefficients must be determined using experimental testing and they have no physical meaning. Sv is the y axis intercept and Sh is the x axis intercept.

Coefficient D is the maximum value (peak value) of Fx (apart from the effect of Sv). C is the shape factor (determines the shape of the peak). Sh and Sv are shifting values; they shift the curve horizontally and vertically (hence the 'h' and 'v'), and if it's recalled correctly they were added in a later stage. Sh represents forces, which appear because of the tire may look a bit like a cone, and Sv represents ply steer forces (which appear because of the direction and method with which the plies are manufactured into the tire).

B, C, D and E variables are functions of the wheel load, slip angle, slip ratio and camber. The product BCD is the slope for $s + Sh = 0$ or C_x in the linear model. The values of these coefficients are expressed as functions of 12 new coefficients b_i which are characteristic of a specific tire but also depend on road conditions and speed.

Calculating these goes as follows;

$$C = b_0 \tag{4.9}$$

$$D = (b_1 * F_Z + b_2) * F_Z \tag{4.10}$$

$$B = \left(\left(b_3 * F_Z{}^2 + b_4 * F_Z\right) * e^{(-b_5 * F_Z)}\right) / (C * D) \tag{4.11}$$

$$E = b_6 * F_Z{}^2 + b_7 * F_Z + b_8 \tag{4.12}$$

$$S_h = b_9 * F_Z + b_{10} \tag{4.13}$$

$$S_v = 0 \tag{4.14}$$

At the end,

$$F_x(s) = D\sin\left[C\arctan\left\{B * (s + Sh) - E\left(B * (s + Sh) - \arctan\left(B * (s + Sh)\right)\right)\right\}\right] + Sv \tag{4.15}$$

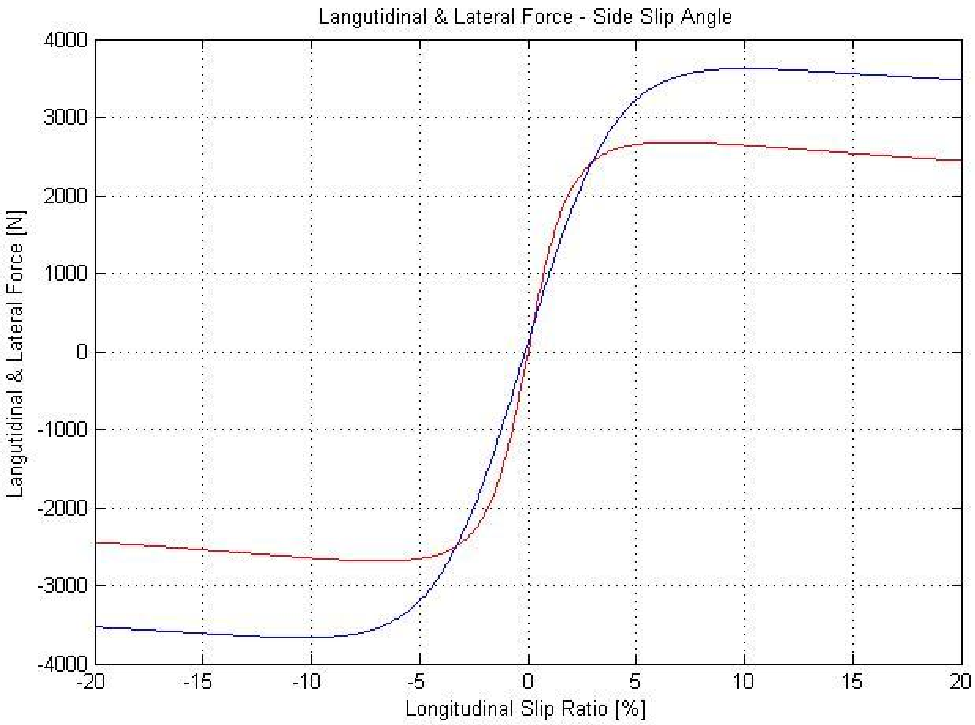

Figure 4.3 : Characteristics of Tire Forces According to Longitudinal Slip

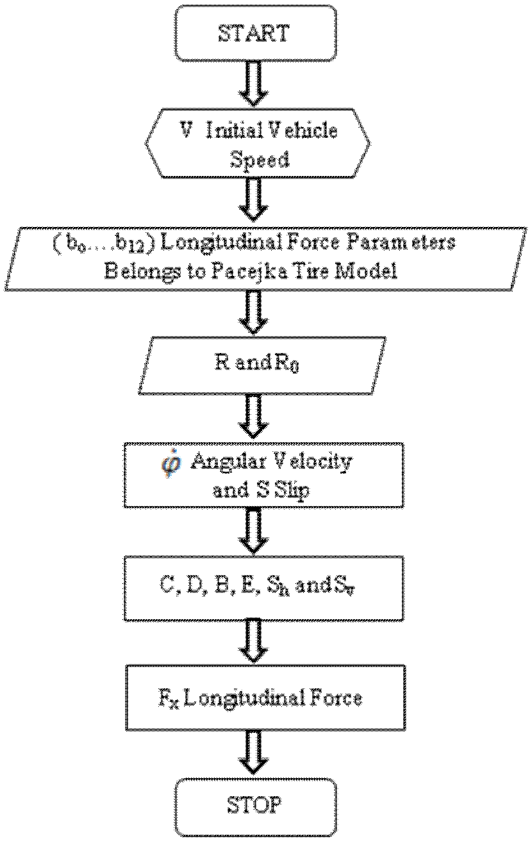

Figure 4.4 : Flow Chart of Pacejka Tire Model

Fz is the normal force. Some interpretations for the Pacejka formula are;

$D + S_v$ is the maximum force that tire can generate at its peak performance.

$B * C * D$ is the longitudinal stiffness of the tire.

$Arctan(B * C * D)$ gives the angle of the curve where it passes through the origin (apart from S_h,S_v). This is a stiffness indicator as well.

The curve usually has the following shape: linear from the origin, peaking (at height $D + S_v$), then gradually coming down again.

The asymptotic height to which the curve tends to go at big inputs is called $Y(s)$ and is equal to D*sin (0.5*PI*C).

The load sensitivity is $b1$, while $b2$ is the constant friction coefficient (the offset of the friction is $b1 * Fz$). The $b2$ value is like the friction coefficient only multiplied by 1000, so $b2 = 1005.6$ means a peak friction coefficient of 1,0056 (apart from S_v).

There are 12 coefficients that some magical constants for Longitudinal Force are required by the formula, $b0 - b12$ Ford Taurus. And these are;

4.2 Internal Combustion Engine & Electric Motor Map

In this thesis the used tire model is the Pacejka tire model. Vehicle Dynamics model, where Pacejka tire model was used, utilizes some equations and maps for some subsystems in Matlab.

These are:

- Electric motor map,
- Internal combustion engine torque map,
- Internal combustion engine fuel consumption map,
- NEDC – New european driving cycle.

The maps used by vehicle dynamics model are demonstrated in the following figures below.

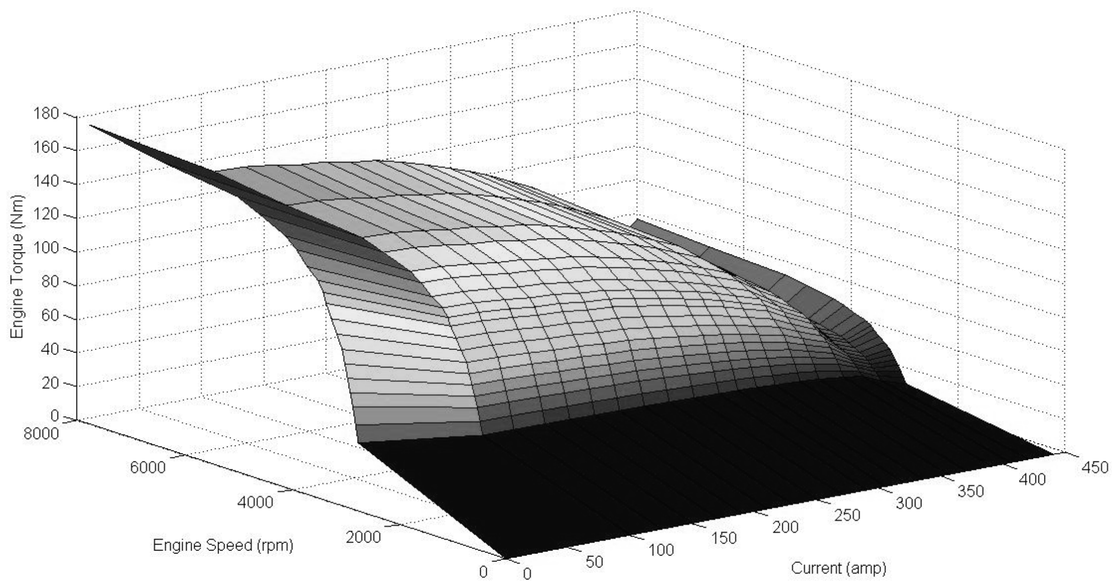

Figure 4.5 : Electric Motor Map

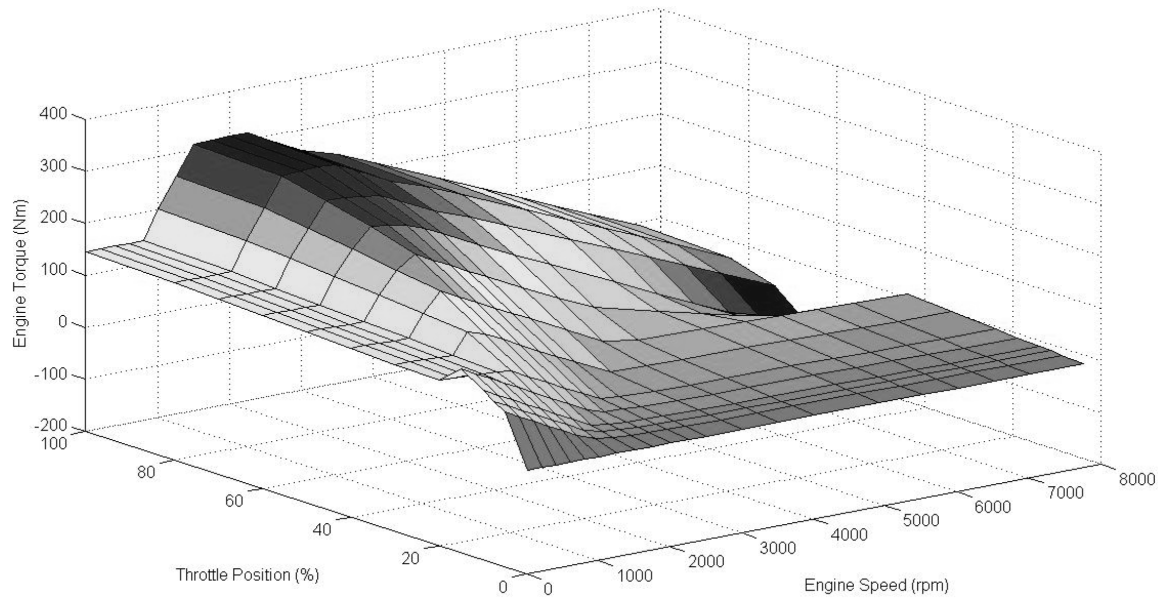

Figure 4.6 : Internal Combustion Engine Torque Map

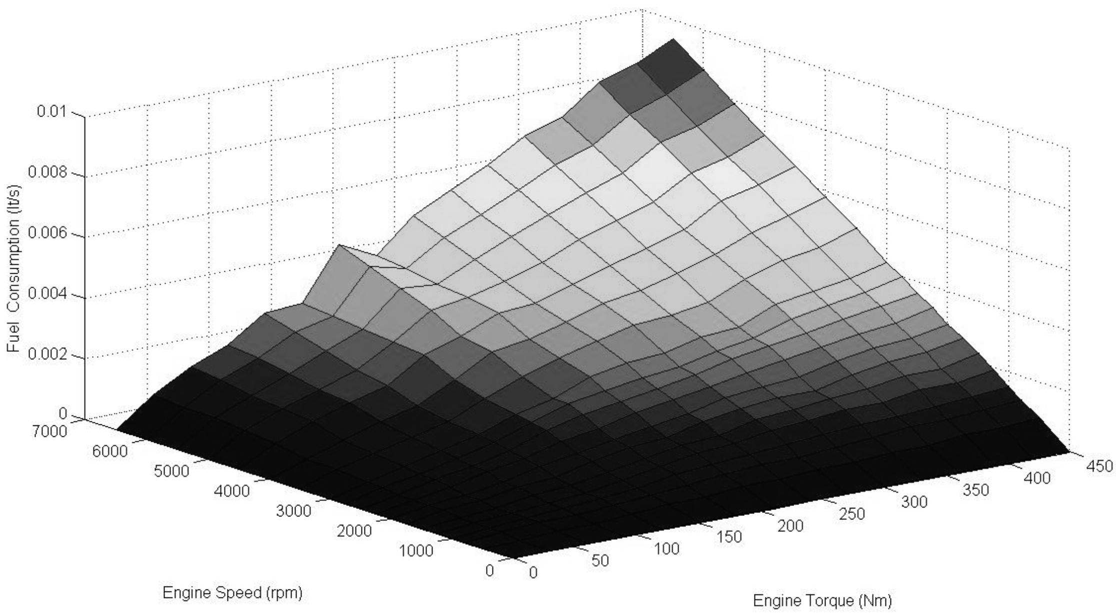

Figure 4.7 : Fuel Comsumption Map

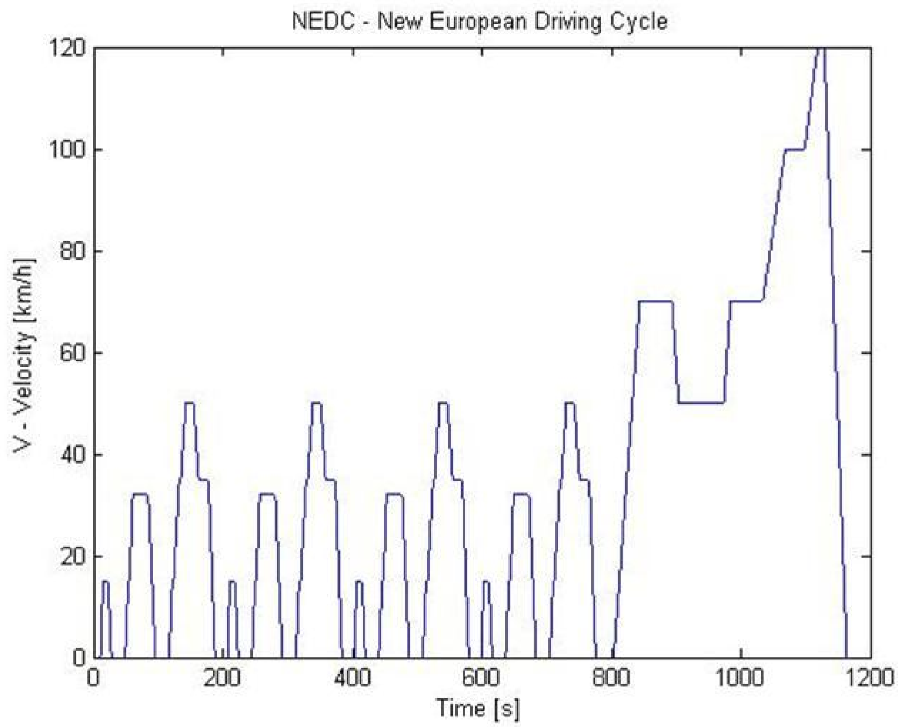

Figure 4.8 : New European Driving Cycle

4.3 Equations of Motion of the Wheel and Calculation of Longitudinal Slip

A wheel is pulled or pushed freely has a vertical load referred to Z. A torque referred to M_R will be added if wheels are driven. The figure below illustrates the motion of the wheel. The wheel centre of gravity SP_R takes the hill along the axis x and rotates by angle of φ_R.

X is here pulling force or thrust and there are additional forces at the tire-ground contact patch in addition to M_R referred to as moving or braking torque and wheel load of Z. Wheel load acts in front of the middle of contact patch by distance *"e"* since pressure distribution on the contact area is asymmetric. m_R is wheel mass and J_R is the inertia of the wheel. If wheel weight is $G_R = m_R * g$, equations of motion can be written as demonstrated below.

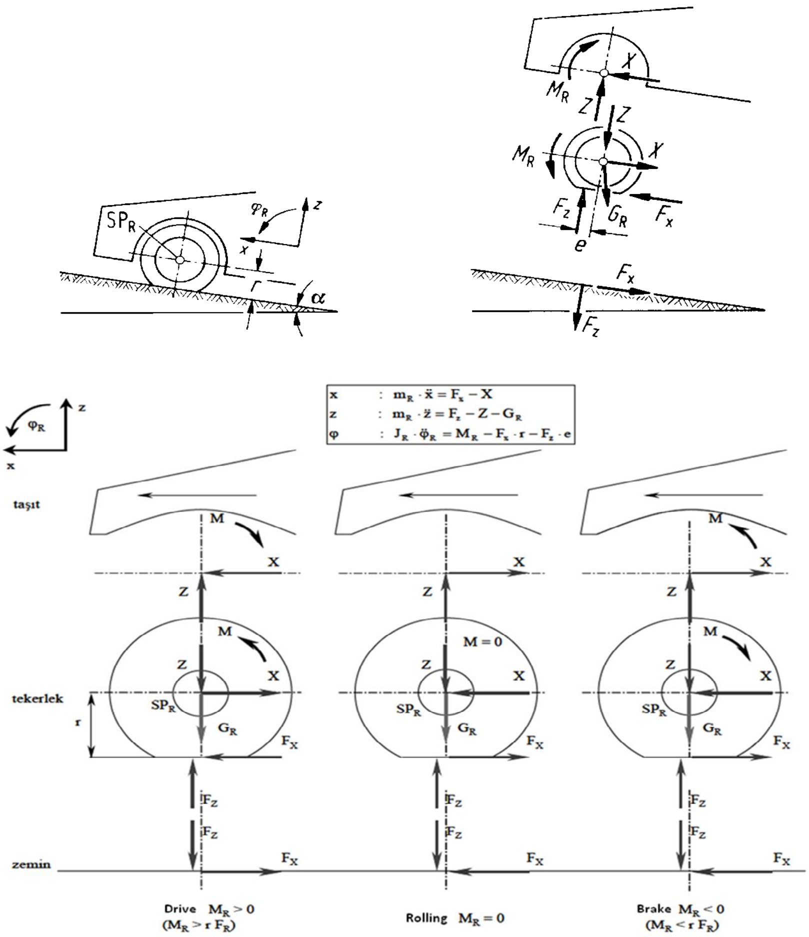

Figure 4.9 : Torques and Forces Affecting on the Wheel

$$m_R\ddot{x} = F_x - X - G_R \sin\alpha \tag{4.16}$$

$$m_R\ddot{z} = F_z - Z - G_R \cos\alpha = 0 \qquad (\ddot{z} = 0) \tag{4.17}$$

There is no acceleration on the direction of z by assuming ground surface is not rough and bumpy.

$$J_R * \ddot{\varphi}_R = M_R - F_x * r - F_z * e \tag{4.18}$$

r : Static wheel radius

In the most general case “Equation of Motion of the Wheel around its rotating axis” can be written as shown below.

$$\ddot{\varphi} = \frac{(M_{drive} - M_{brake}) - F_x * r - F_z * e}{J_R} \quad \textbf{(4.19)}$$

$$M_{nst} = M_{drive} - M_{brake}$$

Here; wheel drive torque is;

$$M_{drive} = M_m * i_g * i_a * \eta_g * \eta_a \quad \textbf{(4.20)}$$

Brake System

The system shows a linear behaviour hence the brake pedal till reached wheel brakes. And this behaviour is basically due to the geometric dimensions of the system was assumed. However, the essence of investigation does not exist in the brake system and whole correlations that depend on the brake systems construction are collected within a single coefficient.

