

Adaptive and Interconnected Suspension Systems for Improving Truck Stability and Ride Quality

by

Yukun Lu

A thesis

presented to the University of Waterloo

in fulfillment of the

thesis requirement for the degree of

Doctor of Philosophy

in

Mechanical and Mechatronics Engineering

Waterloo, Ontario, Canada, 2023

© Yukun Lu 2023

Examining Committee Membership

The following served on the Examining Committee for this thesis. The decision of the Examining Committee is by majority vote.

Supervisor:	Dr. Amir Khajepour Professor, Dept. of Mechanical and Mechatronics Engineering University of Waterloo
Internal Member:	Dr. Stephan Lambert Professor, Dept. of Mechanical and Mechatronics Engineering University of Waterloo
Internal Member:	Dr. Armaghan Salehian Associate Professor, Dept. of Mechanical and Mechatronics Engineering University of Waterloo
Internal-External Member:	Dr. Nasser Lashgarian Azad Associate Professor, Dept. of Systems Design Engineering University of Waterloo
External Member:	Dr. Siamak Arzanpour Associate Professor, School of Mechatronic Systems Engineering Simon Fraser University

Author's Declaration

I hereby declare that I am the sole author of this thesis. This is a true copy of the thesis, including any required final revisions, as accepted by my examiners.

I understand that my thesis may be made electronically available to the public.

Abstract

Truck drivers are constantly exposed to undesirable vibrations caused by uneven road surfaces, especially in long-distance transportation. In addition to being at risk of repetitive motion injuries, drivers also suffer from fatigue, which in turn becomes a potential safety issue. As a way to resolve the issue, a truck is usually equipped with two suspension systems, namely the primary suspension and the secondary (cabin) suspension, in which the cabin suspension mainly acts as a vibration isolation system between the cabin and the rest of the vehicle. Different techniques have been applied to the cabin suspension to improve ride quality, stability, and safety. In most commercial vehicles, conventional passive suspensions are used that contain passive springs and dampers with fixed stiffness and damping properties. Nowadays, adaptive suspension systems are developed to further improve ride quality by dynamically controlling damping characteristics. Besides, interconnected suspension systems have been introduced to enhance the roll stability of the sprung mass and provide better handling and ride quality simultaneously. As a result, this study aims to improve truck drivers' ride comfort and stability by developing semi-active and interconnected suspension systems. A semi-active cabin suspension system is developed based on different multi-objective optimal control approaches to attenuate the vertical, roll, and pitch vibrations transmitted to the cabin. The Model Predictive Control (MPC) is chosen as the benchmark because of its superior performance in handling constraints and prediction, although its high computational costs make it difficult to be implemented in suspension applications. To solve this issue, a novel integrated Skyhook-LQR (Linear Quadratic Regulator) control approach is introduced to take advantage of both Skyhook and LQR while its computational cost is low enough for real-time implementation in suspension control tasks. In addition, two gain-adaptive algorithms are proposed to intelligently adjust the control gains according to the disturbance inputs and/or vehicle states based on onboard sensor measurements. Two novel interconnected suspension systems are designed to minimize the harsh rotational motions during braking or turning maneuvers, namely AIS-ARS and AIS-ARPS. Both configurations have adaptive damping and adjustable rotational stiffness characteristics, which provide excellent vibration attenuation and attitude control performances. All the proposed suspension systems and their mathematical models are examined and validated through laboratory experiments and co-simulations between ADAMS/Car and MATLAB/Simulink.

Acknowledgements

First and foremost, I would like to express my deepest appreciation to my supervisor, Dr. Amir Khajepour, for his invaluable guidance, unwavering support, and constant encouragement. His expertise and scholarly insights have profoundly influenced my academic growth. I am genuinely grateful for the opportunity to finish both my Master's and Ph.D. studies with him. His positive influence extends far beyond the boundaries of academia and into the realms of personal growth and development.

I would like to thank the members of my examining committee, Prof. Stephan Lambert, Prof. Armaghan Salehian, Prof. Nasser Lashgarian Azad, and Prof. Siamak Arzanpour, for their insightful comments and suggestions on my work.

Completing this research would not have been possible without the support of the remarkable fellows at the Mechatronic Vehicle Systems Laboratory. I sincerely appreciate the indispensable help from the technicians, Jeff Graansma, Michael Duthie, Aaron Sherratt, and Ruslan Liang, whose expertise and assistance were crucial in carrying out experimental work and ensuring the smooth operation of the lab. In addition, I would like to thank the post-doctoral fellows, Dr. Amir Soltani and Dr. Alireza Pazooki, for their timely assistance. At the outset of my academic journey, I am grateful to have guidance from Dr. Yanjun Huang, Dr. Yechen Qin, Dr. Yubiao Zhang, Dr. Wenhao Yu, Dr. Yixiao Liang, and Dr. Hong Wang. Throughout my five-year study, my profound thanks to my schoolmates Dr. Shiyi Yang, Ruihe Zhang, Jiaming Zhong, Minghao Ning, Shucheng Huang, Ruizeng Zhang, and Dr. Chen Sun.

Coming to Waterloo after my undergraduate was my first time far away from home. My friends here make my new life much easier and more joyful than expected. My utmost thanks to Ruoyi (Ophelia) Li for helping me settle down in Waterloo and continued help in these years. My heartfelt thanks to Xueman (JoJo) Lyu for her encouragement, care, and fantastic dish. A special thanks to my beloved furry friend, Seven. Her adorable nature and unmatched companionship make my days brighter.

Lastly, I would like to express my deepest gratitude to my extraordinary parents, Yanqun Lu and Hongying Kong. Their unwavering love, support, and understanding have been a constant source of strength and motivation. The pandemic forced us apart for more than four long years, and I am so sorry for the absence from home. See you soon!

Dedication

To my beloved parents,

Hongying Kong & Yanqun Lu

for their endless love and support.

Table of Contents

Examining Committee Membership.....	ii
Author's Declaration	iii
Abstract.....	iv
Acknowledgements	v
Dedication.....	vi
List of Figures.....	xi
List of Tables	xvi
Chapter 1 Introduction.....	1
1.1 Motivations	1
1.2 Contributions	2
1.3 Outline	4
Chapter 2 Literature Review.....	6
2.1 Vehicle Adaptive Suspension Systems.....	6
2.1.1 Passenger car adaptive suspension	7
2.1.2 Cabin adaptive suspension.....	13
2.2 Interconnected Suspension Systems	23
2.2.1 Roll-plane interconnected configuration	25
2.2.2 Pitch-plane interconnected configuration	26
2.3 Vehicle State Observer	28
2.4 Summary.....	31
Chapter 3 Coordinated Semi-Active Suspension Systems for Truck Cabins.....	34
3.1 Modelling of Cab-over-engine Suspension System.....	34
3.1.1 Passive COE suspension system.....	36

3.1.2 Adaptive COE suspension system	45
3.2 Suspension Coordinated Control Strategies	47
3.2.1 Model Predictive Control for semi-active suspension systems	48
3.2.2 Integrated Skyhook-LQR controller for semi-active suspension systems	55
3.2.3 Gain-adaptive algorithms.....	58
3.3 State Estimation.....	63
3.4 Summary.....	65
Chapter 4 Simulation and Experimental Studies of Coordinated Semi-Active Suspensions	66
4.1 Co-simulation	66
4.2 Vibration Assessment.....	68
4.3 Simulation Results.....	70
4.3.1 Observer evaluation.....	70
4.3.2 Cabin: dual bumps at 25 km/h	75
4.3.3 Cabin: offset-bump at 20 km/h	80
4.3.4 Cabin: paved road test	85
4.4 Experimental Results.....	89
4.4.1 Experimental setup	89
4.4.2 Observer evaluation and model validation	94
4.4.3 Performance evaluation	96
4.5 Summary.....	112
Chapter 5 Adaptive Interconnected Suspension Systems with Adjustable Rotational Stiffness....	114
5.1 AIS-ARS.....	114
5.1.1 Topology of the AIS-ARS	114
5.1.2 Mathematical modelling.....	116

5.1.3 Roll stiffness analysis	120
5.1.4 Vertical stiffness analysis	123
5.1.5 Damping characteristics.....	124
5.1.6 Damping control system for AIS-ARS	127
5.2 AIS-ARPS	128
5.2.1 Topology of the AIS-ARPS.....	128
5.2.2 Roll stiffness analysis	132
5.2.3 Pitch stiffness analysis.....	134
5.2.4 Damping characteristics.....	136
5.3 Summary.....	139
Chapter 6 Experimental and Simulation Studies of Adaptive Interconnected Suspension Systems with Adjustable Rotational Stiffness	141
6.1 Simulation Results	141
6.1.1 Heavy truck: steady turn at 45 km/h.....	142
6.1.2 Heavy truck: articulation test.....	142
6.1.3 Heavy truck: speedbumps at 20 km/h.....	144
6.1.4 Heavy truck: paved road test	145
6.1.5 Cabin: double lane change.....	145
6.1.6 Cabin: steady turn at 60 km/h.....	147
6.1.7 Cabin: emergency braking.....	150
6.2 Experimental Results	152
6.3 Summary.....	156
Chapter 7 Conclusions and Future Work.....	157
7.1 Conclusions	157

7.2 Future Work.....	160
References	161
Appendix A High-fidelity models in ADAMS/Car.....	175
Appendix B Prototype of AIS-ARS	177
Appendix C Scaled-down cab-over-engine prototype.....	179

List of Figures

Figure 2.1. Cabin suspension configurations.....	16
Figure 2.2. The three DOFs model in the literature: (a) four-point cabin, (b) passenger car	18
Figure 2.3. Quarter-cabin model.....	18
Figure 2.4. Skyhook control theory	18
Figure 2.5. Sketch of the truck with cabin suspension	19
Figure 2.6. Three DOFs cabin model	20
Figure 2.7. Quarter-truck model	20
Figure 2.8. (a) H_2 arrangement; (b) roll-plane interconnected arrangement.....	26
Figure 2.9. Pitch interconnected configurations [136,137].	28
Figure 3.1. Cabin tilting of a flat-nose truck	35
Figure 3.2. The CAD model of a cab-over-engine suspension system.....	35
Figure 3.3. Schematic of truck cabin system.....	35
Figure 3.4. Drawing of the linkage mechanism.....	36
Figure 3.5. Pitch plane of the cabin system	39
Figure 3.6. Roll plane of the cabin system	39
Figure 3.7. Cabin coordinate located at the revolute joint B	40
Figure 3.8. Equivalent adaptive cabin suspension (quarter).....	46
Figure 3.9. Integrated Skyhook-LQR control.....	56
Figure 3.10. Skyhook control approach.....	57
Figure 3.11. Gain-adaptive algorithm.....	59
Figure 3.12. The sigmoid function for gains selection	60
Figure 3.13. Cabin roll rate response.....	62
Figure 3.14. Sensor location	64
Figure 4.1. High-fidelity flat-nose truck model in ADAMS/Car	66
Figure 4.2. Co-simulation between ADAMS/Car and MATLAB/Simulink	67
Figure 4.3. Dual bumps road profile.....	71
Figure 4.4. Observer estimation of cabin vertical motions (dual bumps at 25 km/h).....	71
Figure 4.5. Observer estimation of cabin pitch motions (dual bumps at 25 km/h).....	72

