

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE
ENGINEERING AND TECHNOLOGY

**SLIDING MODE DISTURBANCE OBSERVER AND ADAPTIVE OPTIMAL
CONTROL ALLOCATION METHOD FOR STABILITY CONTROL OF ROAD
VEHICLES**

Ph.D. THESIS

Murat DEMİRCİ

Department of Control and Automation Engineering

Control and Automation Engineering Programme

MARCH 2014

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

**YOL TAŞITLARI STABİLİTE KONTROLÜ İÇİN KAYMA KIPLI BOZUCU
GÖZLEYİCİ VE UYARLAMALI OPTİMAL KONTROL ATAMA METODU**

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To my spouse and my daughter,

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ABBREVIATIONS

4WS4WD	: Four Wheel Steer Four Wheel Drive
4WAS	: Four Wheel Active Steering
AFS	: Active Front Steering
IWB	: Individual Wheel Braking
7DOF	: Seven Degrees-of-Freedom
2DOF	: Two Degrees-of-Freedom
COG	: Center of Gravity
MIMO	: Multiple Input Multiple Output
HIL	: Hardware-In-The-Loop
BIBO	: Bounded Input Bounded Output

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SLIDING MODE DISTURBANCE OBSERVER AND ADAPTIVE OPTIMAL CONTROL ALLOCATION METHOD FOR STABILITY CONTROL OF ROAD VEHICLES

SUMMARY

This thesis proposes a novel adaptive optimal control allocation method for coordination of the actuators in stability control of road vehicles in addition to sliding mode disturbance observers for yaw moment control and steering control problems in vehicle dynamics control. Moreover, a sliding mode MIMO disturbance observer is also introduced in the thesis and applied to the decoupled control problem of four wheel active steering vehicles. The sliding mode MIMO disturbance observer is applicable to m -input m -output square MIMO uncertain systems where decoupled SISO loops are desired.

Coordination term in vehicle dynamics control refers to distribution of a required yaw moment among available actuators of a vehicle, e.g. front and rear wheel active steering actuators, mechanical braking actuators, electric motors, etc. and allocation of the actuators in an optimal manner to generate the required yaw moment. An efficient, convergent, optimal and stability guaranteed control allocation algorithm is very crucial in vehicle dynamics control for vehicle safety. The algorithm should include vehicle dynamic limitations (tyre friction circle limitation, road adhesion, etc.) and actuator limitations.

If the control allocation is a nonlinear optimization problem, it is difficult to find an analytical solution and static-iterative optimization methods are mostly used in the literature. However, the stability of an optimal control allocation method should be guaranteed in a vehicle safety system and there is no an explicit way to show the stability of a static optimization method. Moreover, the static-iterative optimization based control allocation requires the nonlinear optimization problem to be solved in every time step for every corresponding yaw moment reference and this requires a high computational resource in real-time implementation.

As a solution to the above mentioned issues related to the static optimization based control allocation methods, an adaptive optimal control allocation method based on Lagrangian neural-networks is proposed in the thesis. The proposed control allocation method avoids the requirement of solving the optimization problem explicitly in every time step and the convergence and stability of the control allocation are guaranteed.

The typical presence of large amounts of uncertainty in vehicle dynamics for yaw stability enhancement mandates the use of a robust controller. Time varying parameters and the highly nonlinear nature of the vehicle lateral and yaw motion dynamics make it very suitable for sliding mode control (SMC), which is a widely used robust control method because of its robustness against parameter and model uncertainties as well as external disturbances. Two sliding mode based disturbance observer approaches for vehicle steering and yaw moment control problems are proposed in the thesis. The controllers are based on estimation of equivalent input extended-disturbance on the control input channel. The extended-disturbance includes modeling and parameter uncertainties of the vehicle in addition to external yaw disturbances.

The proposed sliding mode yaw moment controllers and adaptive optimal control allocation method are applied to develop novel integrated vehicle stability control systems for four wheel steer four wheel drive (4WS4WD) electric vehicles and conventional road vehicles. The proposed integrated vehicle stability controllers have a hierarchical control structure in which three layers called upper layer, middle layer and lower layer exist. The upper layer includes the proposed sliding mode yaw moment controllers whose output is virtual yaw moment control input to be generated by the middle and lower layers. The middle layer deals with control allocation by using the proposed adaptive optimal control allocation method and the lower layer includes servo control loops for tracking of reference actuator commands.

In the proposed stability control system for 4WS4WD electric vehicles, the adaptive optimal control allocation algorithm is used to coordinate the four wheel steering actuators and four in-wheel electric motors. On the other hand, in the stability control system developed for conventional road vehicles, it is used to coordinate the active front steering and individual braking actuators. Optimal electrical motor torques, steering correction angles and individual braking torques are calculated by solving a

nonlinear control allocation problem which includes a yaw moment equality constraint, the actuator and tire force inequality constraints and a cost function of adaptive weighted square sum of the tire forces. The proposed stability control systems are tested on a Fiat Doblo Van by simulation and real-time experiments. While the vehicle dynamics are modelled and simulated in CarSim simulation software, the control systems in Matlab/Simulink. The real-time experiments are carried out on a dSPACE MicroAutoBox in hardware-in-the-loop simulations with CarSim RT. The simulation and experimental results show that the proposed control allocation method successfully allocates the actuators in an optimal manner and it is concluded that the proposed stability control systems improve the stability and handling of road vehicles even in critical driving maneuvers.

YOL TAŞITI STABİLİTE KONTROLÜ İÇİN KAYMA KIPLI BOZUCU GÖZLEYİCİ VE UYARLAMALI OPTİMAL KONTROL ATAMA METODU

ÖZET

Her yıl trafiğe çıkan araç sayısı giderek artmaktadır ve milyonlarca insan da trafik kazasında hayatını kaybetmektedir. Trafik kazalarının büyük bir çoğunluğu sürücü hatalarından kaynaklanmaktadır. Bu durum sürüş güvenliğini arttırıcı araç teknolojilerinin geliştirilmesinin gerekliliğini ortaya koymuştur. ABS, TCS, ESP, vb. aktif güvenlik sistemleri artık birçok araçta standart ürün olarak yer almaktadır.

Aktif güvenlik sistemlerinin en önemlilerinden biri literatürde ESP, ESC, VSC, vb. isimlerle anılan araç savrulma stabilizasyonu kontrolcüsüdür. Savrulma stabilizasyonunda amaç, aracın savrulma hızı ve kayma açısını istenen referans değerlerde tutarak aracın yanal dinamiğini iyileştirmektir.

Bu çalışmada, yol taşıtı stabilite kontrol sistemlerinde eyleyicilerin koordinasyonu için kullanılmak üzere yeni bir uyarlamalı optimal kontrol atama metodu ile taşıt dinamik kontrolü için kayma kipli bozucu gözleyicileri önerilmiştir. Konvansiyonel kontrol atama metotlarından farklı olarak kararlılığı ve yakınsaklığı ispatlanmış bir metot önerilmekte ve bu metot ile kontrol atama optimizasyon problemlerinin her kontrolcü çevriminde çözüm gereksinimi ortadan kaldırılmaktadır. Bununla birlikte çok girişli-çok çıkışlı dinamik sistemleri birbirinden bağımsız ve birbirini etkilemeyen tek girişli-tek çıkışlı dinamik sistemlere dönüştürmek için çok girişli-çok çıkışlı kayma kipli bozucu gözleyici önerilmiş ve performansı yol taşıtı dinamik kontrol problemine uygulanarak test edilmiştir.

Taşıt dinamiği kontrol sistemlerinde koordinasyon terimi, taşıtın stabilizasyonu için gerekli savrulma momentinin direksiyon eyleyicileri, mekanik fren eyleyicileri, elektrik motorları, vb. mevcut eyleyiciler arasında dağıtımını ve ihtiyaç duyulan savrulma momentinin üretilmesi amacıyla eyleyicilerin atanmasını ifade etmektedir. Taşıt dinamiği kontrol sistemlerinde verimli, yakınsak, optimal ve kararlı bir kontrol atama algoritması kullanmak taşıt güvenliği açısından çok önemlidir. Eyleyici limitleri ile taşıt dinamiği limitlerinin de kontrol atama algoritmasında

dikkate alınması gerekmektedir. Eğer kontrol atama algoritması, doğrusal olmayan bir optimizasyon problemi içeriyor ise, analitik bir çözüm bulmak oldukça güçtür ve genellikle statik bir optimizasyon metodu ile problem çözülmeye çalışılmaktadır. Bir taşıt güvenliği sisteminde kullanılan kontrol atama algoritmasının kararlı olması ve mutlaka optimal değere yakınsaması gerekir. Fakat, kullanılan bir statik optimizasyon metodunun kararlılığını doğrudan göstermek mümkün değildir. Bununla birlikte kontrol atama probleminin her kontrol çevriminde istenen referans savrulma momenti girişine karşılık çözüm gereksinimi gerçek uygulamada çok fazla bilgisayar eforu ve dolayısıyla hızlı ve güçlü donanımlar gerektirir.

Bu tez kapsamında bu problemlerin çözümü için uyarlamalı optimal kontrol atama metodu önerilmiştir. Önerilen kontrol atama metodu ile kontrol atama optimizasyon problemlerinin her kontrolcü çevriminde çözüm gereksinimi ortadan kaldırılmaktadır. Tez kapsamında önerilen kontrol atama metodunun kararlılığı Lyapunov kararlılık teorisi ile analiz edilmiş ve kararlı olduğu gösterilmiştir.

Taşıt dinamiğindeki belirsizlikler ve bozucular savrulma stabilizasyonu kontrolü için dayanıklı bir kontrol tekniğinin kullanımını gerektirir. Doğrusal olmayan taşıt yanal ve savrulma dinamiği ve taşıt parametrelerinin zamanla değişmesinden dolayı, kayma kipli kontrol tekniği savrulma stabilizasyonu kontrolü için oldukça uygun bir kontrol yöntemidir. Bu çalışma kapsamında, taşıt savrulma stabilizasyonu için iki yeni kayma kipli kontrol tekniği önerilmiştir. Kontrolcüler, genişletilmiş bozucunun kontrol girişi tarafında kestirimi mantığı üzerine temellendirilmiştir. Genişletilmiş bozucu, harici bozuculara ilaveten modelleme ve parametre belirsizliklerini de içermektedir.

Önerilen kayma kipli bozucu gözleyicileri ve uyarlamalı kontrol atama metodu entegre taşıt stabilite kontrol sistemlerinin geliştirilmesinde kullanılmıştır. Tez kapsamında geliştirilen entegre stabilite kontrol sistemleri, içerisinde 3 ayrı katman barındıran hiyerarşik bir yapıdadır. Bu katmanlar üst seviye katmanı, orta seviye katmanı ve alt seviye katmanıdır. Üst seviye katmanda önerilen kayma kipli bozucu gözleyicileri taşıt stabilizasyonunu sağlamak için gerekli sanal savrulma kontrol momentini hesaplarken, orta seviye katmanında çalışan uyarlamalı optimal kontrol atama metodu mevcut eyleyicileri koordine ederek gerekli savrulma momentinin üretilmesini sağlamaktadır. Alt seviye katman ise referans eyleyici komutlarının takip edilmesini sağlayan servo kontrolcülerden oluşmaktadır.

Önerilen entegre stabilite kontrol algoritmalarının performansları, hem elektrikli hem de konvansiyonel Fiat Foblo Van aracı üzerinde hem simülasyon çalışmaları ile hem de gerçek zamanlı donanım içeren simülasyon çalışmaları ile test edilmiştir. Araç modelleme ve simülasyonu için CarSim yazılımı, gerçek zamanlı testler için de dSPACE MicroAutoBox kullanılmıştır. CarSim araç dinamiği modelleri yüksek serbestlik derecesine sahiptir ve kütüphanesinde yer alan dinamik modellerin birçoğu gerçek araç üstü testleri ile doğrulanmıştır. Uygulama aracı olarak kullanılan Fiat Doblo Van aracının hem elektrikli hem de konvansiyonel versiyonu için CarSim ortamında dinamik model oluşturulmuştur. Elektrikli Fiat Doblo aracı dört tekerden bağımsız olarak 4 adet elektrik motoru ile tahrik edilmekte ve hem ön hem de arka aks üzerindeki elektrik motorları ile tekerlerin yönelimi kontrol edilebilmektedir. Elektrikli Fiat Doblo aracı için geliştirilen entegre stabilite kontrol sisteminde aracın stabilizasyonu için gerekli savrulma momenti bu toplam 6 adet elektrik motoru koordine edilerek oluşturulmaktadır. Konvansiyonel araçlar için geliştirilen entegre stabilite kontrol sisteminde ise gerekli savrulma momenti aktif direksiyon sistemi ve dört tekerlekte yer alan fren eyleyicileri koordine edilerek oluşturulmaktadır. Geliştirilen kontrol algoritmaları, Matlab/Simulink ortamında modellenmiştir. CarSim ile Matlab/Simulink entegre bir şekilde çalıştırılarak kontrolcülerin performansları doğrulanmıştır. Gerçek zamanlı testlerde ise, kontrol algoritmaları Matlab/Simulink araçları kullanılarak C koduna çevrilmiş ve dSPACE MicroAutoBox üzerine yüklenerek CarSim ile entegre bir şekilde donanım içeren simülasyon testleri yapılmıştır.

Hem simülasyon hem de deneysel test sonuçları göstermektedir ki, önerilen uyarlanabilir kontrol atama metodu, araç stabilite kontrolünde kullanılan eyleyicileri optimal bir şekilde koordine edebilmekte ve önerilen kayma kipli bozucu gözleyiciler de modelleme ve parametre belirsizliklerine bağlı bozucular ve harici bozucuları başarılı bir şekilde engelleyebilmektedir. Ayrıca önerilen entegre taşıt stabilite kontrol sistemlerinin kritik sürüş manevralarında taşıtın stabilite ve tutunmasını sağlamada oldukça başarılı olduğu testlerle gösterilmiştir. Bununla birlikte gerçek zamanlı donanım içeren simülasyon testleri göstermiştir ki, önerilen uyarlamalı optimal kontrol atama metodu ve kayma kipli bozucu gözleyicileri üzerine temellendirilmiş entegre stabilite kontrol sistemleri, 10ms'den daha küçük çevrim periyotlarında kararlı bir şekilde çalışabilmektedir.

1. INTRODUCTION

1.1 Motivation And Literature Review

Conventional vehicle stability control systems, which are called Vehicle Dynamics Control (VDC), Electronic Stability Program, etc., are based on existing ABS systems in lower layer and individual braking is applied to tires according to upper layer yaw stability controller (van Zanten, 2000). In the upper layer yaw stability controllers, it is aimed to force yaw rate and side slip angle to track their desired reference values. The outputs of the controllers are generally tire reference longitudinal slip or tire reference braking/driving torque for the lower layer system. In the lower layer systems, the tracking controllers for the reference values generated by the upper layer yaw stability controller are employed.

The braking based stability control applications are improved with inclusion of brake-by-wire systems into vehicles. By the development of in-wheel drive or all wheel drive electric vehicles, it is possible to generate the required yaw moment by differential braking and active torque distribution to improve handling and stability of a vehicle. Various studies and researches on braking/driving based yaw moment control were conducted in the literature, e.g. Xiang et al. (2008), Guvenc et al. (2003), Mirzaei et al. (2008), Chung et al. (2003), Cerone et al. (2009), Goodarzi and Esmailzadeh (2007), Tahami et al. (2003), Geng et al. (2009), Shino and Nagai (2001), Zhao and Zhang (2009) and Chung and Yi (2006).

Active differential based yaw moment control systems were also studied in the literature, e.g. Canale et al. (2009), Canale et al. (2008) and Marino and Scalzi (2009). In this approach, an active differential actuator which allows directing the drive torque to only one of the wheels, left or right wheel is used to generate required yaw moment.

Steering based yaw moment control is another approach for improving vehicle handling and stability. Rear wheel steering, front wheel active steering and four wheels active steering based control systems were developed in the literature, e.g. A. Guvenc et al. (2009), She et al. (2007), Guvenc et al. (2004), Hiraoka et al. (2004),

Zheng and Anwar (2009), Yu and Gao (2007), Canale and Fagiano (2008), Li and Yu (2009), Morino et al. (2007) and Yih and Gerdes (2005). Two types of steering systems are used for front wheel active steering in the real implementation. The first one is a steer-by-wire actuator based steering system in which there is no mechanical connection between steering hand wheel and tires. In the vehicles equipped with steer-by-wire actuator, steering correction angle that is an output of the stability controller is added to driver steering input in an electronic control unit and total front tire steering angle is set by a servo motor. The second one is an auxiliary steering actuator based steering system in which the mechanical connection between steering hand wheel and tires is kept but, an additional auxiliary steering actuator is mounted on steering shaft or rack. In the vehicles equipped with auxiliary steering actuator, steering correction angle is added to driver steering input mechanically by an auxiliary servo motor. By proper design of the steering system, this mechanical addition cannot be reflected to the driver. For more details about steering systems, please refer to Anka et al. (2007).

The typical presence of large amounts of uncertainty in vehicle steering control for yaw stability enhancement mandates the use of a robust controller. In the inverse model based disturbance observer approach proposed by Guvenc et al. (2004) and A. Guvenc et al. (2009), a 2DOF vehicle steering controller is designed to improve yaw dynamics of a vehicle. The design is carried out by finding a region in controller parameter space where Hurwitz stability and a mixed sensitivity frequency domain constraint are simultaneously satisfied. It is difficult to tune controller parameters because the filter design must guarantee both the causality of the estimator and the stability of whole control system (She et al. 2007). Time varying parameters and the highly nonlinear nature of the vehicle lateral and yaw motion dynamics make it very suitable for sliding mode control (SMC), which is a widely used robust control method because of its robustness against parameter and model uncertainties, as well as external disturbances.

The other challenge of vehicle steering control is resulted from strong coupling between lateral and yaw motions of vehicle dynamics (Chen and Tan, 1998). The independent control of the yaw rate and the side slip angle (or lateral speed) improves vehicle handling, comfort, safety and lane keeping. Various decoupling solutions for four wheel active steering (4WAS) vehicles were proposed in several papers, e.g. Marino and Scalzi (2010), AGuvenc et al. (2010), Marino and Cinili

(2009) and Ackermann (1994). Guvenc et al. (2010) proposed MIMO disturbance observer for the decoupling problem of the four wheel active steering vehicles. Marino and Scalzi (2010) developed two PI controllers for reference tracking of yaw rate and side slip angle and H_{∞} based optimization approach for the decoupling problem.

The conventional braking based VDC systems which are based on ABS system have some drawbacks for longitudinal vehicle dynamic especially in an emergency braking maneuver on a μ -split road surface. The braking distance for a vehicle equipped with a conventional VDC system on a μ -split road surface is longer than that of unequipped vehicles since the tyre-road friction forces are not utilized completely to stop the vehicle due to additional aim of counteracting to the undesired yaw motion (Ding and Taheri, 2010). Moreover, we know that driving comfort is also worsened when the conventional VDC system is active since the longitudinal speed reduces significantly (Ghike et al. 2009; Trachtler, 2004). If an active steering based stability control system is available, the undesired yaw motion caused by unexpected yaw moment disturbances resulting from a situation like an emergency braking on a μ -split road surface can be attenuated by applying an extra steering correction angle to the front tire steering angle without awareness of the driver and it will be possible to apply the maximum longitudinal tire forces to the tires. Thus, the vehicle can be stopped in a shorter distance. Hence, integration of braking/driving and steering based stability control systems is widely studied in the literature to achieve a more effective and more comfortable solution, e.g. Ding and Taheri (2010), Ali et al. (2008), He et al. (2004), Yang et al. (2009), Mokhiemar and Abe (2002), Nagai et al. (1997) and Hwang et al. (2008).

In integrated vehicle stability control systems, an upper layer includes a yaw moment controller whose output is virtual yaw moment control input to be generated by the middle and lower layers. While the middle layer deals with control allocation and produces actuator reference commands, servo controllers in the lower layer make the actuators to track these reference values. For example, Goodarzi et al. (2007) developed a multilayer type of vehicle dynamics control system in which a motion controller is designed based on optimal control theory to generate the required yaw moment for lateral motion of the vehicle and traction force for longitudinal motion. An analytical solution with fuzzy parameter adaptation is derived for tire reference slip distribution and a sliding mode controller is used to control tire slip. Geng et al.

(2009) proposed a direct yaw moment controller based on LQR theory and an optimal in-wheel motor driving/braking force distributor. A fuzzy yaw moment controller and a fuzzy torque allocation algorithm were developed by Feiqianq et al. (2009). Tahami et al. (2003) combined the upper and middle layers and constructed a fuzzy logic based yaw stability control system whose output is the required torque change of in-wheel motors. The required torque change is symmetrically distributed to create yaw moment difference. The authors do not take into account the saturation limits of the in-wheel motor torques. Moreover, the in-wheel motor torque distribution is very simple and not optimal. Anwar (2005) designed a generalized predictive controller for yaw dynamics in the upper layer. The allocation of dictated yaw moment is achieved by using a heuristic torque distribution strategy for each electromagnetic brake at each wheel. Piyabongkarn et al. (2007) studied the use of torque biasing devices, i.e. electronically-controlled limited slip differential (ELSD) and center coupler. They use a proportional controller to control yaw rate and heuristic decision logic to activate the ELSD or center coupler.

Coordination term in integrated vehicle dynamics control refers to distribution of a required yaw moment among available actuators of a vehicle, e.g. front and rear wheel active steering actuators, mechanical braking actuators, electric motors, etc. and allocation of the actuators in an optimal manner to generate the required yaw moment. An efficient, convergent, optimal and stability guaranteed control allocation algorithm is very crucial in vehicle dynamics control for vehicle safety. The algorithm should include vehicle dynamic limitations (tyre friction circle limitation, road adhesion, etc.) and actuator limitations.

Mokhiamar and Abe (2006) derived an optimization problem for optimal distribution of tire forces in which the square sum of workloads of four wheels are taken as a cost function to be minimized. To simplify the optimization problem, they converted the optimization problem into a linear equation system and it is solved in every time step. They take into account tire friction circle or tire workload in the cost function but, since the friction circle limitation is not considered as an explicit nonlinear inequality constraint to assure tire forces in the friction circle, the solutions of the optimization algorithm may not be feasible. Moreover, the authors also do not take into account the actuator limitations in the optimization problem. Cho et al. (2008) proposed an optimal tire force distribution method in which the friction circle limitation was taken as a nonlinear inequality constraint for integrating active front

steering and conventional electronic stability control system. They derived an analytical solution for optimal tire forces by solving the optimization problem using Lagrange multipliers theory. Since they make some assumptions between lateral and longitudinal tire forces to simplify the optimization problem and to find an analytical solution, the results are not optimal. Moreover, the actuator constraints are not considered in the problem so that the actuators may saturate and the dictated yaw moment cannot be produced by the actuators. Wang and Longoria (2009) proposed a static optimization method called accelerated fixed point algorithm to solve the control allocation problem in which tire slip and slip angles are chosen as allocation variables instead of tire forces to simplify the optimization problem and to avoid the nonlinear friction circle limitation of tire forces. Plumlee et al. (2006) studied a quadratic programming based control allocation for distribution of front/rear wheel steering angles and longitudinal tire forces. Zhao and Zhang (2009) used sequential dynamic programming approach for distribution of longitudinal tire forces in a wheel motored electric vehicle. Ono et al. (2006) also proposed a sequential dynamic programming approach for tire force distribution in a four wheel distributed steering and four wheel distributed traction/braking systems. Tire grip margin and friction circle of each tire are estimated and used in the tire force distribution algorithm but, the authors do not take into account the actuator limitations in the optimization problem.

If the control allocation is a nonlinear optimization problem, it is difficult to find an analytical solution and static-iterative optimization methods are mostly used in the literature. However, the stability of an optimal control allocation method should be guaranteed in a vehicle safety system and there is no an explicit way to show the stability of a static optimization method. Moreover, the static-iterative optimization based control allocation requires the nonlinear optimization problem to be solved in every time step for every corresponding yaw moment reference and this requires a high computational resource in real-time implementation.

To overcome these issues, various dynamic control allocation methods were proposed in the literature. Tjonnas and Johansen (2010) proposed an adaptive control allocation method for active steering and brake allocation problem. The allocation variables are tire slip and steering angles. Acarman (2009) derived a control allocation method by using state-dependent Riccati equation technique for

coordination of individual braking torque and steering angle. Chen and Wang (2013) designed a hierarchical motion control structure for a four wheel independent drive electric vehicle. A dynamic control allocation method with an additional aim of power consumption minimization was used for coordination of four in-wheel electric motors. While the actuator limitations were taken in account, tire grip margin and friction circle limitation of each tire were not considered in the optimization problem. Moreover, it is also not clear whether the four in-wheel motors alone are capable to generate stabilizing yaw moments at high-speed critical vehicle maneuvers. A more recent literature survey on control allocation and its applications on yaw stability control was published by Johansen and Fossen (2013).

1.2 Thesis Objectives and Contributions

This thesis proposes two disturbance observer approaches for vehicle steering control which are based on sliding mode control theory. The controllers estimate the equivalent input extended-disturbance in the control input channel. The equivalent input extended-disturbance term refers to estimation of the extended disturbance in control input side. The extended-disturbance includes modeling and parameter uncertainties of the vehicle in addition to external yaw disturbances caused by side wind, μ -split braking, etc. The aim in the disturbance observer based steering control is to force the vehicle to track a 2DOF reference vehicle dynamics behavior by estimating the equivalent input extended-disturbance which is the front tire steering correction angle for the active steering system. The sliding mode based disturbance observer approaches developed in the thesis are the first known applications to vehicle steering control.

A multi input multi output (MIMO) version of the sliding mode disturbance observer scheme for decoupling MIMO nonlinear uncertain systems into single input single output (SISO) systems is also presented in the thesis. The cross-couplings between inputs and outputs, modeling and parameter uncertainties and external disturbances are treated as an extended disturbance to be attenuated by the MIMO disturbance observer. The proposed MIMO disturbance observer was applied to the four wheel active steering (4WAS) control problem of road vehicles. The proposed MIMO disturbance observer scheme is a novel solution for the decoupling problem of

MIMO nonlinear uncertain systems and is the first known application to four wheel active steering control.

The other important contribution of the thesis is a novel adaptive optimal control allocation method using Lagrangian neural networks which is applied to the control allocation problem in dynamics control of over-actuated vehicles. The proposed optimal control allocation method eliminates the requirement of solving optimization problem in every time step and it is a convergent and stability guaranteed method for the control allocation problems.

Two sliding mode yaw moment observers for yaw moment control problem are also developed in the thesis. The aim in the sliding mode yaw moment observer is to force the vehicle to track a reference vehicle dynamics behavior by estimating the equivalent input extended-disturbance which is the required stabilizing yaw moment.

By summarizing, the contributions of the thesis are

- Application of two sliding mode based disturbance observer approaches for vehicle steering and yaw moment control problems,
- A sliding mode MIMO disturbance observer approach for decoupling of MIMO nonlinear uncertain systems and application of the proposed sliding mode MIMO disturbance observer into the decoupled control problem of four wheel active steering vehicles,
- A novel control allocation method for over-actuated systems, which is an adaptive optimal control allocation method using Lagrangian neural-networks,
- Application of the proposed adaptive optimal control allocation method into coordination of steering actuators and in-wheel electric motors in four wheel steering four wheel drive electric vehicles.
- Application of the proposed adaptive optimal control allocation method into coordination of active front steering and individual braking actuators in conventional road vehicles.

2. VEHICLE DYNAMICS MODELING

The vehicle model used to study vehicle dynamics control has seven degrees-of-freedom. The lateral and longitudinal velocities of the vehicle and the yaw rate constitute three degrees-of-freedom related to the vehicle body. The wheel velocities of the four wheels constitute the other four degrees-of-freedom (Rajamani, 2006). Vehicle free body diagram at yaw plane and nomenclature used to define symbols in vehicle models are given in Figure 2.1 and Table 2.1 respectively.

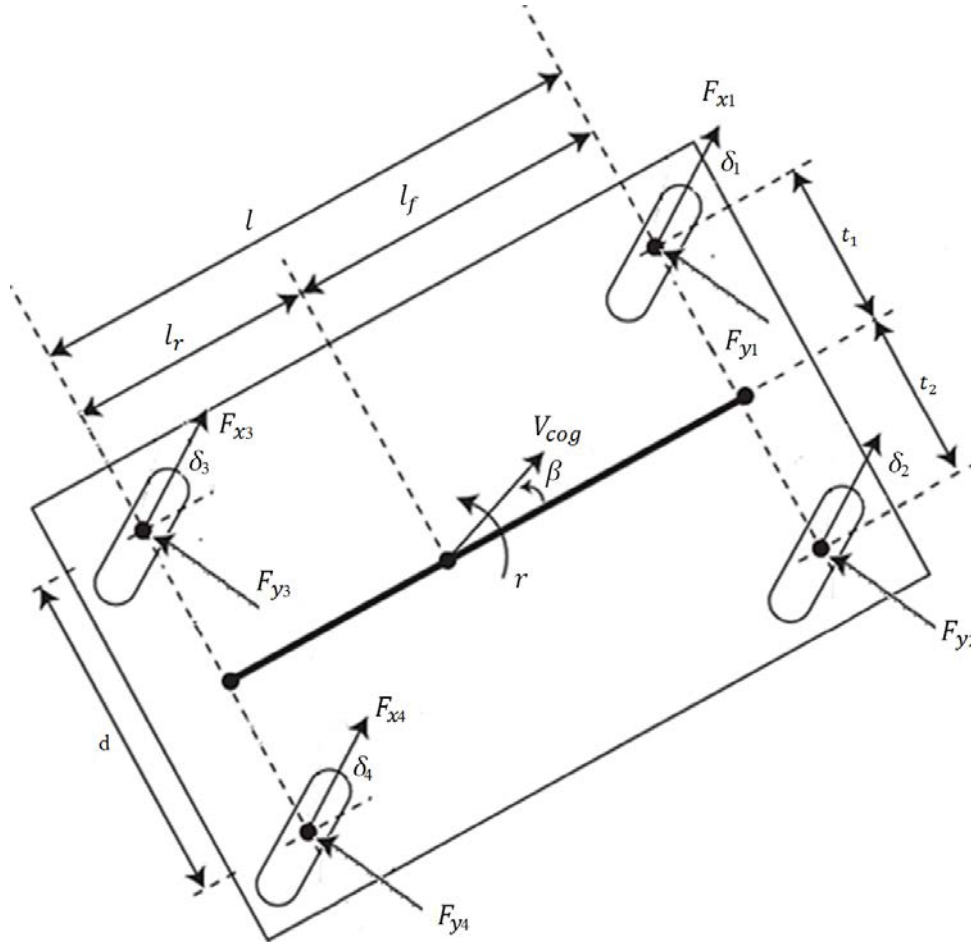


Figure 2.1 : Vehicle free body diagram at yaw plane.

Table 2.1 : Nomenclature.

Symbol	Description	Unit
M_z	Yaw moment control signal	Nm
T_i	Total tire torque (i=1,2,3,4)	Nm
T_{bi}	Tire braking torque (i=1,2,3,4)	Nm
T_{bi_max}	Max. tire braking torque (i=1,2,3,4)	Nm
T_{mi}	ith wheel motor torque (i=1,2,3,4)	Nm
T_{mi_max}	ith wheel max. motor torque (i=1,2,3,4)	Nm
δ_i	Tire steering angle (i=1,2,3,4)	rad
δ_{i_max}	Max. tire steering angle (i=1,2,3,4)	rad
δ_{fc}	Front steering actuator control input	rad
δ_{rc}	Rear steering actuator control input	rad
δ_d	Driver steering input	rad
V_{cog}	Vehicle speed at COG	m/s
V_x	Longitudinal velocity at COG	m/s
V_y	Lateral velocity at COG	m/s
β	Vehicle side slip angle	rad
r	Yaw rate	rad/sec
m	Vehicle mass	kg
F_{xi}	Longitudinal tire force (i=1,2,3,4)	N
F_{yi}	Lateral tire force (i=1,2,3,4)	N
I_z	Moment of inertia about z-axis	kgm ²
I_ω	Tire inertia	kgm ²
F_{xaero}	Aerodynamic force	N
θ	Inclination angle	rad
C_{xi}	Longitudinal tire stiffness	N/rad
$C_{yi} (C_i)$	Lateral tire stiffness	N/rad
C_r	Rear tire cornering stiffness in single-track vehicle	N/rad
C_f	Front tire cornering stiffness in single-track vehicle	N/rad
R_ω	Effective tire Radius	m
ω_i	Tire rotational speed (i=1,2,3,4)	rad/s
$I_{\omega i}$	Tire moment of inertia (i=1,2,3,4)	kgm ²
μ_i	Road friction coefficient (i=1,2,3,4)	-
s_i	Longitudinal tire slip (i=1,2,3,4)	-
α_i	Tire slip angle (i=1,2,3,4)	rad
$l_f (l_r)$	Distance from COG to front (rear) axle	m
$t_1 (t_2)$	Distance from COG to left (right) track	m
h_{cg}	Height of COG	m
d	Track width	m

2.1 7DOF Nonlinear Vehicle Dynamics Model

The 7DOF vehicle dynamics model is composed of lateral, longitudinal and yaw motion dynamics and four wheel tire dynamics. The longitudinal and lateral dynamics are

$$m(\dot{V}_x - rV_y) = -F_{xaero} - mg \sin \theta + \sum_{i=1}^4 (F_{xi} \cos \delta_i - F_{yi} \sin \delta_i) \quad (2.1)$$

$$m(\dot{V}_y + rV_x) = \sum_{i=1}^4 (F_{yi} \cos \delta_i + F_{xi} \sin \delta_i) \quad (2.2)$$

The yaw motion dynamic is

$$I_z \dot{r} = \sum_{i=1}^4 \{ (F_{xi} \sin \delta_i + F_{yi} \cos \delta_i) l_{yi} + (-F_{xi} \cos \delta_i + F_{yi} \sin \delta_i) l_{xi} \} \quad (2.3)$$

where $l_{x1} = l_{x3} = t_1$, $l_{x2} = l_{x4} = -t_2$, $l_{y1} = l_{y2} = l_f$ and $l_{y3} = l_{y4} = l_r$.

One of the important vehicle dynamic states is vehicle side slip angle and it is defined as

$$\beta \approx \frac{v_y}{v_x} \quad (2.4)$$

Then, the vehicle dynamics given in (2.1), (2.2) and (2.3) can be written in a compact form as follows (Kiencke and Nielsen, 2005).

$$\begin{bmatrix} \dot{V}_{COG} \\ \dot{\beta} \\ \dot{r} \end{bmatrix} = - \begin{bmatrix} 0 \\ r \\ 0 \end{bmatrix} + \begin{bmatrix} \frac{1}{m} \cos \beta & \frac{1}{m} \sin \beta & 0 \\ -\frac{1}{m} \sin \beta & \frac{1}{mV_{COG}} \cos \beta & 0 \\ 0 & 0 & \frac{1}{I_z} \end{bmatrix} \begin{bmatrix} \sum F_x \\ \sum F_y \\ \sum M + M_z \end{bmatrix} \quad (2.5)$$

where the total tire forces and yaw moment are

$$\begin{bmatrix} \sum F_x \\ \sum F_y \end{bmatrix} = \sum_{i=1}^4 \begin{bmatrix} \cos \delta_i & -\sin \delta_i \\ \sin \delta_i & \cos \delta_i \end{bmatrix} \begin{bmatrix} F_{xi} \\ F_{yi} \end{bmatrix} - \begin{bmatrix} F_{xaero} + mg \sin \theta \\ 0 \end{bmatrix} \quad (2.6)$$

$$\sum M = \sum_{i=1}^4 \{ (F_{xi} \sin \delta_i + F_{yi} \cos \delta_i) l_{yi} + (-F_{xi} \cos \delta_i + F_{yi} \sin \delta_i) l_{xi} \} \quad (2.7)$$

and M_z is called virtual yaw moment control input which is defined as follows.

$$M_z = \sum_{i=1}^4 \{ (\Delta F_{xi} \sin \delta_i + \Delta F_{yi} \cos \delta_i) l_{yi} + (-\Delta F_{xi} \cos \delta_i + \Delta F_{yi} \sin \delta_i) l_{xi} \} \quad (2.8)$$

On the other hand, the tire rotational dynamic is

$$I_\omega \frac{d\omega_i}{dt} = T_i - R_\omega F_{xi} \quad (2.9)$$

To model tire lateral and longitudinal tire forces, the following Dugoff tire model is widely used in the literature due to its combined tire force model.

$$F_{xi} = f_i C_{xi} s_i \quad (2.10)$$

$$F_{yi} = f_i C_{yi} \alpha_i \quad (2.11)$$

where

$$f_i = \begin{cases} 2 - \gamma_i, & \gamma_i < 1 \\ 1, & \gamma_i \geq 1 \end{cases} \quad (2.12)$$

$$\gamma_i = \frac{\mu_i F_{zi}}{2F_{Ri}} \quad (2.13)$$

$$F_{Ri} = \sqrt{(C_{xi} s_i)^2 + (C_{yi} \alpha_i)^2} \quad (2.14)$$

and the tire longitudinal slips and lateral slip angles are defined as

$$s_i = \frac{R\omega_i - v_i}{\max(\omega_i, v_i)} \quad (2.15)$$

$$\alpha_{1,2} = \delta_{1,2} - \begin{cases} \tan^{-1}\left(\frac{V_y + r l_f}{V_x + r t_1}\right), & i = 1 \\ \tan^{-1}\left(\frac{V_y + r l_f}{V_x - r t_2}\right), & i = 2 \end{cases} \quad (2.16)$$

$$\alpha_{3,4} = \delta_{3,4} - \begin{cases} \tan^{-1}\left(\frac{V_y - r l_r}{V_x + r t_1}\right), & i = 3 \\ \tan^{-1}\left(\frac{V_y - r l_r}{V_x - r t_2}\right), & i = 4 \end{cases} \quad (2.17)$$

Moreover, the tire center velocities and vertical load distributions are

$$v_i = \begin{cases} (V_x + r t_1) \cos \delta_1 + (V_y + r l_f) \sin \delta_1, & i = 1 \\ (V_x - r t_2) \cos \delta_2 + (V_y + r l_f) \sin \delta_2, & i = 2 \\ (V_x + r t_1) \cos \delta_3 + (V_y - r l_r) \sin \delta_3, & i = 3 \\ (V_x - r t_2) \cos \delta_4 + (V_y - r l_r) \sin \delta_4, & i = 4 \end{cases} \quad (2.18)$$

$$F_{zi} = \begin{cases} \frac{m g l_r}{2(l_f + l_r)} - \frac{\Sigma F_x h_{cg}}{2(l_f + l_r)} - \frac{\Sigma F_y h_{cg} l_r}{d(l_f + l_r)}, & i = 1 \\ \frac{m g l_r}{2(l_f + l_r)} - \frac{\Sigma F_x h_{cg}}{2(l_f + l_r)} + \frac{\Sigma F_y h_{cg} l_r}{d(l_f + l_r)}, & i = 2 \\ \frac{m g l_r}{2(l_f + l_r)} + \frac{\Sigma F_x h_{cg}}{2(l_f + l_r)} - \frac{\Sigma F_y h_{cg} l_r}{d(l_f + l_r)}, & i = 3 \\ \frac{m g l_r}{2(l_f + l_r)} + \frac{\Sigma F_x h_{cg}}{2(l_f + l_r)} + \frac{\Sigma F_y h_{cg} l_r}{d(l_f + l_r)}, & i = 4 \end{cases} \quad (2.19)$$

2.2 2DOF Linear Vehicle Dynamics Model

The 2DOF linear vehicle dynamics model is also called single track vehicle model and includes lateral and yaw motion dynamics of a vehicle. Under the following assumptions:

- the side slip angle and steering angle values are small such that,

$$\sin \beta \approx 0, \cos \beta \approx 1, \sin \delta_i \approx 0, \cos \delta_i \approx 1$$

- there is no difference between left and right track such that

$$\delta_1 = \delta_2 = \delta_f, \delta_3 = \delta_4 = \delta_r$$

$$\alpha_1 = \alpha_2 = \alpha_f, \alpha_3 = \alpha_4 = \alpha_r$$

- the inclination and wind forces are neglected
- the vehicle body speed, V_{COG} is constant over a limited time range
- and a linear tire model is used to define lateral forces as follows

$$F_{y1} + F_{y2} = F_{yf} = 2C_f\alpha_f, F_{y3} + F_{y4} = F_{yr} = 2C_r\alpha_r$$

the nonlinear vehicle dynamics model given in (2.5), (2.6) and (2.7) can be reduced into the following 2DOF vehicle dynamis model.

