

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE
ENGINEERING AND TECHNOLOGY

**DESIGN AND SIMULATION OF A STOP AND GO
CONTROL SYSTEM FOR ROAD VEHICLES**

M.Sc. THESIS

Özgür SARIGÜL

Department of Mechanical Engineering

System Dynamics and Control Programme

JUNE 2012

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Thesis Advisor: Assoc. Prof. Dr. Bilin AKSUN GÜVENÇ

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

**YOL ARAÇLARI İÇİN DUR VE KALK
KONTROL SİSTEMİ TASARIMI VE SİMÜLASYONU**

YÜKSEK LİSANS TEZİ

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To my spouse and family,

FOREWORD

Traffic congestion is getting heavier day to day in urban areas, and automotive industry has been developing various driver assistance systems like Lane Keeping and Distance Keeping. Stop and Go Control System is one of the most important driver assistance systems, which is aimed to help drivers in urban traffic by undertaking throttle and brake responsibilities in following preceding vehicles. More efficient traffic flow is also provided with regulating follow speed and follow distance. Further, it has been accepted among safety and comfort systems, and it has been trying to be commercialized by most of the automotive companies. In this thesis, a Stop and Go Control System is designed and simulated in MATLAB/Simulink environment using a Single Track Nonlinear Vehicle Model. Simulation results obtained and presented for two different types of control strategies and various driving scenarios.

First, I would like to express my special thanks to my advisor Assoc. Prof. Dr. Bilin AKSUN GÜVENÇ for her guidance, encouragement, patience, and insight throughout my thesis study.

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May 2012

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ABBREVIATIONS

DAS	: Driver Assistance Systems
CC	: Cruise Control
ACC	: Adaptive Cruise Control
S&G	: Stop and Go
PD	: Proportional Derivative
PI	: Proportional Integral
PID	: Proportional Integral Derivative
NASS	: National Accident Sampling System
GES	: General Estimates System
ITARDA	: Institute for Traffic Acccident Research and Data Analysis
GIDAS	: Grafisch-Interaktives Daten Analyse System
CTH	: Constant Time Headway
VTH	: Variable Time Headway
LQ	: Lineaer Quadratic
MMC	: Model Matching Control
LMS	: Least Mean Square
SMC	: Sliding Mode Control
MPC	: Model Predictive Control
LIDAR	: Light Detection and Ranging
RADAR	: Radio Detection and Ranging
CG	: Center of Gravity
SAE	: Society of Automotive Engineering
RPM	: Revolution per Minute

LIST OF SYMBOLS

V_{target}	: Target vehicle speed
V_{host}	: Host vehicle speed
d	: Relative distance between vehicles
V_r	: Relative velocity
d_{desired}	: Desired distance
T_h	: Headway time
ψ	: Yaw angle
θ	: Pitch angle
ϕ	: Roll angle
XOY	: Local coordinate system
X_oOY_o	: Global coordinate system
F_x	: Tire force in longitudinal direction
F_y	: Tire force in lateral direction
F_{xf}, F_{xr}	: Front and rear tire forces in longitudinal direction
F_{yf}, F_{yr}	: Front and rear tire forces in lateral direction
M_z	: Moment around vertical -z axis
I_z	: Mass moment of inertia with respect to vertical axis -z through CG
m	: Vehicle mass
a	: Vehicle acceleration
a_x, a_y	: Vehicle acceleration in longitudinal and lateral directions
V	: Vehicle velocity at center of gravity
V_f, V_r	: Front and rear vehicle velocities
V_{fx}, V_{rx}	: Front and rear vehicle velocities in longitudinal direction
V_{fy}, V_{ry}	: Front and rear vehicle velocities in lateral direction
V_{tf}, V_{tr}	: Front and rear linear tire velocities
α_f, α_r	: Front and rear tire sideslip angles
β	: Chassis sideslip angle
β_f, β_r	: Front and rear chassis sideslip angles
δ_f, δ_r	: Front and rear wheel angles
$F_{\text{resist/resistance}}$	: Resistance forces
l_f, l_r	: Distance from CG of vehicle to front and rear tires
M_{zd}	: Disturbance moment
r	: Yaw rate
R_{eff}	: Effective tire radius
ω_{tf}, ω_{tr}	: Front and rear tire rotational velocities
s_f, s_r	: Front and rear wheel-slip ratios in longitudinal direction
f_f, f_r	: Front and rear correction values for Dugoff tire force
C_{xf}, C_{xr}	: Longitudinal tire stiffnesses for front and rear tires
C_{yf}, C_{yr}	: Lateral tire stiffnesses for front and rear tires
F_{Rf}, F_{Rr}	: Resultant forces for front and rear tires
μ	: Friction coefficient
F_z	: Normal force on tire

g	: Gravitational acceleration
θ	: Engine throttle angle
θ_D	: Rotation angle at inlet of differential
θ_T	: Rotation angle at outlet of differential
$\theta_{T,f}, \theta_{T,r}$	: Front and rear tire rotation angles
θ_E	: Rotation angle at outlet of engine
ω_D	: Rotational velocity at inlet of differential
ω_T	: Rotational velocity at outlet of differential
ω_E	: Rotational velocity of engine
i_T	: Gear ratio
i_D	: Differential ratio
J_E	: Engine inertia
J_D	: Differential inertia
J_T	: Transmission inertia
J_{eq}	: Equivalent inertia
$J_{T,f}, J_{T,r}$	: Front and rear tire inertias
$J_{eq,f}$	: Equivalent inertia at front axis
$J_{eq,r}$	: Equivalent inertia at rear axis
E	: Kinetic energy of vehicle transmission parts
T_E	: Engine torque
T_D	: Engine torque increased by transmission
T_T	: Engine torque at tires
$T_{net,engine}$	: Net engine torque
η_T	: Transmission efficiency
η_D	: Differential efficiency
$F_{rolling}$	: Tire rolling resistance
$C_{rolling}$	: Rolling resistance coefficient
F_{slope}	: Road slope resistance
γ	: Road slope angle
F_{drag}	: Aerodynamic drag resistance
C_{drag}	: Aerodynamic drag coefficient
ρ	: Air density
A	: Vehicle frontal area
$F_{acceleration}$	: Acceleration resistance
λ	: Rotating mass factor
R_f, R_r	: Front and rear dynamic tire radius
r_f, r_r	: Front and rear static tire radius
$T_{net,f}, T_{net,r}$	: Net engine torque at front and rear tires
T_{brake}	: Brake torque
Δx	: Distance traveled in axis -x
Δy	: Distance traveled in axis -y
$X_r O_1 Y_r$	: Local coordinate system of host vehicle
$P(X_f, Y_f)$	: Position of target vehicle
A	: Transformation matrix
R	: Rotation matrix
X_f^O	: Coordinate vector of target vehicle with respect to global coordinate system
A_{O1}^O	: Transformation matrix of local coordinate system of host vehicle with respect to global coordinate system

$\mathbf{X}_f^{01}$	: Coordinate vector of target vehicle with respect to local coordinate system of host vehicle
$\mathbf{F}_{\text{required}}$	: Required force
$\mathbf{a}_{\text{required}}$	: Required acceleration
$\mathbf{T}_{\text{required,engine}}$	: Required engine torque
$\mathbf{T}_{\text{required,wheel}}$	: Required engine torque at wheels
$\mathbf{a}_{\text{desired}}$	: Desired acceleration
$\mathbf{K}_v, \mathbf{K}_d$	: PD controller coefficients
$\mathbf{d}_{x,\text{desired}}$	: Desired distance between vehicles
$\mathbf{x}_{\text{target}}$	: Position of target vehicle in longitudinal direction
$\mathbf{x}_{\text{host}}$	: Position of host vehicle in longitudinal direction
$\mathbf{sc}$	: Scenario
$\mathbf{ctrl}$	: Controller

DESIGN AND SIMULATION OF A STOP AND GO CONTROL SYSTEM FOR ROAD VEHICLES

SUMMARY

In this thesis, a design methodology and simulation results of a Stop and Go control system for road vehicles have been presented. The main objective of the Stop and Go control system is to help drivers in heavy congested traffic at urban areas by taking the responsibility for repeated throttle and brake applications to follow the preceding vehicle at relatively low speeds. This control strategy is known as one type of longitudinal control in literature. With this control strategy, more efficient traffic flow is also expected by taking the distance keeping logic into consideration.

In the study, it is intended to follow a flexible approach for integration of the control system with the vehicle model. The vehicle model consists of a single track nonlinear vehicle model (also known as “Bicycle Model”) which simply represents a real vehicle characteristics. Engine, driveline, and a nonlinear tire models are also included to complete a realistic vehicle model.

A hierarchical structure was adopted for the Stop and Go controller design. A low-level controller determines reference acceleration values, whereas a high-level controller attempts to track these reference acceleration values by modulating throttle and brake applications. High-level controller also determines different driving scenarios and so different control strategies. Two different control alternatives were offered for the design of the low-level controller for comparison. The first one of these comprises a PID Controller for the simple cruise control mode, and a PD Controller for the Stop & Go mode. The second one comprises again a PID Controller for the simple cruise control mode, and a Fuzzy Logic Controller for the Stop & Go mode. Performance of the Stop and Go controller was analyzed through extensive simulations conducted in MATLAB/Simulink for different driving scenarios.

This thesis consists of six chapters. First chapter gives a short introduction, and in the second chapter Stop and Go Control System is presented. Vehicle model is described in the third chapter. In the fourth chapter, structure of the stop and go controller and different driving scenarios that the vehicle is subjected to during simulations is presented. Chapter five gives results and comparisons for the simulations. Finally, suggestions and conclusions is given in the sixth chapter.

YOL ARAÇLARI İÇİN DUR VE KALK KONTROL SİSTEMİ TASARIMI VE SİMÜLASYONU

ÖZET

Bu tezde, yol araçları için bir Dur ve Kalk kontrol sistemi tasarlanmıştır. Tasarlanan sistem ile simülasyonlar yapılmış ve elde edilen sonuçlar karşılaştırmalı olarak sunulmuştur. Dur ve Kalk kontrol sisteminin ana amacı; kentsel alanlardaki yoğun trafikte nisbi olarak düşük hızlarda öndeki aracı takip edebilmek için art arda gerçekleşen gaz ve fren uygulamaları sorumluluğunu üstlenerek sürücülere yardımcı olmaktır. Bu sistem ile sürücüler özellikle şehir içi trafikte en sık karşılaşılan yorucu ardışık durma ve kalkma işlemlerinden kurtulmuş olur. Dur ve Kalk kontrol sisteminin bu yönüyle öncelikli olarak bir konfor sistemi olduğu söylenebilir. Literatürde bu kontrol stratejisi boylamsal araç kontrolünün bir çeşidi olarak bilinir.

Dur ve Kalk kontrol sisteminden ayrıca mesafe koruma mantığı dikkate alınarak daha verimli bir trafik akışının sağlanması beklenmektedir. Dur ve Kalk kontrol sistemi bunların yanında uzun süreli ve yorucu trafiğin sürücü üzerinde oluşturduğu dikkat dağılması gibi nedenlerden dolayı oluşabilecek kazaların da önüne geçerek aynı zamanda bir güvenlik sistemi haline gelmektedir. Dur ve Kalk kontrol sisteminde, güvenlik kavramının konfordan sonra gelmesinin en önemli nedeni tasarlanan sistemin belirli limitler dahilinde çalışmasıdır. Mesela, aracın ivme değerleri sürücünün konforunu etkilememesi ve panik duygusu oluşturmaması için belirli limitler dahilinde sınırlandırılmıştır. Bu limitlerin dışına çıkılması durumunda sürücünün ses ile ve görsel olarak uyarılarak kontrolü devralması sağlanmaktadır.

Bu çalışmada, tasarlanan kontrol sistemi ile araç modelinin entegrasyonunda esnek bir yaklaşımın takip edilmesi amaçlanmaktadır. Araç modeli MATLAB/Simulink ortamında modellenen, gerçek araç karakteristiklerini basit bir şekilde yansıtan tek izli nonlineer araç modelinden (ayrıca “Bisiklet Modeli” olarak da bilinmektedir) oluşmaktadır. Gerçekçi araç modelini tamamlamak için Simulink’e ayrıca motor, güç aktarma organları ve nonlineer lastik modelleri de dahil edilmiştir. Araç dinamikleri ve modele eklenen her bir mekanizma Simulink ortamında matematiksel denklemler yardımıyla ya da motor modeli gibi look-up tabloları yardımıyla modellenmiştir. Modellenen tüm alt sistemler birleştirilerek bu tez çalışması için gerekli ve yeterli olan araç modeli ortaya çıkmıştır.

MATLAB/Simulink’te oluşturulan araç modeli sürücünden giriş değerleri olarak sadece gaz ve fren komutlarını beklemektedir. Çıkış değerleri olarak ise araç hızı, motor devri, yalpa açısı, şasi kayma açısı, alınan mesafe gibi bilgiler verilebilmektedir. Bu bilgilere özellikle kontrolör tasarımında kontrolcü tarafından ihtiyaç duyulmaktadır. Araç modeline sürücü tarafından girilmesi gereken değerler kontrolcü tasarlandıktan sonra kontrolcü tarafından verilmektedir. Simülasyonlar sırasında gerçek hayata daha uygun senaryolar elde edebilmek adına hem Dur ve

Kalk aracı için hem de önde bulunan aracı temsil etmek için tasarlanan aynı araç modeli kullanılmıştır. Öndeki araçtan da bazı bilgiler Dur ve Kalk aracına aktararak kontrol sisteminin çalışması sağlanmıştır. Aktarılan bu bilgiler Dur ve Kalk kontrol sisteminin en önemli ekipmanlarından biri olan radar sensörü için gerekli bilgilerdir.

Dur ve Kalk kontrolör tasarımı için hiyerarşik bir yapı benimsenmiştir. Bu hiyerarşik yapı bir üst-seviyeli kontrolcü ve bir de alt-seviyeli kontrolcünden oluşmaktadır. Alt-seviyeli kontrolcü Dur ve Kalk aracının önde bulunan aracı takip etmesi için gerekli olan referans ivme değerlerini belirlerken, üst-seviyeli kontrolcü ise gaz ve fren uygulamalarını kullanarak bu referans ivme değerlerini yakalamaya çalışır. Referans ivme değeri hesaplanırken, Dur ve Kalk aracının öndeki araç ile olan bağıl mesafesi ve aralarındaki bağıl hız dikkate alınır. Dolayısı ile Dur ve Kalk kontrol sisteminin, bir hız ve mesafe kontrolü olduğu söylenebilir. Referans ivme değerinin hesaplanmasının nedeni, buradan yola çıkarak önce bu ivmeyi elde etmek için tekerleklerde gerekli olan kuvveti bulmak, sonra da tekerleklerdeki bu kuvveti elde edebilmek için gerekli olan motor torkunu hesaplamaktır. Bu motor torku da sürücü tarafından motora verilen gaz girişi ile elde edilmektedir. Burada istenen, gerekli olan bu gaz girişinin hesaplanmasıdır. Motor modeli Simulink'e motor hartiası olarak look-up tablosu şeklinde eklenmiştir. Motor modelinin girişleri gaz açıklığı ve motor devri iken, çıkışı ise motor torkudur. Dur ve Kalk kontrolcüsünün ihtiyacı olan gaz açıklığının hesaplanması için ise bu motor modelinin tersinin kullanılması gerekmektedir. Ters motor modelinin girişlerini motor devri ve hesaplanan motor torku oluştururken, çıkışını ise gaz açıklığı oluşturur. Ters motor modeli, motor modelindeki verileri kullanarak ve tabloda eksik kalan veriler için de enterpolasyon ve ekstrapolasyon yöntemlerini kullanarak oluşturulmuştur. Ters motor modeli de motor modelinde olduğu gibi sisteme look-up tablosu şeklinde entegre edilmiştir. Gaz açıklığı en düşük değer olan sıfır olduğunda eğer motor tablosunun verdiği negatif tork (motor freni olarak da bilinmektedir) yeterli olmazsa, üst-seviyeli kontrolcü gerekli olan ekstra fren torkunu hesaplayarak frenleri de devreye sokar. Bu şekilde hem takip sırasında gerekli olan hız hem de mesafe üst-seviyeli kontrolcü tarafından korunmuş olur.

Üst-seviyeli kontrolcü ayrıca şehir içi trafikte karşılaşılma ihtimali en yüksek olan değişik sürüş senaryolarını da belirler. Bu senaryolar tasarlanan kontrol sisteminin performansının değerlendirilmesi açısından çok önemlidir. Üst-seviyeli kontrolcü bunların yanında ayrıca her bir senaryo için farklı kontrol stratejilerini de belirler. Belirlenen senaryolar tabii ki sistem limitleri dahilinde belirlenmiştir. Örnek olarak, Dur ve Kalk aracında trafik işaretlerini algılama ile ilgili herhangi bir sistem tasarlanmadığından trafik işaretleri ile ilgili olabilecek muhtemel herhangi bir senaryo da tasarlanmamıştır.

Alt-seviyeli kontrolcünün tasarımında kıyaslama açısından iki farklı kontrolcü alternatifi kullanılmıştır. Bunlardan ilki, basit seyir kontrol modu (hız sabitleyici kontrol) için bir PID Kontrolcüsü ve Dur ve Kalk modu için de bir PD Kontrolcüsüdür. İkincisi, yine basit seyir kontrol modu için bir PID Kontrolcüsü ve Dur ve Kalk modu için ise de bir Bulanık Mantık Kontrolcüsüdür. Burada kıyaslamaya Bulanık Mantık kontrolcüsünün dahil edilmesinin asıl nedeni, bu kontrolcünün insan davranış ve deneyimlerinin de kontrolcü yapısına dahil edilmesine olanak sağlamasından dolayıdır.

MATLAB/Stateflow’da oluşturulan tüm sürüş senaryoları için Simulink’te simülasyonlar yapılmış ve Dur ve Kalk kontrolcüsünün performansı analiz edilmiştir. Belirlenen kritik senaryolar için de PD ve Bulanık Mantık kontrolcülerinin her ikisi için simülasyonlar yapılmış ve aralarında kıyaslamalarda bulunulmuştur.

Bu tez toplam altı bölümden oluşmaktadır. Birinci bölüm Dur ve Kalk kontrol sistemi ile ilgili genel bir bilgi vermekte ve tezin amacından bahsetmektedir. Dur ve Kalk kontrol sistemine ihtiyacın nasıl doğduğu ve hangi gelişmelerden sonra bu noktaya geldiği bu bölümde açıklanmıştır. İkinci bölümde ise Dur ve Kalk kontrol sistemi hakkında daha detaylı bir bilgi verilmiştir. Şehir içi trafiğinin yapısı, Dur ve Kalk kontrol sisteminin fonksiyonel gereklilikleri, sensör yapısı ve literatür araştırmaları ikinci bölümde verilmektedir. Üçüncü bölümde MATLAB/Simulink’te oluşturulan araç modeli tüm detayları ile anlatılmaktadır. Araç modelini oluşturan tüm hareket denklemleri sunulmuş, modele eklenen sistemler detayları ile anlatılmıştır. Ayrıca araç modelinin güvenilirliğini test etme açısından bu bölümde de çeşitli senaryolar ile simülasyonlar yapılmış ve sonuçları verilmiştir. Dördüncü bölümde, Simulink’te ve MATLAB/Stateflow’da oluşturulan Dur ve Kalk kontrolcüsünün yapısı anlatılmış ve simülasyonlar sırasında aracın maruz bırakılacağı değişik sürüş senaryoları sunulmuştur. Alt-seviyeli kontrolcüler ve hesaplama yöntemleri ve üst-seviyeli kontrolcüler bu bölümde tanıtılmıştır. Beşinci bölüm tasarlanan tüm senaryolar için PD kontrolcüsü ile uygulanan simülasyon sonuçlarını ve tasarlanan her iki kontrolcü için de teknik açıdan kritik senaryolarda yapılan kıyaslamaları vermektedir. Son olarak, sonuçlar ve öneriler kısmı altıncı bölümde verilmektedir.

1. INTRODUCTION

Many research and development companies and institutes have been studying actively on driver assistance systems (DAS) and active safety systems since more than two decades considering the potential for improved driving comfort and increased vehicle safety. Longitudinal vehicle control is one of the most attractive subjects among these systems.

Cruise Control (CC) system, an example of longitudinal vehicle control in the context of driver assistance, has already been commercialized by most automotive manufacturers for many years and widely being used at road vehicles. This system simply sets the vehicle speed at a desired value and only gives throttle action to the vehicle. The system can easily be out of order by pressing the brake pedal. CC increases driving comfort at some level especially for long travels.

Adaptive Cruise Control (ACC) is more advanced version of the cruise control system, marketed by some automotive companies [1] and currently under development by others. ACC systems are designed especially for highway applications that has in general homogenous traffic behavior. The vehicle runs at a desired cruise control speed when there is no vehicle in front. When a target vehicle at slower speed is detected by a radar sensor mounted on the bumper of the host vehicle, the throttle and brake system are controlled to maintain the inter-vehicle distance which is set by the control system. Host vehicle follows the target vehicle at the same speed until the vehicle in front is no longer a target vehicle or one of the vehicles changes its lane. In this case, the host vehicle increases its speed again to the cruise control speed. ACC system controls only the longitudinal behavior of the vehicle and if there is a need for steer, driver interface is required. In addition, since comfort comes first for the ACC system, acceleration and deceleration values are limited for the control system. If there is a situation to need hard brake for preventing collision, driver is alerted for action.

Second generation ACC systems are also being developed by the automotive companies to be able to cope with more complex traffic situations that operate at lower speeds in urban areas [2]. In terms of comfort, safety, traffic flow, noise and emissions, second generation ACC will surely increase the capability of the first generation ACC in dense traffic. However, it is almost impossible to combine the characteristics of the first and the second generation ACC together in one control system. Thus, a new concept arose in the literature called “Stop and Go Control (S&G)”.

The aim of the Stop and Go control is a partial automation of the vehicle longitudinal control resulting in a reduction in the task of the driver in a traffic jam. Stop and Go control system controls both speed and distance to preceding vehicles as in the ACC, and so it is aimed to increase comfort and safety of the vehicle by reducing the danger of rear-end collisions. Stop and Go control, however, is not limited to the simple task of following and requires extensive knowledge about traffic behavior. Sensor requirements should be increased to detect the driving environment and the acquired information needs to be assessed for the traffic situation.

In order to understand why there are so many researches and developments on longitudinal control, it would be better to investigate rear-end collisions around the world. As discussed earlier, longitudinal control not only provides comfort for the drivers but also increases safety by decreasing the possibility of rear-end collisions. In the United States, rear-end collisions are the most frequent among all crash types and accounts for 29% of all police-reported crashes with a total number of approximately 1.8 million annually according to a study by Najm et al. on light vehicle crashes [3] using the NASS/GES database. In Japan, ITARDA data shows that rear-end collisions are the most numerous of all crash types [4]. In 2005, they accounted for approximately 32% of all crashes, including approximately 300,000 collisions. In UK, the number of rear-end collisions are around 26% of the approximately 2.7 million crashes [5]. In Turkey, although there is no specific statistical survey on rear-end collisions, Table 1.1 shows that the number of collisions in inhabited areas is significantly higher than the collisions occurred in uninhabited areas. However, when the table is investigated further, it can be seen that the number of killed persons in inhabited areas is less than the ones in uninhabited

areas. This vice versa situation tells that the vehicle speeds are relatively slower which can be considered as general urban traffic.

Table 1.1 : The rate of traffic accidents to total accidents according to the settlement, 2001-2010 [6].

Toplam - Total						
	Kaza Accidents		Ölü Persons killed		Yaralı Persons injured	
2001	409 407		2 954		94 497	
2002	407 103		2 900		94 225	
2003	422 272		2 811		95 607	
2004	537 352		4 427		136 437	
2005	620 789		4 505		154 086	
2006	728 755		4 633		169 080	
2007	825 561		5 007		189 057	
2008 ⁽¹⁾	104 212		4 236		184 468	
2009 ⁽¹⁾	111 121		4 324		201 380	
2010 ⁽¹⁾	116 804		4 045		211 496	
Yerleşim yeri - Inhabited area						
	Kaza Accidents	(%)	Ölü Persons killed	(%)	Yaralı Persons injured	(%)
2001	363 528	88,8	1 309	44,3	62 690	66,3
2002	362 979	89,2	1 215	41,9	62 202	66,0
2003	375 168	88,8	1 129	40,2	63 002	65,9
2004	461 848	85,9	1 599	36,1	79 671	58,4
2005	533 161	85,9	1 647	36,6	91 336	59,3
2006	626 638	86,0	1 638	35,4	99 999	59,1
2007	714 197	86,5	1 748	34,9	113 598	60,1
2008 ⁽¹⁾	71 567	68,7	1 433	33,8	111 064	60,2
2009 ⁽¹⁾	76 429	68,8	1 549	35,8	122 036	60,6
2010 ⁽¹⁾	80 517	68,9	1 365	33,7	129 051	61,0
Yerleşim yeri dışı - Uninhabited area						
	Kaza Accidents	(%)	Ölü Persons killed	(%)	Yaralı Persons injured	(%)
2001	45 879	11,2	1 645	55,7	31 807	33,7
2002	44 124	10,8	1 685	58,1	32 023	34,0
2003	47 104	11,2	1 682	59,8	32 605	34,1
2004	75 504	14,1	2 828	63,9	56 766	41,6
2005	87 628	14,1	2 858	63,4	62 750	40,7
2006	102 117	14,0	2 995	64,6	69 081	40,9
2007	111 364	13,5	3 259	65,1	75 459	39,9
2008 ⁽¹⁾	32 645	31,3	2 803	66,2	73 404	39,8
2009 ⁽¹⁾	34 692	31,2	2 775	64,2	79 344	39,4
2010 ⁽¹⁾	36 287	31,1	2 680	66,3	82 445	39,0

In another study held by Eis et al. [7] on German GIDAS data, most of the vehicle-to-vehicle single rear-end collisions occurred on urban roads. Eis et al. also show that approximately 70% of the crashed cars in vehicle-to-vehicle single rear-end collisions has a crash speed lower than 30 km/h. In addition, Langwieder et al. [8]

conducted a study in Germany and found that the difference in speed between the vehicles at the time of the collision was less than 15 km/h in more than 70% of 496 random sampled rear-end collisions involving personal injury.

1.1 Purpose of Thesis

The goal of this thesis study is to design and simulate a stop and go control system for road vehicles. A single track nonlinear vehicle model is constructed in MATLAB/Simulink with engine, driveline, and tire models. The vehicle model is then controlled by controllers designed in MATLAB/Stateflow for different driving scenarios.

In the first chapter of this thesis, an introduction to the subject is given briefly. Information about development stages of Stop and Go is given and its characteristics described. The need for Stop and Go control is also determined by giving some statistical information. In the second chapter, Stop and Go control is detailed with literature survey. In chapter three, vehicle model is described by dynamic equations and MATLAB/Simulink subsystems are also shown. Chapter four presents the Stop and Go control structure for two different control strategies, and the scenarios that are most frequently seen in Stop and Go traffic are explained and modeled in MATLAB/Stateflow. Simulation results and comparisons for the scenarios explained in chapter four are shown in chapter 5. Lastly, suggestions and conclusions is given in chapter six.

2. STOP AND GO CONTROL SYSTEM

Evolution of Longitudinal Control can be set in order as described earlier: Cruise Control, Adaptive Cruise Control, and Stop and Go Control. In general, the aim of longitudinal control is to take driver responsibilities in longitudinal direction by intervening throttle and braking system of the vehicle. While Cruise Control is aimed to follow a desired speed set by the driver only acting on throttling, Adaptive Cruise Control has the capability of following preceding vehicles at target vehicle speed at a desired distance by controlling both throttle and braking system. When Stop and Go Control System comes into question, it can be considered as an adaptation of Adaptive Cruise Control System into urban stop and go traffic situations.

2.1 Overview on Stop and Go Control System

Stop and Go Control, as one type of longitudinal control, is in fact a question of speed and distance control. While a vehicle equipped with S&G runs at a desired cruise control speed when there is no target in front, following action immediately starts when a target vehicle is detected in front at target vehicle speed and at a desired distance. This situation continues until the target vehicle is no longer a target in case if it changes its lane, accelerates over a specified limit or the host vehicle changes its lane. Lateral control of the vehicle is completely under the driver's responsibility. Stop and Go control is only able to intervene into throttle and brakes. The system requires only two inputs from the driver: one is cruise control speed when there is no target, and the other is headway time that determines the desired distance according to variable host vehicle speed. This is also known as Constant Time Headway (CTH) Control [9].

One point should also be highlighted about Stop and Go control that the system is primarily a comfort system, not a safety system. It serves as a safety system to some extent, but not much. The main reason for this is that it has limits in terms of acceleration and deceleration, which are set by standards [10,11]. In case if there is

an emergency, the system may not be able to stop the vehicle within its deceleration limits, and the driver is alerted.

When compared to an Adaptive Cruise Control with a low level of deceleration, Stop and Go Control must have a much broader view of the driving environment. Simple task of following is no longer valid for Stop and Go as it requires extensive knowledge about driver and traffic behavior.

New sensor concepts need to be developed to detect the driving environment and the acquired information should be assessed for the traffic situation. Models are also necessary to identify the traffic state and appropriate actions should be determined for each situation.

2.2 Stop and Go Driving Environment and Requirements

Stop and Go Control, in general, works at low speeds up to zero velocity. The aim is to take over driver's tedious tasks such as accelerating, decelerating, and stopping at a proper distance to preceding vehicle in an environment called congested traffic with the characteristics listed as follows [2]:

- All lanes are full of vehicles
- No possibility to choose travel speed
- Little or no speed variations inside and outside the driving lane
- Maximum velocity is less than 30-40 km/h
- Minimum velocity is 0 km/h
- Longer time periods of stopped traffic
- Frequent stops followed by intermittent starts
- Relatively long duration

By giving longitudinal assistance to the driver, Stop and Go control has the potential to harmonize the traffic flow and so traffic safety is increased. The system, however, should provide feedback to the driver about its operation mode frequently so that the driver can override the system at any time.

Stop and Go Control is expected to have at least the following functional requirements [12]:

- Maintain a safe distance to preceding vehicle
- Slow down behind decelerating vehicle, eventually make a full-stop if required
- Slow down and stop behind stopped vehicle
- Autonomous “go” from standstill when stopped vehicle moves
- “go” when initiated by driver in case no target vehicles exist
- Control vehicle speed up to set-speed when no preceding vehicles are present
- Manage standstill condition even on slopes
- Manage near cut-ins from adjacent lanes comfortably
- Recognize and manage lane changes initiated by the driver
- Harmonize perturbed traffic flows
- Inform driver when system limits are reached
- Switch-off when brake pedal is activated
- Limit vehicle speed when set-speed is reached
- Adjust headway according to driver preference

It might also be possible for Stop and Go control system to adjust headway and vehicle speed according to road class, road attributes and roadway curvature if it is integrated with, for example, a vehicle navigation system [13].

As a first generation Stop and Go control system, it is not able to cope with the following functions [2]:

- React to cross traffic from side streets
- React to traffic signals and signs
- Avoid a collision under every circumstance

2.3 Control Problem for Stop and Go Control System

As discussed earlier in the thesis, stop and go control is a speed and distance control like in adaptive cruise control. The system deals with some variables shown on Figure 2.1.

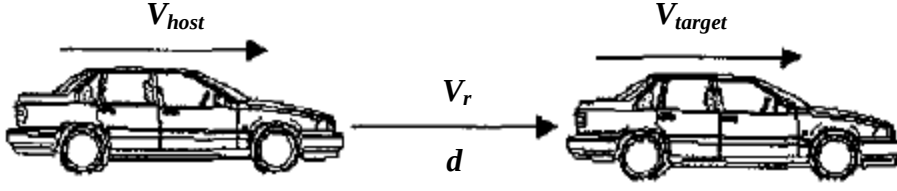


Figure 2.1 : Vehicle following scenario.

These variables can be listed as follows:

V_{target} : Target vehicle speed

V_{host} : Host vehicle speed

d : Relative distance between the vehicles

V_r : Relative velocity

Stop and go control in longitudinal direction attempts to maintain a desired distance between the host and target vehicles with a speed at target vehicle's velocity. This control is achieved by using the measured relative distance and relative velocity. Derivatives of these values can also be used in the control system as will be discussed later in Fuzzy Logic Control subsection.

There are different concepts for determination of desired distance in literature [14]. Constant Space Headway Control [15], Constant Time Headway Control (CTH) [9], which is used in this thesis, and Variable Time Headway Control (VTH) [16].

Desired distance is expressed in CTH concept as follows:

$$d_{desired} = T_h \cdot V_{host} \quad (2.1)$$

where T_h is headway time set by the driver. From the formula, it is obvious that desired distance is linearly proportional to the host vehicle speed. During the following, when the system is in steady state, desired distance can also be expressed as follows:

$$d_{desired} = T_h \cdot V_{host} = T_h \cdot V_{target} \quad (2.2)$$

While Stop and Go system is analyzed in terms of control problem, system nonlinearities should also be taken into account. Both the controlled system and the controllers have some nonlinearities. Engine model, tire model, transmission, and vehicle kinematics can be given as examples to controlled system nonlinearities. Throttle-brake transitions and lane changes are examples to controller nonlinearities.

Target detection and assessment of target vehicle maneuvers can also be counted as another control problem. The stop and go control system should detect the right target, for example, when there are two or more vehicles in one lane, or the vehicle in adjacent lane is so close to the host vehicle's lane. The closest one is determined as the target vehicle in this case. In addition, lane change and curving for the target vehicle is physically the same maneuvers, but the system distinguishes these two scenarios by assessing the next steps.

2.4 Literature Survey on Stop and Go Control

In general, it can be said that researchers in academia and industry conduct their study on Stop and Go Control as an extension to Adaptive Cruise Control [17]. As discussed earlier in Section 2.2, basic requirements for realizing a stop and go control system are explained in detail in [2]. Congested traffic flow properties described and dependency of driver to the traffic flow mentioned.

In [18], control structure for Stop and Go resembles the one in ACC and desired distance is given with an empirical expression based on human behavior. Low-level throttle and brake controllers are also introduced. While desired acceleration is produced by using inverse of an engine look-up table, desired brake torque is obtained by controlling the master cylinder pressure.

Importance of driver behavior in Stop and Go is pointed out in [19]. ACC system having ability to operate at S&G situations in urban traffic is tuned with a data collected from 20 different drivers to represent an average driver's behavior. There are many other studies in the literature based on human drivers driving characteristics [20-23].

Another and maybe the most important point about Stop and Go Control are control algorithms. Different control algorithms for speed and distance control are presented in the literature. Linear Quadratic (LQ) optimal control theory has been used to design vehicle desired acceleration for distance control in [10]. Here, the control algorithm consists of a distance control algorithm and throttle/brake control algorithm for acceleration tracking. Vehicle tests of this LQ optimal control carried out in [24].

In another study [25], a simple PI controller is used for reliability and robustness because of not knowing true car dynamics in detail. However, the system is capable of car following in low-speed situations in urban areas as well as in high-speed situations.

A new inter-distance reference model-based control approach is adopted in [26,27], which can be used both in cruise control and stop-and-go scenarios. The proposed model is non-linear and provides dynamic solutions, which verify comfort and safety criteria simultaneously. An important property of this proposed structure concerns the fact that the control design could be meet independent of the model design, permitting the additional control loop responsible only for model-matching between the actual system and the desired reference dynamics. In [28], another Model Matching Control (MMC) controller based on Sliding Mode Control (SMC) method is proposed for longitudinal acceleration tracking control for Stop and Go control system. In this study, nonlinearities of the vehicle acceleration response on low-speed condition are investigated and a nominal trans-function model of the vehicle longitudinal dynamic system is obtained with LMS system identification technique. MMC controller, including a SMC feedback compensator, is designed by using this model. Thus, advantage of the two control methods as control robustness and rapid response is combined.

Robust grey-box closed loop control is implemented in [29,30] for stop-and-go control. A robust strategy within an algebraic framework is proposed considering that many vehicle/road interaction factors (road slope, rolling resistance, aerodynamic forces) are very poorly known and measurements are quite noisy. A reference longitudinal acceleration is generated as a feedforward control, and a feedback term is also introduced. This closed-loop not only behaves a typical PID controller, but also estimates linear or nonlinear unmodeled dynamics.

There is a considerable amount of researches on Fuzzy Logic Control for stop and go applications [31-33]. Moreover, Genetic Algorithm is used to tune a PI controller in [34], which is used to determine the desired acceleration for stop and go control system. Model Predictive Controller (MPC) is adopted to design the controller for a Stop and Go system in [35]. Finally, in [36], Sliding Mode Controller (SMC) is used to design the controller for Stop and Go system.