Figure 4.6. Offset-bump road profile.....	73
Figure 4.7. Observer estimation of cabin vertical motions (offset-bump at 20 km/h).....	73
Figure 4.8. Observer estimation of cabin pitch motions (offset-bump at 20 km/h).....	74
Figure 4.9. Observer estimation of cabin roll motions (offset-bump at 20 km/h).....	74
Figure 4.10. Cabin frequency-weighted vertical acceleration (dual bumps at 25 km/h).....	75
Figure 4.11. Cabin vertical velocity (dual bumps at 25 km/h)	76
Figure 4.12. Cabin vertical displacement (dual bumps at 25 km/h)	76
Figure 4.13. Cabin frequency-weighted pitch acceleration (dual bumps at 25 km/h).....	77
Figure 4.14. Cabin pitch rate (dual bumps at 25 km/h)	77
Figure 4.15. Cabin pitch angle (dual bumps at 25 km/h)	78
Figure 4.16. Adaptive gain scheduling (dual bumps at 25 km/h).....	79
Figure 4.17. Comparison of damping forces (dual bumps at 25 km/h).....	79
Figure 4.18. Cabin frequency-weighted vertical acceleration (offset-bump at 20 km/h).....	80
Figure 4.19. Cabin vertical velocity (offset-bump at 20 km/h)	81
Figure 4.20. Cabin vertical displacement (offset-bump at 20 km/h).....	81
Figure 4.21. Cabin frequency-weighted roll acceleration (offset-bump at 20 km/h)	82
Figure 4.22. Cabin roll rate (offset-bump at 20 km/h).....	83
Figure 4.23. Cabin roll angle (offset-bump at 20 km/h).....	83
Figure 4.24. Cabin frequency-weighted pitch acceleration (offset-bump at 20 km/h).....	84
Figure 4.25. Cabin pitch rate (offset-bump at 20 km/h)	84
Figure 4.26. Adaptive gain scheduling (offset-bump at 20 km/h).....	85
Figure 4.27. Paved road profiles.....	86
Figure 4.28. Combined ISO road profile	87
Figure 4.29. Comparison of computational time (combined ISO road)	87
Figure 4.30. CAD drawing of the scaled-down cabin model	90
Figure 4.31. Test rig for the adaptive suspension systems	91
Figure 4.32. Schematic of the Stewart Platform coordinates	92
Figure 4.33. Passive damper tuning.....	94
Figure 4.34. Observer evaluation: body vertical velocity.....	95

Figure 4.35. Observer evaluation: body vertical displacement	95
Figure 4.36. Observer evaluation: body roll angle	96
Figure 4.37. Observer evaluation: body pitch angle	96
Figure 4.38. Stewart Platform movement: paved road	97
Figure 4.39. Scaled-down cabin roll acceleration (pure roll motion)	99
Figure 4.40. Scaled-down cabin roll acceleration (pure roll motion)	99
Figure 4.41. Scaled-down cabin roll rate (pure roll motion)	99
Figure 4.42. Scaled-down cabin roll rate (pure roll motion)	100
Figure 4.43. Scaled-down cabin roll angle (pure roll motion)	100
Figure 4.44. Scaled-down cabin roll angle (pure roll motion)	100
Figure 4.45. Adaptive gain of roll (pure roll motion)	101
Figure 4.46. Scaled-down cabin pitch acceleration (emergency brake and acceleration)	102
Figure 4.47. Scaled-down cabin pitch acceleration (emergency brake and acceleration)	103
Figure 4.48. Scaled-down cabin pitch rate (emergency brake and acceleration)	103
Figure 4.49. Scaled-down cabin pitch rate (emergency brake and acceleration)	103
Figure 4.50. Scaled-down cabin pitch angle (emergency brake and acceleration)	104
Figure 4.51. Scaled-down cabin pitch angle (emergency brake and acceleration)	104
Figure 4.52. Adaptive gain of pitch (emergency brake and acceleration)	104
Figure 4.53. Scaled-down cabin vertical acceleration (offset-bump at 20 km/h)	106
Figure 4.54. Scaled-down cabin vertical acceleration (offset-bump at 20 km/h)	106
Figure 4.55. Scaled-down cabin roll acceleration (offset-bump at 20 km/h)	107
Figure 4.56. Scaled-down cabin roll acceleration (offset-bump at 20 km/h)	107
Figure 4.57. Scaled-down cabin roll rate (offset-bump at 20 km/h)	108
Figure 4.58. Scaled-down cabin roll rate (offset-bump at 20 km/h)	108
Figure 4.59. Scaled-down cabin roll angle (offset-bump at 20 km/h)	108
Figure 4.60. Scaled-down cabin roll angle (offset-bump at 20 km/h)	109
Figure 4.61. Scaled-down cabin pitch acceleration (offset-bump at 20 km/h)	109
Figure 4.62. Scaled-down cabin pitch acceleration (offset-bump at 20 km/h)	109
Figure 4.63. Scaled-down cabin pitch rate (offset-bump at 20 km/h)	110

Figure 4.64. Scaled-down cabin pitch rate (offset-bump at 20 km/h)	110
Figure 4.65. Scaled-down cabin pitch angle (offset-bump at 20 km/h)	110
Figure 4.66. Scaled-down cabin pitch angle (offset-bump at 20 km/h)	111
Figure 4.67. Adaptive gain of roll (offset-bump at 20 km/h)	111
Figure 4.68. Adaptive gain of pitch (offset-bump at 20 km/h).....	111
Figure 5.1. Schematic of the roll-plane AIS-ARS	115
Figure 5.2. Roll stiffness comparison of the roll-plane AIS-ARS	122
Figure 5.3. Damping control system for AIS-ARS	128
Figure 5.4. Schematic of the roll and pitch coupled interconnected suspension	129
Figure 5.5. Comparison of the dual-axle suspension roll stiffness.....	133
Figure 5.6. Nonlinear pitch stiffness.....	136
Figure 6.1. Fully loaded truck built in ADAMS/Car.....	141
Figure 6.2. Truck suspension relative roll angle (steady-turn at 45 km/h)	142
Figure 6.3. Articulation test road profile	143
Figure 6.4. Lateral load transfer (articulation test - fully loaded).....	143
Figure 6.5. Lateral load transfer (articulation test - half loaded).....	143
Figure 6.6. Multiple speedbumps road profile.....	144
Figure 6.7. Truck chassis frame vertical acceleration (speedbumps at 20 km/h)	144
Figure 6.8. Truck suspension relative displacement (speedbumps at 20 km/h)	145
Figure 6.9. Cabin suspension relative roll angle (DLC at 60 km/h)	146
Figure 6.10. Cabin roll rate (DLC at 60 km/h)	147
Figure 6.11. Cabin suspension relative roll angle at different pressures (DLC at 60 km/h).....	147
Figure 6.12. Vehicle trajectory (steady turn at 60 km/h).....	148
Figure 6.13. Cabin suspension relative roll angle (steady turn at 60 km/h)	149
Figure 6.14. Cabin roll rate (steady turn at 60 km/h)	149
Figure 6.15. Cabin roll acceleration (steady turn at 60 km/h)	149
Figure 6.16. Cabin suspension relative roll angle at different pressures (steady turn at 60 km/h) .	150
Figure 6.17. Vehicle longitudinal motion (emergency brake).....	151
Figure 6.18. Cabin suspension relative pitch angle (emergency brake)	151

Figure 6.19. Cabin pitch rate (emergency brake)	152
Figure 6.20. The CAD drawing of the roll-plane interconnected suspension	153
Figure 6.21 CAD drawing of the test-rig arrangement and main components.....	153
Figure 6.22. Laboratory experimental test rig	154
Figure 6.23. Hydraulic cylinder movements: from nominal position to maximum roll angle	154
Figure 6.24. Hydraulic cylinder displacements (a), and equivalent roll angle (b).....	155
Figure 6.25. IS-ARS roll moment.....	155
Figure 6.26. IS-ARS roll stiffness	156
Figure A.1 The truck assembly example provided in ADAMS/Car.....	175
Figure A.2 The redesigned flat-nose cabin system.....	176
Figure A.3 The modelling of the linkage mechanism in ADAMS/Car.	176
Figure B.1. Hydraulic system modelled in MATLAB/SimScape	178

List of Tables

Table 3.1. Parameters of the COE system	44
Table 4.1. The control gains of cabin suspension controllers.....	67
Table 4.2. Performance evaluation (paved road test)	88
Table 4.3. Experimental study: paved road test.....	97
Table 4.4. Percentage of improvement compared to passive suspension (pure roll motion)	101
Table 4.5. Percentage of improvement compared to passive suspension (emergency brake)	105
Table 4.6. Percentage of improvement compared to passive suspension (offset-bump).....	112
Table 4.7. Summary of percentage of improvement compared to passive suspension	113
Table 5.1. Parameters of the AIS-ARS system.....	122
Table 5.2. Parameters of the AIS-ARPS system	134
Table 6.1. Performance evaluation (paved road test)	145

Chapter 1

Introduction

1.1 Motivations

Vehicle suspensions are an interface to isolate road irregularities and forces to the vehicle body that greatly affect ride quality, handling, and roll stability. Compared to passenger cars, rollover accidents in commercial vehicles cause more significant damages and injuries. In the U.S., there are more than 15,000 rollovers of commercial trucks each year [1], resulting in severe accidents and economic losses. Vehicle rollovers occur during extreme cornering maneuvers that are known as untripped rollovers. On the other hand, driving on off-road surfaces, even at low speeds, might result in a rollover due to the high center of gravity (CG) and heavy load, which is called a tripped rollover. Besides, an out-of-control heavy vehicle poses severe hazards and injuries to its operator, other cars, and pedestrians. Hence, the roll stability requirements of heavy vehicles are much stricter than passenger cars. Many studies have been conducted to solve this issue. In general, there are seven main factors affect the commercial vehicle's roll stability, i.e., vehicle mass, the height of the CG location, the height of the vehicle roll center, suspension roll stiffness, effective track width, tire characteristics, and frame flexibility. However, a transportation vehicle has much less control over CG location or roll center height. In addition, the vehicle track width is usually determined by the relevant national standards and regulations, and changes are impractical. Thus, the suspension roll stiffness and tire characteristics are possible means to improve the roll stability of a heavy vehicle. Between these two, the suspension roll stiffness is more vital for maintaining the vehicle roll stability.

The vibration attenuation ability is another critical requirement for heavy trucks. It benefits both ride comfort and cargo transportation. Compared to a passenger car, a truck usually has two suspension systems, the primary and the secondary (cabin) suspensions. The cabin suspension mainly acts as a vibration isolation system between the cabin and the rest of the vehicle. It is designed to improve ride quality by minimizing undesired vibrations without compromising stability and handling performance.

Overall, developing reliable suspension systems is a crucial task in the automotive industry, by which vehicle stability, ride quality, and safety can be guaranteed. In this study, the roll stability of a heavy truck is investigated because of its high CG location and huge loads. An adaptive interconnected suspension with adjustable roll stiffness (AIS-ARS) is developed, which has excellent potential to enhance trucks' roll stability and improve the threshold of rollover. Technically, the interconnected

topology provides a nonlinear roll stiffness characteristic. In other words, the suspension roll stiffness increases with the suspension's relative roll angle. Meanwhile, the adjustable roll stiffness allows users to easily change the suspension roll stiffness according to driver preferences, load, vehicle model, etc. Also, it can be conveniently tuned to fit different vehicle models and applications. Besides, the adaptive damping functionality provides outstanding vibration attenuation performance to avoid cargo damage.

In addition, the key to improving ride quality and lessening motion injuries is to minimize the magnitude of undesired vibrations by refining the cabin suspension system. One of the main approaches is to equip the cabin with a semi-active suspension, in which the damping characteristic is controllable in real-time. A novel coordinated suspension control strategy, integrated Skyhook-LQR, is introduced in this study. As the name implies, this approach takes advantage of both Skyhook and LQR (Linear Quadratic Regulator) to improve the cabin's ride quality in vertical, roll, and pitch directions simultaneously. Considering that it is impossible to have only one set of weightings/gains to achieve optimum performance under various disturbances, two gain-adaptive algorithms (disturbance-based and state-based) are developed to further enhance the overall performance of the integrated Skyhook-LQR. Technically, the gain-adaptive algorithms intelligently adjust the LQR's weighting matrix according to the measured disturbance inputs or vehicle states. One can choose the proper gain-adaptive algorithm depending on the available onboard sensors. Note that these gain-adaptive algorithms can also be used with other optimal control strategies. Furthermore, a novel adaptive interconnected suspension with adjustable roll and pitch stiffness (AIS-ARPS) is introduced since the large roll and pitch angles are another source of discomfort and instability during emergency braking, acceleration, and cornering. As expected, the system enhances the suspension anti-roll and anti-pitch properties, resulting in better cabin attitude and stability control.

All the approaches introduced above are examined through both simulation and experimental studies. A scaled-down cabin prototype is designed and fabricated to evaluate the proposed adaptive damping control algorithms. A horizontal hydro-pneumatic interconnected suspension prototype is designed and assembled to examine the functionalities of the AIS-ARS and AIS-ARPS. Overall, the studies validate the feasibility and effectiveness of the proposed approaches for both research and industrial uses.