The brake torque for any wheel of front or rear axle is,

$$M_{brake} = k_{Brake\ System} \cdot F_p$$

In this case, the brake system connects wheel brake forces to the brake pedal linearly.

F_x Longitudinal force in equation of motion of the wheel is the most important part between wheel and vehicle model. Longitudinal and lateral slip and vertical wheel load play a key part in computation of angular acceleration.

If the equation of motion of the wheel is investigated more intensive, the components,

i. Positive or negative driving shaft torque that originate from the engine,
ii. Torque effects resulting from braking system,
iii. Longitudinal force, which is occurred in consequence of longitudinal slip, diagonal motion, vertical load and ground conditions,
iv. Rolling resistance.

Constitute the basis of the longitudinal simulation while having delicate balance among themselves.

Linear dynamic is realized with this equation. Critical point here is the balance between torques and longitudinal force.

For instance, when a driving torque, which is x Nm at t time, is applied to pneumatic tire, longitudinal force is developed at the tire-ground contact patch. At the same time, if this force that is subjected to driving torque is less than driving torque, wheel will start to rotate more than free rolling and a phenomenon is usually referred to as longitudinal slip will occur. Maximization of this case is referred to as slip-max and slip is converged to value of 1 at this time. But this phenomenon has to be considered while time is progressing.

M_{net} – Net torque of wheel was chosen in accordance with the equation of motion of the wheel to get better understanding in spite of M_R.

And the equation of torque balance becomes,

$$J_R * \ddot{\varphi}_R = M_{net} - (f_R * F_z + F_x) * r \qquad \textbf{(4.22)}$$

On the smooth ground, a phrase for wheels which has no driving and braking torque on and rolling with non slip, is given by;

$$F_x = -F_R \qquad \textbf{(4.23)}$$

$$\alpha = 0,\ \ddot{x} = 0,\ \ddot{\varphi}_R = 0,\ M_R = 0 \qquad \textbf{(4.24)}$$

$$F_x = X \qquad \textbf{(4.25)}$$

$$-F_x = F_z * e/r \qquad \textbf{(4.26)}$$

$$-F_x = F_R \qquad \textbf{(4.27)}$$

$-F_x$ is the rolling resistance here and represented as F_R. Rolling resistance is related to vertical wheel load referred to as F_z.

$$F_R = f_R * F_z \tag{4.28}$$

$$f_R = e/r \tag{4.29}$$

f_R is referred to as rolling resistance coefficient.

Pneumatic tires on longitudinal direction are almost rigid. There is a part without stretch on tire structure throughout longitudinal direction of tire due to steel elements. Longitudinal deformation of tire doesn't exceed one percent of it even in excessive strains. Rolling circumference of the wheel is assumed as constant and diameter corresponds to rolling circumference is referred to as dynamic diameter. Dynamic radius of wheel R_0 is obtained by rolling circumference U. Distance covered at each rolling for a wheel rotates without slip is given by $\varphi_0 = 2\pi = 360°$

$$U = 2\pi R_0 \tag{4.30}$$

is rolling perimeter as shown above.

R is a fictive radius, while a torque acts onto the wheel. The difference between R_0 and R can be defined on the case of slip at one (S = 1). Relation between linear path and angular motions is given by;

$$x = R_0 * \varphi_0 = R * \varphi \tag{4.31}$$

And similarly, relations for velocities becomes,

$$\dot{x} = R_0 * \dot{\varphi}_0 = R * \dot{\varphi} \tag{4.32}$$

When a wheel is spinning and a driving torque is applied on it, wheel rotates without the equivalent translator progression therefore, $r\dot{\varphi} > V$ and a positive value for slip occurs. If a wheel is rotating at a certain angular velocity but linear velocity of the wheel centre is zero. The longitudinal slip will be 100%. This is often observed on an icy surface, where the driven wheels are spinning at high angular velocities, while the vehicle doesn't move forward. And the conditions of this case becomes as below,

$$x \approx 0, \ \varphi \to \infty, \ R = 0 \quad \textbf{(4.33)}$$

On the other hand, when a wheel is locked up and a braking torque is applied on it, a stretching of the tread elements occurs prior to entering the contact area in contrast with the compression effect for a driven wheel. the distance that wheels rotate will be greater than free rolling. The severity of braking is often measured by the skid of wheel. For a locked wheel, the angular velocity $\varphi = 0$ is zero, whereas the linear velocity centre is not zero. Under this condition the skid is denoted 100%. And conditions of braking case become like,

$$x \neq 0, \ \varphi = 0, \ R \to \infty \quad \textbf{(4.34)}$$

Static radius of wheel r_{statik} is independent from slip. Phrases for the case of braking and driving are given by,

Braking: $$S = 1 - \frac{R_0 * \dot{\varphi}}{V} \quad \textbf{(4.35)}$$

Driving: $$S = 1 - \frac{V}{R_0 * \dot{\varphi}} \quad \textbf{(4.36)}$$

Discrimination is done for the phrases of braking and driving below. Slip of a wheel has both braking and driving torque on it is obtained positively between the values of zero and one (0…1). The sign of longitudinal force of Pacejka (F_x) is obtained from the sign of net torque, which is established by braking and driving torque that effect onto wheels together.

Furthermore, a non-dimensional statement is obtained by dividing F_x longitudinal force to F_z wheel load.

$$\mu = F_x / F_z \quad \textbf{(4.37)}$$

This statement is referred adhesion. μ, adhesion is a function of slip of S. It has approximately the same value for the case of moving and brake.

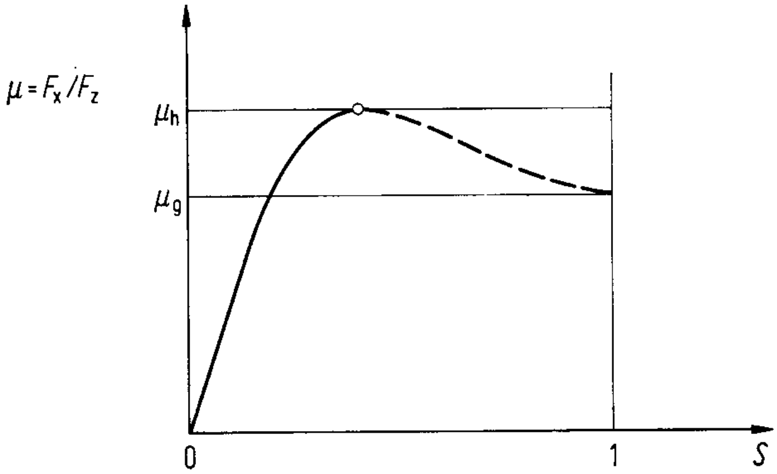

Figure 4.10 : Coefficient of Force Joint (adhesion) and Slip

The highest value of adhesion is obtained between 0,2…0,4 values of S (slip). If slip is increased some more, μ is decreased and μ_g is arrived on the case of 100 % slip (S = 1).

4.4 Euler's Method for First Order Differential Equations

Textbooks on differential equations often give the impression that most differential equations can be solved in closed form, but experience does not bear this out. It remains true that solutions of the vast majority of first order initial value problems cannot be found by *analytical* means. Therefore, it is important to be able to approach the problem in other ways. Today there are many different methods that produce numerical approximations to solutions of differential equations. Here, we use the oldest and simplest such method, originated by Euler to solve our general first order initial-value problems, which calculate linear and angular velocity at time. It is called the *tangent line method* or the *Euler method*. It uses a fixed step size "dt" and generates the approximate solution.

The dynamic behaviour of systems is an important subject. A mechanical system involves displacements, velocities, and accelerations. An equation that involves one or more derivatives of the unknown function is called an ordinary differential equation. Euler method is a numerical procedure for solving first-order differential equations with a given initial value.

The problems of solving an ODE are classified into initial-value problems (IVP) and boundary- value problems (BVP), depending on how the conditions at the endpoints of the domain are specified. All the conditions of an initial-value problem are specified at the initial point. On the other hand, the problem becomes a boundary-value problem if the conditions are needed for both initial and final points. The ODE in the time domain are initial-value problems, so all the conditions are specified at initial time, such as $t = 0, v = 0.1, \varphi = 0,\ M_m = 0\ and\ s = 0$ here we use. For notations, we use $t,\ s\ and\ M_m$ as an independent variable.

How it works:

Considering our first order differential equations given below;

$$F_x(s) = Dsin\left[Carctan\left\{B(s + Sh) - E\left(B(s + Sh) - actan\left(B(s + Sh)\right)\right)\right\}\right] + Sv \tag{4.38}$$

$$a = F_x(s)/m \tag{4.39}$$

$$\frac{dv}{dt} = a \rightarrow v = \int_{t0}^{ti} a * dt \tag{4.40}$$

$$\ddot{\varphi} = \frac{(M_m/r*ik*nk) - F_x - (F_z*1000*f_R)*r}{Jr} \tag{4.41}$$

$$\frac{d\dot{\varphi}}{dt} = \ddot{\varphi} \rightarrow \dot{\varphi} = \int_{t0}^{ti} \ddot{\varphi} * dt \tag{4.42}$$

$$t_{(ti+dt)} = t_{(ti)} + dt \tag{4.43}$$

Starting at some time, to and step size dt; the value of $v_{(to+dt)}$ can then be approximated by the value of $v_{(to)}$ plus the time step multiplied by the slope of the function, which is the derivative of $v_{(t)}$

$$v = \int_{t0}^{ti} a * dt \tag{4.44}$$

$$v_{(ti+dt)} = v_{(ti)} + dt * a_{(ti)} \tag{4.45}$$

$$t_{(ti+dt)} = t_{(ti)} + dt \tag{4.46}$$

Similarly; starting at some time, to and step size dt for the approximation of angular velocity $\dot{\varphi}$, the value of $\dot{\varphi}_{(to+dt)}$ can then be approximated by the value of $\dot{\varphi}_{(to)}$ plus the time step multiplied by the slope of the function, which is the derivative of $\dot{\varphi}_{(t)}$

$$\dot{\varphi} = \int_{t0}^{ti} \ddot{\varphi} * dt \tag{4.47}$$

$$\dot{\varphi}_{(ti+dt)} = \dot{\varphi}_{(ti)} + dt * \ddot{\varphi}_{(ti)} \tag{4.48}$$

$$t_{(ti+dt)} = t_{(ti)} + dt \tag{4.49}$$

So, if we can calculate the value of dv/dt and $d\dot{\varphi}/dt$ at time t_0 using the equations above, then we can generate an approximation for the value of $v\ and\ \dot{\varphi}$ at time $t_0 + dt$ using those equations given above. We can then use this new value of y (at to) to find dv/dt and $d\dot{\varphi}/dt$ (at to) and repeat. Although this seems circular, if properly used it can generated an approximate solution. This is referred to as Euler's method.

So we can basically state this simple background Euler Method for the first order differential equations as follows;

1) Starting at time to, choose a value for dt, and give initial condition $v_{(to)}\ and\ \dot{\varphi}_{(to)}$.
2) From $v_{(to)}\ and\ \dot{\varphi}_{(to)}$ calculate the derivative of $v_{(to)}\ and\ \dot{\varphi}_{(to)}$ $at\ t = to + dt$.

3) From this value find an approximate value for $v_{(to+dt)}$ and $\dot{\varphi}_{(to+dt)}$

4) Let $t_i = t_i + dt$, and

$$v_{(ti)} = v_{(ti)} + dt * a_{(ti)} \; and \; \dot{\varphi}_{(ti)} = \dot{\varphi}_{(ti)} + dt * \ddot{\varphi}_{(ti)}$$

5) Repeat steps 2 and 3, 4 … until the solution is finished.

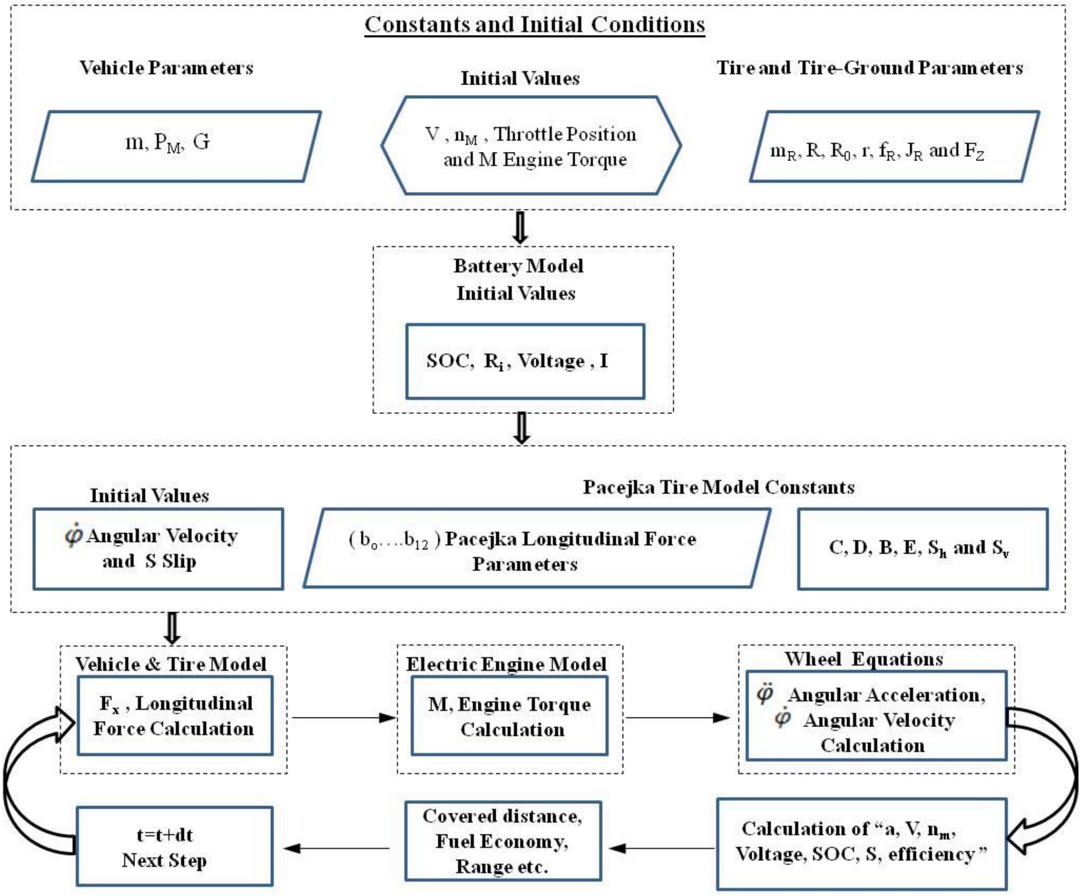

Figure 4.11 : Flow Chart of "Tire & Vehicle Model" in Detail

EV Model is also controlled by a virtual driver and driven in accordance with a specific road in Matlab programme which works as data-driven. The vehicle model is fed with a maximum step of 1 ms as in the following input variables are Time, Longitudinal force, Throttle Position, Brake Pedal Force, Electric Motor Moment, Current, State of Charge in the Matlab programme written for electric vehicle..

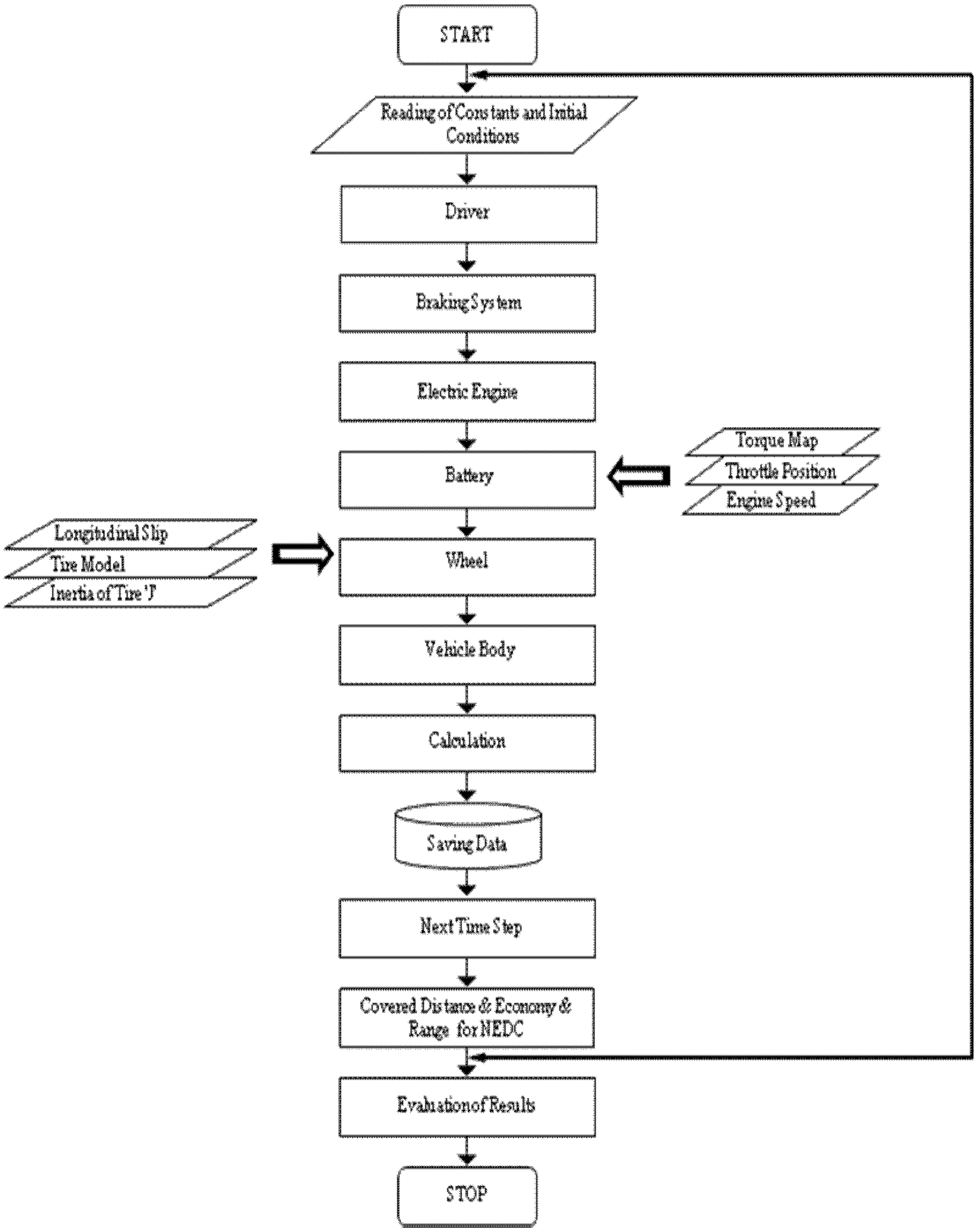

Figure 4.12 : Calculus Flow of Whole Vehicle Model

In the meantime, the magnitudes of vehicle motion (position, velocity & acceleration) should be kept in values and the vehicle is controlled by the acceleration pedal & brake pedal in the direction requested. The programme reads all the data & parameters of the vehicle and maps belong to the electric vehicle system & subsystem at the same time.