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} b_{11} & b_{12} \\ b_{21} & b_{22} \end{bmatrix} \begin{bmatrix} \delta_f \\ \delta_r \end{bmatrix} + \begin{bmatrix} 0 \\ b_{23} \end{bmatrix} M_z + \begin{bmatrix} Z_1 \\ Z_2 \end{bmatrix} \quad (2.20)$$

with

$$\begin{aligned} a_{11} &= -\frac{2(C_f + C_r)}{mV_{COG}} & a_{12} &= -1 - \frac{2(C_f l_f - C_r l_r)}{mV_{COG}^2} & a_{21} &= -\frac{2(C_f l_f - C_r l_r)}{I_z} \\ a_{22} &= -\frac{2(C_f l_f^2 + C_r l_r^2)}{I_z V_{COG}} & b_{11} &= \frac{2C_f}{mV_{COG}} & b_{12} &= \frac{2C_r}{mV_{COG}} \\ b_{21} &= \frac{2C_f l_f}{I_z} & b_{22} &= -\frac{2C_r l_r}{I_z} & b_{23} &= \frac{1}{I_z} \end{aligned}$$

and

$$M_z = \sum_{i=1}^4 \Delta F_{yi} l_{yi} - \Delta F_{xi} l_{xi} \quad (2.21)$$

where Z_1 and Z_2 represent modelling and parameter uncertainties and external disturbances.

2.3 CarSim Software and Fiat Doblo Van Model

CarSim is a vehicle dynamics simulation software and it can simulate the performance of vehicles in response to driver control inputs (steering, throttle, brakes, clutch, and shifting) in a given environment (road geometry, coefficients of friction, wind). Due to modular structure of CarSim modeling environment, it is possible to model different vehicle dynamic models from conventional vehicles to electric vehicles. The dynamics models of the CarSim are realistic and highly nonlinear models and most of its sub-component models are validated through the road tests. CarSim supports to make co-simulations with Matlab/Simulink models. Moreover, with additional RT options, it is also possible to make real-time hardware-in-the-loop simulations with well-known rapid controller prototyping systems like dSPACE targets (DS1005, DS1006, DS1103, MicroAutoBox), LabVIEW RT target, RT-Lab Target, etc.

The vehicle dynamics control systems developed in the thesis were tested on CarSim environment (Figure 2.3). The test vehicle is a Fiat Doblo Van (Figure 2.2). The conventional version of the Fiat Doblo Van is included in CarSim software but different powertrain and steering configurations of the Fiat Doblo Van were modeled in CarSim to simulate the performances of the proposed control systems. A distributed electric powertrain version of the Fiat Doblo Van with four wheel steering actuators was modeled to simulate the proposed integrated vehicle stability control system for four wheel steer four wheel drive electric vehicles.



Figure 2.2 : Fiat Doblo Van.

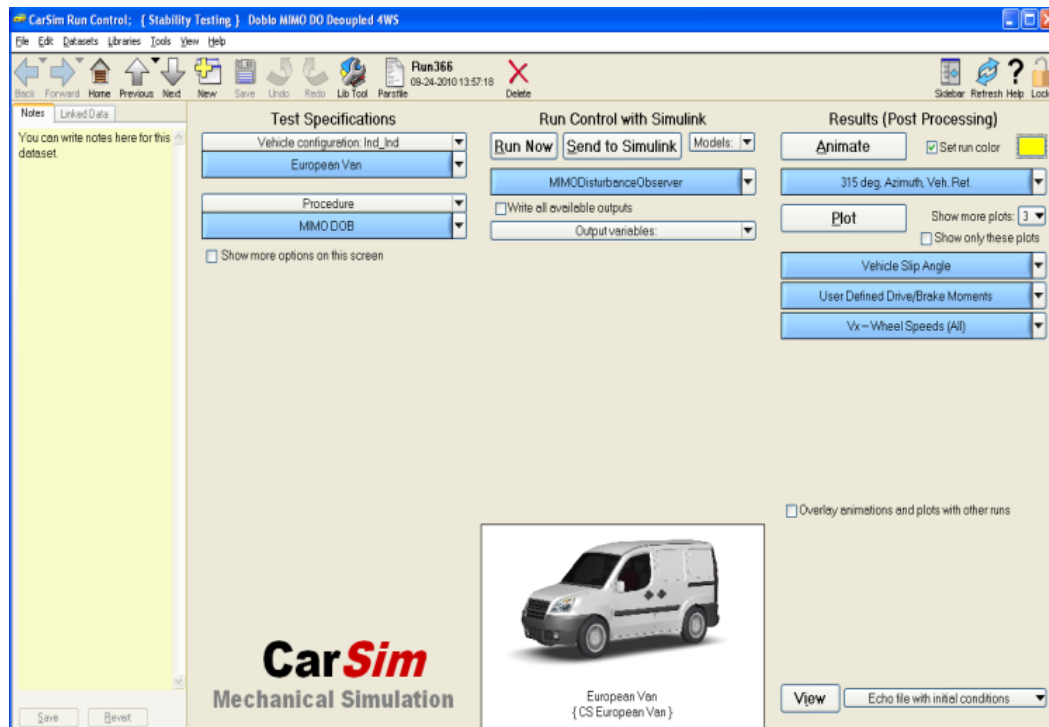


Figure 2.3 : Fiat Doblo Van modeled in CarSim.

As shown in Figure 2.4, it is possible to define various vehicle dynamics inputs (outputs) in the model as an external input (outputs) due to the modular structure of CarSim simulation environment. The subcomponents of the vehicle model can be model in a third party simulation environment (e.g. Matlab/Simulink, LabView, etc.) and can be co-simulated with CarSim.

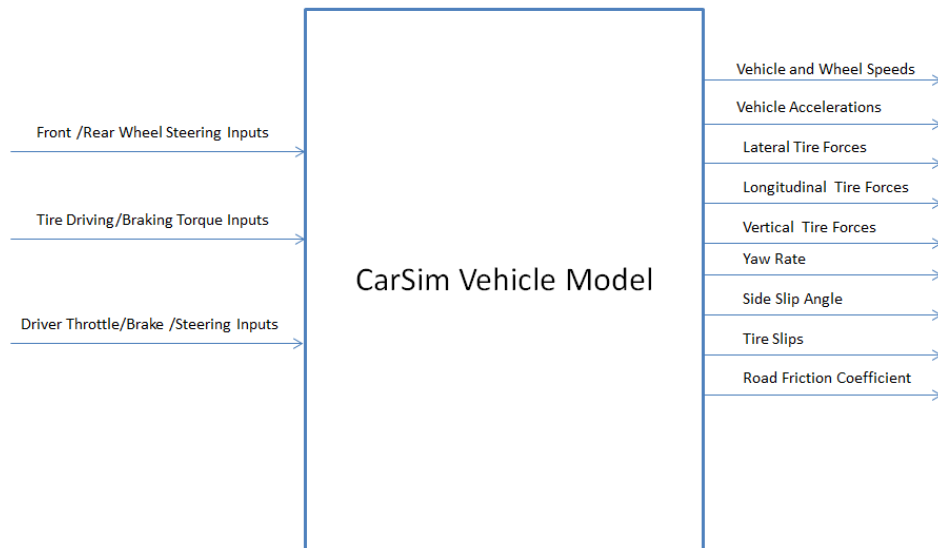


Figure 2.4 : Inputs and Outputs of a CarSim Model.

The CarSim models use a high degree-of freedom nonlinear vehicle dynamics with nonlinear tire models which also include highly nonlinear coupled dynamics. In addition to the 7DOF vehicle dynamics model given above, powertrain and steering dynamics, roll, pitch and vertical motion dynamics are also included in the model.

3. CONTROL PROBLEMS AND DESIGN METHODS AND TOOLS

3.1 Vehicle Steering Control Problem

The aim of the vehicle steering controllers given in the thesis is to force the vehicle to track a 2DOF reference vehicle dynamics behavior by estimating the equivalent input extended-disturbance which is the front tire steering correction angle for active steering system. The extended-disturbance includes modeling and parameter uncertainties of the vehicle in addition to external yaw disturbances caused by side wind, μ -split braking, etc.

By excluding the uncertainty and disturbance terms and applying Laplace transformation to (2.20) in the Chapter 2 for only front steering case ($\delta_r = 0$), $\beta(s)$ and $r(s)$ can be derived as

$$\beta(s) = \frac{(b_{11}s - b_{11}a_{22} + b_{21}a_{12})\delta_f(s) + b_{23}a_{12}M_z(s)}{(s - a_{11})(s - a_{22}) - a_{21}a_{12}} \quad (3.1)$$

$$r(s) = \frac{(b_{21}s + b_{11}a_{21} - b_{21}a_{11})\delta_f(s) + b_{23}(s - a_{11})M_z(s)}{(s - a_{11})(s - a_{22}) - a_{21}a_{12}} \quad (3.2)$$

where s is Laplace variable then, transfer functions from $\delta_f(s)$ to $r(s)$ and $\delta_f(s)$ to $\beta(s)$ are achieved as follows.

$$G_{r\delta}(s) = \frac{r(s)}{\delta_f(s)} = \frac{b_{21}s + b_{11}a_{21} - b_{21}a_{11}}{(s - a_{11})(s - a_{22}) - a_{21}a_{12}}$$

$$G_{\beta\delta}(s) = \frac{\beta(s)}{\delta_f(s)} = \frac{b_{11}s - b_{11}a_{22} + b_{21}a_{12}}{(s - a_{11})(s - a_{22}) - a_{21}a_{12}} \quad (3.3)$$

The steady-state values of the transfer functions in (3.3) can be calculated as follows.

$$\begin{aligned}\lim_{s \rightarrow 0} sG_{r\delta}(s) &= \frac{b_{11}a_{21} - b_{21}a_{11}}{a_{11}a_{22} - a_{21}a_{12}} \\ \lim_{s \rightarrow 0} sG_{\beta\delta}(s) &= \frac{b_{21}a_{12} - b_{11}a_{22}}{a_{11}a_{22} - a_{21}a_{12}}\end{aligned}\quad (3.4)$$

then, the following input gains, K_β and K_r are defined.

$$\begin{aligned}K_r &= \frac{b_{11n}a_{21n} - b_{21n}a_{11n}}{a_{11n}a_{22n} - a_{21n}a_{12n}} \\ K_\beta &= \frac{b_{21n}a_{12n} - b_{11n}a_{22n}}{a_{11n}a_{22n} - a_{21n}a_{12n}}\end{aligned}\quad (3.5)$$

where $a_{11n}a_{22n} - a_{21n}a_{12n} \neq 0$. Thus, the desired reference vehicle dynamics used in vehicle steering control are

$$\dot{x}_d = A_d x_d + B_{\delta d} \delta_d \quad (3.3)$$

with

$$A_d = \begin{bmatrix} \frac{-1}{\tau_d} & 0 \\ 0 & \frac{-1}{\tau_d} \end{bmatrix} \quad B_{\delta d} = \begin{bmatrix} \frac{K_\beta}{\tau_d} \\ \frac{K_r}{\tau_d} \end{bmatrix} \quad x_d = \begin{bmatrix} \beta_d \\ r_d \end{bmatrix} \quad (3.4)$$

$$\tau_d = \frac{b_{11n}a_{21n} - b_{21n}a_{11n}}{b_{21n}(a_{11n}a_{22n} - a_{21n}a_{12n})}$$

where δ_d is the desired command input signal. Nominal values of the vehicle parameters, a_{ij} and b_{ij} are used for calculation input gains, K_β and K_r given in (3.5) and the time constant, τ_d given in (3.7). The vehicle parameters are given in the Appendix B. Moreover, a vehicle speed dependent implementation of the reference vehicle dynamics is also considered in the reference vehicle dynamics. The steady-state gains and the time constant depend on the vehicle speed. The dependence of the reference vehicle dynamics on vehicle speed makes the steering controllers longitudinal speed-scheduled and the scheduling allows the controller to track the true reference vehicle dynamics (Guvenc et al. 2004).

The desired yaw rate and sideslip angle dynamics given in (3.6) and (3.7) cannot always be obtained. Moreover, it is not safe to try and to obtain the above desired reference dynamics if the road friction coefficient is unable to provide tire forces to support high yaw rate and side slip angle references. Hence, the desired reference vehicle dynamics are limited as follows (Rajamani, 2006).

$$r_d \leq 0.85 \frac{\mu g}{V} \quad (3.5)$$

$$\beta_d \leq \tan^{-1}(0.02\mu g) \quad (3.6)$$

where μ is the road friction coefficient and g is the acceleration of gravity.

Similar to the equation (2.20), 2DOF vehicle dynamics model for front wheel steering vehicles is

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} b_{11} \\ b_{21} \end{bmatrix} \delta_f + \begin{bmatrix} \varrho_1 \\ \varrho_2 \end{bmatrix} \quad (3.7)$$

where $\varrho = [\varrho_1 \ \varrho_2]^T$ includes unmodeled nonlinear dynamics, parameter uncertainties and external disturbances. The vehicle dynamics in (3.10) can be rewritten in terms of the reference vehicle dynamics given in (3.6) and (3.7) as follows.

$$\begin{aligned} \dot{x} &= A_d x + (A - A_d)x + B_{\delta d} \delta_f + (B_{\delta} - B_{\delta d}) \delta_f + \varrho \\ y &= Cx \end{aligned} \quad (3.8)$$

where $x = [\beta \ r]^T \in R^{2 \times 1}$ is the state vector, $y \in R^{2 \times 1}$ is the output vector, $\delta_f \in R^1$ is the front wheel steering input, $A \in R^{2 \times 2}$ and $B_{\delta} \in R^{2 \times 1}$ are the state and input matrix defined below, $C \in I_{2 \times 2}$ is an identity matrix and β , r and δ_f are available signals.

$$A = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}, \quad B_{\delta} = \begin{bmatrix} b_{11} \\ b_{21} \end{bmatrix} \quad (3.9)$$

If an extended-disturbance term, N is defined as

$$N = (A - A_d)x + (B_{\delta} - B_{\delta d}) \delta_f + \varrho \quad (3.10)$$

and under the condition that there exists an equivalent input extended disturbance (EIED), $v \in R^1$ in the control input channel, (3.11) can be written as

$$N = B_{\delta d} v \quad (3.11)$$

where $B_{\delta d} = [b_{\delta d1} \quad b_{\delta d2}]^T$ and $b_{\delta d1} \neq 0$ and $b_{\delta d2} \neq 0$. It is shown by She et al. (2008) that if $(A_d, B_{\delta d}, C)$ is observable and controllable and if $(A_d, B_{\delta d}, C)$ has no zeros on the imaginary axis, then there always exist an EIED in the control input channel. Then, the dynamics in (3.11) becomes

$$\dot{x} = A_d x + B_{\delta d} (\delta_f + v) \quad (3.12)$$

The detailed derivations of the extended-disturbance term, N with a sample application are given in the Appendix A.

Two different sliding mode disturbance observer based vehicle steering controllers are proposed in the thesis. The aim in the disturbance observer based steering control is to force the vehicle to track a 2DOF reference vehicle dynamics behavior by estimating the equivalent input extended-disturbance, v and compensating it by a front tire steering correction angle for active steering system in this work.

Two types of steering systems are used for front wheel active steering in the real implementation. The first one is a steer-by-wire actuator based steering system in which there is no mechanical connection between steering hand wheel and tires. In the vehicles equipped with steer-by-wire actuator, steering correction angle which is output of the controller is added to driver steering input in an electronic control unit and total front tire steering angle is set by a servo motor. The second one is an auxiliary steering actuator based steering system in which the mechanical connection between steering hand wheel and tires is kept but, an additional auxiliary steering actuator is mounted on steering shaft or rack. In the vehicles equipped with auxiliary steering actuator, steering correction angle is added to driver steering input mechanically by an auxiliary servo motor. By proper design of the steering system, this mechanical addition cannot be reflected to the driver.

3.2 Integrated Vehicle Stability Control Problem

In hierarchically integrated vehicle stability control systems, there exist layers called upper layer, middle layer, lower layer. The upper layer includes a yaw moment

controller whose output is virtual yaw moment control input to be generated by the middle and lower layers. Yaw rate and side slip angle are the controlled states and virtual yaw moment is the output of the yaw moment controller. It is aimed to force the vehicle to track a reference vehicle dynamics behavior. The presence of large amounts of uncertainty in vehicle dynamics mandates the use of a robust yaw moment controller. Time varying parameters and the highly nonlinear nature of the vehicle lateral and yaw motion dynamics make it very suitable for sliding mode control which is a widely used robust control method because of its robustness against parameter and model uncertainties as well as external disturbances.

The reference vehicle dynamics are generated using linear two degree-of-freedom vehicle dynamics. Road friction coefficient, longitudinal vehicle speed and driver steering input information are used in the reference vehicle dynamics. Driver steering input and yaw rate can be measured by using dedicated low cost sensors but the measurement of side slip angle and road friction coefficient requires high cost sensors which are not affordable for a stability control system in road vehicles. Since the vehicle is driven by four electric motors mounted in wheels, it is not suitable to derive the vehicle speed information from wheel speed sensors so that the vehicle speed should also be measured or estimated. Numerous estimation techniques were proposed for estimation of side slip angle, road friction coefficient and vehicle speed in the literature, e.g. Han and Huh (2011), Anderson and Bevely (2010), Cheli et al. (2007), Stephant et al. (2007) and Piyabongkarn et al. (2009).

The middle layer deals with control allocation and produces actuator reference commands. The control allocation term in vehicle dynamics control refers to distribution of a required yaw moment among available actuators of a vehicle, e.g. front and rear wheel active steering actuators, mechanical braking actuators, electric motors, etc. and allocation of the actuators in an optimal manner to generate the required yaw moment. An efficient, convergent, optimal and stability guaranteed control allocation algorithm is very crucial in vehicle dynamics control for vehicle safety. The algorithm should include vehicle dynamic limitations (tyre friction circle limitation, road adhesion, etc.) and actuator limitations. If the control allocation is a nonlinear optimization problem, it is difficult to find an analytical solution and static-iterative optimization methods are mostly used in the literature. However, the stability of an optimal control allocation method should be guaranteed in a vehicle safety system and there is no an explicit way to show the stability of a static

optimization method. Moreover, the static-iterative optimization based control allocation requires the nonlinear optimization problem to be solved in every time step for every corresponding yaw moment reference and this requires a high computational resource in real-time implementation.

The lower layer includes the drive systems and servo control loops for tracking the reference actuator commands calculated in the middle layer. Since the scope of this work does not include the lower layer actuator control, it is assumed here that all of the servo control loops are well tuned and the reference values are successfully tracked by the actuators.

3.2.1 Yaw Moment Control Problem

The aim in the yaw moment controller is to force the vehicle to track a reference vehicle dynamics behavior. The reference vehicle dynamics model for yaw moment control is

$$\dot{x}_d = A_d x_d + B_{md} M_d \quad (3.13)$$

with

$$A_d = \begin{bmatrix} \frac{-1}{\tau_d} & 0 \\ 0 & \frac{-1}{\tau_d} \end{bmatrix} \quad B_{md} = \begin{bmatrix} \frac{K_t}{\tau_d} \\ \frac{K_m}{\tau_d} \end{bmatrix} \quad x_d = \begin{bmatrix} \beta_d \\ r_d \end{bmatrix} \quad M_d = \frac{K_r}{K_m} \delta_d \quad (3.14)$$

where r_d and β_d are the yaw rate and vehicle side slip angle references respectively, K_r , K_t and K_m are the input gains, M_d is the desired virtual yaw moment input and τ_d is the time constant, which are defined as follows.

$$\begin{aligned} K_t &= \frac{b_{23}a_{12}}{a_{11}a_{22}-a_{21}a_{12}} & K_m &= \frac{b_{23}a_{11}}{a_{21}a_{12}-a_{11}a_{22}} \\ K_r &= \frac{b_{11}a_{21}-b_{21}a_{11}}{a_{11}a_{22}-a_{21}a_{12}} & \tau_d &= \frac{b_{11}a_{21}-b_{21}a_{11}}{b_{21}(a_{11}a_{22}-a_{21}a_{12})} \end{aligned} \quad (3.15)$$

The above desired reference yaw rate and side slip angle dynamics cannot always be obtained. It is not safe to try and obtain the desired reference dynamics if the road friction coefficient unable to provide the forces to support high yaw rate and side slip angle dynamics. Hence, the reference yaw rate and side slip angle dynamics are limited. These limitations are given in (3.8) and (3.9). Moreover, vehicle speed

dependent implementation of the reference vehicle dynamics is also considered here to track a true reference vehicle dynamics and hence, the input gains and the time constant defined in (3.18) depend on vehicle speed. Nominal values of the vehicle parameters, a_{ij} and b_{ij} given in the Appendix A are used for calculation the input gains.

The vehicle dynamics given in (2.20) can be written as follows in terms of the reference dynamics,

$$\dot{x} = A_d x + (A - A_d)x + B_{md}M_z + (B_m - B_{md})M_z + \rho \quad (3.16)$$

where $x = [\beta \ r]^T$ is state vector, $M_z \in R^1$ is the virtual yaw moment control input, $\rho = [\rho_1 \ \rho_2]^T$ is modeling and parameter uncertainties and external disturbances, $A \in R^{2 \times 2}$ and $B_m \in R^{2 \times 1}$ are state and input matrix. Similar to the case of the vehicle steering control derivations, there exists an equivalent input extended disturbance (EIED), $M_{ze} \in R^1$ in the control input channel such that

$$B_{md}M_{ze} = (A - A_d)x + (B_m - B_{md})M_z + \rho \quad (3.17)$$

then, the vehicle dynamics in (3.19) can be written as

$$\dot{x} = A_d x + B_{md}(M_z + M_{ze}) \quad (3.18)$$

where $B_{md} = [b_{md1} \ b_{md2}]^T$ and $b_{md1} \neq 0$ and $b_{md2} \neq 0$. It is shown by She et al. (2008) that if (A_d, B_{md}, C) is observable and controllable and if (A_d, B_{md}, C) has no zeros on the imaginary axis, then there always exist an EIED in the control input channel. Two different sliding mode disturbance observer based yaw moment controllers are proposed in the thesis. The aim in the disturbance observer based yaw moment control is to force the vehicle to track a 2DOF reference vehicle dynamics behavior by estimating and compensating the equivalent input extended-disturbance, M_{ze} . Since $x = [\beta \ r]^T$ and M_z are available signals, M_{ze} with the above structure can be observed with a suitable disturbance observer structure.

3.2.2 Integrated Stability Control Problem in 4WS4WD Electric Vehicles

The yaw moment control problem was given in the above section. Once the required yaw moment for stabilization of the vehicle is calculated using the yaw moment

controller, the remaining tasks are to optimally distribute the tire forces which generate the required yaw moment and to allocate the actuators. The actuators of 4WS4WD electric vehicles is given in Figure 3.1.

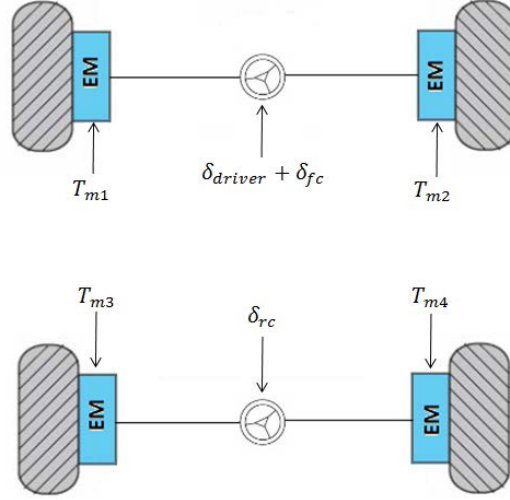


Figure 3.1 : The actuators of the 4WS4WD electric vehicle.

While distributing the tire forces, the nonlinear friction circle limitations of the tire forces should be taken into account. The nonlinear friction circle limitation is

$$(F_{xi})^2 + (F_{yi})^2 \leq (\mu_i F_{zi})^2 \quad (3.19)$$

For four wheel steering four wheel drive electric vehicles, front and rear wheel steering actuators and four electric motors mounted in wheels can be used for yaw moment generation. Actuator limitations should also be considered for a feasible control allocation. The limitations for steering actuators and electric motors are

$$-T_{mi_max} \leq T_{mi} \leq T_{mi_max} \quad (3.20)$$

$$-\delta_{i_max} \leq \delta_i \leq \delta_{i_max} \quad (3.21)$$

For mapping the tire forces into real actuator control signals, i.e. T_{mi} and δ_i , wheel rotational dynamics and simple tire model are considered. Since the wheel rotational dynamic is

$$I_\omega \frac{d\omega_i}{dt} = T_{mi} - T_{bi} - F_{xi}R \quad (3.22)$$

the motor torques are changed as $\Delta T_{mi} = \Delta F_{xi} R$. In a same manner, for small tire side slip angles, since the tire steering angle can be determined as

$$\delta_i \approx \begin{cases} \beta + \frac{l_f r}{V_{COG}} + \alpha_i, & i = 1, 2 \\ \beta - \frac{l_r r}{V_{COG}} + \alpha_i, & i = 3, 4 \end{cases} \quad (3.23)$$

where $\alpha_i \approx \frac{F_{yi}}{c_i}$, the tire steering angle are changed as $\Delta \delta_i \approx \frac{\Delta F_{yi}}{c_i}$. Thus, for four wheel steering four wheel drive electric vehicles, Optimal Tire Force Distribution Problem (OTFDP) is formulated as follows.

$$\begin{aligned} \min \quad & f(u) = \sum_{i=1}^4 (\gamma_i \Delta F_{xi}^2 + \rho_i \Delta F_{yi}^2) \\ \text{s. t.} \quad & h(u) = \sum_{i=1}^4 (\Delta F_{yi} l_{yi} - \Delta F_{xi} l_{xi}) - M_z = 0 \\ & (F_{xi} + \Delta F_{xi})^2 + (F_{yi} + \Delta F_{yi})^2 \leq (\mu_i F_{zi})^2 \\ & -T_{mi_max} \leq T_{mi} + \Delta F_{xi} R \leq T_{mi_max} \\ & -\delta_{i_max} \leq \delta_i + \frac{\Delta F_{yi}}{c_i} \leq \delta_{i_max} \end{aligned} \quad (3.24)$$

where $u = [\Delta F_{x1} \ \Delta F_{x2} \ \Delta F_{x3} \ \Delta F_{x4} \ \Delta F_{y1} \ \Delta F_{y2} \ \Delta F_{y3} \ \Delta F_{y4}]^T$, γ_i and ρ_i are longitudinal and lateral weighting factors are chosen by considering the tire workload such as

$$\gamma_i = \left(\frac{\Delta F_{xi}}{\mu_i F_{zi}} \right)^2, \quad i = 1, 2, 3, 4 \quad (3.25)$$

$$\rho_i = \left(\frac{\Delta F_{yi}}{\mu_i F_{zi}} \right)^2, \quad i = 1, 2, 3, 4 \quad (3.26)$$

A novel integrated vehicle dynamics control system for four wheel steering four wheel drive (4WS4WD) electric vehicles are proposed in the thesis and it is composed of an adaptive optimal control allocation method using Lagrangian neural networks for optimal distribution of tire forces and a sliding mode yaw moment observer for robust control of yaw dynamics. The proposed vehicle dynamics control scheme is composed of three layers and illustrated in Figure 3.2. The sliding mode disturbance observer bases the work proposed by Abidi and Sabanovic (2007). An equivalent input extended-disturbance term is defined and it refers to estimation of the extended disturbance in control input side. The disturbance includes modeling

and parameter uncertainties of the vehicle in addition to external yaw disturbances caused by side wind, yaw disturbance moment, etc. After estimation the extended disturbance, it is compensated such that the vehicle dynamics are forced to track the reference vehicle dynamics. Time varying parameters and the highly nonlinear nature of the vehicle lateral and yaw motion dynamics make it very suitable for sliding mode control which is a widely used robust control method because of its robustness against parameter and model uncertainties as well as external disturbances.

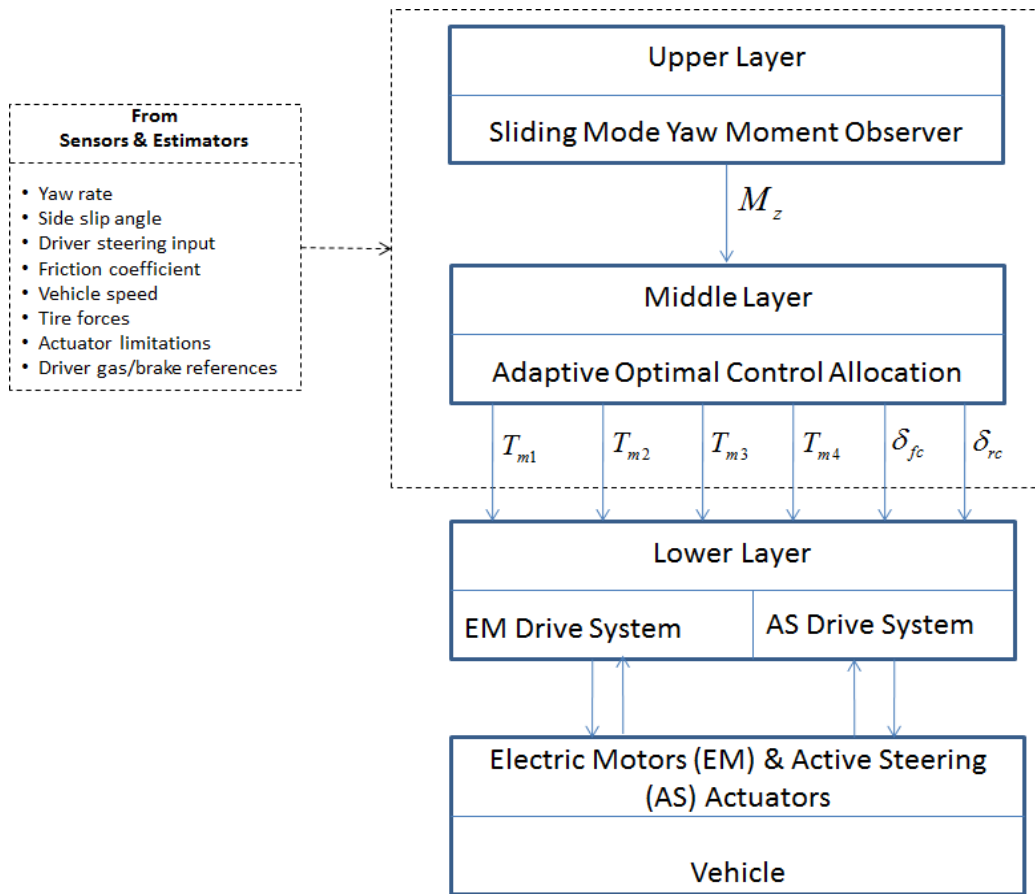


Figure 3.2 : The proposed hierarchical vehicle dynamics control diagram for 4WS4WD electric vehicles.

On the other hand, the proposed adaptive optimal control allocation method bases an optimization solution method proposed by Zhang and Constantinidis (1992) and Wah and Wang (1999). To tackle the problems in the static-iterative control allocation methods mentioned before, this work proposes a Lagrangian neural network approach to adaptively solve the nonlinear optimization problem, which avoids the requirement of solving the optimization problem explicitly in every time step and the

convergence and stability of the control allocation are guaranteed. The optimization problem is handled by Lagrange multipliers theory.

3.2.3 Integrated Stability Control Problem By Coordination of AFS and Individual Braking Actuators

The integrated stability control problem for the vehicles equipped with individual braking and active front steering (AFS) actuators is given in this section. The proposed integrated vehicle stability controller has a hierarchical control structure (Figure 3.3) in which three layers called upper layer, middle layer and lower layer exist.

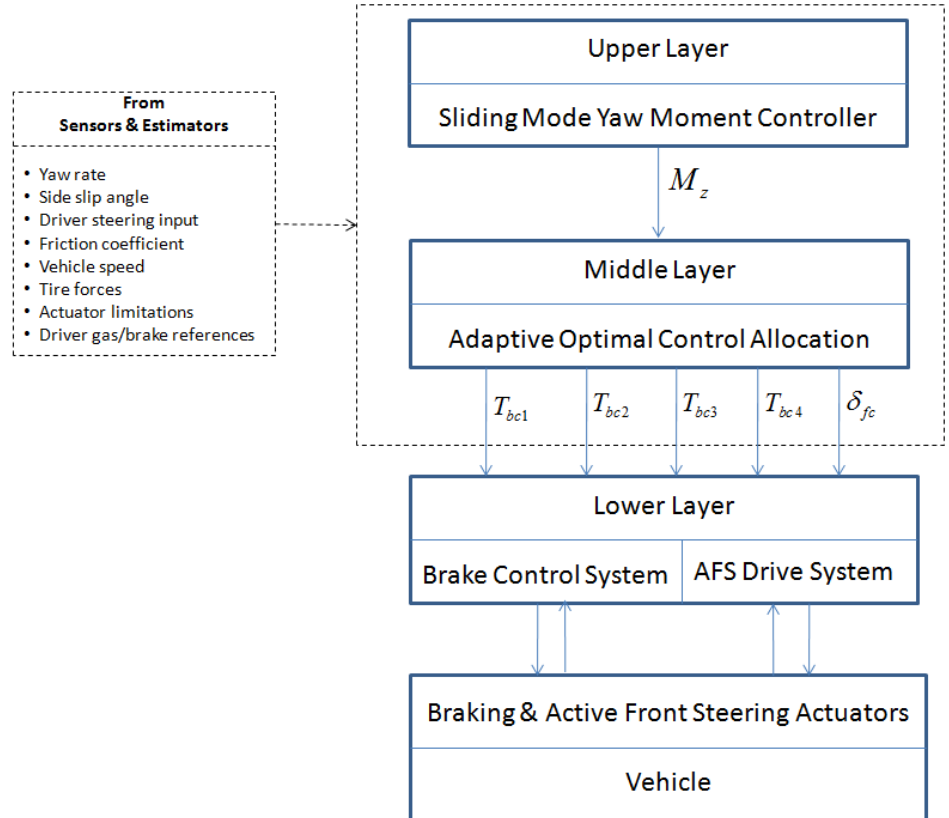


Figure 3.3 : The proposed hierarchical vehicle dynamics control diagram for conventional vehicles.

The upper layer includes a yaw moment controller whose output is virtual yaw moment control input to be generated by the middle and lower layers. The yaw moment controller bases the control structure proposed by Talole and Phadke (2008). The middle layer deals with control allocation. The required virtual yaw moment to stabilize the vehicle are generated by five actuators which are four individual braking actuators and active front steering actuator. Optimal braking torques and steering

correction angle are calculated by solving a nonlinear control allocation problem in which a yaw moment equality constraint, the actuator and tire force inequality constraints and a cost function of adaptive weighted square sum of the tire forces exist. The lower layer includes the braking and active front steering systems. Servo control loops for reference braking torques and steering angle calculated by the middle layer are included in the drive systems. Since the scope of this work does not include the lower layer actuator control, it is assumed here that all of the servo control loops are well tuned and the reference values are successfully tracked by the actuators. A well-controlled steer-by-wire Active Front Steering (AFS) actuator is modeled in s-domain as follows (Ackermann et al., 2002).

$$\frac{\eta\delta_f}{\delta_d + \delta_{fc}} = \frac{w_a^2}{s^2 + 2\zeta_a w_a s + w_a^2} \quad (3.27)$$

where $\zeta_a = 0.7$, $w_a = 2\pi \cdot 10\text{Hz}$, η , δ_f , δ_d and δ_{fc} are steering ratio, front tire steering angle, driver steering input and steering control input generated by the control allocation respectively.

An individually actuated braking system is considered in this study. A well-controlled braking actuator model is

$$\frac{T_{bi}}{T_{bdi} + T_{bci}} = \frac{w_b^2}{s^2 + 2\zeta_b w_b s + w_b^2} \quad (3.28)$$

where $\zeta_b = 0.7$, $w_b = 2\pi \cdot 10\text{Hz}$, $i=1,2,3,4$, T_{bi} , T_{bdi} and T_{bci} are the tire braking torque, the driver braking input and the braking torque control input generated by the control allocation respectively.

Derivation of the vehicle dynamics model for yaw moment controller design is given in the previous section. After calculating the required stabilizing yaw moment by the yaw moment controller in the upper layer, the remaining task is to generate this stabilizing yaw moment by allocating the individual braking and active front steering actuators. Similar to the optimal tire force distribution problem of 4WS4WD electric vehicles given in the previous section, the following nonlinear optimal tire force distribution problem is formulated for distributing the required yaw moment, M_z between the available actuators.

$$\begin{aligned}
\min \quad & f(u) = \sum_{i=1}^4 \gamma_i \Delta F_{xi}^2 + \sum_{j=1}^2 \rho_j \Delta F_{yj}^2 \\
\text{s. t.} \quad & h(u) = \sum_{j=1}^2 \Delta F_{yj} l_{yj} - \sum_{i=1}^4 \Delta F_{xi} l_{xi} - M_z = 0 \\
g(u) = & \begin{cases} (F_{xi} + \Delta F_{xi})^2 + (F_{yi} + \Delta F_{yi})^2 - (\mu_i F_{zi})^2 \leq 0, & i = 1,2 \\ (F_{xi} + \Delta F_{xi})^2 + (F_{yi})^2 - (\mu_i F_{zi})^2 \leq 0, & i = 3,4 \\ T_{bi} - \Delta F_{xi} R \leq T_{bi_max}, & i = 1,2,3,4 \\ -\delta_{j_max} - \delta_j - \frac{\Delta F_{yj}}{c_j} \leq 0, & j = 1,2 \\ \delta_j + \frac{\Delta F_{yj}}{c_j} - \delta_{j_max} \leq 0, & j = 1,2 \\ \Delta F_{xi} < 0, & i = 1,2,3,4 \end{cases} \quad (3.29)
\end{aligned}$$

where $u = [\Delta F_{x1} \ \Delta F_{x2} \ \Delta F_{x3} \ \Delta F_{x4} \ \Delta F_{y1} \ \Delta F_{y2}]^T$, γ_i and ρ_j are longitudinal and lateral weighting factors. It can be seen that the optimization problem includes one equality and sixteen inequality constraints. The weighting coefficients, γ_i and ρ_i in the cost function are chosen by considering the tire workload such as

$$\gamma_i = \left(\frac{\Delta F_{xi}}{\mu_i F_{zi}} \right)^2, \quad i = 1,2,3,4 \quad (3.30)$$

$$\rho_j = \left(\frac{\Delta F_{yj}}{\mu_j F_{zj}} \right)^2, \quad j = 1,2 \quad (3.31)$$

The solution of the optimization problem and coordination of the actuators are handled by the proposed novel adaptive optimal control allocation algorithm.

