

Springer Tracts in Mechanical Engineering

Saad Kashem
Romesh Nagarajah
Mehran Ektesabi

Vehicle Suspension Systems and Electromagnetic Dampers

 Springer

Springer Tracts in Mechanical Engineering

Board of editors

Seung-Bok Choi, Inha University, Incheon, South Korea

Haibin Duan, Beijing University of Aeronautics and Astronautics, Beijing,
P.R. China

Yili Fu, Harbin Institute of Technology, Harbin, P.R. China

Carlos Guardiola, Universitat Politècnica de València, València, Spain

Jian-Qiao Sun, University of California, Merced, USA

About this Series

Springer Tracts in Mechanical Engineering (STME) publishes the latest developments in Mechanical Engineering - quickly, informally and with high quality. The intent is to cover all the main branches of mechanical engineering, both theoretical and applied, including:

- Engineering Design
- Machinery and Machine Elements
- Mechanical structures and Stress Analysis
- Automotive Engineering
- Engine Technology
- Aerospace Technology and Astronautics
- Nanotechnology and Microengineering
- Control, Robotics, Mechatronics
- MEMS
- Theoretical and Applied Mechanics
- Dynamical Systems, Control
- Fluids mechanics
- Engineering Thermodynamics, Heat and Mass Transfer
- Manufacturing
- Precision engineering, Instrumentation, Measurement
- Materials Engineering
- Tribology and surface technology

Within the scopes of the series are monographs, professional books or graduate textbooks, edited volumes as well as outstanding PhD theses and books purposely devoted to support education in mechanical engineering at graduate and post-graduate levels.

Indexed by SCOPUS and Springerlink.

To submit a proposal or request further information, please contact: Dr. Leontina Di Cecco Leontina.dicecco@springer.com or Li Shen Li.shen@springer.com.

Please check our Lecture Notes in Mechanical Engineering at <http://www.springer.com/series/11236> if you are interested in conference proceedings. To submit a proposal, please contact Leontina.dicecco@springer.com and Li.shen@springer.com.

More information about this series at <http://www.springer.com/series/11693>

Saad Kashem • Romesh Nagarajah •
Mehran Ektesabi

Vehicle Suspension Systems and Electromagnetic Dampers

 Springer

Saad Kashem
Swinburne University of
Technology Sarawak
Kuching, Sarawak
Malaysia

Romesh Nagarajah
Swinburne University of Technology
Melbourne, VIC
Australia

Mehran Ektesabi
Swinburne University of Technology
Melbourne, VIC
Australia

ISSN 2195-9862 ISSN 2195-9870 (electronic)
Springer Tracts in Mechanical Engineering
ISBN 978-981-10-5477-8 ISBN 978-981-10-5478-5 (eBook)
DOI 10.1007/978-981-10-5478-5

Library of Congress Control Number: 2017946769

© Springer Nature Singapore Pte Ltd. 2018

This work is subject to copyright. All rights are reserved by the Publisher, whether the whole or part of the material is concerned, specifically the rights of translation, reprinting, reuse of illustrations, recitation, broadcasting, reproduction on microfilms or in any other physical way, and transmission or information storage and retrieval, electronic adaptation, computer software, or by similar or dissimilar methodology now known or hereafter developed.

The use of general descriptive names, registered names, trademarks, service marks, etc. in this publication does not imply, even in the absence of a specific statement, that such names are exempt from the relevant protective laws and regulations and therefore free for general use.

The publisher, the authors and the editors are safe to assume that the advice and information in this book are believed to be true and accurate at the date of publication. Neither the publisher nor the authors or the editors give a warranty, express or implied, with respect to the material contained herein or for any errors or omissions that may have been made. The publisher remains neutral with regard to jurisdictional claims in published maps and institutional affiliations.

Printed on acid-free paper

This Springer imprint is published by Springer Nature
The registered company is Springer Nature Singapore Pte Ltd.
The registered company address is: 152 Beach Road, #21-01/04 Gateway East, Singapore 189721, Singapore

Author's Declaration

I hereby declare that I am the sole author of this manuscript. To the best of my knowledge, the document contains no material previously published or written by another person except where due reference is made in the text.