5. RESULTS AND EVALUATION

In the last chapter, calculations are done by Matlab programmes relating to the longitudinal dynamics software of electric and internal combustion engine vehicle models. And the results were evaluated by energy consumption and range.

Simulation Results of Internal Combustion Engine Vehicle For ECE-15 City Driving Cycle

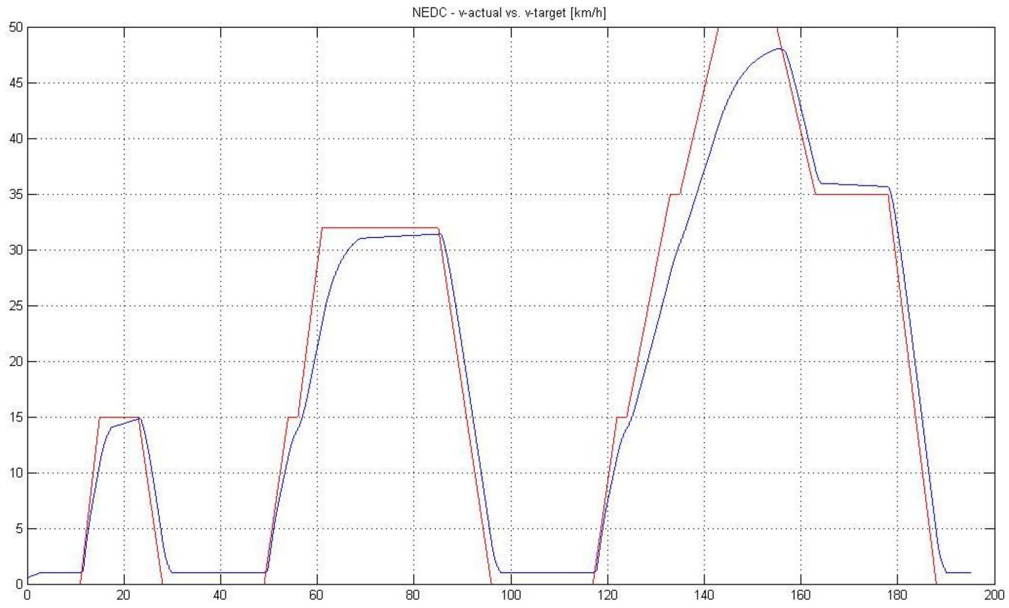

Figure 5.1 Actual & Target Speed of Vehicle

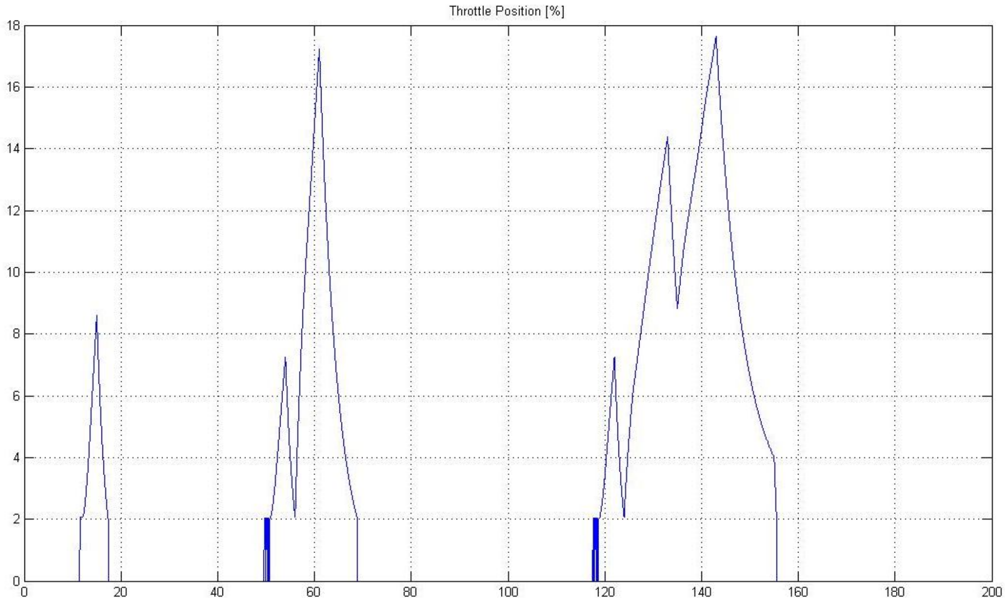

Figure 5.2 Throttle Position

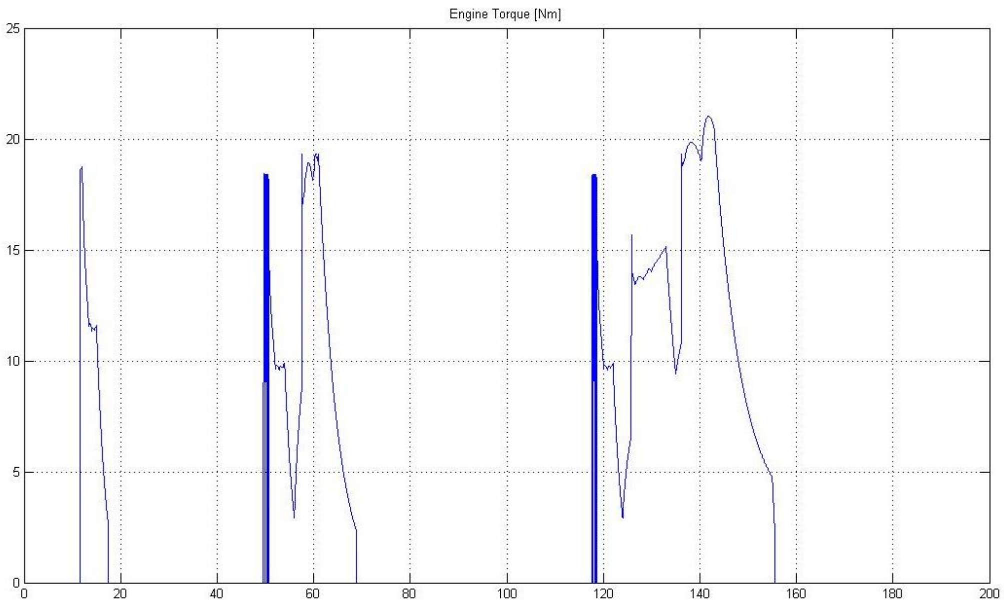

Figure 5.3 Engine Torque

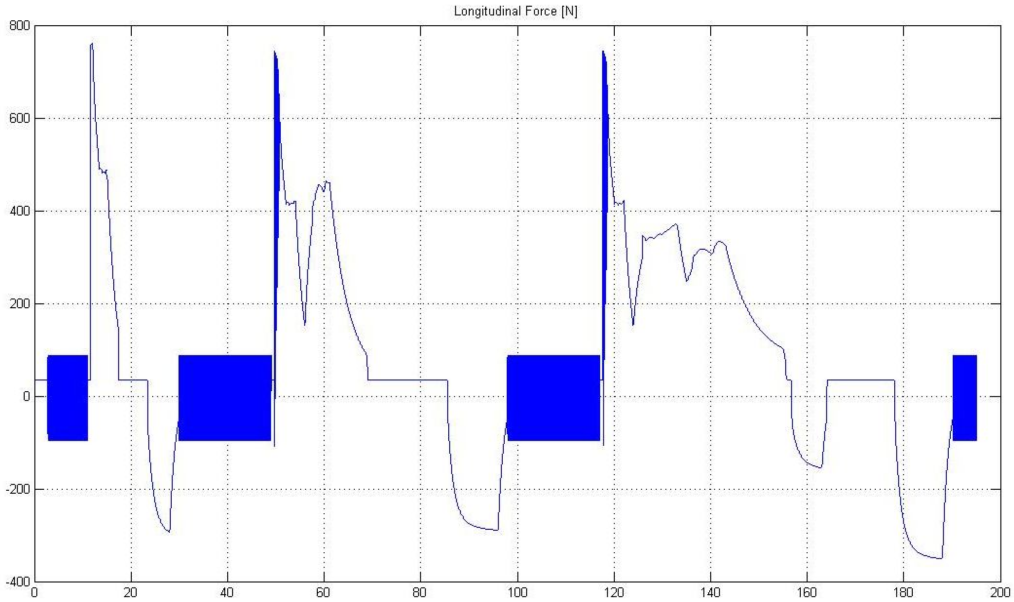

Figure 5.4 Longitudinal Force

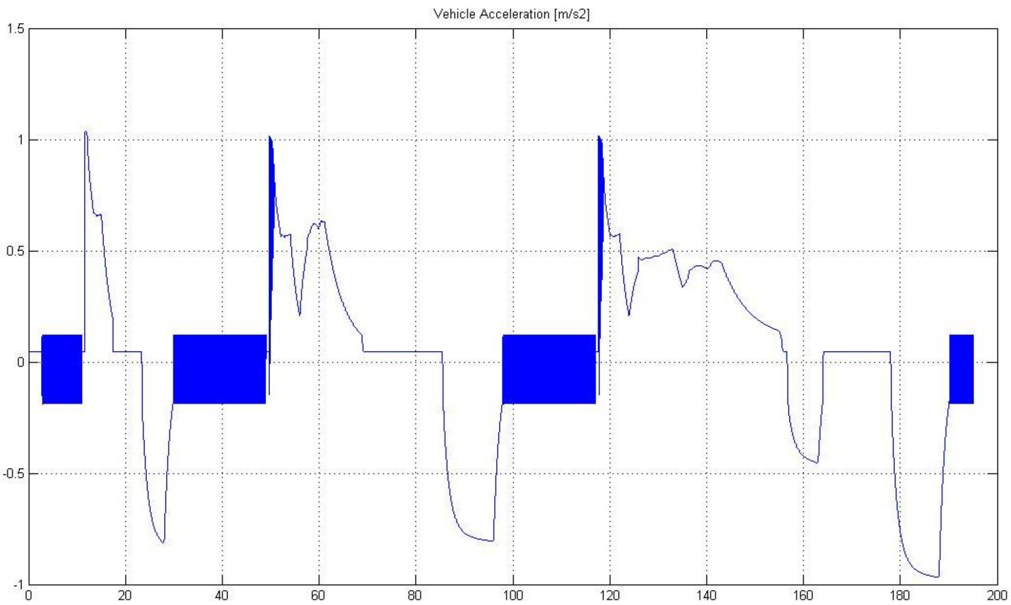

Figure 5.5 Vehicle Acceleration

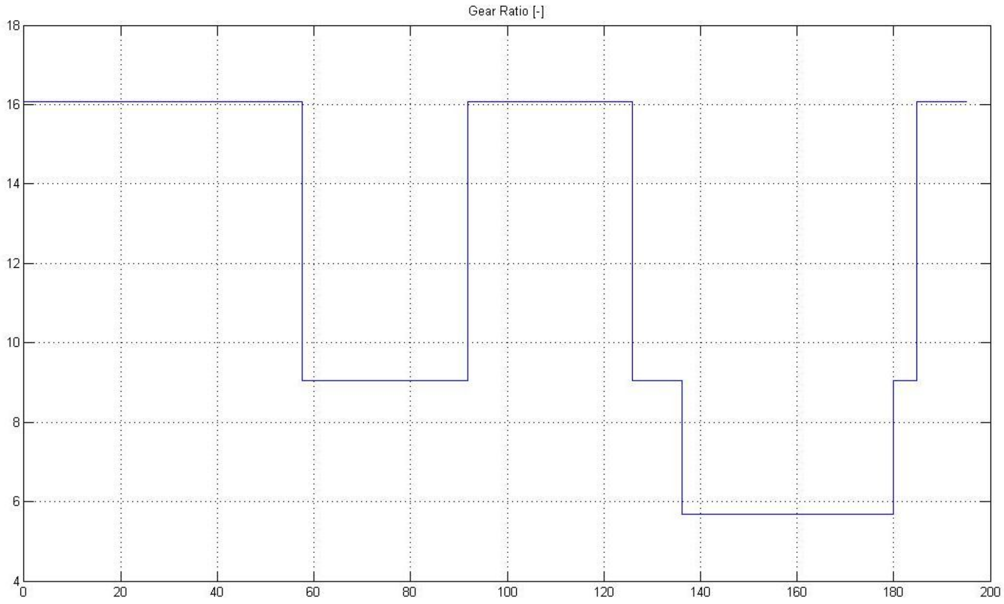

Figure 5.6 Gear Ratio

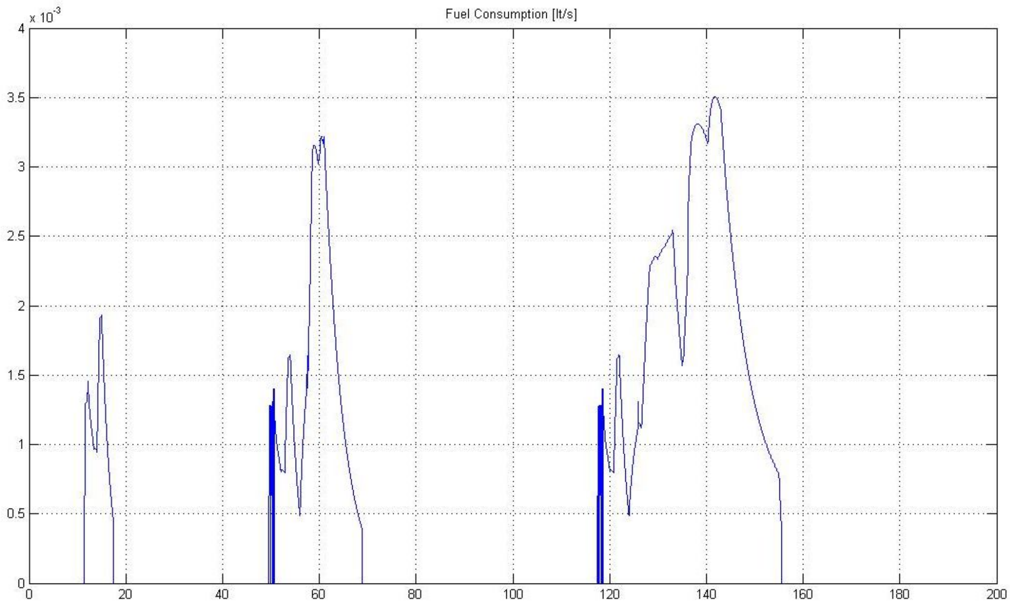

Figure 5.7 Fuel Consumption (lt/s)

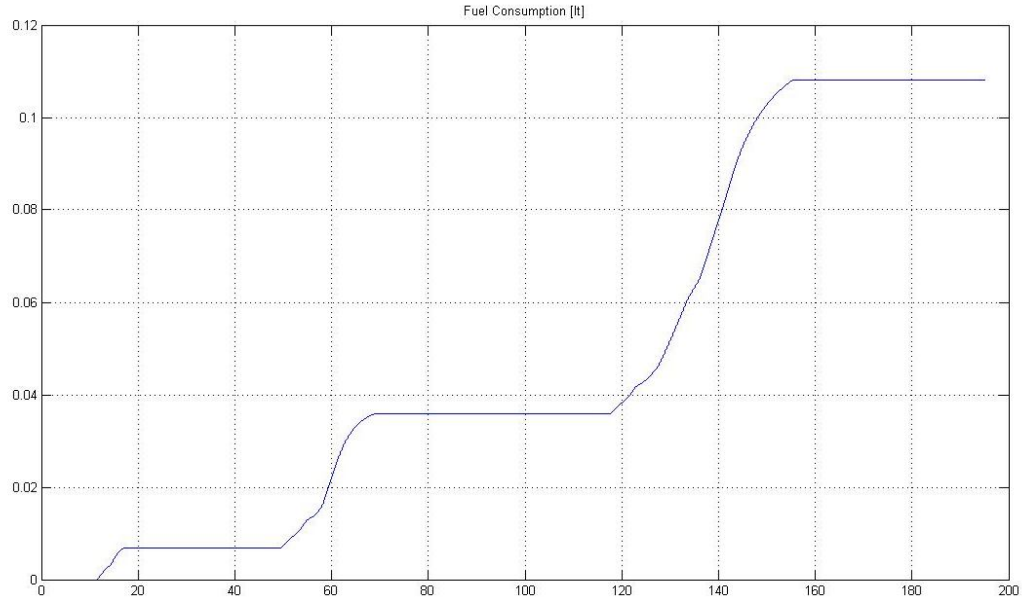

Figure 5.8 Fuel Consumption (lt)