3.3 Model Reference Sliding Mode Control Approach For Yaw Moment and Steering Control Problems

A sliding mode model reference control approach with modeling and parameter uncertainty and disturbance estimation is derived in this section. As it is derived in (3.15) and (3.21) in the vehicle steering and yaw moment control problems that the vehicle dynamics model for controller design has a single input multiple output dynamics with the equivalent input extended disturbance term as follows.

$$\dot{x} = A_d x + B_d(u + n) \quad (3.32)$$

where $x \in R^{2 \times 1}$ is the state vector, $A_d \in R^{2 \times 2}$ is the nominal state matrix, $B_d \in R^{2 \times 1}$ is the control input matrix, $u \in R^1$ is the control input and $n \in R^1$ is the equivalent

input extended disturbance in the control input side, which represents the modeling and parameter uncertainties and external disturbances. Similar to the work proposed by Talole and Phadke (2008), the sliding mode model reference controller is derived by choosing the following sliding surface.

$$\sigma = \varphi x + z \quad (3.33)$$

with

$$\dot{z} = -\varphi A_d x - \varphi B_d u_d \quad (3.34)$$

$$z(0) = -\varphi x(0) \quad (3.35)$$

with $\varphi = [\varphi_1, \varphi_2] \in R^{1 \times 2}$, $\varphi_i > 0$ and u_d is the desired command input signal. If the equations (3.36), (3.37) and (3.38) are examined, it can be seen that if a control input u is designed ensuring sliding, then $\dot{\sigma} = 0$ implies

$$\dot{x} = A_d x + B_d u_d \quad (3.36)$$

which results perfect reference model tracking. By choosing a sliding surface as in (3.36), (3.37) and (3.38), the reaching mode in which the sliding mode controller lacks of robustness is eliminated such that the system is in sliding surface at the beginning of time, $\sigma(0) = 0$ and the controller tries to keep the system in sliding surface by forcing the derivative of the sliding surface to zero.

If the control input, u is defined as follows,

$$u = u_d + u_\sigma + u_n \quad (3.37)$$

where

$$u_\sigma = -k_\sigma \sigma, \quad k_\sigma > 0 \quad (3.38)$$

and u_n is

$$u_n(s) = \frac{-1}{\tau \varphi B_{\delta d}} \left(\sigma(s) + \frac{k_\sigma \varphi B_{\delta d}}{s} \sigma(s) \right) \quad (3.39)$$

the derivative of the sliding surface is forced to zero. Here, u_n is an estimation of the equivalent input extended disturbance, n , i.e. $u_n = -\hat{n}$.

Substituting the control input terms in (3.41) and (3.42) in to (3.35) yields

$$\dot{x} = A_d x + B_d(u_d - k_\sigma \sigma + \tilde{n}) \quad (3.40)$$

where $\tilde{n} = n - \hat{n}$.

3.4 Sliding Mode Disturbance Observer Approach For Yaw Moment and Steering Control Problems

By considering the system dynamics given in (3.35) and applying the sliding mode disturbance observer (SMDO) approach proposed by Abidi and Sabanovic (2007), the disturbance observer is of the following form,

$$\dot{\hat{x}} = A_d \hat{x} + B_d(u - u_c) \quad (3.41)$$

where $u_c \in R^1$ is an estimation of the equivalent input extended disturbance, n . If a sliding surface, $\sigma \in R^1$

$$\sigma = \lambda(\hat{x} - x) \quad (3.45)$$

with $\lambda = [\lambda_1, \lambda_2] \in R^{1 \times 2}$, $\lambda_i > 0$ is selected and the disturbance observer is designed according to Lyapunov stability theory, the following observer control input, u_c in discrete-time domain ensures sliding

$$u_c(k) = u_c(k-1) + K_u(D_u \sigma(k) + \frac{\sigma(k) - \sigma(k-1)}{T}) \quad (3.46)$$

where T is the sampling time and K_u and D_u are design parameters. More details about the derivation steps of the observer control input are given in the paper of Abidi and Sabanovic (2007).

After estimating the equivalent input extended disturbance in the control input side, the following control input, u is defined to compensate the disturbance and to make the system dynamics follow the reference dynamics,

$$u = u_d + u_c \quad (3.47)$$

then, (3.35) becomes

$$\dot{x} = A_d x + B_d(u_d + u_c + n) \quad (3.48)$$

With the control structure given in (3.46), if $\sigma \rightarrow 0 \Rightarrow x \rightarrow \hat{x} \Rightarrow u_c \rightarrow -n$ then, the system dynamics in (3.48) converges to the reference dynamics given in (3.39).

3.5 Adaptive Optimal Control Allocation Using Lagrangian Neural Networks

Control allocation is one issue to be dealt with when designing controllers for overactuated dynamic systems. An efficient, convergent, optimal and stability guaranteed control allocation algorithm is very crucial in a control system. The algorithm should include system dynamic limitations and actuator limitations. If the control allocation is a nonlinear optimization problem, it is difficult to find an analytical solution and static-iterative nonlinear programming methods are mostly used in the literature. However, the stability of an optimal control allocation method should be guaranteed in a control system and there is no an explicit way to show the stability of a static-iterative optimization method. Moreover, the static-iterative optimization based control allocation requires the nonlinear optimization problem to be solved in every time step for every corresponding virtual control input dictated by the high level controller and this requires a high computational resource in real-time implementation.

As a solution to the above mentioned issues related to the static-iterative optimization based control allocation methods, a novel adaptive optimal control allocation method based on Lagrangian neural-networks is proposed in this study. The proposed control allocation method avoids the requirement of solving the optimization problem explicitly in every time step and the convergence and stability of the control allocation are guaranteed.

Allocation term refers to distribution of a total control effort among available actuators in an optimal manner. Consider an overactuated dynamic system,

$$\dot{x} = A(x) + B_u u \quad (3.49)$$

where $A(x) \in R^n$, $B_u \in R^{n \times m}$ is the input matrix, $x(t) \in R^n$ is the state and $u(t) \in R^m$ is the control input. If $rank(B_u) = k < m$, B_u can be written as

$$B_u = B_\xi B \quad (3.50)$$

where $B_\xi \in R^{n \times k}$ and $B \in R^{k \times m}$. Then, by defining a total control effort which is also called virtual control input as $\xi = Bu$, the system dynamic is achieved as

$$\dot{x} = A(x) + B_\xi \xi \quad (3.51)$$

where $\xi \in R^k$.

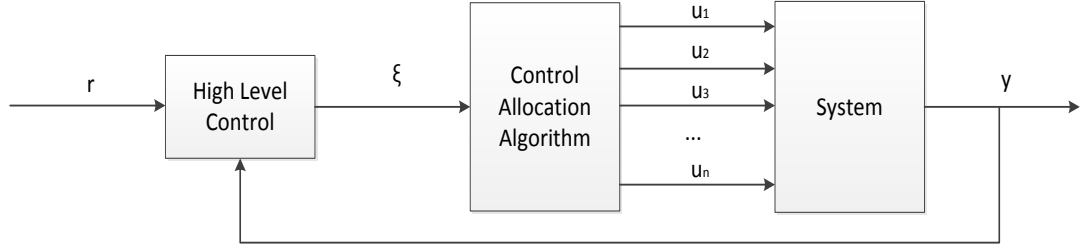


Figure 3.4 : Control diagram for an overactuated system.

A high level controller that is used to track a reference input can be designed by using any suitable nonlinear control method. Once the total control effort is calculated using the high level controller, the remaining tasks are to allocate the actuators of the system and to optimally distribute the total control effort among the available actuators. Thus, the following optimal control allocation problem can be defined.

$$\begin{aligned}
 & \text{minimize} \quad f(u) \\
 & \text{subject to} \quad g(u) = \begin{cases} g_1(u) = u_{i_min} - u_i \\ g_2(u) = u_i - u_{i_max} \\ \dots \\ g_r(u) \end{cases} \leq 0 \\
 & \quad \quad \quad h(u) = \xi - Bu = 0
 \end{aligned} \quad (3.52)$$

If the problem is formulated by well-known Lagrange Multiplier Theorem, the Lagrangian function becomes

$$L(u, \eta, \lambda, s) = f(u) + \lambda^T h(u) + \sum_{j=1}^r \eta_j (g_j(u) + s_j^2) \quad (3.53)$$

where $\lambda = [\lambda_1, \lambda_2, \lambda_3, \dots, \lambda_k]^T \in R^k$ and η_j are the Lagrangian multipliers and s_j is slack variable. A Karush-Khun-Tucker (KKT) point of the optimization problem is defined as a point (u^*, η^*, λ^*) satisfying the conditions below (Zhang and Constantinidis, 1992).

$$\frac{\partial f(u^*)}{\partial u_l} + \sum_{i=1}^k \lambda_i^* \frac{\partial h_i(u^*)}{\partial u_l} + \sum_{j=1}^r \eta_j^* \frac{\partial g_j(u^*)}{\partial u_l} = 0, \quad l = 1, 2, \dots, m \quad (3.54a)$$

$$h_i(u^*) = 0, \quad i = 1, 2, \dots, k \quad (3.54b)$$

$$g_j(u^*) \leq 0, \quad j = 1, 2, \dots, r \quad (3.54c)$$

$$\eta_j^* \cdot g_j(u^*) = 0, \quad j = 1, 2, \dots, r \quad (3.54d)$$

$$\eta_j^* \geq 0, \quad j = 1, 2, \dots, r \quad (3.54e)$$

If the gradients of equality and active inequality constraints are linearly independent and KKT conditions are hold, then u^* is local minimum point of the optimization problem. Since the optimization is nonlinear and it is difficult to find an analytical solution, the conventional approach is to solve the optimization problem by using a static iterative algorithm. If dimension of the problem is high and fast convergence of the algorithm in a limited time step is desired, it requires very high computational resources. As an alternative solution to the static-iterative algorithms, a neural network that will always converge to a KKT point of the optimization problem is proposed by Zhang and Constantinidis (1992). It is shown that the stationary point of the following neural network is a KKT point of the optimization problem and the stationary point will be converged if the stationary point is asymptotically stable and the neural network is globally Lyapunov stable.

$$\frac{du}{dt} = -K_u \frac{\partial L(u, \eta, \lambda, s)}{\partial u} \quad (3.55a)$$

$$\frac{d\lambda}{dt} = K_\lambda \frac{\partial L(u, \eta, \lambda, s)}{\partial \lambda} \quad (3.55b)$$

$$\frac{d\eta}{dt} = K_\eta \frac{\partial L(u, \eta, \lambda, s)}{\partial \eta} \quad (3.55c)$$

$$\frac{ds}{dt} = -K_s \frac{\partial L(u, \eta, \lambda, s)}{\partial s} \quad (3.55d)$$

where $K_u \in \mathbb{R}^{m \times m}$, $K_\lambda \in \mathbb{R}^{k \times k}$, $K_\eta \in \mathbb{R}^{r \times r}$ and $K_s \in \mathbb{R}^{r \times r}$ are diagonal matrices which determine convergence rates. The disadvantage of the slack variable based design given above is that when the search trajectory is inside a feasible region, some satisfied constraints may still pose some effect on the Lagrangian function, leading to possible oscillations and divergence when a local minimum lies on the boundary of the feasible region (Wah and Wang, 1999). To avoid this disadvantage, a new neural network based on the MaxQ method is proposed by Wah and Wang (1999). The

inequality constraints can be converted into equality constraints by using the MaxQ method instead of the slack variable method. The MaxQ method eliminates the need for slack variables and decreases the dimension of the problem. The Lagrangian function of the MaxQ method is

$$L(u, \eta, \lambda) = f(u) + \lambda^T h(u) + \sum_{j=1}^r \eta_j p_j(u) \quad (3.56)$$

where

$$p_j(u) = \max(0, g_j(u)) \quad (3.57)$$

and the corresponding neural network that converges to a KKT point of the optimization problem is

$$\frac{du}{dt} = -K_u \frac{\partial L(u, \eta, \lambda)}{\partial u} \quad (3.58a)$$

$$\frac{d\lambda}{dt} = K_\lambda \frac{\partial L(u, \eta, \lambda)}{\partial \lambda} = K_\lambda h(u) \quad (3.58b)$$

$$\frac{d\eta}{dt} = K_\eta \frac{\partial L(u, \eta, \lambda)}{\partial \eta} = K_\eta p(u) \quad (3.58c)$$

where $K_u \in \mathbb{R}^{m \times m}$, $K_\lambda \in \mathbb{R}^{k \times k}$ and $K_\eta \in \mathbb{R}^{r \times r}$ are diagonal matrices which determine convergence rates.

To show the global stability of the neural network, a Lyapunov function, V is defined as follows.

$$V(u, \lambda, \eta) = \frac{1}{2} \left(\left(\frac{\partial L}{\partial u} \right)^T K_u \left(\frac{\partial L}{\partial u} \right) + \left(\frac{\partial L}{\partial \eta} \right)^T K_\eta \left(\frac{\partial L}{\partial \eta} \right) + \left(\frac{\partial L}{\partial \lambda} \right)^T K_\lambda \left(\frac{\partial L}{\partial \lambda} \right) \right) \quad (3.59)$$

Derivative of the Lyapunov function is

$$\frac{dV}{dt} = \frac{\partial V}{\partial u} \frac{du}{dt} + \frac{\partial V}{\partial \lambda} \frac{d\lambda}{dt} + \frac{\partial V}{\partial \eta} \frac{d\eta}{dt} \quad (3.60)$$

$$\begin{aligned} \frac{dV}{dt} = & \left(\left(\frac{\partial L}{\partial u} \right)^T K_u \left(\frac{\partial^2 L}{\partial u^2} \right) + \left(\frac{\partial L}{\partial \eta} \right)^T K_\eta \left(\frac{\partial^2 L}{\partial \eta^2} \right) + \left(\frac{\partial L}{\partial \lambda} \right)^T K_\lambda \left(\frac{\partial^2 L}{\partial \lambda^2} \right) \right) \frac{du}{dt} \\ & + \left(\left(\frac{\partial L}{\partial u} \right)^T K_u \left(\frac{\partial^2 L}{\partial u \partial \lambda} \right) \right) \frac{d\lambda}{dt} + \left(\left(\frac{\partial L}{\partial u} \right)^T K_u \left(\frac{\partial^2 L}{\partial u \partial \eta} \right) \right) \frac{d\eta}{dt} \end{aligned} \quad (3.61)$$

By substituting the dynamic equations given in (3.58) in to (3.61), one can achieve

$$\frac{dV}{dt} = -\left(\frac{\partial L}{\partial u}\right)^T K_u \left(\frac{\partial^2 L}{\partial u^2}\right) K_u \left(\frac{\partial L}{\partial u}\right) \quad (3.62)$$

If $\left(\frac{\partial^2 L}{\partial u^2}\right) > 0$ then, the derivative of the Lyapunov function is negative,

$$\frac{dV}{dt} < 0 \quad (3.63)$$

Hence $V(u, \lambda, \eta)$, is a Lyapunov function for the system and the neural network is Lyapunov stable.

On the other hand, the local characteristic of the equilibrium is determined by the linearized system. If the first order equations given in (3.58) are linearized at the equilibrium point (u^*, λ^*, η^*) , the linearized system is

$$\begin{bmatrix} \frac{du}{dt} \\ \frac{d\lambda}{dt} \\ \frac{d\eta}{dt} \end{bmatrix} = - \underbrace{\begin{bmatrix} K_u & 0 & 0 \\ 0 & K_\lambda & 0 \\ 0 & 0 & K_\eta \end{bmatrix}}_{\tilde{K}} \underbrace{\begin{bmatrix} \frac{\partial^2 L}{\partial u^2} & \frac{\partial^2 L}{\partial u \partial \lambda} & \frac{\partial^2 L}{\partial u \partial \eta} \\ -\frac{\partial^2 L}{\partial \lambda \partial u} & 0 & 0 \\ -\frac{\partial^2 L}{\partial \eta \partial u} & 0 & 0 \end{bmatrix}}_G \begin{bmatrix} u - u^* \\ \lambda - \lambda^* \\ \eta - \eta^* \end{bmatrix} \quad (3.64)$$

Since $K > 0$, if the real part of the each eigenvalue of G is strictly positive, (u^*, λ^*, η^*) is asymptotically stable point of the system. Assuming $[\hat{x} \ \hat{y} \ \hat{z}]$ is the complex conjugate of a complex vector $[x \ y \ z]$, $Re(\alpha)$ represents the real part of a complex eigenvalue, α of G and the corresponding non-zero eigenvector is defined as

$$\begin{aligned} (x, y, z) &\neq (0, 0, 0) \\ x &\in \mathbb{C}^n \ y \in \mathbb{C}^m \ z \in \mathbb{C}^r \end{aligned} \quad (3.65)$$

we have

$$Re \left\{ [\hat{x}^T \ \hat{y}^T \ \hat{z}^T] G \begin{bmatrix} x \\ y \\ z \end{bmatrix} \right\} = Re(\alpha)(\|x\|^2 + \|y\|^2 + \|z\|^2) \quad (3.66)$$

Substituting the definition of G in (3.64) into the left side of (3.66) yields

$$Re \left\{ \begin{bmatrix} \hat{x}^T & \hat{y}^T & \hat{z}^T \end{bmatrix} G \begin{bmatrix} x \\ y \\ z \end{bmatrix} \right\} = Re \left\{ \hat{x}^T \frac{\partial^2 L}{\partial u^2} x + \hat{x}^T \frac{\partial^2 L}{\partial u \partial \lambda} y + \hat{x}^T \frac{\partial^2 L}{\partial u \partial \eta} z - \hat{y}^T \frac{\partial^2 L}{\partial \lambda \partial u} x - \right. \\ \left. \hat{z}^T \frac{\partial^2 L}{\partial \eta \partial u} x \right\} \quad (3.67)$$

Then,

$$Re(\alpha)(\|x\|^2 + \|y\|^2 + \|z\|^2) = Re \left\{ \hat{x}^T \frac{\partial^2 L}{\partial u^2} x \right\} \quad (3.68)$$

If $\frac{\partial^2 L}{\partial u^2} > 0$, $Re(\alpha) > 0$ and then G is strictly positive. Hence, (u^*, λ^*, η^*) is asymptotically stable point of the neural network.

3.6 Input-Output Decoupling Problem in MIMO Systems and Sliding Mode MIMO Disturbance Observer

Couplings between inputs and outputs of a system are a problem to be solved when the desired dynamics to be tracked are decoupled. A robust input-output decoupling solution should be designed for the system. Input-output decoupling means a situation where each control input only affects one output and none of the other outputs.

A multi input multi output (MIMO) version of the sliding mode disturbance observer (SMDO) for decoupling MIMO nonlinear uncertain systems into single input single output (SISO) systems is proposed in this section. The proposed MIMO SMDO is applicable to m -input m -output square MIMO systems. After decoupling by the MIMO SMDO, it is possible to design independent tracking controllers for each SISO loop by considering only nominal SISO dynamics.

This study is based on the disturbance observer schemes proposed by Talole and Phadke (2008) and Abidi and Sabanovic (2007). Consider an m -input m -output MIMO plant as shown in Figure 3.5 with the following dynamics

$$\begin{aligned} \dot{X} &= (A_n + \Delta A)X + (B_n + \Delta B)U + d \\ Y &= CX \end{aligned} \quad (3.69)$$

where $X \in R^m$ is the state vector, $Y \in R^m$ is the output vector, $U \in R^m$ is the control input, $A_n = \text{diag}\{a_{n1}, a_{n2}, \dots, a_{nm}\}$ and $B_n = \text{diag}\{b_{n1}, b_{n2}, \dots, b_{nm}\}$ are the

nominal state and input matrices respectively, $\Delta A \in R^{m \times m}$ and $\Delta B \in R^{m \times m}$ are the modelling and parameter uncertainties and $d \in R^m$ is the external disturbance.

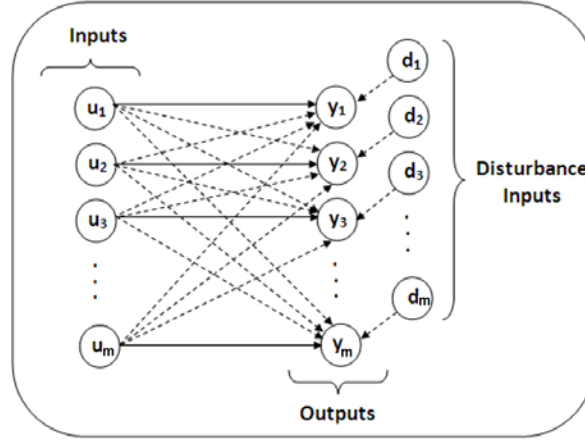


Figure 3.5 : A MIMO plant input-output coupling diagram.

Under the condition that there exists an equivalent input extended-disturbance (EIED), $U_e = [u_{e1}, u_{e2}, \dots, u_{em}]^T \in R^m$ in the control input channel,

$$B_n U_e = \Delta A X + \Delta B U + d \quad (3.70)$$

It is shown by She et al. (2008) that if (A_n, B_n, C) is observable and controllable and if (A_n, B_n, C) has no zeros on the imaginary axis, then there always exist an EIED in the control input channel. Then, the MIMO system dynamics given in (3.69) can be written as follows.

$$\dot{X} = A_n X + B_n (U + U_e) \quad (3.71)$$

The aim in the sliding mode MIMO disturbance observer is to design an observer which estimates the equivalent input extended disturbance in the control input side, U_e . To design the MIMO disturbance observer based on sliding mode control theory, the following sliding surface, σ is selected

$$\sigma = K(X - Z) \quad (3.72)$$

with

$$\begin{aligned} \dot{Z} &= A_n X + B_n (U + U_c) \\ Z(0) &= X(0) \end{aligned} \quad (3.73)$$

where $K = \text{diag}\{k_1, k_2, \dots, k_m\}$, $k_i > 0$, $U_c = [u_{c1}, u_{c2}, \dots, u_{cm}]^T \in R^m$ is the observer control input, $Z \in R^m$ and $\sigma = [\sigma_1, \sigma_2, \dots, \sigma_m]^T \in R^m$. If the observer is derived by using Lyapunov stability theory with the following Lyapunov function, V

$$V = \frac{\sigma^T \sigma}{2} \quad (3.74)$$

then, similar to the design steps given in the study of Abidi and Sabanovic (2007), by deriving sliding mode disturbance observer for the MIMO case, one can get the following control law of the MIMO disturbance observer in discrete time domain.

$$U_c(k) = U_c(k-1) + \frac{H}{T} \{(T\Gamma + I)\sigma(k) - \sigma(k-1)\} \quad (3.75)$$

where T is sampling time, $I_{m \times m}$ is the identity matrix, Γ and H are the design parameters.

After the estimation of the extended disturbance in the control input side, by defining the following control input to cancel the disturbance and to make the system dynamics track the nominal plant,

$$U = U_n - U_c \quad (3.76)$$

the system dynamics given in (3.71) becomes

$$\dot{X} = A_n X + B_n (U_n - U_c + U_e) \quad (3.77)$$

The MIMO SMDO architecture given above is actually composed of multiple SISO SMDO loops which can be designed separately as if they were separate SISO SMDO loops. Each SISO SMDO loops decouple the desired input-output dynamics by treating the effect of the couplings as a disturbance in addition to the modelling and parameter uncertainties and external disturbances. After decoupling by the MIMO SMDO, it is possible to design independent tracking controllers for each loop by considering only nominal SISO dynamics since each SISO SMDO loops make the related input-output dynamics track the nominal dynamics. It should be noted here that the proposed MIMO SMDO is applicable to m -input m -output square MIMO uncertain systems where decoupled SISO loops are desired and the matching condition is satisfied.

4. DESIGN OF VEHICLE DYNAMICS CONTROL SYSTEMS

4.1 Vehicle Steering Control

This study proposes two vehicle steering controllers based on the disturbance observer approaches, i.e. sliding mode disturbance observer and model reference sliding mode control with extended disturbance estimation. The aim in the vehicle steering controllers is to force the vehicle to track a 2DOF reference vehicle dynamics behavior by estimating the equivalent input extended-disturbance which is the front tire steering correction angle for active steering system. The sliding mode based disturbance observer approaches developed in this thesis are the first known applications to vehicle steering control.

By considering the vehicle dynamics in (3.15) and applying the sliding mode disturbance observer (SMDO) approach given in Chapter 3, the SMDO for vehicle steering control is of the following form.

$$\dot{\hat{x}} = A_d \hat{x} + B_{\delta d}(\delta_f - \delta_c) \quad (4.1)$$

where δ_c is the steering correction angle for the active front steering system. Then, the control law given in (3.46) becomes

$$\delta_c(k) = \delta_c(k-1) + K_\delta(D_\delta \sigma(k) + \frac{\sigma(k) - \sigma(k-1)}{T}) \quad (4.2)$$

where K_δ and D_δ are the design parameters, T is the sampling time and σ is the sliding surface function defined in (3.45). The steering control input is

$$\delta_f = \delta_d + \delta_c \quad (4.3)$$

where δ_d is the driver steering input and then, the resulting system with disturbance feedback is

$$\dot{x} = A_d x + B_{\delta d}(\delta_d + \delta_c + n) \quad (4.4)$$

The stability of the control law given in (4.2) is proved by using Lyapunov stability theory and the sliding surface, σ asymptotically converges to zero. It can be derived that if

$$\sigma \rightarrow 0 \Rightarrow x \rightarrow \hat{x} \Rightarrow \delta_c \rightarrow -n \quad (4.5)$$

then, the system dynamics in (3.15) converges to

$$\dot{x} = A_d x + B_{\delta d} \delta_d \quad (4.6)$$

which yields that the reference vehicle dynamics are tracked successfully. Similar to the structure in the work of Abidi and Sabanovic (2007), the structure of the sliding mode disturbance observer based vehicle steering control is given in Figure 4.1.

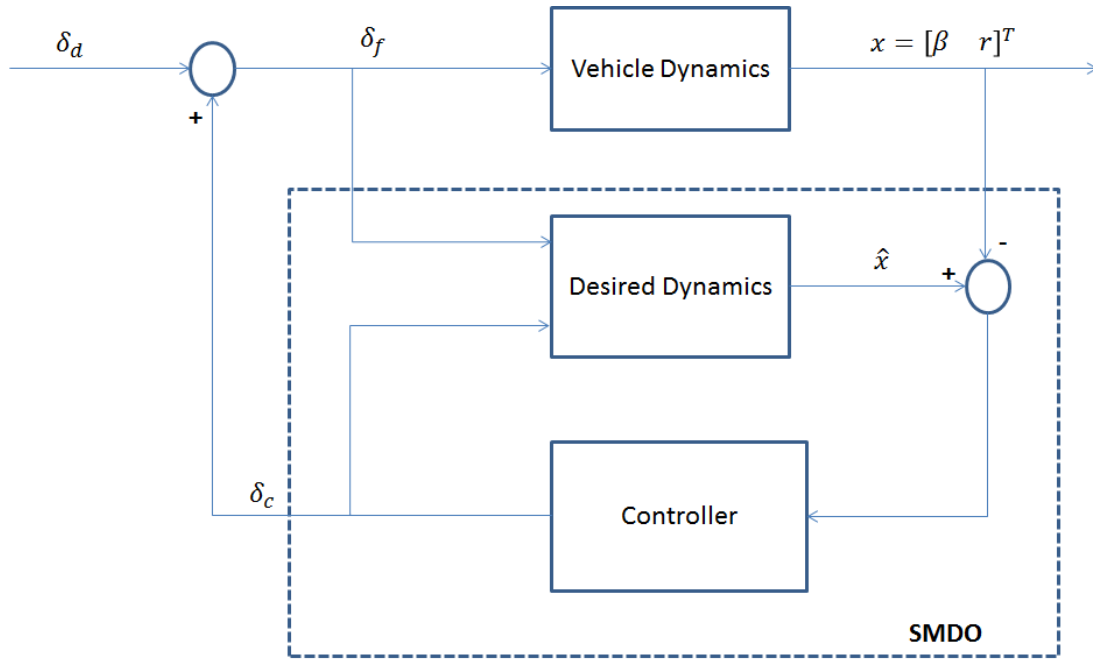


Figure 4.1 : The SMDO based vehicle steering control.

On the other hand, the model reference sliding mode control with extended disturbance estimation (SMEDE) method is also applied to the vehicle steering control problem. By considering the vehicle dynamics in (3.15) and using the control approach that is derived in the Chapter 3, the vehicle steering controller is

$$\delta_f = \delta_d + \delta_\sigma + \delta_n \quad (4.7)$$

with δ_d is the driver steering input, $\delta_\sigma = -k_\sigma \sigma$, $k_\sigma > 0$ where σ is defined in (3.36) and δ_n in s-domain is

$$\delta_n(s) = \frac{-1}{\tau\varphi B_{\delta d}} \left(\sigma(s) + \frac{k_\sigma \varphi B_{\delta d}}{s} \sigma(s) \right) \quad (4.8)$$

where the filter time constant, τ and $\varphi \in R^2$ are a design parameter. Then, by substituting (4.7) in to (3.15) and (3.36), the resulting system and sliding surface dynamics with $\delta_n = -\hat{n}$ are

$$\dot{x} = A_d x + B_{\delta d} (\delta_d - k_\sigma \sigma + \tilde{n}) \quad (4.9)$$

$$\dot{\sigma} = \varphi B_{\delta d} (\tilde{n} - k_\sigma \sigma) \quad (4.10)$$

where $\tilde{n} = n - \hat{n}$. Here, $\hat{n}$ is the estimated value of n . From Talole and Phadke (2008), the estimation of the equivalent-input extended disturbance is

$$\hat{n} = F(s)n \quad (4.11)$$

where $F(s)$ is a first order filter with unity dc gain and time constant, τ . Then, one can obtain

$$\dot{\tilde{n}} = -\frac{1}{\tau} \tilde{n} + \dot{n} \quad (4.12)$$

Combining the dynamics in (4.10) and (4.12) gives the following error dynamics

$$\begin{bmatrix} \dot{\tilde{n}} \\ \dot{\sigma} \end{bmatrix} = \begin{bmatrix} -\frac{1}{\tau} & 0 \\ \varphi B_{\delta d} & -k_\sigma \varphi B_{\delta d} \end{bmatrix} \begin{bmatrix} \tilde{n} \\ \sigma \end{bmatrix} + \begin{bmatrix} 1 \\ 0 \end{bmatrix} \dot{n} \quad (4.13)$$

It can be seen that the system matrix of the error dynamics in (4.13) is stable implying the stability for the error dynamics. As the error dynamics are driven by $\dot{n}$, it is obvious that for bounded $|\dot{n}|$, bounded input bounded output (BIBO) stability is assured for the error dynamics. Moreover, if the rate of change of the extended disturbance is negligible, i.e. $\dot{n} \approx 0$ then, the error dynamics are asymptotically stable. However, it is also shown in Talole and Phadke (2009) that asymptotically stability for the error dynamics can always be assured if some higher derivative of the extended disturbance is equal to zero. For example, if $\dot{n} \neq 0$ but some higher derivative of n is zero then, the asymptotically stability for the error dynamics can be guaranteed by choosing an appropriate higher-order filter.

It is assumed here that the states of the vehicles are available. Yaw rate can be measured by using a low-cost sensor but, direct measurement of sideslip angle and road friction coefficient requires high-cost sensors that are practically not applicable in the vehicle. Robust estimation of sideslip angle and road friction coefficient can be used. Various estimation algorithms are proposed in the literature to estimate sideslip

angle and road friction coefficient, e.g. Han and Huh (2011), Anderson and Bevly (2010), Cheli et al. (2007), Stephant et al. (2007) and Piyabongkarn et al. (2009).

The performance of the proposed vehicle steering control approaches were tested on a Fiat Doblo Van modelled in CarSim environment and the simulation experiments are given in the next chapter. The performance of the proposed vehicle steering control systems is also compared with the performance of the inverse model based disturbance observer approach that is a well-known disturbance observer method for motion control.

4.2 Lateral and Yaw Motion Decoupling Control in 4WAS Vehicles

The MIMO SMDO constructed in the previous chapter is applied to the decoupled control of four wheel active steering (4WAS) dynamics which are composed of two input (rear wheel steering angle and front wheel steering angle) and two outputs (vehicle side slip angle and yaw rate). If the virtual yaw moment input is considered as a yaw moment disturbance input, the linear 2DOF vehicle dynamics that are derived in the Chapter 2 are

$$\underbrace{\begin{bmatrix} \dot{r} \\ \dot{\beta} \end{bmatrix}}_{\dot{X}} = \underbrace{\begin{bmatrix} a_{22} & a_{21} \\ a_{12} & a_{11} \end{bmatrix}}_A \underbrace{\begin{bmatrix} r \\ \beta \end{bmatrix}}_X + \underbrace{\begin{bmatrix} b_{21} & b_{22} \\ b_{11} & b_{12} \end{bmatrix}}_B \underbrace{\begin{bmatrix} \delta_f \\ \delta_r \end{bmatrix}}_{\delta} + \underbrace{\begin{bmatrix} \rho_1 \\ \rho_2 \end{bmatrix}}_{\rho} \quad (4.14)$$

where ρ includes unmodeled nonlinear dynamics, parameter uncertainties and external disturbances. It is desired to achieve two independent input-output dynamics which are the dynamics from front wheel steering input (δ_f) to yaw rate output (r) and from rear wheel steering input (δ_r) to vehicle side slip angle (β). Moreover, it is desired to cancel the affects of rear wheel steering angle on yaw rate and front wheel steering angle on vehicle side slip angle. The desired nominal dynamics are (AGuvenc et al. 2010)

$$\underbrace{\begin{bmatrix} \dot{r} \\ \dot{\beta} \end{bmatrix}}_{\dot{X}} = \underbrace{\begin{bmatrix} a_{n1} & 0 \\ 0 & a_{n2} \end{bmatrix}}_{A_n} \underbrace{\begin{bmatrix} r \\ \beta \end{bmatrix}}_X + \underbrace{\begin{bmatrix} b_{n1} & 0 \\ 0 & b_{n2} \end{bmatrix}}_{B_n} \underbrace{\begin{bmatrix} \delta_f \\ \delta_r \end{bmatrix}}_{\delta} \quad (4.15)$$

where

$$\begin{aligned}
a_{n1} &= -\frac{1}{\tau_{n1}} & a_{n2} &= -\frac{1}{\tau_{n2}} \\
b_{n1} &= \frac{K_r}{\tau_{n1}} & b_{n2} &= \frac{K_b}{\tau_{n2}} \\
K_r &= \frac{a_{12}b_{22} - b_{12}a_{22}}{a_{11}a_{22} - a_{12}a_{21}} & K_b &= \frac{a_{21}b_{11} - b_{21}a_{11}}{a_{11}a_{22} - a_{12}a_{21}}
\end{aligned}$$

The four wheel steering vehicle dynamics given in (4.14) can be written in terms of nominal vehicle dynamics by reminding the equivalent input extended disturbance concept given in the Chapter 3 as follows.

$$\dot{X} = A_n X + B_n(\delta + \delta_e) \quad (4.16)$$

where δ_e is the equivalent input extended disturbance in the control input channel such that $B_n \delta_e = (A - A_n)X + (B - B_n)\delta + \rho$. With a steering control input defined as $\delta = \delta_n - \delta_c$, (4.16) becomes

$$\dot{X} = A_n X + B_n(\delta_n - \delta_c + \delta_e) \quad (4.17)$$

where δ_c is the observed value of δ_e . Then, by adapting the sliding mode MIMO disturbance observer given in the Chapter 3, the control law in discrete-time domain yields

$$\delta_c(k) = \delta_c(k-1) + \frac{H}{T} \{(T\Gamma + I)\sigma(k) - \sigma(k-1)\} \quad (4.18)$$

where T is the sampling time, $I \in \mathbb{R}^{2 \times 2}$ is the identity matrix, $H \in \mathbb{R}^{2 \times 2} > 0$ and $\Gamma \in \mathbb{R}^{2 \times 2} > 0$ are design parameters with diagonal matrices.

The positive design parameters, Γ and H are tuned manually here but, it can also be adjusted adaptively by using an adaptive tuning algorithm which aims to minimize a cost function of sliding surface and the parameter estimation errors. The performance of the proposed MIMO SMDO is compared to the results of the MIMO IMDO proposed by A Guvenc et al. (2010). Moreover, the MIMO SMDO is also tested on a Fiat Doblo Van equipped with four wheel active steering actuators and the simulation experiments are given in the next chapter. The Fiat Doblo Van is modelled in CarSim environment with a high degrees-of-freedom nonlinear vehicle dynamics.

4.3 Vehicle Stability Control for 4WS4WD Electric Vehicles

This study involves a layered vehicle dynamics control system which is composed of an adaptive optimal control allocation method using Lagrangian neural networks for optimal distribution of tire forces and the sliding mode yaw moment observer for robust control of yaw dynamics (Figure 3.2). The proposed optimal control allocation method avoids the requirement of solving the optimization problem explicitly in every time step for every corresponding virtual yaw moment input dictated by the upper layer. The aim in the sliding mode yaw moment observer is to force the vehicle to track a reference vehicle dynamics behavior by estimating the equivalent input extended-disturbance which is the required stabilizing virtual yaw moment. The proposed layered stability control scheme has been tested on a four wheel drive four wheel steer electric vehicle which is modeled in CarSim. Both of the sliding mode disturbance observer and the optimal control allocation method are the first known applications to the stability control problem of road vehicles.

The yaw moment control problem was given in the previous chapter. By considering the vehicle dynamics given in (3.21) and adapting the sliding mode disturbance observer approach derived in in the Chapter 3, the disturbance observer is of the following form

$$\dot{\hat{x}} = A_d \hat{x} + B_{md}(M_z - M_c) \quad (4.19)$$

where M_c is the observer control input. Then, the observer control input in discrete-time domain becomes

$$M_c(k) = M_c(k-1) + K_m(D_m \sigma(k) + \frac{\sigma(k) - \sigma(k-1)}{T}) \quad (4.20)$$

where T is the sampling time, K_m and D_m are the design parameters. The virtual yaw moment control input is

$$M_z = M_d + M_c \quad (4.21)$$

where M_d is the commanded virtual yaw moment input calculated by using driver steering input and then, the resulting system with disturbance feedback is

$$\dot{x} = A_d x + B_{md}(M_d + M_c + M_{ze}) \quad (4.22)$$

The stability of the control law given in (4.20) is proved by using Lyapunov stability theory and the sliding surface, σ asymptotically converges to zero. It can be derived that if

$$\sigma \rightarrow 0 \Rightarrow x \rightarrow \hat{x} \Rightarrow M_c \rightarrow -M_{ze} \quad (4.23)$$

then, the system dynamics in (3.21) converges to

$$\dot{x} = A_d x + B_{md} M_d \quad (4.24)$$

which yields that the reference vehicle dynamics are tracked successfully.