Dr. Saad Kashem

Acknowledgements

It is a pleasure to thank all the people who made this possible. It is my great pleasure to offer warm thanks to Professor Saman Halgamuge who is the assistant dean of the Melbourne School of Engineering at the University of Melbourne. The effort and time he took to help me to validate the designed full car analytical model were outstanding.

I have been privileged to work with and learn from Timothy Barry and Mehedi Al Emran Hasan. They helped me to learn MATLAB/Simulink. I am also grateful to them for helping me to get through the difficult times and for all the emotional support.

It is my pleasure to thank Jason Austin, Simon Lehman and Alex Barry who worked with me to set up and experiment the Quanser suspension plant. From my supervision of their undergraduate final-year project on active suspension system, I have learned many things.

I wish to thank Dr. Durul Huda for his time and patience in teaching me about the dynamics of the full car model. I would like to thank the many people who have taught me science, including my high school teachers (especially Abdul High) and my undergraduate faculties at East West University (especially Md. Ishfaqur Raza PhD, Dr. Ruhul Amin, Dr. Anisul Haque, Dr. Mohammad Ghulam Rahman, Dr. Khairul Alam, Dr. Tanvir Hasan Morshed), for their wise advice, helping with various applications and so on.

Lastly, and most importantly, I wish to thank my parents. They supported me and loved me. To them I dedicate this book.

And special thanks to almighty Allah who made this book possible.

Abstract

A suspension system is an essential element of a vehicle to isolate the frame of the vehicle from road disturbances. It is required to maintain continuous contact between a vehicle's tyres and the road. In order to achieve the desired ride comfort and road handling performance, many types of research have been conducted. A new modified skyhook control strategy with an adaptive gain that dictates the vehicle's semi-active suspension system is presented. The proposed closed-loop feedback system first captures the road profile input over a certain period. Then it calculates the best possible value of the skyhook gain for the subsequent process. Meanwhile, the system is controlled according to the new modified skyhook control law using an initial or previous value of the skyhook gain. In this book, the proposed suspension system is compared with passive and three other recently reported skyhook controlled semi-active suspension systems through a virtual environment with MATLAB/Simulink as well as an experimental analysis with Quanser suspension plant. Its performances have been evaluated in terms of ride comfort and road handling performance. The model has been validated in accordance with the international standards of admissible acceleration levels ISO2631 and human vibration perception. This control strategy has also been employed on the full car model to improve the isolation of the vibration and handling performance of the road vehicle.

This book also describes the development of a new analytical full vehicle model with nine degrees of freedom, which uses the new modified skyhook strategy to control the full vehicle vibration problem. Nowadays, many researchers are working on active tilting technology to improve vehicle cornering. But in those work, the effect of road bank angle is not considered in the control system design or in the dynamic model of the tilting standard passenger vehicles. The non-incorporation of road bank angle creates a non-zero steady-state torque requirement. Therefore, in this manuscript, this phenomenon was addressed while designing the direct tilt control and the dynamic model of the full car model.

This book has indicated the potential of the SKDT suspension system in improving cornering performances of the vehicle and paves the way for future work on vehicle's integrated system for chassis control.

Keywords Quarter-car • Vehicle • Suspension • Semi-active • Skyhook • Adaptive • Control • Damper • Quanser

Contents

1	Introduction	1
2	Control Strategies in the Design of Automotive Suspension Systems	9
3	Vehicle Suspension System	23
4	Design of Semi-active Suspension System	39
5	Full Car Model Cornering Performance	65
6	Simulation of Full Car Model	79
7	Experimental Analysis of Full Car Model	143
8	Conclusions and Recommendations	171
	Appendix A	177
	Appendix B	179
	Appendix C	187
	References	199