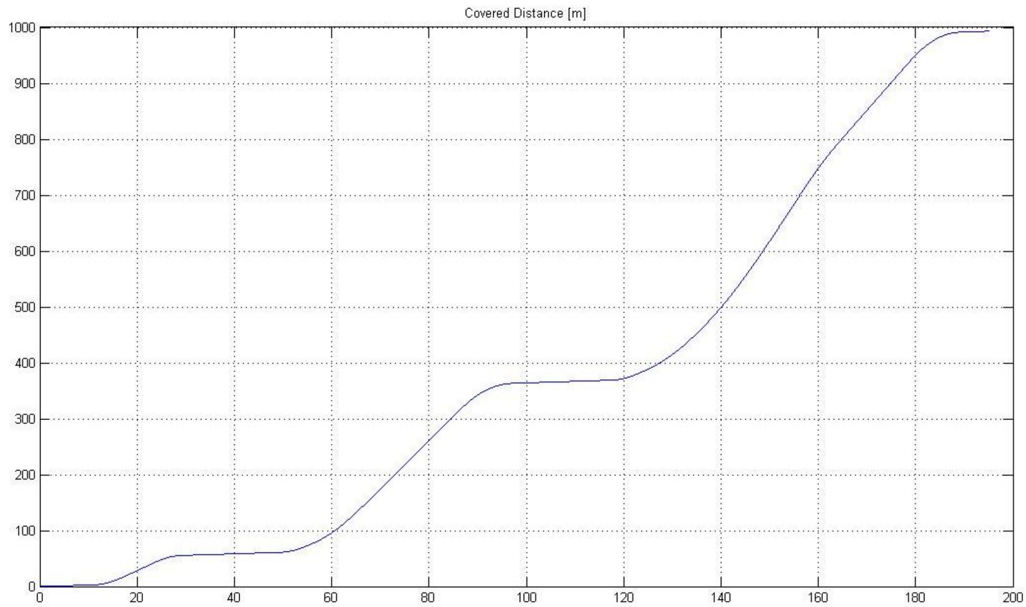

Figure 5.9 Covered Distance

Simulation Results of Internal Combustion Engine Vehicle For EUDC Freeway Driving Cycle

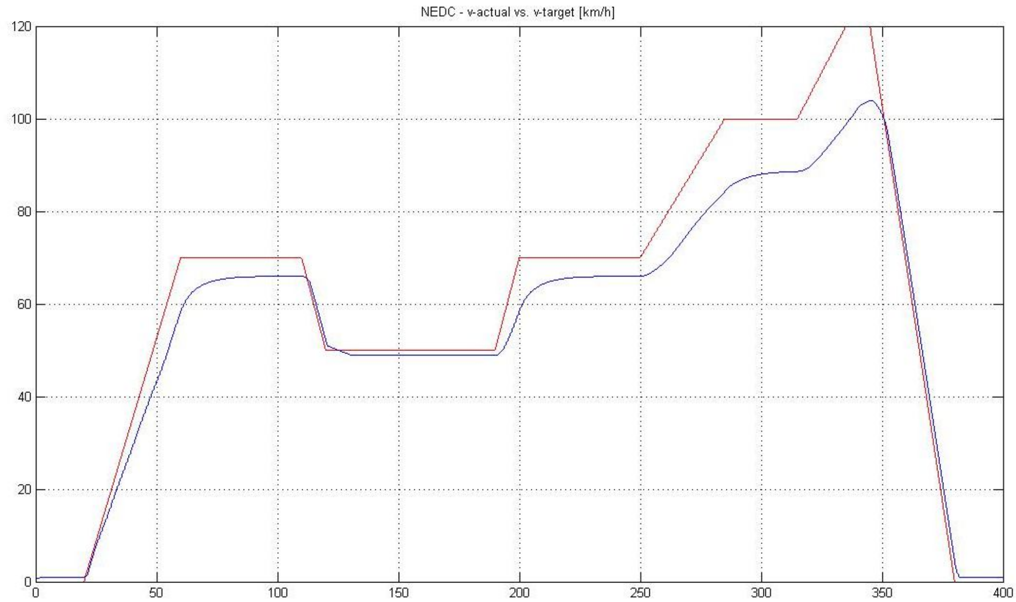

Figure 5.10 Actual & Target Speed of Vehicle

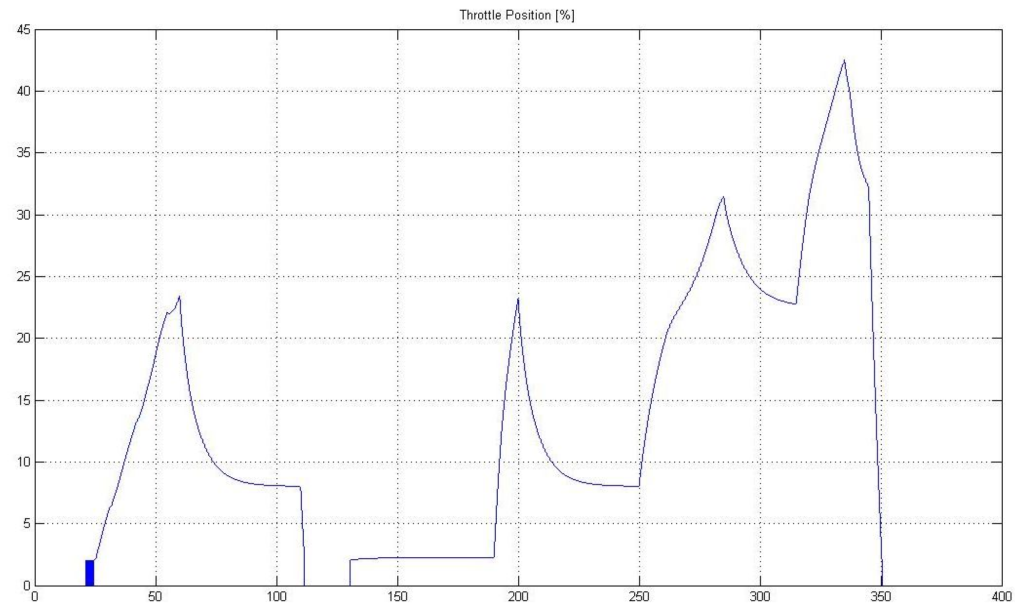

Figure 5.11 Throttle Position

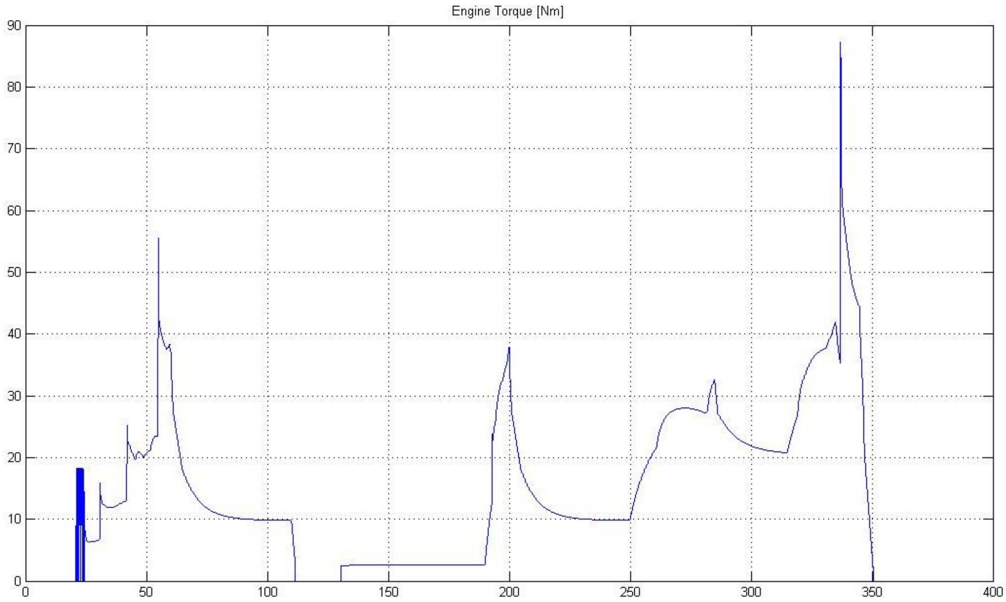

Figure 5.12 Engine Torque

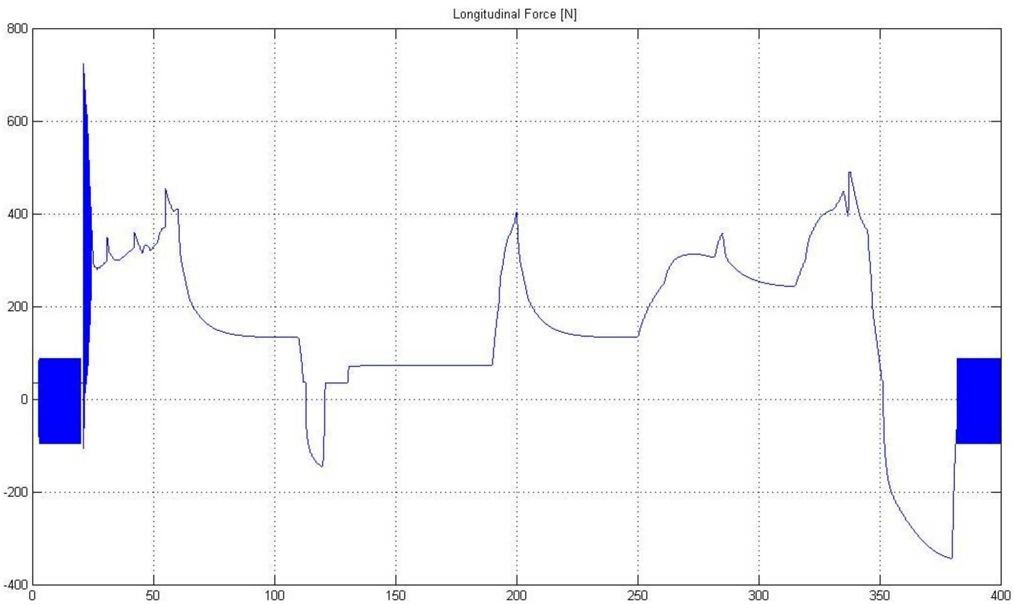

Figure 5.13 Longitudinal Force

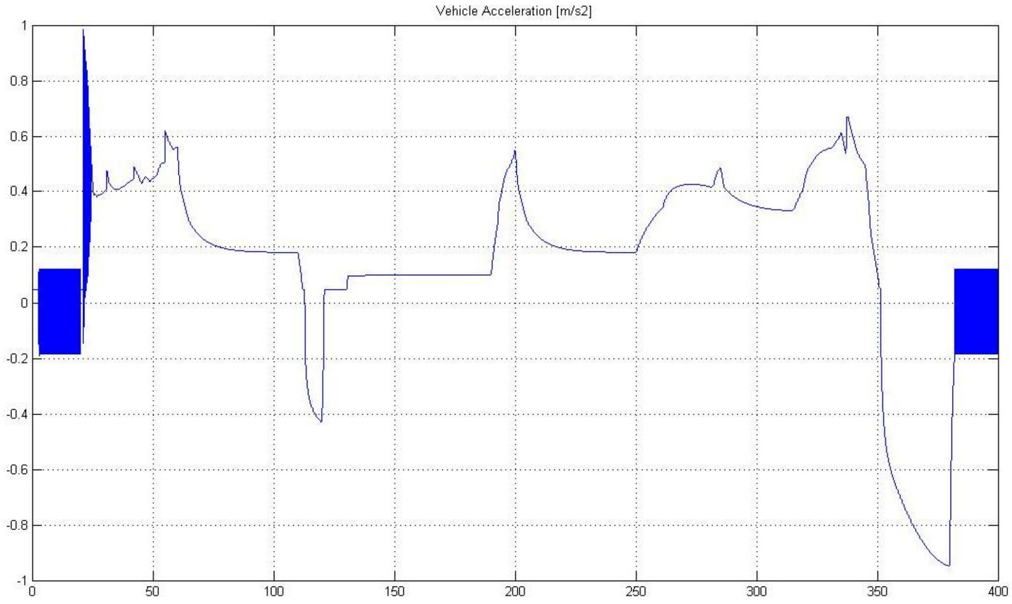

Figure 5.14 Vehicle Acceleration

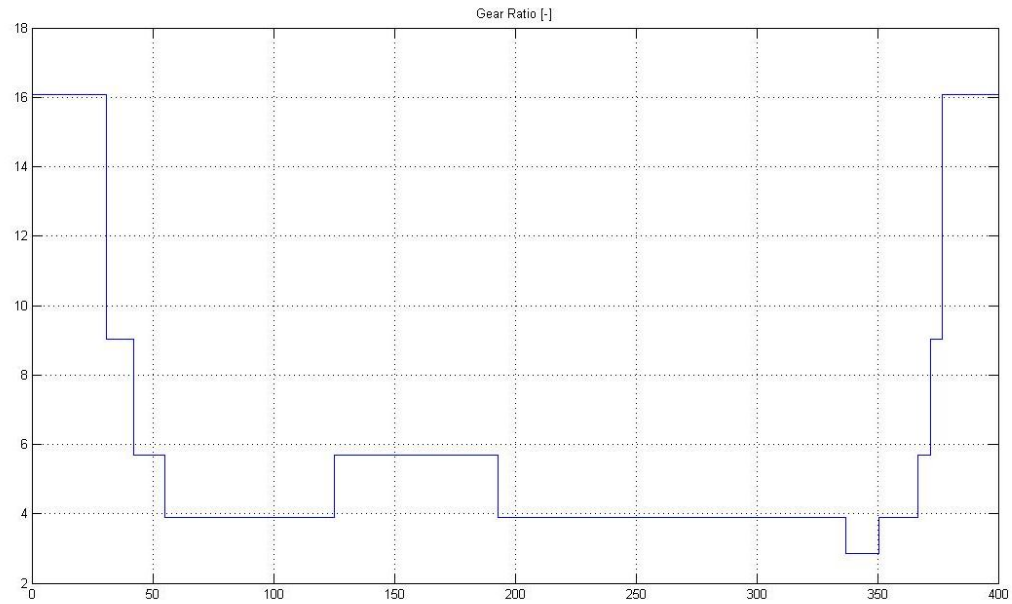

Figure 5.15 Gear Ratio

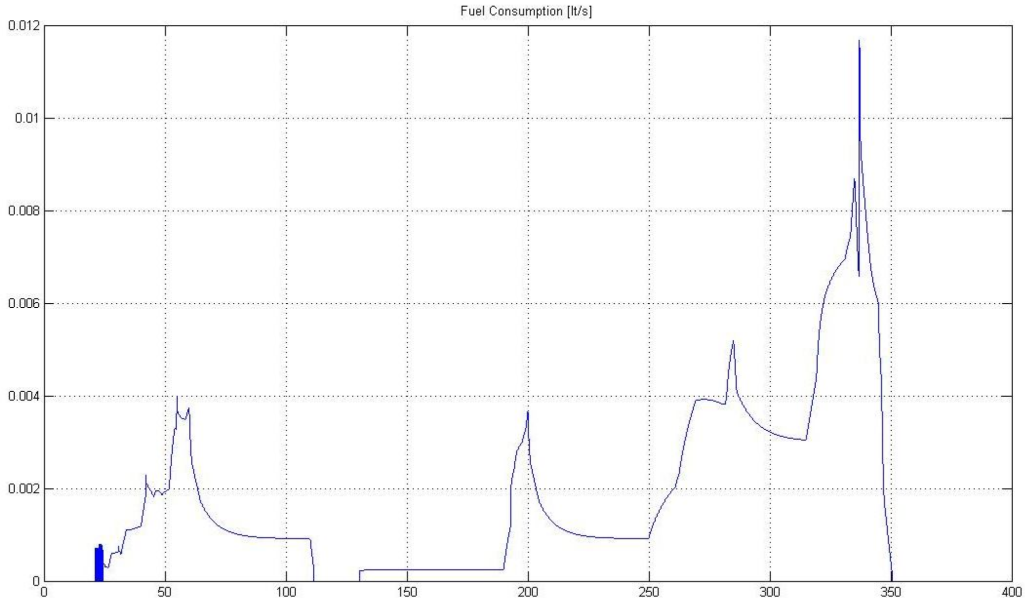

Figure 5.16 Fuel Consumption (lt/s)

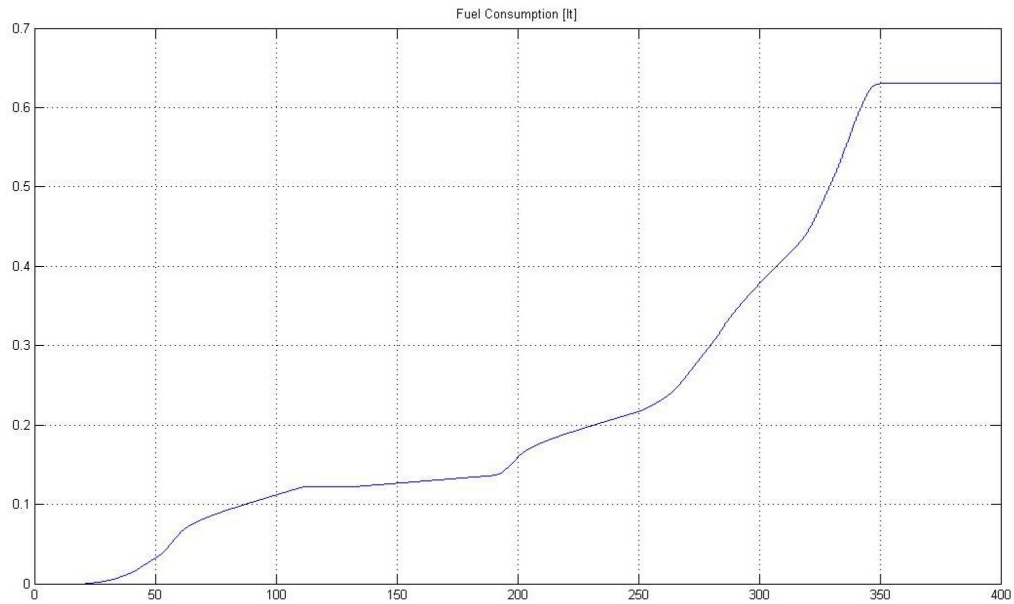

Figure 5.17 Fuel Consumption (lt)