Once the required yaw moment for stabilization of the vehicle is calculated using the sliding mode yaw moment observer, the remaining tasks are to optimally distribute the tire forces which generate the required yaw moment and to allocate the actuators. The optimal tire force distribution problem for 4WS4WD electric vehicles was given in (3.27). By redefining the inequality constraints given in (3.27) as follows.

$$g(u) = \left\{ \begin{array}{l} (F_{xi} + \Delta F_{xi})^2 + (F_{yi} + \Delta F_{yi})^2 - (\mu_i F_{zi})^2 \leq 0 \\ -T_{mi_max} - T_{mi} - \Delta F_{xi} R \leq 0 \\ T_{mi} + \Delta F_{xi} R - T_{mi_max} \leq 0 \\ -\delta_{i_max} - \delta_i - \frac{\Delta F_{yi}}{C_i} \leq 0 \\ \delta_i + \frac{\Delta F_{yi}}{C_i} - \delta_{i_max} \leq 0 \end{array} \right\}, i = 1, 2, 3, 4 \quad (4.25)$$

The adaptive optimal control allocation method based on Lagrangian Neural Networks which is proposed in the previous chapter is applied to solve the above nonlinear tire force distribution problem. It was shown that if a Lagrangian function of the MaxQ method is defined as follows

$$L(u, \eta, \lambda) = f(u) + h(u) \cdot \lambda + \sum_{j=1}^{20} \eta_j p_j(u) \quad (4.26)$$

with

$$p_j(u) = \max(0, g_j(u)) \quad (4.27)$$

where $f(u)$ and $h(u)$ are defined in (3.27), λ and η_j are the Lagrange multipliers, the stationary point, (u^*, η^*, λ^*) of the following adaptive dynamic system is a KKT point of the optimization problem and the stationary point will be converged if the stationary point is asymptotically stable and the adaptive dynamic system is globally Lyapunov stable.

$$\frac{du}{dt} = -K_u \frac{\partial L(u, \eta, \lambda)}{\partial u} \quad (4.28a)$$

$$\frac{d\lambda}{dt} = K_\lambda \frac{\partial L(u, \eta, \lambda)}{\partial \lambda} = K_\lambda h(u) \quad (4.28b)$$

$$\frac{d\eta}{dt} = K_\eta \frac{\partial L(u, \eta, \lambda)}{\partial \eta} = K_\eta p(u) \quad (4.28c)$$

where $K_u \in \mathbb{R}^{8 \times 8}$, $K_\lambda \in \mathbb{R}^1$ and $K_\eta \in \mathbb{R}^{20 \times 20}$ are diagonal matrices which determine convergence rates. Then, the optimal tire forces can be determined by the following adaptive dynamic system.

$$\frac{d\Delta F_{xi}}{dt} = k_{f_{xi}} \left(2\Delta F_{xi}\gamma_i + \frac{2}{(\mu_i F_{zi})^2} \Delta F_{xi}^3 - \lambda l_{xi} + 2\eta_i (F_{xi} + \Delta F_{xi}) + R(\eta_{i+4} - \eta_{i+8}) \right) \quad (4.29a)$$

$$\frac{d\Delta F_{yi}}{dt} = k_{f_{yi}} \left(2\Delta F_{yi}\rho_i + \frac{2}{(\mu_i F_{zi})^2} \Delta F_{yi}^3 + \lambda l_{yi} + 2\eta_i (F_{yi} + \Delta F_{yi}) + C_i^{-1}(\eta_{i+16} - \eta_{i+12}) \right) \quad (4.29b)$$

$$\frac{d\eta_j}{dt} = k_{\eta_j} \max(0, g_j(u)) \quad (4.29c)$$

$$\frac{d\lambda}{dt} = k_\lambda (\sum_{i=1}^4 (\Delta F_{yi} l_{yi} - \Delta F_{xi} l_{xi}) - M_c) \quad (4.29d)$$

where $k_{f_{xi}}$, $k_{f_{yi}}$, k_{η_j} and k_λ are the convergence gains to be selected, $i = 1, 2, 3, 4$ and $j = 1, 2, 3, \dots, 20$. Since

$$\frac{\partial^2 L}{\partial \Delta F_{xi}^2} > 0, \frac{\partial^2 L}{\partial \Delta F_{yi}^2} > 0, \quad i = 1, 2, 3, 4 \quad (4.30)$$

the proposed adaptive dynamic system in (4.29) is Lyapunov stable and (u^*, λ^*, η^*) is asymptotically stable point of the adaptive dynamic system. Existence of the conditions in (4.30) are shown in the Appendix C.

Once the optimal tire forces $(\Delta F_{xi}, \Delta F_{yi})$ are determined, the electric motor and steering actuator commands are calculated as

$$T_{mi} = \frac{T_{md}}{4} + R \cdot \Delta F_{xi} \quad (4.31)$$

$$\delta_{fc} = \eta \cdot \left(\frac{\Delta F_{y1} + \Delta F_{y2}}{2 \cdot C_{fn}} \right) \quad (4.32)$$

$$\delta_{rc} = \eta \cdot \left(\frac{\Delta F_{y3} + \Delta F_{y4}}{2 \cdot C_{rn}} \right) \quad (4.33)$$

where T_{md} is drive/brake input of the driver, η is steering ratio, R is tire radius, C_{fn} and C_{rn} are nominal front and rear cornering stiffnesses respectively. The proposed algorithms are applied to the stability control of a Fiat Doblo Van equipped with four wheel steering actuators and four electric motors mounted in wheel. The controllers are modeled in Matlab/Simulink environment and a co-simulation is performed with CarSim in which a high degree-of-freedom Doblo Van model is run. The simulation experiments and the results are given in the next chapter. Moreover, real-time experiments is also carried out on a dSPACE MicroAutoBox control hardware with CarSim RT and the results are given in the Chapter 6.

4.4 Vehicle Stability Control by Coordination of AFS and Individual Brakes

A robust integrated vehicle dynamics control by coordination of active front steering and individual braking actuators is proposed in this section. The coordination of the actuators is handled by the adaptive optimal control allocation algorithm given in the Chapter 3. Moreover, a sliding mode yaw moment controller based on the model reference sliding mode control with extended disturbance estimation was developed. The yaw moment controller is used to force a vehicle to track a reference vehicle dynamics behavior. The output of the yaw moment controller is a virtual yaw moment control input which will be generated by the available actuators and is a dictated target constraint for the control allocation problem. On the other hand, it is also shown in this section that the proposed adaptive optimal control allocation method may compensate partial actuator failures and may assure an active fault tolerant vehicle dynamics control system.

The integrated vehicle stability control problem with hierarchical control structure was given in Chapter 3 (Figure 3.3). Model following sliding mode control approach proposed by Talole and Phadke (2008) is adapted for yaw moment control problem. By considering the vehicle dynamics given in (3.21) and using the control approach that is derived in the Chapter 3, the virtual yaw moment control input, M_z is defined as

$$M_z = M_d + M_\sigma + M_n \quad (4.34)$$

with

$$M_\sigma = -k_\sigma \sigma, \quad k_\sigma > 0 \quad (4.35)$$

where the sliding surface, σ is defined in (3.36), M_d is the commanded virtual yaw moment input calculated by using driver steering input and the disturbance cancelation input, M_n is

$$\widehat{M}_n(s) = \frac{-1}{\tau\varphi B_{md}} \left(\sigma(s) + \frac{k_\sigma \varphi B_{md}}{s} \sigma(s) \right) \quad (4.36)$$

Where the filter time constant, τ and $\varphi \in R^2$ are a design parameter. The comments and derivations about the error dynamics and stability proof of the proposed yaw moment controller are similar to the case of the vehicle steering controller and hence the proposed control structure is BIBO stable.

Similar to the optimal tire force distribution problem of 4WS4WD electric vehicles given in the previous section, the nonlinear optimal tire force distribution problem was formulated in (3.32) for distributing the required yaw moment, M_z between the available actuators.

By defining the following Lagrangian function of the MaxQ method

$$L(u, \eta, \lambda) = f(u) + h(u) \cdot \lambda + \sum_{k=1}^{16} \eta_k p_k(u) \quad (4.37)$$

where $p_k(u) = \max(0, g_k(u))$ and by applying the adaptive optimal control allocation method proposed in the Chapter 3, the optimal tire forces can be determined by the following adaptive dynamic system.

$$\frac{d\Delta F_{xi}}{dt} = \Gamma_{f_{xi}} \left(2\Delta F_{xi} \gamma_i + \frac{2}{(\mu_i F_{zi})^2} \Delta F_{xi}^3 - \lambda l_{xi} + 2\eta_i (F_{xi} + \Delta F_{xi}) + \eta_{i+12} - R\eta_{i+4} \right) \quad (4.38a)$$

$$\frac{d\Delta F_{yj}}{dt} = \Gamma_{f_{yj}} \left(2\Delta F_{yj} \rho_j + \frac{2}{(\mu_j F_{zj})^2} \Delta F_{yj}^3 + \lambda l_{yj} + 2\eta_j (F_{yj} + \Delta F_{yj}) + C_j^{-1} (\eta_{j+12} - \eta_{j+8}) \right) \quad (4.38b)$$

$$\frac{d\eta_k}{dt} = \Gamma_{\eta_k} \max(0, g_k(u)) \quad (4.38c)$$

$$\frac{d\lambda}{dt} = \Gamma_\lambda \left(\sum_{j=1}^2 \Delta F_{yj} l_{yj} - \sum_{i=1}^4 \Delta F_{xi} l_{xi} - M_z \right) \quad (4.38d)$$

where $\Gamma_{f_{xi}}$, $\Gamma_{f_{yj}}$, Γ_{η_k} and Γ_λ are the convergence gains to be designed, $i = 1, 2, 3, 4$, $j = 1, 2$ and $k = 1, 2, \dots, 16$. Then, the individual braking actuator torque commands and the steering actuator angle command are calculated as

$$T_{bci} = R \cdot \Delta F_{xi} \quad (4.39)$$

$$\delta_{fc} = \eta \left(\frac{\Delta F_{y1} + \Delta F_{y2}}{2C_{fn}} \right) \quad (4.40)$$

where η is steering ratio, R is tire radius and C_{fn} are nominal front tire cornering stiffness. To prevent a slip on a braked wheel, a wheel slip controller which simply provides ratio, ϕ_b for braking torque is used as follows.

$$T_{bi}^* = \phi_b T_{bi} \quad (4.41)$$

where ϕ_b is a control gain given below. It is given in the Chapter 2 that the tire slip, s_i is

$$s_i = \frac{\omega_i R - V_{xi}}{\omega_i R} \quad (4.42)$$

By adapting the sliding mode controller proposed by Abidi and Sabanovic (2007) and defining the following sliding surface

$$\sigma_i = s_{ith} - |s_i| \quad (4.43)$$

where s_{ith} is a tire slip threshold then, the wheel slip controller in discrete-time is

$$\phi_b(k) = \begin{cases} 1 & , s_i \leq s_{ith} \\ \phi_b(k-1) + K_\phi (D_\phi \sigma_i(k) + \frac{\sigma_i(k) - \sigma_i(k-1)}{T}), & s_i > s_{ith} \end{cases} \quad (4.44)$$

where K_ϕ and D_ϕ are design parameters and T is sample time.

5. SIMULATION RESULTS

The vehicle dynamics control algorithms developed in the previous chapters were tested on a Fiat Doblo Van modeled in CarSim environment. The CarSim model uses a high degree-of freedom nonlinear vehicle dynamics with nonlinear tire models which also include highly nonlinear dynamics. Since the dynamic models of CarSim are realistic models which truly represent the vehicle dynamics, the results are a valuable indicate of the performances of the proposed control algorithms.

To make simulation experiments on the vehicle for different powertrain and steering configurations, four vehicle models are developped on CarSim. One model is front wheel active steering model of the vehicle for testing the proposed sliding mode disturbance observer based vehicle steering control approaches and integrated stability control approach by coordination of AFS and individual brakes, the other model is four wheel steering model of the vehicle for testing the proposed lateral and yaw motion decoupling controller and the last model is four wheel steering four wheel drive electric vehicle model for testing the proposed integrated stability control algorithms, i.e. the sliding mode based yaw moment controllers and the adaptive optimal control allocation method, for 4WS4WD electric vehicles. The simulation experiments are performed by running CarSim software and Matlab/Simulink simultaneously since the proposed vehicle dynamics control approaches are modelled in Matlab/Simulink environment while the vehicle in CarSim.

5.1 Results For Vehicle Steering Control

The simulation experiments of the proposed sliding mode disturbance observer based vehicle steering control approaches in driving maneuvers and cross-wind disturbance tests are shown in this section. To show the robustness of the controllers, the test simulations are repeated for different road conditions and different vehicle speeds. The simulation environment is illustrated in Figure 5.1. The Doblo model in CarSim

exports the states of the vehicle like vehicle side slip angle, yaw rate, vehicle speed, etc. while the controllers in Matlab/Simulink environment send the front wheel steering correction angle input to CarSim.

The proposed disturbance observer approaches designed using sliding mode control theory, i.e. sliding mode disturbance observer (SMDO) and sliding mode model reference control with extended-disturbance estimation (SMEDE) is compared with the inverse model based disturbance observer approach that is a well-known disturbance observer in motion control. The inverse model based disturbance observer (IMDO) approach for vehicle steering control problem was studied before by Guvenc et al. (2004) and A. Guvenc et al. (2009).

The performances of the proposed approaches are tested in two different scenarios which are double lane change (DLC) maneuver and cross-wind disturbance test. The DLC maneuver are repeated for two operating conditions which are

- *Case 1: Driving on a dry road surface ($\mu=0.85$) with 140km/h initial vehicle speed*
- *Case 2: Driving on an wet road surface ($\mu=0.5$) with 100km/h initial vehicle speed*

To make a numerical comparison between the performance results of the control approaches, the defined metrics are yaw rate maximum tracking error and root mean square error, E_{r_max} and E_{r_rms} respectively defined as follows.

$$E_{r_max} = \max(|r(t) - r_d(t)|)$$

$$E_{r_rms} = \sqrt{\frac{1}{t_f - t_0} \int_{t_0}^{t_f} (r(t) - r_d(t))^2 dt} \quad (5.1)$$

where t_0 and t_f are the starting and the final test times respectively.

In the implementation of IMDO and SMEDE, the same low-pass filter is chosen as follows.

$$Q(s) = \frac{1}{0.08s + 1} \quad (5.2)$$

5.1.1 Double lane change maneuver

In the double lane change maneuver (DLC), a driver steering input as shown in Figure 5.2 is applied to the vehicle with an initial speed, no braking and no throttle input. The aim is to evaluate the performances of the proposed controllers in extreme driving maneuvers in which the vehicle dynamics is highly nonlinear and there are highly nonlinear modeling and parameter uncertainties. The DLC maneuver is repeated for two operating conditions, i.e. Case 1 and Case 2.

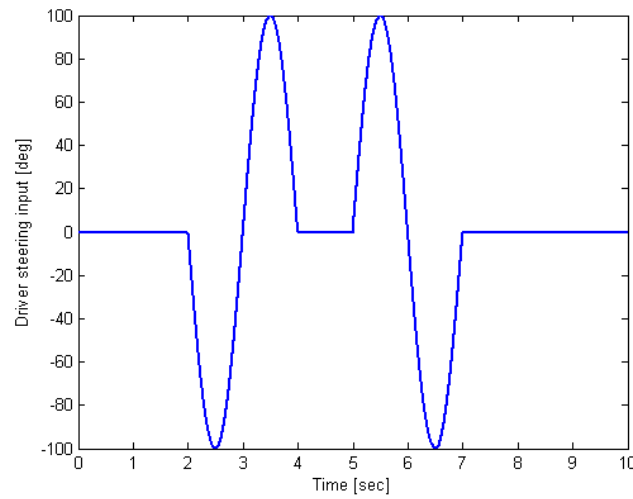


Figure 5.2 : Driver steering input in DLC maneuver.

Figure 5.3 and Figure 5.4 show the obtained results for Case 1 and Case 2 respectively. Figure 5.5 shows the displacements of the controlled and uncontrolled vehicles for Case 1 and Case 2. The metrics which shows the tracking performances obtained in the double lane change maneuvers are reported in Table 5.1. By examining the figures, it can be seen that the vehicle equipped with the proposed controllers makes DLC maneuvers successfully for all cases while the uncontrolled vehicle cannot and the vehicle drifts out. The performances of the controllers are very close to each other. All of them give practically the same results but, it can be seen from Table 5.1 that better results are achieved for the approaches designed by using sliding mode control theory, i.e. SMDO and SMEDE. The proposed vehicle steering controllers are robust against modeling and parameter uncertainties and they are very successful to improve handling and stability of the vehicle even in extreme driving maneuvers.

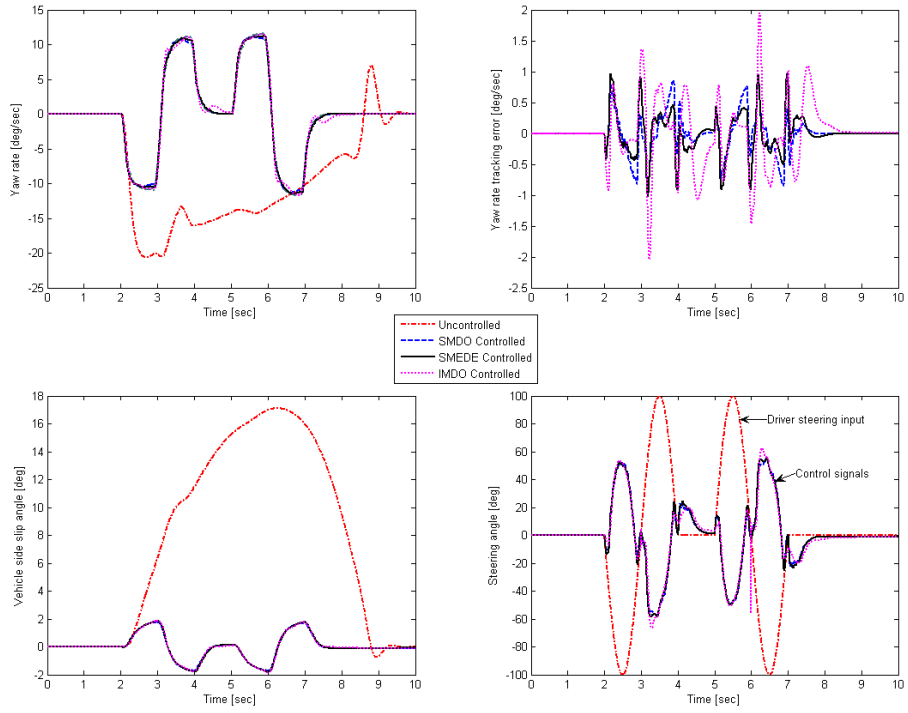


Figure 5.3 : The obtained results in DLC maneuver for Case 1.

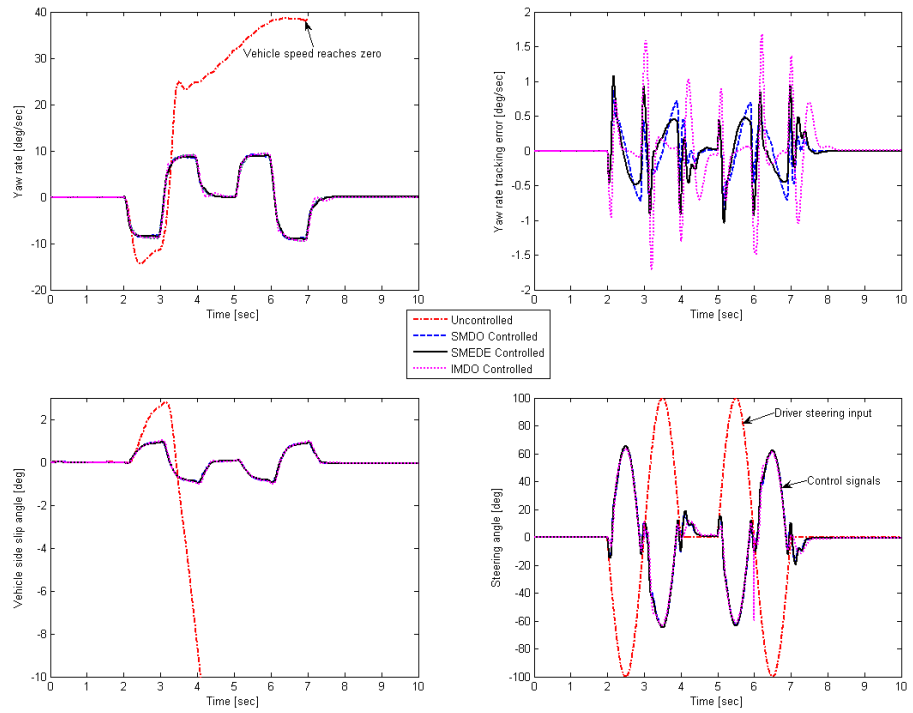


Figure 5.4 : The obtained results in DLC maneuver for Case 2.

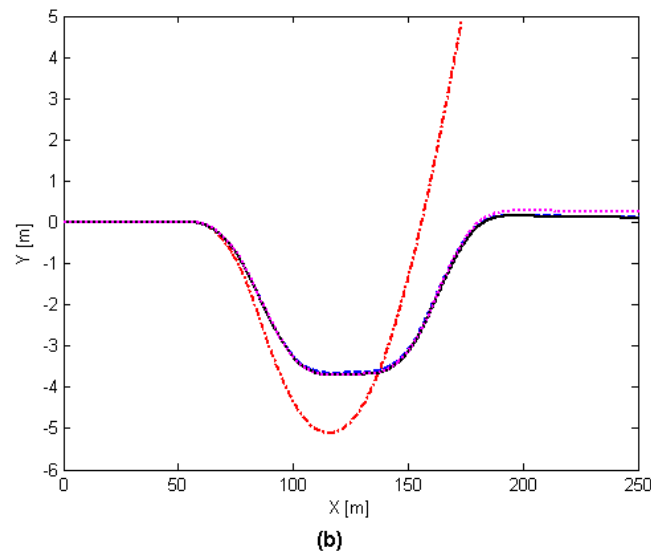
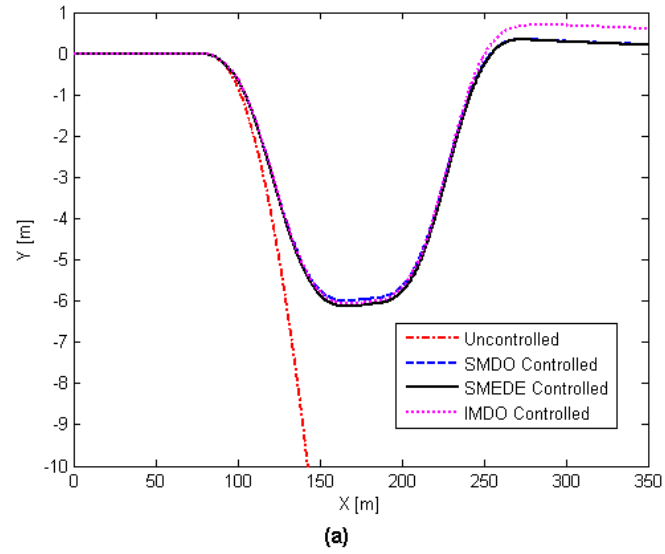


Figure 5.5 : The displacements of the controlled and uncontrolled vehicle in DLC maneuver (a) the results for Case 1 (b) the results for Case 2.

Table 5.1 : Yaw rate rms & max. tracking errors in DLC maneuvers.

Metric	Cases	SMDO	SMEDE	IMDO
E_{r_rms}	Case 1	0.2847	0.2865	0.5108
	Case 2	0.2768	0.2949	0.4411
E_{r_max}	Case 1	0.9893	1.015	2.055
	Case 2	0.8898	1.084	1.724

5.1.2 Cross-wind disturbance test

In the cross-wind disturbance (CWD) test, a test procedure which is given in CarSim is used. In this test procedure, the vehicle enters in to a test track shown in Figure 5.6 in which there exist fan blowers at each side and specific positions of the icy-road to create a 100km/h side wind.

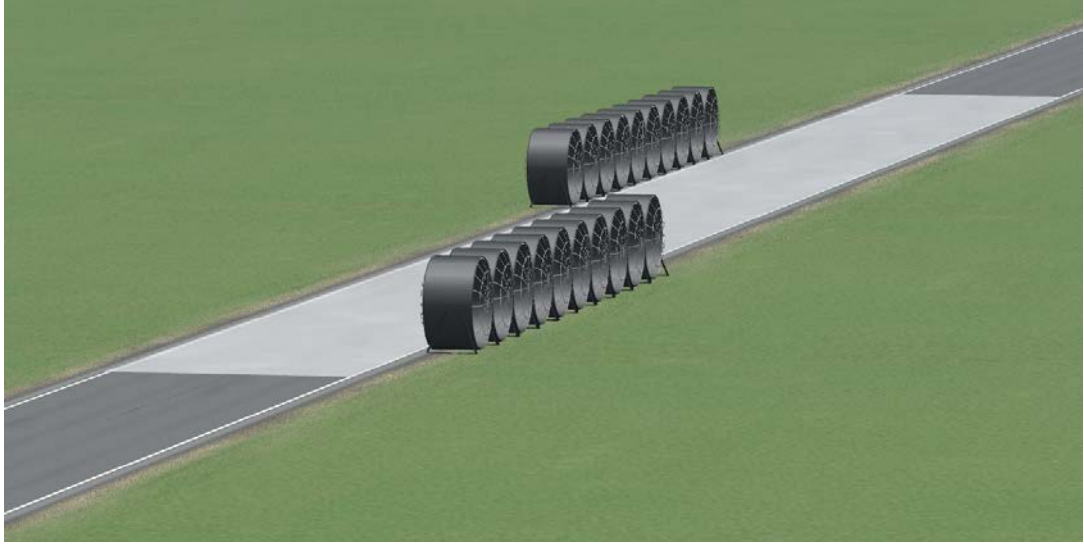


Figure 5.6 : The CWD test track with a low-friction road surface.

The aim is to evaluate the performances of the proposed controllers against external yaw disturbances. The initial speed of the vehicle is 80km/h and neither braking nor throttle has been applied to the vehicle during the test. Figure 5.7 shows the obtained results of the yaw rate and the vehicle trajectories for uncontrolled and controlled cases. It can be seen that all of the control approaches reject the effects of the wind disturbance in an effective way even in low- friction roads while the uncontrolled vehicle drifts out. The obtained performances of the controllers are very close to each other and practically the same behavior is achieved but, smaller tracking error and faster tracking response are achieved with the proposed approaches of SMDO and SMEDE which are designed by using sliding mode control theory.

Inverse model based disturbance observer and sliding mode disturbance observer are two powerfull disturbance observer approaches in motion control. However, inverse model based disturbance observer lacks of robustness against high frequency nonlinear disturbances. Moreover, tuning of the parameters of inverse model based disturbance observer is very difficult since the filter design must guarantee both the causality of the estimator and the stability of whole control system (She et al. 2007).

On the other hand, the proposed sliding mode disturbance observer is a chattering-free approach while the conventional sliding mode disturbance observers which are proposed in the literature mostly includes chattering in the control signal.

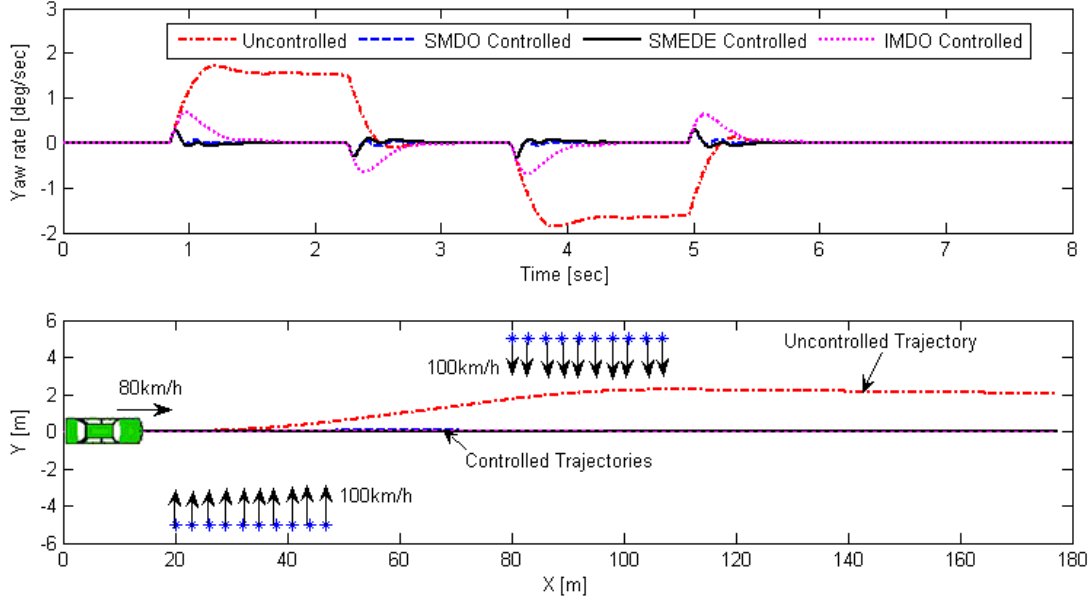


Figure 5.7 : The obtained results for the CWD test in CarSim.

5.2 Results for Lateral and Yaw Motion Decoupling Control

5.2.1 Simulation study with linear vehicle dynamics

In the first group of simulation experiments for lateral and yaw motion decoupling control, the proposed MIMO SMDO is applied to the same vehicle that is given in the paper of A Guvenc et al. (2010) and the performances of the proposed MIMO SMDO, the MIMO IMDO (Figure 5.8) and PID controllers are compared. Two separate PID controllers are used to control two independent control loops which are the yaw rate and side slip angle control loops. While the PID gains of K_{p1} , K_{i1} and K_{d1} are belongs to the yaw rate control loop K_{p2} , K_{i2} and K_{d2} are for the sideslip angle control loop. The numerical values of the vehicle parameters, the PID gain values and the time constants of the Q-filters of the MIMO IMDO used in the simulations by A Guvenc et al. (2010) are given in Table 5.2.

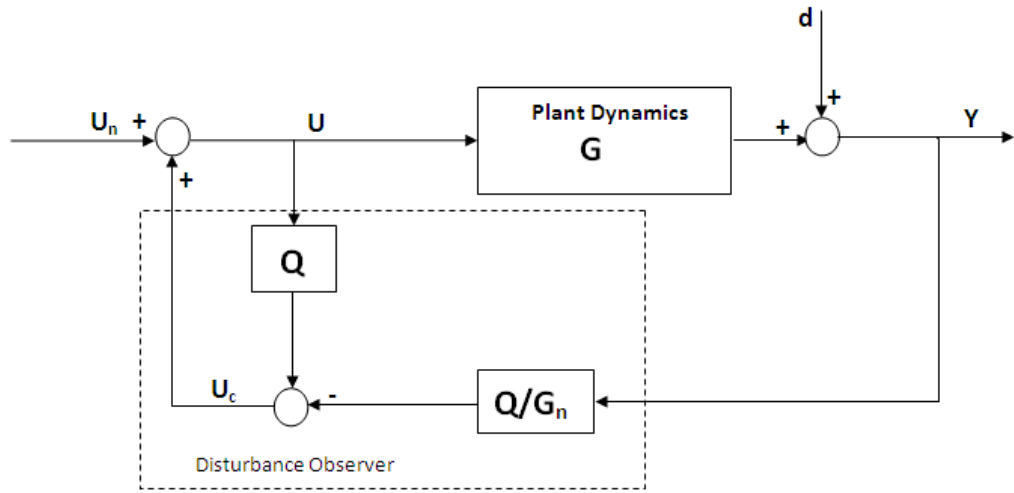


Figure 5.8 : Inverse model based disturbance observer (IMDO).

Table 5.2 : The vehicle parameters and the PID gain values.

Symbol	Value	Symbol	Value
V_{COG}	30m/s	m	1296kg
l_f	1.25m	K_{p1}	0.4
l_r	1.32m	K_{i1}	2
I_z	1750kgm ²	K_{d1}	0.03
τ_{n1}	0.14s	K_{p2}	0.7
τ_{n2}	0.2s	K_{i2}	3.7
C_f	84000N/rad	K_{d2}	0.06
C_r	96000N/rad	τ_{Q1}	0.04s
		τ_{Q2}	0.06s

The sampling time, T is $1ms$ and design parameters of the MIMO SMDO used in (4.18) are as follows.

$$\Gamma = \begin{bmatrix} 5 & 0 \\ 0 & 5 \end{bmatrix} \quad H = \begin{bmatrix} 0.01 & 0 \\ 0 & 0.01 \end{bmatrix}$$

The simulation results for front and rear wheel steering angle step inputs are shown in Figure 5.9 and Figure 5.10 respectively. It can be seen that the yaw rate only responds to the front steering angle while the response of the vehicle side slip angle to the front steering angle is attenuated as expected. The reference yaw rate dynamic is tracked successfully. In a same way, the vehicle side slip angle only responds to the rear wheel steering angle while the response of the yaw rate to the rear wheel

steering angle is attenuated as expected and the reference vehicle side slip angle dynamic is tracked successfully.

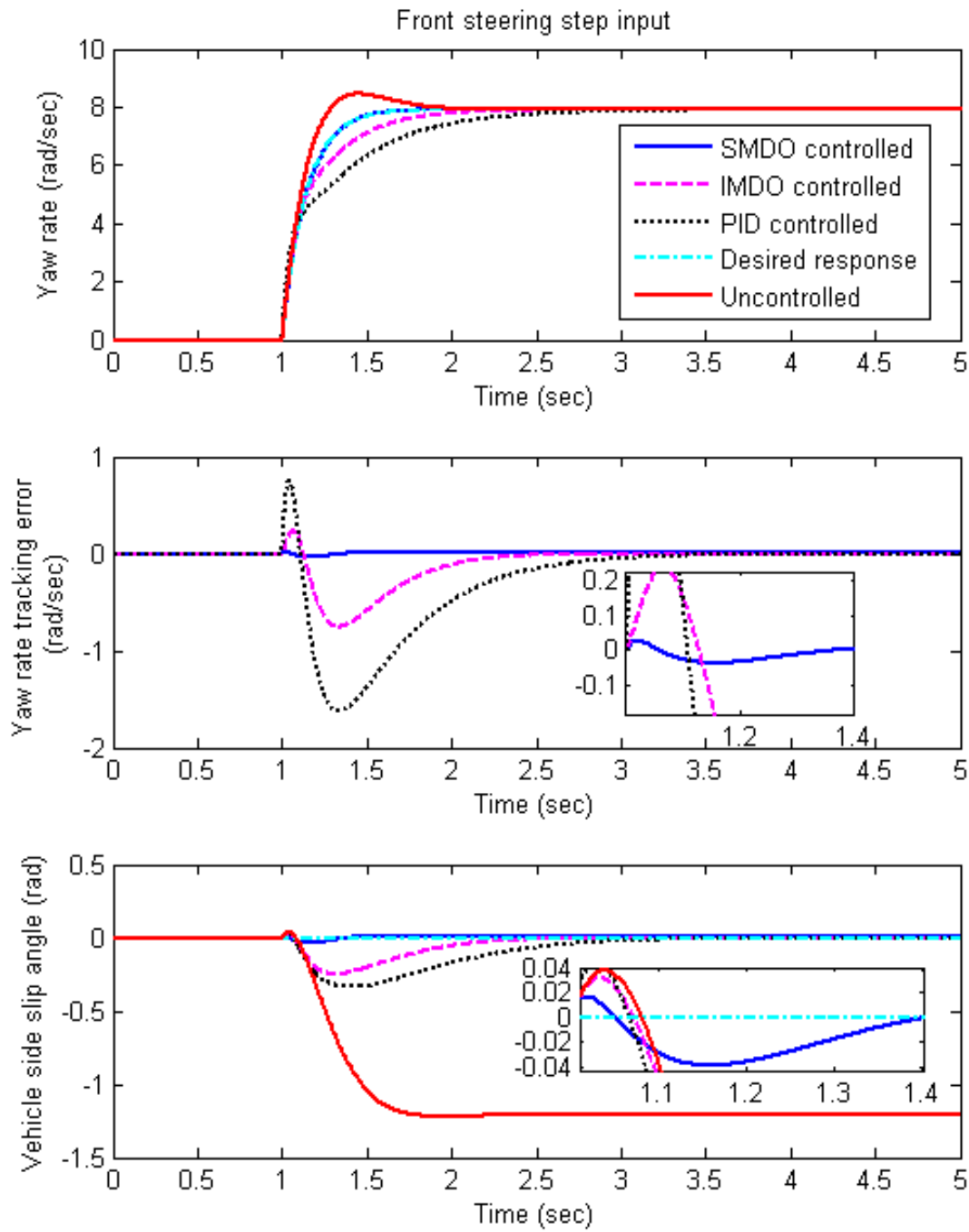


Figure 5.9 : Front wheel step steering input results for linear vehicle dynamics.

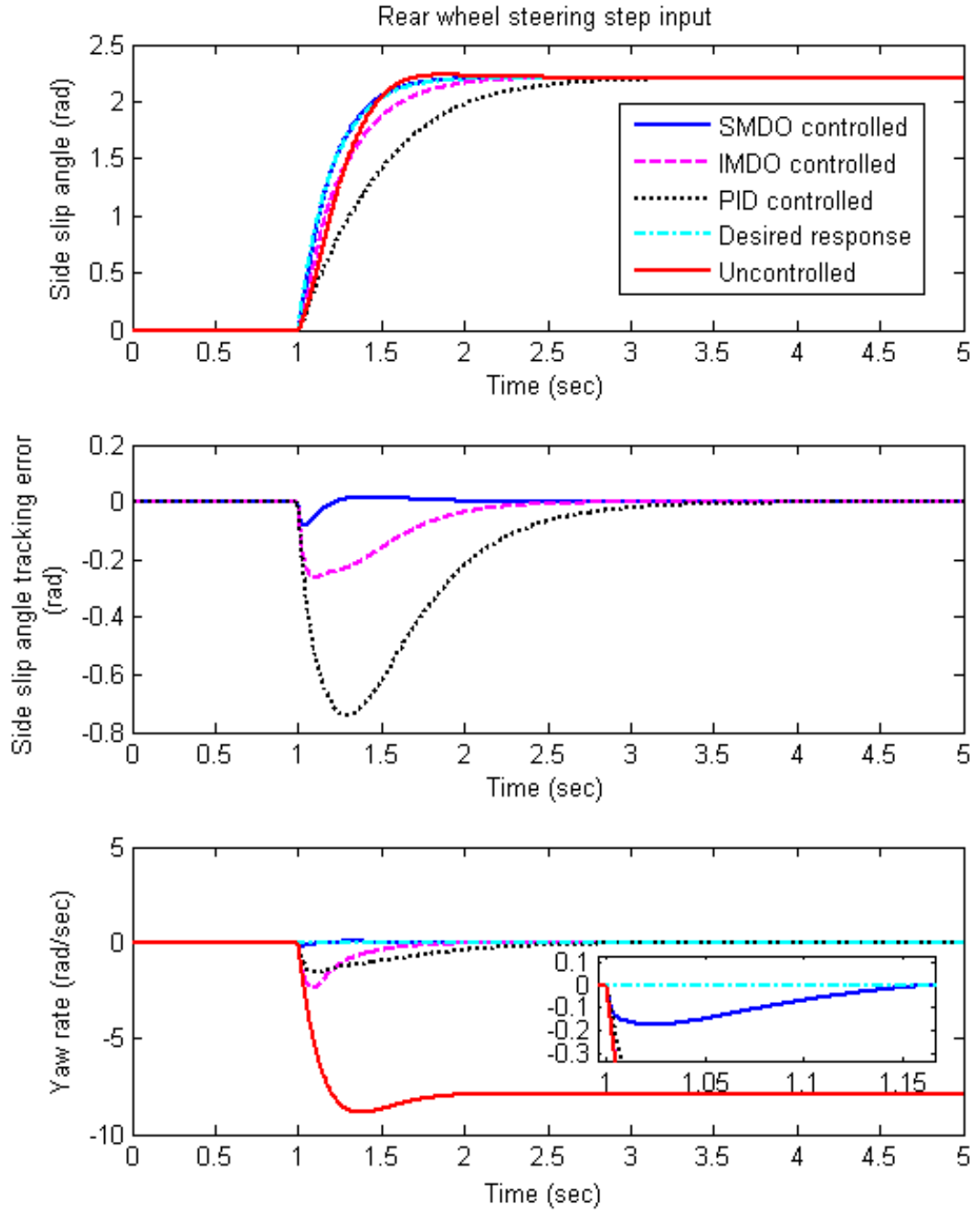


Figure 5.10 : Rear wheel step steering input results for linear vehicle dynamics.