1.2 Contributions

Aiming to improve the heavy truck's ride quality and roll stability, the contributions of this work are summarized as follows:

- A novel Skyhook-LQR coordinated suspension control strategy is introduced by taking advantage of the Skyhook and Linear Quadratic Regulator (LQR) approaches, which significantly improves truck cabin dynamics in vertical, roll, and pitch directions. Most importantly, it requires affordable computational power for real-time implementation on automotive-grade microcontrollers.
- Two gain-adaptive algorithms are introduced that aim to further optimize the control performance of the integrated Skyhook-LQR controller. Technically, the weighting matrix of the LQR algorithm is intelligently adjusted according to the disturbances and/or vehicle states measured by the onboard sensors. The integrated Skyhook-LQR coordinated controller, along with the gain-adaptive algorithms, have been verified through experimental studies.
- Two hydro-pneumatic interconnected suspension systems are introduced. One is called adaptive interconnected suspension with adjustable roll stiffness (AIS-ARS), while the other is called adaptive interconnected suspension with adjustable roll and pitch stiffness (AIS-ARPS). The AIS-ARS is implemented on the truck's primary suspension to enhance roll stability, articulation, and vibration attenuation. The AIS-ARPS is used on a cabin suspension to provide anti-roll and anti-pitch properties and improve ride quality. The adaptive damping characteristic provides comfort ride experience, while the adjustable stiffness functionality allows designers and users to customize the suspension rotational stiffness. Moreover, the manufacturers can conveniently tune the suspension stiffness properties for different vehicle types and applications and remove the heavy anti-roll bars. The proposed adjustable roll/pitch stiffness technique has been experimentally tested and verified.

The outcomes from this study have been successfully transferred into patents and journal articles, and more publications are undergoing. The achievements attained so far are listed as follows:

- Patents:

1. A hydro-pneumatic interconnected suspension with adaptive damping and adjustable roll stiffness (filed)
2. A roll stiffness adjustment method for hydro-pneumatic interconnected suspensions (filed)
3. An active stiffness adjustment method for air springs (filed)

- Book:

1. Goodarzi, A., Lu, Y., & Amir Khajepour. (2023). Vehicle Suspension System Technology and Design. Springer Nature. <https://doi.org/10.1007-978-3-031-21804-0>

- Papers:

1. Lu, Y., Khajepour, A., Soltani, A., Li, R., Zhen, R., Liu, Y., & Wang, M. (2023). Gain-adaptive Skyhook-LQR: a coordinated controller for improving truck cabin dynamics. *Control Engineering Practice*, 130, 105365.
2. Lu, Y., Khajepour, A., Hajiloo, R., Korayem, A. H., & Zhang, R. (2023). A New Integrated Skyhook-Linear Quadratic Regulator Coordinated Control Approach for Semi-Active Vehicle Suspension Systems. *Journal of Dynamic Systems, Measurement, and Control*, 145(3), 034501.
3. Lu, Y., Khajepour, A., Soltani, A., Wang, Y., Zhen, R., Liu, Y., & Li, G. (2022). Cab-Over-Engine truck cabins: A mathematical model for dynamics, driver comfort, and suspension analysis and control. *Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics*, 14644193221141596.
4. Lu, Y., Khajepour, A., Liu, Y., & Zhen, R. (2022). Adaptive cabin suspension systems of commercial vehicles: a review of the state-of-art and future trends. *International Journal of Heavy Vehicle Systems*, 29(1), 33-47.
5. Lu, Y., Khajepour, A., & Soltani, A. (2022). A coordinated control system for truck cabin suspension based on model predictive control. *International Journal of Heavy Vehicle Systems*, 29(5).

1.3 Outline

The rest of the thesis is organized as follows:

Chapter 2 reviews the commonly used cabin configurations and their mathematical models, various adaptive suspension systems, vehicle state observers, and interconnected suspension systems introduced in past studies.

Chapter 3 introduces the two control systems developed for a cabin semi-active suspension system: MPC and integrated Skyhook-LQR, together with two gain-adaptive algorithms that can intelligently adjust the control gains to further improve the overall suspension control performance. Meanwhile, the mathematical model of the cabin-over-engine (COE) system is derived based on the Lagrangian methodology considering the tilting mechanism. The model is used in the MPC control and simulations.

Chapter 4 demonstrates the simulation and experimental studies on state estimation, adaptive damping control, and gain-adaptive techniques. Specifically, the experimental test rig and the co-simulation between MATLAB/Simulink and ADAMS/Car are introduced in detail.

Chapter 5 introduces two novel hydro-pneumatic interconnected suspension systems, namely AIS-ARS and AIS-ARPS, and the analysis of their mathematical modelling, adaptive damping, and adjustable stiffness characteristics.

Chapter 6 examines the feasibility and performance of the proposed interconnected suspension systems through simulation and experimental studies.

Chapter 7 concludes the results and contributions of this thesis. It also provides some suggestions for future work.

Chapter 2

Literature Review

This chapter begins by reviewing vehicle suspension systems, truck cabin configurations, and their associated mathematical models. Subsequently, various adaptive suspension systems and vehicle state observers are discussed. Lastly, prior studies regarding interconnected suspension systems are appraised.

2.1 Vehicle Adaptive Suspension Systems

The suspension is one of the critical systems in a vehicle. Generally, its fundamental requirements can be concluded as follows [2]:

- Guarantee vehicle stability when doing turning maneuvers, braking, and accelerating;
- Provide comfort ride experience on various road roughness;
- Maintain proper vehicle posture;
- Avoid excessive suspension deflection that causes hard bump-stop;

With the rapid developments of automated driving, a train-like ride comfort level is required because people expect to read, sleep, and drink/eat in autonomous cars [3]. However, the conventional passive suspension design must compromise between two conflict objectives: ride comfort and handling stability. Thus, active and semi-active suspension systems are developed to optimize suspension performances. Active suspensions are fundamentally different from traditional suspension systems. Technically, a conventional suspension stores energy in a spring and dissipates it via a shock absorber whose damping characteristics are fixed. An active suspension can also store and dissipate energy, but the shock absorbers are replaced with high-speed actuators. It can handle bumps and rough roads much better by adjusting the actuator forces in real time. Fully active systems can control body motions (pitch, roll, and bounce) and wheel hub motion over a wide range of frequencies (0 to 30 Hz). During the 1980s, few vehicles used active suspension technology, most of which were test vehicles from Lotus and Volvo [4]. During the 1990s, the interest in active suspensions widely declined due to their cost, complication, and high power consumption. To solve these issues, semi-active suspensions received a lot of attention in the automotive industry. Nowadays, they are widely used in high-end cars. The main working principle of a semi-active system is the damping control that provides a distinct advantage in both ride quality and handling stability. This system includes adaptive shock absorbers as its main part,

multiple sensors, such as height sensors and IMU, to detect road conditions and vehicle motions, and an Electronic Control Unit (ECU) that controls the damping properties of the adaptive shock absorbers to improve vehicle comfort and handling stability.

2.1.1 Passenger car adaptive suspension

Many control approaches have been applied to adaptive suspension systems of passenger cars, such as Skyhook, Groundhook, LQR, Linear Quadratic Gaussian (LQG), Model Predictive Control (MPC), Acceleration Driven Damping (ADD), Linear Parameter Varying (LPV), H_∞ , fuzzy logic, etc. This section focuses on the adaptive damping control for improving ride comfort that can be implemented in either passenger cars or commercial vehicles [5].

The Skyhook and Groundhook approaches were widely used in adaptive suspensions, and their hybrid algorithms were introduced in some studies [6]. For example, Liu et al. [7] introduced a generalized Skyhook-Groundhook (GenHook) hybrid strategy, in which the weighting coefficient of Skyhook and Groundhook in impedance-based control was determined via the 95th percentile aggregate fourth-power force. The simulation results demonstrated that the proposed controller could significantly enhance ride comfort and maintain stability under different working conditions. Past studies showed that Skyhook performed very well in minimizing the sprung mass' vertical acceleration around its resonance frequency but not in higher frequency ranges. However, the ADD approach has complementary characteristics to Skyhook (SH), which is good at isolating the vibrations beyond the sprung mass' natural frequency. Thus, the combination of these two comfort-oriented control algorithms was proposed in the literature, namely SH-ADD. It aimed to provide the quasi-optimal performance of ride quality. A crossover frequency α was selected near the first-order resonance frequency where SH and ADD were switched. In detail, the control algorithms of Skyhook, ADD, and SH-ADD are presented as follows [8,9]:

$$c_{SH} = \begin{cases} c_{min}, & \text{if } \dot{z}_s(\dot{z}_s - \dot{z}_u) < 0 \\ c_{max}, & \text{if } \dot{z}_s(\dot{z}_s - \dot{z}_u) \geq 0 \end{cases}$$

$$c_{ADD} = \begin{cases} c_{min}, & \text{if } \ddot{z}_s(\dot{z}_s - \dot{z}_u) < 0 \\ c_{max}, & \text{if } \ddot{z}_s(\dot{z}_s - \dot{z}_u) \geq 0 \end{cases}$$

$$c_{SH-ADD} = \begin{cases} c_{SH}, & \text{if } \ddot{z}_s^2 - \alpha^2 \dot{z}_s^2 < 0 \\ c_{ADD}, & \text{if } \ddot{z}_s^2 - \alpha^2 \dot{z}_s^2 \geq 0 \end{cases}$$

Moreover, Nie et al. [10] developed a modified SH-ADD with a frequency selector for a semi-active suspension to balance ride quality and road holding. The algorithm was validated through numerical analysis and experiment on a quarter-car test rig, which validated that the proposed controller could balance ride comfort and road holding around first- and second-order resonance frequencies. Besides, it reduced the chattering of the sprung mass acceleration, which was a common issue of the ‘on-off’ control algorithms.

In addition, H_2 and H_∞ control approaches were frequently mentioned in past studies, which included formal consideration of robustness with respect to model/parameter uncertainties [11]. Therefore, they were often referred to as robust controllers. Li et al. [12] developed an output feedback based H_∞ controller for an active quarter car suspension system. It was designed to ensure the asymptotic stability of the closed-loop system with H_∞ disturbance attenuation level and output constraints, and to improve ride comfort and road holding. Moreover, Du et al. [13] implemented a H_∞ optimal controller on a hydraulically interconnected suspension designed for three motion modes (bounce, pitch, and roll) using a common quadratic Lyapunov function. Li et al. [14] introduced a state feedback H_2 controller for a quarter-car active suspension system with an adjustable inerter, a spring, and a damper in parallel. It aimed to improve ride quality, suspension deflection, and tire deflection performance. In the simulation, the inerter tracked the desired control force generated by the state feedback H_2 algorithm. The results demonstrated that the ride quality and tire deflection could be optimized with slight deterioration of the unsprung mass’ natural frequency, and the suspension deflection was improved at high frequencies.

Some novel strategies were also introduced. For example, Zheng et al. [15] developed a novel framework of cloud-aided nonlinear active full car suspension, in which the backstepping control strategy was employed to handle the nonlinear characteristics updated in a remote cloud. The control reference was chosen based on the suspension aggregate information and road conditions, by which the control algorithm was processed on the central computational server. The uncertain parameters and unmodeled dynamics were dealt with by an adaptive algorithm. The simulation based on a 7 DOFs full car model demonstrated that the proposed strategy improved ride comfort by more than 80% compared to the passive suspension. Besides, Tang et al. [16] investigated a Takagi-Sugeno fuzzy controller for a semi-active quarter-car suspension equipped with an MR damper. The proposed approach was evaluated on a quarter-car test rig and compared to a Skyhook controller. The experimental results demonstrated that the TS fuzzy controller could effectively attenuate the undesired vibration by 10%

to 30%. Moreover, Zhang et al. [17] introduced an adaptive neural network control scheme for an active suspension system by using bioinspired nonlinear dynamics. In detail, a novel constructive predictor was designed to address the input delay. Then, the neural networks were adopted to approximate the uncertain dynamics. A finite-time adaptive control was implemented to update the neural network weights and minimize vibrations. The theoretical analysis and experimental results vindicated the efficiency of suppressing vibrations and low energy consumption by more than 44%. Additionally, Nguyen et al. introduced a fractional-order derivative-based sliding mode control (FD-SMC) for a half-car semi-active suspension system with MR dampers [18], which aimed to minimize the sprung mass' accelerations in vertical and pitch directions. The proposed controller consisted of I-SmDs, which relied on measured data processed by the new filter CoFilter [19] and the FD-based SMC technique. The FD-SMC determined the control force, and I-SmDs estimated the current intensity applied to MR dampers to generate the desired damping forces. The experimental test demonstrated that the proposed controller could deal with the dynamic response of the system and the noisy measurement very well. Specifically, the vertical and rotational accelerations of the sprung mass were reduced to a large extent and the road-holding ability was improved to provide safer driving conditions. An objective-oriented hierarchy control strategy that integrated with active braking and active front steering systems was developed by Lu et al. [20], in which a fuzzy control approach was used for the semi-active suspension system and a sliding mode control technique was utilized for braking and steering systems. The simulation results demonstrated that the proposed global integrated controller could effectively improve the ride comfort, lateral stability, and braking safety. Coric et al. [21] introduced a collocation-type control variable optimization method to enhance vehicle ride comfort and maintain wheel-holding ability through a fully active suspension (FAS) system. A quarter-car and a full 10 DOFs vehicle models were used in this study. In addition, the cost function was extended to include the FAS energy consumption and wheel damage resilience costs. Na et al. [22] utilized a novel approximation-free control (AFC) technique to improve the efficiency and reliability of quarter-car active suspension systems. The major advantage of this approach was that it incorporated the unmolded dynamics of the hydraulic actuator and eliminated the need for laborious parameter tuning. Furthermore, the control strategy employed prescribed performance functions (PPFs) that ensured both transient and steady-state suspension performance for ensuring the safe operation of the adaptive suspension system. The experiments based on a practical quarter-car active suspension test rig demonstrated that this control technique

outperformed several other control methods and provided superior performance with improved computational efficiency.