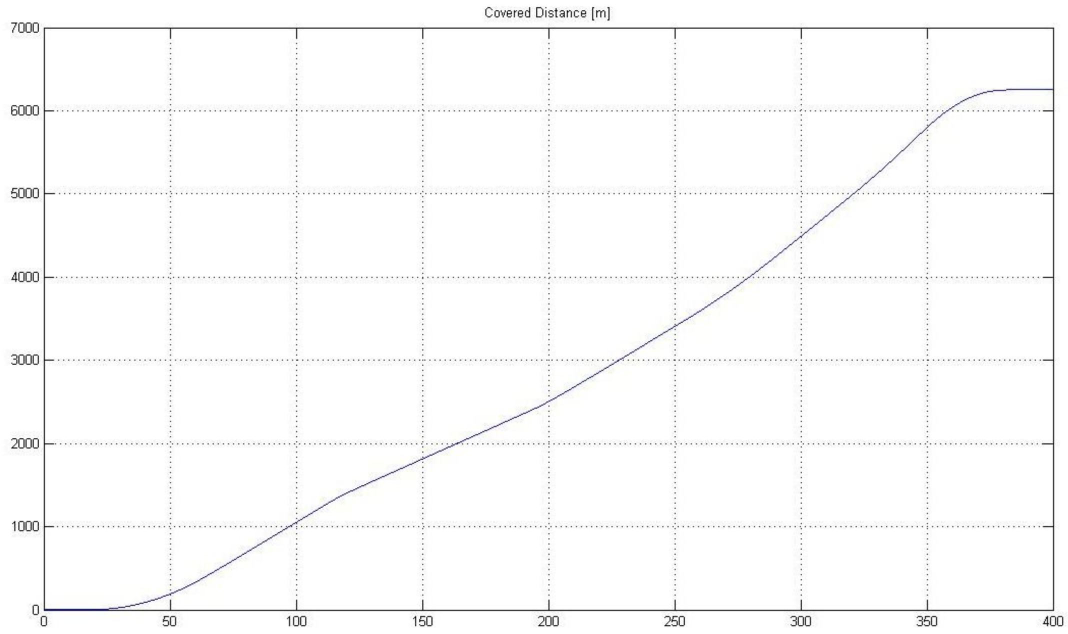

Figure 5.18 Covered Distance

Simulation Results of Electric Vehicle For ECE-15 City Driving Cycle

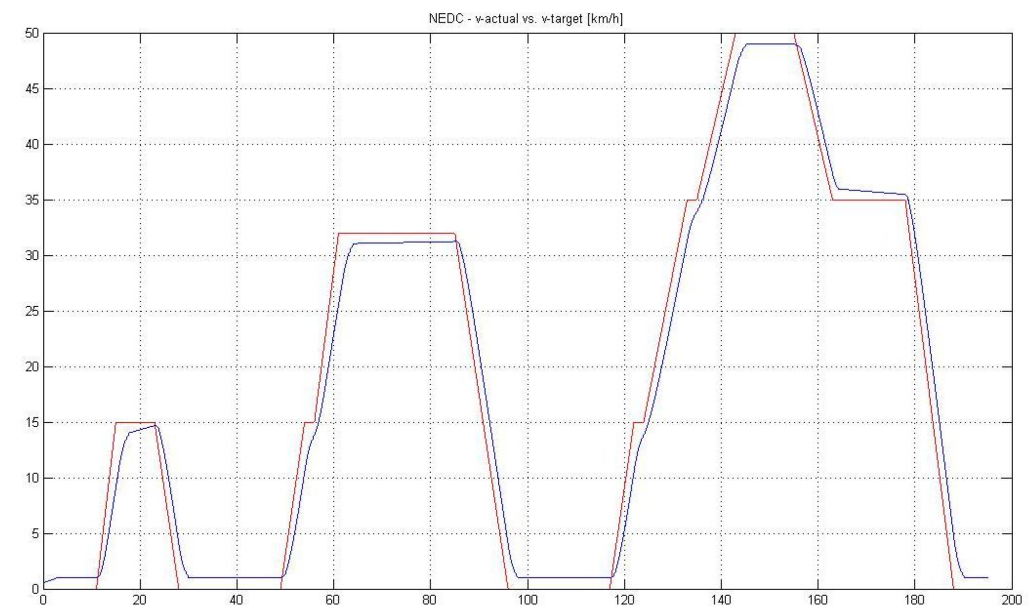

Figure 5.19 Actual & Target Speed of Vehicle

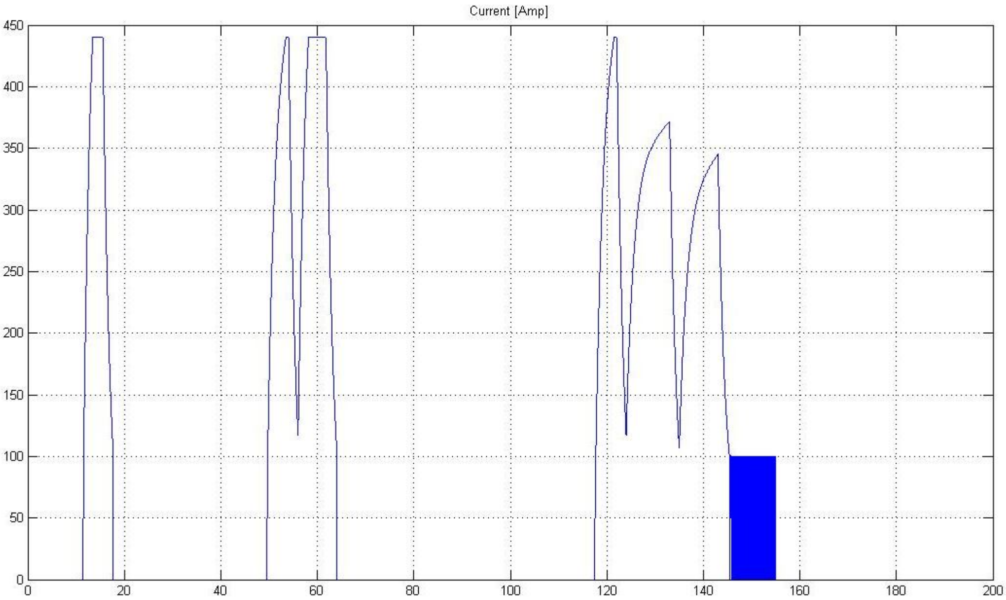

Figure 5.20 Current

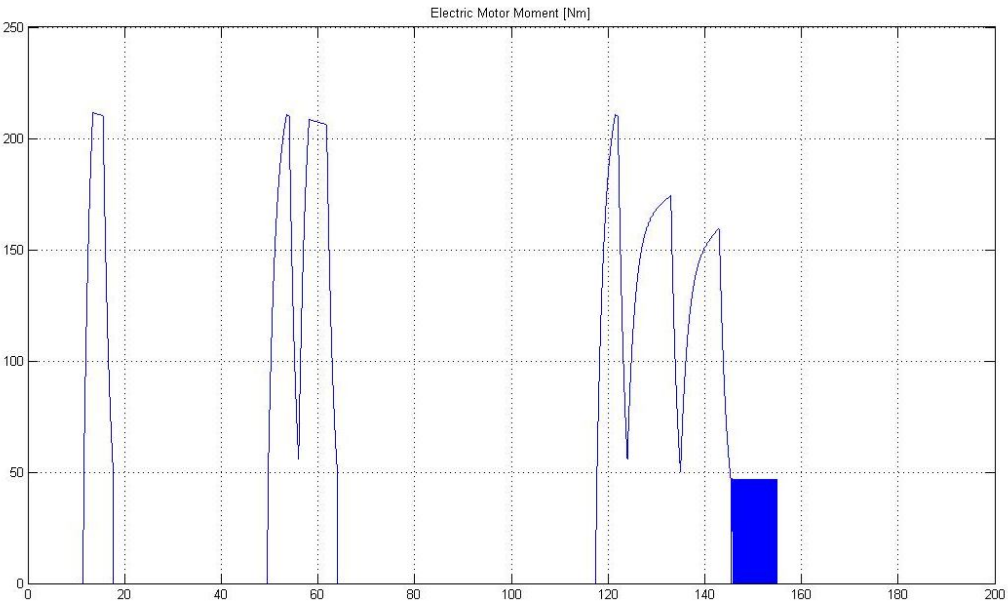

Figure 5.21 Electric Motor Torque

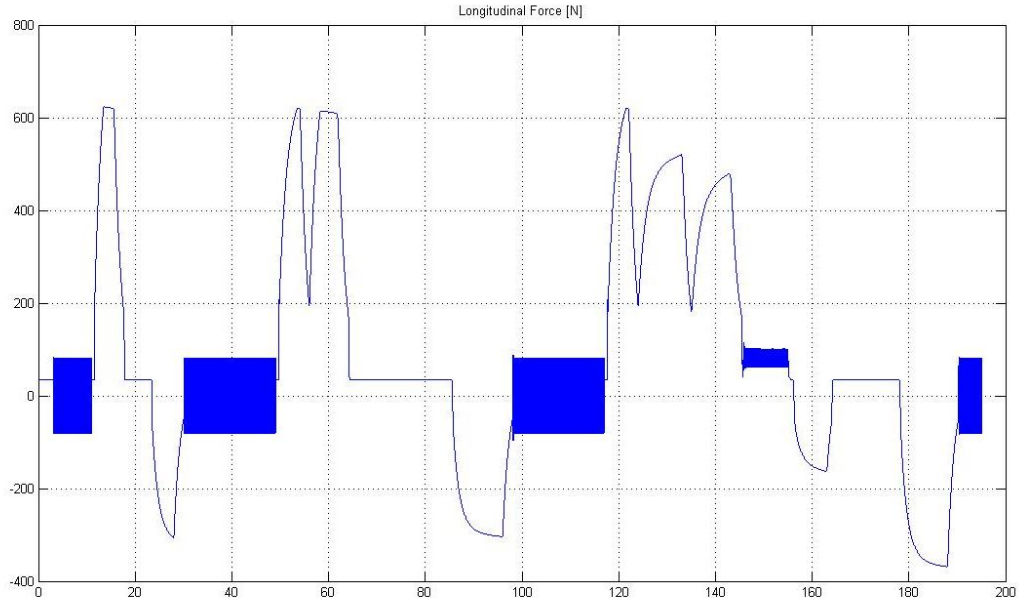

Figure 5.22 Longitudinal Force

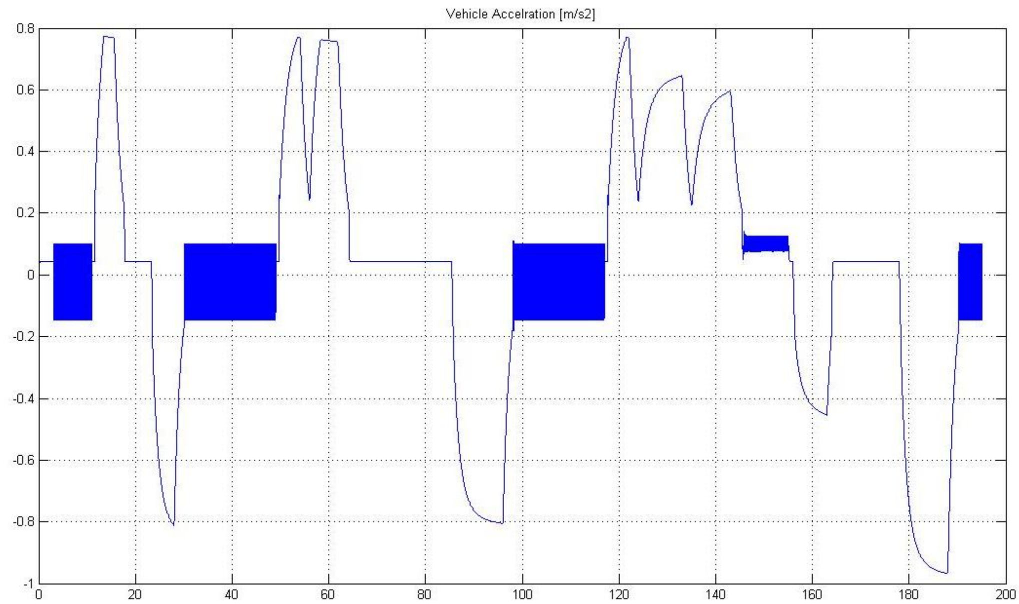

Figure 5.23 Electric Vehicle Acceleration

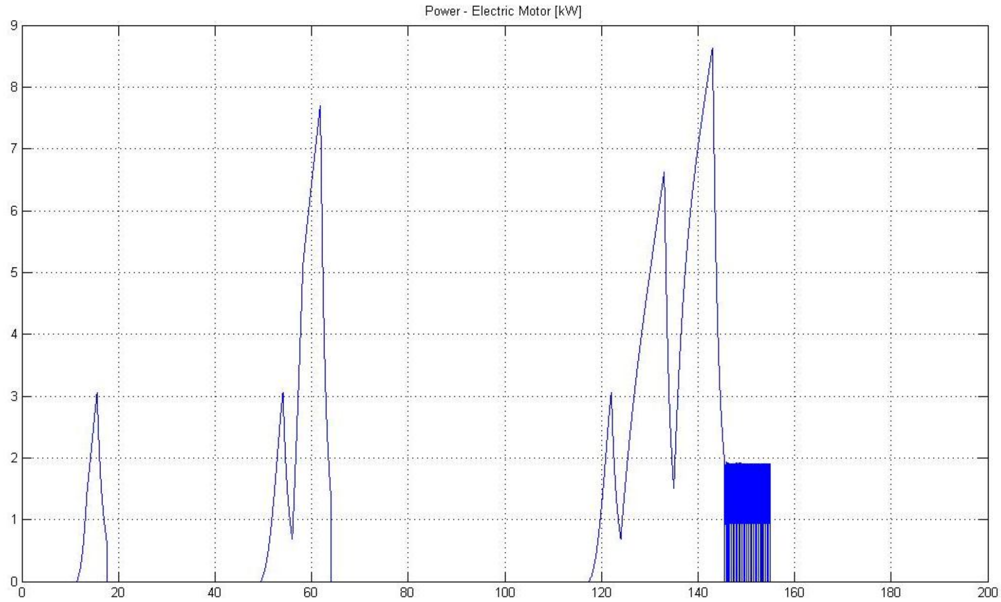

Figure 5.24 Power of EV

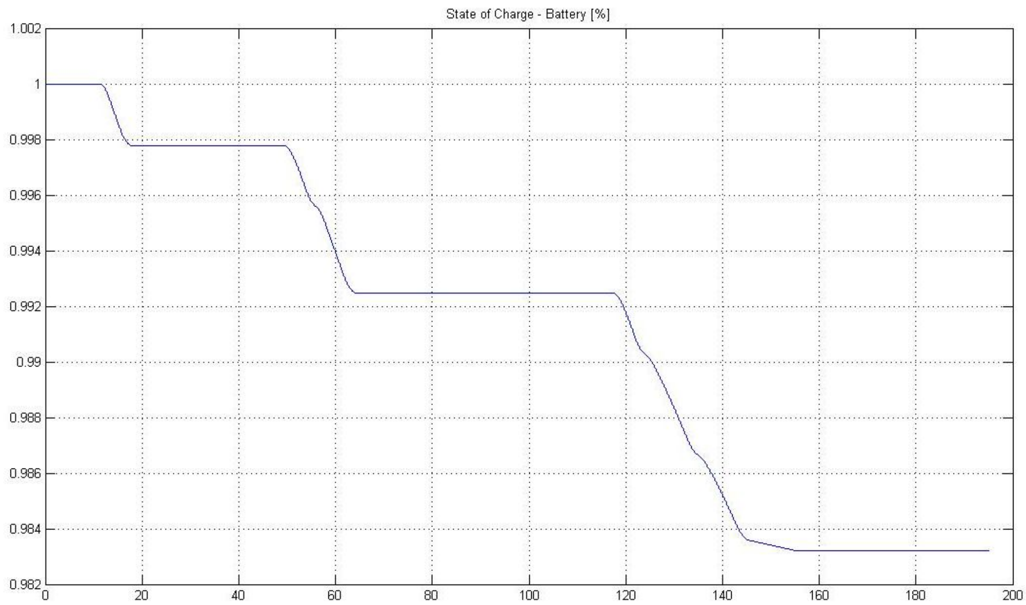

Figure 5.25 State of Charge (SOC)