The external disturbance rejection test simulation is carried out. A step yaw moment input is applied to the system and the results are plotted in Figure 5.11. It is clear that excellent disturbance rejection performance is achieved. The yaw rate and the vehicle side slip angle are not affected much by side wind disturbance input.

When the performances of the MIMO SMDO, the MIMO IMDO and PID control are compared, it can be seen from Figure 5.9, Figure 5.10 and Figure 5.11 that better results are achieved with the proposed MIMO SMDO.

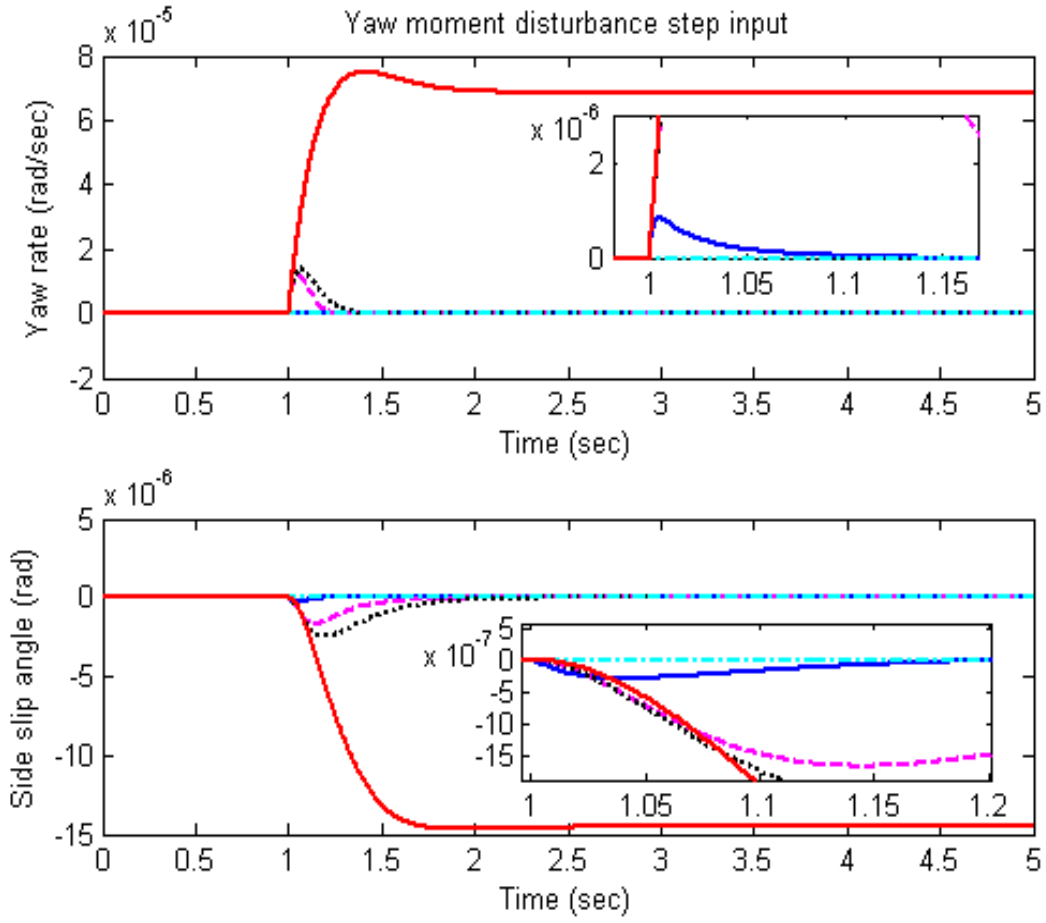


Figure 5.11 : Yaw moment disturbance input results for linear vehicle dynamics.

5.2.2 Simulation study with Nonlinear vehicle dynamics

The simulation study given above are carried out for a simple linear four wheel active steering dynamics with simple tire model. In this section, the proposed MIMO SMDO is applied to a Fiat Doblo Van equipped with four wheel steering actuators which is modeled in CarSim environment. The simulations are performed by running CarSim software and Matlab/Simulink simultaneously since the proposed MIMO SMDO is modelled in Matlab/Simulink environment while the vehicle in CarSim. The simulation environment is illustrated in Figure 5.12. The simulations studied in the previous section are repeated for the Fiat Doblo Van. In the simulations, the vehicle enters into the maneuver with a speed of 120km/h on a dry concrete surface. A front and rear wheel steering step inputs are applied to the vehicle. The results are shown in Figure 5.13 and Figure 5.14 respectively.

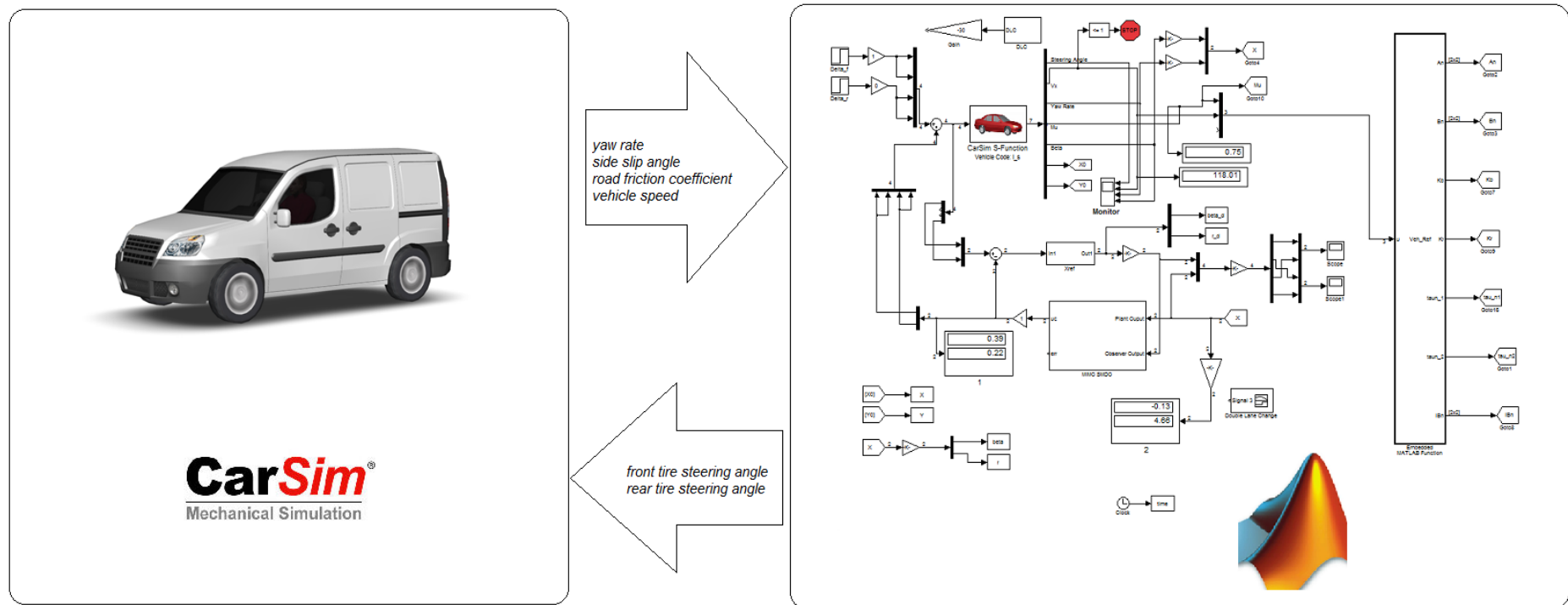


Figure 5.12 : CarSim and Matlab/Simulink co-simulation and data exchange diagram for the lateral and yaw motion decoupling control.

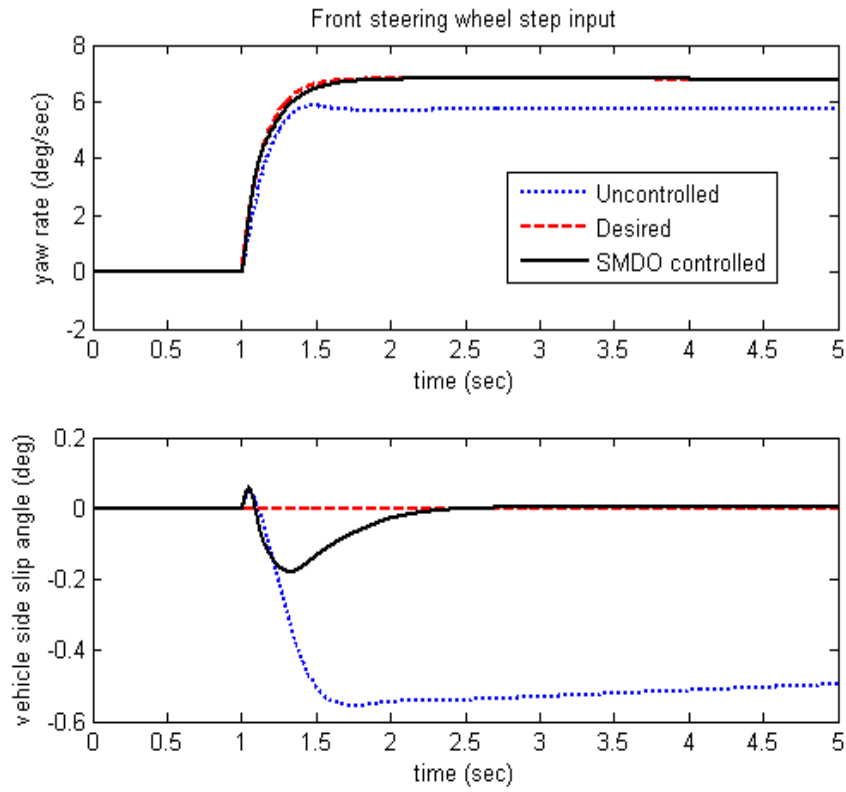


Figure 5.13 : Front wheel step steering input results for the Doblo Van.

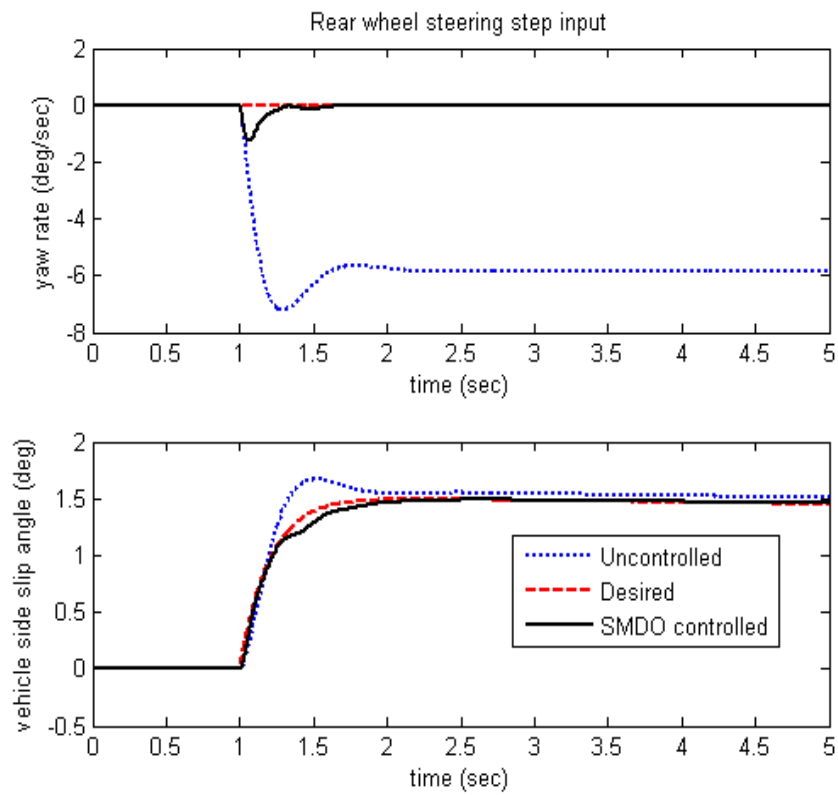


Figure 5.14 : Rear wheel step steering input results for the Doblo Van.

It can be seen from the figures that the yaw rate only responds to the front steering angle while the vehicle side slip angle only responds to the rear wheel steering angle. The response of the vehicle side slip angle to the front steering angle and the response of the yaw rate to the rear wheel steering angle are attenuated and the reference yaw rate and side slip angle dynamics are tracked successfully.

To see the external disturbance rejection performance of the MIMO SMDO, a 40km/h side wind force is applied to the vehicle at 0.5s. The results are given in Figure 5.15. The figure shows that excellent disturbance rejection performance is achieved. The yaw rate and the vehicle side slip angle are not affected much by side wind disturbance input.

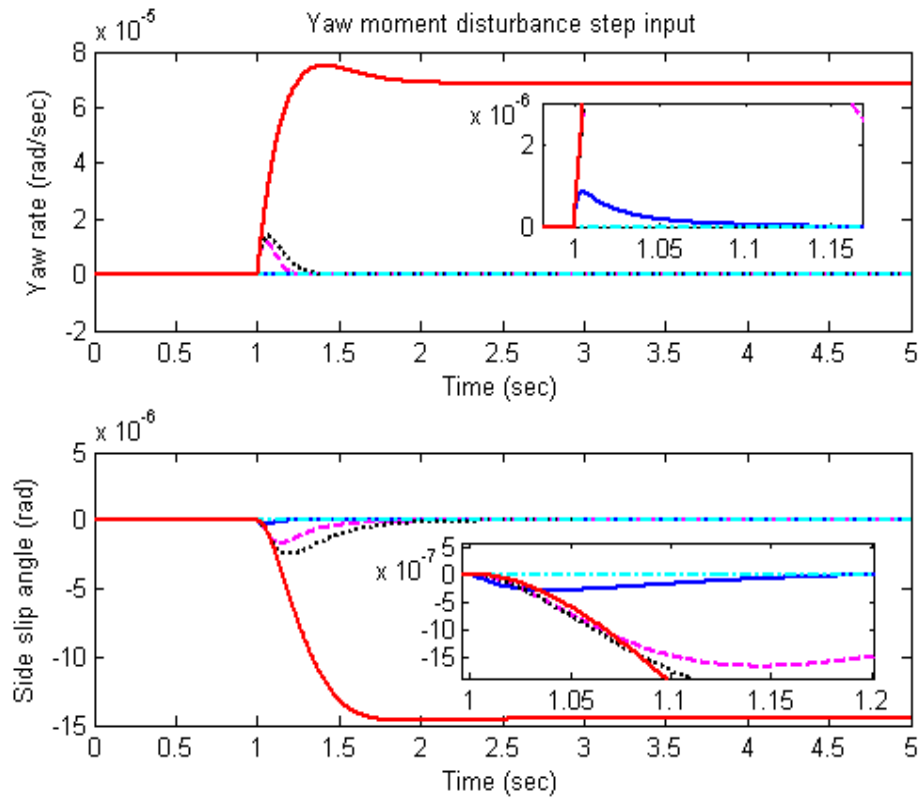


Figure 5.15 : Side wind disturbance input results for the Doblo Van.

The simulation results of CarSim European Van validate the perfect input-output decoupling and disturbance rejection performances of the proposed MIMO SMDO. Strong couplings between lateral and yaw motion dynamics are removed successfully by the MIMO SMDO. After the decoupling of the dynamics, it is possible to design independent servo controllers by only considering nominal plant dynamics in the outer loop for the lateral and yaw motion references related to lane

keeping control, collision avoidance, stability and handling control, etc. With such a decoupled vehicle dynamics, it is possible to assign any arbitrary yaw rate and side slip angle references. A zero lateral speed can easily be achieved while referencing yaw rate, which yields improved comfort and maneuverability and stable cornering maneuvers. Moreover, a zero yaw rate for a stable obstacle avoidance maneuver or lane keeping can be obtained with a simple outer linear servo control loop.

5.3 Results for Integrated Stability Control of 4WS4WD Electric Vehicles

The integrated vehicle dynamics control for 4WS4WD electric vehicles was tested on a Fiat Doblo Van. The vehicle is equipped with four axle motors and front and rear steering actuators. The control system and the vehicle powertrain is modeled in Matlab/Simulink and it is co-simulated with CarSim (Figure 5.16).

5.3.1 Double lane change maneuver

The first test scenario is Double Lane Change (DLC) maneuver. In the DLC maneuver, a driver steering input as shown in Figure 5.2 and a fixed 20% throttle input have been applied to the vehicle with initial speeds of 130km/h and 120km/h on a wet road and an icy road respectively. The aim is to evaluate the performance of the proposed control approach in extreme driving maneuvers in which the vehicle dynamics is highly non-linear and there are highly non-linear modeling and parameter uncertainties. By examining Figure 5.17 and Figure 5.18, it can be seen that the vehicle equipped with the proposed controller makes DLC maneuvers successfully while the uncontrolled vehicle cannot and the vehicle drifts out. The control system is very successful to improve handling and stability of the vehicle even in extreme driving maneuvers. The actuator efforts that are allocated using the proposed adaptive optimal control allocation approach are given in Figure 5.19, Figure 5.20, Figure 5.21 and Figure 5.22 for the two DLC maneuvers. The results show that the yaw moment is successfully distributed between the actuators in an optimal manner.

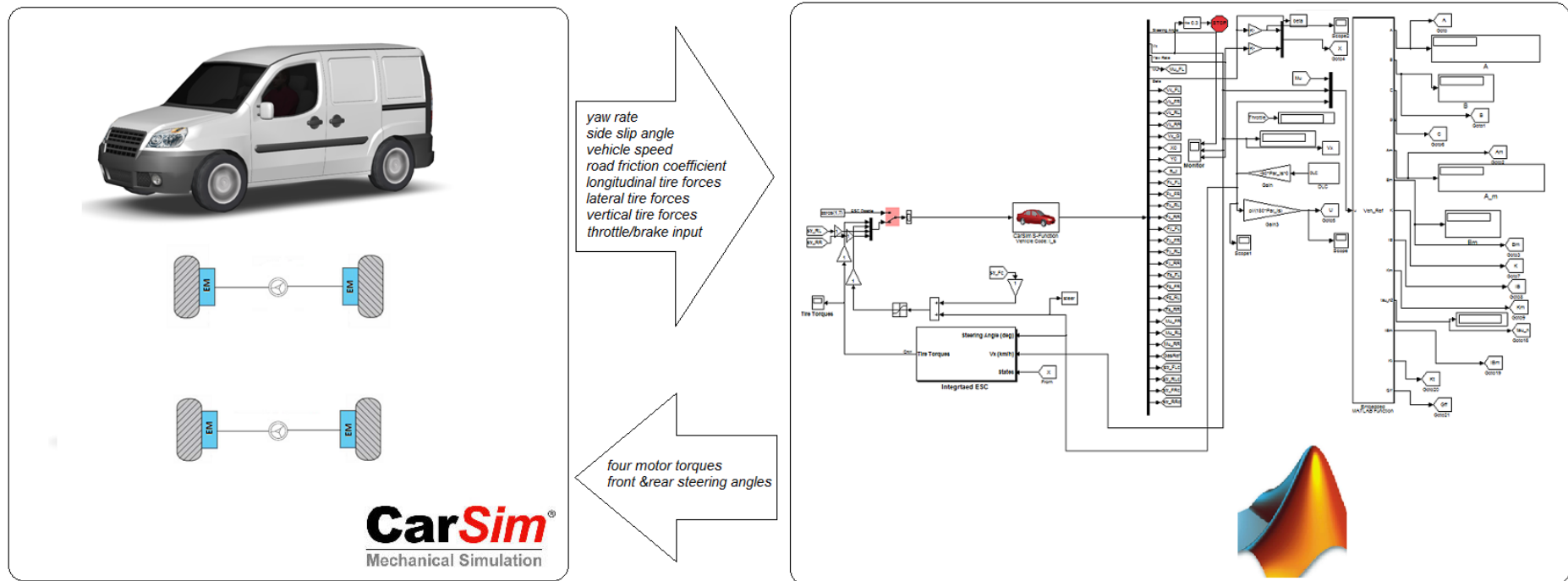


Figure 5.16 : CarSim and Matlab/Simulink co-simulation and data exchange diagram for the integrated stability control of 4WS4WD vehicles.

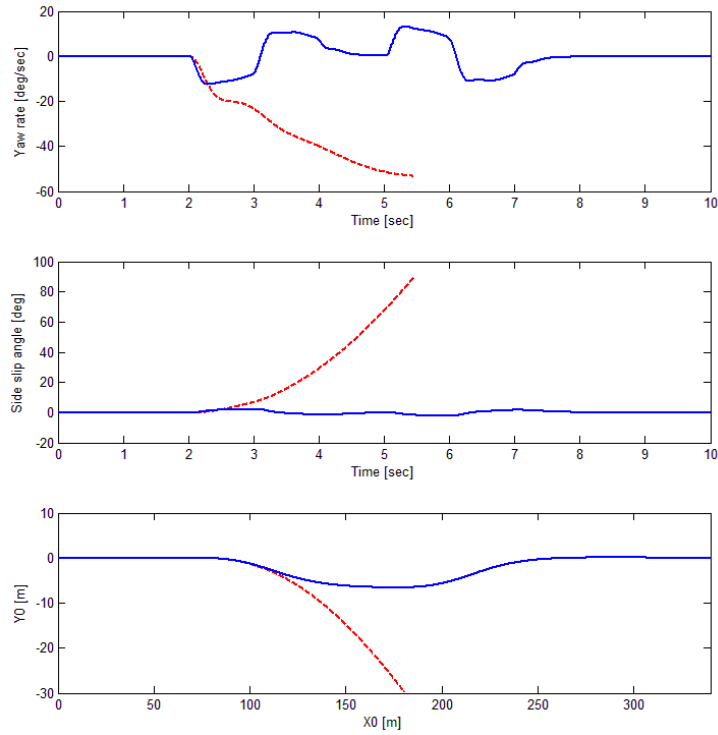


Figure 5.17 : The results in DLC maneuver with initial speed of 130km/h on a wet road (red-dot line: uncontrolled vehicle, blue-continuous line controlled vehicle).

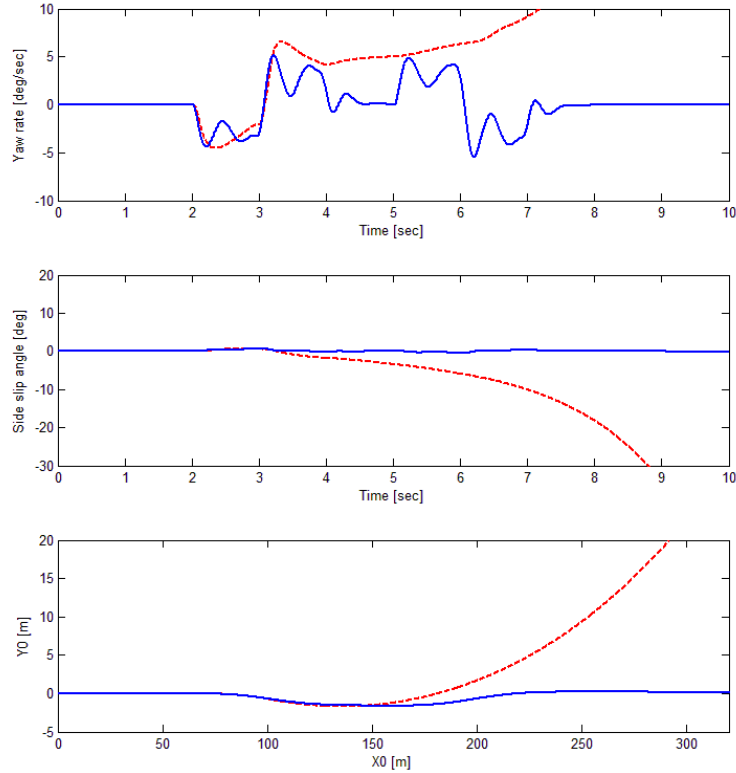


Figure 5.18 : The results in DLC maneuver with 120km/h on an icy road (red-dot line: uncontrolled vehicle, blue-continuous line: controlled vehicle).

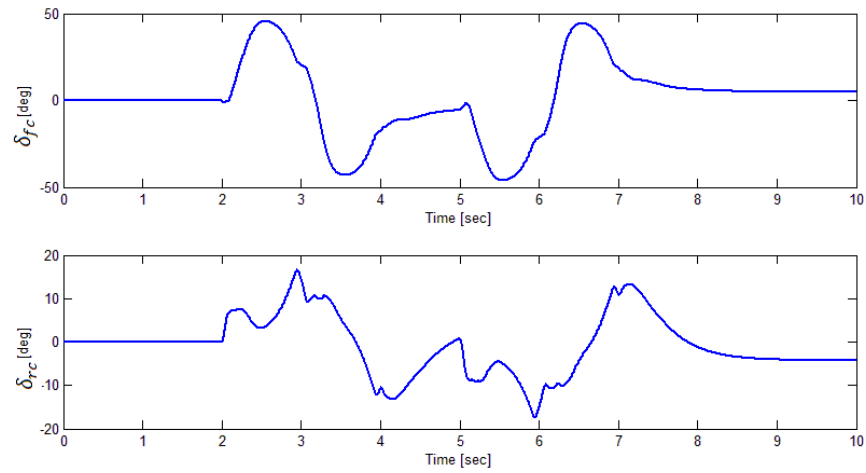


Figure 5.19 : The steering angles in the DLC maneuver with initial speed of 130km/h on a wet road.

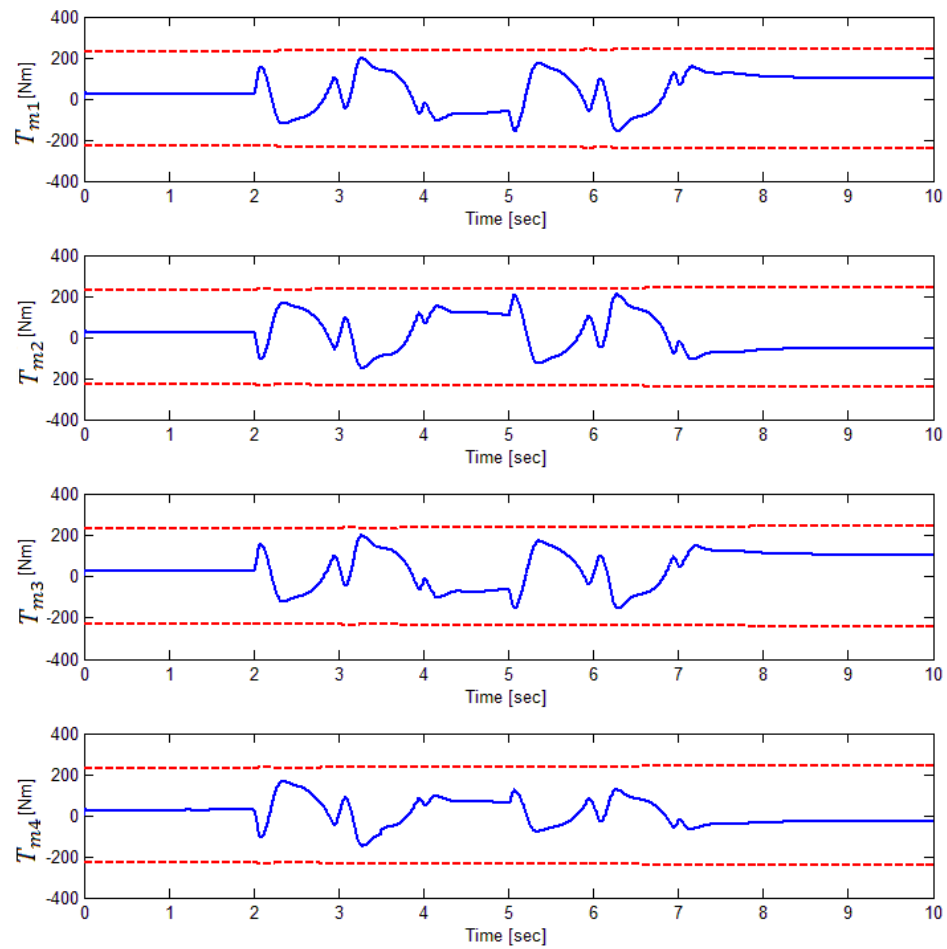


Figure 5.20 : The motor torques in the DLC maneuver with initial speed of 130km/h on a wet road (Red-dot lines shows the motor torque limits)

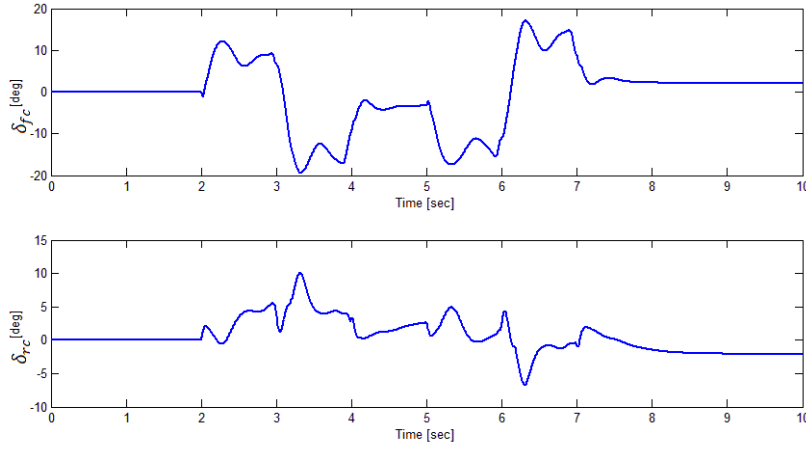


Figure 5.21 : The steering angles in the DLC maneuver with initial speed of 120km/h on an icy road.

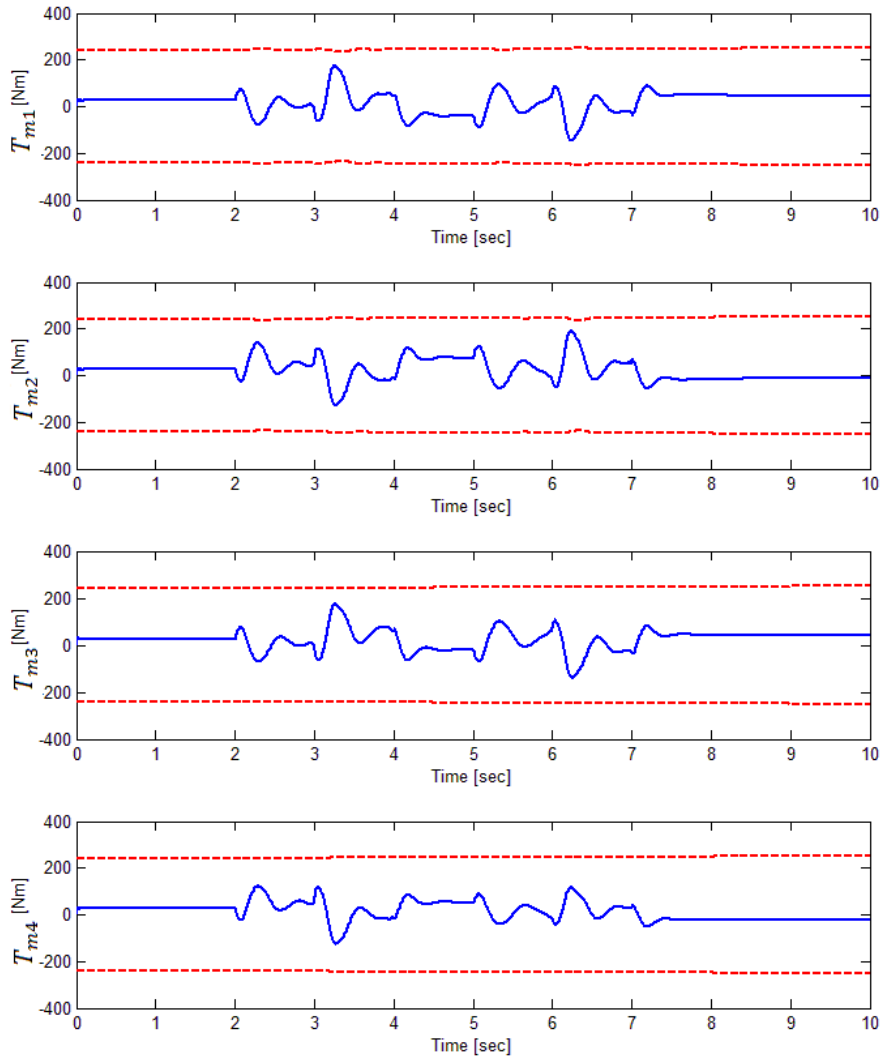


Figure 5.22 : The motor torques in the DLC maneuver with initial speed of 120km/h on an icy road (Red-dot lines shows the motor torque limits)

5.3.2 Cross-wind disturbance test

The other test scenario is cross-wind disturbance (CWD) test in which the aim is to evaluate the performance of the proposed vehicle dynamics control system against external yaw disturbances. In this test, the vehicle enters in to a test track in which there exist fan blowers at each side and specific positions of an icy road to create a 100km/h side wind. The initial speed of the vehicle is 120km/h and 20% throttle input are applied to the vehicle during the test. Figure 5.23 shows the obtained results of the yaw rate and the vehicle trajectories for uncontrolled and controlled cases. It can be seen that the proposed control system rejects the effects of the wind disturbance in an effective way even in low-friction road while the uncontrolled vehicle drifts out. The allocated actuator efforts are shown in Figure 5.24 and Figure 5.25. The simulation results for double lane change maneuver and cross wind disturbance rejection test carried out in CarSim environment shows that the proposed control allocation approach is very successful in vehicle dynamic stabilization and optimal control allocation.

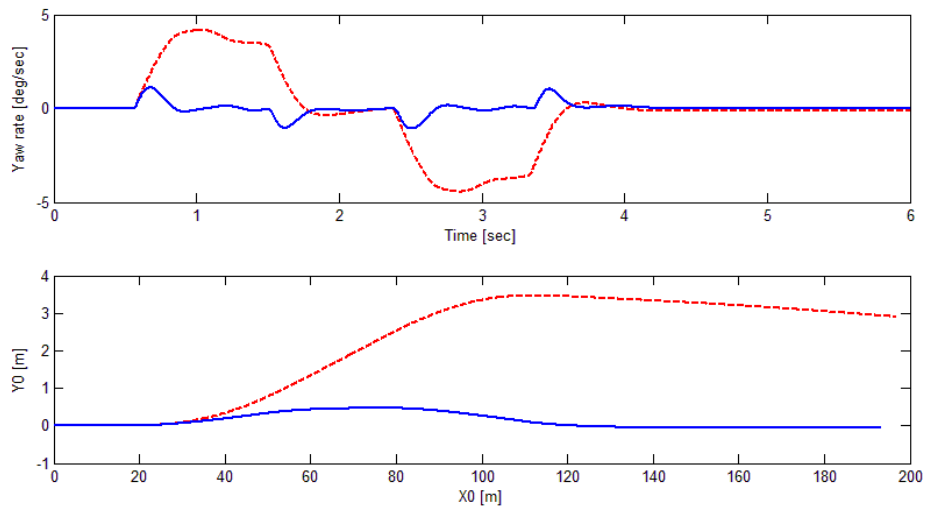


Figure 5.23 : The obtained results for the CWD test (red-dot line: uncontrolled vehicle, blue-continuous line: controlled vehicle).

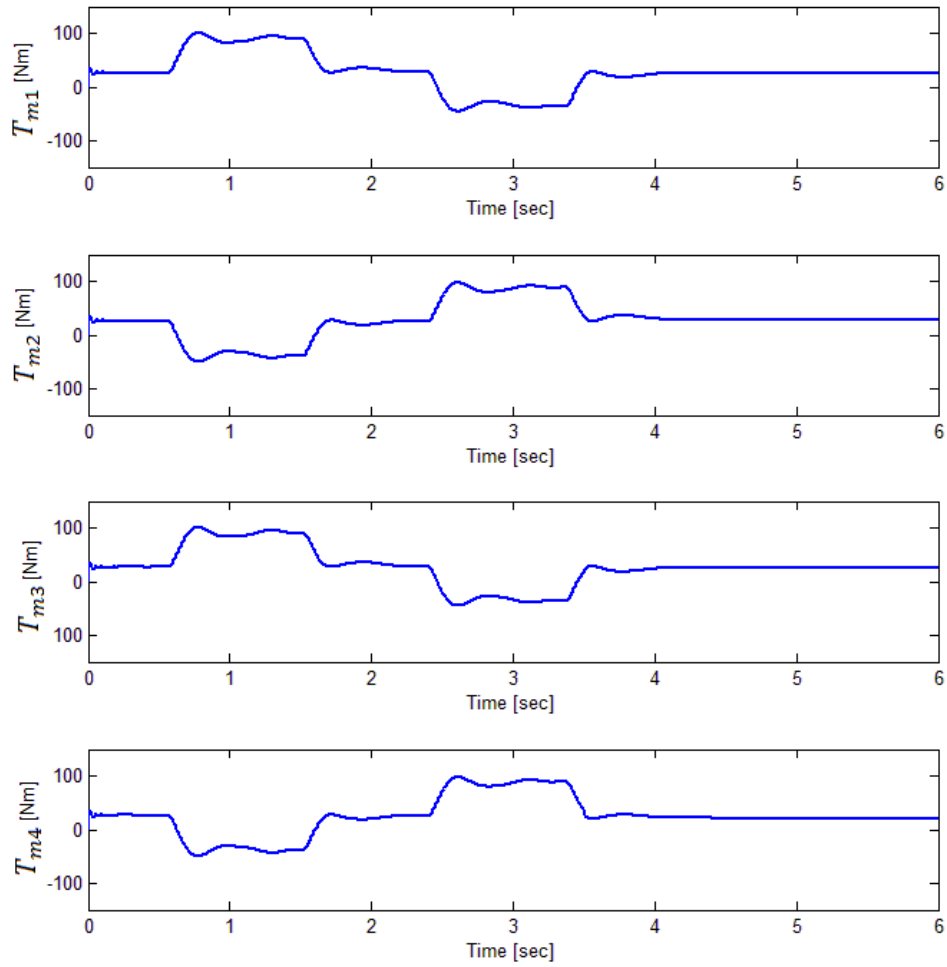


Figure 5.24 : The motor torques in the CWD test.