▪ **Model-based control**

Different model-based control techniques have been implemented in vehicle adaptive suspension systems. Among them, the MPC and LQR were the most popular algorithms in past studies. These two control approaches can solve multi-objective optimization problems, by which the vehicle dynamics and ride comfort in vertical, roll, and pitch directions can be analyzed and optimized. For example, Koch and Kloiber [23] introduced an adaptive controller which dynamically interpolated between differently tuned LQRs governed by the dynamics wheel load and the suspension deflection. Its performance was examined on a quarter-car test rig with a linear electrical motor. Compared to a fast adaptive Skyhook controller, the proposed approach could overcome conservatism and offer significant ride comfort while keeping the limits on the suspension deflection and the wheel load. Meanwhile, a framework was designed to ensure the stability of the adaptive suspension system.

Compared to the LQR, the MPC algorithm can adequately handle the semi-active constraint, also known as the damping dissipativity constraint [24]. Nguyen et al. [25] designed an MPC controller for a semi-active suspension based on a 7 DOFs full car model. An observer that could estimate the road profile over the prediction horizon was introduced. This MPC controller with road estimation gave a trade-off between ride comfort and handling stability. Meanwhile, it guaranteed the physical constraints of semi-active dampers. The simulation results illustrated the effectiveness of the proposed approach. Similar work has also been done by Gohrle [26]. However, the high computational cost of the MPC algorithm made it difficult and expensive to be implemented in practical applications. Therefore, various strategies have been proposed to maintain the accuracy of the prediction but reduce the computational cost at the same time, such as hybrid MPC [27], fast MPC [28], explicit MPC [29], nonlinear MPC [30], etc. Johan et al. [31] introduced an explicit MPC for an active suspension system. It was assessed through simulation and experiment on a sport utility vehicle (SUV). The simulation results showed that the vibrations in vertical, roll, and pitch directions could be reduced by 10% to 35% compared to the passive suspension. The experimental results validated that the proposed controller sufficiently minimized the vibration in the range of 0~15 Hz. However, this algorithm was processed on a dSPACE MicroAutoBox with 16MB flash memory which was more powerful than an automotive-grade microcontroller. Morato et al. [32] designed a fast real-time LPV MPC controller for a semi-

active suspension with a sampling frequency of 200 Hz. The simulation results illustrated that the vertical vibrations were attenuated to a large extent, but the roll performance was at the same level as the passive suspension. The experimental test was done on a testbed called INOVE Soben-Car, but the algorithm was processed on a 2.4GHz 8GB RAM Macintosh computer instead of an automotive-grade microcontroller. Furthermore, a hybrid horizon-varying (HV) MPC controller with road preview was developed for autonomous vehicles on uneven roads [33]. This comfort optimization strategy combined vehicle speed planning and road-preview semi-active suspension control with a novel road data processing method, by which the control system could fully utilize the road information with a fixed preview length while adapting to speed variation. The simulation results verified the benefits of the hybrid HV-MPC method, especially in improving ride comfort and realizing multi-objective optimization (vertical vibration, driving time, and speed variation).

The damping forces were mostly considered as the control inputs in previous studies, which resulted in linear control plants with nonlinear state-dependent constraints. This issue increased the requirement of computational memory. Two different techniques were introduced by Kjellberg and Sundell [30] to solve this problem. Firstly, a full-complexity nonlinear MPC with 18 states was proposed. The full-car model was reformulated, which used damping coefficients as the control signal instead of damping forces. This method resulted in a nonlinear control plant with linear constraints. Secondly, the full car model was simplified by removing the wheel dynamics to reduce the execution time [34]. Hence, the number of states was reduced to 15. Meanwhile, the sampling time and prediction horizon were tuned to decrease the computational cost. The proposed real-time nonlinear MPC was finally implemented on a Volvo S90 equipped with semi-active dampers and a vision system, but the controller was run on a MicroAutoBox. Overall, it is noticeable that the real-time MPCs introduced in the past studies were not comprehensively examined on any automotive-grade microcontrollers. From the authors' point of view, the reasons can be concluded as follows:

- 1) The QP solvers employed in most of the studies are ready-to-use packages/software, such as *quadprog* provided by MATLAB, *qpOASES* [35], *ODYS* [36], *NASOQ* from the University of Toronto [37], *OSQP* from the University of Oxford [38], etc. They usually require relatively high flash memories for processing, which causes a big problem for practical application.
- 2) The prediction algorithm is an important feature of MPC, but it also needs a huge amount of flash memory. Meanwhile, the computational cost is highly dependent on the prediction horizon setting.

3) Some studies focused on full car dynamics, resulting in a complicated mathematical model with more states that naturally increased the computational cost.

▪ **Gain-adaptive approach**

Most existing works focused on non-adaptive control techniques, in which the controller gains were independent of the disturbance inputs. Since the driving conditions are randomly changed, using only one set of gains is impossible to achieve the optimum control performance all the time. Hence, some gain-adaptive algorithms are introduced in the literature, which can be classified into two main categories, i.e., parameter-adaptive suspension due to system nonlinear and uncertain properties [39] and road-adaptive suspension because of randomly changed road surfaces. Considering various driving conditions of a commercial vehicle, this study mainly focuses on road-adaptive techniques by adjusting the gains and/or weighting matrices of the damping control algorithm.

As one of the most commonly used suspension control techniques, Skyhook control was widely utilized for developing adaptive control systems in the literature. Nguyen et al. [40] designed a road frequency adaptive control system for a quarter-car semi-active suspension to improve both ride comfort and handling performances in all frequency ranges of road disturbances. Their control law was extended from the conventional Skyhook approach, and its gains were scheduled based on the estimated disturbance frequencies. To determine the gains of the modified Skyhook algorithm, a fully active LQR controller was utilized to generate the optimum damping force (benchmark). The scheduled gains aimed to minimize the force differences between the modified Skyhook and the active LQR using the Minimum Norm Criterion [41]. The performance of the proposed controller was evaluated through numerical simulations and only compared to the passive suspension system, which showed that the adaptive Skyhook could provide better ride comfort and handling ability. Similarly, Hong et al. [42] developed a road-adaptive Skyhook controller for a semi-active Macpherson suspension. The proposed approach extended the conventional Skyhook-Groundhook control scheme and scheduled its gains according to road conditions based on the measured sprung mass' vertical acceleration. Five sets of optimal gains were presented for different ISO roads from class A (very good) to class E (very poor) to describe the road roughness. The hardware-in-the-loop simulations examined the control algorithm on a quarter-car test rig, demonstrating that the proposed semi-active suspension achieved a competitive control performance by adopting the road-adaptive law. Moreover, Yi and Song [43] developed a road-adaptive Skyhook controller using Neural Network based on a road detection algorithm. The modified

Skyhook approach considered both ride quality and dynamic tire force. The proposed road detection algorithm determined and updated the Skyhook gains according to the estimated frequency contents of road disturbances based on the measured sprung mass acceleration. Its performance was validated through a quarter-car test rig. The experimental results showed that the proposed road-adaptive Skyhook controller provided adequate damping for the wheel hop frequency and enhanced the ride comfort and road holding compared to the classic Skyhook approach.

Besides the Skyhook approach, other control techniques were utilized to achieve road-adaptive functionality. Koch et al. [44] designed a road-adaptive LQR controller for a quarter-car active suspension system. With a switching control structure, the adaptive algorithm scheduled the LQR gains according to the dynamic wheel load and the suspension deflection. Specifically, there were six pre-defined sets of LQR weighting matrices for different primary control objectives, such as comfort-oriented, intermediate, safety-oriented, etc. The switching logic switched between six sets of weighting matrices to maximize the overall suspension performance. The simulation results showed that the proposed road-adaptive LQR controller improved the ride comfort by 20% compared to the passive suspension and 11% compared to the non-adaptive LQR approach.

Similarly, Fialho and Balas [45] presented a framework for developing a road-adaptive suspension controller via LPV control and nonlinear back-stepping techniques. The gains of the LPV algorithm were adaptively switched between two pre-tuned values based on the measured suspension deflections, and a first-order filter was implemented to avoid discontinuous gain transition.

2.1.2 Cabin adaptive suspension

Truck drivers are constantly exposed to undesirable vibrations caused by uneven road surfaces, especially in long-haul transportation. Extensive research has been done on lower back problems experienced by truck drivers due to vibrations created when driving on rough roadways, braking, turning maneuvers, and engine operations [46]. Quantitatively, intensive low-frequency vibrations from 0.5 Hz to 10 Hz are transmitted to the driver seat if the road-induced vibrations are in the frequency range of 2 Hz to 20 Hz [47,48]. As a result, drivers suffer from motion injuries, such as neck and back pain. Statistically, male truck drivers are four times more likely to suffer from a herniated disk than the average male [49]. According to Deprez et al. [46], lower back pain is a major cause of industrial disability in middle-aged workers accounting for 20% of all work-related injuries. Furthermore, constant vibrations can also lead to driver fatigue and possibly have further implications on driving

safety. Unsurprisingly, the negative impact on the economy is substantial. Work-related injuries experienced by truck drivers in the United States result in an estimated loss of \$90 billion annually [46]. The European Parliament continues to work on regulations to restrict the harm of whole-body vibration. One of the results is the Machinery Directive 89/392/EEC and its amendments of 1996 which force vehicle manufacturers to develop vibration isolation techniques to reduce the effects of noise and vibration to the lowest possible level [50].

At the end of the 19th century, the first truck and passenger car emerged simultaneously [51]. Passenger cars and buses were designed for passenger transportation, which require a high level of comfort. While trucks were developed to transport goods as efficient and durable machines [52]. In the 1920s and 1930s, the first sleeper berth was developed [53] to accommodate drivers frequently needing to pass the night in their trucks. Cabins were initially welded to the chassis frame, which resulted in an uncomfortable ride experience. No further changes were made to the truck cabin suspension until the 1960s. The rubber mounts installed between the chassis frame and the cabin substantially improved ride comfort. In the same period, the first tilt cabin was invented. It is still one of the standard configurations used today because the tilt structure greatly facilitates the drive-line inspection [51]. During the 1970s and 1980s, various steel and air springs were used to refine the cabin suspension design.

Looking back at history, truck manufacturers put a lot of attention and effort toward enhancing drivers' ride comfort, such as developing seat suspension systems, cabin suspension systems, and improved ergonomic layouts of the cabin interior [54]. Most seat suspension systems are designed to isolate vertical vibrations only, but operators are exposed to vibrations in all three planes (vertical, pitch, and roll). As a result, cabin suspension systems have received a lot of attention over the past two decades as they can eliminate vibrations in all three directions.

Nowadays, there are three types of cabin suspension arrangements in general: front, rear, and full [55]. In the USA, most tractor semi-trailers are equipped with rear cabin suspension, and full cabin suspension is commonly used in Europe [56].