List of Figures

Fig. 1.1	Rear suspension system without wheel of a vehicle	2
Fig. 1.2	The passive, semi-active and active suspension system	2
Fig. 2.1	An ideal skyhook configuration	15
Fig. 2.2	A schematic of the groundhook control system	17
Fig. 2.3	Narrow commuter vehicle	18
Fig. 2.4	(a) Vehicle tilt by suspension, (b) vehicle tilt by actuator	18
Fig. 2.5	Nissan Land Glider	21
Fig. 3.1	Suspension system	24
Fig. 3.2	Passive suspension system	24
Fig. 3.3	Semi-active suspension system	26
Fig. 3.4	Active suspension system	26
Fig. 3.5	(a) Ideal quarter-car model, (b) simplified quarter-car model	27
Fig. 3.6	Mass spring characteristics	29
Fig. 3.7	Mass-spring-damper configuration	29
Fig. 3.8	Two degrees of freedom horizontal multiple mass spring damper	30
Fig. 3.9	Vertical multiple mass spring-damper configuration	31
Fig. 3.10	Forces acting at a point	32
Fig. 3.11	(a) Low-bandwidth suspension model, (b) high-bandwidth suspension model	33
Fig. 3.12	The road profile	34
Fig. 3.13	(a) Comparison between passive suspension models 1–6, (b) comparison between passive suspension models 1 and 7–11	35
Fig. 4.1	Schematic of the suspension systems based on proposed modified skyhook control system with adaptive skyhook gain	42

Fig. 4.2 (a) The time histories of three classes of roads, (b) power spectral density of three classes of road 45

Fig. 4.3 The time history of road profile 46

Fig. 4.4 The sprung-mass acceleration of the passive and semi-active suspension systems 47

Fig. 4.5 The ride comfort performance comparison 48

Fig. 4.6 The road-handling performance comparison 49

Fig. 4.7 Human vibration sensitivity test in frequency domain 50

Fig. 4.8 Quanser suspension plant 52

Fig. 4.9 Quanser suspension plant: (a) *front top panel view*, (b) Quanser suspension system *side view*, (c) Quanser suspension plant. *Front bottom panel view*, (d) Quanser suspension system *bottom view*, (e) Quanser suspension system *bottom view* 53

Fig. 4.10 The Quanser quarter-car model experimental setup 55

Fig. 4.11 The Quanser suspension plant modelled in Simulink 56

Fig. 4.12 DC-micro motor characteristics curve 59

Fig. 4.13 The sprung-mass acceleration of the passive and semi-active suspension systems (a) in a simulation environment, (b) in the experimental setup 60

Fig. 4.14 The ride comfort performance comparison (a) in simulation environment, (b) through experimental setup 61

Fig. 4.15 The road-handling performance comparison (a) in simulation environment, (b) through experimental setup 62

Fig. 4.16 Vertical vibration of car suspension in frequency domain 63

Fig. 5.1 A schematic diagram of a full vehicle active suspension system 66

Fig. 5.2 Free body diagram of a bicycle model 68

Fig. 5.3 Stable and unstable lateral forces acting on a static vehicle 69

Fig. 5.4 (a) Acting torque on the vehicle body, (b) *front view* of the tilting vehicle 71

Fig. 5.5 Driving scenario one 73

Fig. 5.6 Driving scenario two 74

Fig. 5.7 Driving scenario three 74

Fig. 5.8 Driving scenario four 75

Fig. 6.1 Simulink model 81

Fig. 6.2 The frequency domain response of the car body vertical acceleration to road class A: (a) at narrow frequency range and (b) at broad frequency range 82

Fig. 6.3 The frequency domain response of the car body pitch angular acceleration to road class A: (a) at narrow frequency range and (b) at broad frequency range 83

Fig. 6.4 The time domain response of vehicle body vertical acceleration to road class A: (a) full trajectory and (b) short time span 84

Fig. 6.5 The time domain response of vehicle pitch angular acceleration to road class A: **(a)** full trajectory and **(b)** short time span 85

Fig. 6.6 The time domain response of vehicle pitch angular acceleration to road class A: **(a)** full trajectory and **(b)** short time span 86

Fig. 6.7 The frequency domain response of the car body vertical acceleration to road class B: **(a)** at narrow frequency range and **(b)** at broad frequency range 87