2.5 Sensory Requirements for Stop and Go Control System

The most important matter in a stop and go control system, without doubt, is the information derived from the vehicle environment detection as the level of support to the driver will be based on this. Medium to high level of driving dynamics such as acceleration and deceleration, and frequent and sudden lane changes are the characteristics of a stop and go traffic. Therefore, sensory requirements for a stop and go system are higher than for an ACC range sensor in terms of coverage, dynamics, and robustness [2]. Stop and go control systems will most likely incorporate multiple sensors that are connected to a bus-type of hardware architecture for easy data access. The sensory system should also track multiple targets. The sensory system must be able to detect passenger vehicles, trucks, motorcyclists, and cyclists in all circumstances. Pedestrians can also be seen as targets. Nevertheless, a reliable detection method still does not exist. A concealed installation/mounting should be considered as well as multiple utilization of sensors for different functions such as parking aid, ACC, traffic sign recognition, etc.

Range, relative velocity and azimuth (or lateral position) variables need to be measured for object detection. Acquiring of extents of the sensed objects are quite important but so hard. The width of the object becomes quite relevant if its position is close to the host vehicle, especially in adjacent lanes where they could pose a possible threat.

2.5.1 Sensor geometry

Sensory requirement for the driving environment detection is much more extended for a stop and go control system when compared to standard ACC. A wider field of view is necessary in order to detect vehicles in adjacent lanes and track targets in curves with a small radius especially in mid and short range.

A possible starting point for setting up the sensor geometry requirements is based on the determination of the space required to maneuver a vehicle driven by an average driver as a function of the vehicle speed. Looking at Figure 2.2, it can be seen that the average driver uses a much larger level of lateral acceleration while cornering at lower speeds than at higher speeds.

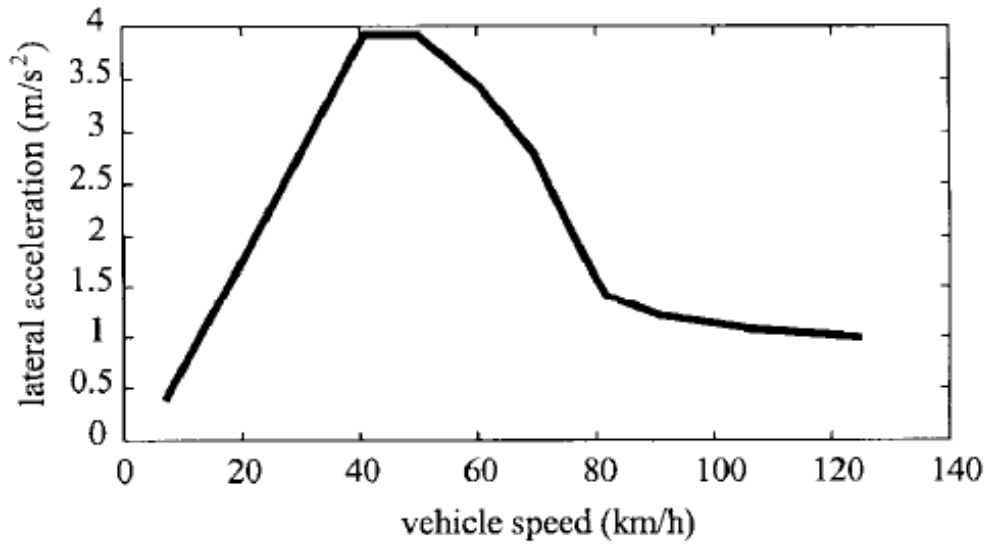


Figure 2.2 : Relationship between lateral acceleration and vehicle speed [37].

Maneuverability space can be determined by combining the knowledge from Figure 2.2 and the space required to stop a vehicle with a maximum deceleration (combined with a signal processing delay time) is given. Figure 2.3 shows the space of maneuverability based on Figure 2.2 with 4 m/s^2 brake deceleration and 0.5 sec processing time.

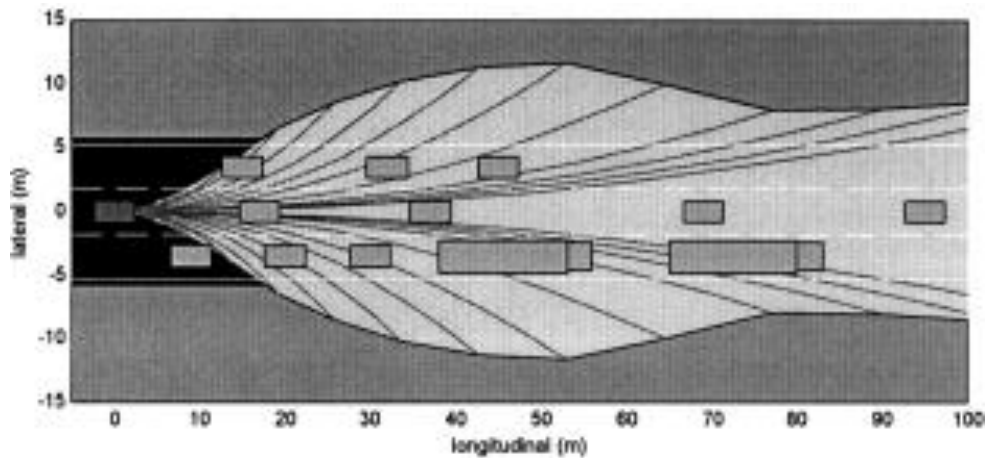


Figure 2.3 : Space of maneuverability based on Figure 2.2 [2].

Various sensor configurations can be put together based on Figure 2.3 so that the space of maneuverability is covered as much as possible. Examples of various sensor configurations can be seen in [2].

The azimuth angle of the short range radar, which can be used in stop and go control system applications, is in between $16\text{-}25^\circ$, and the detection range is around 30-40 m [11].

2.5.2 Sensor options

Four types of object detection such as radar, laser, vision and ultrasonic are technically feasible and applicable in a vehicle environment at the moment [2]. Each of them has its own technical and economical advantages and disadvantages. However, only radar and laser type sensors are currently being used in ACC and S&G systems.

In Japan, the first generation ACC systems were based on laser range sensors (Lidar-based sensors). The laser-based systems are more sensitive to adverse weather conditions (reducing the maximum detection range in rainy conditions, creating “ghost” objects due to road spray) but have the potential for a larger angular vision (azimuth) and a better angular resolution (especially with a scanning principle) than radar-based systems. Laser-based systems are cheaper than radar-based systems in terms of cost perspective. Because of their good angular resolution, scanning laser sensors have a potential to become usable for stop and go control systems. The range of interest for a stop and go system (30-40 m) and the speeds of travel are small enough (30-40 km/h), thus sensor deterioration due to adverse weather conditions becomes less significant.

In Europe, most of the current ACC systems available are based on 77 GHz radar sensors (see Figure 2.4). The main advantages of the radar-based systems are that the relative velocity can be measured directly and that the performance does not deteriorate under adverse weather conditions (such as heavy rain and road spray). One of the key issues with radar-based sensors are their capability to create a sufficiently large enough angular vision with correspondingly good angular resolution. Better angular resolution can be offered by mechanically-scanned systems but they are most likely more expensive than fixed-beam radars. Currently, several manufacturers are working on 24 GHz radar sensors as an alternative for ultrasonic parking-aid sensors (see Figure 2.5). With a sufficiently large range and good enough angular resolution, these sensors might become valuable for a stop and go control system. 24 GHz short range radars could also provide enhanced performance and responsiveness during challenging driving situations such as vehicle “cut-ins” in urban traffic. Figure 2.6 shows different usage area of 24 GHz short range radar in automotive safety and control applications. Ranges and angular reaches (azimuth) of short and long range radars can be seen in Figure 2.7 together.



Figure 2.4 : 77 GHz long range radar [38].



Figure 2.5 : 24 GHz short range radar [38].

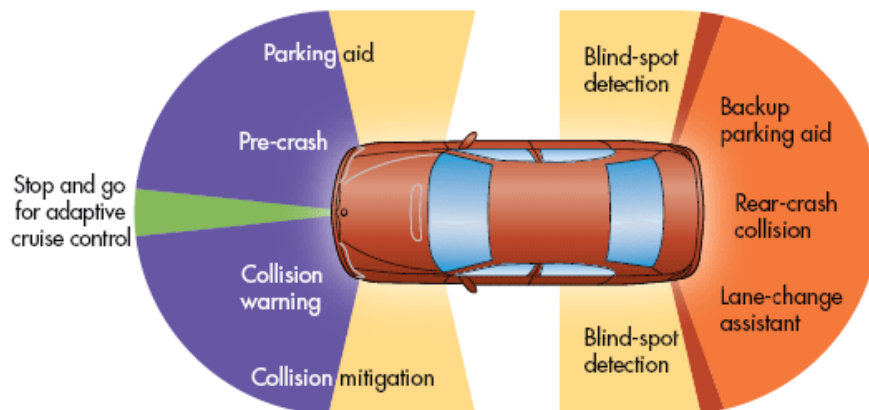


Figure 2.6 : Short range radar based assistance and safety applications [39].

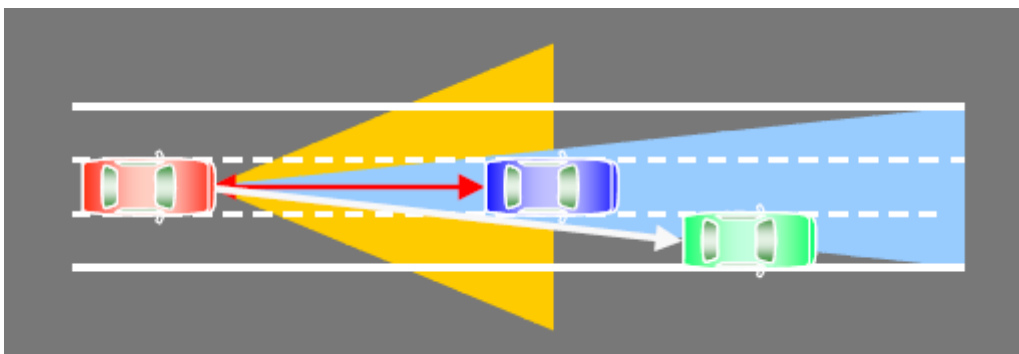


Figure 2.7 : View of short and long radar ranges and angular visions [38].

2.5.3 Multi-sensor fusion

Multiple sensors approach will most likely be used on a stop and go control system with reasonable overlap in detection ranges, possibly using different detection principles. A sensor fusion concept is then required [2].

Fusion not only enables and stabilizes fast-object information through redundant information and models, but also deals with inhomogeneous and unsynchronized data. In addition, a sensor malfunctioning detecting mechanism is available. The sensor fusion module framework is formed by a multiple-target tracking module (see Figure 2.8). It generates continuous object trajectories from individual measurements and maintains tracking even if sensor signals drop out for a short duration of time. The core of the tracking module is a Kalman filter [40]. This filter is a prerequisite for continuous tracking and fusion of redundant sensor information and contains a model that describes the object dynamics (trajectories) through differential equations.

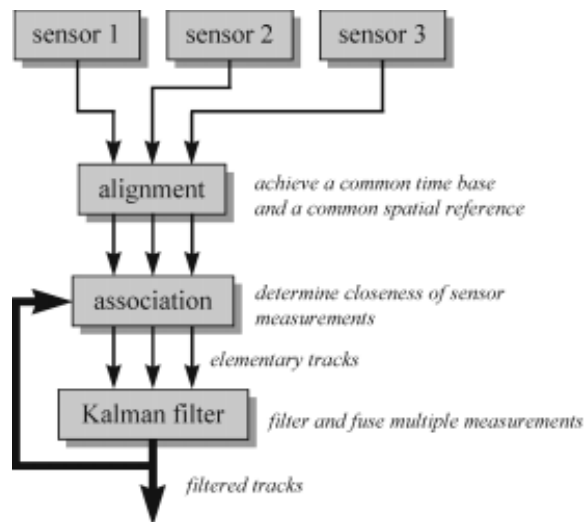


Figure 2.8 : Multiple-target tracking and sensor fusion concept [2].

3. LONGITUDINAL VEHICLE MODELING

It is crucial to design a vehicle model that accurately represents dynamics, geometry, driveline, and engine behavior of the vehicle so that a reliable controller for Stop and Go Control System can then be designed and tested by simulations. The vehicle model consists of sets of equations that will provide responses to inputs from the driver, such as throttle, brake, gearshift and steering, as close as possible to real vehicle responses. It is also possible to establish virtual connections to the vehicle's driver inputs with the help of designed controller. MATLAB/Simulink is used in this thesis to design and simulate the vehicle model.

3.1 Vehicle Model

Vehicle model consists of subsystems that include dynamics of the mechanical systems of the vehicle. These systems can be defined as vehicle dynamic system that expresses longitudinal and lateral dynamics of the vehicle, engine model that generates power to the vehicle, powertrain and transmission models that transmits the engine power to the wheels, and tire model that interact with ground. In addition, some resistance effects on the vehicle are also included into the model so that it becomes more realistic. All these vehicle subsystems will be presented in this chapter.

First, coordinate system of the vehicle should be specified before starting vehicle modeling. As seen in Figure 3.1, this coordinate system is located at center of gravity (CG) of the vehicle. Lateral and longitudinal movements of the vehicle based on global coordinate system are referenced to this 3-axis local coordinate system.

Local coordinate system moves together with the vehicle, as it is located on the vehicle. In Figure 3.1, x , y , and z denotes longitudinal, lateral, and vertical movements, respectively. Corresponding motions around these directions can also be seen as roll, pitch, and yaw motions.

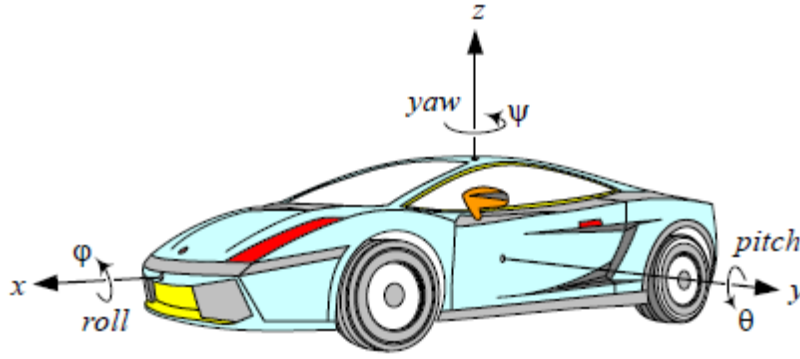


Figure 3.1 : Local coordinate system of a vehicle [41].

There is one more coordinate system on the vehicle that is located on the tires to express independent motions of the tires from the chassis of the vehicle (see Figure 3.2). The movements on this coordinate system will later be first transformed into the local coordinate system, and then into the global coordinate system.

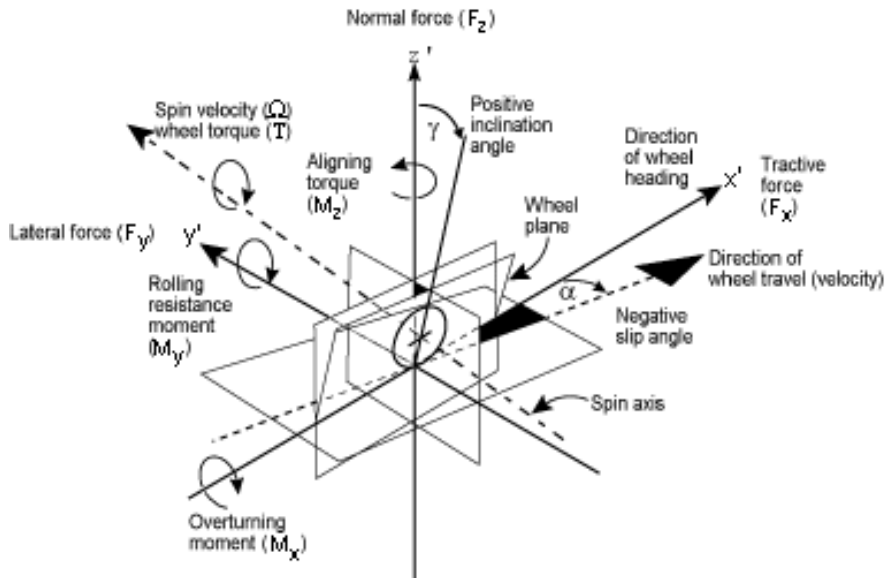


Figure 3.2 : SAE wheel coordinate system [42].

3.1.1 Vehicle dynamics

In this thesis, a Single Track Nonlinear Vehicle Model (also commonly known as “Bicycle Model”) is used to represent vehicle dynamics. Figure 3.3 shows the model that front and rear wheels are combined on a single axis passing through the center of gravity of the vehicle [43,44]. All dynamic equations will be based on this model.

Single Track Nonlinear Vehicle Model includes longitudinal and lateral dynamics as well as yaw dynamics. The angle ψ shown in Figure 3.3 between the vehicle local coordinate and global coordinate is named as yaw angle.

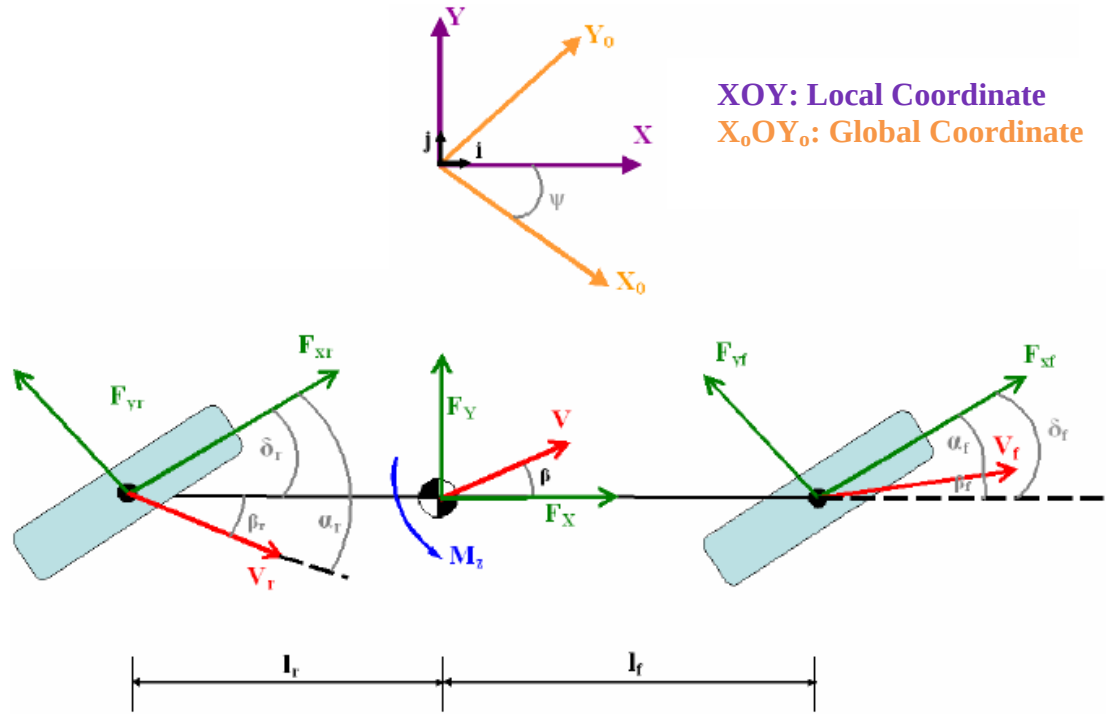


Figure 3.3 : Single track vehicle model [36].

When equations of motion for the vehicle are written, velocity vector V at center of gravity of the vehicle can be expressed as follows [42]:

$$\vec{V} = V \cdot \cos \beta \cdot \vec{i} + V \cdot \sin \beta \cdot \vec{j} \quad (3.1)$$

Derivative of this equation gives acceleration:

$$\begin{aligned} \vec{a} = \frac{d\vec{V}}{dt} = & \left[\dot{V} \cos \beta - V (\sin \beta) \dot{\beta} \right] \vec{i} + \left[\dot{V} \sin \beta + V (\cos \beta) \dot{\beta} \right] \vec{j} + \\ & + V \cos \beta \frac{d\vec{i}}{dt} + V \sin \beta \frac{d\vec{j}}{dt} \end{aligned} \quad (3.2)$$

Derivative of the unity vectors in (3.2) expresses motion of the global coordinate with respect to the local coordinate. These can be written as:

$$\frac{d\vec{i}}{dt} = \dot{\psi} \cdot \vec{k} \times \vec{i} = \dot{\psi} \cdot \vec{j} \quad (3.3a)$$

$$\frac{d\vec{j}}{dt} = \dot{\psi} \cdot \vec{k} \times \vec{j} = -\dot{\psi} \cdot \vec{i} \quad (3.3b)$$

Then, the acceleration term takes the following form:

$$\begin{aligned}\vec{a} = \frac{d\vec{V}}{dt} = & \left[\dot{V} \cos \beta - V (\sin \beta) \dot{\beta} \right] \cdot \vec{i} + V \cos \beta \dot{\psi} \cdot \vec{j} + \\ & + \left[\dot{V} \sin \beta + V (\cos \beta) \dot{\beta} \right] \cdot \vec{j} - V \sin \beta \dot{\psi} \cdot \vec{i}\end{aligned}\quad (3.4)$$

$$\Rightarrow \vec{a} = \underbrace{\left[\dot{V} \cos \beta - V (\dot{\beta} + \dot{\psi}) \sin \beta \right] \cdot \vec{i}}_{a_x} + \underbrace{\left[\dot{V} \sin \beta + V (\dot{\beta} + \dot{\psi}) \cos \beta \right] \cdot \vec{j}}_{a_y}\quad (3.5)$$

When the force equations are written with respect to X and Y coordinates;

$$\sum F_x = ma_x = m \left[\dot{V} \cos \beta - V (\dot{\beta} + \dot{\psi}) \sin \beta \right] \quad (3.6)$$

$$\sum F_y = ma_y = m \left[\dot{V} \sin \beta + V (\dot{\beta} + \dot{\psi}) \cos \beta \right] \quad (3.7)$$

$$\sum M_z = I_z \ddot{\psi} \quad (3.8)$$

can be obtained.

m denotes the mass of the vehicle in these equations, whereas β denotes the chassis sideslip angle at the vehicle CG. I_z is the mass moment of inertia with respect to a vertical axis -z through the CG.

Net force in the X direction includes longitudinal and lateral forces of the front and rear tire in the X direction, and also resistance forces.

$$\sum F_x = F_{xf} \cos \delta_f - F_{yf} \sin \delta_f + F_{xr} \cos \delta_r - F_{yr} \sin \delta_r - \sum F_{resist} \quad (3.9)$$

Resistance forces will be investigated later in this chapter in detail.

Net force in the Y direction can be written similarly:

$$\sum F_y = F_{yf} \cos \delta_f + F_{xf} \sin \delta_f + F_{yr} \cos \delta_r + F_{xr} \sin \delta_r \quad (3.10)$$

Moment equation can also be written as follows:

$$\sum M_z = F_{yf} \cos \delta_f l_f - F_{yf} \cos \delta_r l_r + F_{xf} \sin \delta_f l_f - F_{xr} \sin \delta_r l_r + M_{zd} \quad (3.11)$$

where l_f and l_r are distance from the CG of the vehicle to the front and rear tires, respectively. δ_f and δ_r are the wheel angles, and M_{zd} is the disturbance moment.

When all of these equations are expressed in matrix form;

$$\begin{bmatrix} \sum F_x \\ \sum F_y \\ \sum M_z \end{bmatrix} = \begin{bmatrix} \cos \delta_f & \cos \delta_r \\ \sin \delta_f & \sin \delta_r \\ l_f \sin \delta_f & -l_r \sin \delta_r \end{bmatrix} \begin{bmatrix} F_{xf} \\ F_{xr} \end{bmatrix} + \begin{bmatrix} -\sin \delta_f & -\sin \delta_r \\ \cos \delta_f & \cos \delta_r \\ l_f \cos \delta_f & -l_r \cos \delta_r \end{bmatrix} \begin{bmatrix} F_{yf} \\ F_{yr} \end{bmatrix} + \begin{bmatrix} \sum F_{resist} \\ 0 \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ M_{zd} \end{bmatrix} \quad (3.12)$$

and

$$\begin{bmatrix} -\sin \beta & \cos \beta & 0 \\ \cos \beta & \sin \beta & 0 \\ 0 & 0 & 1 \end{bmatrix} \cdot \begin{bmatrix} mV(\dot{\beta} + r) \\ m\dot{V} \\ I_z \dot{r} \end{bmatrix} = \begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} \quad (3.13)$$

sets of equations are obtained [45]. r is called the yaw rate, which is equal to the derivative of the yaw angle ψ .

In equation (3.13), inverse of the first matrix is equal to itself. Then, the equation can be written as:

$$\begin{bmatrix} mV(\dot{\beta} + r) \\ m\dot{V} \\ I_z \dot{r} \end{bmatrix} = \begin{bmatrix} -\sin \beta & \cos \beta & 0 \\ \cos \beta & \sin \beta & 0 \\ 0 & 0 & 1 \end{bmatrix} \cdot \begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} \quad (3.14)$$

To be able to define the vehicle speed at the front and rear tire axes with respect to the vehicle speed at the CG, the following sets of equations are used.

First, the velocity vector at the front tire $\vec{V}_f$ is equal to the sum of velocity at the CG and the velocity due to yaw motion:

$$\vec{V}_f = \vec{V} + \dot{\psi} \vec{k} \times l_f \vec{i} = V \cos \beta \vec{i} + V \sin \beta \vec{j} + \dot{\psi} l_f \vec{j} \quad (3.15)$$

Rearranging gives;

$$\vec{V}_f = V \cos \beta \vec{i} + (V \sin \beta + \dot{\psi} l_f) \vec{j} = V_{fx} \vec{i} + V_{fy} \vec{j} \quad (3.16)$$

velocity expression at local coordinate. The angle between the tire velocity and the vehicle chassis can be expressed as follows:

$$\beta_f = \tan^{-1} \left(\frac{V_{fy}}{V_{fx}} \right) = \tan^{-1} \left(\frac{V \sin \beta + \dot{\psi} l_f}{V \cos \beta} \right) = \tan^{-1} \left(\tan \beta + \frac{\dot{\psi} l_f}{V \cos \beta} \right) \quad (3.17)$$

Linear velocity of the tire on its own axis is given as:

$$V_{tf} = V_f \cos \alpha_f \quad (3.18)$$

Finally, scalar value of the tire velocity is:

$$V_f = |\vec{V}_f| = \sqrt{(V \cos \beta)^2 + (V \sin \beta + \dot{\psi} l_f)^2} \quad (3.19)$$

The sets of equations above can be written for the rear tires similarly as follows:

$$\vec{V}_r = \vec{V} + \dot{\psi} \vec{k} \times (-l_r \vec{i}) = V \cos \beta \vec{i} + V \sin \beta \vec{j} - \dot{\psi} l_r \vec{j} \quad (3.20)$$

$$\vec{V}_r = V \cos \beta \vec{i} + (V \sin \beta - \dot{\psi} l_r) \vec{j} = V_{rx} \vec{i} + V_{ry} \vec{j} \quad (3.21)$$

$$\beta_r = \tan^{-1} \left(\frac{V_{ry}}{V_{rx}} \right) = \tan^{-1} \left(\frac{V \sin \beta - \dot{\psi} l_r}{V \cos \beta} \right) = \tan^{-1} \left(\tan \beta - \frac{\dot{\psi} l_r}{V \cos \beta} \right) \quad (3.22)$$

$$V_{tr} = V_r \cos \alpha_r \quad (3.23)$$

$$V_r = |\vec{V}_r| = \sqrt{(V \cos \beta)^2 + (V \sin \beta - \dot{\psi} l_r)^2} \quad (3.24)$$

3.1.2 Tire model

Tires are one of the most important components in a vehicle that affect the dynamics significantly. Modeling of a pneumatic tire is so complex and hard in general. While there are close models established with the help of finite element methods, these are quite complex and time wasting models in terms of simulations. The tire model that will be used in this thesis is commonly used, and modeling is relatively easy. There are two commonly used tire models in the literature for the modeling of the tire dynamic forces.

The first one is called Pacejka or Magic Tire Formula [46], and consists of empirical sets of equations. This model calculates the longitudinal force F_x , lateral force F_y , and moment M_z by the help of longitudinal and sideslip angles.

An important feature for the tire forces F_x and F_y is that they do not change independent of each other. One is increasing while the other is decreasing, or vice versa like *sine* and *cosine* functions. These forces are expressed as independent of each other in Pacejka model, not considering the dependency between each other.

The other commonly used tire model is known as Dugoff tire model [47], which considers the dependency of tire forces F_x and F_y to each other. In this model, longitudinal tire forces change with longitudinal slip, whereas lateral forces are functions of lateral tire sideslip angles α_f and α_r . Dugoff tire model is used in this thesis while modeling the vehicle.

Dugoff tire model equations can be expresses with the following sets of equations:

First, the longitudinal wheel-slip ratio for each tire is defined as:

$$s_{f,r} = \begin{cases} \frac{R_{eff} \cdot \omega_{tf,tr} - V_{tf,tr}}{R_{eff} \cdot \omega_{tf,tr}} & R_{eff} \cdot \omega_{tf,tr} > V_{tf,tr} \quad (driving) \\ \frac{R_{eff} \cdot \omega_{tf,tr} - V_{tf,tr}}{V_{tf,tr}} & R_{eff} \cdot \omega_{tf,tr} < V_{tf,tr} \quad (braking) \end{cases} \quad (3.25)$$

Here, the slip value changes between -1 to 0 for braking, whereas it changes between 0 to 1 for driving.

The simplified Dugoff tire model is given by:

$$F_{xf} = f_f C_{xf} s_f \quad (3.26)$$

$$F_{xr} = f_r C_{xr} s_r \quad (3.27)$$

$$F_{yf} = f_f C_{yf} \alpha_f \quad (3.28)$$

$$F_{yr} = f_r C_{yr} \alpha_r \quad (3.29)$$

where C_x denotes the longitudinal tire stiffness, and C_y denotes the lateral (cornering) tire stiffness. These values also show that the forces change linearly with longitudinal and lateral slip values. The coefficients f_f and f_r are the correction values that provide the forces not to change after a certain slip values, and also result in expression of the forces F_x and F_y with respect to each other. The equations for f_f and f_r can be expressed as follows:

$$f_{f,r} = \begin{cases} 1 & F_{Rf,Rr} \leq \frac{\mu F_z}{2} \\ \left(2 - \frac{\mu F_z}{2 F_{Rf,Rr}}\right) \frac{\mu F_z}{2 F_{Rf,Rr}} & F_{Rf,Rr} > \frac{\mu F_z}{2} \end{cases} \quad (3.30)$$

$$F_{Rf,Rr} = \sqrt{(C_{xf,xr} s_{f,r})^2 + (C_{yf,yr} \alpha_{f,r})^2} \quad (3.31)$$

$$F_z = \frac{mg}{2} \quad (3.32)$$

3.1.3 Engine model

The engine is modeled as a static map by using values for a diesel mid-sized passenger car. This map is implemented in MATLAB/Simulink as a look-up table, whose inputs are throttle and engine speed, whereas the output is engine torque.

Figure 3.4 shows different torque curves obtained for different throttle inputs. These values are for the engine speeds between 800-4800 revolutions per minute (rpm). As can be understood from the curves, the engine gives its maximum torque not at maximum engine speed. However, minimum torque is obtained at maximum engine speed for any throttle. One more important point for the engine is that it can produce

negative torque at lower throttle inputs, and gives friction effect at driveline of the vehicle. This feature of the engine will be used later in this thesis to decrease the vehicle speed at some level.

Three-dimensional version of the engine map can also be seen in Figure 3.5.

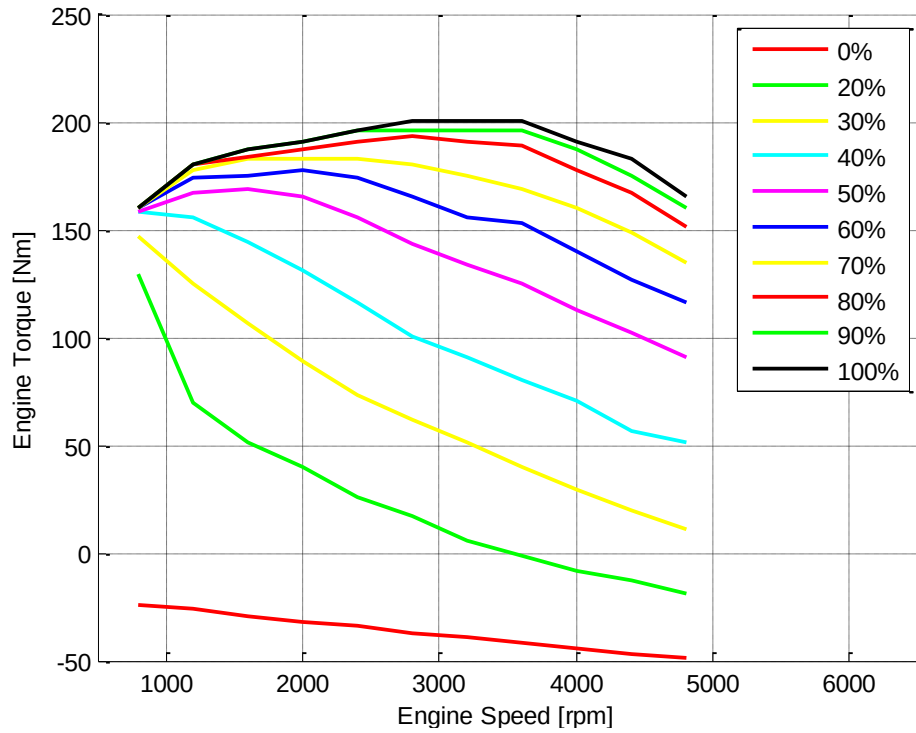


Figure 3.4 : Two-dimensional static engine map.

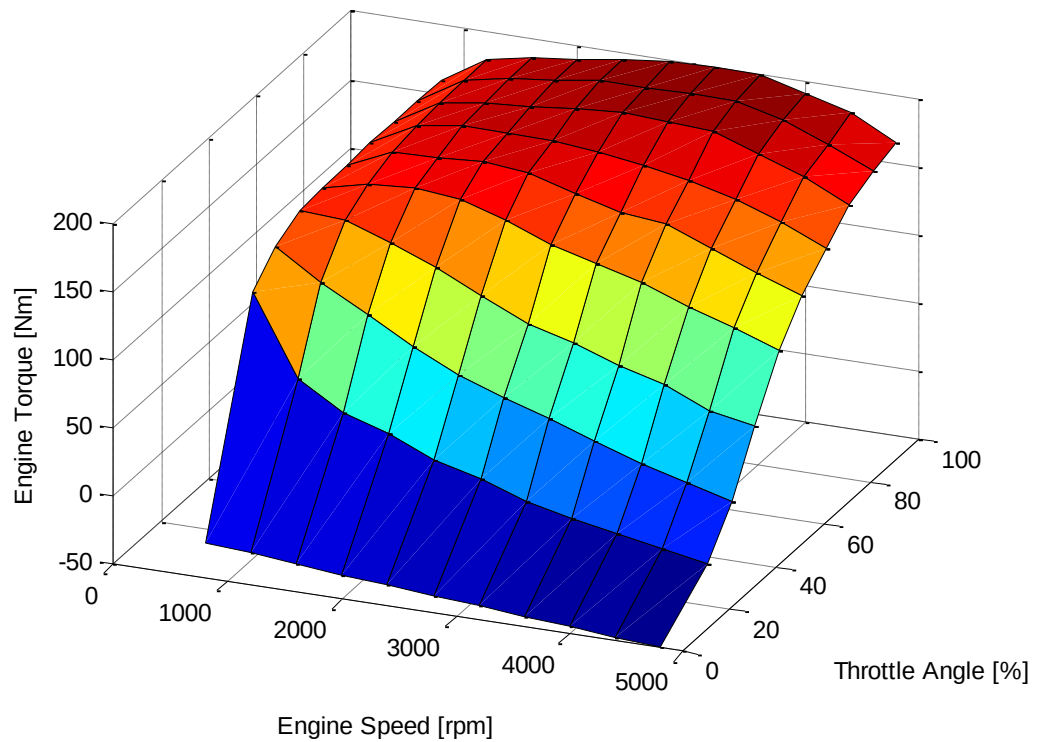


Figure 3.5 : Three-dimensional static engine map.

3.1.4 Driveline

Vehicle driveline is given the name for a set of mechanism that transmits the engine power to the wheels. While the power is transmitted to the wheels, rotational velocity is decreased to increase the torque. This is achieved by some gear mechanisms, and so gear ratios. In addition, different dynamics due to rotating masses in the driveline arises in terms of inertia. General view of the driveline of a rear-wheel driven vehicle is shown in Figure 3.6. In this model, shafts between the engine-transmission, transmission-differential, and differential-wheels are accepted as rigid and their inertial dynamics are included in the engine, transmission, differential, and wheels. Transmission of the driveline is provided by an automatic transmission. However, no torque converter is considered and continuously full transmission is accepted. Thus, inertial effects of the torque converter are included in the engine inertia. Further, inertial effects of the shaft in the front tires are included in the front tires.