Front suspension: In the front cabin suspension system, as shown in Figure 2.1(a), the front side consists of springs and shock absorbers. The rear part is pinned to the chassis frame, allowing small pitch and vertical movements. However, this arrangement is hard to install because the flexible control linkages are needed, and the crashworthiness results in additional constraints on the front cab isolators

[57]. Nevertheless, the front cabin suspension system can effectively attenuate vertical vibrations and reduce the fore-aft motion. Flat-nose tractor manufacturers can consider it an alternative to full suspended cabin suspension.

Rear suspension: As shown in Figure 2.1(b), the rear cabin suspension configuration is widely used in the commercial vehicle industry because it is relatively easy to install. Usually, the cabin's forepart is pinned to the chassis frame, while the rear part is equipped with coil or air springs and shock absorbers. The guiding mechanism usually includes transverse arms, a Panhard rod, and/or two lateral dampers [58]. Note that the suspension damping property is mainly constrained by stroke length, and the stiffness is highly related to the cabin's vertical and pitch natural frequencies [59]. In general, the natural frequency of the cabin suspension with air springs often lies in 1~1.4 Hz [60], and that with steel springs lies in 1.8~3 Hz [61]. Moreover, the rear cabin suspension attenuates the backslap since it is partially decoupled from the frame bending, but it increases vertical vibrations simultaneously [57]. Due to the relatively low roll stiffness, the truck steering maneuvers result in additional roll motion and increased pitch motion when braking or accelerating. This results in a significant drawback of the cabin suspension system: poor attitude behaviour. In the early years, the roll motion was limited by using a stiff anti-roll bar; however, this was once again a source of discomfort for the driver [51]. Based on this configuration, the tilted cabin gains much attention in the industry because it is easy to install and facilitates the drive-line inspection. A helpful review of the tilting cabin mechanism can be found in [62].

Full suspension: The full (4-point) cabin suspension system emerged to further improve heavy-truck drivers' ride comfort. As shown in Figure 2.1(c), all four corners are installed with coil [63] or air springs [64] and dampers. Its guiding mechanism generates a slight radial movement which causes the fore-aft movement [58]. Nowadays, lateral shock absorbers are sometimes used in addition to stabilizers to optimize the cabin's roll dynamics further [65,66]. Air springs are widely used in the full cabin suspension system. In this case, the height levelling system requires external valves, linkage components, and pipelines to connect airbags and air supplies. It can maintain a constant suspension natural frequency despite changes in the cabin load. Besides, the cabin tilting function can be achieved using a front tilting arrangement [67], which can be tilted up to 45 degrees for chassis maintenance purposes [68]. The full-floating arrangement provides the best ride quality because of its potential to improve ride comfort in vertical, roll, and pitch directions. Meanwhile, it enhances crash safety and reduces the dynamic forces at mounting points [58]. Hence, the full cabin suspension system is

frequently equipped in heavy vehicles nowadays [69] and upgraded to various adaptive suspension systems.

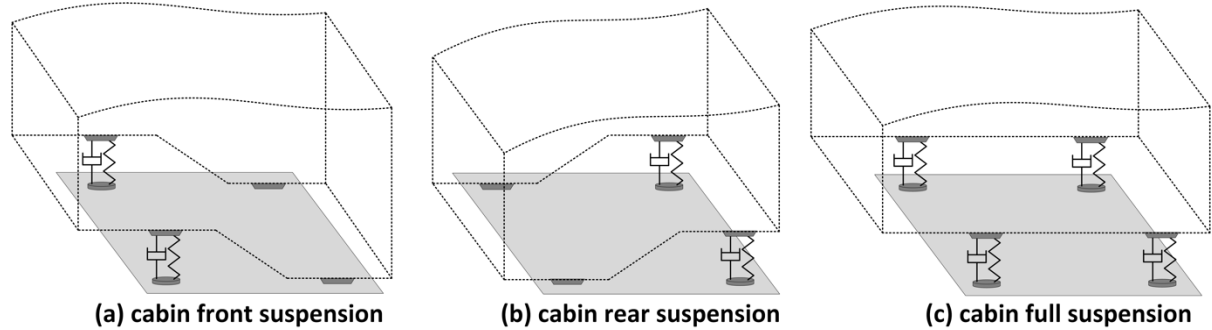


Figure 2.1. Cabin suspension configurations

In addition to the cabin suspension systems, there are two commonly used cabin styles: cab-over-engine (COE) and long-nose cabin. Cab-over-engine, also known as flat-nose cabin, is widely used in Asia and Europe because of their strict regulations for truck length. Besides, it is popular in the commercial vehicle industry because of its easy installation and improved maneuverability. Many manufacturers produce various COEs, such as Volvo, Scania, Mack, Isuzu, Hino, Fuso, Iveco, etc. The flat-nose cabin sits above the engine and hence provides better visibility. Long-nose truck cabin features a design where the engine is mounted in front of the driver's compartment and has an elongated hood that extends out in front of the vehicle. Some car manufacturers, such as Freightliner, Kenworth, Peterbilt, and Western Star, are well-known for producing heavy-duty trucks with traditional long-nose designs. This type of cabin design is popular in North America, particularly for heavy-duty trucks used in long-haul transportation. The long-nose design provides several benefits, including improved engine cooling, easier maintenance access, and better weight distribution. Additionally, many drivers prefer this type of cabin's aesthetic appeal and classic style.

In past studies, the cabin was usually modelled as a three DOFs rigid body system in Figure 2.2(a) or even a simple one-DOF mass-spring-damper base excitation system [70]. In detail, one widely used four-point cabin suspension was modelled as a simplified system of the passenger car suspension shown in Figure 2.2(b) without considering the unique tilting mechanisms in the cabin suspension [71,72]. For example, Ekberg and Hansson [73] used this simple four-point cabin suspension model for control system development, in which each strut consisted of a spring and an adaptive damper. Similarly, Marcu [74] developed a cabin suspension model for a class 8 Volvo VN 770 semi-truck. At the front,

the cabin was mounted to the frame through bushings on each side that were equivalently considered as stiff springs and dampers. At the rear, the cabin suspension contained two air springs and two adaptive dampers for control purposes. For simplification, some studies were mainly focused on the cabin's vertical dynamic, while its roll and pitch motions were not considered. Iersel [75] and Soliman [76] developed a four DOFs quarter-truck model and a nine DOFs half-truck model, respectively. Both truck models were equipped with a single DOF cabin model with a variable rate damper. Similarly, Hansson [77] introduced a disturbance-adaptive active cabin suspension system for agricultural tractors based on a single DOF cabin model, whereby the vertical actuator force worked in parallel with a passive linear spring.

Vibration exposure to seated drivers causes drive fatigue and leads to health disorders. This issue is particularly critical for heavy truck and off-road vehicle operators, so it is essential to provide a healthy ride quality for long-haul drivers [78]. Although the passive full cabin suspension system provides acceptable ride comfort and various tuning methods have been investigated by researchers [63,79,80], the overall cabin dynamics can be further optimized by applying advanced control systems. This study reviews and compares suspension control techniques for improving ride comfort, such as Skyhook [81], LQR [73], ADD [75], LPV [75], H_∞ [82], fuzzy logic [83], etc. These adaptive cabin suspension systems are thoroughly reviewed and discussed in the following paragraphs.

▪ Skyhook control

In the early 1970s, Karnopp and Crosby [84] introduced the concept of the semi-active suspension system based on the well-known Skyhook control. Because of its performance and computational efficiency, the Skyhook control approach is widely implemented in adaptive cabin suspension systems. Based on the cabin suspension system shown in Figure 2.3, the concept of the Skyhook theory is that the cabin is connected to an imaginary sky through a damper c_{sky} , and a spring k_s installed between the cabin and the chassis frame, as shown in Figure 2.4. The detailed introduction can be found in Section 3.2.2.1. This comfort-oriented control law focuses on reducing the sprung mass acceleration by generating a damping force proportional to the absolute sprung mass velocity. Due to its popularity and efficiency, the Skyhook approach is often utilized as the benchmark.

Tong and Amirouche [81] introduced a Skyhook-based semi-active cabin suspension to achieve optimal vertical and pitch dynamics. In this suspension system, the cabin was supported by both front and rear suspensions that consisted of passive springs and adaptive dampers, as shown in Figure 2.5. A

3-axle tractor semi-trailer with a separated cabin was modelled as an 8 DOFs system. The simulation results showed that the driver's absorbed power dropped by 30% using the proposed controller, and both vertical and pitch accelerations were reduced. With the addition of a controllable seat suspension system, more than 60% of the driver's absorbed power could be reduced. Moreover, Nakano et al. [85] designed a self-powered active cabin suspension system based on the conventional Skyhook approach. An electric generator was installed in the primary suspension system to regenerate vibration energy and store it in a capacitor. The actuator installed in the cabin suspension achieved active vibration control using the energy stored in the capacitor. Numerous simulations showed that the self-powered active cabin suspension system had better vibration isolation performance than a semi-active or passive system.

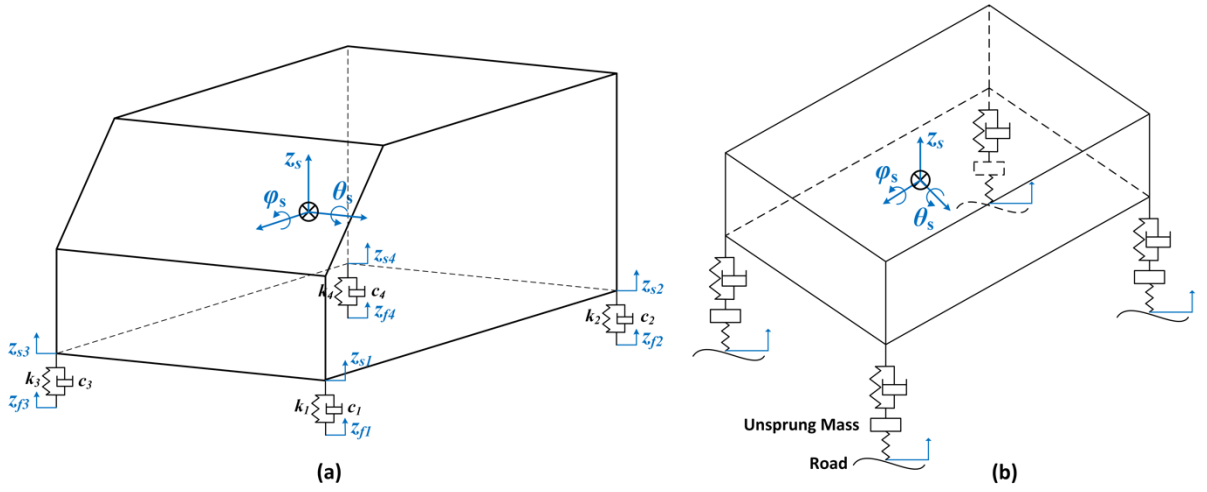


Figure 2.2. The three DOFs model in the literature: (a) four-point cabin, (b) passenger car

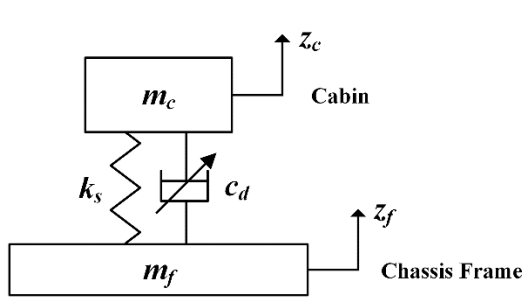


Figure 2.3. Quarter-cabin model

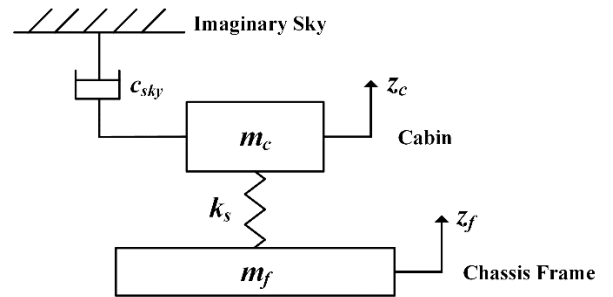


Figure 2.4. Skyhook control theory

Several modified Skyhook algorithms have been introduced in the literature to further optimize the control performance of the Skyhook strategy. Marcu [74] proposed a variation of the Skyhook theory,

named no-jerk Skyhook, which was applied to a rear semi-active cabin suspension system of a class 8 Volvo VN 770 semi-truck. A hierarchical semi-active control (HSAC) approach was developed and implemented to allow the no-jerk Skyhook controller to adapt to various road conditions. The cabin was modelled as a 3-DOF rigid body. In the front, the cabin was mounted to the frame through bushings modelled as relatively stiff springs and dampers on each side. In the rear, the suspension system contained two air springs and two dampers. The corresponding bond graph model was introduced in this work. The simulation results showed that the proposed controller was most effective in the low-frequency range (3 Hz to 5 Hz), but there were no benefits at high-frequency ranges (above 10 Hz). The laboratory testing also showed that the no-jerk Skyhook controller significantly improved the ride comfort around its natural frequency, especially when dealing with transient excitations. The no-jerk policy provided noticeably smoother control signals than the classic skyhook law. Nevertheless, this work mainly focused on the cabin's vertical motions, while its roll and pitch dynamics were not fully considered that were sources of discomfort because of the cabin's high CG location. Ekberg and Hansson [73] applied the modal Skyhook theory for optimizing 4-point cabin active and semi-active suspension systems. As shown in Figure 2.6, each strut consisted of a spring and an adaptive damper. The modal Skyhook was developed based on the motions at the cabin's CG location instead of only relying on the suspension deflection velocities. This method was then implemented in ADAMS/Car for analysis. In some cases, the roll angle barely changed with an individual Skyhook controller, but the vertical acceleration and the pitch angle were considerably suppressed. The modal Skyhook controller could flexibly reduce cabin roll and pitch angles by tuning its user-defined parameters. Overall, the simulation results showed that the overall performance of the modal Skyhook was much better than that of the traditional Skyhook, especially in terms of the cabin's roll and pitch dynamics.