Fig. 6.8 The frequency domain response of the car body pitch angular acceleration to road class B: **(a)** at low frequency and **(b)** at broad frequency range 88

Fig. 6.9 The time domain response of vehicle body vertical acceleration to road class B: **(a)** full trajectory and **(b)** short time span 89

Fig. 6.10 The time domain response of vehicle pitch angular acceleration to road class B: **(a)** full trajectory and **(b)** short time span 90

Fig. 6.11 The time domain response of the vehicle sprung mass m_1 vertical displacement to road class B: **(a)** full trajectory and **(b)** short time span 91

Fig. 6.12 The frequency domain response of the car body vertical acceleration to road class C: **(a)** at narrow frequency range and **(b)** at broad frequency range 92

Fig. 6.13 The frequency domain response of the car body pitch angular acceleration to road class C: **(a)** at narrow frequency range and **(b)** at broad frequency range 93

Fig. 6.14 The time domain response of vehicle body vertical acceleration to road class C: **(a)** full trajectory and **(b)** short time span 94

Fig. 6.15 The time domain response of vehicle pitch angular acceleration to road class C: **(a)** full trajectory and **(b)** short time span 95

Fig. 6.16 The time domain response of the vehicle sprung mass m_1 vertical displacement to road class C: **(a)** full trajectory and **(b)** short time span 96

Fig. 6.17 The frequency domain response of the car body vertical acceleration to the combined road: **(a)** at narrow frequency range and **(b)** at broad frequency range 97

Fig. 6.18 The frequency domain response of the car body pitch angular acceleration to the combined road: **(a)** at narrow frequency range and **(b)** at broad frequency range 98

Fig. 6.19 The time domain response of vehicle body vertical acceleration to the combined road: **(a)** full trajectory and **(b)** short time span 99

Fig. 6.20 The time domain response of vehicle pitch angular acceleration to the combined road: **(a)** full trajectory and **(b)** short time span 100

Fig. 6.21 The time domain response of the vehicle sprung mass m_1 vertical displacement to the combined road: **(a)** full trajectory and **(b)** short time span 101

Fig. 6.22 The response of steering and bank angle in driving scenario one: (a) desired tilting angle, (b) required actuator force 102

Fig. 6.23 The vehicle body vertical acceleration for driving scenario one: (a) full trajectory and (b) short time span 103

Fig. 6.24 The pitch angular acceleration for driving scenario one: (a) full trajectory and (b) short time span 104

Fig. 6.25 The roll angular acceleration for driving scenario one: (a) full trajectory and (b) short time span 105

Fig. 6.26 The lateral acceleration for driving scenario one: (a) full trajectory and (b) short time span 107

Fig. 6.27 The vehicle sprung mass m_1 's vertical displacement for driving scenario one: (a) full trajectory and (b) short time span 108

Fig. 6.28 The rollover threshold in driving scenario one: (a) full trajectory and (b) short time span 109

Fig. 6.29 The response of steering and bank angle in driving scenario two: (a) desired tilting angle, (b) required actuator force 110

Fig. 6.30 The vehicle sprung mass m_1 's vertical displacement for driving scenario two: (a) full trajectory and (b) short time span 111

Fig. 6.31 The vehicle body vertical acceleration for driving scenario two: (a) full trajectory and (b) short time span 112

Fig. 6.32 The pitch angular acceleration for driving scenario two: (a) full trajectory and (b) short time span 113

Fig. 6.33 The roll angular acceleration for driving scenario two: (a) full trajectory and (b) short time span 114

Fig. 6.34 The lateral acceleration for driving scenario two: (a) full trajectory and (b) short time span 115

Fig. 6.35 The rollover threshold in driving scenario two: (a) full trajectory and (b) short time span 116

Fig. 6.36 The response of steering and bank angle in driving scenario three: (a) desired tilting angle, (b) required actuator force 117

Fig. 6.37 The vehicle sprung mass m_1 's vertical displacement for driving scenario three: (a) full trajectory and (b) short time span 118

Fig. 6.38 The vehicle body vertical acceleration for driving scenario three: (a) full trajectory and (b) short time span 119