As a result, J_E represents the inertia of the engine, torque converter and transmission inertias with respect to rotational velocity of the engine, J_D represents the inertia of differential and transmission parts rotating with a velocity of ω_D , and finally J_T represents the inertia of tires, wheels, brake disks and thrust shafts rotating with a velocity of ω_T .

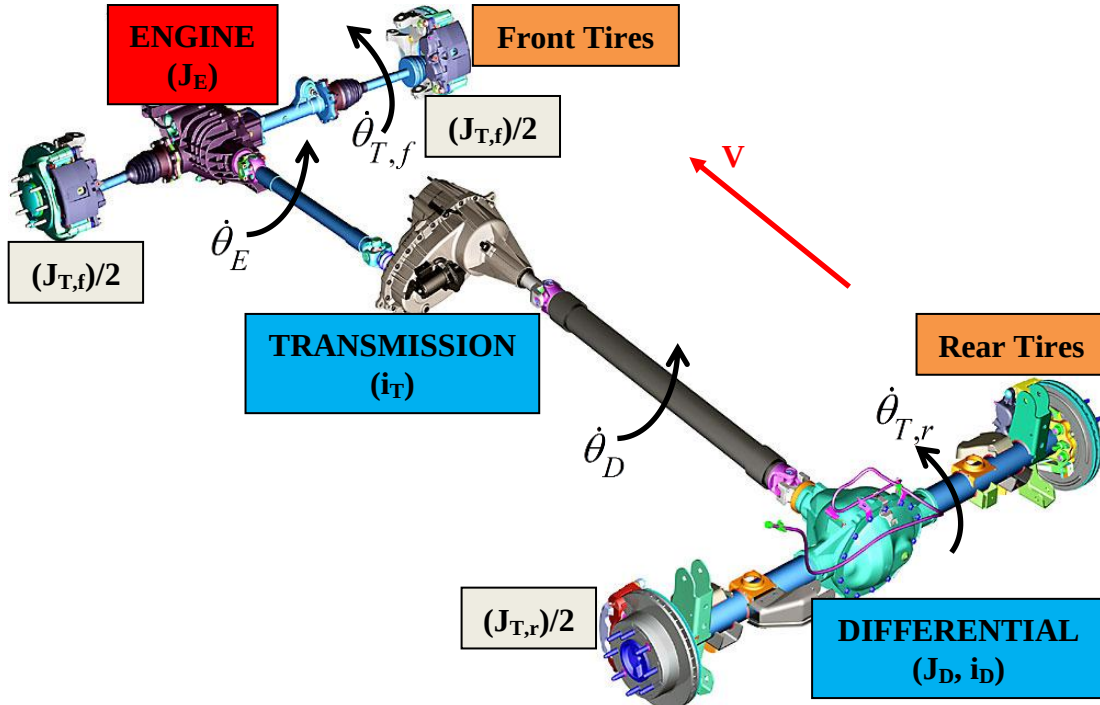


Figure 3.6 : View of a vehicle driveline.

Torque and velocity expressions through the driveline can be expressed with the following sets of equations;

$$\omega_D = \dot{\theta}_D = \dot{\theta}_E / i_T \quad (3.33)$$

$$\omega_T = \dot{\theta}_T = \dot{\theta}_D / i_D = \dot{\theta}_E / (i_D \cdot i_T) \quad (3.44)$$

$$T_D = T_E \cdot i_T \quad (3.55)$$

$$T_T = T_D \cdot i_D = T_E \cdot i_D \cdot i_T \quad (3.66)$$

where T_E is the engine torque, T_D is the increased torque by transmission, T_T is the torque changed by differential and transmitted to the wheels, i_T is the gear ratio and i_D is the differential (final drive) ratio.

It should be noted that all inertial effects on the powertrain expressed with respect to the wheel axes during both converting the engine rotational velocity to tire velocity and vice versa. Kinetic energy of the rotational masses can be written as follows:

$$E = \frac{1}{2} J_E \omega_E^2 + \frac{1}{2} J_D \omega_D^2 + \frac{1}{2} J_T \omega_T^2 \quad (3.77)$$

When the rotational velocities are defined in terms of tire rotational velocity, the following expression is found:

$$E = \frac{1}{2} J_E (\omega_T i_D i_T)^2 + \frac{1}{2} J_D (\omega_T i_D)^2 + \frac{1}{2} J_T \omega_T^2 \quad (3.88)$$

Similarly, energy expression with equivalent inertia can be written as:

$$E = \frac{1}{2} J_{eq} \omega_T^2 \quad (3.99)$$

From equations (3.38) and (3.39), equivalent inertia can be derived as:

$$J_{eq} = J_E i_D^2 i_T^2 + J_D i_D^2 + J_T \quad (3.40)$$

From equation (3.40), the effect of gear and final drive ratios can be seen. This expression involves also decreasing inertias of components from engine through rear tires into rear tires. For the front tires, similar method should be used to define the equivalent inertia to express inertias of front tires, brake disks and rotating axes in

terms of front tires. This is also included in the model and independent of the gear state. Equivalent inertias used in this model are given in Table 3.1.

Table 3.1 : Equivalent inertia values [48].

Gear	$J_{eq,r}$ (kg.m ²)
1	27.77
2	11.39
3	6.03
4	4.23

Finally, it should not be forgotten that the losses due to friction and gear backlashes in the driveline results in change of transmitted torque to the wheels. Thus, net torque can be expressed as follows:

$$T_{net,engine} = T_E \cdot i_D \cdot i_T \cdot \eta_T \cdot \eta_D \quad (3.41)$$

where η_T and η_D denotes the transmission and differential efficiencies, respectively. Gearbox efficiencies are defined as approximately 90% in the literature, so these values are taken as 0.9.

3.1.5 Transmission model

It is a must for a Stop and Go control system that the transmission is automatic. Automatic transmission model and gearshift logic is modeled in SIMULINK/Stateflow. The base for the gearshift is a gearshift map. This map foresees increasing or decreasing of the gear by considering the throttle position. Figure 3.7 illustrates the gearshift map.

According to Figure 3.7, gearshift occurs at only one vehicle speed for a desired gear and throttle position. There is hysteresis between the up and down curves of the gearshifts to prevent any continuous gearshifts.

As previously mentioned, gearshift logic according to the gearshift map is modeled in Stateflow. Figure 3.8 shows Simulink model for the Automatic transmission.

In Figure 3.8, threshold calculation block includes the gearshift map shown in Figure 3.7 as a look-up table whose inputs are the current gear state and throttle position, and outputs are up and down thresholds. These threshold values are fed to the shift-logic Stateflow block with vehicle speed (in km/h) for the gear state calculation. Then, gear ratio is determined with a look-up table by inputting the gear state.

Looking at the overall Automatic transmission model, inputs are the vehicle speed in km/h and throttle, whereas the outputs are the gear state and gear ratio.

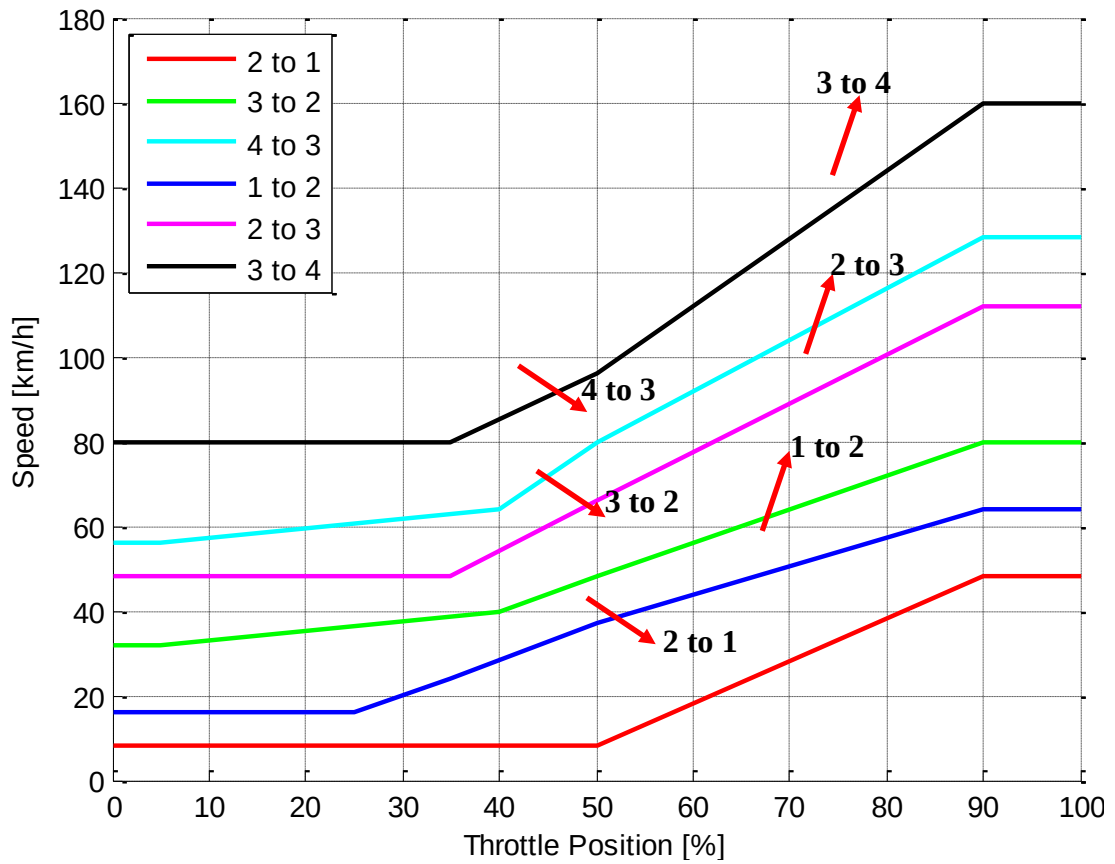


Figure 3.7 : Gearshift map.

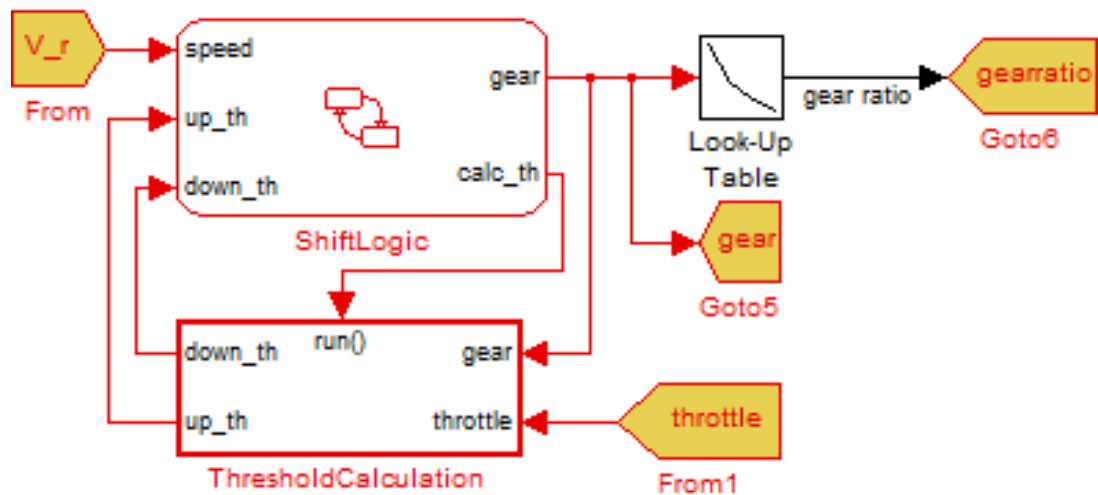


Figure 3.8 : Simulink model for the Automatic transmission.

Gear ratios used in the model is gathered from the same reference for the equivalent inertia values in [48]. Here, a four-speed automatic transmission is used in the model

which has been a wide usage up to recent years. Table 3.2 shows the gear ratios used in the automatic transmission.

Table 3.2 : Gear ratios [48].

Gear	i_T
1	3.154
2	1.913
3	1.281
4	0.951

Stateflow block shown in Figure 3.8 can be seen in detail in Figure 3.9. The block consists of two states that are parallel to each other. The first state is responsible for sending the current gear state, whereas the second state is responsible for deciding the gearshift and makes the first state act accordingly. The second state starts simulation in steady state and behaves according to the current vehicle speed and gear state.

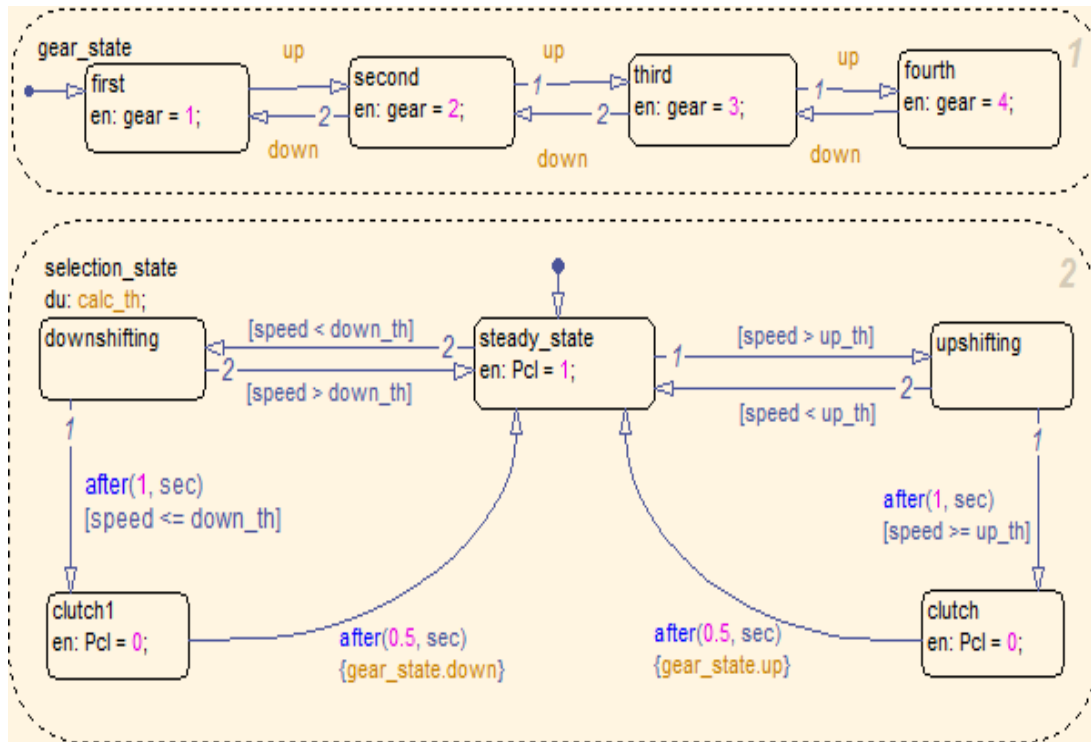


Figure 3.9 : Automatic gearshift logic.

When gear state increase is considered in state-2, the state comes from steady_state substate to upshifting substate. After 1 second wait, if there is no decrease in speed below up-threshold, state-1 is alerted for gear-state up 0.5 second later, which represents the clutch in the transmission. Final reach point is again steady_state. Total 1.5 seconds wait represents real life gearshift procedure.

3.1.6 Resistance forces

Resistance forces should also be included in the vehicle model to be able to represent a real vehicle as much as possible. Although the resistance forces act on different locations of the vehicle, they are assumed to act at the CG of the vehicle. Rolling resistance, slope (gradient) resistance, aerodynamic drag resistance and acceleration resistance can be counted under Resistance forces. Figure 3.10 shows general view of important resistance forces.

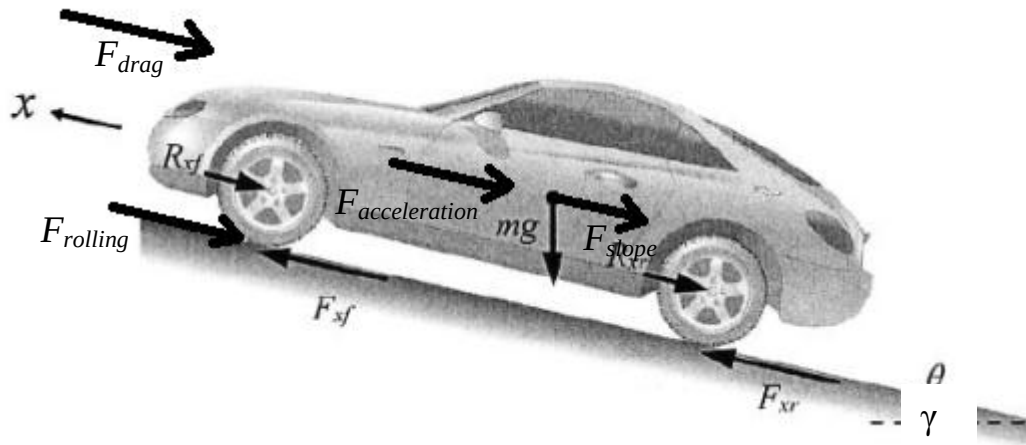


Figure 3.10 : Resistance forces [49].

3.1.6.1 Tire rolling resistance

Tire rolling resistance stems from the deformation in the contact patch for the tire and the road while the tire is rotating. Deformation of the road can be neglected, but the tire is elastic and new material from the tire continuously enters the contact patch as the tire rotates. This material is deflected vertically as it goes through the contact patch and then springs back to its original shape after it leaves the contact patch. Due to the internal damping of the tire material, the energy spent in deforming the tire material is not completely recovered when the material returns to its original shape. This loss of energy can be represented by a force on the tires called the rolling resistance. This resistance can be expressed by the following equation;

$$F_{rolling} = C_{rolling} m.g \quad (3.42)$$

where $C_{rolling}$ is the rolling resistance coefficient, m is the vehicle mass and g is the gravitational acceleration.

The loss of energy in tire deformation also results in a non-symmetric distribution of the normal tire load over the contact patch. Moment due to this unbalanced load is balanced by the moment due to rolling resistance as shown in Figure 3.11.

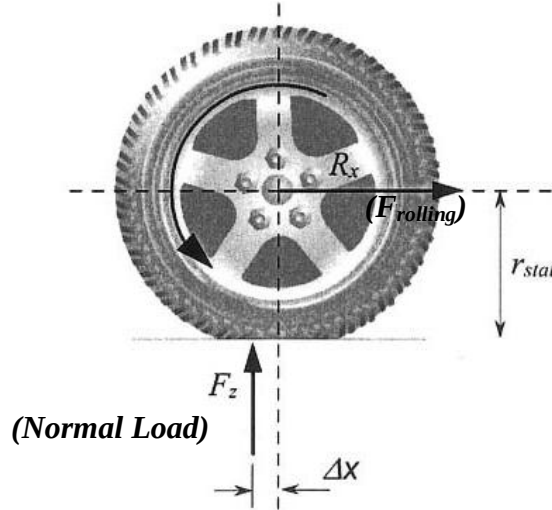


Figure 3.11 : Description of rolling resistance [49].

3.1.6.2 Slope resistance

Slope resistance is the resistance force acting on the vehicle due to its own weight when there is a slope (γ) on the road. This force can be expressed with the following equation:

$$F_{slope} = m.g.\sin(\gamma) \quad (3.43)$$

3.1.6.3 Aerodynamic drag resistance

Aerodynamic drag resistance occurs due to friction by air around the vehicle. This is expressed by:

$$F_{drag} = \frac{1}{2} C_{drag} \rho A V^2 \quad (3.44)$$

where C_{drag} is air resistance coefficient, ρ is air density, A is vehicle frontal area and V is the vehicle speed. ρ can change with atmospheric conditions, but 1.225 kg/m^3 is used in average. A can be found approximately with the formula below for the vehicles between 800-1200 kg [49]:

$$A = 1.6 + 0.00056(m - 765) \quad (3.45)$$

C_{drag} changes with vehicle geometry, and it is said to be taken as 0.35 for a mid-sized passenger car in the literature.

3.1.6.4 Acceleration resistance

Acceleration resistance is the inertial effect of rotating parts in the vehicle. This inertial effect of an accelerating vehicle is given by:

$$F_{acceleration} = \lambda m \ddot{x} \quad (3.46)$$

When net forces acting on the vehicle is written according to D'Alembert principle:

$$\sum F_x + \sum F_{resistance} = 0 \quad (3.47)$$

$$\Rightarrow \sum F_x + \sum F_{drag} + F_{rolling} + F_{slope} + F_{acceleration} = 0 \quad (3.48)$$

Substituting equation (3.46) into equation (3.48) and rearranging gives:

$$\sum F_x + \sum F_{drag} + F_{rolling} + F_{slope} = \lambda m \ddot{x} \quad (3.49)$$

This expression adds the inertial effects of rotating masses in the vehicle to inertial effect of vehicle mass. λ is rotating mass factor and can be expressed by;

$$\lambda = 1 + \sum_{j=1}^n \frac{J_{T,j}}{r_j \cdot R_j} \bigg/ m \quad (3.50)$$

where $J_{T,j}$ expresses all rotating mass inertias, r_j is static wheel radius, R_j is the dynamic wheel radius and m is the vehicle mass. In the right-hand side of the equality, the first term 1 represents the inertia of the vehicle mass and the other term represents inertia of rotating masses. When the second term is written again for all rotating parts;

$$\sum_{j=1}^n \frac{J_{T,j}}{r_j \cdot R_j} = \frac{J_{T,f}}{r_f \cdot R_f} + \frac{J_{T,r} + i_D^2 J_D + i_D^2 i_T^2 J_E}{r_r \cdot R_r} \quad (3.51)$$

is obtained. Here, $J_{T,f}$ and $J_{T,r}$ represent front and rear tires inertial moments, J_D and J_E are differential system and engine inertial moments, r_f and r_r are front and rear tires static radius, and R_f and R_r are front and rear tires dynamic radius, respectively. Static and dynamic radiuses are so close to each other, so can be assumed as equal.

Equation (3.51) shows that λ changes with the gear ratios. Also, the second term is the expression for the equivalent inertial moment. So, the following form can be written:

$$\lambda = 1 + \left(\frac{J_{eq,f}}{r_f \cdot R_f} + \frac{J_{eq,r}}{r_r \cdot R_r} \right) / m \quad (3.52)$$

Rotating mass factors calculated for all gear ratios and obtained with the data in Table 3.1 are given in Table 3.3 below.

Table 3.3 : Rotating mass factors (λ).

Gear	λ
1	1.33
2	1.145
3	1.085
4	1.065

3.1.7 Net moment on the tires

Rotational velocity of the tires can be found by writing moment balance at front and rear tires. Equivalent inertia values for the front and rear tires are used to find this rotational velocity. Related moment balance for the front and rear tires can be written for a rear-wheel driven vehicle according to the Figure 3.10 as follows:

$$T_{net,r} = T_{engine} - T_{brake} - F_{xr} \cdot r_r = J_{eq,r} \cdot \dot{\omega}_{T,r} \quad (3.53)$$

$$T_{net,f} = -T_{brake} - F_{xf} \cdot r_f = J_{eq,f} \cdot \dot{\omega}_{T,f} \quad (3.54)$$

As can be seen from the equations above, engine torque acts only on the rear tires whereas brakes act both on front and rear tires. Regarding the brakes, a detailed dynamic model is not used in the vehicle model. Instead, a first order system with 0.1 second time constant is used to represent the brakes. In addition, longitudinal tire forces make frictional effect on the tires and result in negative torque when multiplied by static tire radius.

Rotational velocity of the tires can be found by integrating the rotational acceleration terms in equations (3.53) and (3.54). It should be noted again that the resistance forces are considered and subtracted from the longitudinal forces in these equations.

Distance traveled in X and Y coordinates in the global coordinate system can also be found for the vehicle simulations with the following equations:

$$\Delta x = \int_0^{t_f} V \cos(\beta + \psi) dt \quad (3.55)$$

$$\Delta y = \int_0^{t_f} V \sin(\beta + \psi) dt \quad (3.56)$$

3.2 Vehicle Modeling in MATLAB/Simulink

All dynamic equations and expressions obtained in Section 3.1 are brought together by the help of MATLAB/Simulink. Sets of equations for a certain mechanical system are grouped as a subsystem, and outputs are sent to the related subsystem if required. Figure 3.12 illustrates the vehicle model in Simulink including all subsystems.

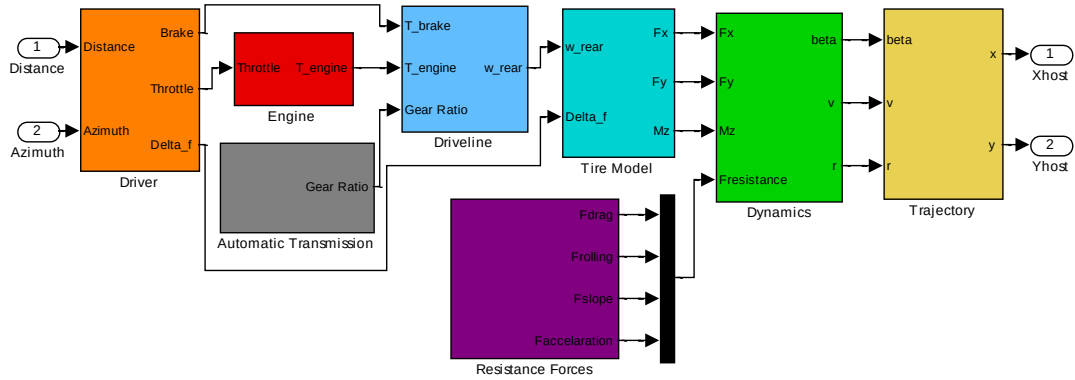


Figure 3.12 : Vehicle model in MATLAB/Simulink.

In addition to the mentioned systems in Section 3.1, driver model is added to the vehicle model as a subsystem to complete the model in terms of driver inputs as throttle, brake and steering. Input to the driver is distance and azimuth data from a radar which will be discussed later in the thesis. The driver model also includes stop and go controllers, which will also be discussed later.

3.2.1 Simulation results obtained with the vehicle model

In this section, different maneuvers such as throttle, brake and steering are given to the vehicle as inputs, and results are obtained and presented. Presented outputs are, for example, vehicle speed, engine speed, gearshifts, and tire rotational speeds.

As the first maneuver, throttle input shown in Figure 3.13 is applied to the vehicle at $t = 0$ and initial vehicle speed is 54 km/h.

Vehicle speed and engine speed obtained with this given throttle position can be seen in Figure 3.14 and Figure 3.15, respectively.

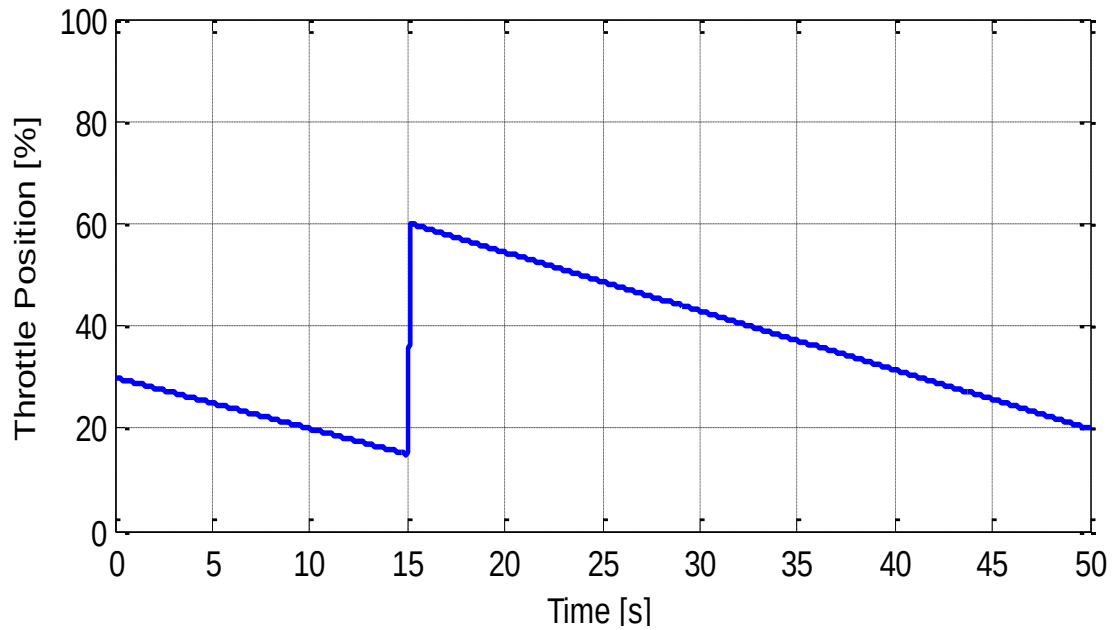


Figure 3.13 : Throttle position.

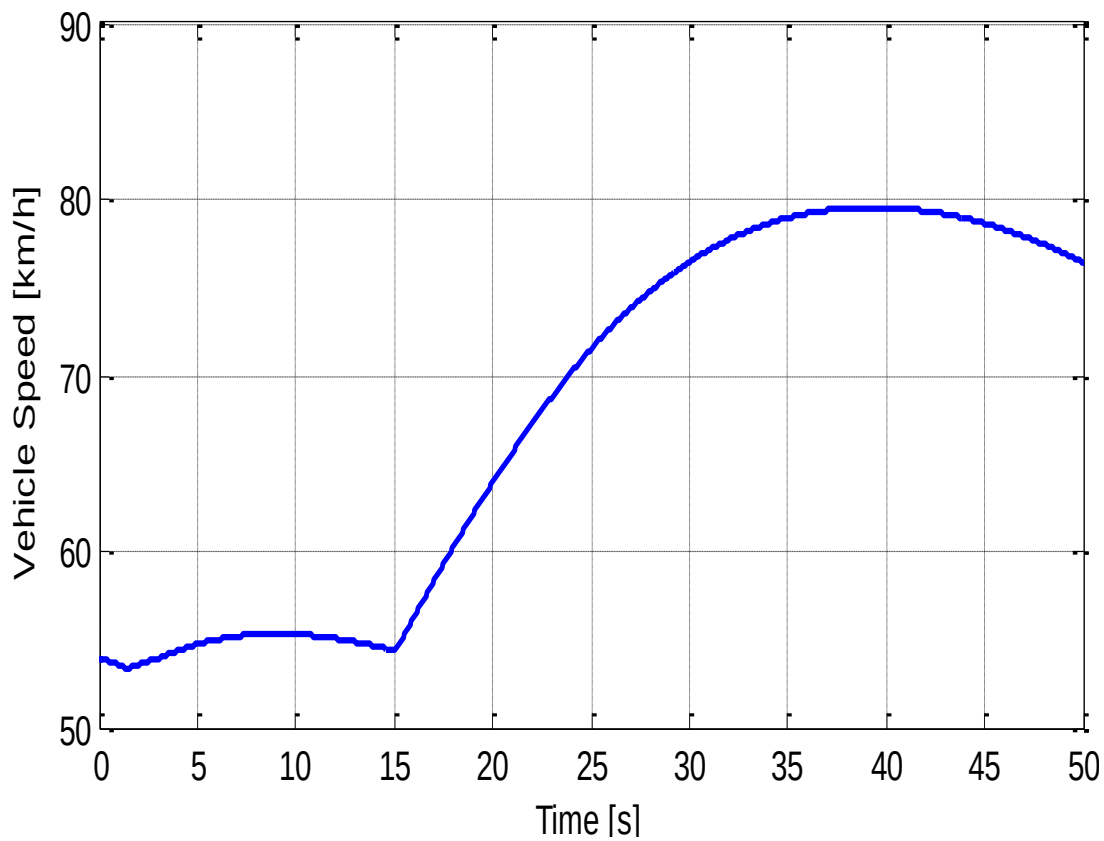


Figure 3.14 : Vehicle speed.

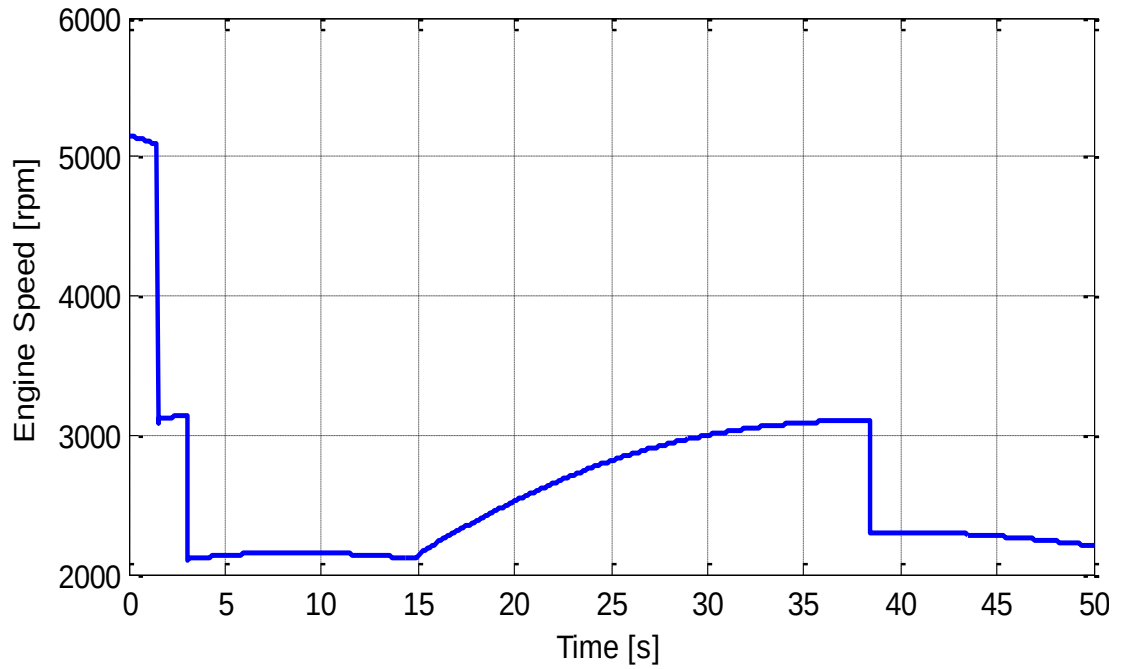


Figure 3.15 : Engine speed.

In Figure 3.15, instant changes represent gearshifts, which can be shown in Figure 3.16. Gearshifts occur at around $t = 1.5, 3$ and 38 seconds. The other abrupt change in Figure 3.15 occurs at $t = 15$ seconds, which represents the abrupt change in throttle position at that time (see Figure 3.13).

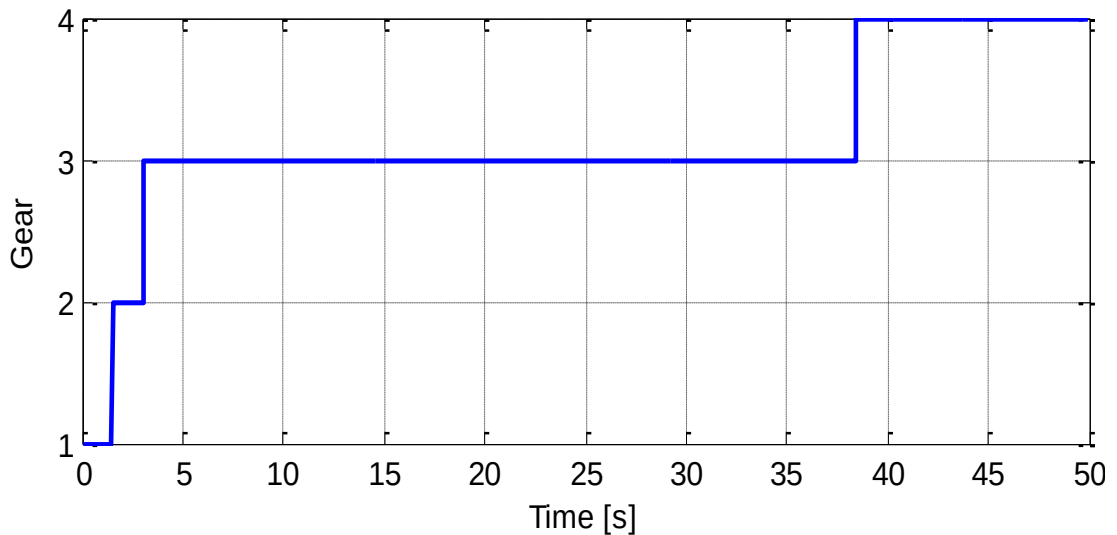


Figure 3.16 : Gearshifts.

Rotational speed and linear speed of the rear and front tires are presented in Figure 3.17 and Figure 3.18, respectively. Rotational speed of the tires is multiplied by the tire radius to be able to compare it with linear speed.