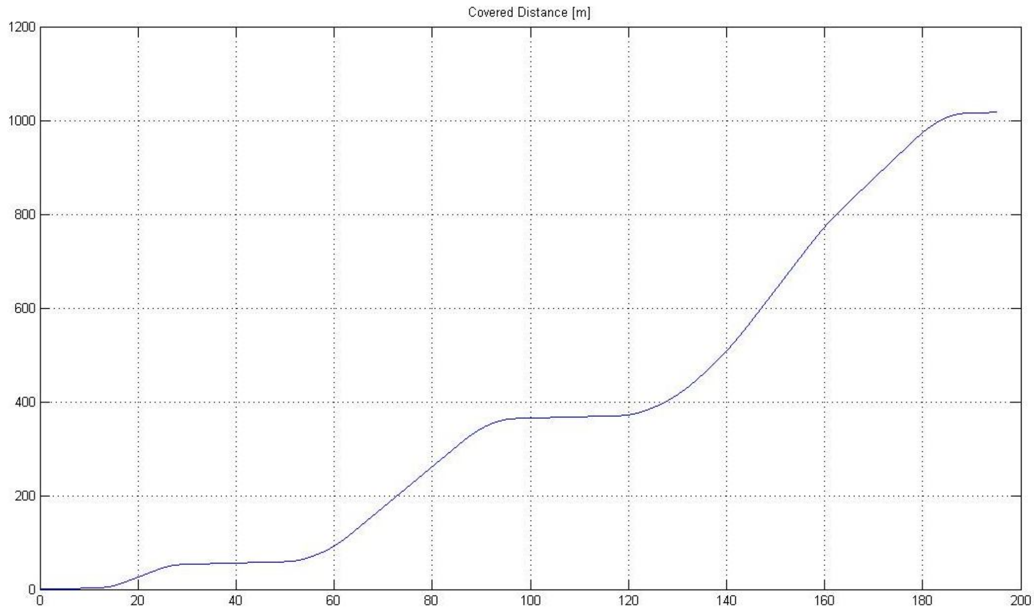

Figure 5.26 Covered Distance

Simulation Results of Electric Vehicle For EUDC Freeway Driving Cycle

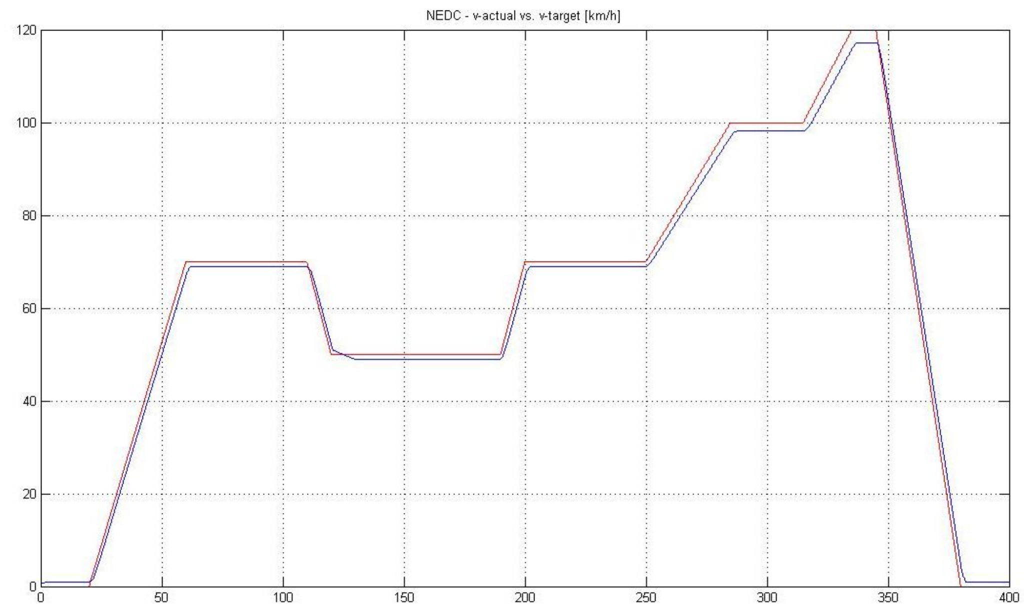

Figure 5.27 Actual & Target Speed of Vehicle

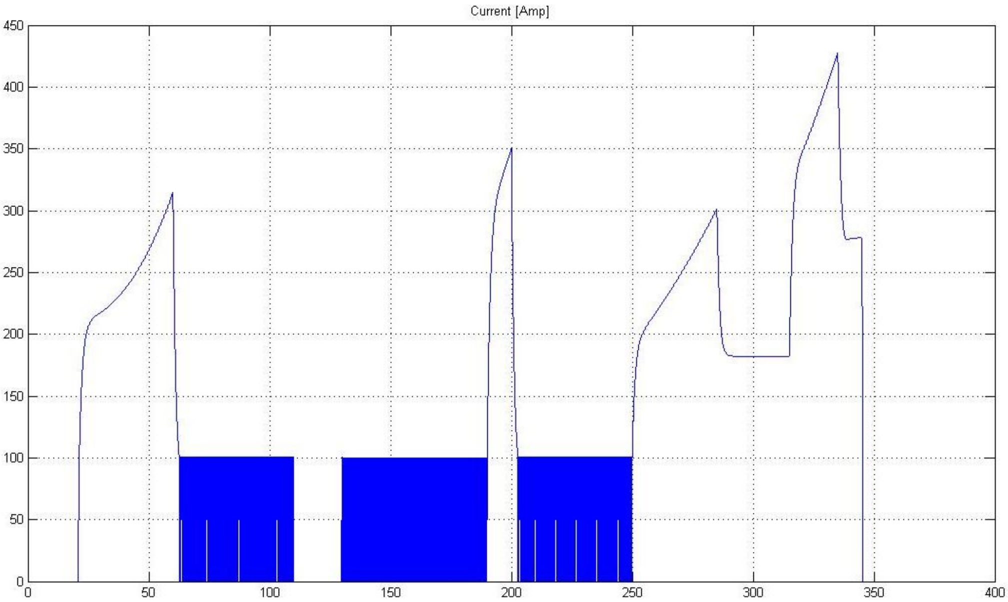

Figure 5.28 Current

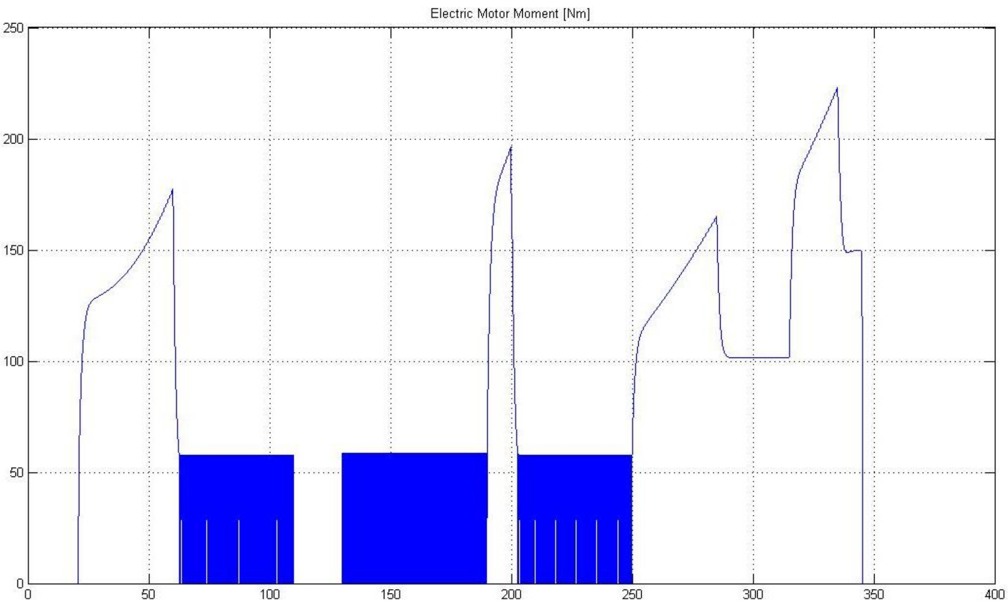

Figure 5.29 Electric Motor Torque

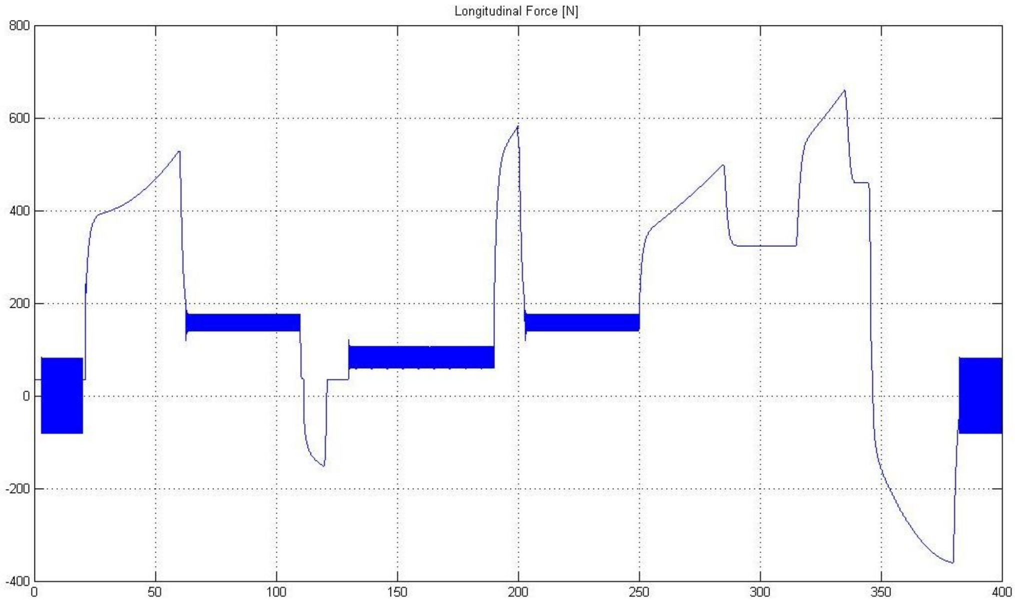

Figure 5.30 Longitudinal Force

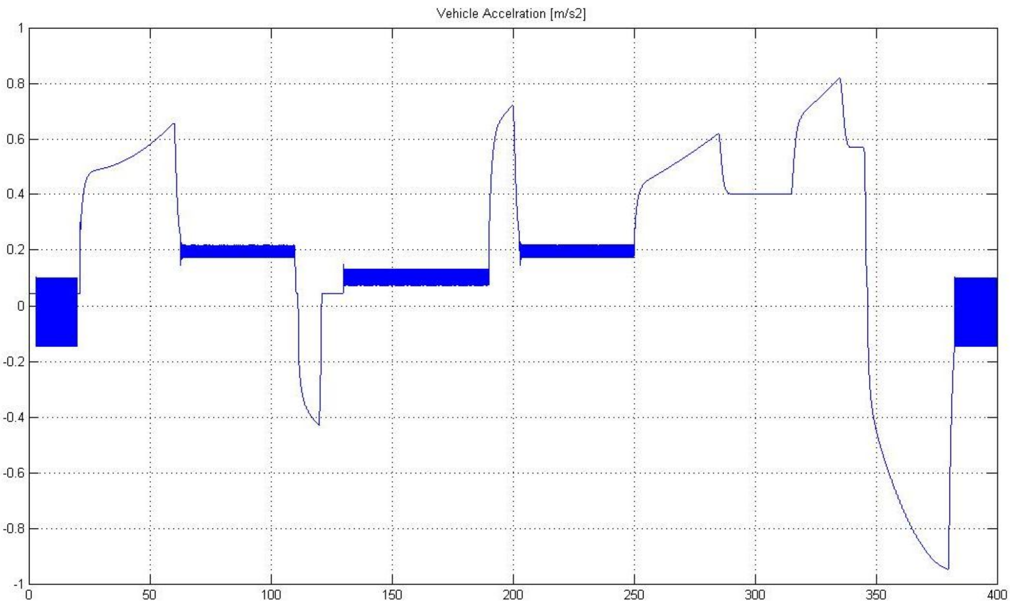

Figure 5.31 Electric Vehicle Acceleration

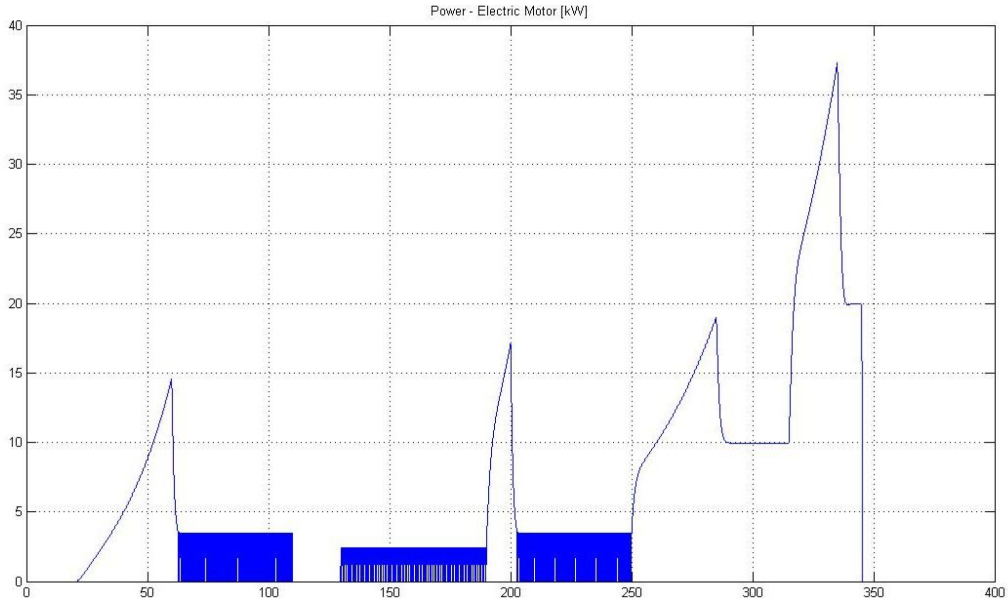

Figure 5.32 Power of EV

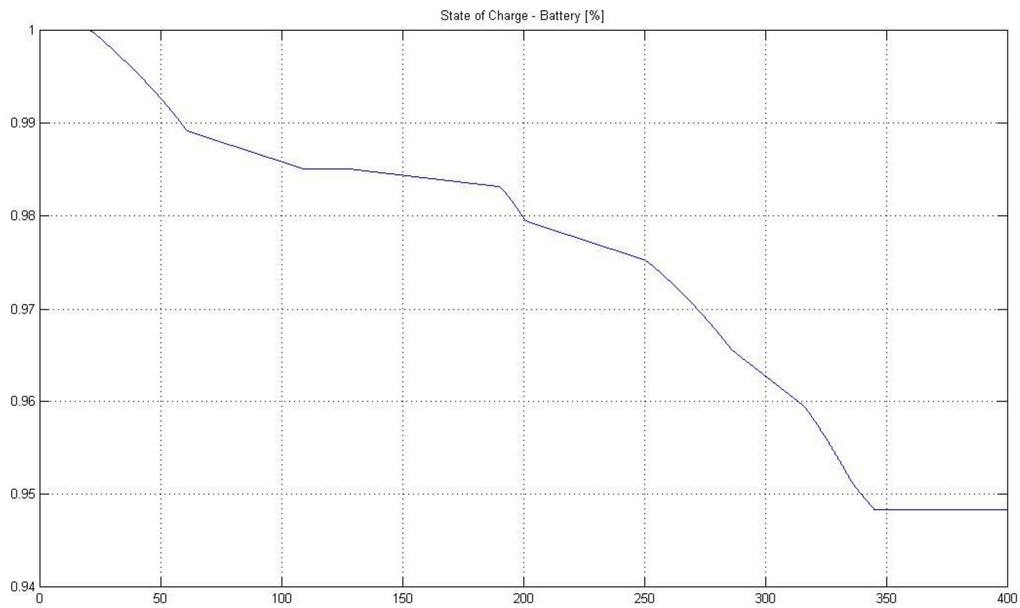

Figure 5.33 State of Charge (SOC)

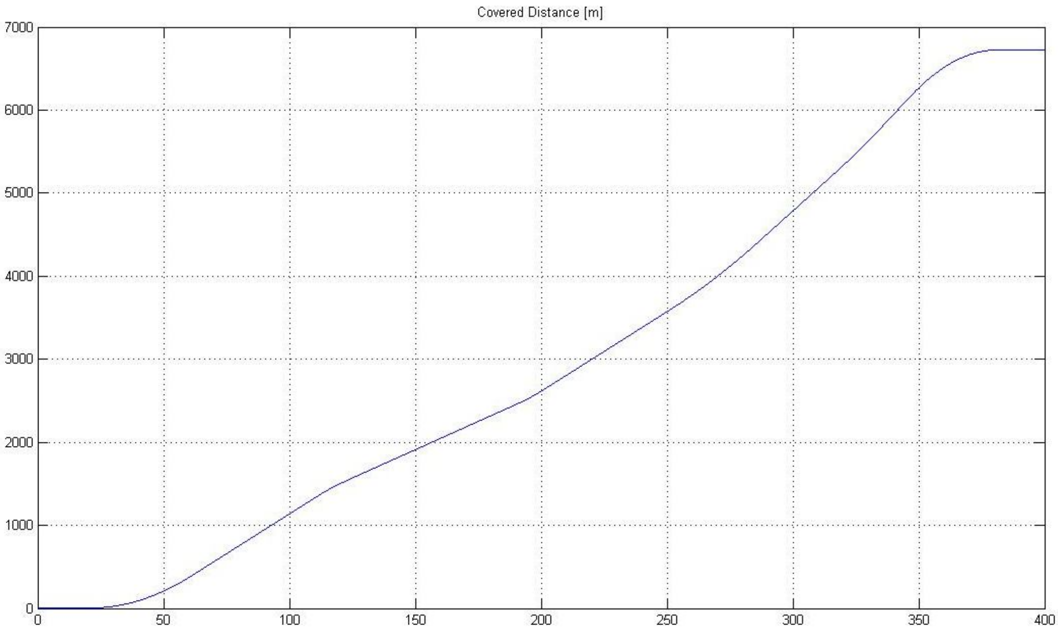

Figure 5.34 Covered Distance

5.1 Energy Consumption

For calculating the costs, current diesel prices were taken to be 3.25 TL/lt and electricity prices of Tedas were taken to be 0.21659 TL/kWh.

Internal Combustion Engine Vehicle

The ICE vehicle is consumed 0.11 lt at city driving cycle and 0.09 lt at freeway driving cycle. The fuel consumption of the ICE vehicle per 100km is about 11.1 lt/100km for city driving cycle and 9,2 lt/100km for freeway driving cycle. And, the costs of consumption for city and freeway driving cycles are in order 36 TL/100km and 29,5 TL/100km.

Electric Vehicle

The battery capacity of electric vehicle is 250 amph, and the energy was taken out of the battery is 0.58 kWh at city driving cycle and 0.44 kWh at freeway driving cycle. The EV energy consumption per 100 km is about 58.1 kWh/100km at city driving cycle and 43.5 kWh/100km at freeway driving cycle. And, the costs are about 12,5 TL/100km for city driving cycle and 9,5 TL/100km for the freeway driving cycle.

$$Energy\ consumption = Q_0 * Voltage * 10^{-3} \quad [kWh] \tag{5.1}$$

5.2 Range

Internal Combustion Engine Vehicle

When a fuel tank of 80 L is considered, the range of the ICE will be about 718 km for city driving cycle and about 875 km for freeway driving cycle.

Full Throttle Position

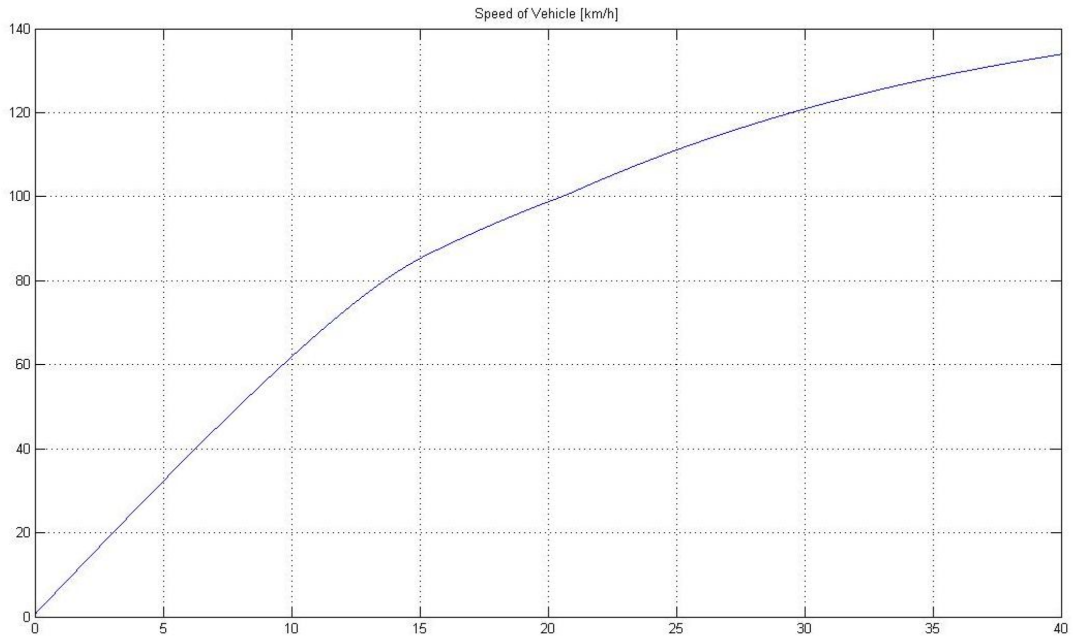

Figure 5.35 ICE Speed at Full Throttle Position

Maximum speed and acceleration performaces of the real vehicle are presented below.