Fig. 6.39 The pitch angular acceleration for driving scenario three: (a) full trajectory and (b) short time span 120

Fig. 6.40 The roll angular acceleration for driving scenario three: (a) full trajectory and (b) short time span 121

Fig. 6.41 The lateral acceleration for driving scenario three: (a) full trajectory and (b) short time span 122

Fig. 6.42 The rollover threshold in driving scenario three: (a) full trajectory and (b) short time span 123

Fig. 6.43 The response of steering and bank angle in driving scenario four: (a) desired tilting angle, (b) required actuator force 124

Fig. 6.44 The vehicle sprung mass m_1 's vertical displacement for driving scenario four: (a) full trajectory and (b) short time span 125

Fig. 6.45 The vehicle body vertical acceleration for driving scenario four: (a) full trajectory and (b) short time span 126

Fig. 6.46 The pitch angular acceleration for driving scenario four: (a) full trajectory and (b) short time span 127

Fig. 6.47 The roll angular acceleration for driving scenario four: (a) full trajectory and (b) short time span 128

Fig. 6.48 The lateral acceleration for driving scenario four: (a) full trajectory and (b) short time span 129

Fig. 6.49 The rollover threshold in driving scenario four: (a) full trajectory and (b) short time span 130

Fig. 6.50 The frequency domain response of the car body vertical acceleration: (a) at narrow frequency range and (b) at broad frequency range 131

Fig. 6.51 The frequency domain response of the car body pitch angular acceleration (a) at narrow frequency range and (b) at broad frequency range 132

Fig. 6.52 The frequency domain response of the car body roll angular acceleration (a) at narrow frequency range and (b) at broad frequency range 133

Fig. 6.53 The response of steering and bank angle in driving scenario four and road class C: (a) desired tilting angle, (b) required actuator force 134

Fig. 6.54 The vehicle body vertical acceleration for driving scenario four and road class C: (a) full trajectory and (b) short time span 135

Fig. 6.55 The pitch angular acceleration for driving scenario four and road class C: (a) full trajectory and (b) short time span 136

Fig. 6.56 The roll angular acceleration for driving scenario four and road class C: (a) full trajectory and (b) short time span 137

Fig. 6.57 The lateral acceleration for driving scenario four and road class C: (a) full trajectory and (b) short time span 138

Fig. 6.58 The vehicle sprung mass m_1 's vertical displacement for driving scenario four and road class C: (a) full trajectory and (b) short time span 139

Fig. 6.59 The rollover threshold in driving scenario four and road class C: (a) full trajectory and (b) short time span 140

Fig. 6.60 Vehicle body vertical acceleration comparison 140

Fig. 6.61 Vehicle body pitch angular acceleration comparison 141

Fig. 6.62 Vehicle body roll angular acceleration comparison 141

Fig. 6.63 Vehicle body lateral acceleration comparison 142

Fig. 6.64 Vehicle road handling performance comparison 142

Fig. 7.1 Quanser simulink model 144

Fig. 7.2 Quanser intelligent suspension plant 146

Fig. 7.3 The vehicle front left sprung mass vertical displacement 146

Fig. 7.4 The frequency response of vehicle body vertical acceleration:
(a) at narrow frequency range and (b) at broad frequency
range 147

Fig. 7.5 The frequency domain response of the car body pitch angular
acceleration: (a) at narrow frequency range and (b) at broad
frequency range 148

Fig. 7.6 The frequency domain response of the car body roll angular
acceleration: (a) at narrow frequency range and (b) at broad
frequency range 149

Fig. 7.7 The response of steering and bank angle in driving scenario four
and road class C: (a) desired tilting angle and (b) required
actuator force 150

Fig. 7.8 The vehicle body vertical acceleration for driving scenario four
and road class C: (a) full trajectory and (b) short time span 151

Fig. 7.9 The pitch angular acceleration for driving scenario four and road
class C: (a) full trajectory and (b) short time span 152

Fig. 7.10 The roll angular acceleration for driving scenario four and road
class C: (a) full trajectory and (b) short time span 153