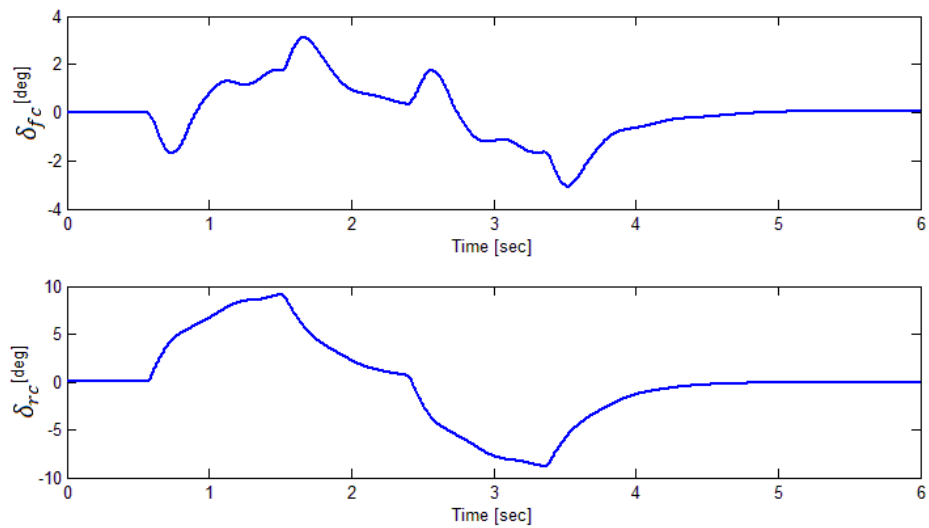


Figure 5.25 : The steering angles in the CWD test

5.4 Results for Integrated Stability Control by Coordination of AFS and Individual Brakes

The integrated vehicle stability control by coordination of individual braking and active front steering actuators was tested on a Fiat Doblo Van. The control system and the vehicle powertrain is modeled in Matlab/Simulink and it is co-simulated with CarSim (Figure 5.26). The simulation experiments and results are given in this section.

5.4.1 Double lane change maneuver

In the DLC maneuver, a driver steering input as shown in Figure 5.2 is applied to the vehicle with initial speeds of 120km/h and 100km/h on wet and icy roads respectively. As before, the aim is to evaluate the performance and robustness of the proposed control approach in extreme driving maneuvers in which the vehicle dynamics are highly non-linear and there are highly non-linear modeling and parameter uncertainties.

By examining Figure 5.27 and Figure 5.28, it can be seen that the vehicle equipped with the proposed controller makes the double lane change maneuvers successfully while the uncontrolled vehicle cannot and the vehicle drifts out. The proposed control system is very successful to improve handling and stability of the vehicle even in extreme driving maneuvers. Vehicle speed, tire speeds and slips for the double lane change maneuvers are given in Figure 5.29 and Figure 5.30. Since the tire slips are remained under the slip threshold, the wheel slip controller is not triggered in these maneuvers. The actuator efforts that are allocated using the proposed adaptive optimal control allocation approach are given in Figure 5.31 and Figure 5.32. The figures show that the yaw moment is successfully distributed between the actuators in an optimal manner.

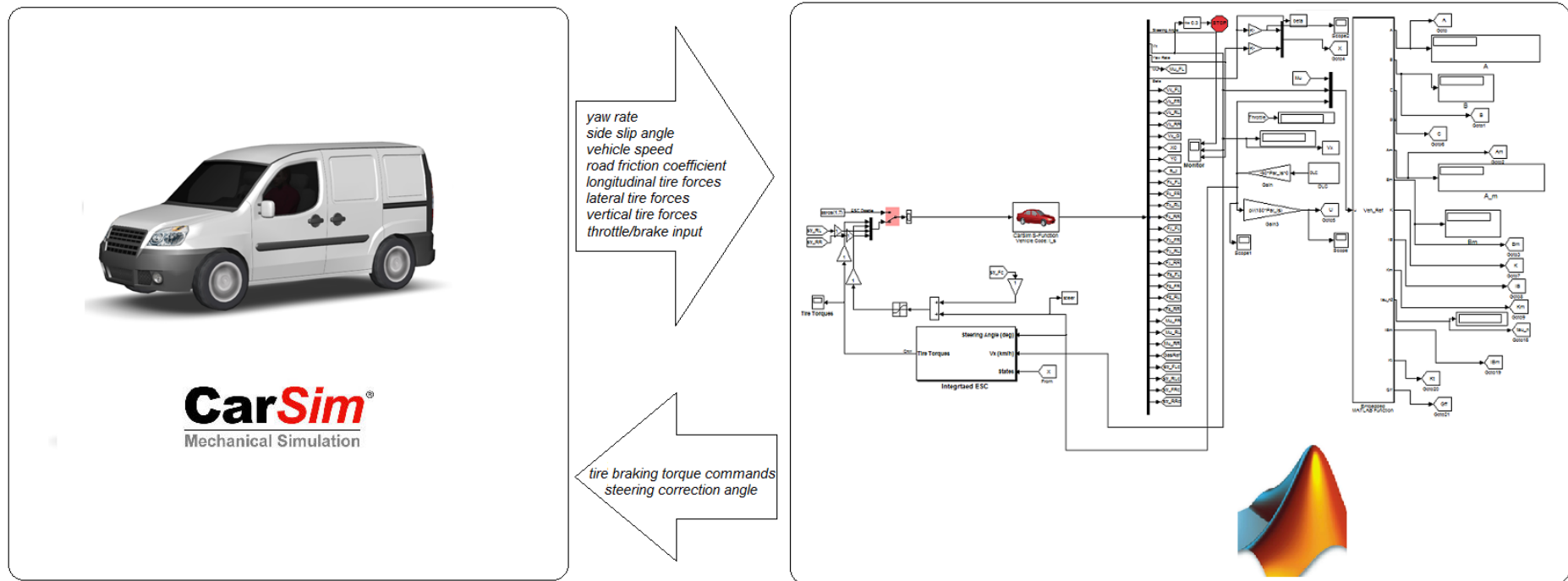


Figure 5.26 : CarSim and Matlab/Simulink co-simulation and data exchange diagram for the integrated stability control by coordination of AFS and Individual braking actuators.

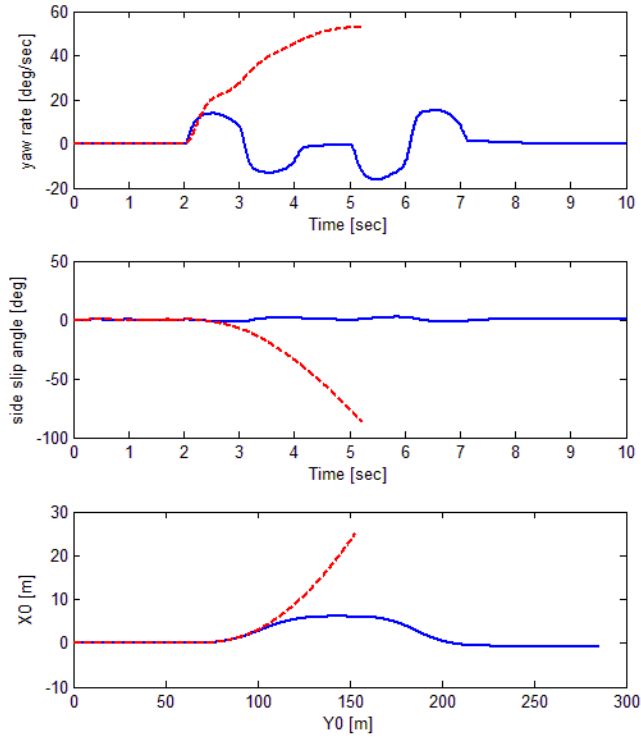


Figure 5.27 : The results in DLC maneuver with initial speed of 120km/h on a wet road (red-dot line: uncontrolled vehicle, blue-continuous line: controlled vehicle).

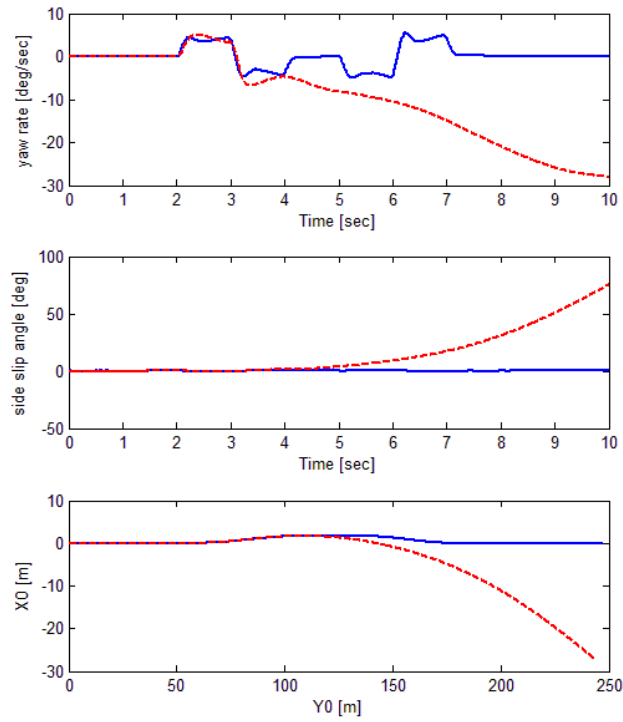


Figure 5.28 : The results in DLC maneuver with 100km/h on an icy road (red-dot line: uncontrolled vehicle, blue-continuous line: controlled vehicle).

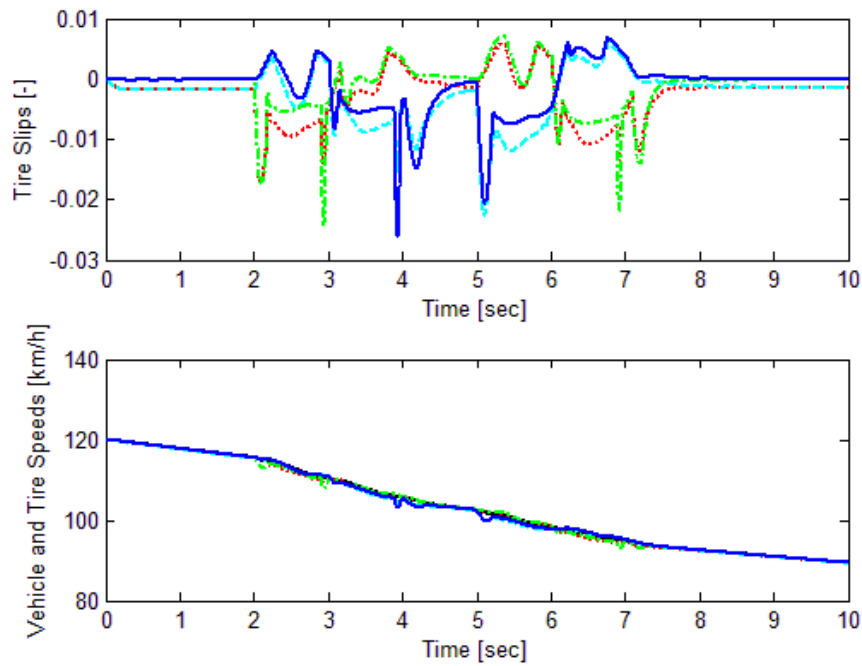


Figure 5.29 : The vehicle speed, tire speeds and slips in the DLC maneuver with initial speed of 120km/h on a wet road

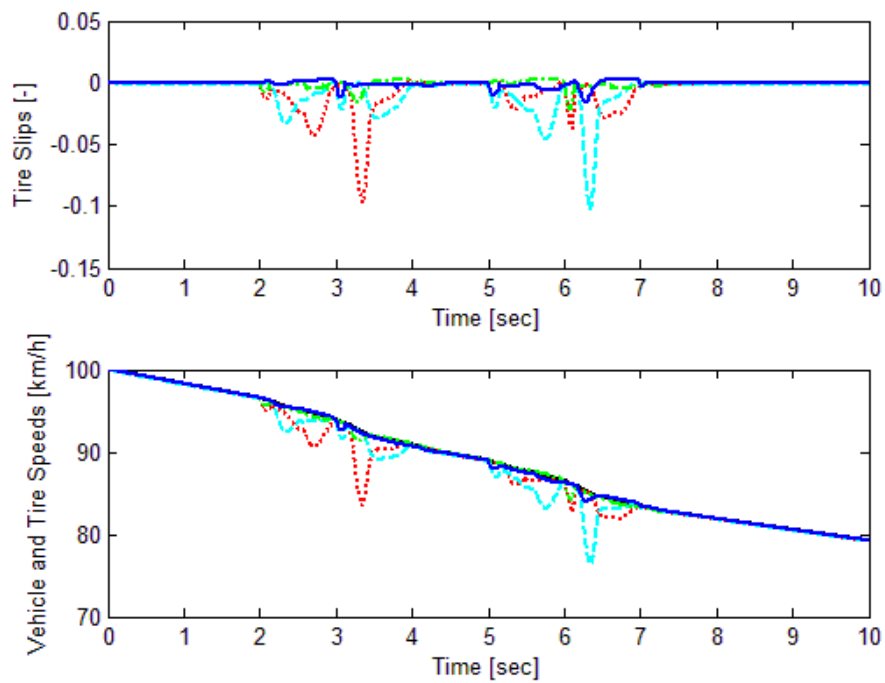


Figure 5.30 : The vehicle speed, tire speeds and slips in the DLC maneuver with initial speed of 100km/h on an icy road

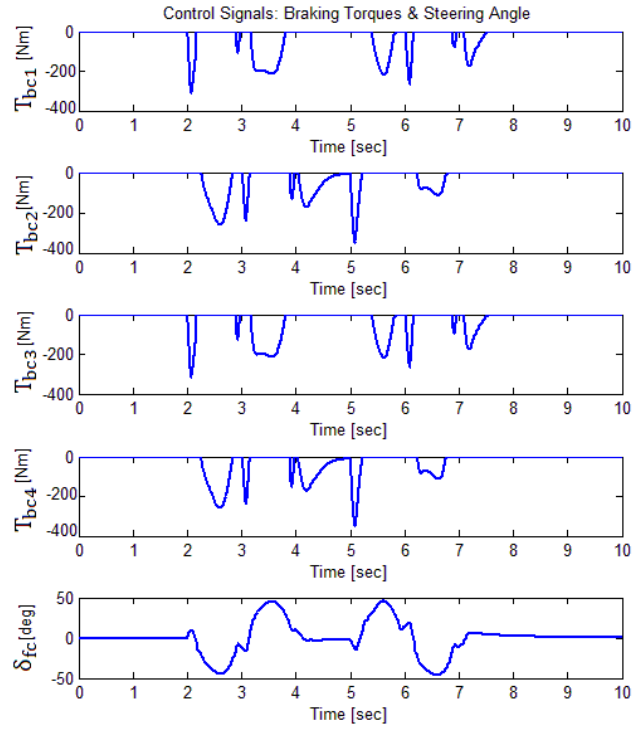


Figure 5.31 : The individual braking torques and corrective steering angle in the DLC maneuver with initial speed of 120km/h on a wet road.

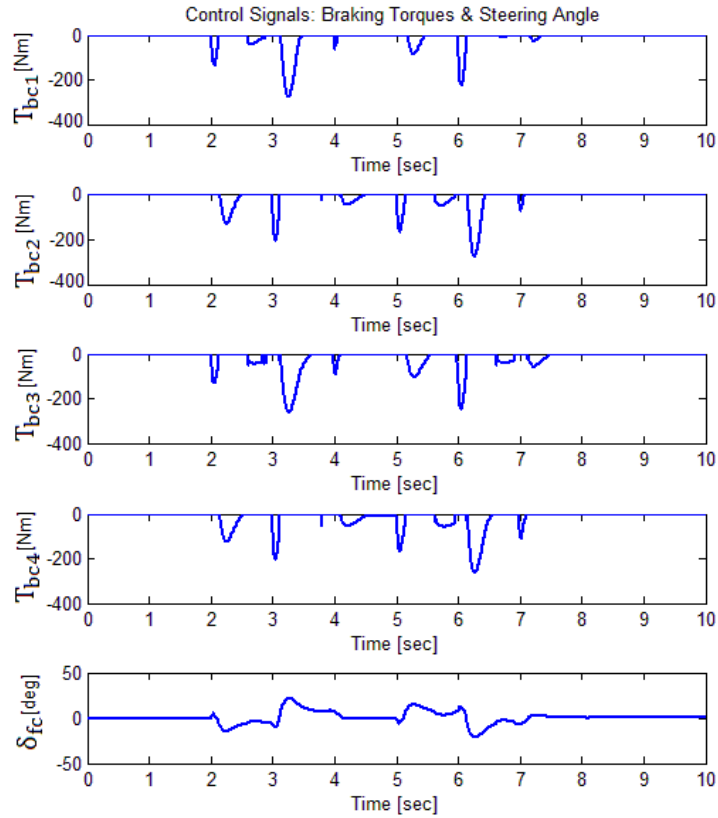


Figure 5.32 : The individual braking torques and corrective steering angle in the DLC maneuver with initial speed of 100km/h on an icy road.

5.4.2 Cross-wind disturbance (CWD) test

The aim in the CWD test is to evaluate the performance and robustness of the proposed control system against external yaw disturbance caused by side winds. The vehicle enters in to a test track with an initial speed of 100km/h in which there exist fan blowers at each side and specific positions of an icy road to create a 100km/h side wind. Figure 5.33 shows the obtained results of yaw rate, side slip angle and vehicle trajectories for the uncontrolled and controlled cases. It can be seen that the proposed control system reject the effects of the wind disturbance in an effective way even in low- friction road while the uncontrolled vehicle drifts out. The allocated actuator efforts are shown in Figure 5.34.

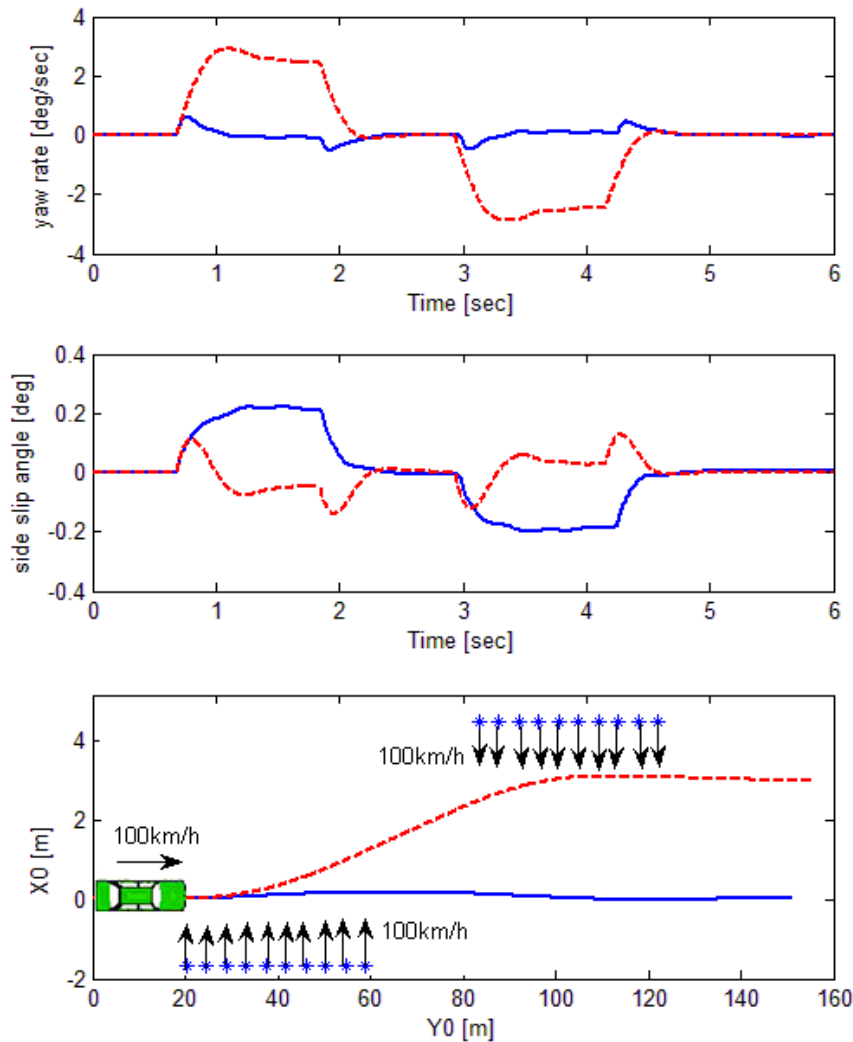


Figure 5.33 : The vehicle states in the CWD test (red-dot: uncontrolled vehicle, blue-continuous: controlled vehicle).

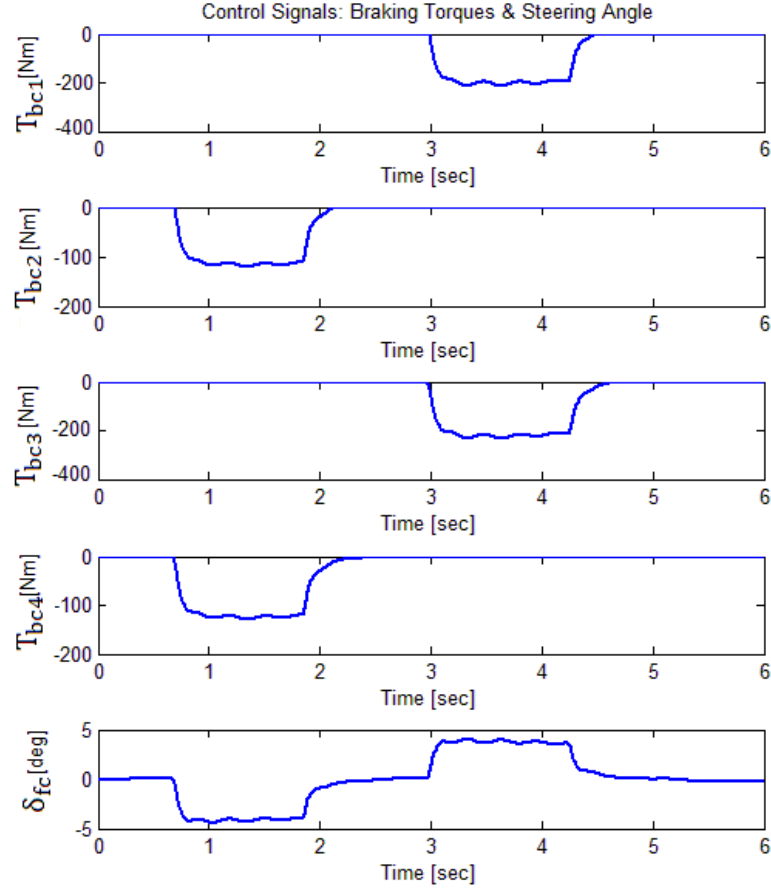


Figure 5.34 : The individual braking torques and corrective steering angle in the CWD test.

5.4.3 μ -split braking maneuver

In this maneuver, the road friction coefficient, μ is different on each side of the vehicle and a hard braking is applied to the vehicle during one second on such a non-uniform friction surface. Initial speed of the vehicle is 65km/h and the left side of the vehicle on an icy surface while right side on a wet surface during braking. Figure 5.35 shows the obtained results of yaw rate, side slip angle and vehicle trajectories for the uncontrolled and controlled cases. The proposed vehicle control system rejects the effects of the yaw disturbance caused by the μ -split braking in an effective way while the uncontrolled vehicle drifts out. Moreover, it can be seen from Figure 5.36 that the tire slips exceed the slip threshold due to hard braking but, the wheel slip controllers are activated and the tire slips are successfully regulated by the proposed wheel slip controller scheme. The control efforts of the actuators are plotted in Figure 5.37. Due to the wheel slip controller, the braking torques of the left

side wheels are reduced to maintain the wheel slips but, the remaining corrective yaw moment is generated by the active front steering actuator.

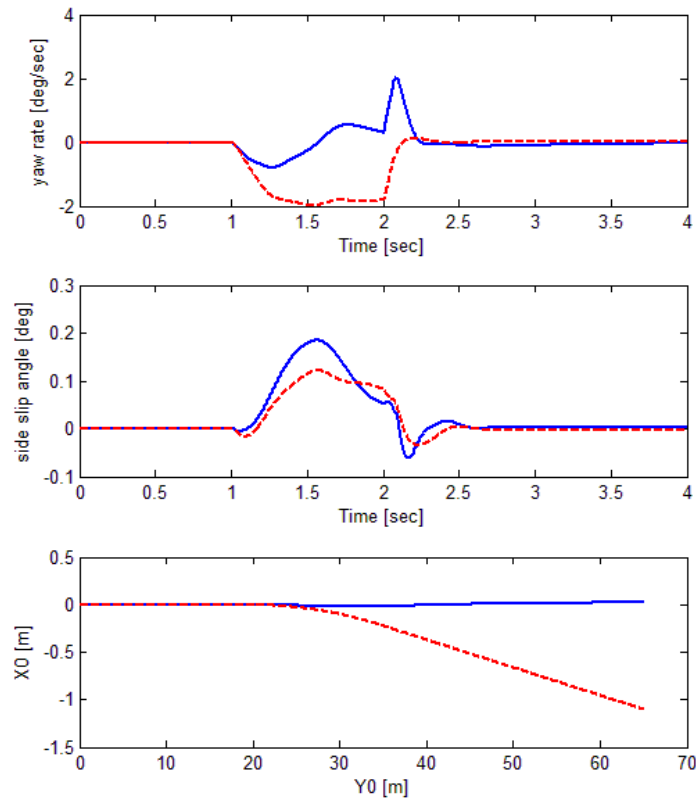


Figure 5.35 : The vehicle states in the μ -split braking maneuver (red-dot: uncontrolled vehicle, blue-continuous: controlled vehicle).

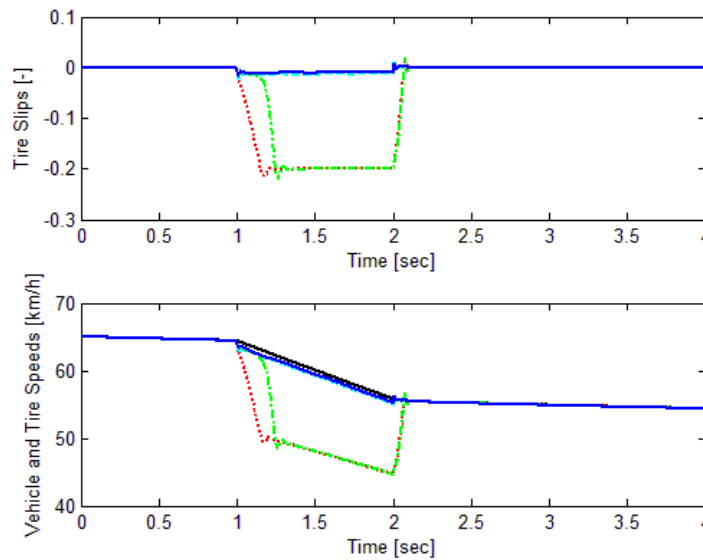


Figure 5.36 : The vehicle speed, tire speeds and slips in the μ -split braking maneuver.

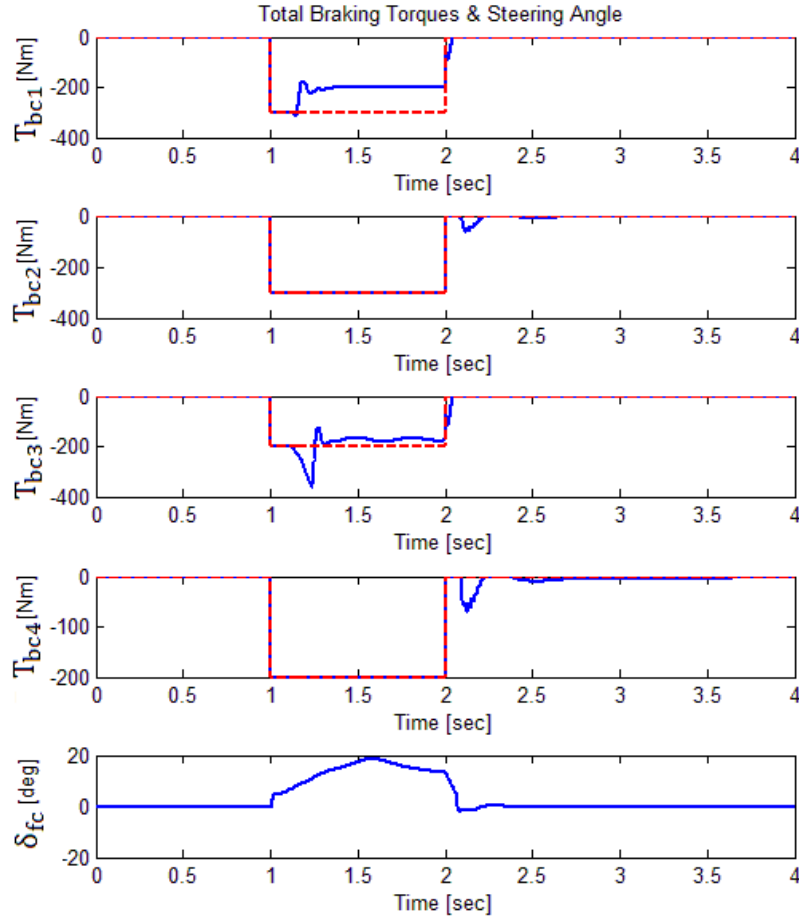


Figure 5.37 : The individual braking torques and corrective steering angle in μ -split braking maneuver.

5.4.4 Actuator failure scenario

Due to its dynamic optimal allocation structure, the proposed adaptive optimal control allocation scheme may compensate partial actuator failures and may assure an active fault tolerant vehicle dynamics control system. To test robustness of the control system against partial actuator failures, the double lane change maneuver with initial speed of 120km/h on a wet road is repeated here but, in addition, assuming the front left and rear right braking actuators are failed at 3rd second of the maneuver ($T_{bc1} = T_{bc4} = 0$). The results given in Figure 5.38 and Figure 5.39 show that the required yaw moment is distributed among the available actuators, i.e. the front right and rear left braking actuators and the active front steering actuator and effect of the front left and rear right braking actuator failures are successfully compensated by the proposed adaptive optimal control allocation.

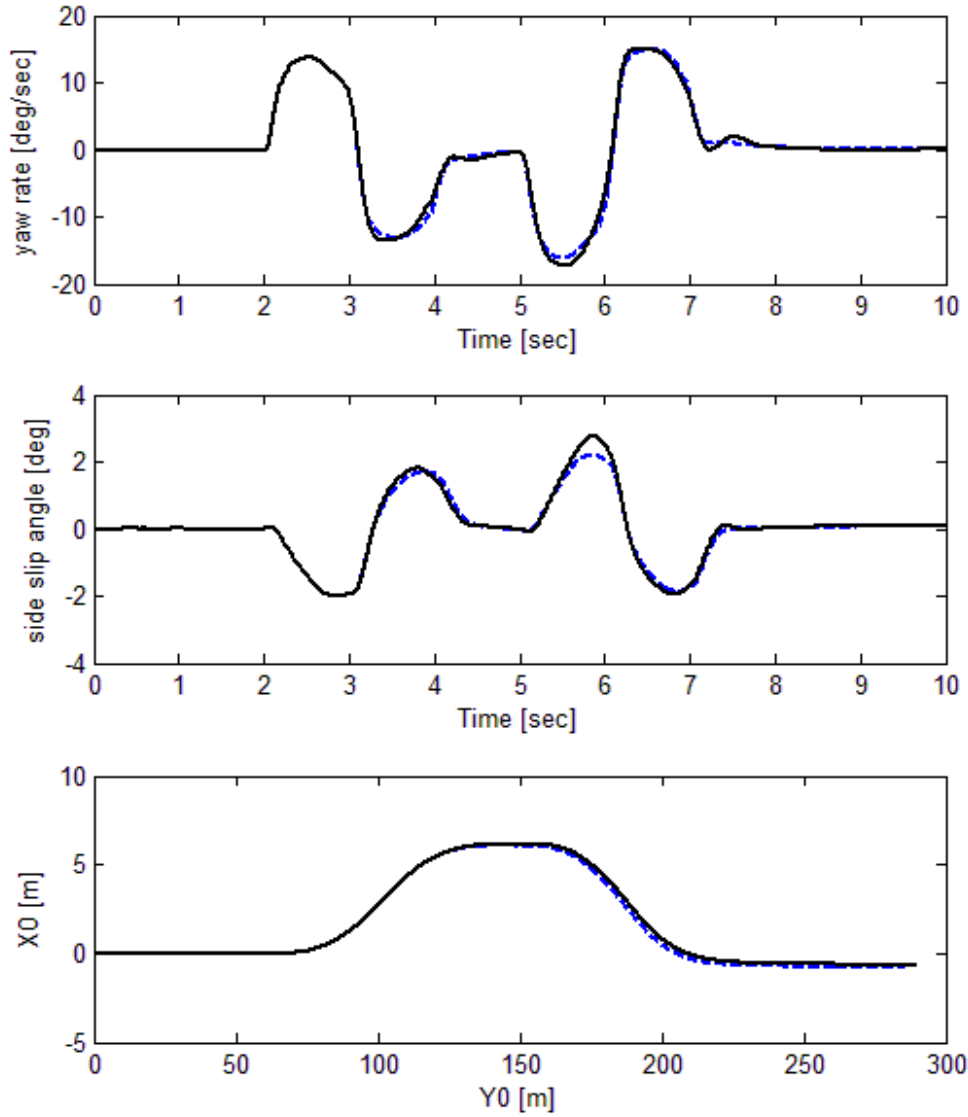


Figure 5.38 : The vehicle states in the actuator failure scenario (blue-dashed: in case of no actuator failure, black-continuous: in case of the front left and rear right braking actuator failures).

The results show that the proposed vehicle dynamics controller improves the stability and handling of the road vehicles and the actuators of active front steering and individual braking are coordinated in an optimal manner to generate the required stabilizing yaw moment. It was also shown that the proposed adaptive optimal control allocation method may compensate partial actuator failures and may assure an active fault tolerant vehicle dynamics control system.

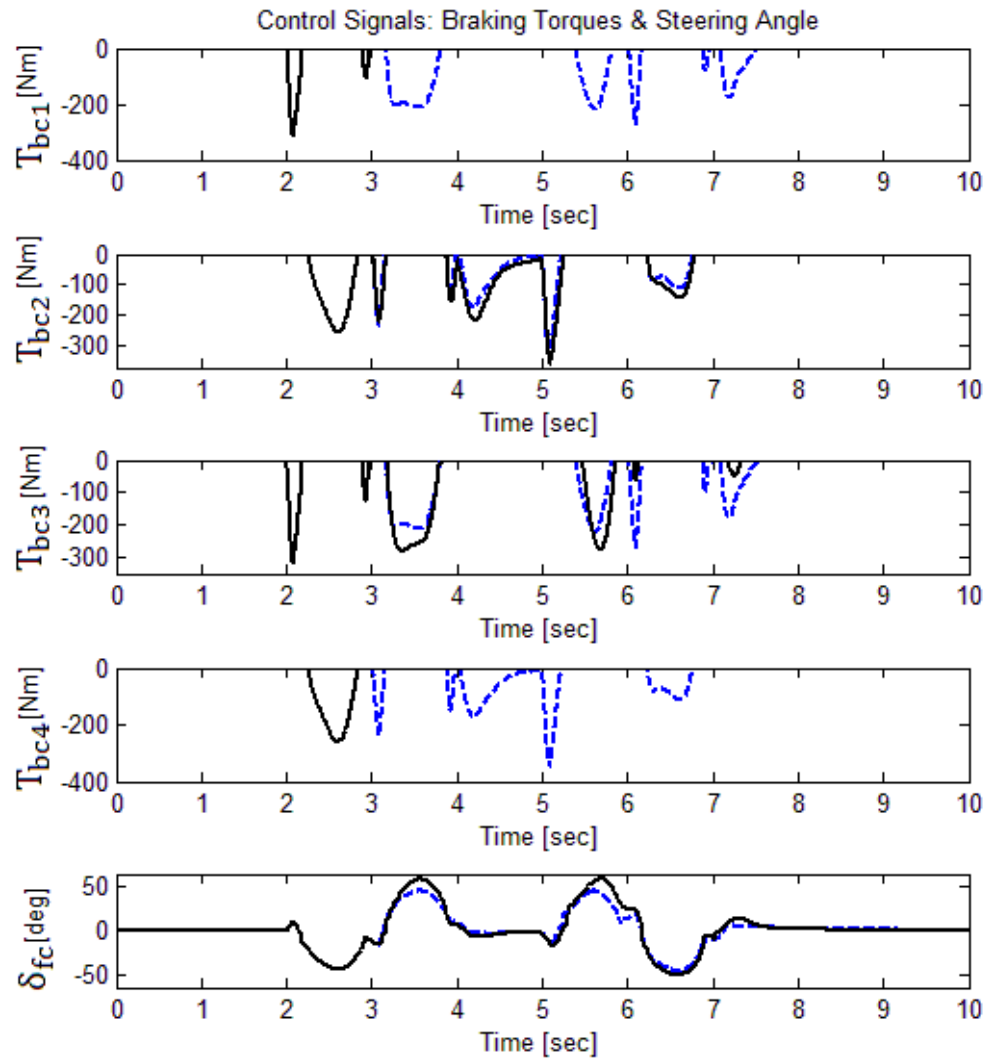


Figure 5.39 : The individual braking torques and corrective steering angle in the actuator failure scenario (blue-dashed: in case of no actuator failure, black-continuous: in case of the front left and rear right braking actuator failures)

6. REAL-TIME EXPERIMENTS

CarSim simulation software with its real-time expansion tool supports real-time hardware in-the-loop simulations for dSPACE embedded control systems. Thus, the proposed integrated stability controllers were implemented on a dSPACE MicroAutobox rapid prototyping system and real-time hardware-in-the-loop experiments in different vehicle maneuvers were carried out to see real time performances of the proposed control algorithms. The test environment are given in Figure 6.1.

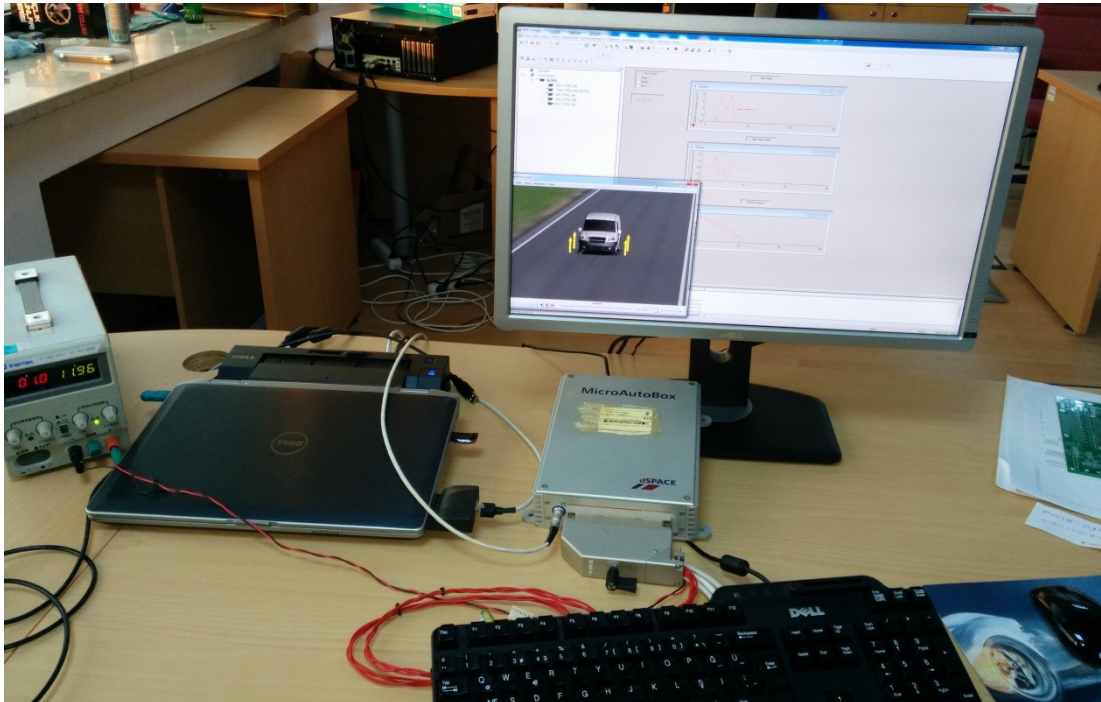


Figure 6.1 : Real-time test environment.