Vmax = 154		(kph)
t1 = 2.2 ;	8-32.3 kph	(s)
t2 = 6.1 ;	8-56.4 kph	(s)
t3 = 20 ;	0-100 kph	(s)
t4 = 26 ;	80-120 kph	(s)
t5 = 22 ;	0-1/4mi	(s)

Our ICE vehicle that was simulated has maximum speed and performances as presented below.

Vmax = 170 (kph)
t1 = 3.8 ; 8-32.3 kph (s)
t2 = 7,8 ; 8-56.4 kph (s)
t3 = 20,5 ; 0-100 kph (s)
t4 = 15,8 ; 80-120 kph (s)
t5 = 9,7 ; 0-1/4mi (s)

Electric vehicle

Our electric vehicle battery is 30 kWh. Thus, the range of the EV will be about 52 km for city driving cycle and the range will be about 70 km for freeway driving cycle.

Full Throttle Position

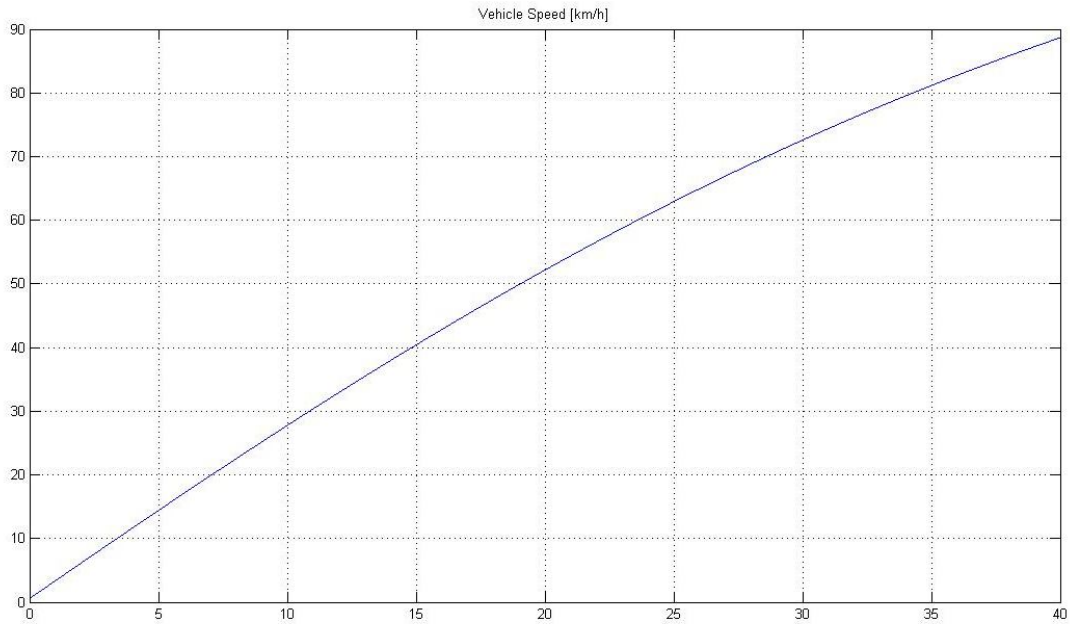

Figure 5.36 EV Speed at Full Throttle Position

Real vehicle maximum speed and acceleration performaces are;

Vmax = 154 (kph)

t1 = 2.2 ; 8-32.3 kph (s)

t2 = 6.1 ; 8-56.4 kph (s)

t3 = 20 ; 0-100 kph (s)

t4 = 26 ; 80-120 kph (s)

t5 = 22 ; 0-1/4mi (s)

Our ICE vehicle that was simulated has maximum speed and performances as presented below.

Vmax = 140 (kph)

t1 = 8,9 ; 8-32.3 kph (s)

t2 = 19,1 ; 8-56.4 kph (s)

t3 = 48 ; 0-100 kph (s)

t4 = 28,7 ; 80-120 kph (s)

t5 = 23,6 ; 0-1/4mi (s)

Evaluation of Simulation Results

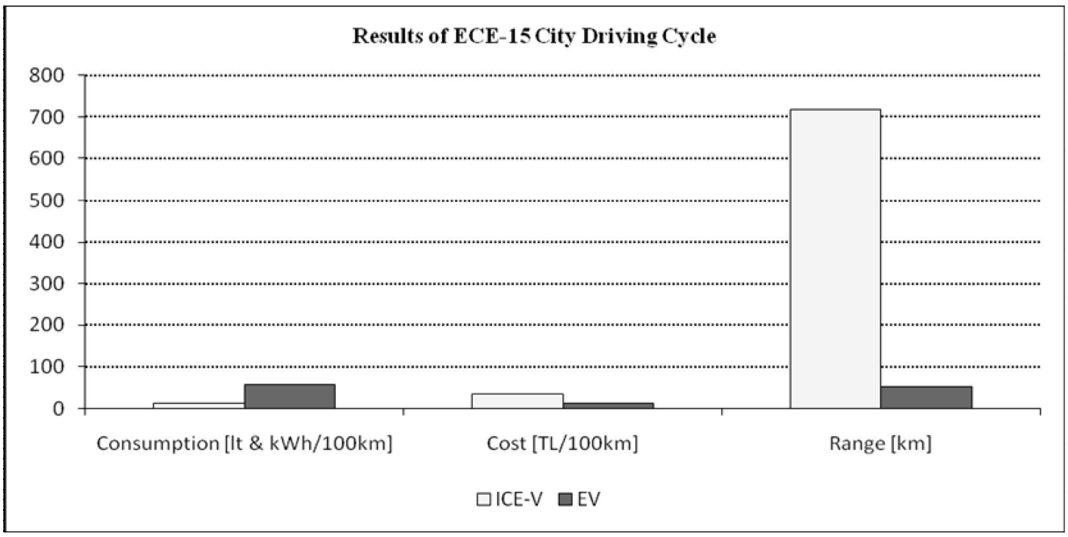

Figure 5.37 ICE & EV Simulation Results at ECE-15

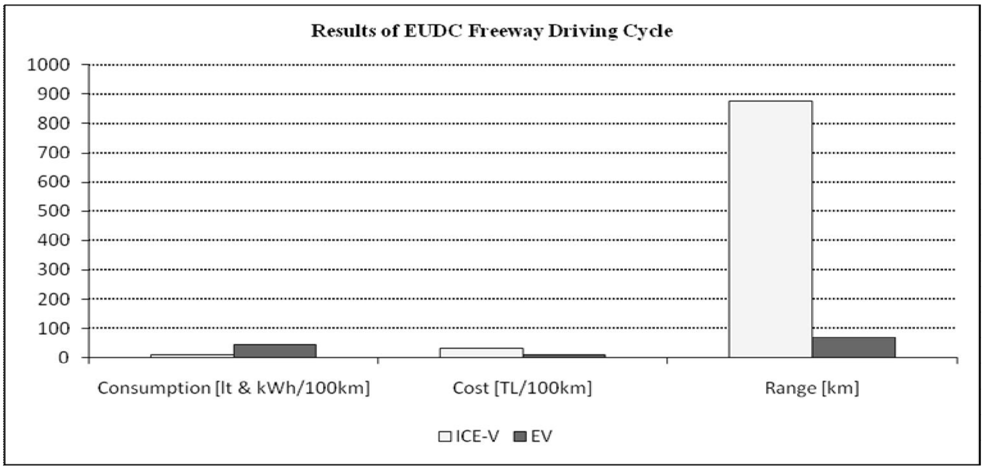

Figure 5.38 ICE & EV Simulation Results at EUDC

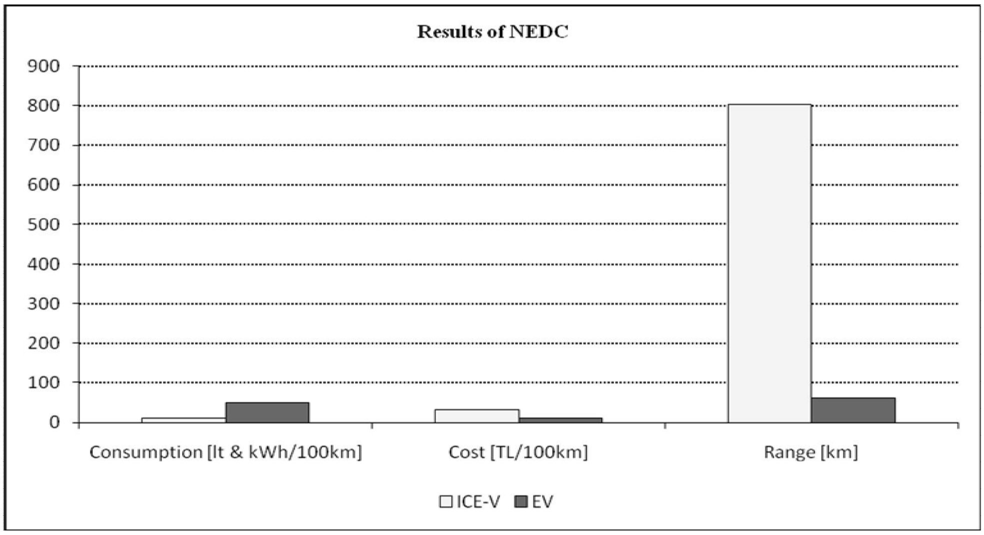

Figure 5.39 ICE & EV Simulation Results at NEDC

Table 5.1 : ICE Vehicle & EV Simulation Results

	IYM-T			E-T		
	Fuel Consumption	Cost	Range	Energy Consumption	Cost	Range
ECE-15	11.1 l/100km	36.1 TL/100km	718km	58.1 kWh/100km	12.6 TL/100km	52km
EUDC	9.2 l/100km	29.9 TL/100km	875km	43.5 kWh/100km	9.4 TL/100km	68km
NEDC	9.9 l/100km	32.2 TL/100km	804km	50 kWh/100km	10.8 TL/100km	61km

6. CONCLUSION AND RECOMMENDATIONS

In this chapter, we discussed in conclusion the energy consumed by electric and internal combustion engine vehicles and their range. Moreover we evaluated electric vehicles and made recommendations with regard to market share, emission, price, battery, range, utilization, user habits and their demands and infrastructure needs:

The EV has a range of 300km, chosen by us because it is difficult to carry the energy for long trips in batteries, but more than sufficient for home to work commuting.
All the systems associated with electric vehicles need to be worked where they are the most efficient in areas. Therefore, algorithms can be generated to work them in the most efficient areas, if requested efficiencies of battery and electric motor obtains.

Market share of electric vehicles are expected to be among 3% and 10% between the years of 2020 and 2025 according to statistics published by ACEA. In 2015; in Europe, 480 thousand electric vehicles; in the USA, 1 million electric vehicles and in the world, 1.7 million electric vehicle circulations are planned in accordance with incentives allocated to purchase of electric vehicles and city projects were generated upon infrastructure investments.

In terms of emissions, electric vehicles have advantages with low rumbling noise levels and zero emission sights. And the prices of electric vehicles are high because of the infrastructure needs and especially their initial purchase costs arising from high battery cost are also high.

The range that electric vehicles can reach with today’s battery technology are limited by 150 – 200 km. Electric vehicles are convenient for the utilization of city driving. This can be demonstrated as a disadvantage of electric vehicles. The technology has to be improved for efficient and economic battery to solve this case. Besides; Service life of batteries is not at requested levels. So; battery parameters kind of temperature, charge time and current depth of battery are managed very well to keep battery life longer.

Widespread of electric vehicles utilization and entry electric vehicles to market are not expected to be spontaneously because of causing significant changes in user habits, infrastructure needs and especially high initial purchase costs arising from high battery cost.

First change in user habits is about to limited range electric vehicles can reach. On the other hand, very low rumbling noise levels and regenerative braking systems that electric vehicles have cause different perception on users and this provides to increase user demands which are seen as the most important factor in the development of electric vehicles market. Furthermore, fast charging (filling) stations users can reach easily to recharge batteries and service stations have been need, which are named as infrastructure requirements of user demands.

This thesis is going to be expanded by lateral dynamics and CO_2emissions. Electric vehicles will be preferred predominantly in the next decade by the improvements at electric vehicle technology. Electric vehicles need to be simulated since the electric vehicles are inevitable. Thus developments about thechnology of electirc vehicles can be become cheaper and steped up. Moreover automotive monopoly may be changed by coming out of electric vehicles on the market and accordingly a chance will regenerated to prodece a turkish made car for the pruposes of turkey.

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APPENDICES

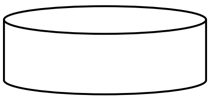
APPENDIX A.1

This shape indicates start and stop points of flow chart.

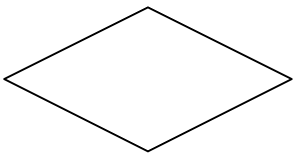
This shape indicates a command of loop, which a process to be renewed for a certain number is provided in an algorithm. Start, step and stop variables of the loop or terms can be specified in the shape. Commands desired to be repeated are appear in the shape.

This shape provides an arithmetic operation to be done in the algorithm. The process desired to be done is written into the shape. It has only input and output parts.

This shape designates input and output commands where to be used. Which variable needs to be read and write into the shape has to be designated in the algorithm.



This shape indicates to read by Disc or write into the disk.



This shape provides to make a decision in an algorithm or either of two options to be implemented according to this decision. Logical term to be controlled is written into the shape. If the term is right during the programme flow, the programme stray to yes part or it stray to no part if the term is wrong. It is a shape has simple input and output parts.

Figure A.1 : Meaning of Figures In the Flowchart.

APPENDIX A.2

Table A.1 : Coefficients for Longitudinal Force

Coefficient	Value	Units
b_0	1.57	Dimensionless
b_1	-48	1/Meganewton
b_2	1005.6	1/kN
b_3	6.8	1/Meganewton
b_4	444	1/kN
b_5	0	1/kN
b_6	0.0034	$1/(kN)^2$
b_7	-0.008	1/kN
b_8	0.66	Dimensionless
b_9	0	1/kN
b_{10}	0	Dimensionless
b_{11}	0	N/kN
b_{12}	0	N

Table A.2 : Conventions For Naming Variables

Variable Names	Abbre viation	Units	Description
Normal Load	F_z	kN	Positive when the tire is penetrating the road
Longitudinal Force	F_x	N	Positive during traction, Negative during braking
Longitudinal Slip	S	%	Negative in braking (100% wheel lock) Positive in traction

Table A.3 : Parameters For Longitudinal Force

Parameters	Description
B	Stifness Factor
C	Shape Factor
D	Peak Factor
E	Curvature Factor
S_h	Horizontal Shift
S_v	Vertical Shift

APPENDIX A.3

Table A.4 : Technical Data of Vehicle.

Symbol	Unit	Description
m	kg	Vehicle mass
g	m/s2	Gravitational acceleration
G	N	Vehicle Weight
m_R	kg	Wheel mass
G_R	N	Wheel Weight
R	m	Fictive radius
R_0	m	Dynamic wheel radius
r	m	Static wheel radius
S	-	Slip
U	m	Rolling circumference
f_R	-	Rolling resistance coefficient
J_R	kgm2	Inertia torque of the wheel
J_m	kgm2	ICE Inertia Torque
J_F	kgm2	Flywheel Inertia Torque
μ	-	Adhesion
φ_R	0	Angle of the wheel
$\dot{\varphi}_R$	1/s	Angular velocity of the wheel
$\ddot{\varphi}_R$	1/s2	Acceleration of the wheel
M_R	Nm	Moving or braking torque
φ	0	Angle of drive wheel
φ_0	0	Angle of free-rolling wheel
$\dot{\varphi}$	1/s	Angular velocity of drive wheel
$\dot{\varphi}_0$	1/s	Angular velocity of free-rolling wheel
x	m	Position
$\dot{x}$	m/s	Velocity of vehicle
$\ddot{x}$	m/s2	Acceleration of vehicle – x

Symbol	Unit	Description
$\ddot{z}$	m/s2	Acceleration of vehicle - z
ρ	kg/m3	Density
A	m2	Frontal area
C_d	-	Drag coefficient
a	m/s2	Vehicle Acceleration
V	m/s	Velocity
λ	-	Slope of angle
α	0	Impact factor
F_L	N	Aerodynamic drag
F_R	N	Rolling Resistance Force
n_m	rpm	Engine or motor speed
M_m	Nm	Engine Torque
η_{EV}	%	Electric motor efficiency
ω_m	rad/s	Angular velocity of EV motor
F_Z	N	Vertical wheel load
F_B	N	Acceleration force
F_{st}	N	Hill climbing wall Force
F_{te}	N	Tractive effort
F_X	N	Longitudinal Force
F_y	N	Lateral Force
M_Z	Nm	Self-Aligning Torque
i_g	-	Gear ratio
i_a	-	Differential ratio
i_k	-	Total cycle ratio
η_K	%	Total efficiency
M_{Brake}	Nm	Brake torque
M_{Drive}	Nm	Drive torque
$k_{Brake\ System}$	-	Brake Coefficient
F_p	N	Brake pedal force

Table A.5 : Technical Data of Battery.