Fig. 7.11 The lateral acceleration for driving scenario four and road class
C: (a) full trajectory and (b) short time span 154

Fig. 7.12 The vehicle sprung mass m_1 's vertical displacement for driving
scenario four and road class C: (a) full trajectory and (b) short
time span 155

Fig. 7.13 The rollover threshold in driving scenario four and road class C:
(a) full trajectory and (b) short time span 156

Fig. 7.14 Vehicle body vertical acceleration comparison 156

Fig. 7.15 Vehicle body pitch angular acceleration comparison 157

Fig. 7.16 Vehicle body roll angular acceleration comparison 157

Fig. 7.17 Vehicle body lateral acceleration comparison 158

Fig. 7.18 Vehicle road handling performance comparison 158

Fig. 7.19 The vehicle rear right sprung mass vertical displacement 158

Fig. 7.20 The frequency response of vehicle body vertical acceleration:
(a) at narrow frequency range and (b) at broad frequency
range 159

Fig. 7.21 The frequency response of vehicle body pitch angular
acceleration: (a) at narrow frequency range and (b) at broad
frequency range 160

Fig. 7.22 The frequency response of vehicle body roll angular acceleration:
(a) at narrow frequency range and (b) at broad frequency
range 161

Fig. 7.23 The response of steering and bank angle in driving scenario four
and road class C: (a) desired tilting angle and (b) required
actuator force 162

Fig. 7.24 The vehicle body vertical acceleration for driving scenario four
and road class C: (a) full trajectory and (b) short time span 163

Fig. 7.25 The pitch angular acceleration for driving scenario four and road class C: **(a)** full trajectory and **(b)** short time span 164

Fig. 7.26 The roll angular acceleration for driving scenario four and road class C: **(a)** full trajectory and **(b)** short time span 165

Fig. 7.27 The lateral acceleration for driving scenario four and road class C: **(a)** full trajectory and **(b)** short time span 166

Fig. 7.28 The vehicle sprung mass m_3 's vertical displacement for driving scenario four and road class C: **(a)** full trajectory and **(b)** short time span 167

Fig. 7.29 The rollover threshold in driving scenario four and road class C: **(a)** full trajectory and **(b)** short time span 168

Fig. 7.30 Vehicle body vertical acceleration comparison 168

Fig. 7.31 Vehicle body pitch angular acceleration comparison 169

Fig. 7.32 Vehicle body roll angular acceleration comparison 169

Fig. 7.33 Vehicle body lateral acceleration comparison 169

Fig. 7.34 Vehicle road handling performance comparison 170

Fig. A1 Determine lateral position acceleration 177

Fig. A2 Determine the front and rear tires lateral forces 178

List of Tables

Table 3.1	The parameters of quarter-car models	36
Table 3.2	Comparison between outputs of the vehicle sprung-mass acceleration	36
Table 4.1	Theoretical road classes on the basis of road roughness	44
Table 4.2	Nominal parameter values used in the simulation	46
Table 4.3	Nomenclature of Quanser suspension system component	54
Table 4.4	Nominal parameter values used in the experiment	57
Table 4.5	The FAULHABER DC-micro motor specification	58
Table 6.1	Nominal parameter values used in the simulation	80
Table 7.1	Nominal parameter values used in the experiment	145

Chapter 1

Introduction

Abstract In this chapter, background of this book has been described. Motivation and methodologies has been depicted in the later section. A brief outline of this manuscript has been included in the last section.