The embedded code for control algorithms developed in Simulink is generated automatically by using Real-Time Workshop toolbox and downloaded in to MicroAutoBox. On the other hand, the dynamics model of the Fiat Doblo Van run in CarSim RT environment. The control algorithms run real-time in MicroAutoBox and the data exchange between CarSim RT and MicroAutoBox is handled by a high speed dSPACE DS821 link board.

6.1 Real-Time Experiments For Stability Control of 4WS4WD Electric Vehicles

The proposed integrated vehicle stability control system for 4WS4WD electric vehicles was tested in real-time hardware-in-the-loop experiments of different vehicle maneuvers and scenarios. The stability control system and the vehicle powertrain is run in dSPACE MicroAutoBox while the high-degrees of freedom nonlinear Fiat Doblo Van model in CarSim RT (Figure 6.2). The design parameters of the sliding mode yaw moment observer given in (4.20) are $K_m = 30$, $D_m = 10$, $\Gamma = [1 \ 10]$ and the sampling time, T is 3ms. On the other hand, the design parameters of the adaptive optimal control allocation algorithm given in (4.29) are $k_{fxi} = k_{fyi} = -1$, $k_{\eta j} = 1 \times 10^4$ and $k_\lambda = 1 \times 10^3$.

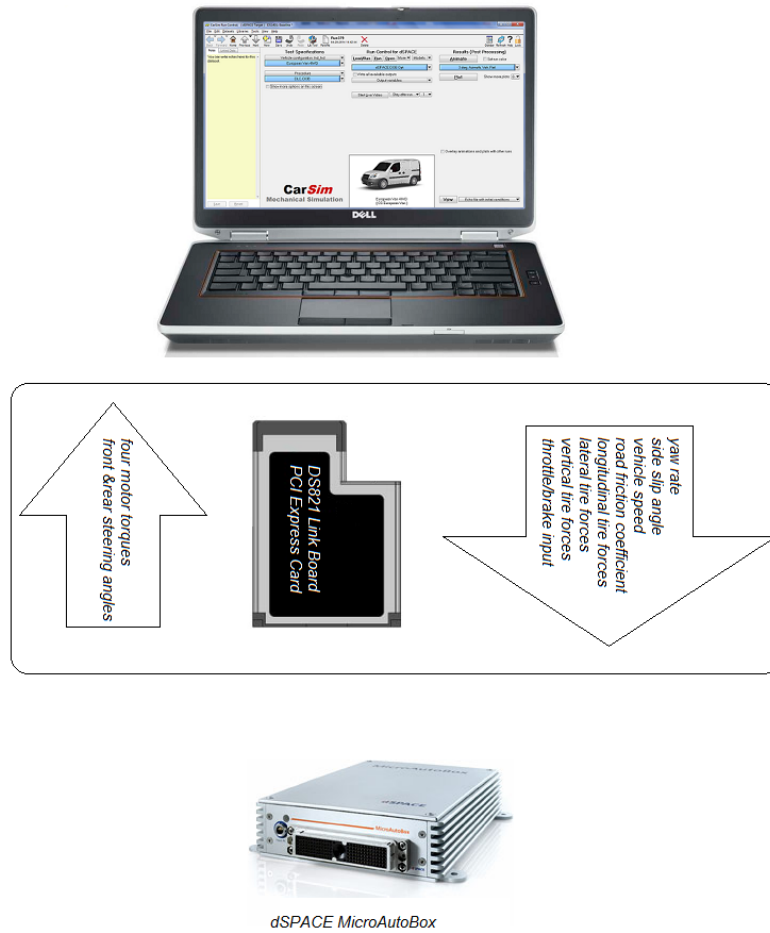


Figure 6.2 : HIL test diagram for the stability control of 4WS4WD electric vehicles.

6.1.1 Double lane change maneuver

The same double lane change maneuver given in simulation tests is repeated in this real-time experiment. During the DLC maneuver, a driver steering input given in Figure 5.2 and a fixed %20 throttle input were applied to the vehicle with initial speed of 130km/h on a wet road. The results for vehicle states and the actuator control signals are given in Figure 6.3 – 6.6.

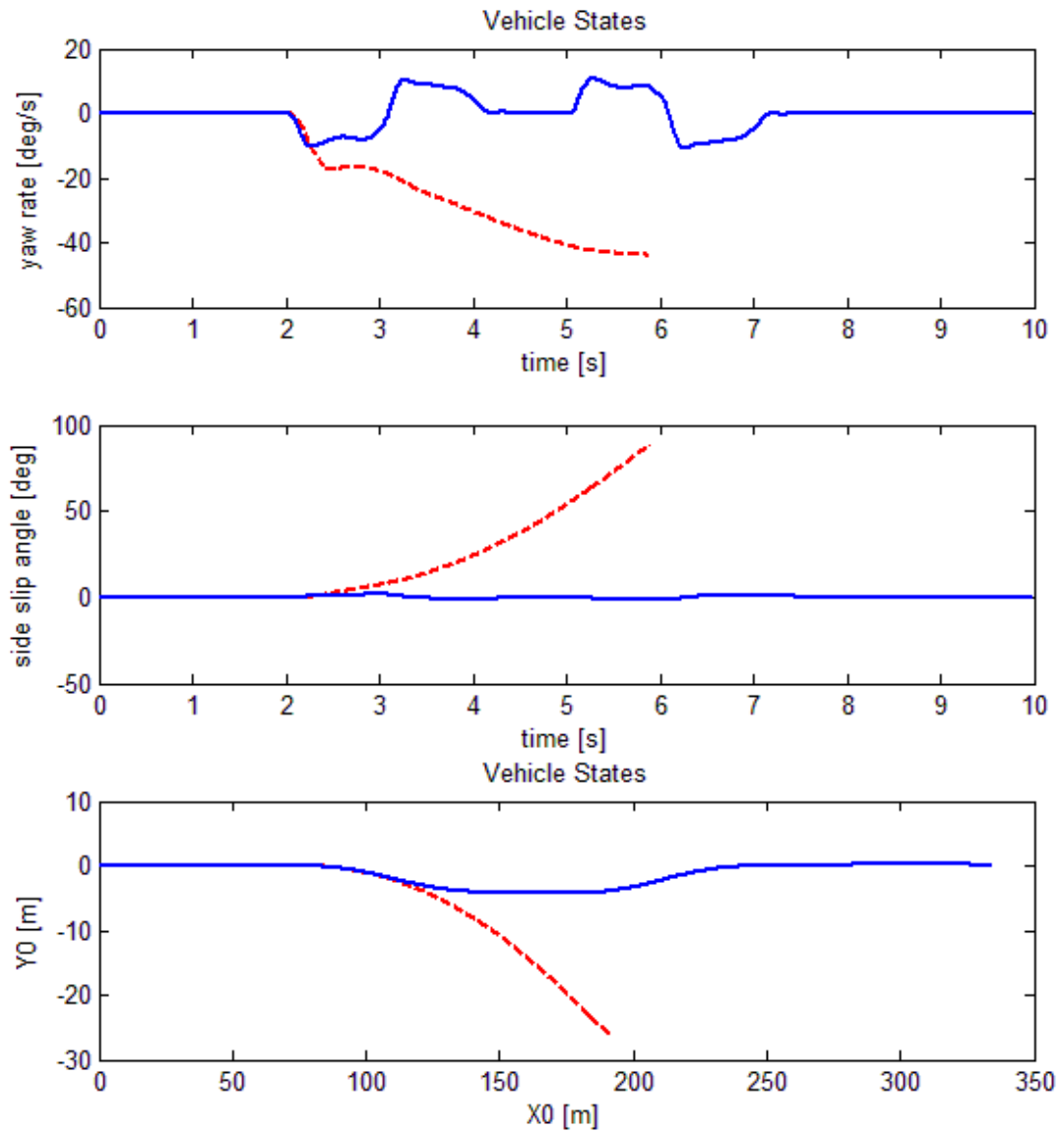


Figure 6.3 : Experimental results of vehicle states in the DLC maneuver (red-dot line: uncontrolled vehicle, blue-continuous line: control vehicle).

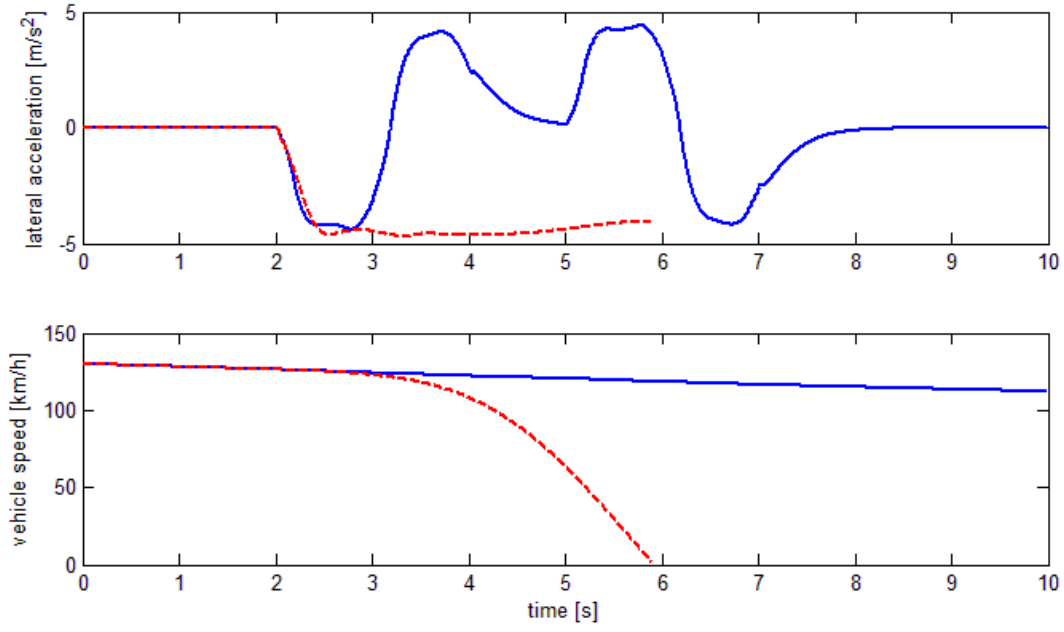


Figure 6.4 : Experimental results of vehicle states in the DLC maneuver (red-dot line: uncontrolled vehicle, blue-continuous line: control vehicle).

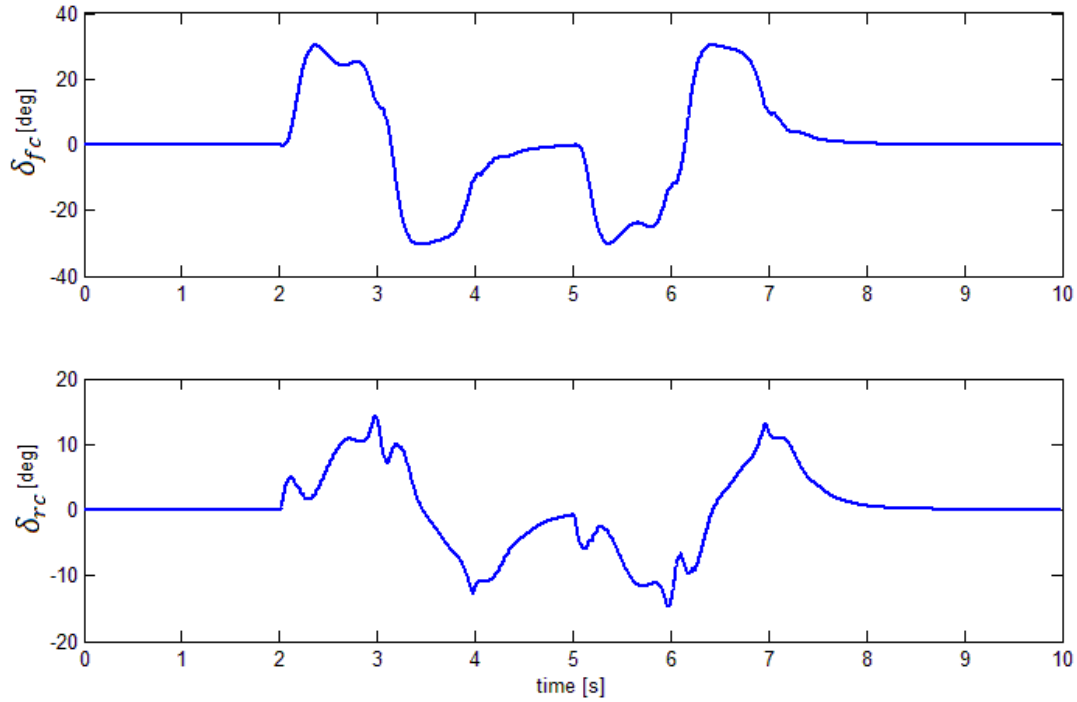


Figure 6.5 : The steering angles in the experiment of DLC maneuver.

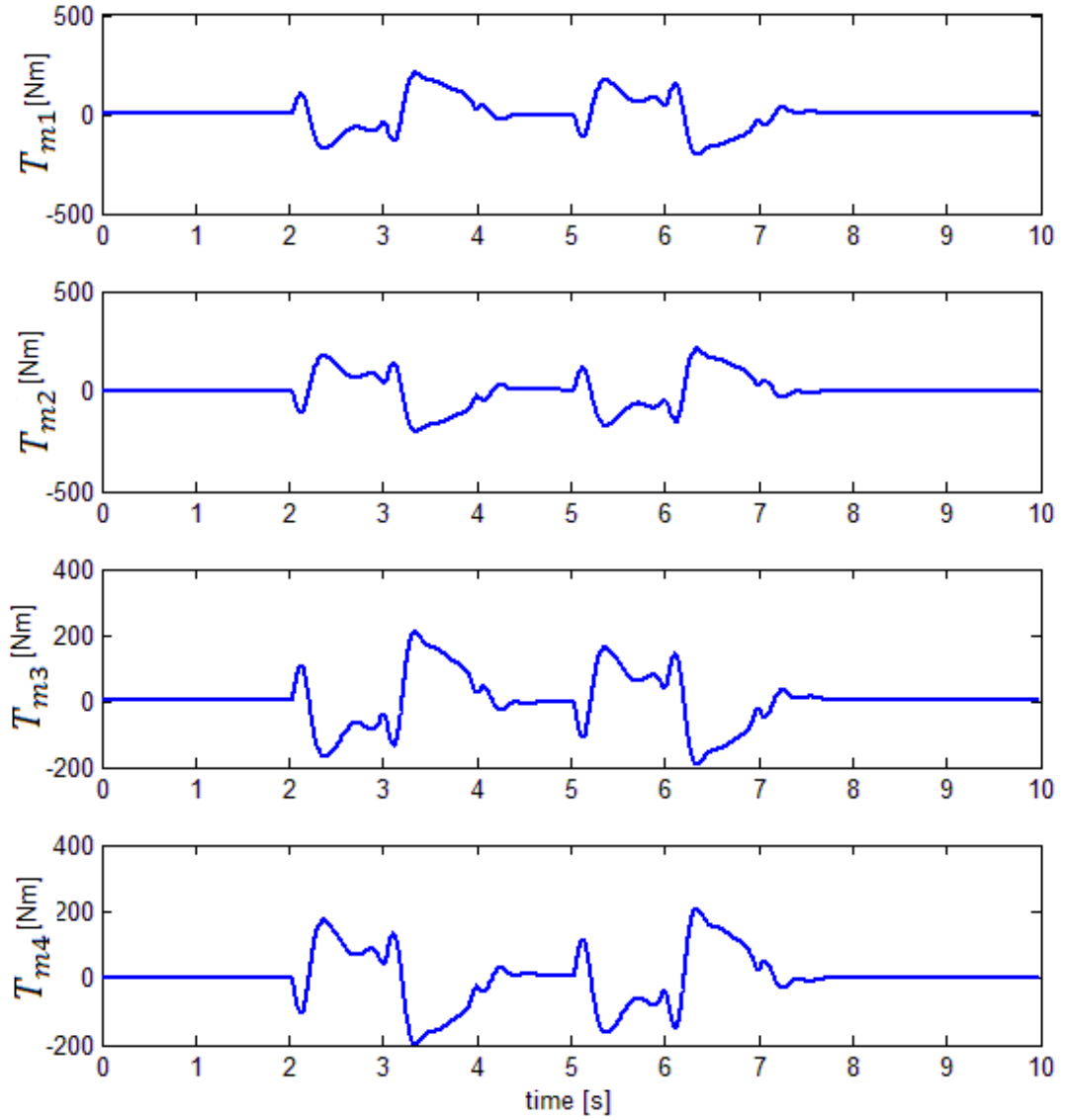


Figure 6.6 : The motor torques in the experiment of DLC maneuver.

6.1.2 Cross-wind disturbance test

The same cross-wind disturbance test given in simulation tests is repeated here. The vehicle enters in to the CWD test track given in Figure 5.6 with an initial speed of the vehicle is 120km/h and %20 throttle input was applied to the vehicle during the test. The vehicle states, the allocated four electric motor torques and the steering correction angles are given in Figure 6.7, Figure 6.8 and Figure 6.9 respectively.

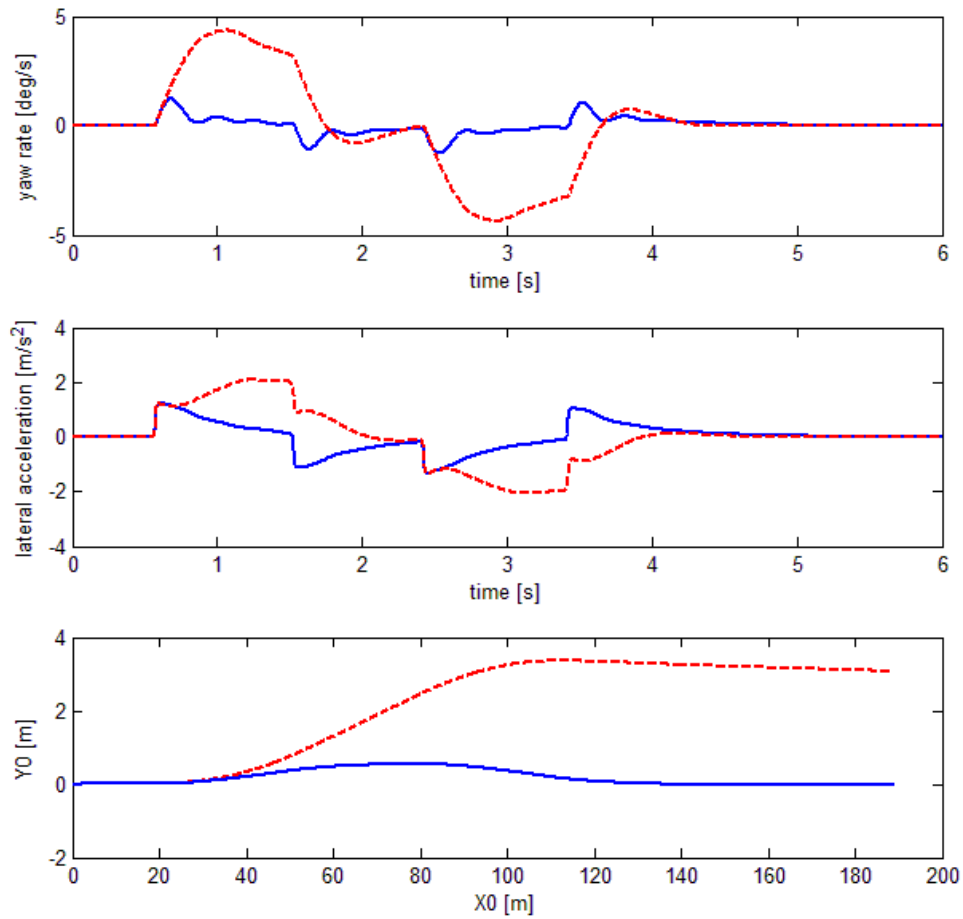


Figure 6.7 : Experimental results of vehicle states in the CWD maneuver (red-dot line: uncontrolled vehicle, blue-continuous line: control vehicle).

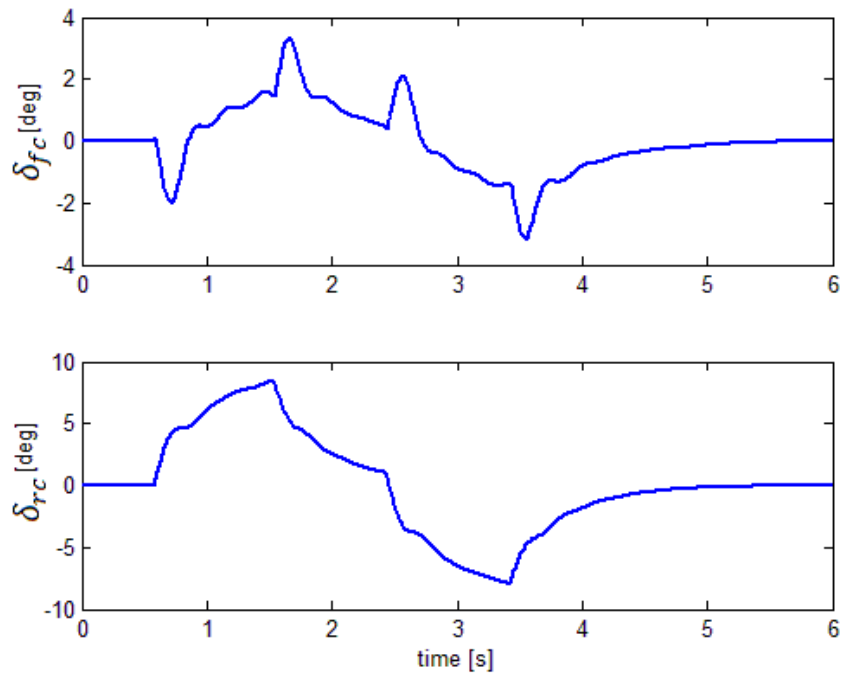


Figure 6.8 : The steering angles in the experiment of CWD maneuver.

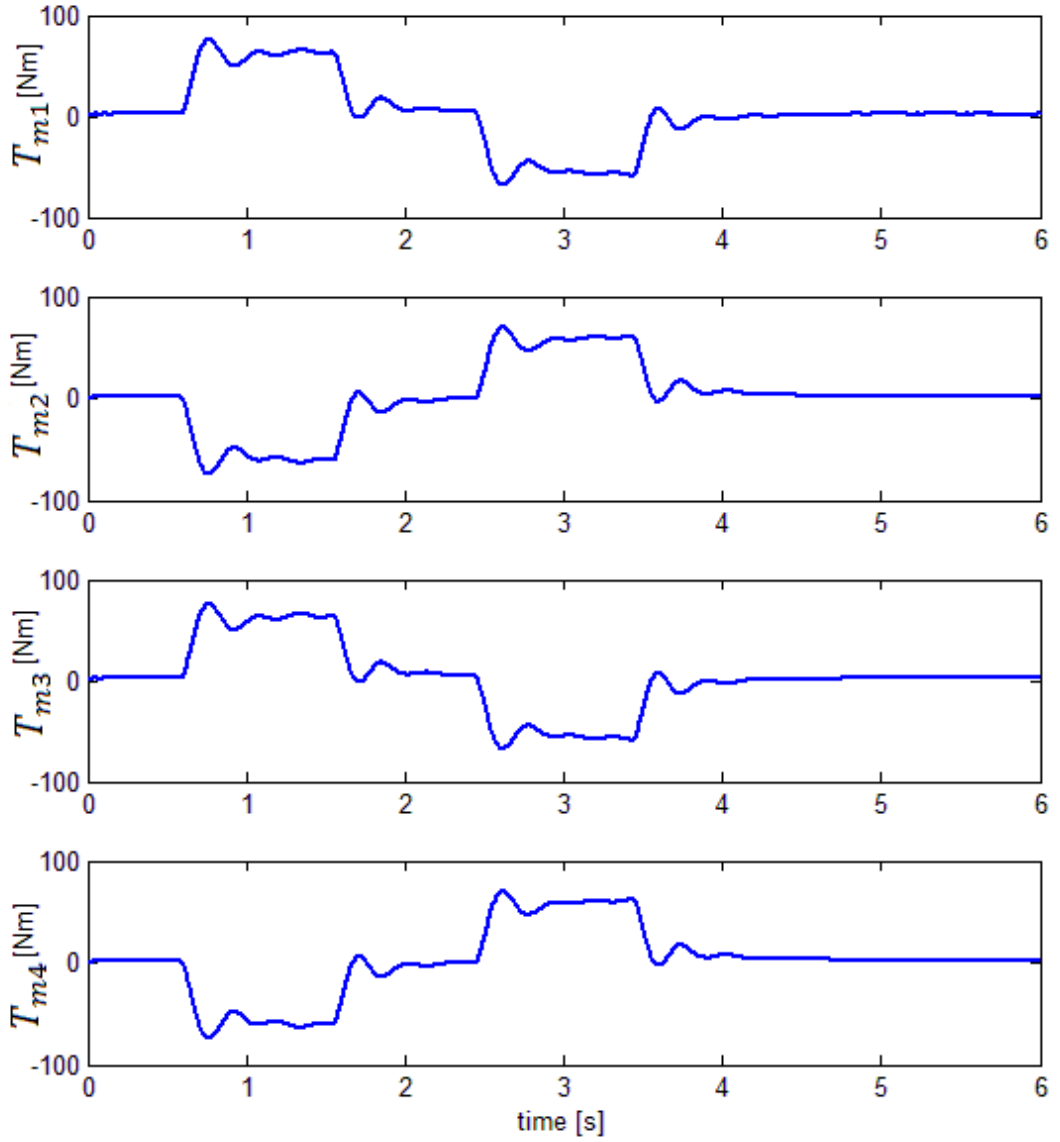


Figure 6.9 : The motor torques in the experiment of CWD maneuver.

6.1.3 Actuator Failure Scenario

In this experimental test, the above DLC maneuver is repeated but, in addition, an actuator failure scenario is considered such that the rear electric motors are assumed to be failed at third second of the maneuver. The aim is to evaluate the performance of the proposed adaptive optimal control allocation algorithm and its capability of compensation of the actuator failure in such an actuator failure. The results are shown in Figure 6.10, Figure 6.11 and Figure 6.12. In the figures, the blue-dashed

lines correspond to the case of normal operation while the black-continuous lines show the case of the failure of the rear electric motors.

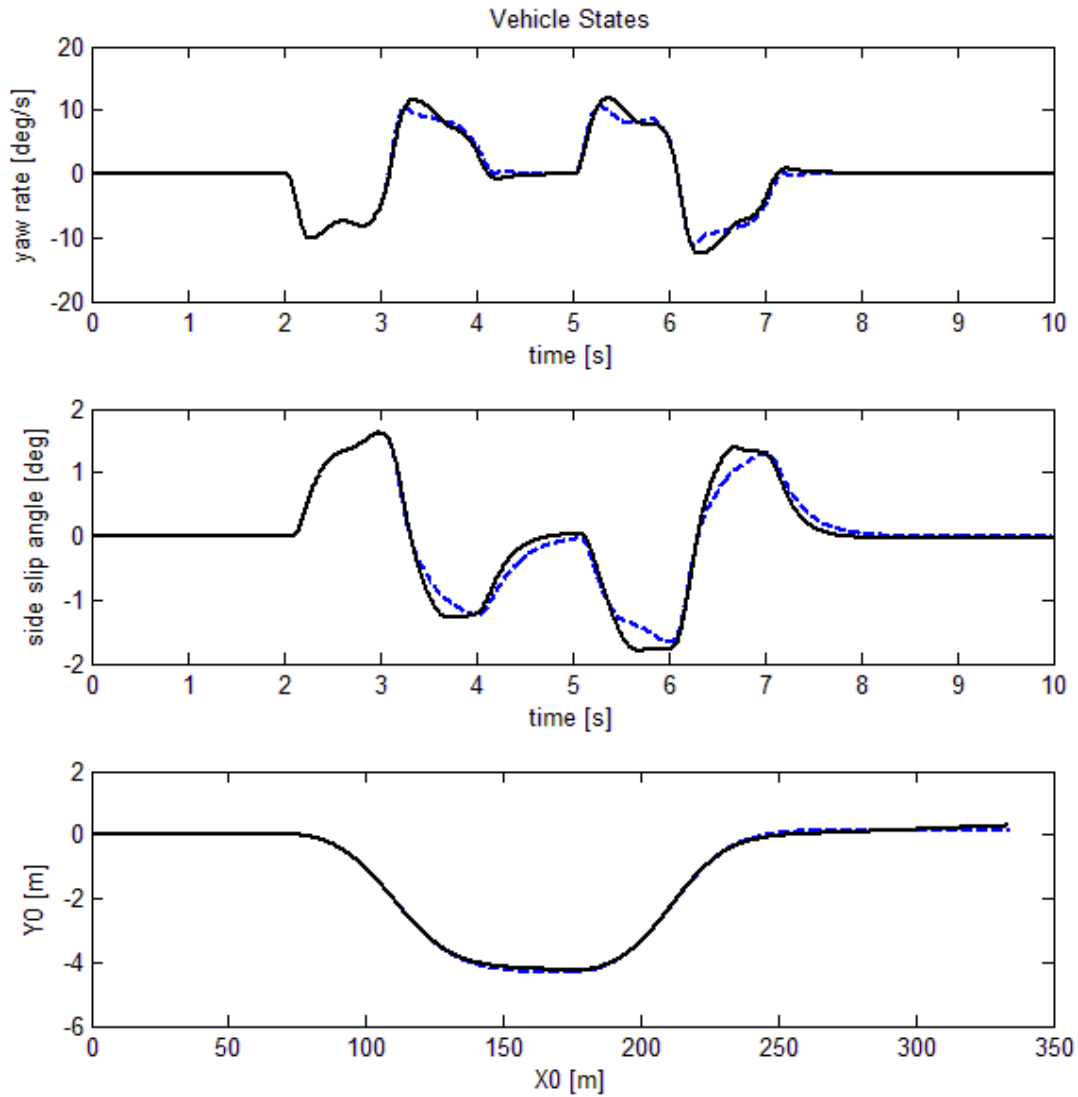


Figure 6.10 : Experimental results of vehicle states in the actuator failure scenario.

The real-time experiments validate the simulation results that the proposed integrated vehicle stability controller improves the stability of the 4WS4WD electric vehicle. The sliding mode yaw moment observer successfully estimates and cancels the modeling and parameter uncertainties and external disturbances. On the other hand, the proposed adaptive optimal control allocation algorithm successfully allocates the actuator control signals in an optimal manner. Moreover, it is also validated in real-time experiments that the proposed adaptive optimal control allocation method may compensate partial actuator failures in the stability control of 4WS4WD electric vehicles.

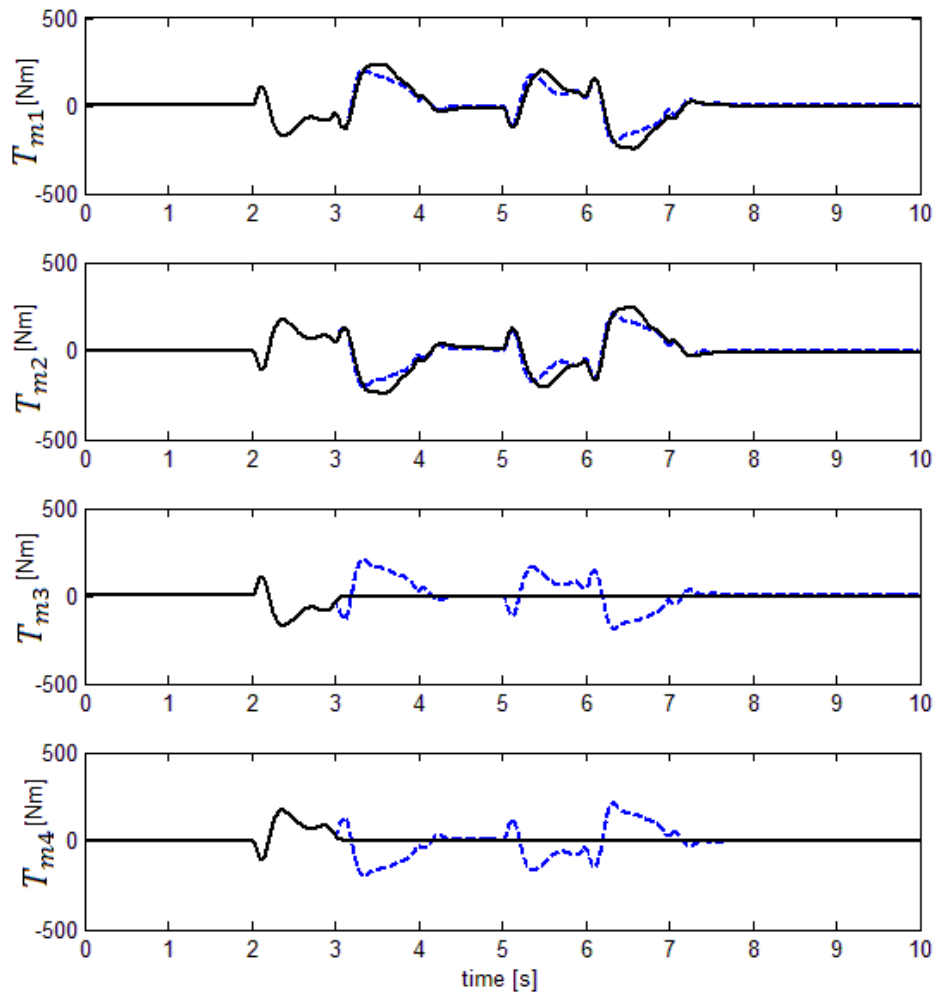


Figure 6.11 : The motor torques in the actuator failure scenario.

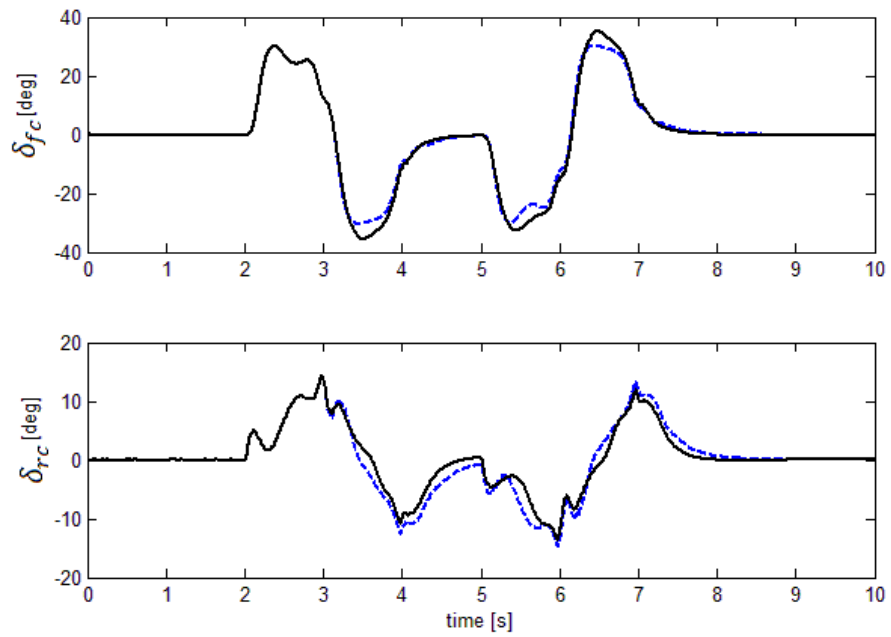


Figure 6.12 : The steering angles in the actuator failure scenario.

6.2 Real-Time Experiments For Integrated Stability Control By Coordination of AFS and Individual Brakes

In this section, it is aimed to implement the proposed integrated vehicle stability control system based on active front steering and individual braking actuators on a real control hardware and to test the performance of it in real-time hardware-in-the-loop experiments. The stability control system and the vehicle powertrain is run in dSPACE MicroAutoBox while the vehicle model in CarSim RT (Figure 6.13).

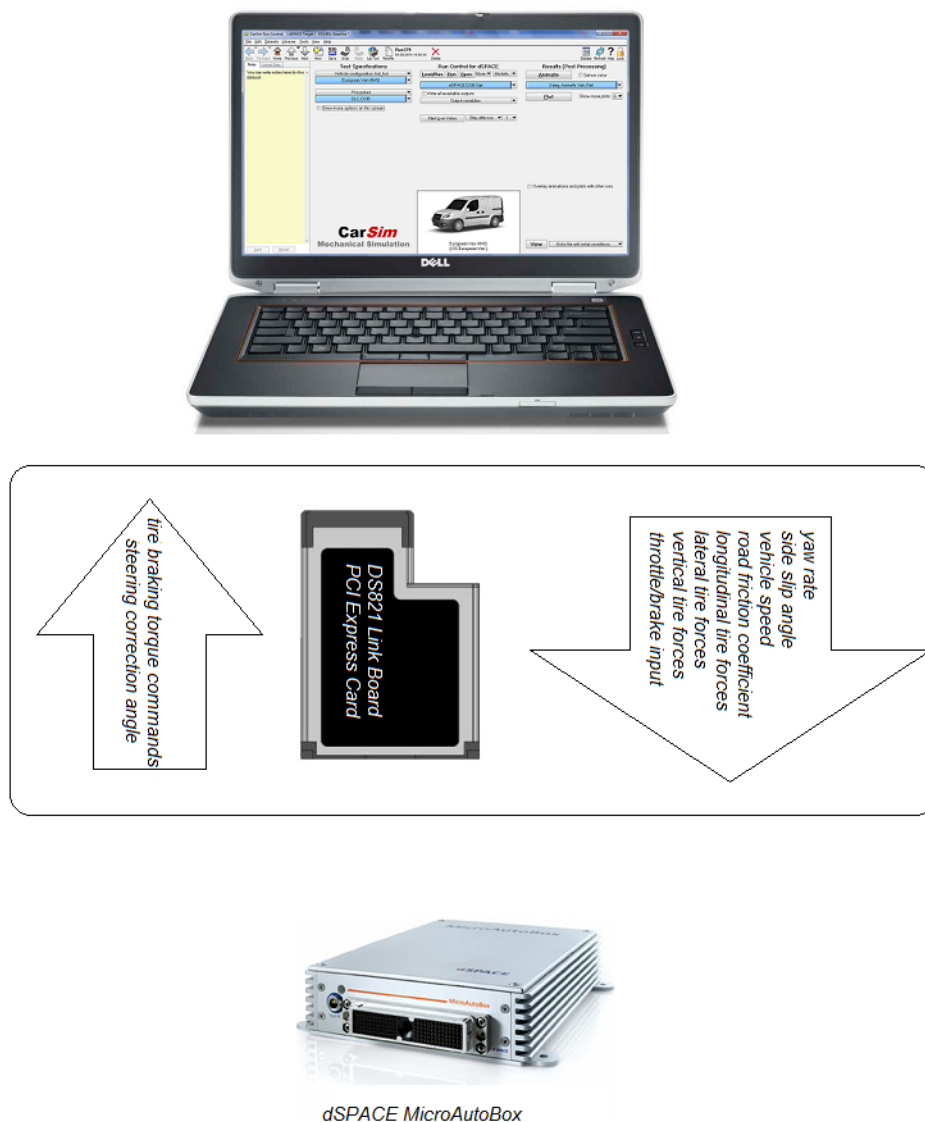


Figure 6.13 : HIL test diagram for the stability control based on AFS and individual braking actuators.