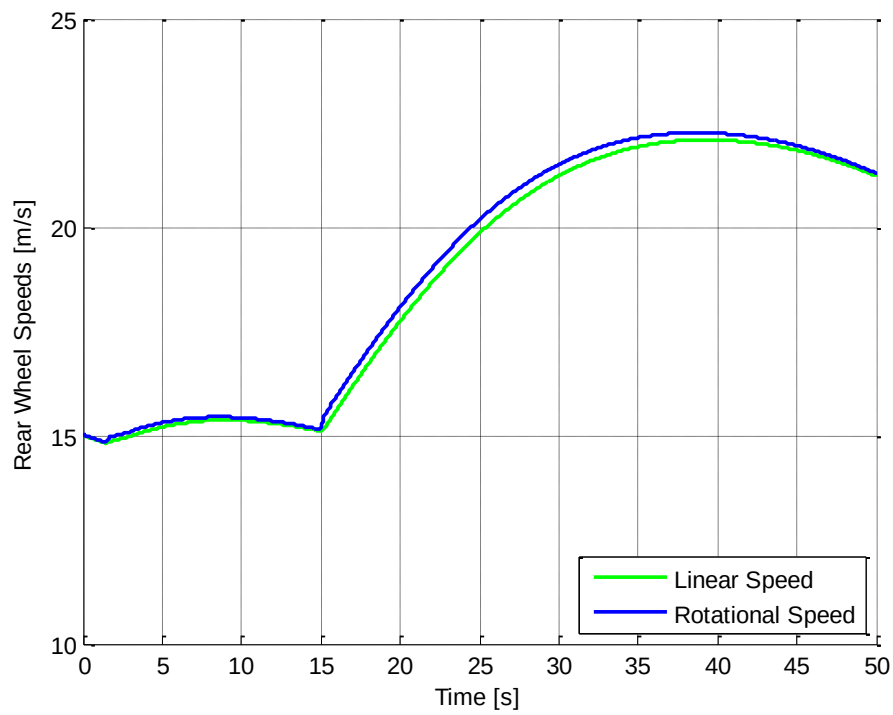


Figure 3.17 : Rotational and linear speeds for the rear tire.

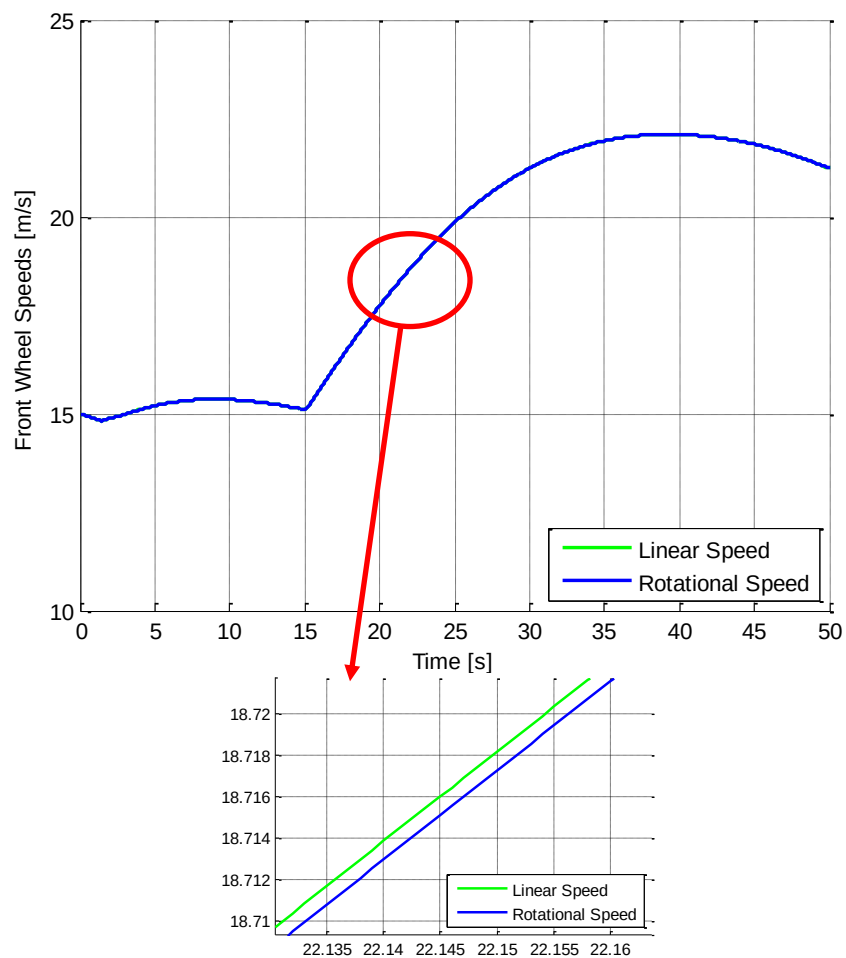


Figure 3.18 : Rotational and linear speeds for the front tire.

As seen in Figure 3.18, although the rotational and linear speeds for the front tire seem on top of each other very little difference can be seen when it is zoomed. Since the rear tires are drive wheels, slip is more significant at rear tires. Slip values for the rear and front tires can be seen in Figure 3.19 and Figure 3.20, respectively.

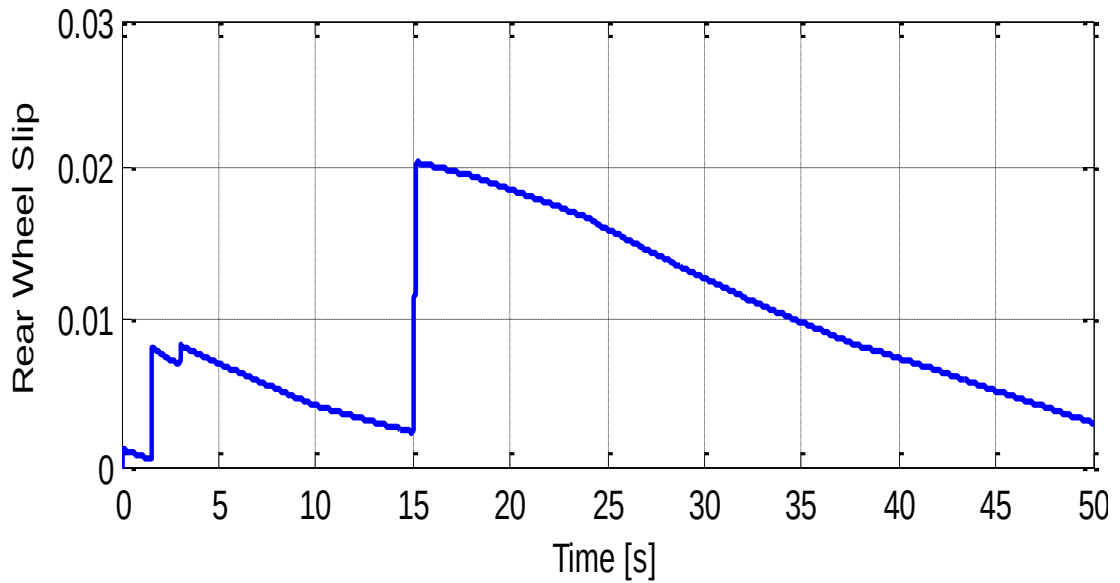


Figure 3.19 : Slip at the rear tire.

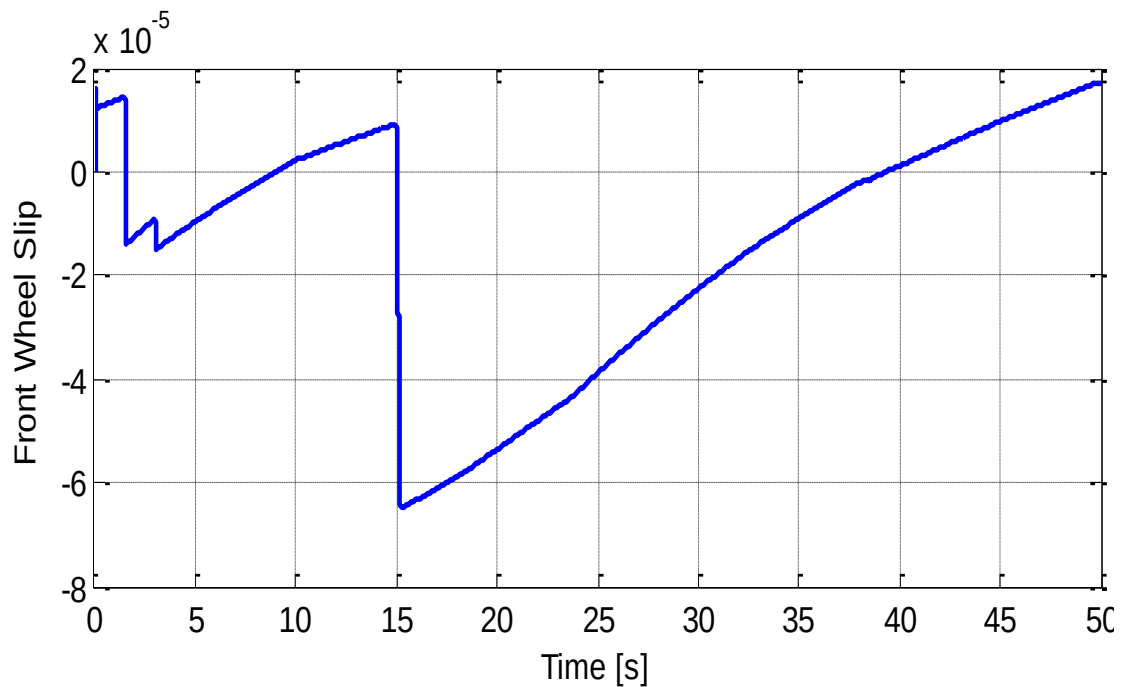


Figure 3.20 : Slip at the front tire.

The second maneuver is a hard brake application. Throttle and brake inputs can be seen in Figures 3.21 and 3.22, respectively. Initial vehicle speed is again 54 km/h.

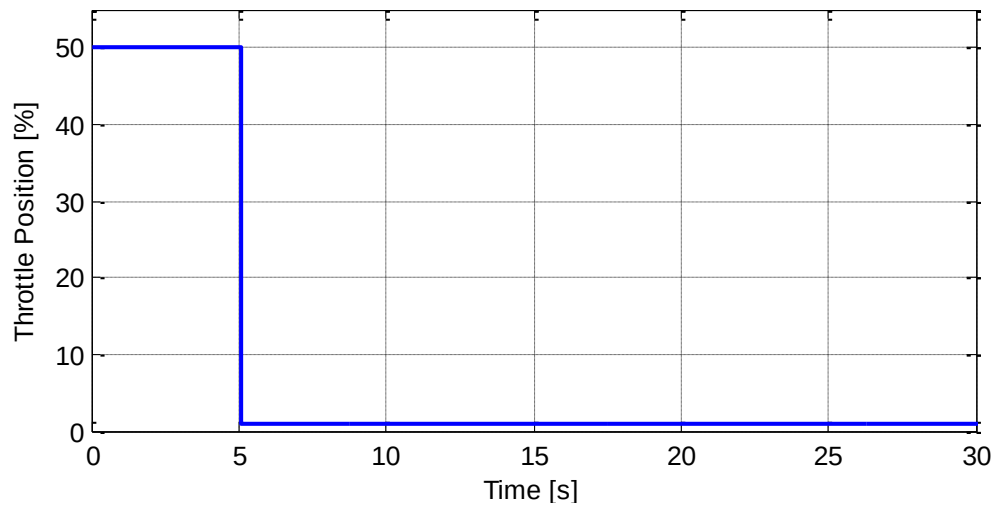


Figure 3.21 : Throttle position.

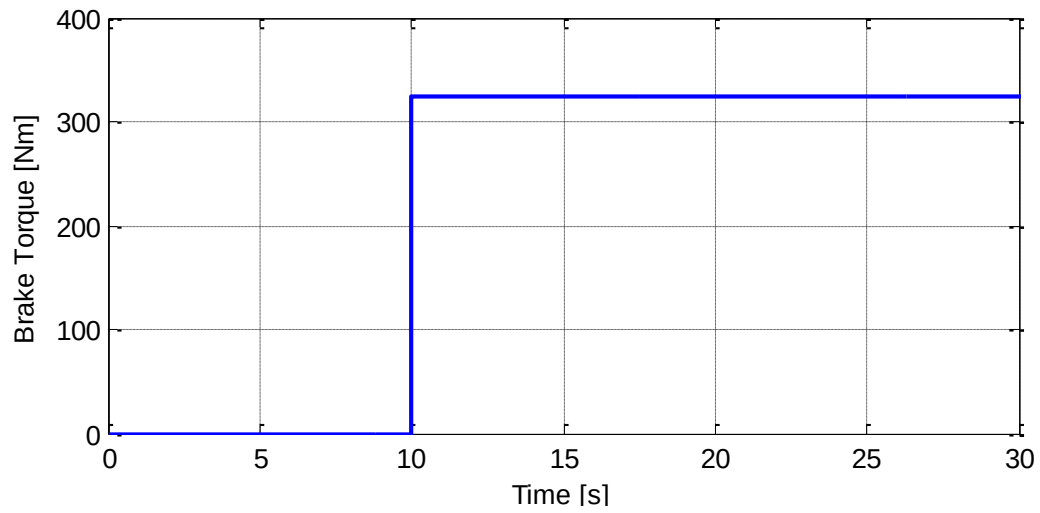


Figure 3.22 : Brake torque.

Vehicle speed obtained by applying this hard brake maneuver can be seen in Figure 3.23. The vehicle comes to a full stop at the time $t = 27$ seconds.

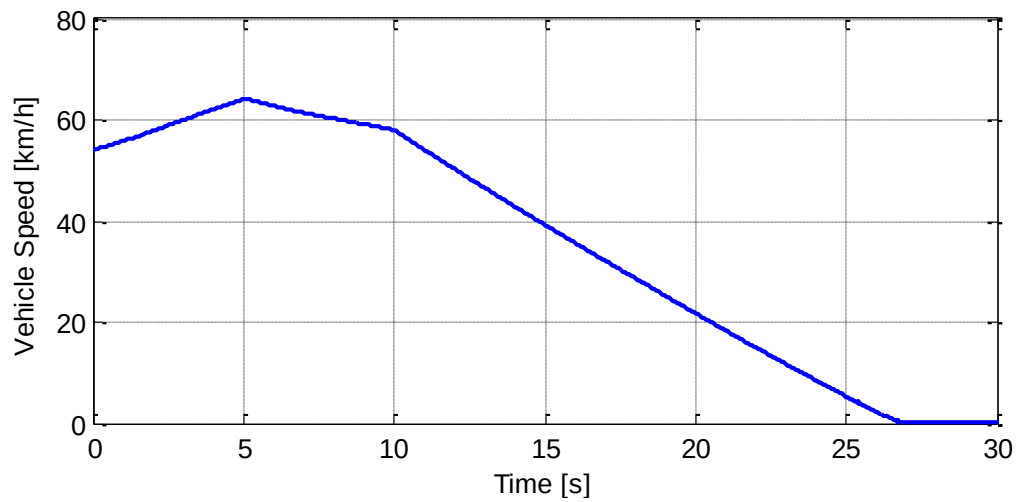


Figure 3.23 : Vehicle speed.

Two different steering maneuvers are applied to the vehicle to investigate lateral response behavior.

The first lateral maneuver is the steering input shown in Figure 3.24. The inputs given when the vehicle speed is 54 km/h with 20% throttle position.

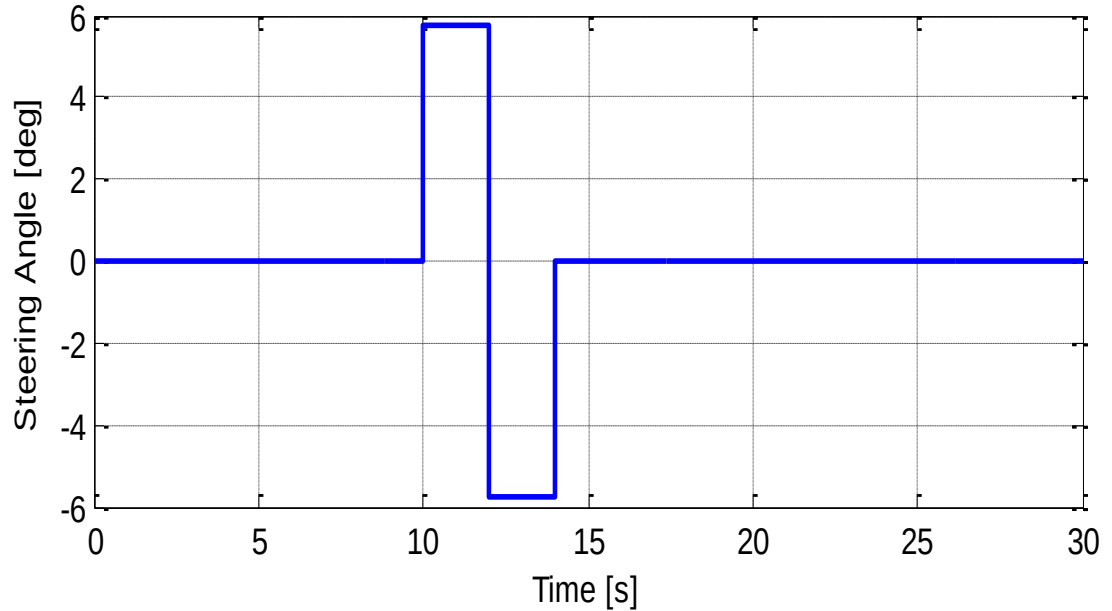


Figure 3.24 : Front wheel steering angle.

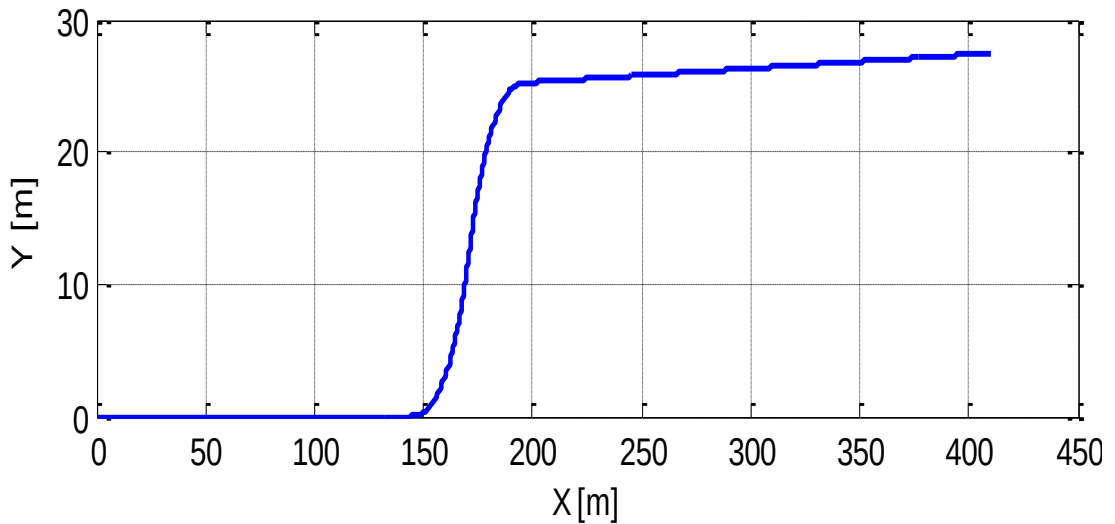


Figure 3.25 : Vehicle coordinates.

Coordinates of the vehicle during the steering maneuver can be seen in Figure 3.25 above.

Vehicle speed oscillates during the steering maneuver because some part of longitudinal force is spent for steering. The oscillation in vehicle speed is shown in Figure 3.26.

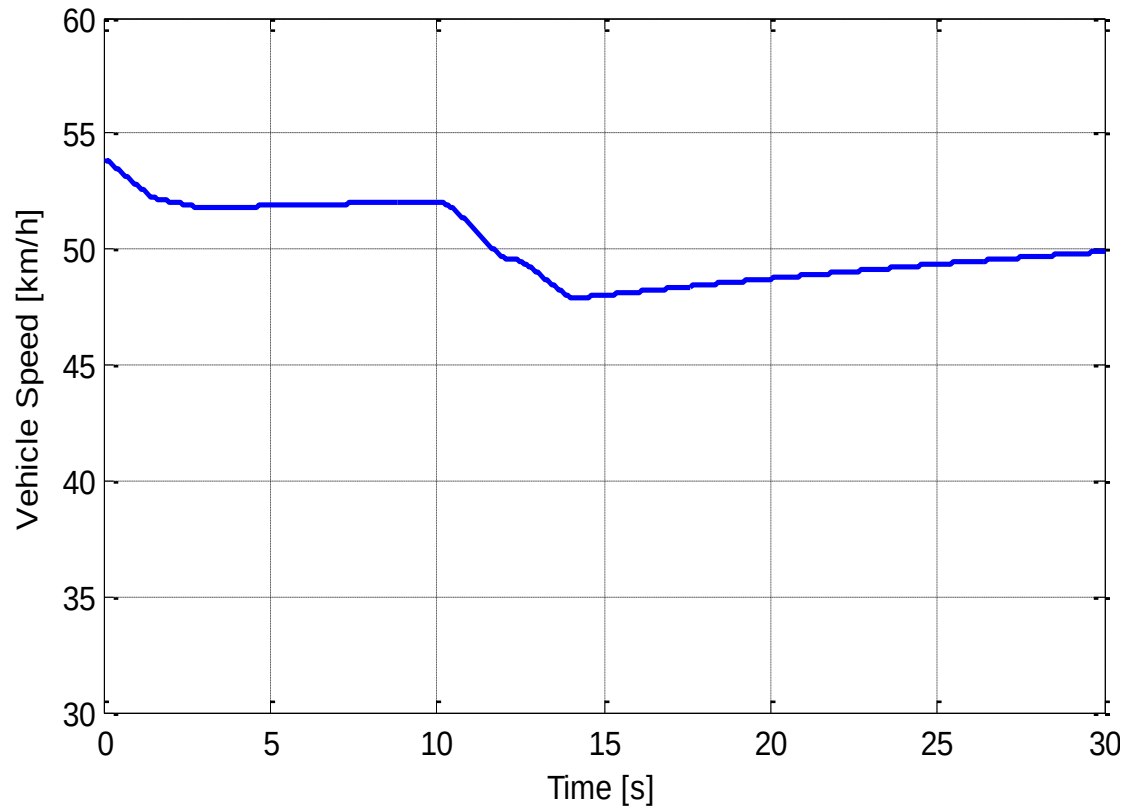


Figure 3.26 : Vehicle speed.

This steering maneuver likes lane change, and resulting yaw rate can be seen in Figure 3.27.

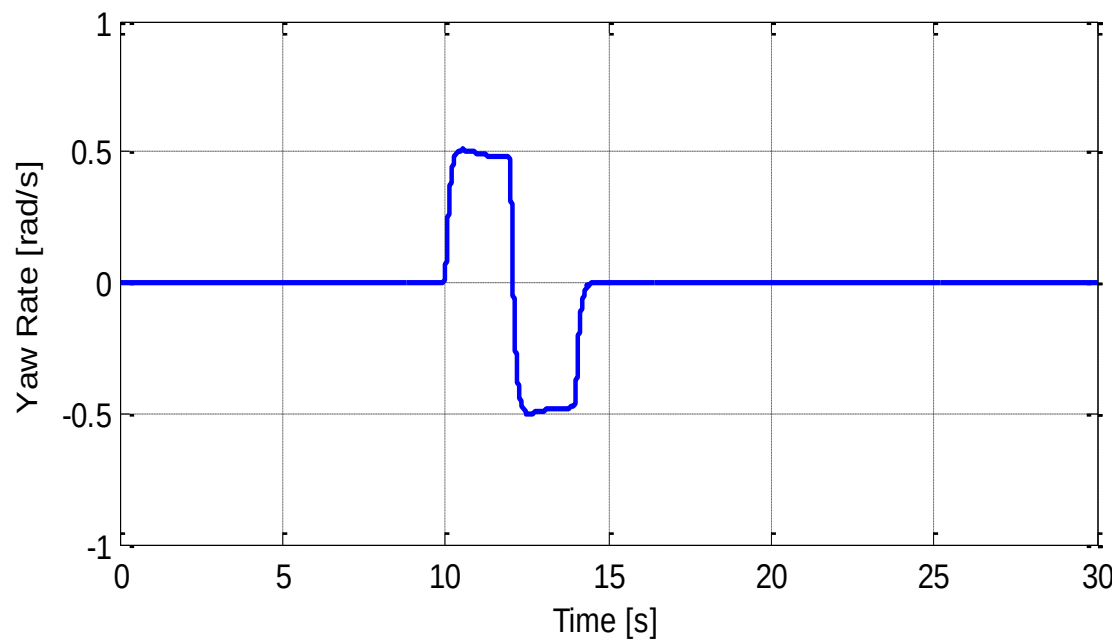


Figure 3.27 : Yaw rate of the vehicle.

This maneuver also increases longitudinal slip, which can be seen in Figure 3.28.

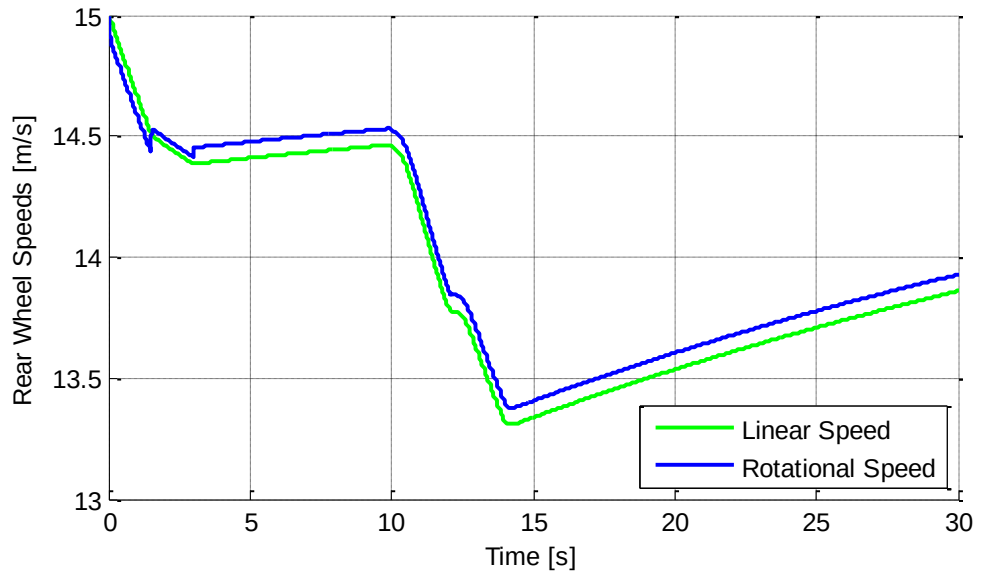


Figure 3.28 : Rotational and linear speeds for the rear tire.

Finally, the steering input shown in Figure 3.29 is applied to the vehicle at a constant speed and constant throttle.

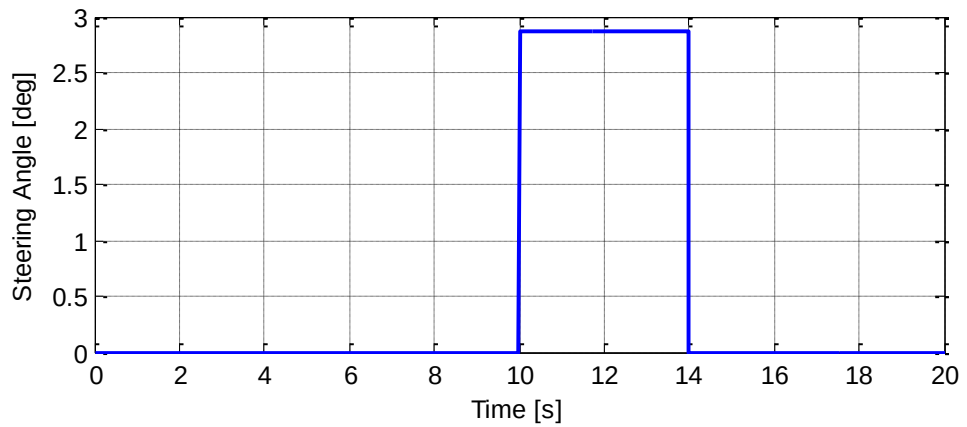


Figure 3.29 : Front wheel steering angle.

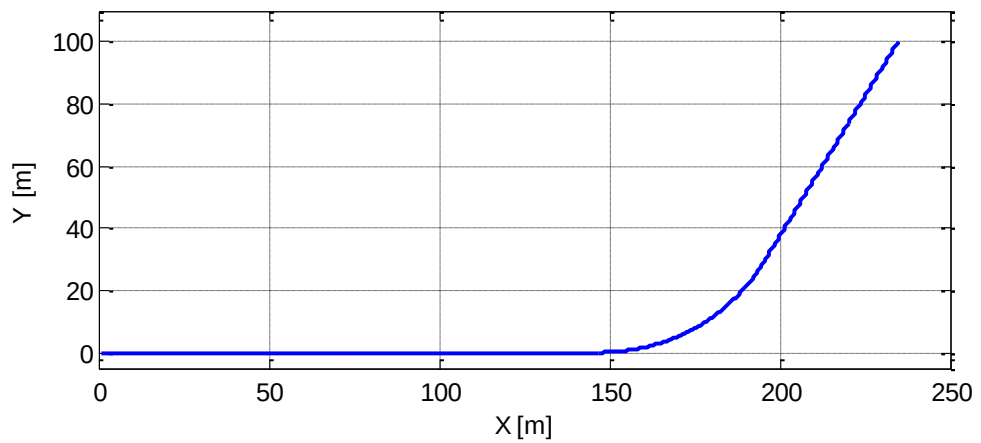


Figure 3.30 : Vehicle coordinates.

Coordinates of the vehicle during this steering maneuver can be seen in Figure 3.30 above.

This steering maneuver likes cornering maneuver, and resulting yaw rate can be seen in Figure 3.31.

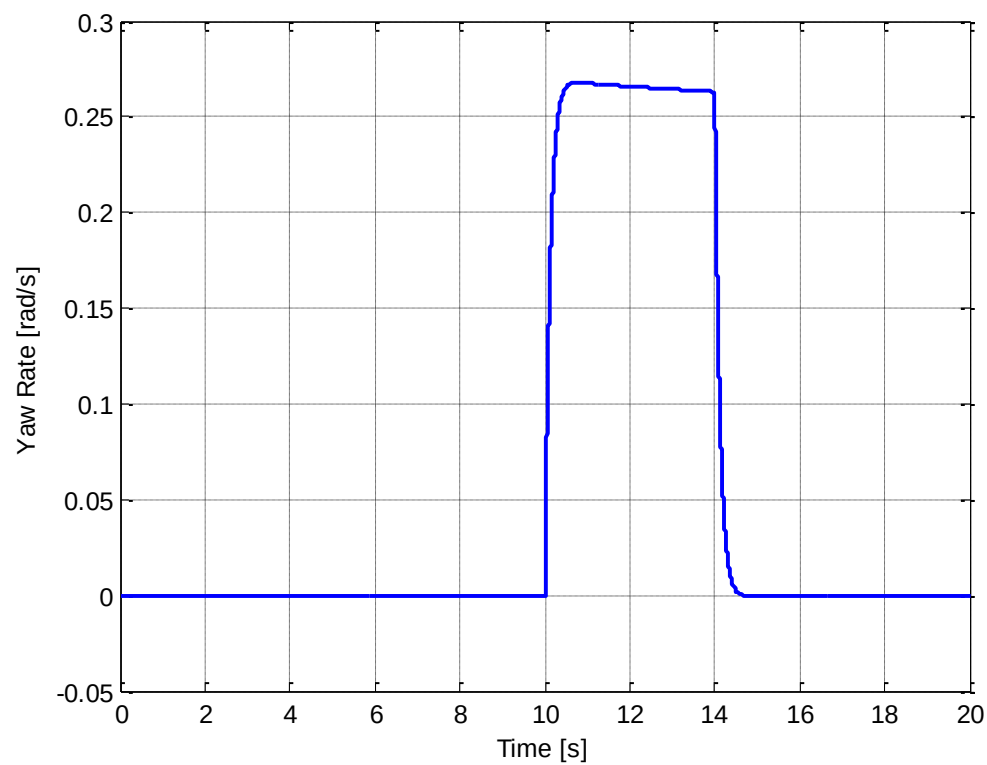


Figure 3.31 : Yaw rate of the vehicle.

4. STOP AND GO CONTROL

Stop and Go Control is a distance and speed control with respect to target vehicles, as already explained in the previous chapters. Designing and modeling a control structure for this system, and providing a good cooperation with the vehicle model subsystems are crucial for testing the design made in terms of reality. Ability of the control structure to intervene into the physical inputs of the vehicle model is also critical for the similar reasons. The control structure should act to the vehicle model as in reality by the help of mechanical systems of the vehicle, not with direct force or another similar direct input. The control structure should also open to human interaction considering any possible human in the loop (HIL) simulator application. Stop and go control system modeled according to the above mentioned criteria is illustrated in Figure 4.1.

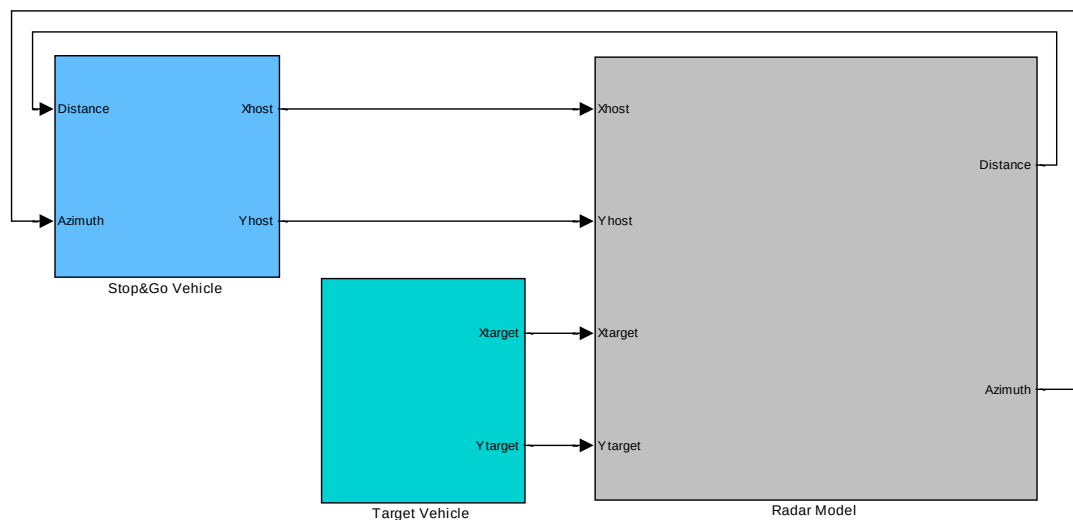


Figure 4.1 : Stop and go control system Simulink model.

As can be seen from the figure above, there are a host vehicle, a target vehicle and a radar model in the system. Host vehicle is equipped with stop and go control system, and target vehicle is added to the system to apply required scenarios for the host vehicle. Radar model is also included in the system for more realistic model representation.

Control structure will be explained in detail later in this chapter. As a short introduction, it can be said that the control structure consists of a hierarchical structure. Lower-level part of this structure includes a continuous and classical control algorithm, whereas higher-level controller includes a discrete and state driven algorithm.

First, scenarios for the stop and go control will be determined in this chapter, where control structure will be presented. Then, sensor model and control structure will be examined.

4.1 Stop and Go Scenarios

The stop and go system is subjected to some specific scenarios in terms of testing the system. The aim of testing the system with these maneuvers is to investigate how the system behaves in real urban traffic conditions. The most common situations are thus composed that can be seen in stop and go driving. Scenarios also determine the limits of the system design, and suitable precautions are taken with the knowledge of situations in advance that may be subjected. In fact, although these scenarios represent most of the situations that the stop and go system may be subjected, investigating and including all possible scenarios is impossible and not in the scope of this thesis. Performance and applicability of the system should be evaluated within the limits of these scenarios.

Eight different scenarios are foreseen for the stop and go driving and illustrated in Table 4.1. Scenarios are explained in detail below.

4.1.1 Scenario 1

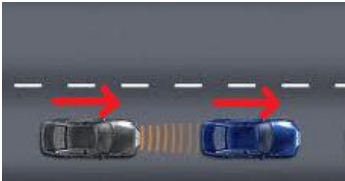
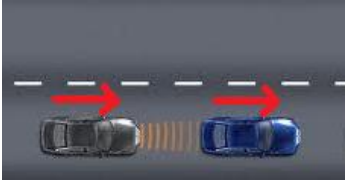
Scenario 1 starts with cruise control mode at a desired speed, and when a target vehicle with lower speed is detected within the radar range, following starts.