Symbol	Unit	Description
Q_0	amps	Battery Nominal Capacity
Q_t	amp	Remaining charge
SOC	-	State of charge
V_0	volts	Voltage
R	miliohms	Battery Internal Resistance
V_{cell}	volts	Battery Voltage
η_{cell}	-	Efficiency of battery
I	amp	Current
P_3	kW	Battery internal power
P_{cell}	kW	Battery terminal power

APPENDIX A.4: Matlab Codes of Electric Vehicle for NEDC

```
% ORAL VATAN, oralvatan@gmail.com
% 20 ARALIK 2010 PAZARTESI
% % MODELLING AND SIMULATION OF LONGITUDINAL DYNAMICS OF
ELECTRIC VEHICLES
% % ELEKTRIKLI BIR TASITIN DOGRUSAL DINAMIGININ
MODELLENMESI VE SIMULASYONU
% % ELECTRIC VEHICLE SIMULATION - NEDC CYCLE

clc;
clear all;
close all;

npoints = 1200001;
dt = 0.001; % time step (sn.)

i=1.0;
k_motor = 1.2;

tic

% Parameters;
b0=1.57;   b1=-48;    b2=1005.6;     b3=6.8;    b4=444;    b5=0;   b6=0.0034;
b7=-0.008;  b8=0.66;   b9=0;          b10=0;     b11=0;     b12=0;

% VEHICLE
m = 1612;           % (kg)
ro = 1.226;         % (kg/m3)
A = 3.6;            % (m2)
cd = 0.394;
lamda = 1;

Vmax = 154;        % (kph)
t_ip1 = 2.2;       % 8-32.3 km/h    (s)
t_ip2 = 6.1;       % 8-56.4 km/h    (s)
t_ip3 = 20;        % 0-100  km/h    (s)
t_ip4 = 26;        % 80-120 km/h
t_dkp = 22;         % 0-1/4mi  (sn.)

mR = 17;            % (kg)
g=9.81;             % (m/s2)
G=m*g;              % (N)

% TIRES
```

```
% 215/75 R16
R=0.364;      % (m)
Ro=0.358;    % (m)
r=0.351;      % (m)
fR=0.011;
Jr=1.283;     % [kgm2]

Fz = 3.14;  %  (kN)

% Pacejka Magic Formula
C=b0; % shape factor
mb=b1*Fz+b2;
D=mb*Fz; % peak value
BCD=(b3*Fz^2+b4*Fz)*exp(-b5*Fz);
B=BCD/(C*D); % stiffness factor
E=b6*Fz^2+b7*Fz+b8; % curvature factor
sh=b9*Fz+b10; % Horizontal shift
sv=0; % Vertical shift

t = zeros(npoints,1); % (s)
t(1) = 0; % (s)

% VEHICLE
a_tasit = zeros(npoints,1);
v = zeros(npoints,1);
s = zeros(npoints,1);
vsetdegeri = zeros(npoints,1);
FL = zeros(npoints,1);
a_tasit(1) = 0.0; % (m/s2)
v(1) = 0.15; % (m/s)
v_kmh(1) = v(1) * 3.6; % (kph)
s(1) = 0;
vsetdegeri(1) = 0;
FL(1) = 0;

% TIRE
S = zeros(npoints,1);
Fx = zeros(npoints,1);
psi_ikinokta =zeros(npoints,1);
psi_birnokta = zeros(npoints,1);
FR = zeros(npoints,1);
psi_birnokta(1) = v(1)/R; % (1/s)
S(1) = (1-v(1)/(psi_birnokta(1)*Ro));
psi_ikinokta(1) = 0;
Fx(1) = 0;
FR(1)=0;

% ENGINE
nm = zeros(npoints,1);
electricmotormoment = zeros(npoints,1);
```

```
Pm = zeros(npoints,1);
elektricmotorverim = zeros(npoints,1);
nm(1) = 0; %(dev/dak)
electricmotormoment(1) = 0; % (Nm)
Pm(1) = 0; % (kW)
elektricmotorverim(1)= 0;   % [%]

% BRAKE SYSTEM
Mb = zeros(npoints,1);
Mb(1) = 0;  % (Nm)

% DRIVER
akim = zeros(npoints,1);
Fp = zeros(npoints,1);
akim(1) = 0;  % [amp]
Fp(1)= 1; % [N]

% BATTERY
soc = zeros(npoints,1);
Vo_soc = zeros(npoints,1);
R_soc = zeros(npoints,1);
Vcell = zeros(npoints,1);
bataryaverim = zeros(npoints,1);
enerjisarfiyati_kWh = zeros(npoints,1);
soc(1) = 1;                 % [%]
Vo_soc(1) = 120;       % [volts]
R_soc(1) = 320;         % [milliohms]
Vcell(1) = 120;           % [volts]
bataryaverim(1)= 0.9;       % [%]
enerjisarfiyati_kWh(1) = 0;
nominalkapasite = 250;       % [Amph]

% ELECTRIC MOTOR MAP
load ElektrikMotoruHaritasi_buildyourownelectricvehicle

% NEDC
load NEDC_avrupacevrimiharitasi_0001zamanadimli_1184sn

% P CONTROL SYSTEM
  P_gaz = 100;
  P_fren = 2;

% NEDC
  vsetdegeri(1:npoints) = hiz(1:npoints);

for adim=1:npoints-1

%    NEDC
     % Driver
       %
```

```
        if v_kmh(adim) > vsetdegeri(adim)
           akim(adim+1) = 0;
           Fp(adim+1) = P_fren * (-vsetdegeri(adim) + v_kmh(adim));
        end

           if v_kmh(adim) < vsetdegeri(adim)
              akim(adim+1) = P_gaz * (vsetdegeri(adim) - v_kmh(adim)) ;
              Fp(adim+1) = 0 ;
           end

              if abs((v_kmh(adim) - vsetdegeri(adim)))< 1
                 akim(adim+1) = 0;
                 Fp(adim+1) = 0;
              end

        if akim(adim+1) < 0
              akim(adim+1) = 0;
        end

           if akim(adim+1) > 440
             akim(adim+1) = 440;
           end

           if Fp(adim+1) < 0
             Fp(adim+1) = 0;
           end

              if Fp(adim+1) > 100
                Fp(adim+1) = 100;
              end

% Longitudinal Force (N)
   Fx(adim+1) = D*sin(C*atan(B*(S(adim)+sh)-E*(B*(S(adim)+sh)-
atan(B*(S(adim)+sh)))))+sv;

% Rolling Resistance Force
        if v(adim)>0
          FR(adim+1) = -1*(Fz*1000*fR);
        else
          FR(adim+1) = 0;
        end

     psi_ikinokta = ((electricmotormoment*i)/r-Mb(adim)/r-Fx-FR(adim))*r/Jr; %
(1/s2)
     psi_birnokta(adim+1) = psi_birnokta(adim) + dt*psi_ikinokta(adim); % (1/s)

     FL(adim+1)=0.5*ro*cd*A*v(adim)*v(adim); % (kgm/s2 = N)
     a_tasit=(2*Fx-(2*Mb/r)-FL(adim+1))/m; % (m/s2)
     v(adim+1) = v(adim) + dt*a_tasit(adim); % (m/s)
     s(adim+1) = s(adim) + dt*v(adim);
```

```
    v_kmh = v.*3.6; % (km/h)

    nm(adim+1) = ((30*psi_birnokta(adim+1))/pi)*i;
      if nm(adim+1)>=7750
        nm(adim+1) = 7750;
      end

    electricmotormoment(adim+1) =
interp2(devirsayisi3,akim3,moment3,nm(adim+1),akim(adim+1)); % (Nm)
    electricmotormoment(adim+1) = electricmotormoment(adim+1) * k_motor;
  % (kW)
    Pm(adim+1) = (electricmotormoment(adim+1)*((2*pi*nm(adim+1))/60))/1000;
% (kW)
    elektricmotorverim(adim+1)
=1/(0.1+(Vcell(adim+1))*akim(adim+1))/((electricmotormoment(adim+1))*(2*pi*n
m(adim+1)/60));

  % Brake System
    Mb(adim+1) = 15.0 * Fp(adim);


% Slip
    S(adim+1) = (1-v(adim+1)/(psi_birnokta(adim+1)*Ro));
% Battery
    Vo_soc(adim+1) = 108.5138 + 21.645*soc(adim) - 10.1587*(soc(adim))^2;
      if soc(adim) <= 0.4
        R_soc(adim+1) = 382.34 - 234.1642*soc(adim) + 195.7467*(soc(adim))^2;
      elseif soc(adim) > 0.4
        R_soc(adim+1) = 320;
      end

    Vcell(adim+1) = Vo_soc(adim+1) - akim(adim+1)*R_soc(adim+1)*0.001;
    soc(adim+1)= soc(adim)-(akim(adim+1)/(nominalkapasite*3600))*dt;

      if soc(adim+1) <0
        soc (adim+1)=0;
      end
   t(adim+1) = t(adim) + dt;
end


% Evaluation
enerjisarfiyati_kWh(npoints) = nominalkapasite*(1-
soc(npoints))*(Vo_soc(npoints))*(10^-3); % [kWh]
tuketim_100km = (100/0.97)* enerjisarfiyati_kWh(npoints);  % [kWh]
maliyet_TL = tuketim_100km * 0.21659;  % [TL]
range_km = (30*100) / tuketim_100km;  % [km]

toc
```

APPENDIX A.5: Matlab Codes of ICE Vehicle for NEDC

```
% ORAL VATAN, oralvatan@gmail.com
% 20 ARALIK 2010 PAZARTESI
% % MODELLING AND SIMULATION OF LONGITUDINAL DYNAMICS OF
ELECTRIC VEHICLES
% % ELEKTRIKLI BIR TASITIN DOGRUSAL DINAMIGININ
MODELLENMESI VE SIMULASYONU
% % ELECTRIC VEHICLE SIMULATION - NEDC CYCLE

clc;
clear all;
close all;

npoints = 1200001;
dt = 0.001; % time step (sn.)

tic

% Parameters;
b0=1.57;   b1=-48;    b2=1005.6;    b3=6.8;    b4=444;    b5=0;   b6=0.0034;
b7=-0.008;  b8=0.66;    b9=0;          b10=0;      b11=0;      b12=0;

% VEHICLE
m = 1612;          % (kg)
ro = 1.226;        % (kg/m3)
A = 3.6;           % (m2)
cd = 0.394;
lamda = 1;

Vmax = 154;       % (kph)
t_ip1 = 2.2;       % 8-32.3 km/h    (s)
t_ip2 = 6.1;       % 8-56.4 km/h    (s)
t_ip3 = 20;        % 0-100  km/h    (s)
t_ip4 = 26;        % 80-120 km/h at 4.23 of axle gear ratio  (s)
t_dkp = 22;         % 0-1/4mi  (sn.)
V_tank = 80;        % (lt.)

mR = 17;           %  (kg)
g=9.81;            %  (m/s2)
G=m*g;             %  (N)

% ICE VEHICLE
% teorik yakit tüketim degerleri (NEDC Urban)
```

```
% ortalama yakittuketimi = 8.9;          % (lt.)
% sehirici yakit tuketimi = 10.5;        % (lt.)
% sehirdisi yakit tuketimi = 7.6;        % (lt.)

% TIRES
% 215/75 R16
R=0.364;      % (m)
Ro=0.358;    % (m)
r=0.351;      % (m)
fR=0.011;
Jr=1.283;    % [kgm2]

 % Gear Box & Differential
% Gear Ratios:
i1=3.8;
i2=2.136;
i3=1.345;
i4=0.921;
i5=0.674;
iR=-3.727;
ia=4.23;
% Total Gear Efficiency
nk1=0.874;
nk2=0.8835;
nk3=0.893;
nk4=0.9025;
nk5=0.9120;
% Total Gear Ratios
ik_1=i1*ia;
ik_2=i2*ia;
ik_3=i3*ia;
ik_4=i4*ia;
ik_5=i5*ia;

% Vertical Wheel Load
Fz = 3.14;  % (kN)
% Pacejka Magic Formula
C=b0; % shape factor
mb=b1*Fz+b2;
D=mb*Fz; % peak value
BCD=(b3*Fz^2+b4*Fz)*exp(-b5*Fz);
B=BCD/(C*D); % stiffness factor
E=b6*Fz^2+b7*Fz+b8; % curvature factor
sh=b9*Fz+b10; % Horizontal shift
sv=0; % Vertical shift

t = zeros(npoints,1);
t(1) = 0; % (s)

% VEHICLE
```

```
a_tasit = zeros(npoints,1);
v = zeros(npoints,1);
s = zeros(npoints,1);
vsetdegeri = zeros(npoints,1);
FL = zeros(npoints,1);
a_tasit(1) = 0.0; % (m/s2)
v(1) = 0.15; % (m/s)
v_kmh(1) = v(1) * 3.6; % (kph)
s(1) = 0;
vsetdegeri(1) = 0;
FL(1) = 0;

% TIRE
S = zeros(npoints,1);
Fx = zeros(npoints,1);
psi_ikinokta =zeros(npoints,1);
psi_birnokta = zeros(npoints,1);
FR = zeros(npoints,1);
psi_birnokta(1) = v(1)/R; % (1/s)
S(1) = (1-v(1)/(psi_birnokta(1)*Ro));
psi_ikinokta(1) = 0;
Fx(1) = 0;
FR(1)=0;

% ENGINE
nm = zeros(npoints,1);
Mm = zeros(npoints,1);
tuketim = zeros(npoints,1);
tuketim_  = zeros(npoints,1);
tuketim_kumulatif= zeros(npoints,1);
Pm = zeros(npoints,1);
nm(1) = 0; %(dev/dak)
Mm(1)= 0; % (Nm)
tuketim(1) = 0; % (lt/s)
tuketim_(1) = 0;
tuketim_kumulatif(1)=0;
Pm(1) = 80; % (kW)

% Brake System
Mb = zeros(npoints,1);
Mb(1) = 0;  % fren momenti (Nm)

% Gear Box
vites_cevrim_orani = zeros(npoints,1);
vites_konumu = zeros(npoints,1);
vites_cevrim_orani(1)=16.074;
vites_konumu(1) = 1.0;

% Driver
gaz = zeros(npoints,1);
```

```
Fp = zeros(npoints,1);
gaz(1) = 0;   % gaz [%]
Fp(1)= 1;  % fren kuvveti [N]

% ICE MAP
load FORD_ictenyanmalimotorharitasi

% ICE Fuel Consumption Map
load FORD_ictenyanmalimotoryakittuketimiharitasi

% NEDC
load NEDC_avrupacevrimiharitasi_0001zamanadimli_1184sn

% P CONTROL SYSTEM
  P_gaz = 2;
  P_fren = 2;

% NEDC
  vsetdegeri(1:npoints) = hiz(1:npoints);

for adim=1:npoints-1

%    NEDC
     % Driver
        if v_kmh(adim) > vsetdegeri(adim)
            gaz(adim+1) = 0;
            Fp(adim+1) = P_fren * (-vsetdegeri(adim) + v_kmh(adim));
        end
            if v_kmh(adim) < vsetdegeri(adim)
                gaz(adim+1) = P_gaz * (vsetdegeri(adim) - v_kmh(adim)) ;
                Fp(adim+1) = 0 ;
            end
                if abs((v_kmh(adim) - vsetdegeri(adim)))< 1
                   gaz(adim+1) = 0;
                   Fp(adim+1) = 0;
                end
                    if 0.1 < Fp(adim+1) && Fp(adim+1) < 0.5
                       Fp(adim+1) = 0;
                    end

        if gaz(adim+1) < 0
           gaz(adim+1) = 0;
        end
            if gaz(adim+1) > 100
               gaz(adim+1) = 100;
            end
                if Fp(adim+1) < 0
                   Fp(adim+1) = 0;
                end
                    if Fp(adim+1) > 100
```

```
                Fp(adim+1) = 100;
              end

% Longitudinal Force (N)
    Fx(adim+1) = D*sin(C*atan(B*(S(adim)+sh)-E*(B*(S(adim)+sh)-
atan(B*(S(adim)+sh)))))+sv;

   % Gear Shifting
       if  v_kmh(adim)< 16
       vites_cevrim_orani(adim+1) = ik_1; nk=nk1;
       end

       if 16 < v_kmh(adim) && v_kmh(adim) < 32
       vites_cevrim_orani(adim+1) = ik_2; nk=nk2;
       end

       if 32 < v_kmh(adim) && v_kmh(adim) < 50
       vites_cevrim_orani(adim+1) = ik_3; nk=nk3;
       end

       if 50 < v_kmh(adim) && v_kmh(adim) < 100
       vites_cevrim_orani(adim+1) = ik_4; nk=nk4;
       end

       if v_kmh(adim) > 100
       vites_cevrim_orani(adim+1) = ik_5; nk=nk5;
       end

% Rolling Resistance Force
   if v(adim)>0
     FR(adim+1) = -1*(Fz*1000*fR);
   else
     FR(adim+1) = 0;
   end

    psi_ikinokta = ((Mm*vites_cevrim_orani(adim)*nk)/r-Mb(adim)/r-Fx-
FR(adim))*r/Jr; % (1/s2)
    psi_birnokta(adim+1) = psi_birnokta(adim) + dt*psi_ikinokta(adim); % (1/s)

    FL(adim+1)=0.5*ro*cd*A*v(adim)*v(adim); % (kgm/s2 = N)
    a_tasit=(2*Fx-(2*Mb/r)-FL(adim+1))/m; % (m/s2)
    v(adim+1) = v(adim) + dt*a_tasit(adim); % (m/s)
    s(adim+1) = s(adim) + dt*v(adim);
    v_kmh = v.*3.6; % (km/h)

    nm(adim+1) = ((30*psi_birnokta(adim+1))/pi)*vites_cevrim_orani(adim+1);
    if nm(adim+1)>= 6400
      nm(adim+1) = 6400;
    end
       if nm(adim+1)<= 750
```

```
        nm(adim+1) = 750;
      end

    Mm(adim+1) =
interp2(devirsayisi,gazkelebegiacikligi,moment,nm(adim+1),gaz(adim+1)); % (Nm)

    tuketim(adim+1) =
interp2(moment2,devirsayisi2,yakittuketimi2,Mm(adim+1),nm(adim+1)); % (lt/s)
    tuketim_(adim+1) = tuketim(adim+1)*25;
      if tuketim(adim+1)<0
        tuketim(adim+1)=0;
      end
    tuketim_kumulatif (adim+1)= tuketim_kumulatif(adim)+tuketim(adim+1)*dt; %
dt

   % Brake System
    Mb(adim+1) = 15.0 * Fp(adim);
   % Slip
    S(adim+1) = (1-v(adim+1)/(psi_birnokta(adim+1)*Ro));

    t(adim+1) = t(adim) + dt; % zaman(s)

end

% Evaluation

tuketim_100km = (100/0.97)*tuketim_kumulatif(npoints);  % [lt]
maliyet_TL = tuketim_100km * 3.25;  % [TL]
range_km = (80*100) / tuketim_100km;  % [km]
```

CURRICULUM VITAE

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