The vehicle is the Fiat Doblo Van used in the previous simulation tests. The design parameters of the model reference sliding mode controller given in (4.35) and (4.36) are the sampling time, T is 3ms, the filter constant, $\tau = T = 3ms$ and $k_\sigma = 1 \times 10^6$. On the other hand, the design parameters of the adaptive optimal control allocation algorithm given in (4.38) are $\Gamma_{f_{xi}} = \Gamma_{f_{yj}} = -0.5$, $\Gamma_{\eta k} = 1 \times 10^3$ and $k_\lambda = 1 \times 10^4$.

6.2.1 Double lane change maneuver

During the DLC maneuver, a driver steering input given in Figure 5.2 was applied to the vehicle with initial speed of 120km/h on a wet road. The results for vehicle states and the actuator control signals are given in Figure 6.14 – 6.18.

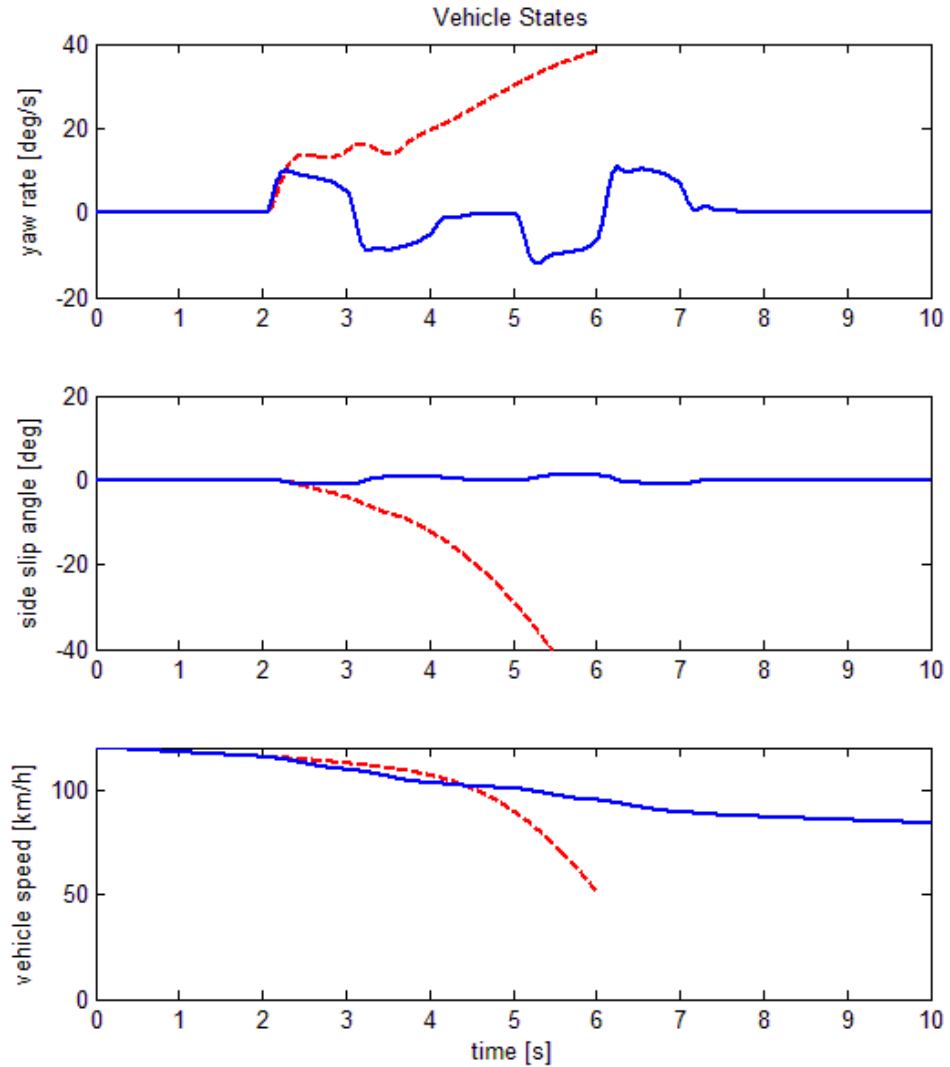


Figure 6.14 : Experimental results of vehicle states in the DLC maneuver (red-dot line: uncontrolled vehicle, blue-continuous line: control vehicle).

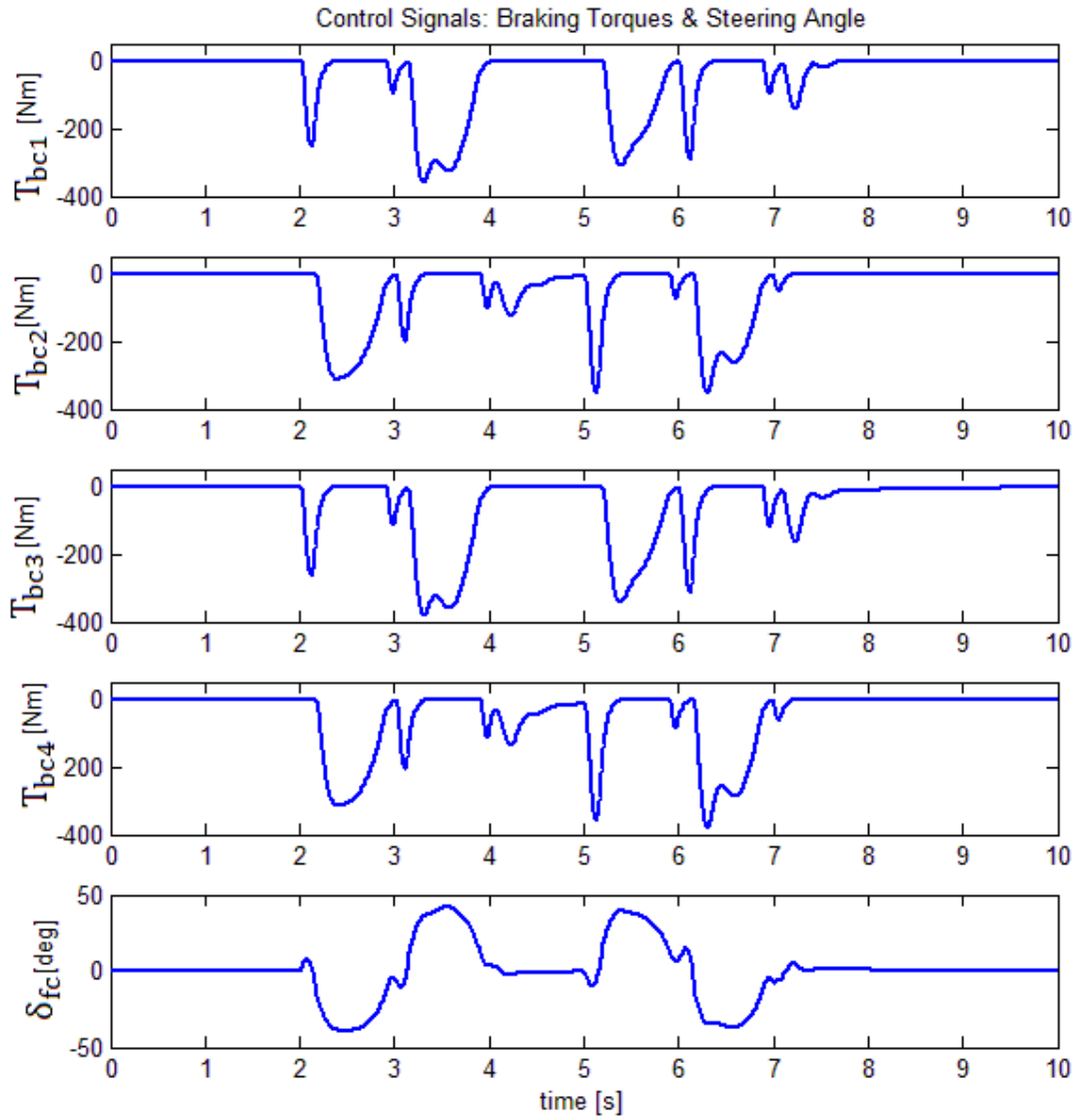


Figure 6.15 : The braking torques and steering correction angle in the experiment of DLC maneuver.

6.2.2 Cross-wind disturbance test

The aim in the cross-wind disturbance (CWD) test is to evaluate real-time performance and robustness of the proposed integrated stability control system against external yaw disturbances. The vehicle enters in to the CWD test track given in Figure 5.6 with an initial speed of 100km/h. The experimental results for the vehicle states are given in Figure 6.16.

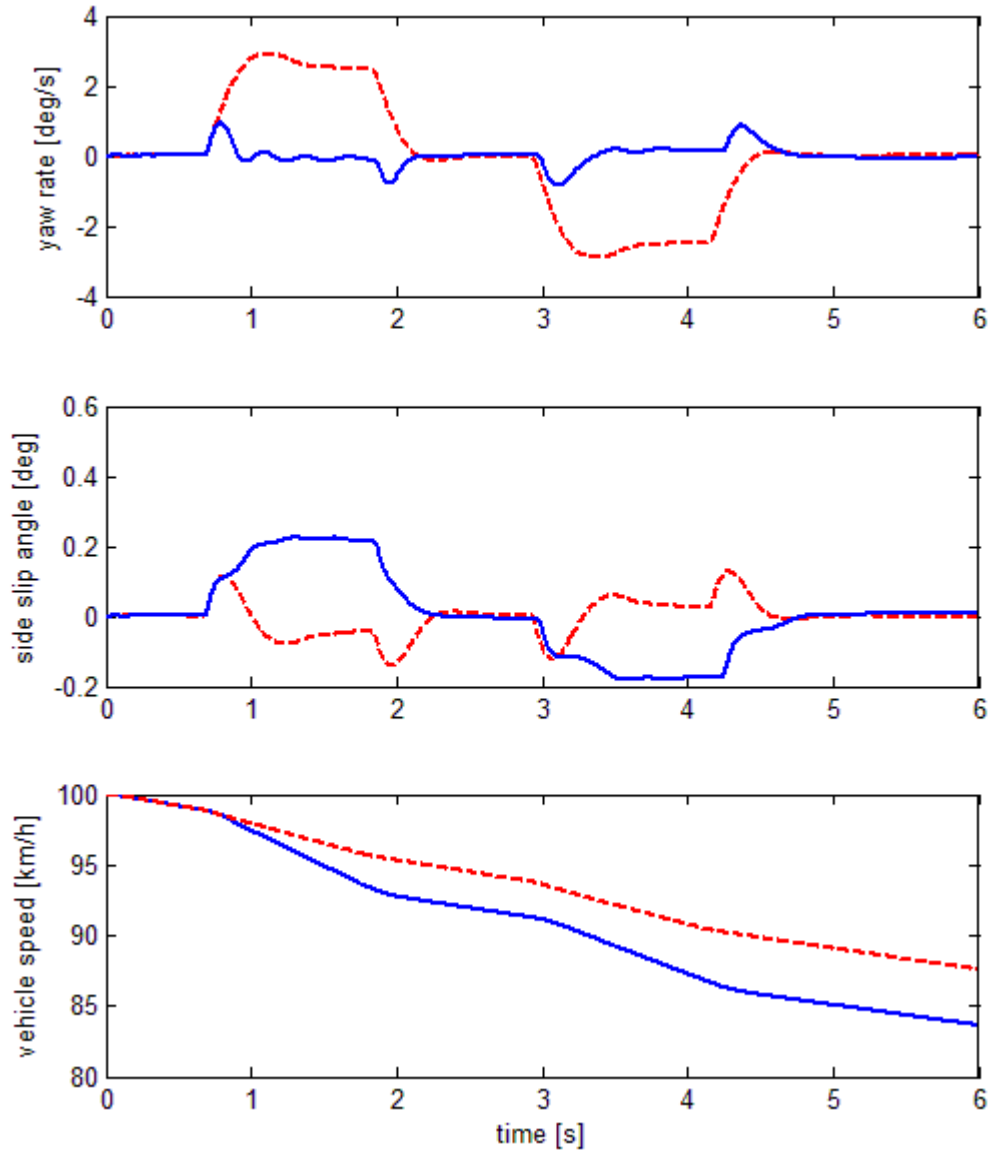


Figure 6.16 : Experimental results of vehicle states in the CWD test track (red-dot line: uncontrolled vehicle, blue-continuous line: control vehicle).

6.2.3 Actuator failure scenario

In this maneuver, fault compensation property of the proposed adaptive optimal control allocation algorithm is tested. Two different actuator failure scenarios are considered. The front braking actuators fail in one case while the rear braking actuators in the other case. The actuator failures are tested in the DLC maneuver given above. Hence, the DLC maneuver is repeated here and it is assumed that the actuators are failed at third second of the maneuver. The experimental results are shown in Figure 6.17 and Figure 6.18. In the figures, the blue-dashed lines correspond to the case of normal operation. While the black-continuous lines show the

case of the failure of the front braking actuators, the red-dash-dotted lines correspond to the case of the failure of the rear braking actuators.

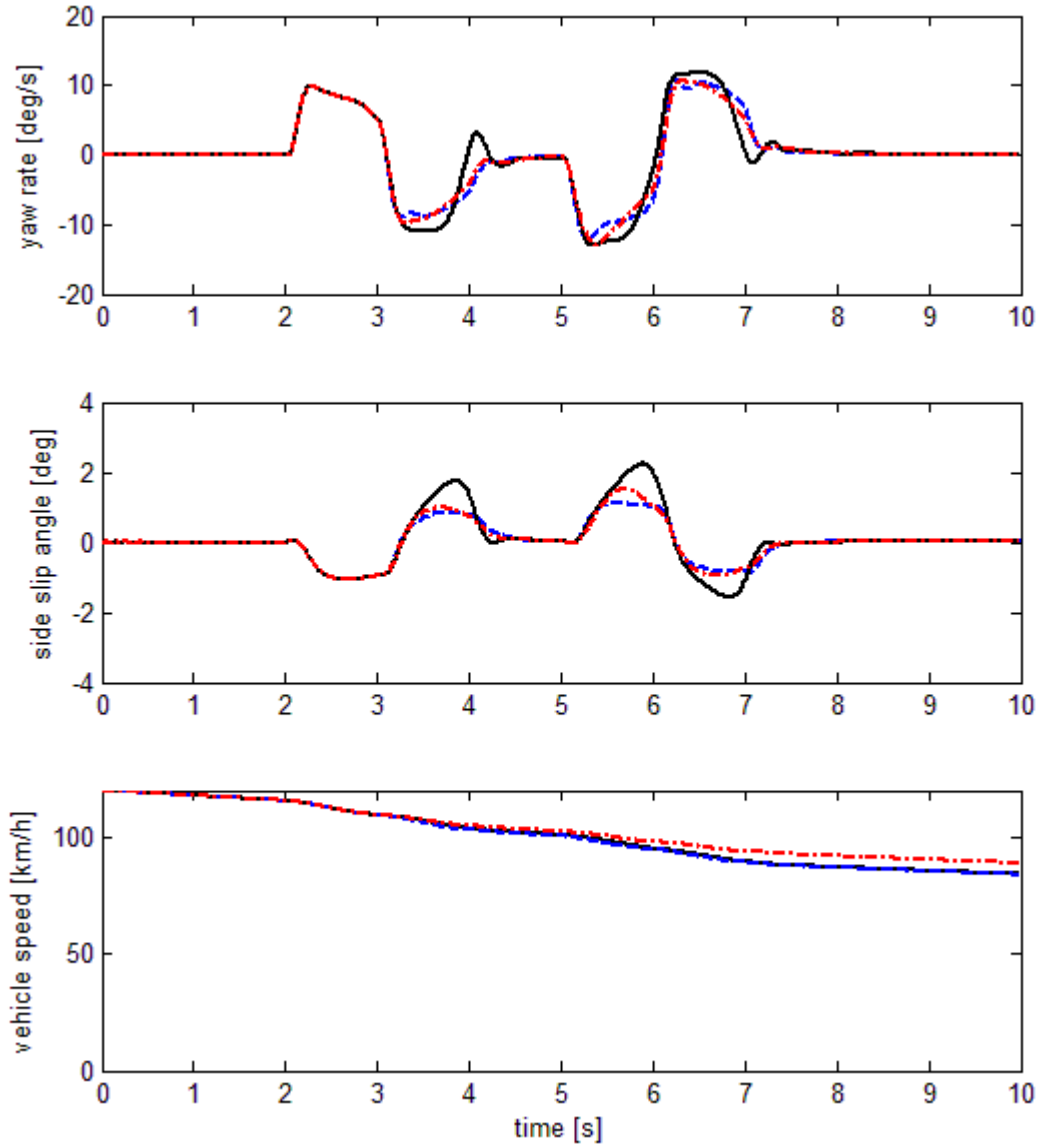


Figure 6.17 : The vehicle states in the braking actuator failure scenarios.

The real-time experiments validate the simulation results that the proposed integrated vehicle stability controller improves the stability of the vehicle. The model reference sliding mode controller successfully tracks the reference vehicle dynamics. The individual braking and front active steering actuators are optimally coordinated by the proposed adaptive optimal control allocation algorithm. It is also validated in real-time experiments that the proposed adaptive optimal control allocation method may compensate partial actuator failures and may assure an active fault tolerant vehicle dynamics control system.

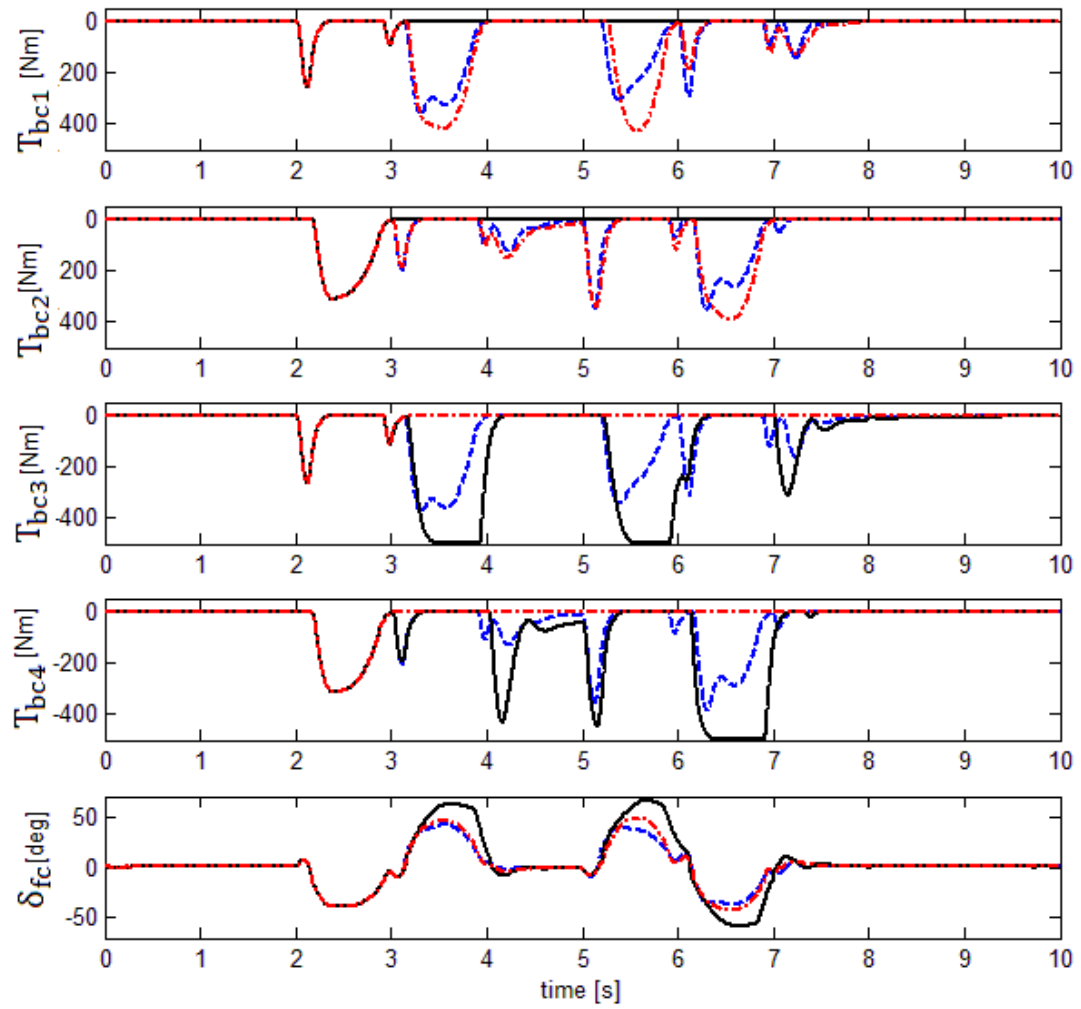


Figure 6.18 : The individual braking torques and steering correction angle in the actuator failure scenarios.

7. CONCLUSIONS

This thesis proposes a novel adaptive optimal control allocation method for coordination of the actuators in dynamics control of over-actuated vehicles in addition to sliding mode disturbance observers for yaw moment control and steering control problems in vehicle dynamics control. Moreover, a sliding mode MIMO disturbance observer is also introduced and applied to the decoupled control problem of four wheel active steering vehicles.

The disturbance observer approaches for vehicle steering control are based on sliding mode control theory. The controllers estimate the equivalent input extended-disturbance in the control input channel. The extended-disturbance includes modeling and parameter uncertainties of the vehicle in addition to external yaw disturbances caused by side wind, μ -split braking, etc. The aim in the disturbance observer based steering control is to force the vehicle to track a 2DOF reference vehicle dynamics behavior by estimating the equivalent input extended-disturbance which is the front tire steering correction angle for active steering system. The steering controllers are tested on a Fiat Doblo Van modelled in CarSim environment. Satisfactory reference tracking performances are achieved with the proposed control approaches during the tests. The stability of the vehicle are improved in extreme driving maneuvers and good disturbance rejection performances are achieved against external yaw disturbances.

The performances of the proposed sliding mode MIMO disturbance observer for input-output decoupling are validated by nonlinear simulations of a realistic vehicle dynamics modelled in CarSim environment. Excellent performance results are achieved such that the yaw rate only responds to the front wheel steering angle input while the vehicle side slip angle only responds to the rear wheel steering angle input. The couplings from the front wheel steering angle to the vehicle side slip angle and from the rear wheel steering angle to the yaw rate are attenuated successfully. With inclusion of the proposed MIMO disturbance observer, it is possible to design

independent servo controllers by only considering nominal plant dynamics in the outer loop for the lateral and yaw motion references related to lane keeping control, collision avoidance, stability and handling control, etc. With such a decoupled vehicle dynamics, it is possible to assign any arbitrary yaw rate and side slip angle references. A zero lateral speed can easily be achieved while referencing yaw rate, which yields improved comfort and maneuverability and stable cornering maneuvers. Moreover, a zero yaw rate for a stable obstacle avoidance maneuver or lane keeping can be obtained with a simple outer linear servo control loop.

Coordination term in vehicle dynamics control refers to distribution of a required yaw moment among available actuators of a vehicle, e.g. front and rear wheel active steering actuators, mechanical braking actuators, electric motors, etc. and allocation of the actuators in an optimal manner to generate the required yaw moment. An efficient, convergent, optimal and stability guaranteed control allocation algorithm is very crucial in vehicle dynamics control for vehicle safety. The algorithm should include vehicle dynamic limitations (tyre friction circle limitation, road adhesion, etc.) and actuator limitations.

If the control allocation is a nonlinear optimization problem, it is difficult to find an analytical solution and static-iterative optimization methods are mostly used in the literature. However, the stability of an optimal control allocation method should be guaranteed in a vehicle safety system and there is no an explicit way to show the stability of a static-iterative optimization method. Moreover, the static optimization based control allocation requires the nonlinear optimization problem to be solved in every time step for every corresponding yaw moment reference and this requires a high computational resource in real-time implementation.

As a solution to the above mentioned issues related to the static optimization based control allocation methods, an adaptive optimal control allocation method based on Lagrangian neural-networks is proposed in the thesis. The proposed control allocation method avoids the requirement of solving the optimization problem explicitly in every time step and the convergence and stability of the control allocation are guaranteed.

The proposed sliding mode yaw moment observers and adaptive optimal control allocation method are applied to develop novel integrated vehicle stability control systems for four wheel steer four wheel drive (4WS4WD) electric vehicles and

conventional road vehicles. The integrated vehicle stability controllers have a hierarchical control structure in which three layers called upper layer, middle layer and lower layer exist. The upper layer includes the proposed sliding mode yaw moment controllers whose output is virtual yaw moment control input to be generated by the middle and lower layers. The middle layer deals with control allocation by using the proposed adaptive optimal control allocation method and the lower layer includes servo control loops for tracking of reference actuator commands.

In the proposed stability control system for 4WS4WD electric vehicles, the adaptive optimal control allocation algorithm is used to coordinate the four wheel steering actuators and four in-wheel electric motors. On the other hand, in the stability control system developed for conventional road vehicles, it is used to coordinate the active front steering and individual braking actuators. Optimal electrical motor torques, steering correction angles and individual braking torques are calculated by solving a nonlinear control allocation problem which includes a yaw moment equality constraint, the actuator and tire force inequality constraints and a cost function of adaptive weighted square sum of the tire forces. The proposed stability control systems are tested on a Fiat Doblo Van by simulation and real-time experiments. While the vehicle dynamics are modelled and simulated in CarSim simulation software, the control systems in Matlab/Simulink. The real-time experiments are carried out on a dSPACE MicroAutoBox in hardware-in-the-loop simulations with CarSim RT. The simulation and experimental results show that the proposed control allocation method successfully allocates the actuators in an optimal manner and it is concluded that the proposed stability control systems improve the stability and handling of road vehicles even in critical driving maneuvers.

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APPENDICES

APPENDIX A: Existence and Derivation of the Equivalent Input Extended Disturbance (EIED)

APPENDIX B: Vehicle Parameters

APPENDIX C: Stability Proofs of the Adaptive Optimal Tire Force Distribution

APPENDIX A: Existence and Derivation of the Equivalent Input Extended Disturbance (EIED)

The extended-disturbance for the vehicle steering control problem was defined in (3.13) as follows,

$$N = (A - A_d)x + (B_\delta - B_{\delta d})\delta_f + \varrho \quad (\text{A.1})$$

By substituting the definitions in (3.7) and (3.12) in to (A.1) yields

$$N = \begin{bmatrix} N_1 \\ N_2 \end{bmatrix} = \begin{bmatrix} a_{11} + \frac{1}{\tau_d} & a_{12} \\ a_{21} & a_{22} + \frac{1}{\tau_d} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} b_{11} - \frac{K_\beta}{\tau_d} \\ b_{21} - \frac{K_r}{\tau_d} \end{bmatrix} \delta_f + \begin{bmatrix} \varrho_1 \\ \varrho_2 \end{bmatrix} \quad (\text{A.2})$$

After rearranging the terms,

$$N = \begin{bmatrix} N_1 \\ N_2 \end{bmatrix} = \begin{bmatrix} (a_{11} + \frac{1}{\tau_d})\beta + a_{12}r + (b_{11} - \frac{K_\beta}{\tau_d})\delta_f + \varrho_1 \\ a_{21}\beta + (a_{22} + \frac{1}{\tau_d})r + (b_{21} - \frac{K_r}{\tau_d})\delta_f + \varrho_2 \end{bmatrix} \quad (\text{A.3})$$

It is shown by She et al. (2008) that if $(A_d, B_{\delta d}, C)$ is observable and controllable and if $(A_d, B_{\delta d}, C)$ has no zeros on the imaginary axis, then there always exist an EIED in the control input channel. Thus,

$$N = B_{\delta d}v \quad (\text{A.4})$$

and the vehicle dynamics can be written as in (3.11).

The aim in the sliding mode disturbance observer based stability control systems proposed in the thesis is to track the two degrees of freedom reference vehicle

dynamics by estimating the EIED in the control input channel and compensating the extended disturbances, N_1 and N_2 for improvement of stability and handling of the vehicle. Since the considered system is a single input two output system, it is not possible to exactly compensate the extended disturbances on both degrees of freedom. Thus, it is aimed to fully compensate the extended disturbance on the yaw axis while reducing the extended disturbance in the side slip angle dynamic. This is meaningful since the main goal during vehicle maneuvers is to regulate yaw rate dynamic while keeping the side slip angle near to zero.

We focus on the effect of the disturbances on the output rather than the disturbances themselves but, it will be good how well the proposed disturbance observer approaches are. Hence, an application of the proposed vehicle steering controllers on Fiat Doblo Van modelled in CarSim is considered. The test maneuver is a double lane change maneuver with an initial speed of 120km/h on a dry road surface and the driver steering input as shown in Figure 5.2. The sliding mode disturbance observer (SMDO) and the model reference sliding mode control with uncertainty and disturbance estimation (SMEDE) approaches for vehicle steering control are simulated. The vehicle states and the extended disturbances are given in Figure A.3 and A.4 for the SMDO and in Figure A.5 and A.6 for the SMEDE respectively. It can be seen that the extended-disturbance in the yaw rate dynamic, N_2 is successfully estimated and compensated while reducing the extended-disturbance, N_1 in the side slip angle dynamic.

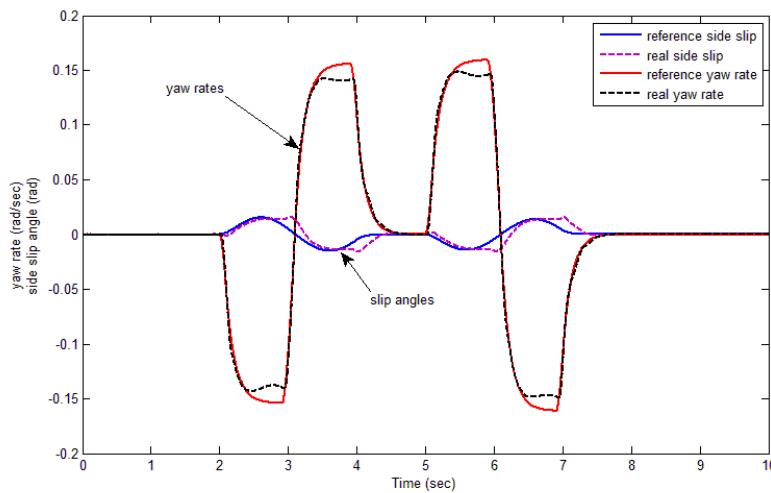


Figure A.1 : The vehicle states for the SMDO.

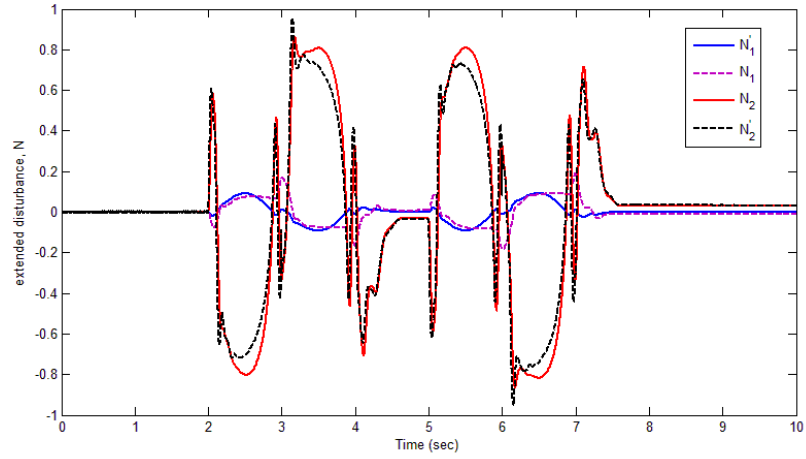


Figure A.2 : The estimated and real extended disturbances for the SMDO.

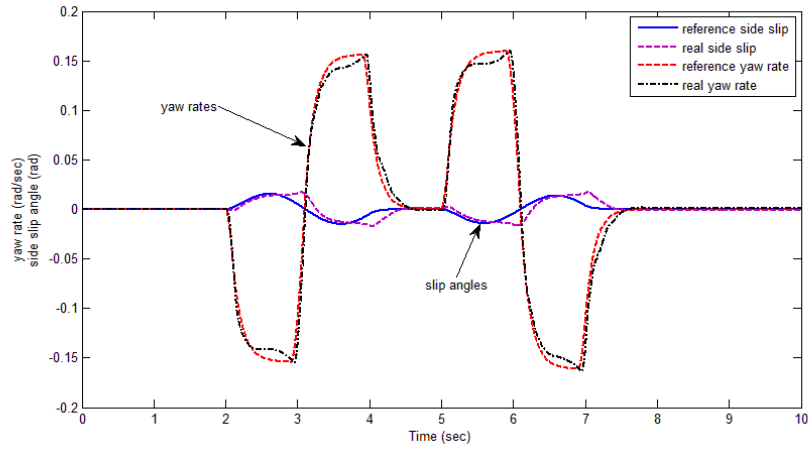


Figure A.3 : The vehicle states for the SMEDE.

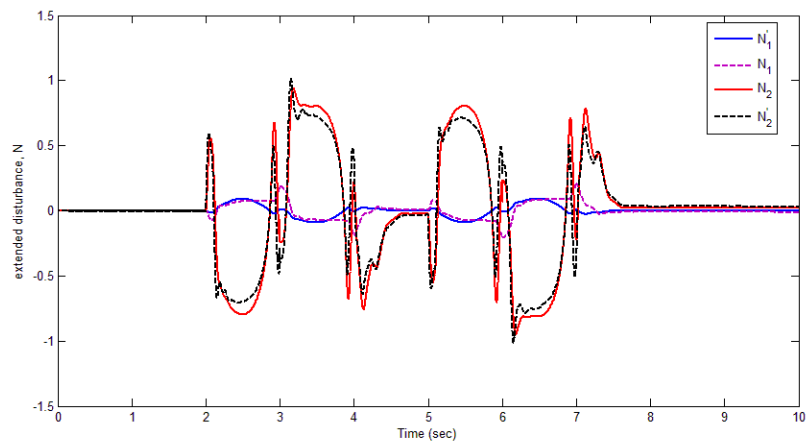


Figure A.4 : The estimated and real extended disturbances for the SMEDE.

APPENDIX B: Vehicle Parameters

The nominal parameters of the Fiat Doblo Van used in the simulations are given in Table B.1.

Table B.1: Vehicle Parameters.

Parameter	Definition	Value
l_{fn}	Distance from COG to front axle	1.1350m
l_{rn}	Distance from COG to rear axle	1.440m
m_n	Vehicle mass	1500kg
I_{zn}	Moment of inertia about z-axis	2975kgm ²
C_{fn}	Front tire cornering stiffness in single-track vehicle model	63369N/rad
C_{rn}	Rear tire cornering stiffness in single-track vehicle model	78610N/rad
h_{cgn}	Height of COG	0.711m
R	Tire radius	0.292m
d	Track width	1.5m

Nominal values of the vehicle parameters, a_{ij} and b_{ij} are used for calculation steady-state gains, K_β and K_r given in (3.5) and the time constant, τ_d given in (3.7). Moreover, a vehicle speed dependent implementation of the reference vehicle dynamics is also considered in the reference vehicle dynamics. Thus, the parameters are

$$\begin{aligned}
 a_{11n}(V_{COG}) &= -\frac{2(C_{fn}+C_{rn})}{m_n V_{COG}} & a_{12n}(V_{COG}) &= -1 - \frac{2(C_{fn}l_{fn}-C_{rn}l_{rn})}{m_n V_{COG}^2} \\
 a_{21n}(V_{COG}) &= -\frac{2(C_{fn}l_{fn}-C_{rn}l_{rn})}{I_{zn}} & a_{22n}(V_{COG}) &= -\frac{2(C_{fn}l_{fn}^2+C_{rn}l_{rn}^2)}{I_{zn} V_{COG}} \\
 b_{11n}(V_{COG}) &= \frac{2C_{fn}}{m_n V_{COG}} & b_{12n}(V_{COG}) &= \frac{2C_{rn}}{m_n V_{COG}} \\
 b_{21n}(V_{COG}) &= \frac{2C_{fn}l_{fn}}{I_{zn}} & b_{22n}(V_{COG}) &= -\frac{2C_{rn}l_{rn}}{I_{zn}} \\
 b_{23n}(V_{COG}) &= \frac{1}{I_{zn}} & &
 \end{aligned} \tag{B.1}$$

APPENDIX C: Stability Proofs of the Adaptive Optimal Tire Force Distribution

Let's show the condition (4.30) holds for the adaptive system given in (4.29). The first derivative of the Lagrangian function given in (4.26) is

$$\frac{\partial L}{\partial \Delta F_{xi}} = 2\Delta F_{xi}\gamma_i + \frac{2}{(\mu_i F_{zi})^2} \Delta F_{xi}^3 - \lambda_{xi} + 2\eta_i(F_{xi} + \Delta F_{xi}) + R(\eta_{i+4} - \eta_{i+8}) \quad (\text{C.1})$$

$$\frac{\partial L}{\partial \Delta F_{yi}} = 2\Delta F_{yi}\rho_i + \frac{2}{(\mu_i F_{zi})^2} \Delta F_{yi}^3 + \lambda_{yi} + 2\eta_i(F_{yi} + \Delta F_{yi}) + C_i^{-1}(\eta_{i+16} - \eta_{i+12}) \quad (\text{C.2})$$

and the second derivatives are

$$\frac{\partial^2 L}{\partial \Delta F_{xi}^2} = 2\gamma_i + \frac{6}{(\mu_i F_{zi})^2} \Delta F_{xi}^2 + 2\eta_i \quad (\text{C.3})$$

$$\frac{\partial^2 L}{\partial \Delta F_{yi}^2} = 2\rho_i + \frac{6}{(\mu_i F_{zi})^2} \Delta F_{yi}^2 + 2\eta_i \quad (\text{C.4})$$

Since $\gamma_i > 0$, $\rho_i > 0$ and $\eta_i > 0$

$$\frac{\partial^2 L}{\partial \Delta F_{xi}^2} > 0, \frac{\partial^2 L}{\partial \Delta F_{yi}^2} > 0 \quad , \quad i = 1, 2, 3, 4 \quad (\text{C.5})$$

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List of Publications and Patents:

- **Murat Demirci**, A. Önder Biliroğlu, Metin Gökaşan, Seta Bogosyan (2010). “Sliding Mode Optimum Control For APU of Series Hybrid Electric Vehicles”, *IEEE International Symposium on Industrial Electronics*, Bari, Italy.
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- **Murat Demirci** and Metin Gokasan (2011). Sliding Mode MIMO Disturbance Observer for Input-Output Decoupling with Application to 4WAS Vehicles, *The Proceedings of the 2011 IEEE International Conference on Mechatronics*, (pp.90-95), Turkey: Istanbul Technical University, April 13-15.
- **Murat Demirci** and Metin Gokasan (2010). Adaptive Chattering-Free Model Reference SMC with Application to Dynamic Stability Control of Road Vehicles, *The Proceedings of the IEEE International Conference on Systems, Man and Cybernetics*, (pp. 574-580), Turkey: Bogazici University, October 10-13.