4.1.2 Scenario 2

In scenario 2, target vehicle increases its speed during following. When the target vehicle's speed increases, stop and go vehicle's speed increases, too in a convenient manner.

Since inter-distance is defined with time headway concept, inter-distance also increases, accordingly.

Table 4.1 : Stop and go system scenarios.

Scenario	S&G State	Condition	Illustration
1	t = 0: Off t = T: On	Target vehicle is detected	
2	On	Target vehicle accelerates during following	
3	On	Target vehicle slows down during following	
4	On	Cut-in vehicle appears	
5	t = 0: On t = T: Off	Target vehicle changes lane	
6	t = 0: On t = T: Off	Host vehicle changes lane	
7	On	Target vehicle stops	
8	On	Target vehicle stops and goes	

4.1.3 Scenario 3

In scenario 3, target vehicle decreases its speed during following. When the target vehicle's speed decreases, stop and go vehicle's speed decreases, too in a convenient manner. If the speed decreases below a certain limit, stopping scenarios become active. Inter-distance also decreases, accordingly.

4.1.4 Scenario 4

A new vehicle arises in scenario 4 with a cut-in between the host and target vehicles. In this case, cut-in vehicle will be the new target vehicle and following continues at the cut-in vehicle. For the simulation purposes, instead of adding a third vehicle into the system, target vehicle's position got back to represent the cut-in vehicle.

4.1.5 Scenario 5

Target vehicle changes its lane during following in scenario 5 with steering inputs, and it is no longer a target vehicle. Host vehicle then continues travel with a predefined cruise control speed until a new target vehicle is detected.

4.1.6 Scenario 6

In scenario 6, host vehicle changes lane this time. When the lane change is occurred by steering inputs for the host vehicle, the control system switches to no-target mode and continues travel with a predefined cruise control speed until a new target vehicle is detected.

4.1.7 Scenario 7

Scenario 7 represents the stopping situation. This scenario starts when the vehicle speed falls below approximately 3 m/s. Slowdowns at higher speeds are considered as scenario 3. Full stop is assumed when the vehicle speed falls below approximately 2 m/s. In this situation, headway time also decreases to provide closer follow distances.

4.1.8 Scenario 8

Scenario 8 starts when the stopped vehicle in scenario 7 goes. Repeated stop and go situations are provided to represent the general behavior of vehicles in heavy congested urban traffic.

4.2 Radar Model

Sensor system is one of the most important parts of a stop and go control system. Including sensor system into the model with its physical properties provides the control system's and simulation's reality. Thus, sensor model is included into the system to represent the measurements.

A continuous wave radar sensor is taken as the base for longitudinal sensor modeling in the stop and go control system. Sensor properties are selected by considering available sensor usage in similar applications. Radar range is taken as 40 meter and reach angle is 25 degree. Effect of dynamic behavior of the sensor is not considered during modeling. The reason for this is that sensor dynamics is much quicker when compared to the modeled mechanical systems. Measurement delay of the sensor is also neglected considering that the wave sent is in light speed and signal processing time is low enough to be neglected when compared to the sample time of simulation. In addition, possible error and disturbances are not included in the model. Measurement is assumed as exact and true.

Longitudinal distance, longitudinal relative velocity and azimuth angle are fed to the control system by sensor. Azimuth angle is used to determine the target vehicles position, and it is used in lane change scenarios.

X and Y coordinates of the vehicle is explained in Section 3, and expressed with Equations (3.55) and (3.56). Direction of the vehicle is also expressed with yaw angle ψ , which is the angle between local and global coordinates.

Inputs to the radar block in the vehicle model can be seen in Figure 4.1. In addition to coordinates of host and target vehicles, yaw angle of the host vehicle is also fed to the radar block. The help of these data provides expressing the radar coordinate system in terms of host vehicle coordinate system. The reason of feeding the target vehicle's position is to find the relative position to the host vehicle.

First, target vehicle coordinate system will be expressed with respect to the host vehicle coordinate system by the help of coordinate transformation matrices.

Figure 4.2 presents the global coordinate system, local coordinate system of the host vehicle and position of the target vehicle.

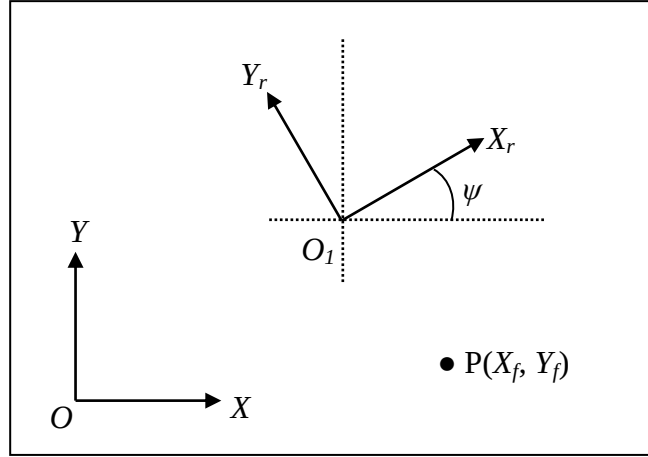


Figure 4.2 : Vehicle coordinates.

The symbols in Figure 4.2 represent the followings; XOY is the global coordinate system, $X_rO_1Y_r$ is the local coordinate system of the host vehicle, $P(X_f, Y_f)$ is the position of the target vehicle, and ψ is the yaw angle of the host vehicle (also the angle between the local and global coordinates).

The aim is to express the target vehicle's position P with respect to the host vehicle coordinate system. Transformation matrix of the host vehicle with respect to the global coordinate system should be written first. General form of the matrix is in the form below:

$$A = \begin{bmatrix} R & \vdots & X_o \\ \dots & \vdots & \dots \\ 0 & \vdots & 1 \end{bmatrix} \quad (4.1)$$

where R is the rotation matrix of the two coordinates expressed by Euler angles with respect to each other. X_o is the location of the origin O_1 with respect to O . Transformation matrix is then given by:

$$A_{\equiv O_1}^O = \begin{bmatrix} \cos(\psi) & -\sin(\psi) & 0 & X_r \\ \sin(\psi) & \cos(\psi) & 0 & Y_r \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 1 \end{bmatrix} \quad (4.2)$$

Coordinate vector of the target vehicle with respect to the global coordinate is also given by the following expression:

$$\vec{X}_f^O = \begin{bmatrix} X_f \\ Y_f \\ 0 \\ 1 \end{bmatrix} \quad (4.3)$$

Coordinates of the target vehicle with respect to global coordinates in terms of host vehicle coordinates and transformation matrix can be expressed by:

$$\vec{X}_f^O = A_{\equiv O_1}^O \cdot \vec{X}_f^{O_1} \quad (4.4)$$

The necessary expression here is $\vec{X}_f^{O_1}$, which can be found by multiplying both sides of Equation (4.4) by inverse of the $A_{\equiv O_1}^O$.

$$\left[A_{\equiv O_1}^O \right]^{-1} \cdot \vec{X}_f^O = \left[A_{\equiv O_1}^O \right]^{-1} \cdot A_{\equiv O_1}^O \cdot \vec{X}_f^{O_1} \Rightarrow \vec{X}_f^{O_1} = \left[A_{\equiv O_1}^O \right]^{-1} \cdot \vec{X}_f^O \quad (4.5)$$

This operation is performed within the radar block in the model as can be seen in Figure 4.3.

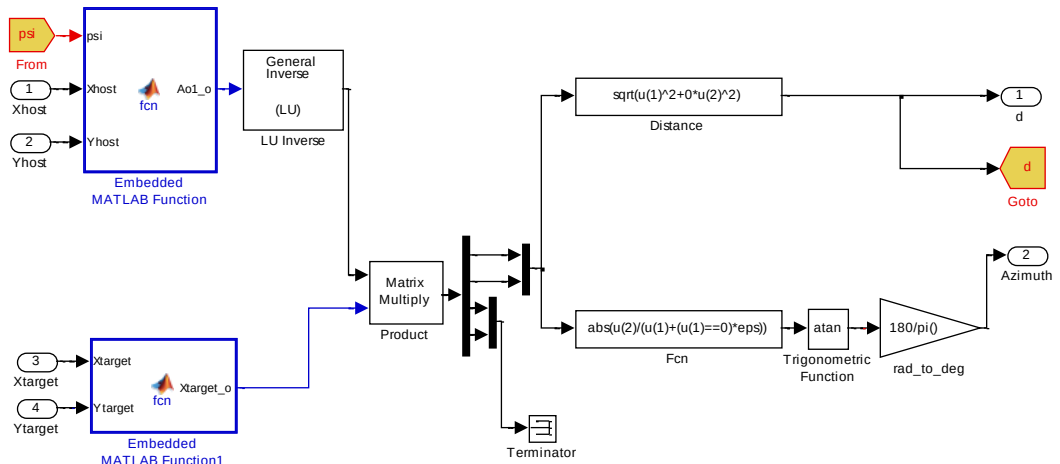


Figure 4.3 : Radar block.

The radar block provides reliable calculation of all types of rotation scenarios of the host vehicle, and distances in x and y coordinates between the two vehicles even when the host vehicle passes the target vehicle.

Azimuth angle is calculated by using x and y outputs with the equation below:

$$Azimuth = \frac{180}{\pi} \arctan\left(\frac{y}{x}\right) \quad (4.6)$$

Distance between the vehicles is also calculated by using x and y outputs with the equation below:

$$d = \sqrt{x^2 + y^2} \quad (4.7)$$

Calculation of relative speed normally performed by differentiating the distance between the vehicles, which is measured by the radar. Since the host vehicle measures its own speed, target vehicle speed can be found by adding the relative speed to host vehicle speed. However, target vehicle speed is fed directly to the radar model for relative speed calculation. Azimuth and distance values are fed to the host vehicle and then to the stop and go control system.

4.3 Control Structure

It should be noted that the system that will be controlled is completely in a nonlinear structure. System nonlinearities have already been discussed in Section 2.3. Engine model, tire model, transmission, and vehicle kinematics can be given as examples to controlled system nonlinearities. Selected control methods and logical operations are also examples to nonlinearities in the control structure.

A hierarchical structure was adopted for the Stop and Go controller design. Two-level of control is applied in this hierarchical structure. A high-level controller determines the scenario that the vehicle is subjected to, and then related low-level controller is triggered. Data measured by the radar are used to guess the scenario with also looking at the yaw rate and steering inputs of the host vehicle. The selected scenario triggers the related low-level controller. One part of the low-level controller determines reference acceleration values, whereas another part of the controller attempts to track these reference acceleration values by modulating throttle and brake applications. With the hierarchical structure, state changes in the high-level controller can prevent low-level controller's outputs.

4.3.1 Low-level control

As explained above, one part of the low-level controller determines reference acceleration values, whereas the other part attempts to track these reference acceleration values by modulating throttle and brake applications. In addition, simple cruise control in no target situation is another part of the low-level controller.

First, the method how the desired acceleration values are converted to throttle and brake inputs will be discussed in this section. Then, principle of calculating the desired acceleration value will be explained. Here, two different control alternatives are used in stop and go mode for calculation of the desired acceleration for comparison. The first one is a PD type speed and distance feedback controller, and the second one is a Fuzzy Logic Controller.

After calculating the desired acceleration, required force at CG of the vehicle to provide this acceleration is found by multiplying the desired acceleration with the vehicle mass. It can be assumed that this force at CG of the vehicle is supplied by the longitudinal force at rear tires, so required torque at the wheels is the multiplication of this force by tire radius. To be able to represent this force with respect to the engine, transmission and final drive ratios are used with transmission efficiencies. Thus, required torque at the engine is calculated. Since the engine speed is known, required throttle can be found by using the engine map.

Mathematical expressions of the mentioned calculations can be given by the following equations:

$$F_{required} = a_{required} \cdot m \quad (4.8)$$

$$T_{required, wheel} = (F_{required} + F_{resistance}) \cdot r_r \quad (4.9)$$

$$T_{required, engine} = \frac{T_{required, wheel}}{i_D \cdot i_T \cdot \eta_D \cdot \eta_T} \quad (4.10)$$

From Equation (4.9), it should be noted that resistance forces should be added to the required force to be able to find the total required force. The engine should give required torque to provide the required acceleration, and also it should cope with resistance forces.

Engine map gives torque values as the output, as previously explained. The inputs are throttle and engine speed. For the control purpose, the engine map will be converted in another form to give throttle position as the output. The inputs will be the torque and engine speed. Interpolation method is used to form the new engine form, which is called Inverse Engine Map.

When the inverse engine map is being formed, all torque values in the engine map are written in the input row of the look-up table. Input columns of the look-up table is filled with engine speed. All known throttle positions from the engine map is written as the output cells of the look-up table. While torque T_1 corresponds to the inputs engine speed ω_1 and throttle θ_1 in the engine map, throttle θ_1 corresponds to the inputs engine speed ω_1 and torque T_1 in the inverse engine map. Since the engine map does not provide throttle positions for all engine speed and torque combinations, inverse engine map cannot be filled completely. The empty spaces are filled by interpolation method. For example, given $T_1 < T_3 < T_2$ with ω_1 - θ_1 - T_1 and ω_1 - θ_2 - T_2 combinations, in which only T_3 is unknown. θ_3 can be found with the following interpolation expression:

$$\theta_3 = \frac{T_3 - T_1}{T_2 - T_1} \cdot (\theta_2 - \theta_1) + \theta_1 \quad (4.11)$$

All empty cells are filled with this interpolation method. However, the engine cannot give some torque values at some engine speeds. For example, maximum throttle position is not seen at maximum engine torque at low engine speeds. So, for the higher torque values, 100% throttle position is written. Similar situation occurs for the torque values below the minimum throttle position. 0% throttle position is written in this case.

2D and 3D inverse engine map is shown in Figures 4.4 and 4.5, respectively.

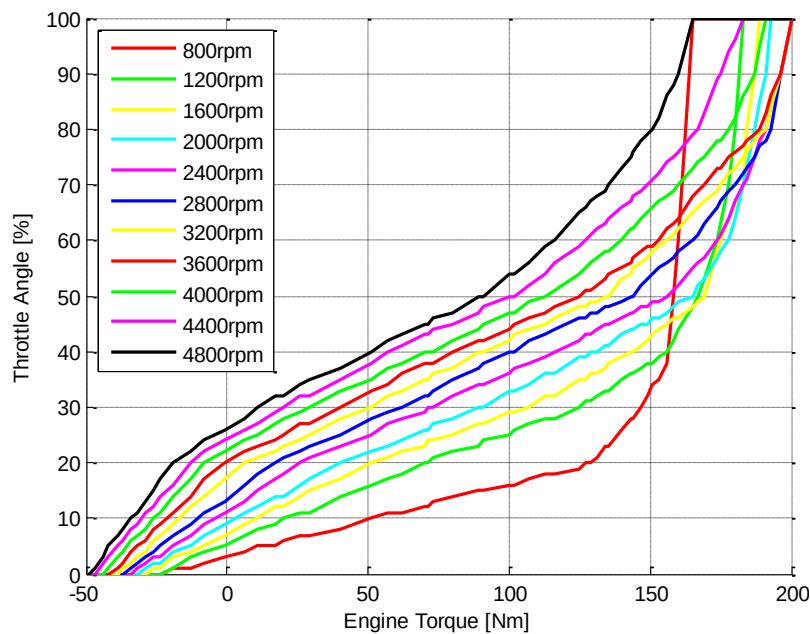


Figure 4.4 : 2D inverse engine map.

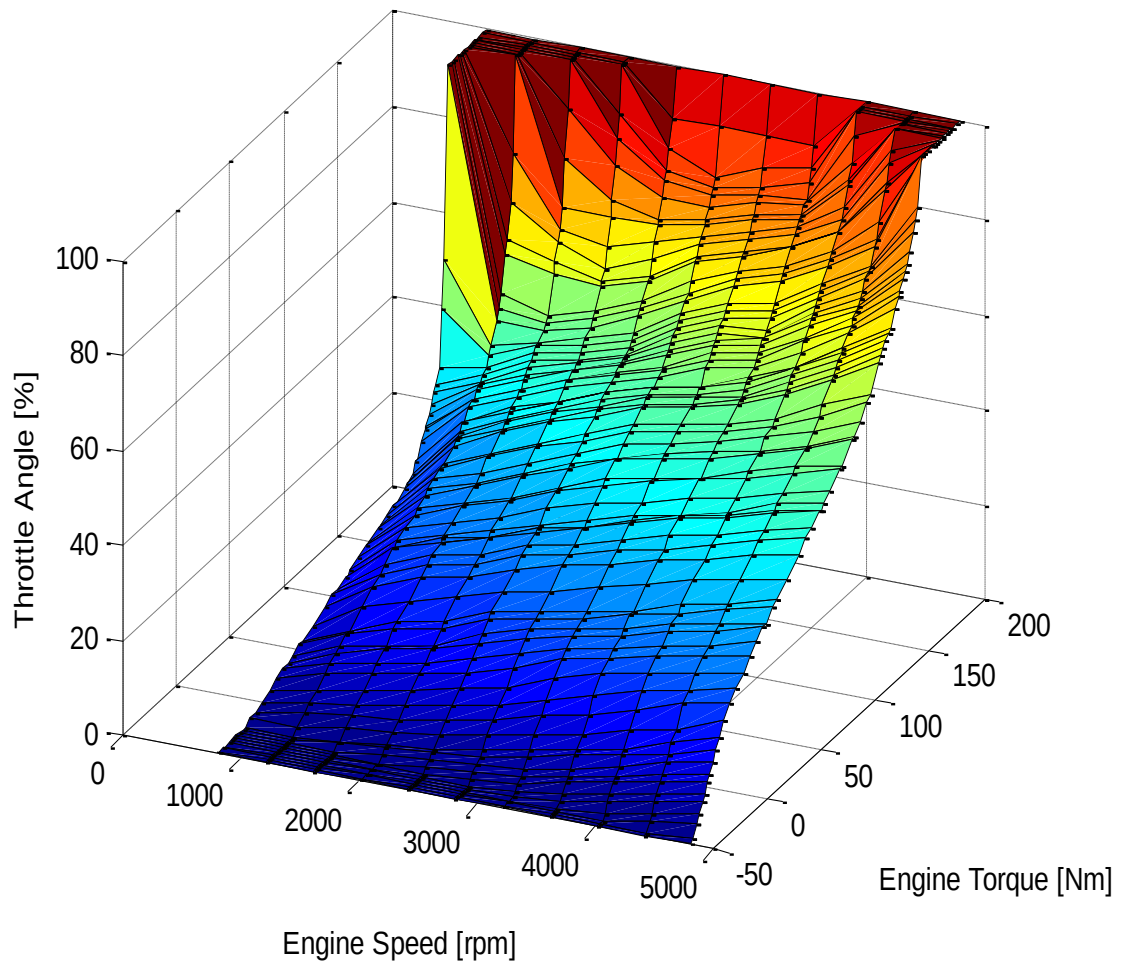


Figure 4.5 : 3D inverse engine map.

4.3.1.1 Cruise control

The first controller in the low-level control is cruise controller, which sets the vehicle speed at a desired value. This controller is activated when there is no target vehicle or target vehicle is no longer a target. Driver can also activate the cruise controller manually.

Control structure is a PID type controller and is based on inverse engine map logic. Controller gives required throttle inputs to cope with the resistance forces to maintain the desired speed. The error between the set speed and the vehicle speed is multiplied by PID coefficients to find required force.

Required force is supplied by the engine or brakes, according to the error is positive or negative. However, since the engine can supply negative torque at low throttle positions (see Figure 3.4), negative torque requirement is supplied by the engine to a

certain level. When the requirement exceeds the negative torque supplied by the engine, brakes applied.

This logic is applied in the model by using a look-up table including minimum engine torque values for all engine speeds. Required torque value is compared with the minimum engine torque values and if the difference is negative, brakes are applied too.

Simulink model of the cruise controller can be seen in Figure 4.6, and inside of the torque calculation subsystem is shown in Figure 4.7.

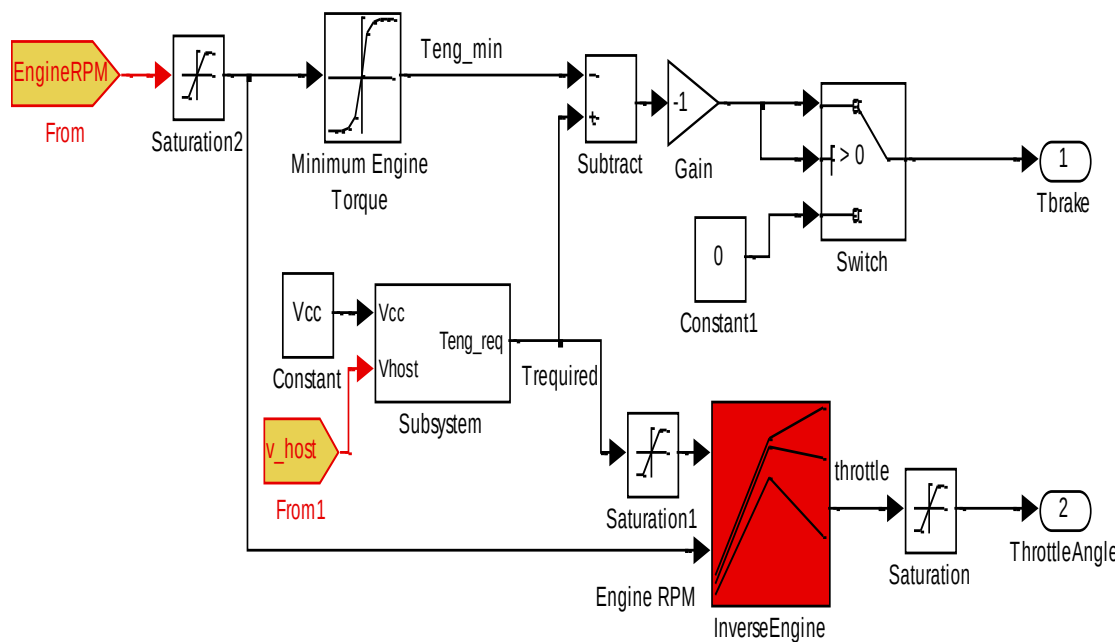


Figure 4.6 : Simulink model of the cruise controller.

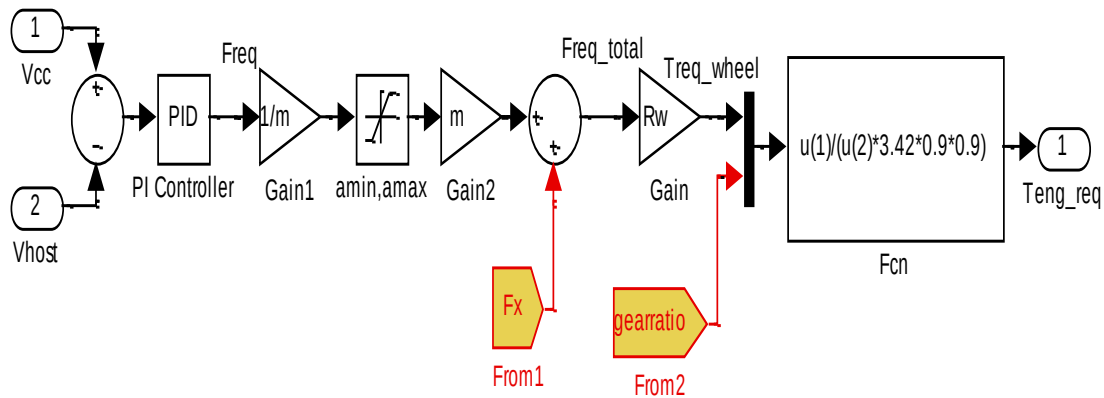


Figure 4.7 : Simulink model for the required torque calculation.

4.3.1.2 PD control

PD control is one of the two control alternatives in the low-level controller of the stop and go control system, which calculates desired acceleration values for target vehicle following purposes. A PD type controller is used to perform these actions. As a starting point of investigation of the control structure, headway time $T_{headway}$ should be mentioned. It is said to be follow time of the target vehicle, meaning that the host vehicle reaches the target vehicle's current position after the headway time. Corresponding follow distance is calculated by multiplying the headway time with the host vehicle speed.

The main aim of a PD type controller is to control distance and relative speed together. The controller equation that will calculate the *desired acceleration* can be given with the following expression;

$$a_{desired} = K_v \cdot (V_{target} - V_{host}) + K_d \cdot (d_x - d_{x,desired}) \quad (4.12)$$

where,

$$d_{x,desired} = T_{headway} \cdot V_{host} + 2 \quad (4.13)$$

K_v and K_d in Equation 4.12 are the controller coefficients. $d_{x,desired}$ is *desired distance* with 2 m safe stopping gap, which changes proportional to the host vehicle speed. This results in a continuously variable controller parameter providing smooth acceleration and deceleration profiles.

After calculating the desired acceleration, a saturation operator is added to the controller model to define the acceleration limits. This is important in terms of stop and go control system to be a comfort system by preventing sudden and high acceleration values. Minimum and maximum acceleration values are selected considering that there is no panic situation for passengers during acceleration and deceleration. Thus, $a_{desired}$ is limited in a range $\pm 3 \text{ m/s}^2$, which is also foreseen in the literature approximately. Desired acceleration is converted to required force by simply multiplying it with the vehicle mass:

$$F_{required} = m \cdot a_{desired} \quad (4.14)$$

Controller's task could be said to calculate a suitable acceleration value in accordance with the received data. K_v and K_d coefficients are so important in calculating the desired acceleration. These coefficients should be tuned with high sensitivity considering that system behavior completely changes with their change.

Controller coefficients in this PD type stop and go control system are selected by trial and error method. The criteria in selecting the coefficients are; utilizing the desired acceleration range optimally, minimizing the follow time and spending minimum energy by keeping the controller inputs as low as possible. Simulink model of the controller can be seen in Figure 4.8, and inside of the torque calculation subsystem is shown in Figure 4.9.

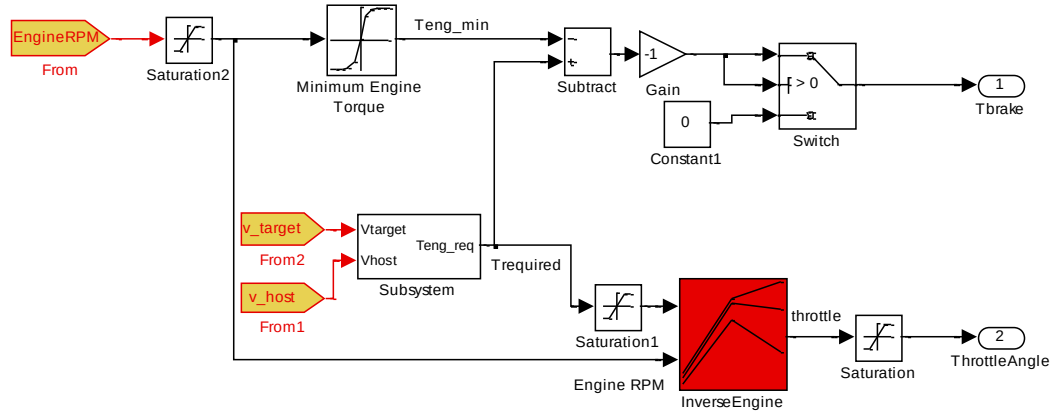


Figure 4.8 : Simulink model of the PD controller.

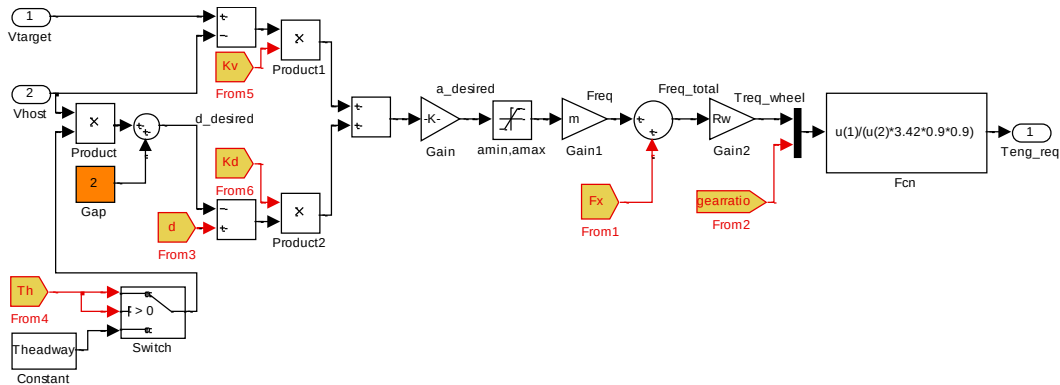


Figure 4.9 : Simulink model for the required torque calculation.

4.3.1.3 Fuzzy logic control

The second alternative in the low-level controller of the stop and go control system for the following purposes is Fuzzy Logic Controller. The main aim of the Fuzzy Logic Control in this stop and go control system is to find a throttle and brake control

algorithm to calculate desired acceleration values that result in necessary speed of the host vehicle, while keeping a desired safe distance from the target vehicle [50].

There are a number of studies on fuzzy logic control in the literature for driving automation [51-54]. In general, it can be said that the use of fuzzy logic in control systems has two main features [32]. The first one is that exact mathematical model of the system is not needed by fuzzy controllers. This is very important for nonlinear systems, such as cars, whose linearization is very difficult. The second feature of fuzzy logic control is that it tries to resemble human drivers' behavior and experience by mimicking their reactions. Users' subjective knowledge can also be integrated into the system, which is in sure a very useful feature to represent human behavior [55].

As discussed above, the purpose of using fuzzy logic in stop and go control is to describe a system, which controls throttle and brake cooperatively. In general, a driver first releases the throttle, then uses the engine brake, and then presses the brake to slow down the vehicle. Fuzzy logic tries to mimic these human actions by using fuzzy rules in target vehicle following situations.

MATLAB/Fuzzy Logic Toolbox is used to model the controller and it is embedded into the low-level control system in Simulink. The general form of the rules in Fuzzy Logic Toolbox is “**if** x [**or/and** z] **then** y [**and** w]”, where x and z are fuzzy conditions for input variables, y and w are fuzzy actions for output variables, and “[]” stands for repetition, which means that a rule can have several conditions and several actions. Inference system used is Mamdani-type [56], and the defuzzification operator is centroid. Output of any rule in this configuration is calculated by considering the degree of truth of input variables. After the degree of truth of all inputs are evaluated, output is multiplied by the lesser or the larger one according to the logical operator is “**and**” or “**or**”, respectively.

Values of input and output variables are gathered from fuzzy sets, which are represented by membership functions. These membership functions can be formed by different shapes, such as triangular, trapezoidal, etc. There are 11 built-in membership function types in Matlab Fuzzy Logic Toolbox. The membership functions are selected by the user by experience. In this thesis, while all the input membership function types are trapezoidal, shapes of the output membership

functions are singletons. Singletons are very commonly used in practical control systems applications. After the aggregation process, there is a fuzzy set for each output variable that needs defuzzification. It is much more efficient in many cases to use singletons as the output membership function rather than a distributed fuzzy set as it can be thought of as a pre-defuzzified fuzzy set. It enhances the efficiency of the defuzzification process because it greatly simplifies the computation required by the more general Mamdani method, which finds the centroid of a two-dimensional function. Rather than integrating across the two-dimensional function to find the centroid, it uses the weighted average of a few data points. Therefore, value of an output variable is calculated as:

$$y_{out} = \frac{\sum_i w_i y_i}{\sum_i w_i} \quad (4.15)$$

where w_i represents the weight of an i th rule and y_i is the value of the output y inferred by the i th. The weight of a rule means its contribution to the overall control action.

Fuzzy logic control in this thesis is composed of four input variables, one output variable, and nine rules. The first input is “*speed error*”, which is the difference between the host vehicle and target vehicle speeds, and expressed as:

$$SpeedError = HostVehicleSpeed - TargetVehicleSpeed \quad (4.16)$$

As the name implies, it is the error and the rules are designed to reduce its value to zero. Membership functions of the speed error are illustrated as:

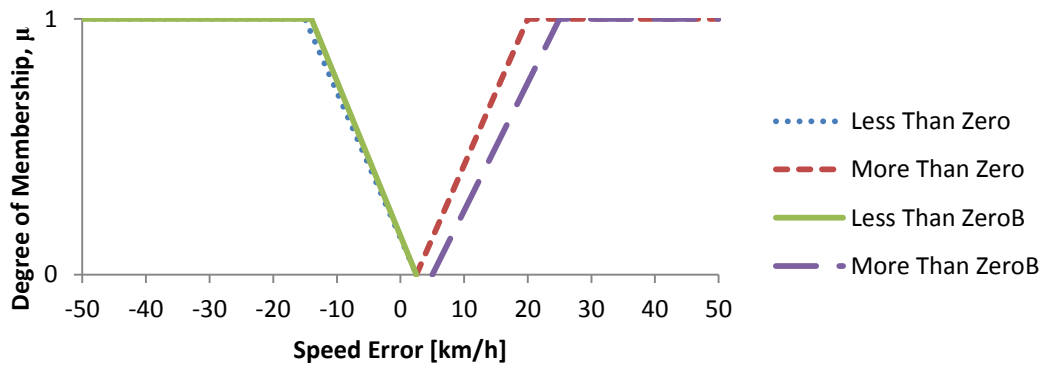


Figure 4.10 : Membership functions for the input speed error [km/h].

The addition of the letter “B” at the end of the membership function names means that it is for brake actuation. The others are for throttle actuation. The reason for the little difference on the figure between the throttle and brake actuations is that there is a slight time difference in actuation in real life. For example, driver first releases the throttle to use engine brake, and when the engine brake is not sufficient to slow down the vehicle, brakes are applied.

The second input is “*acceleration*”. This is the derivative of the speed at time t . Membership function shapes of the input acceleration are shown in Figure 4.11 and it is calculated as:

$$Acceleration_t = \frac{HostVehicleSpeed_t - HostVehicleSpeed_{t-1}}{\Delta t} \quad (4.17)$$

It should be noted here that the result of the acceleration may give oscillating values, and if this is the case, filtering should be used to obtain smoother values.

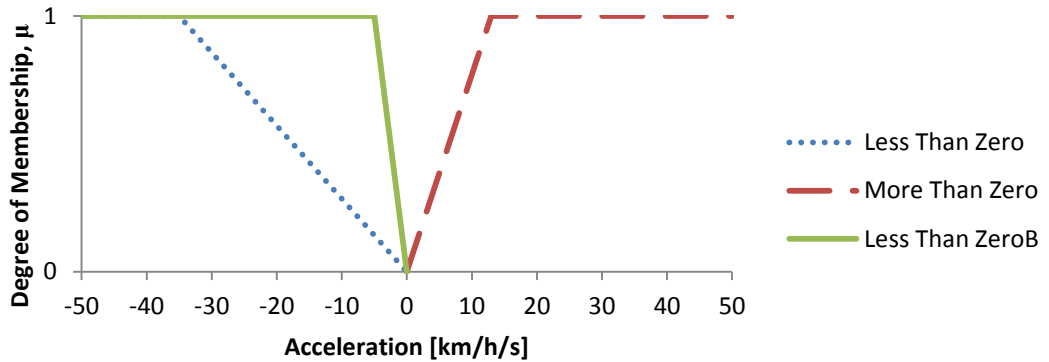


Figure 4.11 : Membership functions for the input acceleration [km/h/s].

As can be seen in the figure, there is no membership function named “More Than ZeroB”. The reason for this is that none of the rules requires this for brake actuation, which will be seen later in this section.

With the inclusion of the speed error and the acceleration terms as inputs to the fuzzy logic control, it resembles to a classic PD control. While the rules involving the speed error behave like a proportional controller component by adjusting the throttle, the rules involving the acceleration behave like a derivative component by smoothing out the actuation of the same command, similar to the damping effect of a D term in PD control.

Input three is “time headway error”. This is the input corresponding to the input “distance” in PD control, but expressed in a different manner. It is given with the following expression;

$$TimeHeadwayError = TimeHeadway_{current} - TimeHeadway_{desired} \quad (4.18)$$

where $TimeHeadway_{current}$ is the current time headway, and $TimeHeadway_{desired}$ is the desired time headway. Membership function shapes of the fuzzy variable time headway error are shown in Figure 4.12.