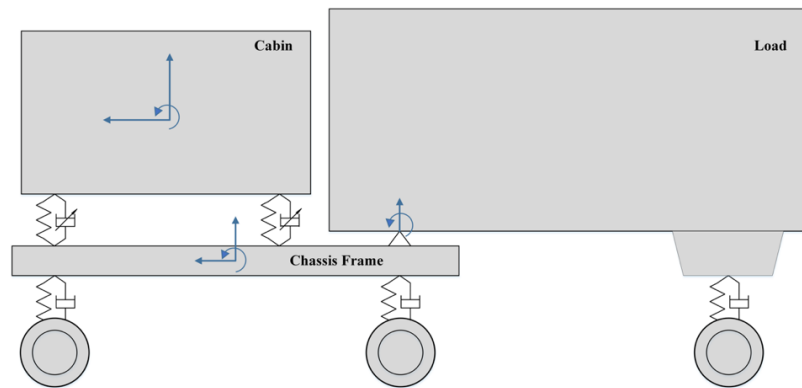


Figure 2.5. Sketch of the truck with cabin suspension

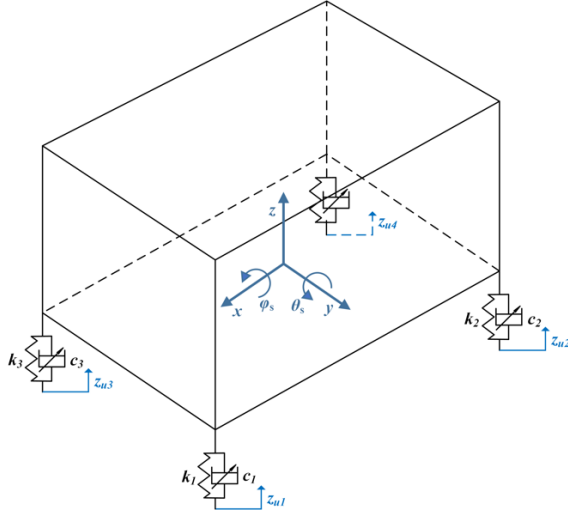


Figure 2.6. Three DOFs cabin model

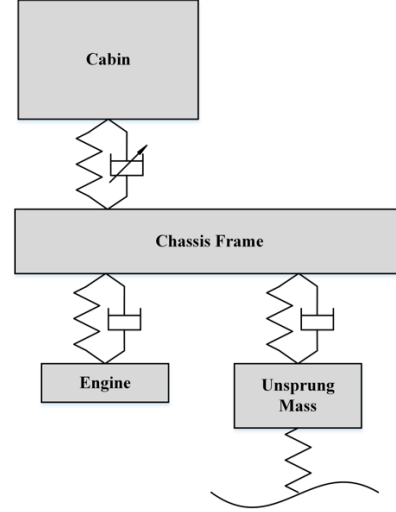


Figure 2.7. Quarter-truck model

▪ Other control approaches

In addition to the modal Skyhook strategy, Ekberg and Hansson [73] implemented the LQR approach for developing a 4-point cabin active suspension system. In their work, the objective function included the cabin's displacement, angles, and velocities in vertical, roll, and pitch directions. The control signals (damping forces) were calculated to minimize the terms in the objective function. The LQR controller was evaluated in ADAMS/Car, which showed that the active LQR controller improved the ride comfort by more than 30%, and the cabin's roll and pitch dynamics could be adjusted by tuning penalty matrices. Overall, the active modal Skyhook and the active LQR provided comparative performances.

Iersel [75] focused on minimizing the undesired vibration of the cabin in the vertical direction. A 4-DOF quarter-truck model was used with a variable rate damper in the cabin suspension, as shown in Figure 2.7. Two popular control theories were tried in his work: mixed Skyhook and ADD control and the LQ/LPV approach. The altered ADD algorithm was applied on a semi-active cabin suspension system as follows:

$$F_d = \begin{cases} F_{max}, & \text{if } \ddot{z}_c(\dot{z}_c - \dot{z}_f) \geq 0 \\ F_{min}, & \text{if } \ddot{z}_c(\dot{z}_c - \dot{z}_f) < 0 \end{cases}$$

which has a similar formulation as the 2-state classic Skyhook algorithm. Interestingly, the traditional Skyhook strategy performed better in low-frequency ranges, while the ADD algorithm acted better in

high-frequency ranges. Hence, combining these two methods had a more significant potential for improving overall ride comfort.

Besides, the proposed LQ/LPV strategy was inspired by H_∞ /LPV. The desired damping force was defined as

$$F_d = c_{nom}(\dot{z}_c - \dot{z}_f) + F_c$$

in which the additional force F_c was calculated by an LQ-optimal controller by solving the algebraic Riccati equation. An LPV parameter ρ was later introduced to capture the system's nonlinear characteristics, which consisted of force boundaries and damper dissipativity constraints. With the proposed LPV parameter, the penalty matrix R of the LQ controller was re-defined as

$$R(\rho) = 1 \times 10^{-3} + \rho \cdot 10^3$$

by which the limitations and force change rate characteristics of semi-active dampers could be satisfied. For example, with the increase of ρ , the control force $F_c \rightarrow 0$, which caused the suspension to act as a passive system. The simulation results showed that the LQ/LPV strategy performed significantly better than the 2-state Skyhook and ADD controllers. Additionally, the author pointed out that those so-called clipped control strategies were not optimal for semi-active suspension systems. The constraints inherent to the semi-active dampers should be considered during the control design process.

Hiromatsu et al. [82] developed a fully active electromechanical cabin suspension actuated by an electric motor. The controller was designed based on the H_∞ feedback control theory and tested on a full-scaled experimental platform and a truck. The motor's output torque was converted to vertical force with a ball screw and an air spring in parallel. The control commands at all four suspension corners were augmented by applying frequency-weighted gains. The control signals were calculated by solving the feedback H_∞ problem to minimize the given objective function under the worst possible disturbance. The robustness against cabin mass perturbation was confirmed on the Nyquist diagram. The experiment results showed that the proposed controller eliminated cabin vibration and the effects of chassis frame bending. Moreover, this active suspension system could be economically produced using 100W class AC servo motors. It should be noted that this study only focused on the vertical dynamics of the cabin excluding roll and pitch directions.

Soliman [76] developed a hybrid cabin suspension system based on a simple linear feedback control law whereby the vibration energy was stored in the capacitor and provided to the DC motor for cabin

control. The proposed control system was simulated on a 9-DOF half-truck model. As expected, the simulation results showed better ride comfort regarding the cabin's vertical acceleration compared to a semi-active suspension equipped with a two-state Skyhook controller. Since the cabin suspension was modelled as a 1-DOF model, its roll and pitch motions were not considered.

As an alternative to damping control, some studies have focused on adaptive air spring stiffness control. Yan et al. [83] introduced a genetic algorithm optimized fuzzy logic controller (GA-FLC) to improve the cabin's vertical dynamics by adjusting the spring rates. Its control performance was evaluated by a nonlinear cabin suspension model built in ADAMS under four different road excitations. The simulation results showed that both the cabin's vertical displacement and acceleration decreased by 30% to 40% after equipping the proposed GA-optimized fuzzy controller. Moreover, Graf and Maas [86] connected air springs and the air supply through fast-switching valves. In this way, the mass flow of the compressed air was controlled, as well as the spring rate. By applying a hybrid Skyhook and Groundhook principle, the cabin accelerations in vertical, roll, and pitch directions were attenuated. According to the road tests with the phototype vehicle, the overall frequency-weighted acceleration [87] was reduced by 40%.

Instead of studying the cabin system of commercial trucks, some researchers have focused on the cabin developments of vibratory rollers in vertical and pitch directions [88,89]. Nguyen et al. [90] applied the PID-fuzzy control to the cabin isolation mounts of soil compactors. A 7-DOF nonlinear single drum vibratory roller was built to evaluate the control performance. The proportional, integral, and derivative gains of the PID were adjusted by the fuzzy control law. The simulation results showed that the ride comfort could be remarkably improved in both vertical and pitch directions, especially in a low-frequency range below 4 Hz. Similarly, Jiao and Nguyen [91] compared this PID-fuzzy controller against two different control approaches: the multi-objective genetic algorithm (MOGA) and the genetic algorithm based PID (GA-PID). According to their simulation results, the fuzzy logic PID (FLC-PID) approach reduced the weighted RMS acceleration by 30% in the vertical direction and more than 45% in the pitch direction, which provided better control performance than GA-PID and MOGA approaches.

▪ **Road adaptive control**

Previous research showed that active and semi-active cabin suspension systems could significantly improve ride quality. However, their control gains and weighting matrices were fixed. Generally, a

controller with fixed parameters must be well-tuned to avoid harsh obstacles and high-level vibrations, which limits the damping potential under usual driving conditions [77]. Thus, an optimal controller with road-adaptive gain-tuning is a reasonable solution [92], by which the adaptive gains are updated based on disturbances and suspension travels.

Iersel [75] pointed out that the suspension deflection significantly impacted the driver's comfort, but the available working spaces were not fully used with fixed control gains. A road adaptive approach was then introduced and applied to 2-state Skyhook and LQ/LPV controllers. The simulation results showed that with different levels of road roughness (especially for lower ones), the proposed gain-adaptive semi-active suspension systems reached a better ride comfort with up to 45% improvement compared to non-adaptive ones. However, the proposed gain scheduling method under discrete obstacle conditions was not thoroughly studied.

Hansson [77] introduced a disturbance-adaptive active cabin suspension system for agricultural tractors based on the time-invariant state feedback theory. The vertical actuator force worked in parallel with a passive linear spring. An adaptive algorithm based on varied parameters in the penalty matrices was applied. The gains of both controller and observer were adjusted according to the chassis frame's motion. The control feedback gains were then calculated with the updated penalty matrices by iterating the Riccati equation. The gains of penalty matrices depended on the suspension travel, so the controller could always strive to find an optimum solution based on the space availability. Simulating the proposed adaptive controller under different road profiles showed that the active suspension performed better than the passive mode, and the adaptive methodology increased the control performance regardless of road conditions.

2.2 Interconnected Suspension Systems

As mentioned earlier in the preceding section, truck drivers' ride comfort can be effectively improved by equipping adaptive cabin suspension systems, while the truck (primary) suspension system is mainly responsible for handling and roll stability. Due to the size and weight of commercial vehicles, they can pose more significant risks on the road than passenger cars, so they are subject to additional regulations and safety requirements. Moreover, heavy vehicles easily cause road damage [93] since they put more pressure on the road surface. The roads designed for heavy vehicle traffic often have thicker, stronger pavement and additional support structures, such as bridges or overpasses, to handle the increased

weight and pressure. To extend road service life, truck manufacturers investigated advanced suspension systems to reduce the pressure and shock transmitted to the road surface [94].