1.1 Background

One of the most important considerations of the present automotive industry is to provide passenger safety, through optimal ride comfort and road holding, for a large variety of vehicle manoeuvres and road conditions. The comfort and safety of the passenger travelling in a vehicle can be improved by minimizing the body vibration, roll and heave of the vehicle body through an optimal road contact for the tyres. The system in the vehicle that provides these actions is the vehicle suspension, i.e. a complex system consisting of various arms, springs and dampers that separate the vehicle body from the tyres and axles (Fig. 1.1). In general, vehicles are equipped with fully passive suspension systems due to their low cost and simple construction. The passive suspension consists of springs, dampers and anti-roll bars with fixed characteristics. The major drawback of the passive suspension design is that you cannot simultaneously maximize both vehicle ride and handle performance. To achieve better ride performance, a “soft” suspension needs to be introduced to maintain contact between the vehicle body and the tyre. The “soft” suspension easily absorbs road disturbances. That is why most of the luxury cars employ “soft” suspensions to provide a comfortable ride. The second characteristic of vehicle performance is the road handling. This refers to a vehicle’s ability to maintain contact between the vehicle’s tyre and the road during turns and other dynamic manoeuvres. This can be achieved by “stiff” suspensions as seen in sports cars. The challenge of the passive suspension system is in achieving the right compromise between the two characteristics of vehicle performance which will best suit the targeted consumer. However, by introducing the active or semi-active suspension system in the vehicle (Fig. 1.2), a more desirable compromise can be achieved between the benefits of the soft and stiff suspension system.

The active or semi-active suspension systems are incorporated with the active components, such as actuators and semi-active dampers, coupled with various



Fig. 1.1 Rear suspension system without wheel of a vehicle

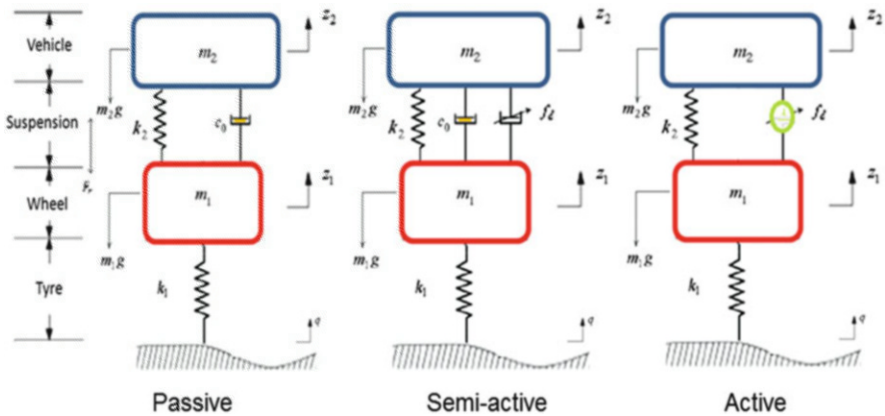


Fig. 1.2 The passive, semi-active and active suspension system

dynamic control strategies. With active components, these systems can provide adjustable spring stiffness and damping coefficients adapted to various road conditions.

Since the early 1970s, many types of active and semi-active suspension systems have been proposed to achieve better control of damping characteristics. Although the active suspension system shows better performance in a wide frequency range, its implementation complexity and cost prevent wider commercial applications. That is why the semi-active suspension system has been widely studied to achieve

high levels of performance in terms of vehicle suspension system. To control the damper of the semi-active suspension system, many control strategies including skyhook surface sliding mode control [1], neural network control [2], H-infinity control [3], skyhook control, ground hook control, hybrid control [4, 5], fuzzy logic control [6, 7], neural network-based fuzzy control [8], neuro-fuzzy control [9], discrete time fuzzy sliding mode control [10], optimal fuzzy control [11], and adaptive fuzzy logic control [12, 13] have been explored. Between all of the above control systems, the skyhook control proposed by Karnopp et al. in 1974 [14] is widely used since it yields the best compromise between vehicle performance and practical implementation of semi-active suspension systems.

In the past few decades, researchers have modified the basic skyhook control strategy by adding some variations and have named them optimal, modified or adaptive type skyhook control strategies [15, 16]. But in most of these studies, skyhook gain (SG) of the control strategy remains as a constant value, and it is usually chosen from a set of values as suited for the vehicle in the simulation environment. One of the major goals of this manuscript is to present a new modified skyhook control strategy with adaptive SG.

This control strategy has also been employed on the full car model to improve the isolation of the vibration and handling the performance of the road vehicle. The full car model designed in this manuscript has nine degrees of freedom, and those are the heave modes of four wheels and the heave, lateral, roll, pitch and yaw modes of the vehicle body.