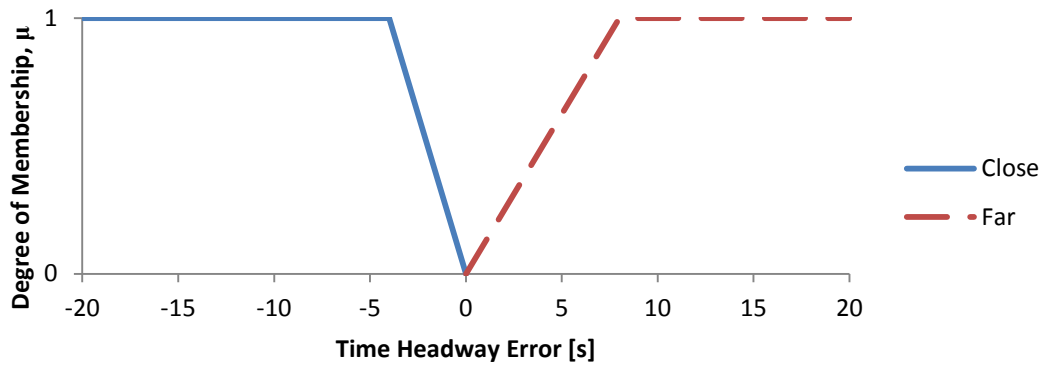


Figure 4.12 : Membership functions for the input time headway error [s].

The definition of the expression “time headway” is already given in the preceding section. Time headway current, adapted for this study, is calculated with the following formula;

$$TimeHeadway_{current} = \frac{x_{target} - x_{host} - 2}{v_{host}} \quad (4.19)$$

where x_{target} and x_{host} are the coordinates of the target and the host vehicles, respectively, and v_{host} is the host vehicle speed. It should be noted that the distance between the vehicles is reduced by 2 m for safety reasons. Time headway desired is the target headway time, which is set as 1 s in normal drive, and reduced to 0.5 s in stopping situations that will be seen later in results chapter.

The last input is “derivative of time headway”. As the name implies, it is the variation of the current time headway with time, and it is calculated with the following equation:

$$DerivativeTimeHeadway = \frac{TimeHeadway_i - TimeHeadway_{i-4}}{4\Delta t} \quad (4.19)$$

The reason for selecting the last four iterations in the calculation is that values of this variable oscillate between positive and negative values. So, this equation gives an average value and this simple filtering is sufficient to smooth the variable. Membership function shapes of this variable can be seen in Figure 4.13.

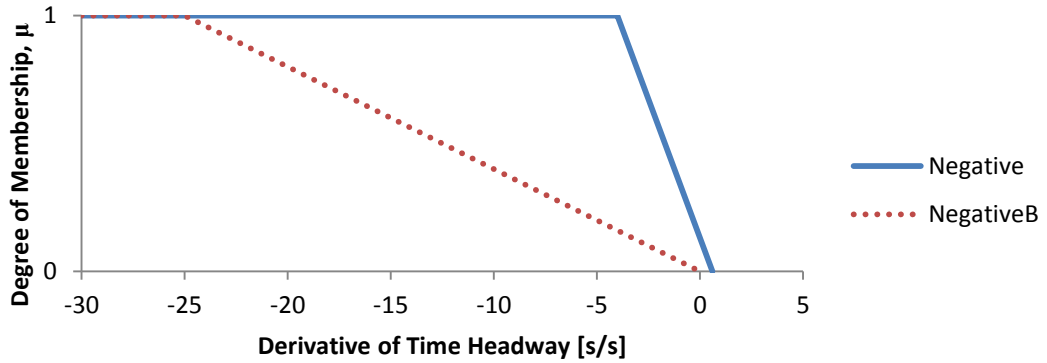


Figure 4.13 : Membership functions for the input derivative of time headway [s/s].

Output of the fuzzy logic control is “*desired acceleration*” as described earlier in this section. Minimum and maximum values of the desired acceleration are limited between $\pm 3 \text{ m/s}^2$ for comfort reasons, which is already explained in the previous section. Membership function shapes of the output variable desired acceleration are shown in Figure 4.14.

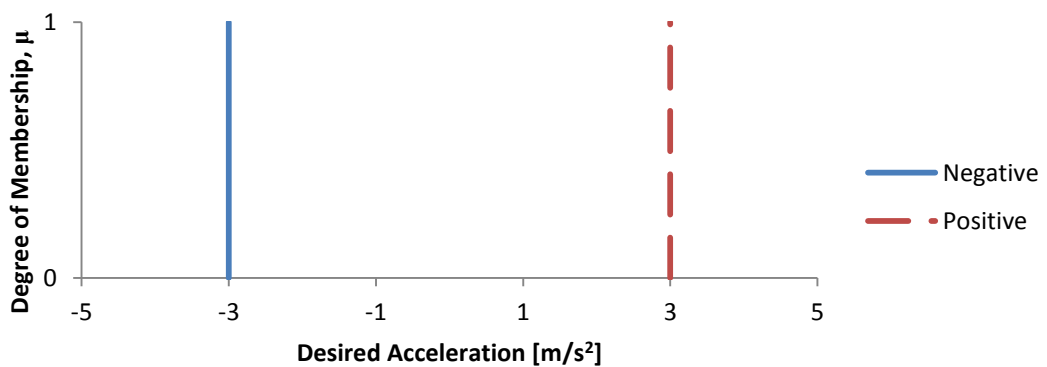


Figure 4.14 : Membership functions for the output acceleration [m/s²].

According to the degree of truth of the output variable, the desired acceleration values change in $\pm 3 \text{ m/s}^2$ range.

Finally, rules constituting the most important part of the fuzzy logic control are described. There are nine rules in the center of the fuzzy logic system that manages the inputs and gives an output as the control action. These rules are:

Rule 1: **If** (*SpeedError* is *MoreThanZero*)

then (*DesiredAcceleration* is *Negative*) (0.1)

Rule 2: **If** (*SpeedError* is *LessThanZero*) **and** (*TimeHeadwayError* is *Far*)

then (*DesiredAcceleration* is *Positive*) (1)

Rule 3: **If** (*Acceleration* is *MoreThanZero*)

then (*DesiredAcceleration* is *Negative*) (0.1)

Rule 4: **If** (*Acceleration* is *LessThanZero*) **and** (*TimeHeadwayError* is *Far*)

then (*DesiredAcceleration* is *Positive*) (0.1)

Rule 5: **If** (*TimeHeadwayError* is *Close*) **and** (*DerivativeTimeHeadway* is *Negative*)

then (*DesiredAcceleration* is *Negative*) (0.1)

Rule 6: **If** (*TimeHeadwayError* is *Close*) **and** (*DerivativeTimeHeadway* is *NegativeB*)

then (*DesiredAcceleration* is *Negative*) (1)

Rule 7: **If** (*SpeedError* is *MoreThanZeroB*)

then (*DesiredAcceleration* is *Negative*) (1)

Rule 8: **If** (*SpeedError* is *LessThanZeroB*) **and** (*TimeHeadwayError* is *Far*)

then (*DesiredAcceleration* is *Positive*) (0)

Rule 9: **If** (*Acceleration* is *LessThanZeroB*) **and** (*TimeHeadwayError* is *Far*)

then (*DesiredAcceleration* is *Positive*) (0)

Name of the variables in the rules are expressed here as defined in Fuzzy Logic Toolbox. The numbers in parenthesis at the end of each rule sentences represent weight of the rules.

The first five rules are considered as the rules for throttle actuation, trying to emulate human action in pressing and releasing the throttle. The weight of a rule is 1 and the output is positive when pressing the throttle is required, and corresponding values are 0.1 and negative when releasing is required. This means that when throttle is required and degree of truth is one, then the acceleration is maximum. When there is no need for throttle and degree of truth is one, then the acceleration is negative and approximately equal to the maximum engine brake. Nevertheless, weight of the rule 4 is 0.1 although the output is positive. The reason for this is that the variable

acceleration is included in fuzzy logic control to smooth the output, which is already explained previously in this section.

The last four rules are designed to represent human behavior in brake actuation. Maximum deceleration is applied when there is a need for brake and degree of truth is one. When there is no need for brake, weight of the rules are zero to assign desired acceleration value zero, which mimicks human brake pedal release action. It should be noted that the brake controller does not fully emulate the throttle controller. As can be seen in Figure 4.11, there is no membership function named “MoreThanZeroB” and corresponding rule is not included, which is: “**If** (*Acceleration* is MoreThanZeroB) **then** (*DesiredAcceleration* is Negative)”. The reason is that this is a derivative rule which is aimed to smooth the behavior of the acceleration situation. However, engine braking in throttle rule 3 is enough to achieve this goal, and the brake pedal does not need to be applied.

About the rules, some points can be made to explain meanings of them in detail. As already explained in this section, rules 1 and 2 involving the *speed error* adjust the throttle, and rules 3 and 4 involving the *acceleration* smooth out the throttle actuation by assigning convenient *desired acceleration* values as output. Moreover, the variable *time headway error* is added to the rules 2 and 4 to counteract the effect of the *speed error* and *acceleration* parts when the target vehicle is relatively close. Rule 5 is designed to release the throttle when the target vehicle is close and the time headway is reducing fast. Rule 6 is the same as rule 5 except the second part, in which membership functions Negative and NegativeB differs in shape allowing rule 5 acts first as engine brake, and if this is insufficient rule 6 acts after rule 5 for braking. In a similar manner, rule 7 is activated when the engine brake in rule 1 is insufficient. Rule 8 is the complementary rule and acts with rule 2 cooperatively. This means that the brakes are released with rule 8 before the throttle in rule 2 is applied. The last one is rule 9, which is the derivative part of the brake controller smoothing the control actions, and works cooperatively with rule 4.

To sum up, fuzzy logic control for stop and go operation is made up of five rules for representing throttle actuation and four rules for representing brake actuation. Simulink model of the Fuzzy Logic controller is the same as PD controller shown in Figure 4.8, but inside of the torque calculation subsystem is different up to the end of *desired acceleration* calculation, and illustrated in Figure 4.15.

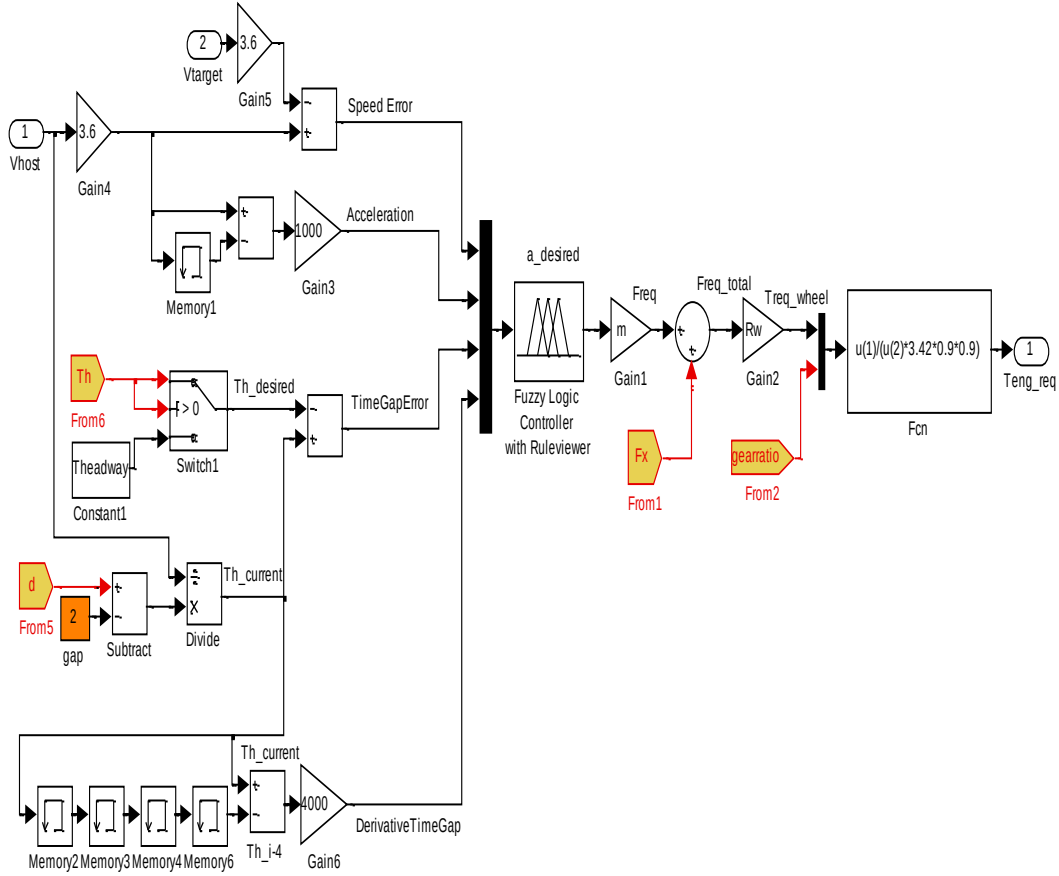


Figure 4.15 : Simulink model for the required torque calculation.

4.3.2 High-level control

In the scope of high-level control; scenarios are differentiated, suitable control system is activated according to the selected scenario, calculated acceleration and force variables are formed, and brake and/or engine usage is decided according to these variables. As can be understood, there is no mathematical structure in high-level control as in low-level controllers. High-level control consists of flow diagrams differentiated to certain states. Simulink/Stateflow is used to constitute this structure in Simulink controller model.

First, scenario differentiation and designing of this structure in the model is described. The stateflow block shown in Figure 4.16 is used to differentiate the scenarios. There are four substates within the scenario differentiation state. These are longitudinal, lateral, stop, and no-target states. For the no-target state, the distance should be either less than zero or greater than the radar range. Distance is less than zero means passing the target vehicle. No-target state is not considered as a scenario in stop and go system, and expressed as scenario 0 for simulation purpose.

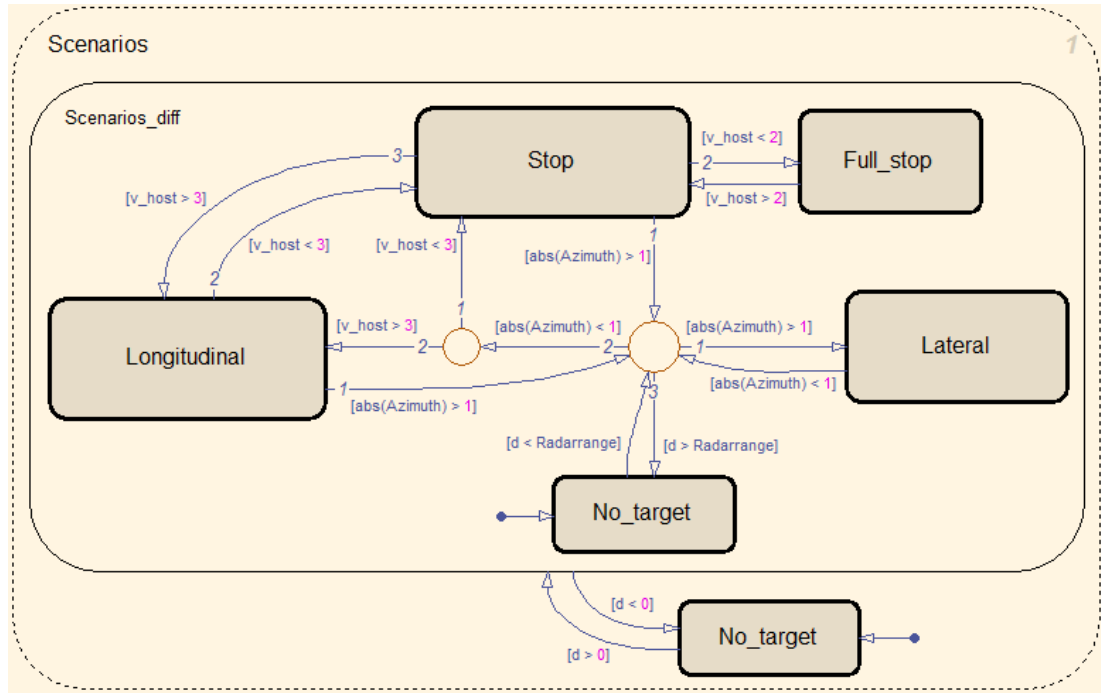


Figure 4.16 : Stateflow block for scenario differentiation.

If a target vehicle is detected within the radar range, no-target state is leaved and stateflow block checks if the scenario is longitudinal or lateral. This is done by the azimuth angle measured by the radar.

If the azimuth angle relates longitudinal scenarios, longitudinal state is activated. One more condition for passing through the longitudinal scenarios is that the vehicle speed should be greater than 3 m/s. When the inside of the longitudinal state is investigated, it can be seen that the main effect in selecting longitudinal scenarios is the target vehicle speed. Thus, target vehicle speed is observed with one second time intervals to be sure that the scenario is not changed with sudden changes. If the level of change is above a certain limit within one second, then slowdown or speed-up state is decided. Scenario 1 is activated if the target vehicle speed is constant, Scenario 2 is activated if the change in target vehicle speed is positive, and Scenario 3 is activated if the change in target vehicle speed is negative. Scenario 4 represents a cut-in vehicle presence. This scenario is applied by getting the target vehicle's position back at a certain time, and it is differentiated by checking the change in the distance.

Lateral state is activated if there is a change in azimuth angle. Lateral state represents lane changes of the host and the target vehicles. Host vehicle lane change is differentiated by checking yaw angle of the host vehicle together with the azimuth

angle. This is represented with Scenario 5. If azimuth changes and there is no change in host vehicle's yaw angle, then it is understood that the target vehicle changes its lane and Scenario 6 is activated.

The last state represents vehicle's stop scenarios. When the host vehicle speed falls below 3 m/s, Scenario 7 is differentiated. The vehicle is assumed to come to a full stop if the speed falls below 2 m/s. As will be seen later in results chapter, the vehicle speed can fall far below this speed. This limit is set just to differentiate the full stop scenario and assign some values for simulation purposes. Headway time T_h is also decreased to 0.5 second in full stop mode to follow the target vehicle with a closer distance. If the vehicle goes after the full stop, Scenario 8 is activated representing repeated stops and goes.

After differentiating the scenarios, another stateflow system is used to run the related low-level controllers. This stateflow diagram is shown in Figure 4.17.

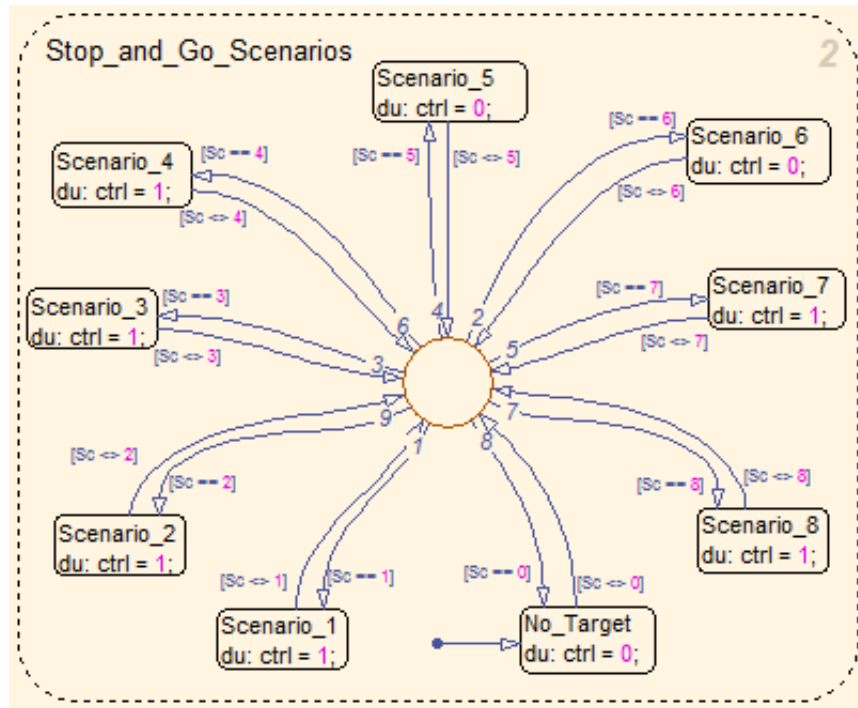


Figure 4.17 : Stateflow block for scenario substructures.

As can be seen from the figure; scenarios no-target, 5 and 6 use the same controller as cruise controller, whereas the others use stop and go controller (PD or Fuzzy Logic controller).

While cruise control is used as a stand-alone controller in no-target situation, it can be activated after stop and go control in scenarios 5 and 6. Although there are only

two types of controllers in the control structure at a time, all scenarios are shown to provide visual attention. As previously described, these controllers are the low-level controllers, which propose desired acceleration values with longitudinal control equations. Decision on throttle and brake transitions are also given in these low-level controllers although they are accounted of part of high-level control. One more point should be raised here that if the target vehicle speed exceeds the cruise control set speed in scenario 2, then it will no longer be a target vehicle and no-target situation applies. The reason for this is that the stop-and-go control system does not have the right to exceed the speed set by the driver.

After describing the stop and go control structure, its performance will be tested in the next chapter for PD control and Fuzzy Logic control, and comparisons will be made.

5. SIMULATION RESULTS

All scenarios, which the stop and go vehicle might be subjected to in urban traffic, are described in the previous chapter in detail. These scenarios are simulated in MATLAB/Simulink environment and results of the simulations are presented in this chapter in the order of scenarios. Results will be given for the PD control first, and then comparisons will be made with the Fuzzy Logic control.

5.1 Results for PD Control

All scenarios are simulated for stop-and-go PD control and presented here, separately.

5.1.1 Scenario 1

In scenario 1, a target vehicle with a slower speed is detected while the host vehicle is traveling in cruise control mode (no-target mode). Host vehicle set speed is 10 m/s, whereas the target vehicle set speed is 5 m/s. Distance between the vehicles is 85 m at $t = 0$ s. Radar range is 40 m and headway time $T_h = 1$ s.

In Figure 5.1, as the host vehicle detects the target vehicle at around $t = 8$ seconds, host vehicle slows down to the target vehicle's speed and following starts.

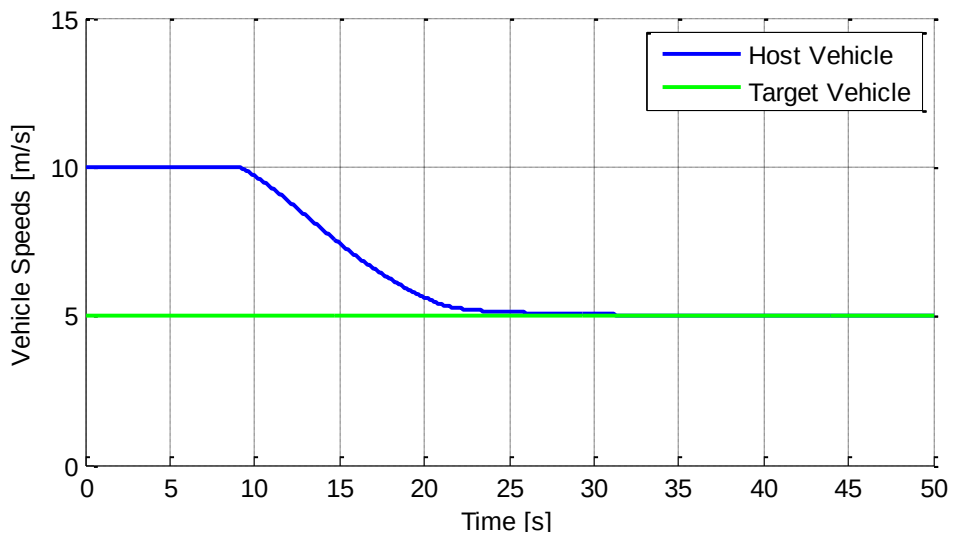


Figure 5.1 : Host and target vehicle speeds.

Figure 5.2 shows the distance between the vehicles and compares it with the desired distance, which is equal to the host vehicle speed since headway time T_h is selected as 1 for the stop and control system (2 m safety distance is added). A small discrepancy between the actual and the desired distance is probably due to deficiencies in tuning of the controller coefficients. However, actual distance is greater than the desired one, so the following is said to be with a safe distance. A deep investigation is required to determine controller coefficients to be able to obtain the best results.

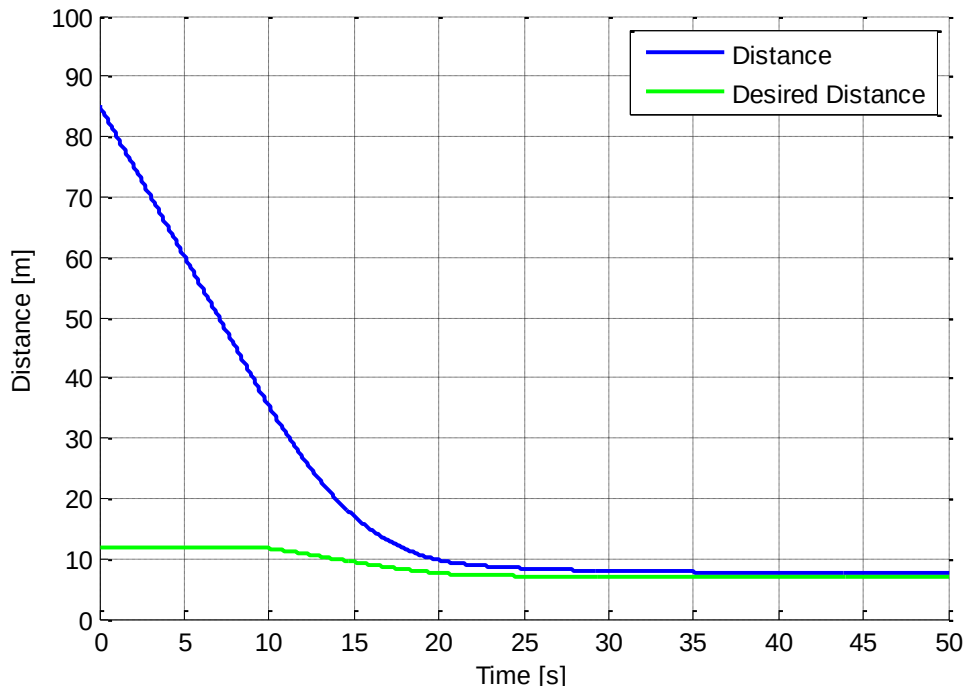


Figure 5.2 : Distance between the vehicles.

Engine speed during the scenario can be seen in Figure 5.3. The smooth change in the engine speed implies that there is no gearshift. Figure 5.4 shows the gearshifts.

Change in the throttle angle is illustrated in Figure 5.5. It suddenly falls to 0 when the target vehicle is detected at $t = 0$ s, and remains like this until the target vehicle speed and desired distance is reached.

Figure 5.6 shows the engine torque, which suddenly falls to a negative value when the throttle angle is zero. It increases in the negative region because of the decrease in the engine speed. Negative engine torque helps brakes whose torque is shown in Figure 5.7.

Differentiation and activation of scenario 1 is shown in Figures 5.8 and 5.9.

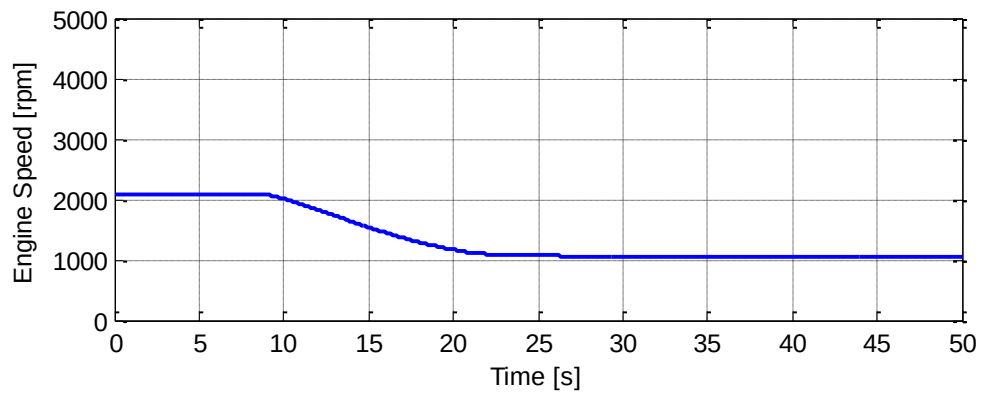


Figure 5.3 : Engine speed.

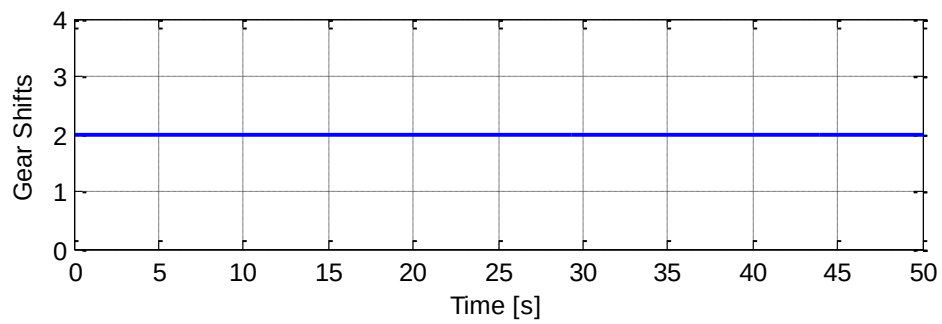


Figure 5.4 : Gearshifts.

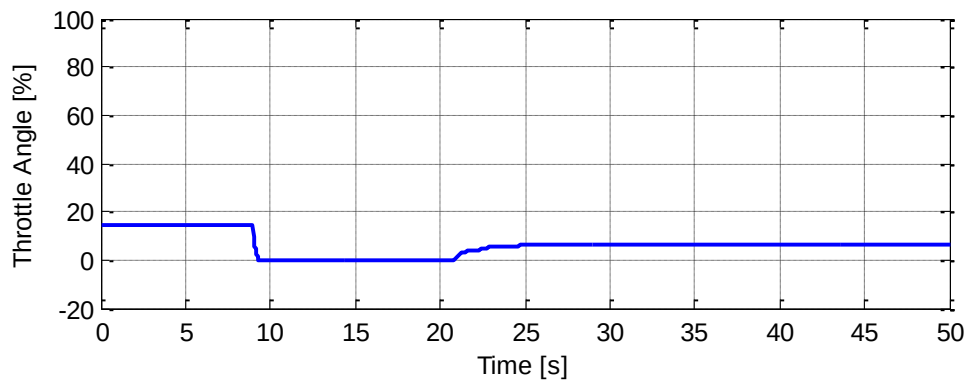


Figure 5.5 : Throttle angle.

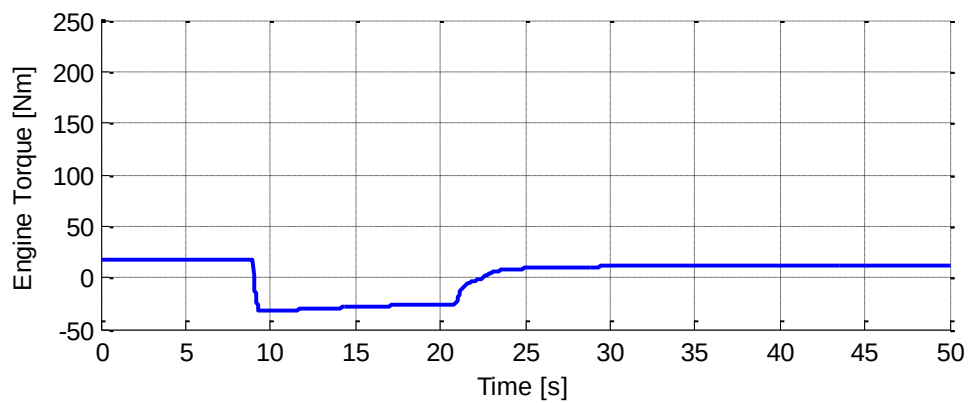


Figure 5.6 : Engine torque.

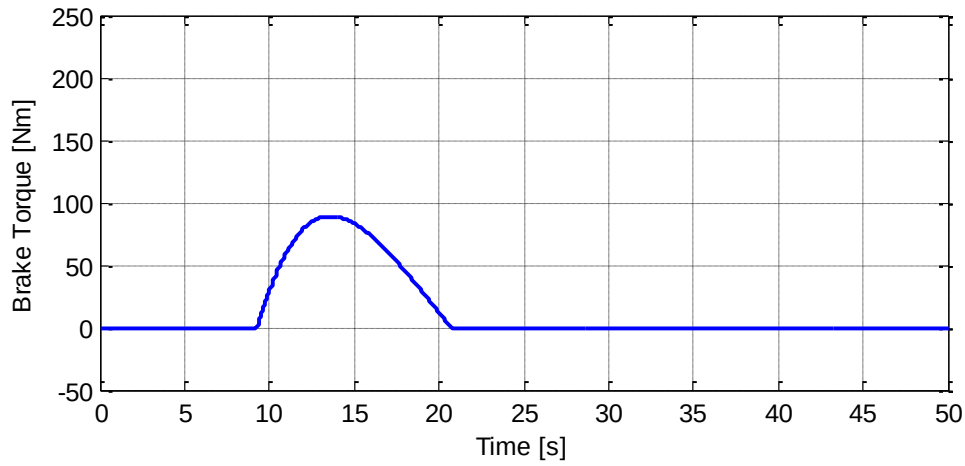


Figure 5.7 : Brake torque.

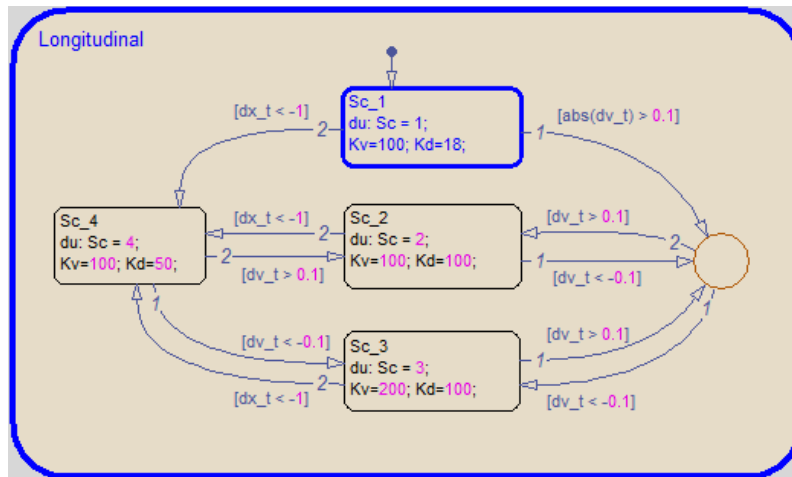


Figure 5.8 : Scenario 1 is differentiated in Stateflow.

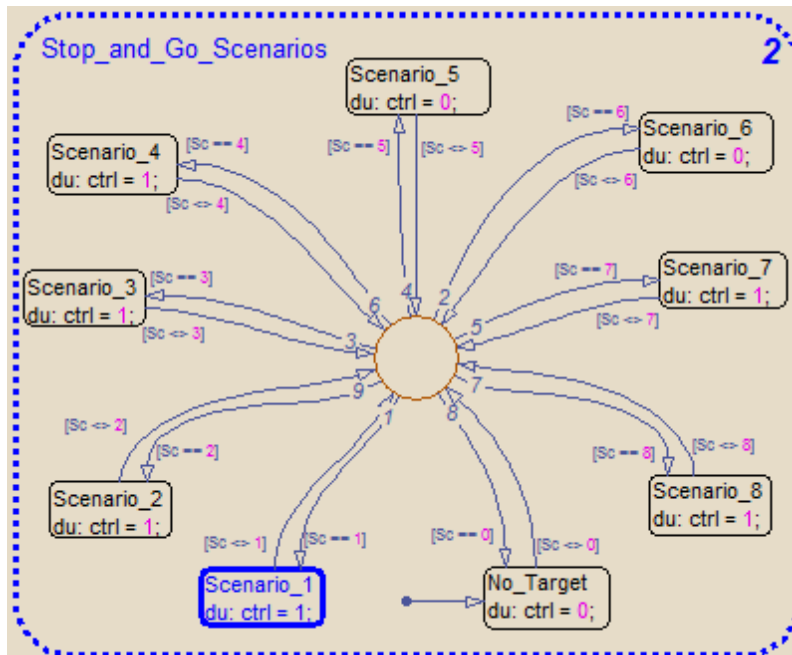


Figure 5.9 : Scenario 1 is activated in Stateflow.

5.1.2 Scenario 2

Scenario 2 is performed after scenario 1. At $t = 47$ s, reference speed of the target vehicle is increased from 5 m/s to 8 m/s. Host vehicle catches the target vehicle with a little time delay as can be seen in Figure 5.10.