Many studies have been conducted to evaluate and improve heavy vehicles' roll stability. In general, there are seven main factors that affect the commercial vehicle's roll stability, i.e., vehicle mass, the height of the CG location, the height of the vehicle roll center, suspension roll stiffness, effective track width, tire characteristics, and frame flexibility. Ervin [95] investigated the influences of size and mass variations on heavy-duty trucks' roll stability. A static rollover threshold (SRT) was used in [96] for the vehicle stability assessment, defined as the maximum lateral acceleration level in g's beyond which static rollover of a vehicle occurs. The study pointed out that a 10% axle load increment results in a 2.5% g reduction of SRT. Besides, increasing the track width by 1% could improve the roll stability limitation by around 3%. Meanwhile, the vehicle CG location also affects the roll stability to a large extent. However, a transportation vehicle has much less flexibility in adjusting CG location or roll center height. In addition, the vehicle track width is usually determined by the relevant national standards and regulations, and changes are impractical. Thus, the suspension roll stiffness and tire characteristics are possible means to improve the roll stability of a heavy vehicle. Between these two, the suspension roll stiffness is more vital for maintaining the vehicle roll stability. In the literature, various suspension designs and control systems have been developed to improve roll stability and limit lateral load transfer, such as (active) anti-roll bars [97], Electronic Stability Control (ESC) for cars and trucks [98,99], Roll Stability Control (RSC) for articulated vehicles and high CG cars [100,101], and Balanced Pneumatic Suspension (BPS) [102,103] for heavy commercial vehicles.

The strong coupling between the vehicle's ride and roll dynamics poses complicated design challenges associated with conflicting requirements between ride comfort and directional stability, especially in heavy commercial vehicles exposed to wide load variations [104]. One approach to ease this conflict is using different interconnected structures, such as mechanical, hydraulic, and pneumatic connections. The most popular mechanical interconnection is the anti-roll bar, which applies additional roll stiffness to achieve better roll stability. However, it adds extra weight to the vehicle chassis and somewhat deteriorates ride comfort [105]. Moreover, the hydraulic interconnected system has received much attention in recent decades because of its anti-roll and/or anti-pitch abilities. Various techniques have been developed for different kinds of vehicles, such as tri-axle trucks [106], tractor semi-trailers [107], buses [108], racing cars (McLaren 720s [109]), passenger cars (Citroen 2CV [110]), ambulance [111], rough terrain vehicle [112], mining vehicle [113], crane vehicle [114], etc. This configuration

has also been investigated as a self-steering system for the high-speed train [115]. Overall, the main advantage of this type of suspension is improved handling and ride quality, especially when the vehicle is driven on uneven or rough road surfaces. Additionally, it can reduce the roll and pitch of the vehicle to improve stability and safety.

Specifically, the hydro-pneumatic interconnected suspension has excellent potential in anti-roll and anti-pitch control. Thus, the harsh rotational motions of the vehicle can be improved without compromising the ride comfort. By interconnecting all the suspension struts, such optimization objectives can be achieved. Although the shock absorber works well with transient and fast-changing inputs like road disturbances, it becomes ineffective when dealing with slow-changing motions, such as braking, accelerating, and turning maneuvers. In these situations, higher roll and pitch stiffness is preferred for reducing the vehicle rotational motions. Hence, the hydro-pneumatic interconnected suspension system, which can provide adjustable rotational stiffness and damping, is a good approach for improving vehicle handling stability during steady maneuvers. Meanwhile, the suspension stiffness and damping properties can be easily assigned by designers. A review of passive interconnected suspension systems can be found in [116].

2.2.1 Roll-plane interconnected configuration

One of the interconnected configurations in the literature is the H_2 arrangement that has been patented by Kinetics Pty Ltd. [117]. As shown in Figure 2.8(a), four double-acting hydraulic cylinders and two nitrogen accumulators are diagonally connected through hoses and ten flow control valves. The valve resistance along the pipelines provides the damping, and the nitrogen gas compression in the accumulators generates the stiffness. Implementing the proposed interconnected suspension system can enhance the roll stability by providing increased roll stiffness and preserving a relatively soft ride at the same time. A comprehensive frequency analysis of this configuration can be found in [118]. The same configuration has been widely studied in research works [107,119,120,121,122,123,124,125,126], and its performance has been evaluated by laboratory and road tests in [108,127,128,129]. This configuration has also been extended to tri-axle applications [106,130,131,132].

Another roll-plane interconnected suspension with external gas chambers and valves has been discussed in [114,133,134], as shown in Figure 2.8(b). The lower and upper chambers are diagonally connected, providing a relatively high and adjustable roll stiffness. Moreover, Cao introduced different kinds of integrated single-gas-chamber and twin-gas-chamber suspension struts [135], which could be

interconnected through hydraulic, pneumatic, or hybrid fluidic couplings. Its highlight feature was the strut's large effective working area, by which the system working pressure was reduced to a large extent. This study concluded that the stiffness and damping characteristics in the roll and pitch were enhanced, and those of the heave remained relatively soft.

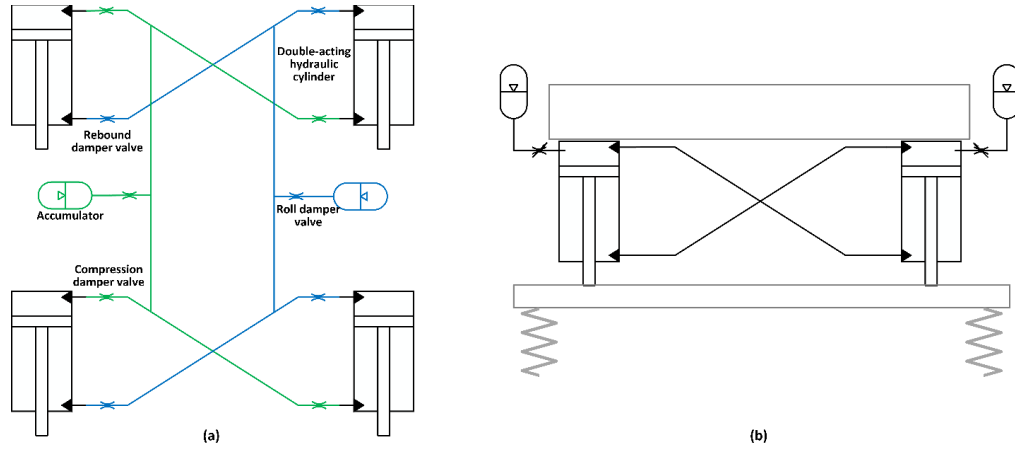


Figure 2.8. (a) H₂ arrangement; (b) roll-plane interconnected arrangement.

2.2.2 Pitch-plane interconnected configuration

Other than roll stiffness, pitch stiffness is another important characteristic that affects the vehicle stability and handling performance. Pitch refers to the vehicle's tendency to rotate about its front-to-back axis, which can be caused by acceleration or braking. Having an appropriate pitch stiffness is essential for several reasons. First, it helps maintain vehicle stability during acceleration and braking, improve handling, and reduce the risk of accidents. Second, it can improve ride quality by reducing the uncomfortable pitch motion when travelling over bumps or uneven surfaces. Finally, it can improve the vehicle's overall performance by reducing the weight transfer, which can improve traction and grip.

Anti-roll bars are standard on many vehicles, but anti-pitch bars are not as commonly used. Many vehicles' suspension systems are designed to reduce pitch without needing an additional anti-pitch bar. For example, most vehicles use a combination of shock absorbers, springs, and other suspension components designed to resist pitch and provide a stable ride. In addition, adding an anti-pitch bar could negatively affect vehicle handling and ride quality, e.g., an anti-pitch bar could make the suspension stiffer and less comfortable, which could be a disadvantage for passengers. It could also make the vehicle more difficult to control in certain situations, such as when driving on slippery roads. However,

some vehicles may require higher levels of pitch stiffness. For example, high-performance sports cars or race cars often require very high pitch stiffness to optimize the handling performance. These vehicles typically have powerful engines designed to accelerate and brake quickly, which can cause significant pitch motions. High pitch stiffness can help to reduce the effects of these forces and provide better control and handling. Similarly, some off-road vehicles, such as trucks or SUVs used for heavy-duty towing or hauling, may require higher pitch stiffness to maintain stability and control under heavy loads. Overall, pitch stiffness is still essential for a vehicle system, but it cannot be maintained by simply adding an anti-pitch bar due to the compact space of the chassis. Thus, some pitch-plane interconnected suspension systems were invented to enhance pitch stiffness without sacrificing ride comfort.

Zhang et al. [113,136] introduced a bounce and pitch plane hydraulically interconnected suspension (HIS) for mining vehicles with a focus on improving ride quality and pitch stiffness, as shown in Figure 2.9(a). In detail, a lumped-mass vehicle model with a mechanical-hydraulic coupled suspension system was developed using the free-body diagram method. The impedance of the hydraulic system was modelled according to the transfer matrix method, which was integrated to form the dynamic equations of the mechanical-hydraulic coupled system. The simulation study indicated that the proposed HIS system effectively suppressed the pitch motion to guarantee vehicle handling performance and favourably provided soft vertical stiffness to improve the ride quality. Meanwhile, the force distribution between the front and rear wheels was more reasonable, and the sprung mass vibration decay rate increased somewhat. However, the suspension roll stiffness was not considered by this interconnected configuration.

Furthermore, Xu and Zhang [137] developed a roll and pitch independently controlled hydraulically interconnected suspension. As shown in Figure 2.9(b), the inside four cylinders and fluid circuits formed roll-plane HIS, and the additional four cylinder outside formed pitch-plane HIS. The modal analysis showed the modes-decoupling property of the proposed system. Specifically, the roll-plane HIS increases roll stiffness without affecting other modes. This study demonstrated that the vehicle ride comfort and handling stability could be improved simultaneously by integrating the roll and pitch plane HIS. Meanwhile, it decoupled the four vehicle modes (bounce, roll, pitch, and warp) that could be independently tuned without compromise.

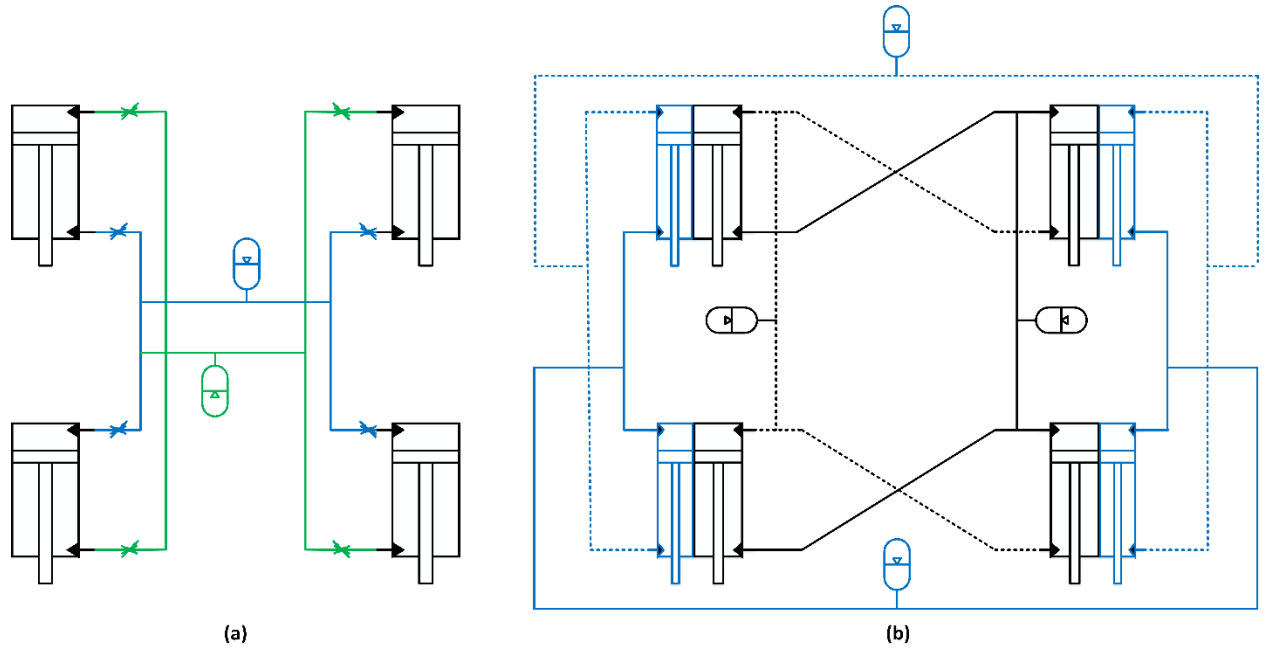


Figure 2.9. Pitch interconnected configurations [136,137].