Nowadays, some researchers have focused on active steering control to improve vehicle cornering [17–19]. Three types of active steering control strategies have been proposed. These are the four-wheel active steering system (4WAS), the front wheel active steering system (FWAS) and the active rear wheel steering system (RWAS). The four-wheel active steering system (4WAS) is the combination of the rear active steering system and the front active steering system. In the FWAS system, the front wheel steer angle is determined by the steering angle generated due to the driver's direct steering input and a resultant corrective steering angle input that is produced by the design of the active front wheel steering controller.

Vehicle performance during cornering has been improved by most of the car manufacturers by using electronic stability control (ESC). Car manufacturers use different brand names for ESC, such as Volvo named it DSTC (Dynamic Stability and Traction Control); Mercedes and Holden called it ESP (Electronic Stability Program); and DSC (Dynamic Stability Control) is the term used by BMW and Jaguar, but despite the term used, the processes are almost the same. To avoid oversteering and understeering during cornering, ESC extends the brake and different torque on each wheel of the vehicle. But ESC reduces the longevity of the tyre as the tyre skids while random braking. To overcome this problem, a vehicle can be tilted inward via an active or semi-active suspension system.

The concept of "active tilting technology" has become quite popular in narrow tilting road vehicles and modern railway vehicles. Now in Europe, most new high-speed trains are fitted with active tilt control systems, and these trains are used as regional express trains [20, 21]. To tilt the train inward during cornering, tilting

actuators are used as an element of the secondary active suspension system. These actuators are named as bolsters. In a road vehicle, actuators are also used to affect the vehicle roll angle via an active suspension system. Since the beginning of the 1950s, there has been extensive work done in developing the narrow tilting vehicle by both the automotive industry [22–25] and academic researchers [26–30].

This particular small and narrow geometric property of the vehicle poses stability problems when the vehicle needs to corner or change a lane. There are also two types of control schemes that have been used to stabilize the narrow tilting vehicle. These control schemes are defined as direct tilt control (DTC) and steering tilt control (STC) systems as detailed in [27, 31, 32]. A typical passenger vehicle body can be tilted up to ten degrees as the maximum suspension travel is around 0.25 m. Then, the lateral acceleration of the tilted vehicle caused by gravity can reach a maximum of about 0.17 g [33]. Since the lateral acceleration produced by normal steering manoeuvres is around 0.3–0.5 g, the active or semi-active suspension systems have the potential of improving vehicle ride handling performance [33]. Semi-active or active suspension systems can act promptly to tilt the vehicle with the help of semi-active dampers or actuators. However, the active suspension systems need to avoid over-sensitive reaction to driver's steering commands for vehicle safety. Recently Bose Corporation presented the Bose suspension system [34] in which the high-bandwidth linear electromagnetic dampers improved vehicle cornering. It is able to counter the body roll of the vehicle by stiffening the suspension while cornering. Car giant Nissan has developed a four-wheeled ground vehicle named Land Glider [35]. The vehicle body can lean into a corner up to 17° for sharper handling considering the speed, steering angle and yaw rate of the vehicle. In addition, in the works stated above and other research, the effect of road bank angle is neither considered in the control system design nor in the dynamic model of the tilting standard passenger vehicles [26, 27, 31, 32, 36–44]. Not incorporating the road bank angle creates a non-zero steady-state torque requirement. So this phenomenon needs to be addressed while designing the tilt control and the dynamic model of the full car model. To lean a vehicle which incorporates the road bank angle, the response time of the actuator or semi-active damper plays an important role.

The majority of the semi-active suspension systems use pneumatic or hydraulic solutions as the actuator or semi-active damper [45–49]. These systems are characterized by high force and power densities but suffer from low efficiencies and response bandwidths. Commercial systems incorporating electromagnetic elements (combine rotary actuators and mechanical elements) illustrate the properties of the magneto-rheological fluids in damper technology to provide adjustable spring stiffness. However, linear electromagnetic actuators appear as a better solution for a semi-active suspension system in respect of their high force densities, form factor, and response bandwidth. The motivation and the methodology of this manuscript are described in the next section.