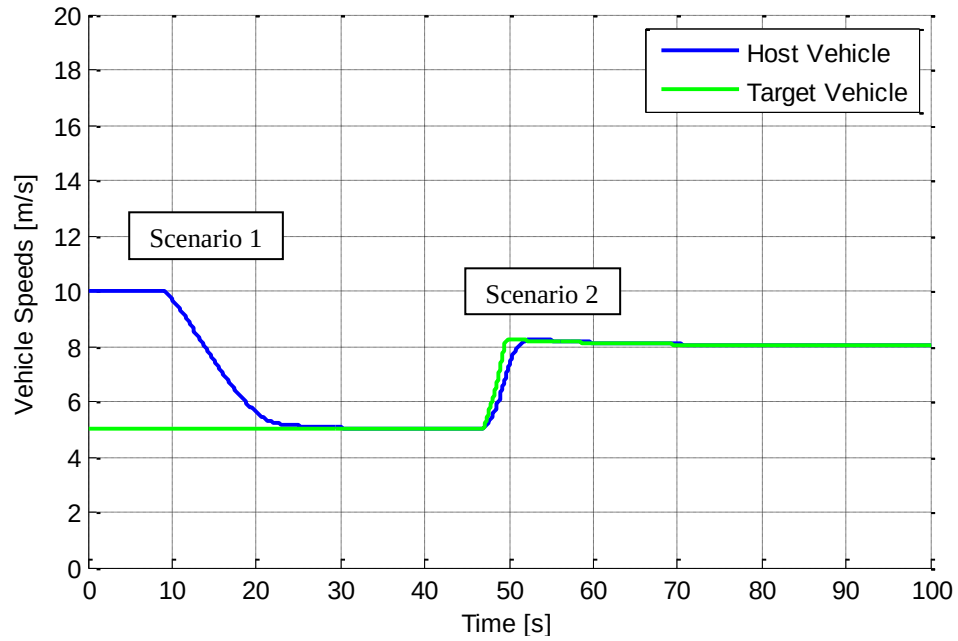


Figure 5.10 : Vehicle speeds.

Figure 5.11 shows the distance between the vehicles. It increases with the increase in host vehicle speed. However, the discrepancy with the desired distance is decreased as the speed and distance errors are relatively lower compared to scenario 1.

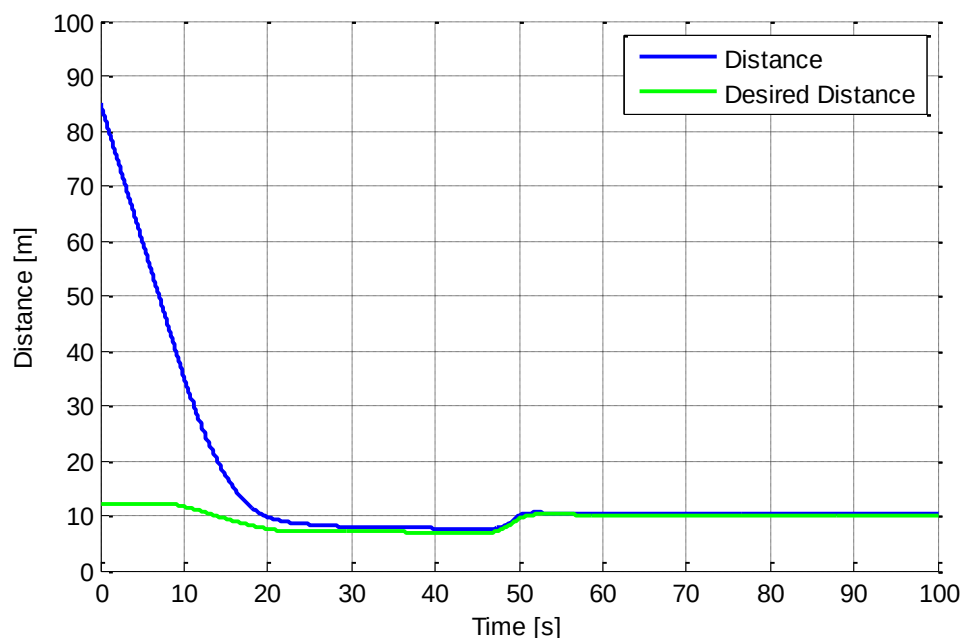


Figure 5.11 : Distance between the vehicles.

Engine speed is shown in Figure 5.12. When the target vehicle speed increases at $t = 47$ s, the host vehicle engine speed increases to be able to catch the target vehicle.

Figure 5.13 shows the decrease in gear state to increase the engine engine torque, which is shown in Figure 5.14. A sharp and short increase in engine torque at around $t = 52$ s is due to the increase in gear state again.

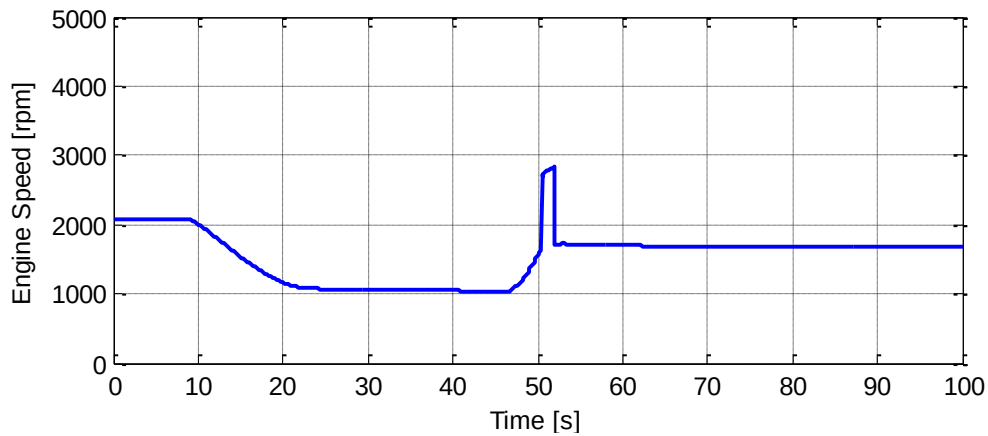


Figure 5.12 : Engine speed.

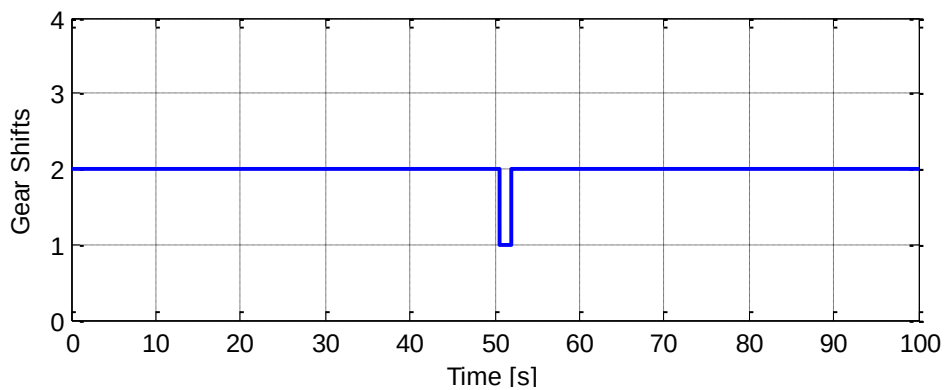


Figure 5.13 : Gearshifts.

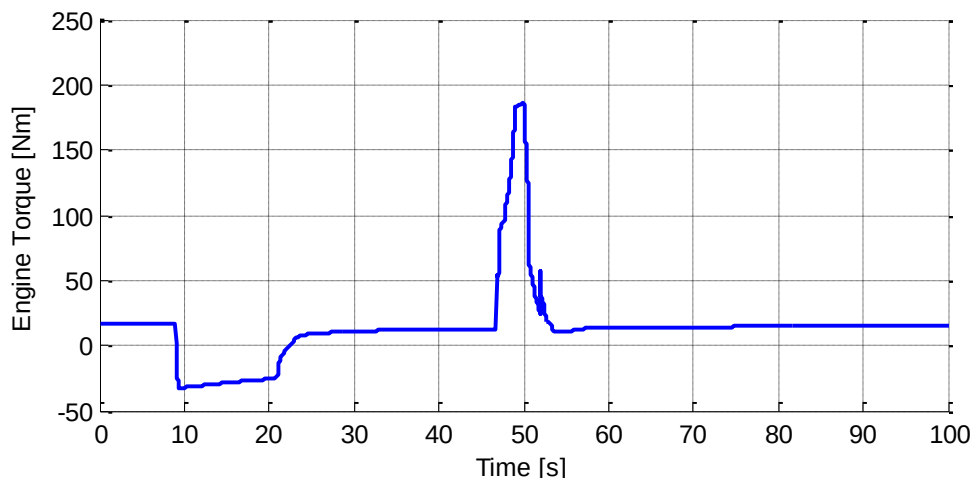


Figure 5.14 : Engine torque.

5.1.3 Scenario 3

Scenario 3 is performed after scenarios 1 and 2. At $t = 97$ s, reference speed of the target vehicle is decreased from 8 m/s to 6 m/s. Host and target vehicle speeds in this scenario can be seen in Figure 5.15.

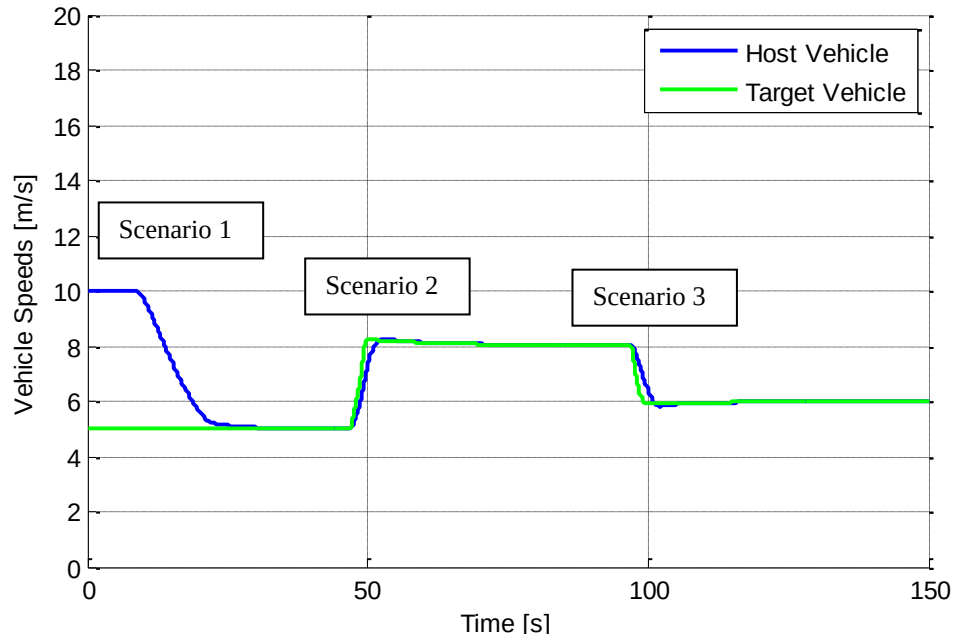


Figure 5.15 : Vehicle speeds.

Figure 5.16 shows the distance between the vehicles. Distance error is very close to zero as in scenario 2, where tuning of the PD coefficients is relatively easy compared to scenario 1.

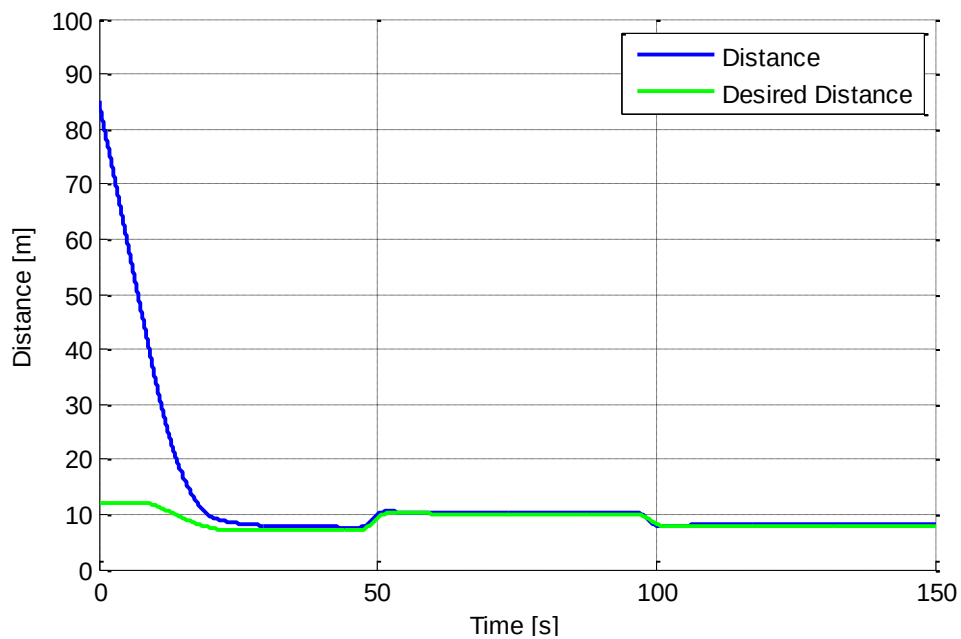


Figure 5.16 : Distance between the vehicles.

Engine and brake torques are shown in Figure 5.17 and 5.18, respectively. There is no sharp change in these values, meaning that there is no change in gear state as can be seen in Figure 5.19.

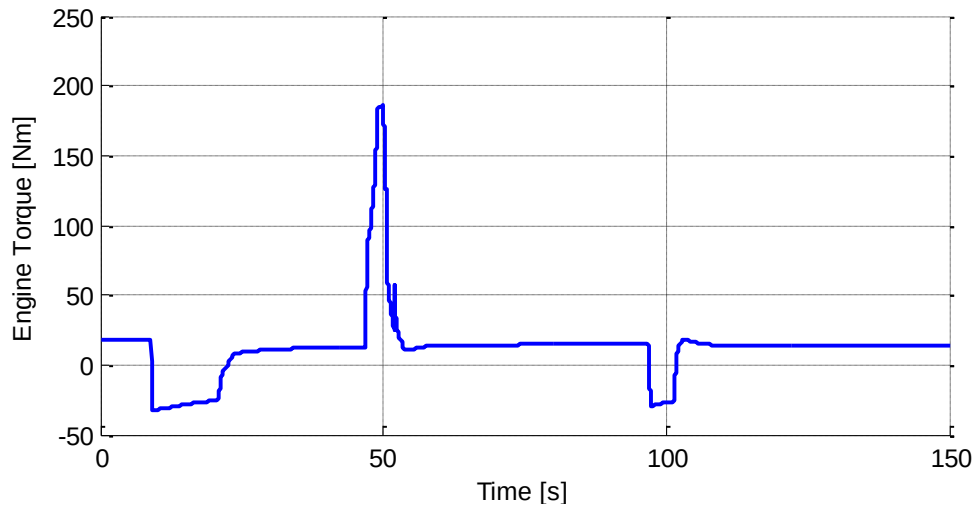


Figure 5.17 : Engine torque.

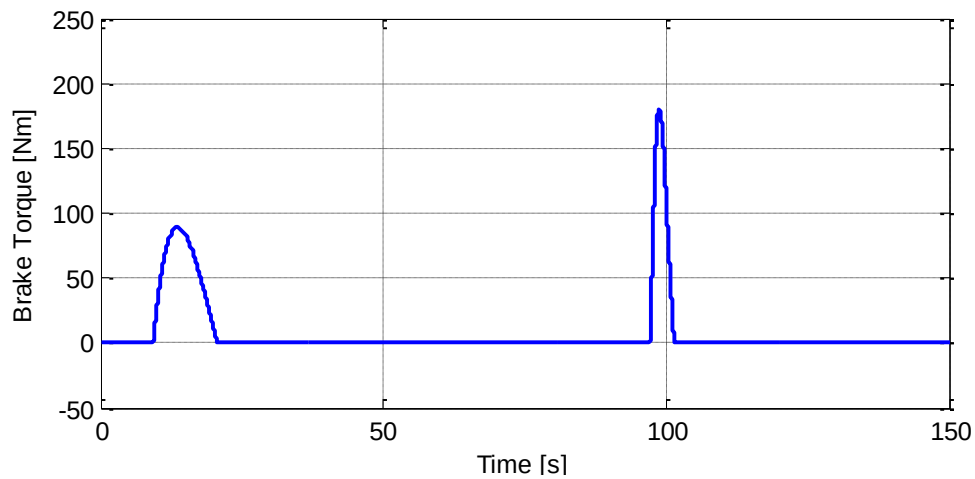


Figure 5.18 : Brake torque.

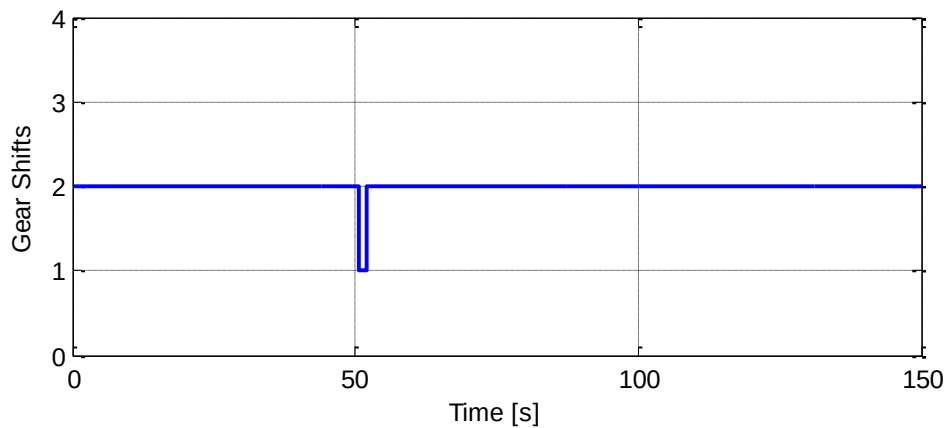


Figure 5.19 : Gearshifts.

5.1.4 Scenario 4

Scenario 4 is a cut-in vehicle situation. This scenario starts after scenario 1. Cut-in occurs at $t = 47$ s. Cut-in vehicle speed is also 5 m/s like the target vehicle.

Vehicle speeds and distance between the vehicles are shown in Figures 5.20 and 5.21, respectively. As can be seen from the figures, host vehicle restores both the speed and the distance values in a reasonable time.

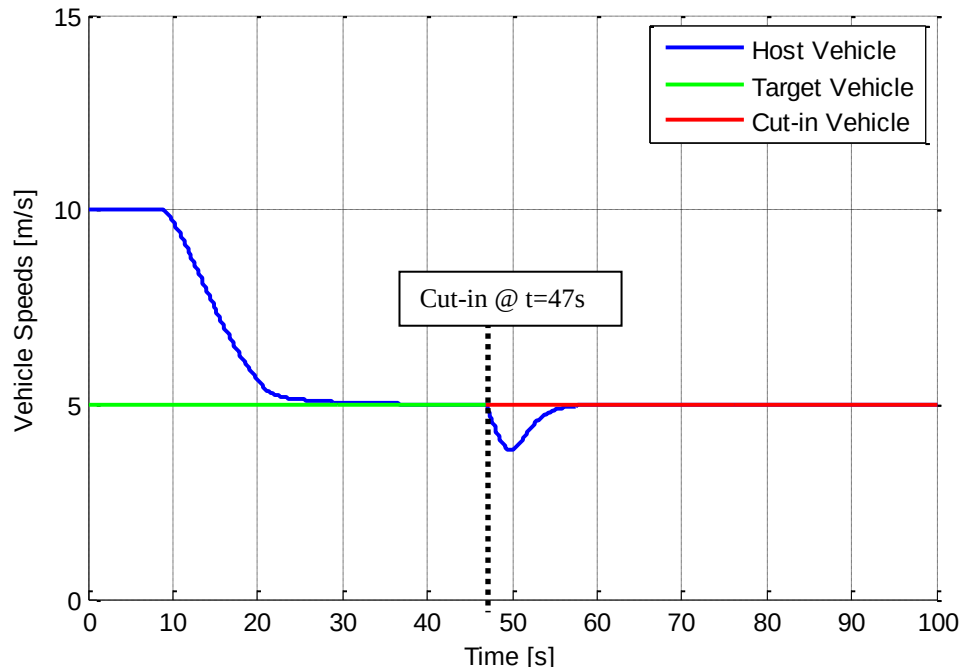


Figure 5.20 : Vehicle speeds.

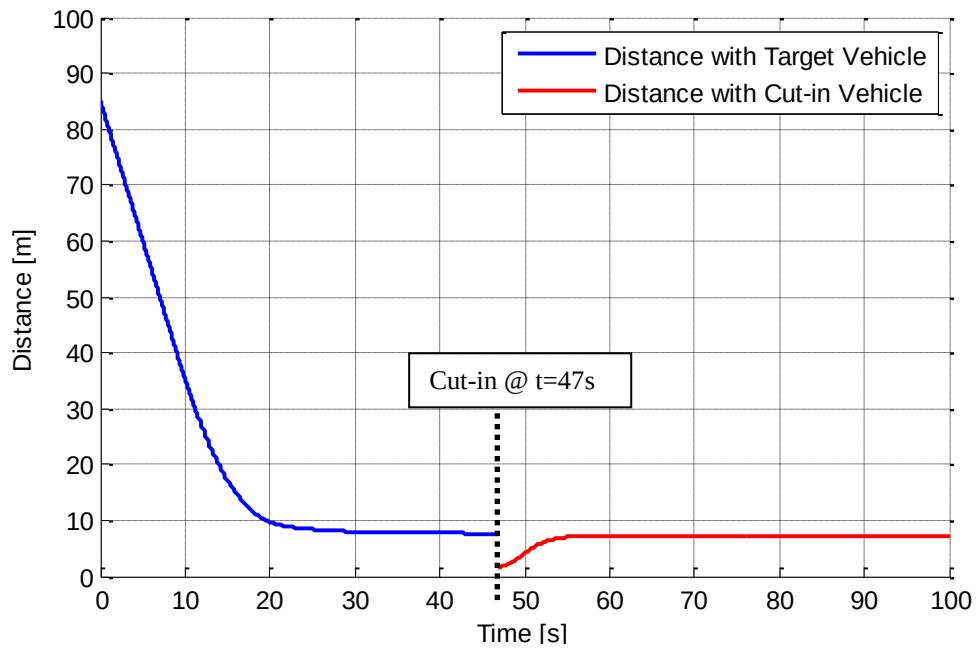


Figure 5.21 : Distance between the vehicles.

Engine and brake torques are shown in Figure 5.22 and 5.23, respectively. It can be inferred from the figures that there is no change in gear state as there is no unexpected sharp changes in these values.

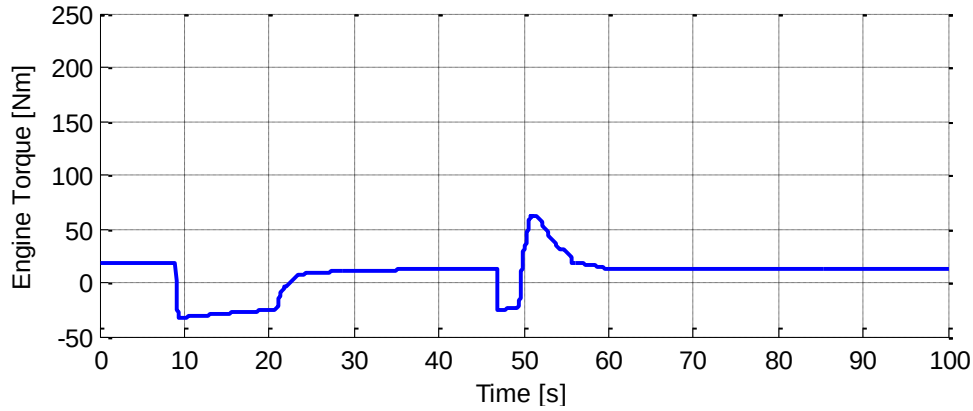


Figure 5.22 : Engine torque.

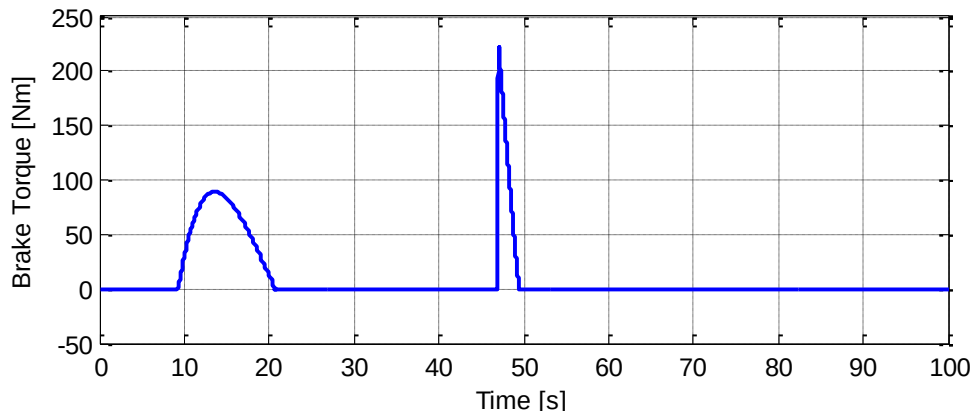


Figure 5.23 : Brake torque.

5.1.5 Scenario 5

Scenario 5 covers the lateral mode of the stop-and-go control. The target vehicle is provided to change its lane by steering inputs at $t = 37$ s as shown in Figure 5.24.

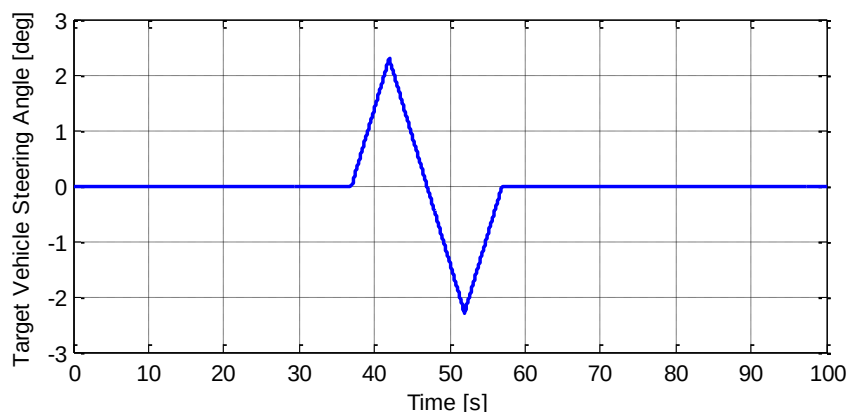


Figure 5.24 : Target vehicle steering input.

Figure 5.25 shows the coordinates of the target vehicle because of this steering.

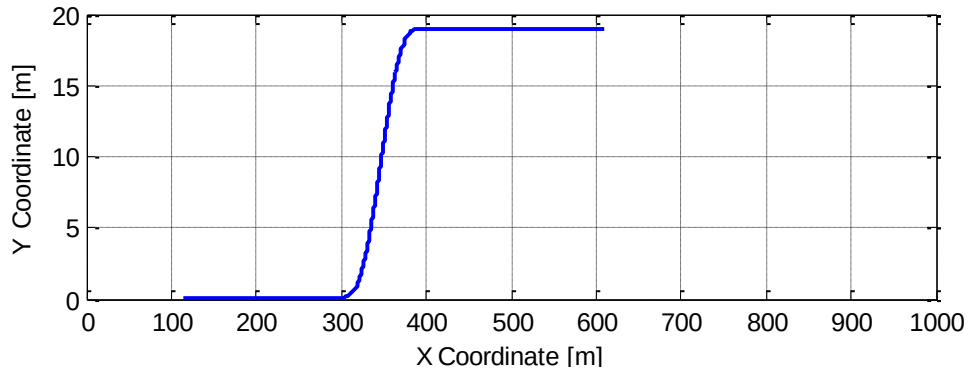


Figure 5.25 : Target vehicle coordinates.

When the target vehicle changes its lane, it is no longer a target vehicle as can be seen in Figure 5.26. Host vehicle set speed is 10 m/s initially, and then decreases to 5 m/s as in scenario 1. When the lane change occurs at $t = 37$ s, the host vehicle increases its speed to the set speed of 10 m/s again in cruise control mode.

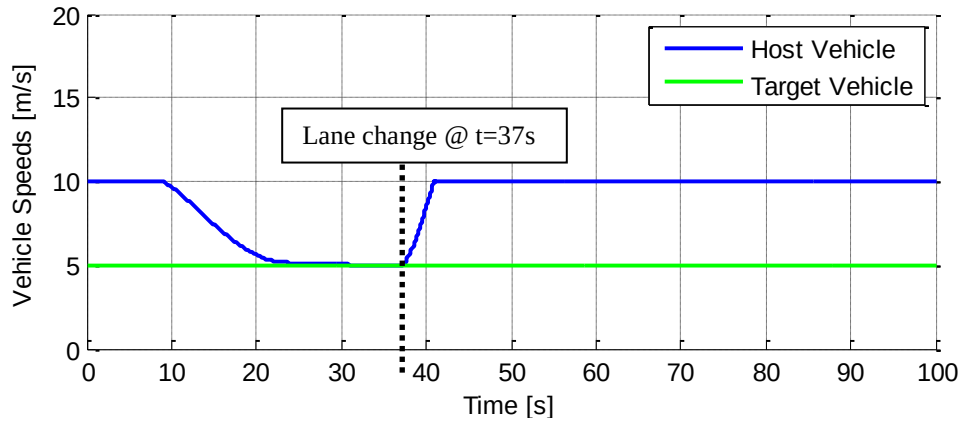


Figure 5.26 : Vehicle speeds.

Differentiation of scenario 5 is shown in Figure 5.27 below.

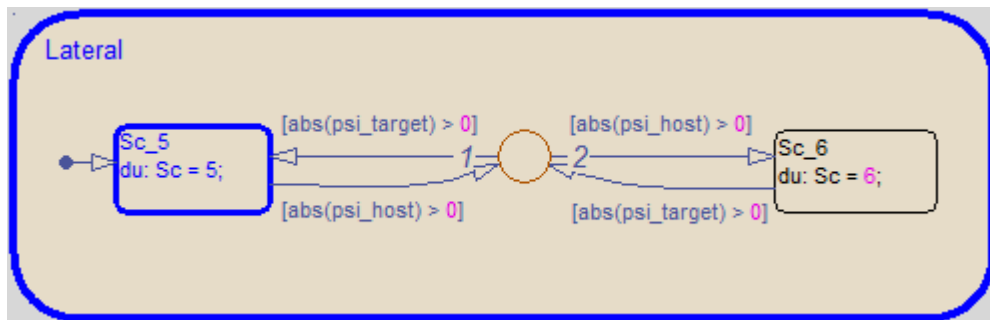


Figure 5.27 : Scenario 5 is differentiated in Stateflow.

Activation of scenario 5 is shown in Figure 5.28.

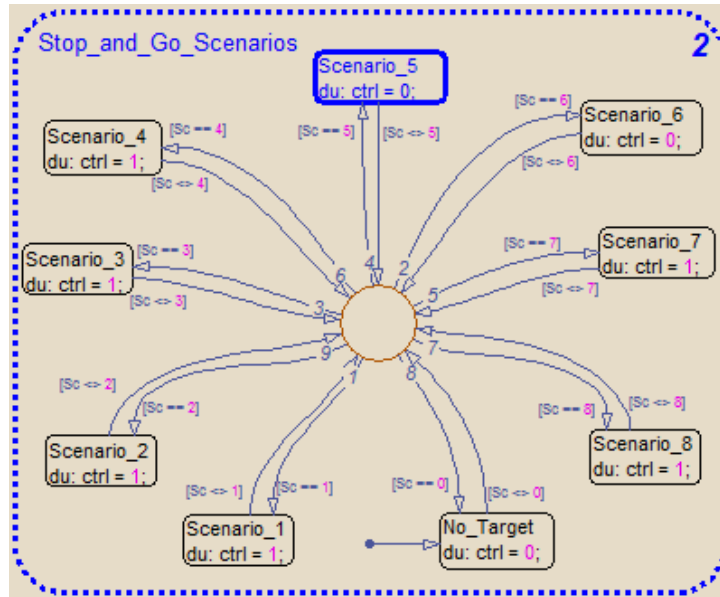


Figure 5.28 : Scenario 5 is activated in Stateflow.

5.1.6 Scenario 6

In scenario 6, host vehicle changes lane this time. The steering input shown in Figure 5.29 is applied to the host vehicle at $t = 37$ s. Coordinates of the host vehicle is illustrated in Figure 5.30 as a results of this steering input.

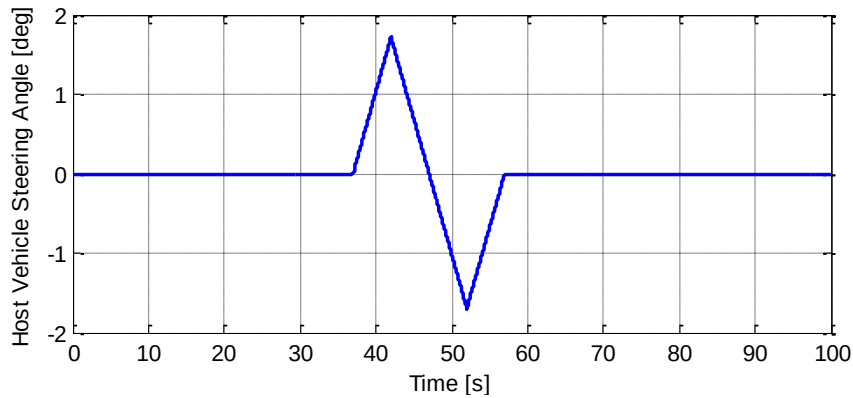


Figure 5.29 : Host vehicle steering input.

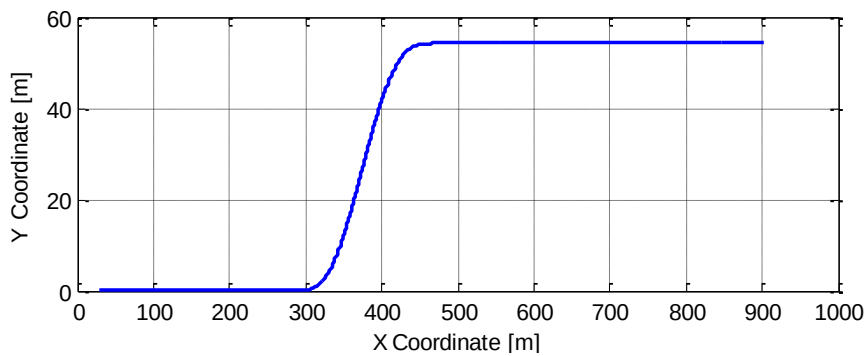


Figure 5.30 : Host vehicle coordinates.

When the host vehicle changes its lane, it leaves following the target vehicle as can be seen in Figure 5.31. When the lane change occurs at $t = 37$ s, the host vehicle increases its speed to the set speed of 10 m/s again in cruise control mode.

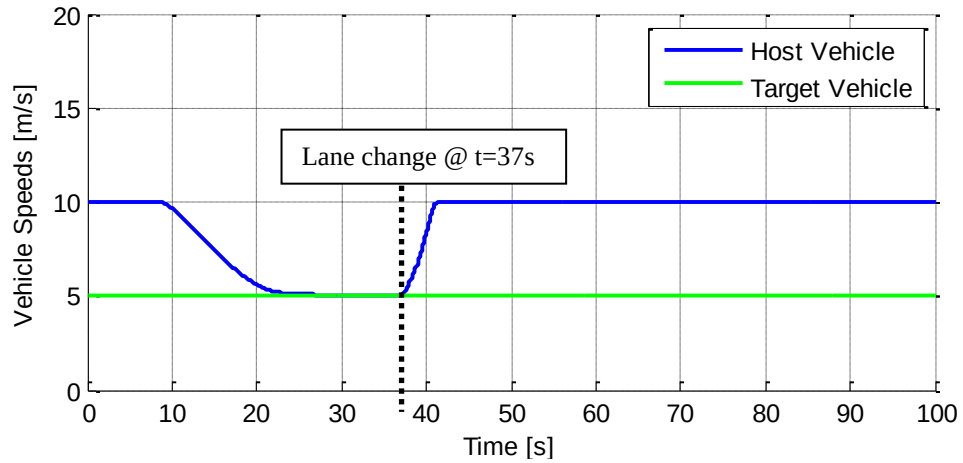


Figure 5.31 : Vehicle speeds.

5.1.7 Scenario 7

This scenario represents stop mode of the control system. The scenario starts after scenario 1 and the vehicle speeds can be seen in Figure 5.32. The target vehicle starts decreasing its speed at $t = 47$ s from 5 m/s to 0.1 m/s, and the host vehicle makes effort to follow the target vehicle by decreasing its speed, too. 0.1 m/s speed is technically accepted as zero in simulations, because 0 m/s vehicle speed makes some equations infinity in the vehicle model and the simulations are aborted.

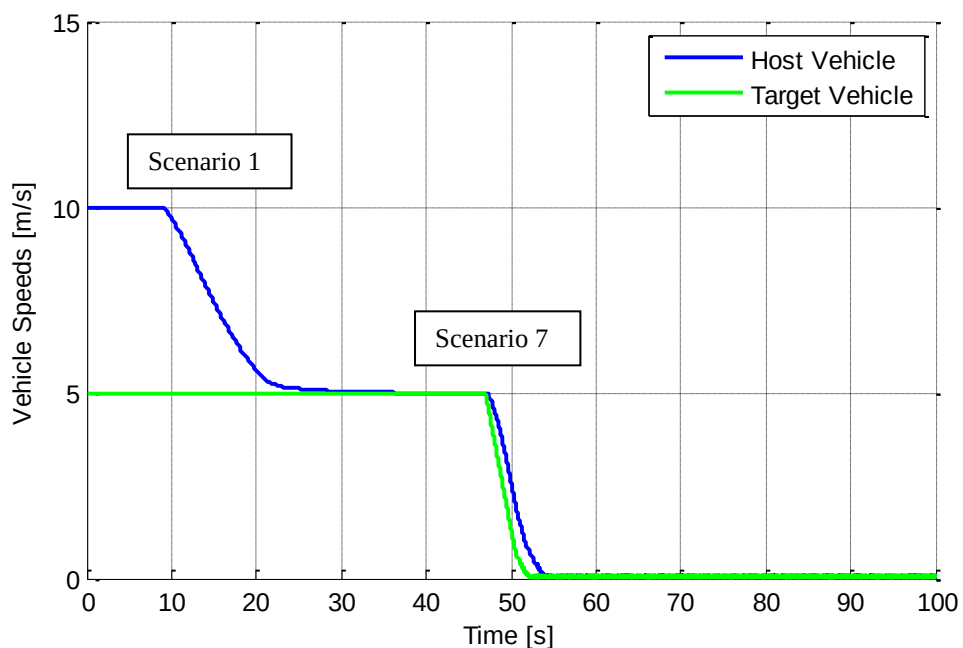


Figure 5.32 : Vehicle speeds.

Very little change in the slope of the host vehicle speed at the velocity of 2 m/s is seen in the figure, which is due to the decrease in headway time T_h from 1 to 0.5 s. As discussed earlier, this is done to follow the target vehicle at a closer distance when the traffic gets heavier.

This situation can also be seen in Figure 5.33. The desired distance suddenly falls from 4 m to 3 m. Final desired distance is around 2.05 m as 2 m gap for safety plus 0.05 m half of the host vehicle speed.

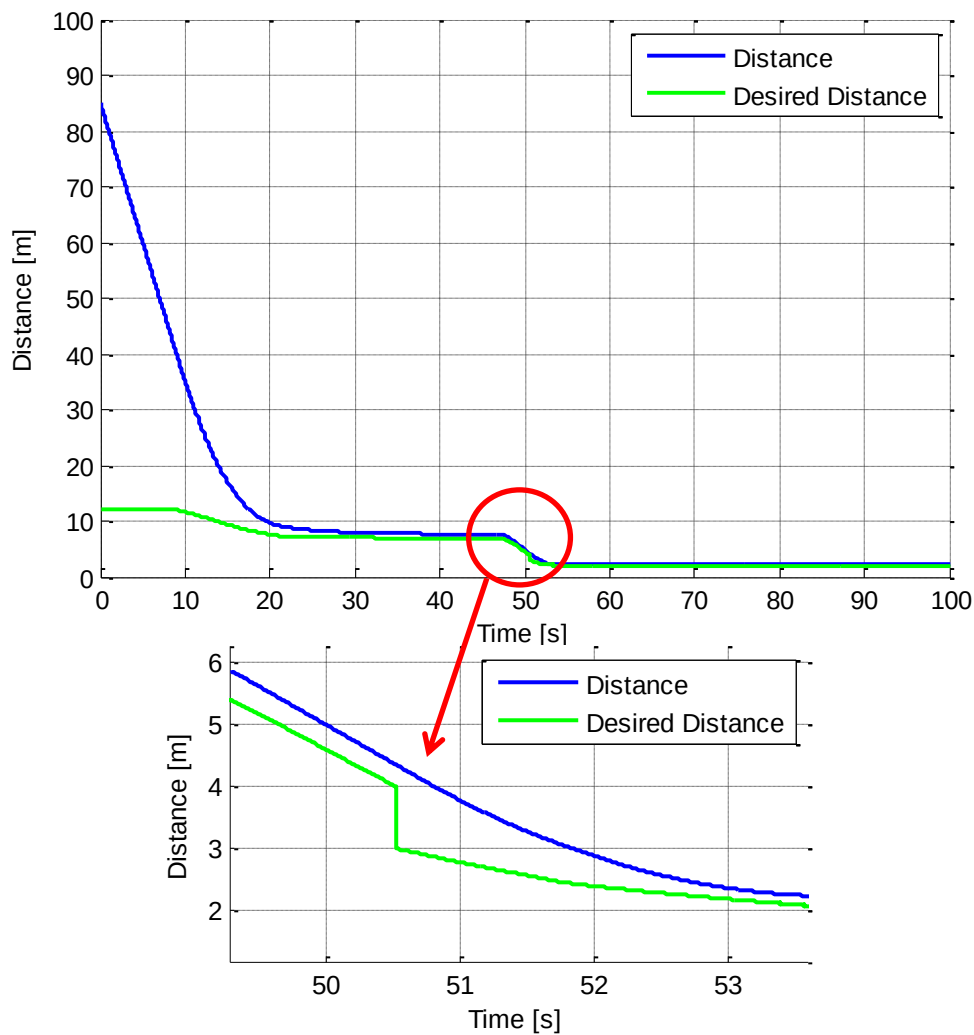


Figure 5.33 : Distance between the vehicles.

5.1.8 Scenario 8

Finally, scenario 8 represents repeated stops and goes. This scenario occurs after scenario 1. The target vehicle goes after stopping, and the host vehicle follows. The target vehicle changes its speed between different values representing common situations in stop-and-go traffic.

Vehicle speeds can be seen in Figure 5.34. Stopping speed is selected as 0.5 m/s because of hard simulation parameters below this speed.

As can be seen from the sub-figure in Figure 5.34, the target vehicle speed oscillates around the stopping speed of 0.5 m/s, so the host vehicle speed oscillates accordingly. However, the host vehicle follows the target vehicle speed even in this extremely hard situation.

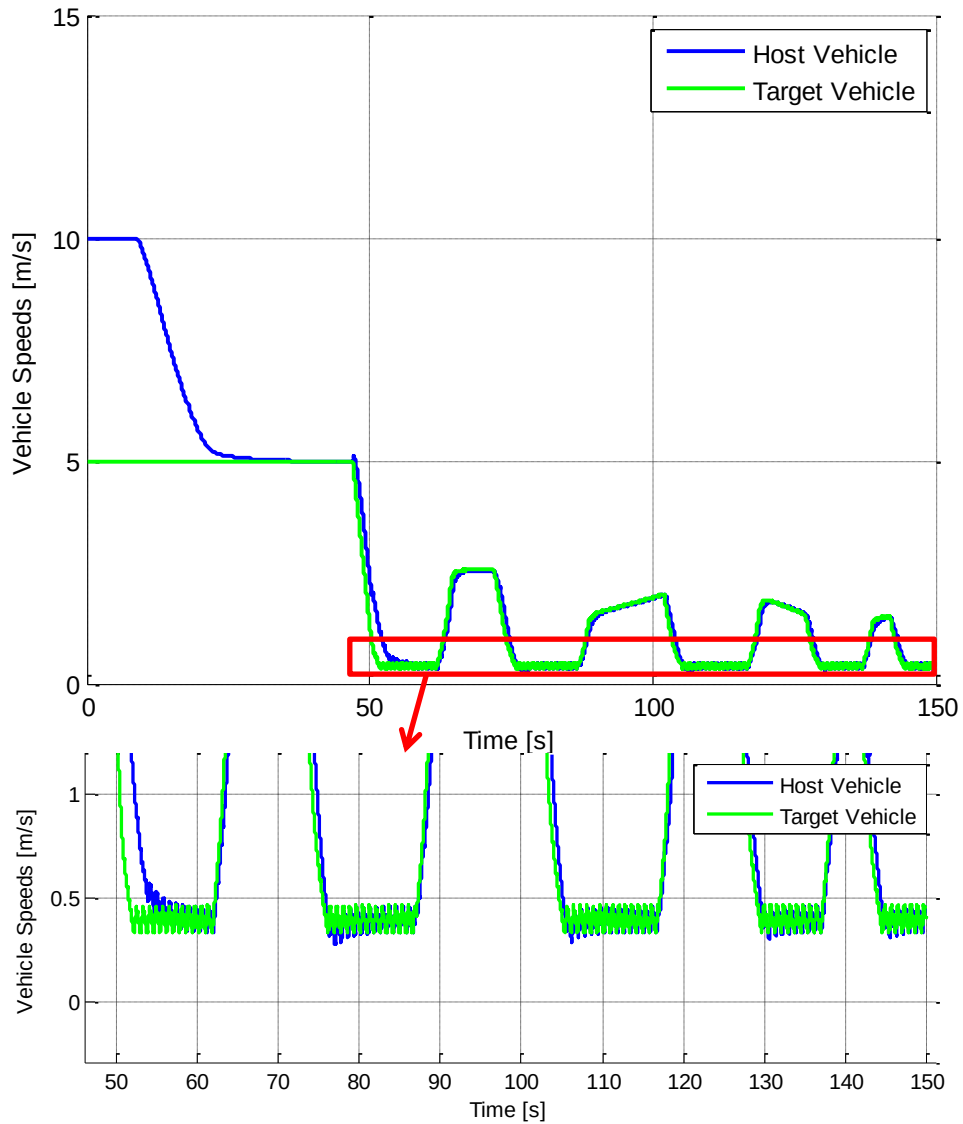


Figure 5.34 : Vehicle speeds.

Distance between the vehicles is shown in Figure 5.35. It can be seen from the sub-figure that the distance in stopping situations is around 2.25 m, which is equal to 2 m safety gap plus 0.25 m of half the host vehicle speed, where headway time T_h is equal to 0.5 s.

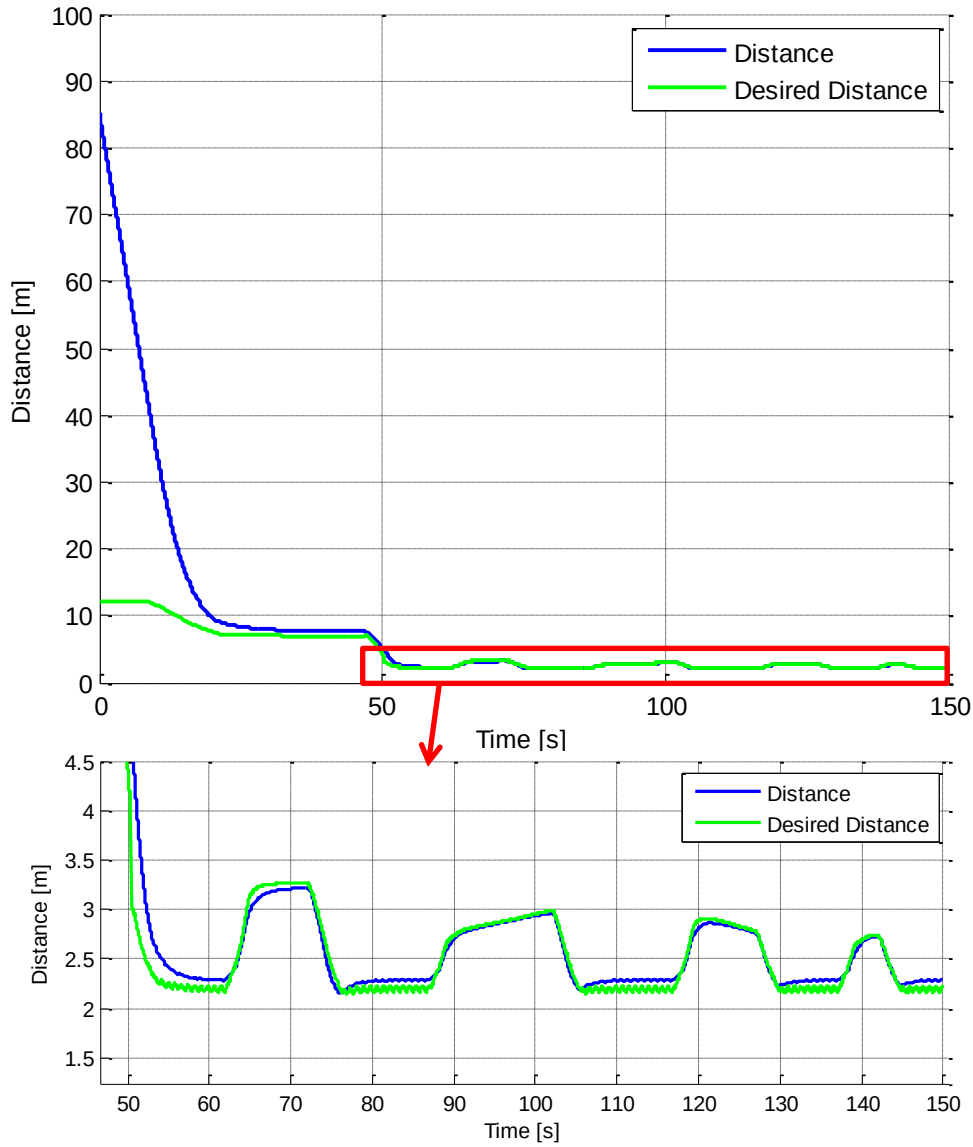


Figure 5.35 : Distance between the vehicles.

5.2 Comparison of PD Control with Fuzzy Logic Control

In section 5.1, Stop and Go Control System is tested with PD Control and results are presented for all scenarios, which are described in section 4.2. In this section, PD Control will be compared with Fuzzy Logic Control, which is already explained in section 4.3.1.3 in detail.

Initially, comparisons will be made for scenarios 1, 2, and 3 together. Figure 5.36 shows the vehicle speeds for the three scenarios, consecutively. Looking at the figure, it can be said that both of the two controllers give good results for speed following in general.

In scenarios 1 and 3, fuzzy logic can be said to show relatively better performance in terms of time and exactness while catching the target vehicle speed. However, the situation is vice versa for scenario 2. It can be inferred from these comparisons that fuzzy logic gives better results in scenarios where the speed is decreasing, whereas PD control gives better results in scenarios where the speed is increasing.

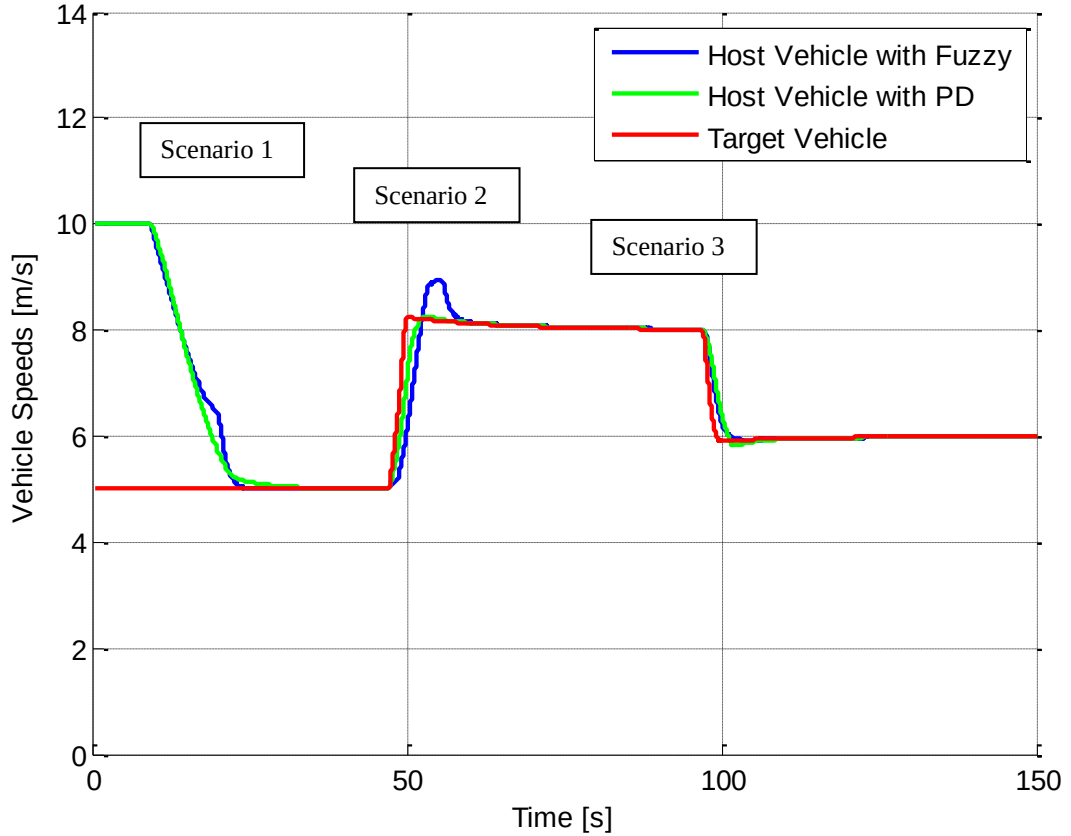


Figure 5.36 : Comparison of vehicle speeds for scenarios 1, 2, and 3.

Figure 5.37 illustrates the distance with the target vehicle for both fuzzy logic and PD controls for the three scenarios. For scenario 1, distance with the target vehicle for PD control in section 5.1.1 should be remembered here that the vehicle could not follow the target vehicle exactly with the desired distance. However, fuzzy logic gives significantly better results in distance following for scenario 1. In scenario 2, PD is better in performance since it does not show overshoot like fuzzy logic, and approaches steady state in a relatively shorter time. In scenario 3, performance of the fuzzy logic controller is again slightly better.

It can again be inferred from the figure that fuzzy logic control is better in scenarios where the speed is decreasing, and PD control is better in scenarios where the speed is increasing.

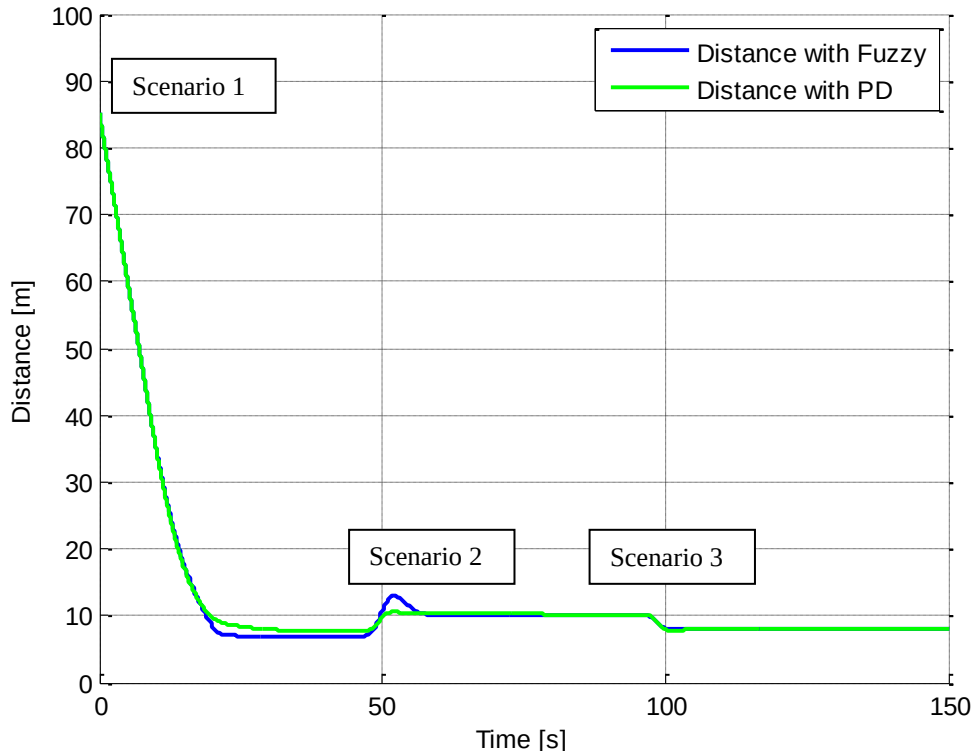


Figure 5.37 : Comparison of follow distance for scenarios 1, 2, and 3.

The second comparison is made for scenario 4, in which the vehicles are subjected to a hard condition, where a cut-in vehicle appears resulting in a sudden decrease in follow distance. This scenario starts after scenario 1.

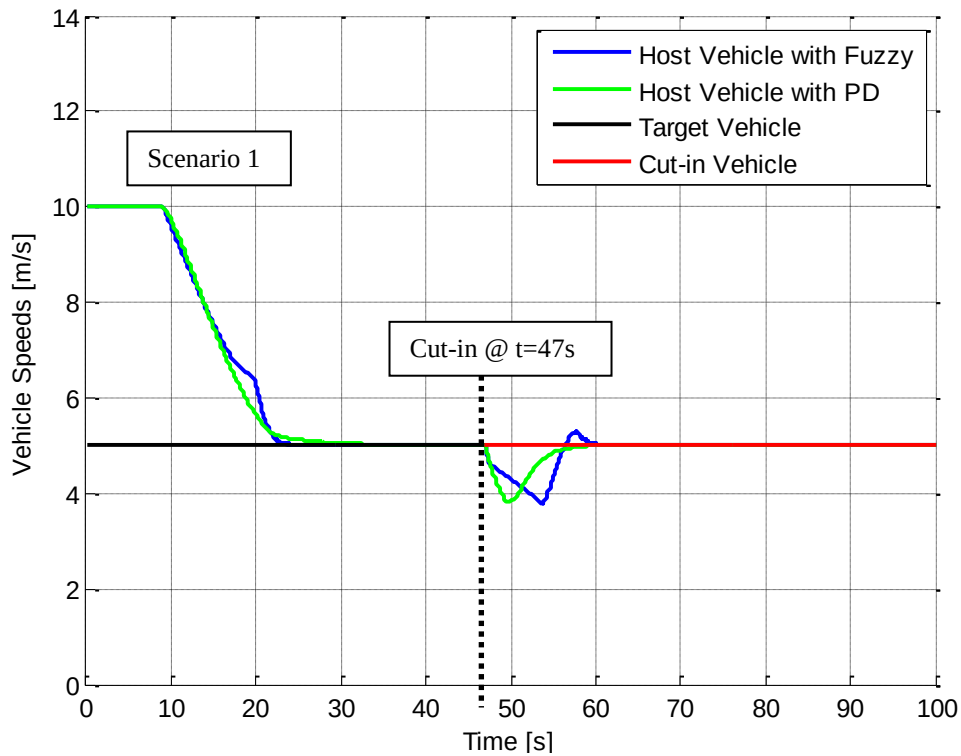


Figure 5.38 : Comparison of vehicle speeds for scenario 4.

Figure 5.38 shows comparison of the vehicle speeds for scenario 4. Here, PD control gives significantly quicker response without any overshoot when compared to fuzzy logic control.

In Figure 5.39, distance with the target vehicle and with the cut-in vehicle are illustrated for both PD and fuzzy logic controls. From the sub-figure, it can be seen that the PD controller gives quicker response without any overshoot. However, PD control cannot follow both the target and the cut-in vehicles with desired distance, which is equal to 7 m ($v_{host} \times T_h + \text{safety gap} = 5 \times 1 + 2 = 7$ m). Fuzzy logic follows the target vehicles exactly with the desired distance (steady-state error is zero).

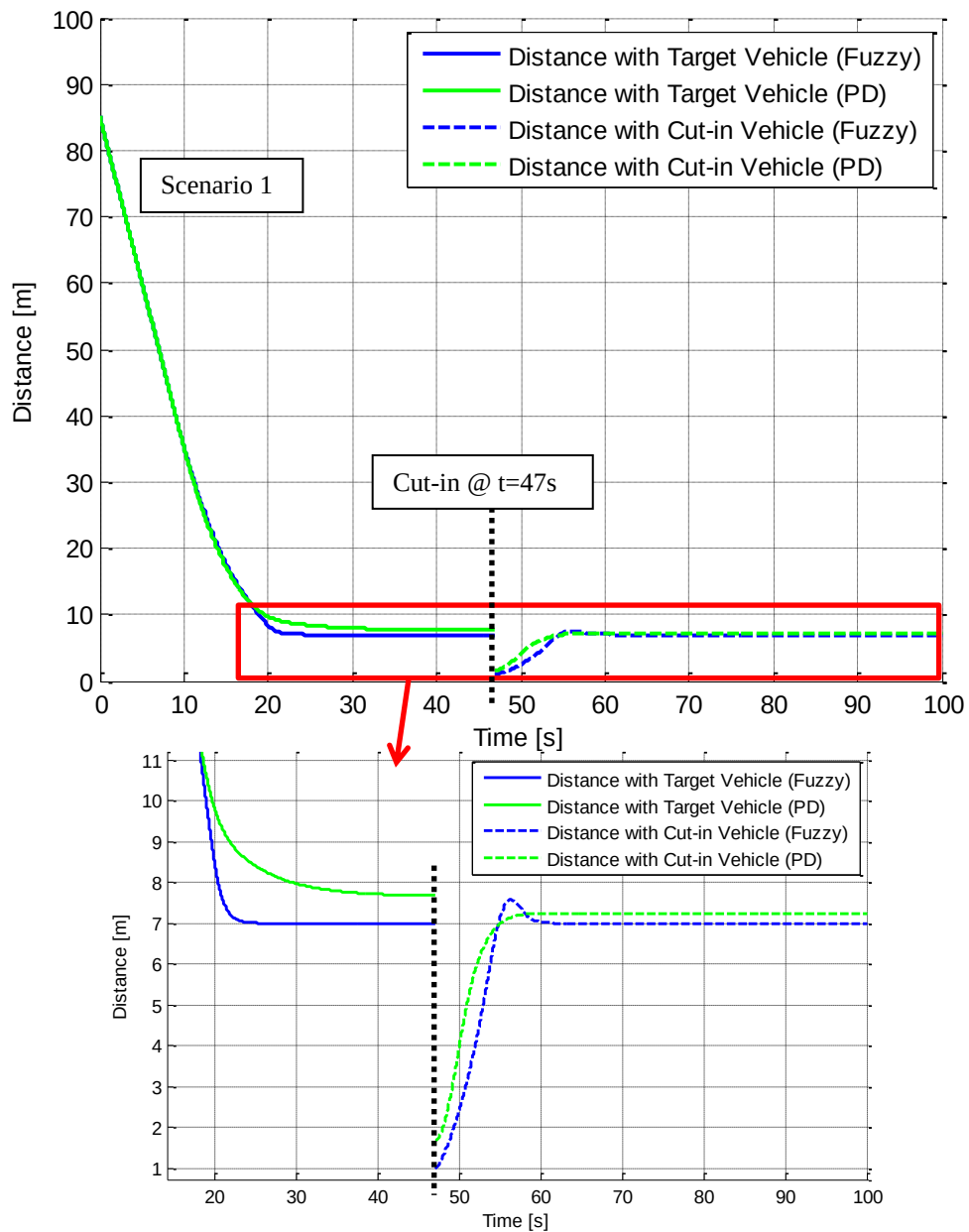


Figure 5.39 : Comparison of follow distance for scenario 4.

Finally, PD and fuzzy logic controllers are compared for scenario 8. Figures 5.40 and 5.41 show the vehicle speeds and follow distances for both controllers, respectively. The scenario starts after scenario 1.

As can be seen from the sub-figures, fuzzy logic controller shows overshoot both in speed decreasing and speed increasing situations for both figures. In addition, time to reach steady-state is longer in fuzzy logic control. Although steady-state error is not zero in PD control, it can be said that PD control is superior in performance for both speed and distance following for scenario 8.

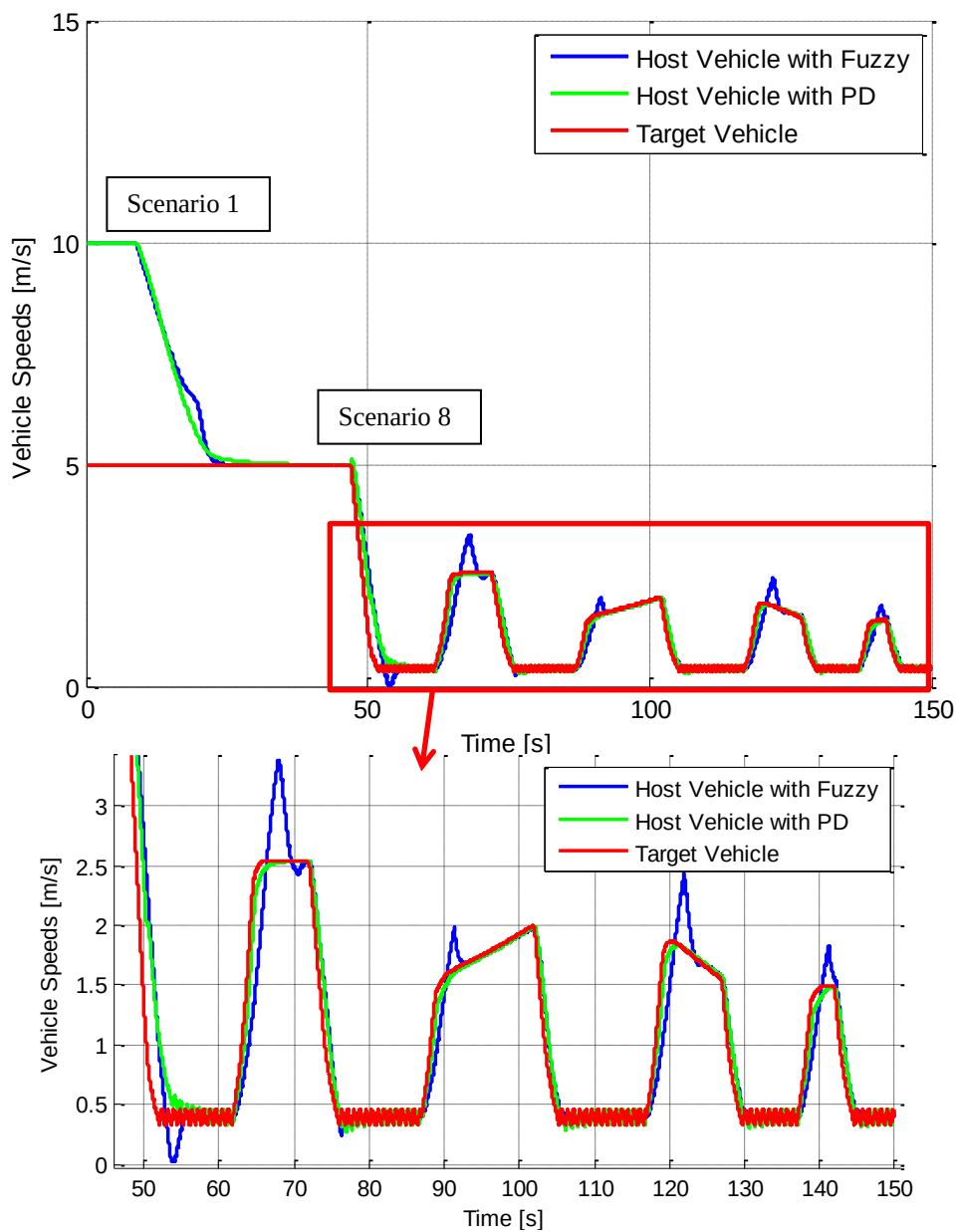


Figure 5.40 : Comparison of vehicle speeds for scenario 8.

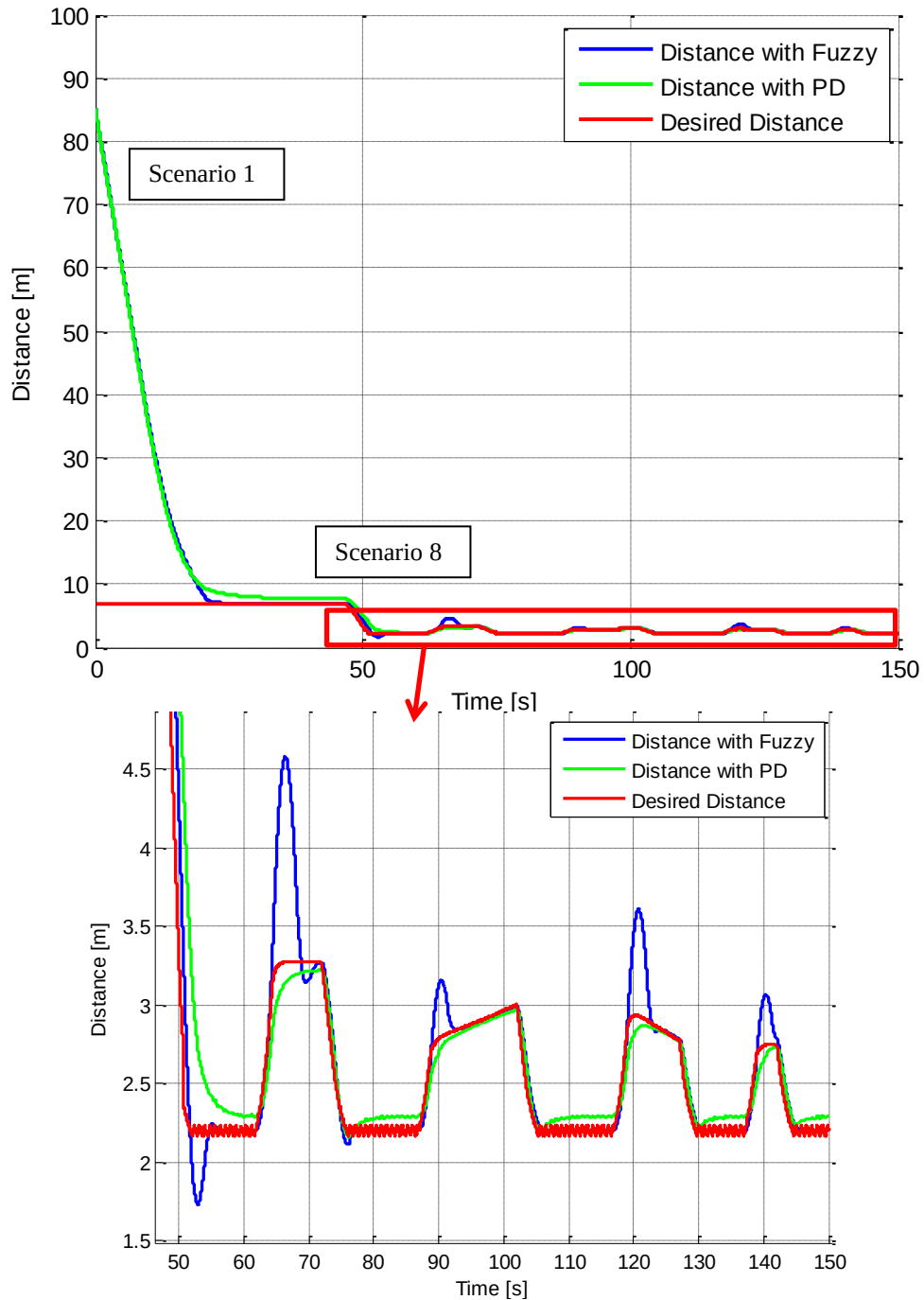


Figure 5.41 : Comparison of follow distance for scenario 8.

Since scenario 8, which represents stop and go mode of the control system, is the hardest scenario among all eight scenarios, controllers' performance can be best compared by looking this scenario.

It can be concluded that PD control shows better performance than fuzzy logic control in stop and go control system. However, both controllers can be improved by changing controllers' parameters and coefficients with more trial and errors.

6. CONCLUSIONS AND RECOMMENDATIONS

A stop and go control system is designed in this thesis for road vehicles that are mostly used in urban traffic. Two different types of controllers, named PD and Fuzzy Logic, are designed for comparison.

Performance of the control system is tested through extensive simulations for both controllers by the help of a vehicle model constructed in MATLAB/Simulink environment. Satisfactory results obtained from the simulations, which show the success of the designed vehicle model and the controllers, and results are presented with comparisons.

During the design of the vehicle model, almost all mechanical systems of a real vehicle are considered and their dynamics are included. A hierarchical control structure is adopted to provide differentiation of all possible scenarios that the vehicle might be subjected to in urban traffic, and to provide intervention to different systems of the vehicle.

The stop and go control system designed in this thesis is used to control a vehicle that is modeled in computer environment. It is also possible for the control system to control a real vehicle. Some additions and modifications should be included in the controller to be able to intervene into related systems of a real vehicle. In addition, inevitable disturbances and sensor noise in real life, which are neglected in this thesis, should also be considered in the design of the controller.

The vehicle equipped with stop and go control system in this study deals only with one target vehicle. However, there are so many vehicles in real urban traffic and scenarios related with potential targets should also be included in the controller structure.

As a future study, effects of stop and go control system on traffic flow can be investigated. As discussed in the thesis, stop and go control system is expected to harmonize traffic flow and thus increase the urban traffic capacity.

Stop and go control system can also be combined with platoon systems as a future study. Further, inter-vehicle communication systems could be added to this combination [57]. All these systems would help increasing traffic capacity and providing safe intelligent vehicles.

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