2.3 Vehicle State Observer

A vehicle state observer is a system that estimates the internal states of a vehicle based on sensor measurements and a dynamic model. The observer uses a set of mathematical equations to filter and process the sensor data to produce an accurate estimation of the vehicle states. Vehicle state observers are commonly used in various applications, including vehicle control, navigation, autonomous driving, etc.

With the rapid developments of the vehicle adaptive suspension systems, many non-measurable vehicle motions need to be estimated to achieve better control performance, such as the body rotational angles, body vertical displacement and velocity, the movements of the unsprung mass, etc. An adaptive suspension system mainly consists of adaptive dampers or actuators, springs, and ECU embedded with the control algorithms and other necessary components. Technically, the performances of these advanced control algorithms can be optimized and improved if more accurate measurements of the vehicle motions are available. Thus, the corresponding observers that can accurately estimate the unmeasurable vehicle states significantly impact the overall control performances. In past studies, the observers designed for linear vehicle dynamic systems are well-known from both deterministic (Luenberger) and stochastic (Kalman) points of view [138]. The Luenberger approach is to choose a

constant observer gain matrix to place the eigenvalues of the estimation error at proper positions [139]. The Kalman approach aims to minimize the error covariance matrix, which determines the observer gain matrix by solving the Riccati equation [140]. Various approaches have been introduced in the literature for developing vehicle state observers based on the quarter-car model, such as Kalman filter related theories [141,142,143], modified Luenburger observer [144,145], sliding mode control [146,147,148], reduced-order observer [149,150], H_∞ [151], H_∞ /LPV [152], extended state observer [153,154], fuzzy control [155], type-2 fuzzy neural network observer [156], etc. Moreover, some observers were developed to estimate both vertical and pitch motions based on the half-car model, such as sliding mode control [157], LQG [158], Luenburger observer [159], nonlinear extended state observer [160], spherical simplex unscented Kalman filter [161], etc.

Various techniques based on the full-car model with different sensor combinations were introduced in the literature to estimate the vertical, roll, and pitch motions of the vehicle body and the road disturbances. Bolandhemmat [162] gave a comprehensive study on the Kalman filter (KF), extended Kalman filter (EKF), unscented Kalman filter (UKF), aided Kalman filter (AKF), and supervisory Kalman filter (SKF). A Cadillac SRX was used for experimental validation. Its sensing system consisted of one 6-axis IMU, four suspension height sensors, and eight accelerometers at both ends of the four suspension struts. The KF observer embedding with minimal realization was tested, and the estimation accuracy with different combinations of sensors was compared. In the first case, one 6-axis IMU and four wheel accelerometers were used, which measured the vertical acceleration, roll rate, pitch rate of the vehicle body, and the vertical accelerations of each wheel. However, the suspension relative displacements, pitch angle, and roll angle of the vehicle body could not be accurately estimated, but it provided acceptable estimations of the wheel-hub velocities, suspension relative velocities, and sprung mass vertical velocity. In the second case, four suspension height sensors and one lateral accelerometer were used. Although the observability matrix had full rank, the KF could not estimate all states accurately due to the lack of information on road disturbances. The vehicle body's vertical velocity, roll rate, pitch rate, the absolute velocities of wheels, and suspension relative velocities were not precisely estimated. Besides, the UKF observer with nine accelerometers' measurements was evaluated by running the computer simulation instead of a road test. Because of the unknown road profiles, the UKF could not provide an acceptable estimation in terms of the sprung mass's vertical displacement, roll angle, pitch angle and wheels' vertical displacements.

Similarly, Ren et al. [163] developed a suspension state observer based on an unscented Kalman filter (UKF), which performed better than the extended Kalman filter (EKF) in terms of robustness and convergence speed. In their approach, four suspension heights and the vertical accelerations at the four corners of the sprung mass were considered as available measurements. Based on these signals, the vertical velocity, roll rate, and pitch rate of the sprung mass and the vertical velocities of the unsprung mass were estimated. The accuracy was validated by co-simulation between MATLAB/Simulink and CarSim on three typical road profiles. The simulation results showed that the proposed UKF observer could accurately estimate the vehicle states and had low sensitivity to unknown road disturbances.

Kim et al. [164] presented an improved discrete Kalman filter with unknown inputs, which could effectively and simultaneously estimate the state variables and road roughness without any prior information about the vehicle suspension control system. The road roughness input estimator was evaluated using an in-vehicle test bed with a laser-type profilometer, providing unparalleled accuracy and precision. Moreover, the state estimation performance of the proposed observer for a suspension control system was validated using CarSim software to strengthen the effectiveness and practicality of this approach.

Moreover, Nguyen et al. [25] designed a full-car observer for a semi-active suspension to estimate vehicle states and road disturbances. The available measurements were vertical accelerations of the vehicle body at its four corners and the vertical displacements of the four wheels. The simulation results presented that the extended observer could accurately estimate the road profile, but the performances of state estimation were not mentioned. With the same estimation objectives, Morato et al. [32] developed an augmented H_2 observer with the measurements of suspension relative displacements and the vertical accelerations at each corner of the vehicle body. Besides, the suspension deflection velocities were computed numerically by a derivative filter. The observer was validated based on the experimental results, which could accurately estimate the road profile and acceptable estimations of the sprung mass' roll and pitch angles. Dugard et al. [165] designed an H_∞ observer to estimate the unmeasurable vehicle state variables under unknown disturbances. There were seven measurements required in their study, i.e., the vertical accelerations at three corners of the vehicle body and the vertical accelerations of the four unsprung masses. The observer robustness test demonstrated that the H_∞ observer was not sensitive to the parametric uncertainties, e.g., the changes of the damping coefficients did not affect the estimation accuracy. The experimental results showed relatively good estimations of velocities and accelerations of the sprung and unsprung mass.

Kjellberg and Sundell [30] tested their suspension controller and the observer on a Volvo S90, equipped with three single-axis accelerometers and four level sensors that measured the position and velocity of each wheel. The authors pointed out that other signals could be obtained based on those measurements with sufficient accuracies, such as suspension deflections and velocities and the sprung mass's vertical velocity, roll rate, pitch rate and corresponding accelerations. Since the absolute vertical position, roll angle, and pitch angle could not be measured with the available sensors, a workaround idea was introduced based on the assumption of the small slope of the road.

Wang, Chang, and Tian [166] developed an extended-state observer to estimate road disturbance, uncertainty dynamics, and vehicle real-time states. In their work, the sprung mass vertical displacement, roll angle, and pitch angle were considered as measurable states, and they were then used for estimating the vertical velocity, roll rate, and pitch rate of the vehicle body. A virtual prototype built in ADAMS was used to evaluate the proposed observer and controllers. It showed that the extended-state observer could estimate the disturbances, but the estimation accuracy of the vehicle body motions was not explained.

Huang et al. [167] presented an unknown system dynamics estimator (USDE) constructed using first-order low-pass filter operations and requiring only one tuning parameter. Unlike traditional state observers, the proposed USDE did not need function approximators like neural networks and fuzzy-logic systems. This reduced the computational burden and eliminated the need for intermediate variables, thus simplifying the entire computational process. Furthermore, the proposed USDE ensured system convergence and had trivial parameter tuning, which made it a highly practical and efficient solution for suspension control systems. To handle the time-varying input delay, a predictor-based compensation strategy was developed. This study combined the USDE and the predictor to design a feedback controller that effectively attenuated vibrations and simultaneously retained the required suspension performances.

2.4 Summary

In this chapter, vehicle adaptive suspension systems based on various control techniques were first reviewed and discussed. Past studies of suspensions have significantly contributed to enhancing drivers' ride comfort and vehicle stability. Compared to the passive suspension, the conventional Skyhook control was efficient in improving vehicle vertical dynamics but sometimes caused unsatisfactory rotational motions. Nevertheless, the Skyhook concept was computationally efficient for

real-time applications and was often considered as the benchmark. Several advanced Skyhook strategies led to greater adaptations, whereby the controller could realize coordinated control performance instead of only focusing on the motions at each corner. Recent works mainly focused on optimal control strategies that had great potential to improve vehicle dynamics in all three directions, i.e., vertical, roll, and pitch. However, the suspension mechanical limitations and semi-active damper dissipative constraint were not fully considered. This problem resulted in the so-called clipped control, which was not the optimal solution for semi-active suspension systems. Various MPC controllers were developed to solve this issue, by which the system constraints were incorporated into the algorithms. However, its computational cost is too expensive for practical applications. Moreover, researchers had tried to develop road-adaptive algorithms to further optimize the control performance regardless of different road conditions. However, the simulation scenarios used in their evaluations were not comprehensive. Finally, although the simulation results showed considerable improvements, very few had been validated through experimental studies. Besides the damping control, vehicle attitude and stability control requirements are vital in the automotive industry. Different interconnected suspension systems were introduced in the literature to enhance vehicle anti-roll and anti-pitch properties. However, their functionalities were limited, and the manufacturing cost was more expensive than conventional suspensions.

Based on the review of the past studies, some research topics on vehicle suspension systems can be investigated:

- **Coordinated vibration attenuation:** The majority of past studies aimed to reduce cabin's vertical acceleration and, to a lesser extent, pitch vibrations. However, the tilt motions in roll and pitch directions are also the primary sources of discomfort. As such, damping control systems that can improve cabin dynamics in all three directions should be developed.
- **Advanced semi-active suspension systems:** A semi-active suspension system requires less power than the active one, but the research on addressing the damper dissipative constraint and the mechanical limitation should be thoroughly considered. Meanwhile, the control algorithm need to be concise enough to be processed on an automotive-grade microcontroller.
- **Gain-adaptive control approach:** Vehicles may drive on various kinds of roads, such as highways, rough paved roads, off-roads, etc. It is not possible to find one set of gains that can handle varieties of driving conditions. One potential solution is to intelligently adjust the control parameters (such as objective functions, penalty matrices, gains, etc.) according to the disturbances. This approach

requires road classifiers to provide accurate disturbance information for the control system. In addition, proper gain-tuning rules need to be developed to guarantee robustness performances on any road.

- **Intelligent hydro-pneumatic interconnected suspensions:** Different from the conventional suspensions, the damping and stiffness characteristics of the interconnected suspension can be conveniently tuned or adjusted. Meanwhile, one can adjust its anti-roll and/or anti-pitch properties by arranging different interconnected configurations, which gives the designers more flexibility to meet any specific requirements.

Chapter 3

Coordinated Semi-Active Suspension Systems for Truck Cabins

This chapter starts with the dynamic modelling of a cab-over-engine (COE) with a tilting mechanism. The accuracy of COE model is validated through the experimental study that can be found in Sections 4.4. It provides crucial insights into the performance of the COE suspension system and serves as a solid foundation for further analysis and development. Building upon this groundwork, two general optimal controllers are developed for vehicle suspension systems, based on the MPC and the integrated Skyhook-LQR approach. These controllers aim to improve the ride quality and stability. The MPC controller employs a predictive control strategy, which takes into account the future behavior of the vehicle and optimizes the control inputs accordingly. On the other hand, the integrated Skyhook-LQR algorithm combines the Skyhook control strategy with the LQR approach, which results in a robust and effective control solution. In addition, two gain-adaptive algorithms are introduced to further optimize the overall performance of the integrated Skyhook-LQR controller on various road conditions. Finally, the chapter is concluded by a thorough summary, which provides insights into the benefits and limitations of the proposed control algorithms.

3.1 Modelling of Cab-over-engine Suspension System

As discussed in Section 2.1.2, the cabin suspension system is usually modelled as a simplified passenger car suspension in the past studies. However, the COE cabin has a unique tilting mechanism, by which the cabin can be easily tilted forward for engine repair or powertrain inspection. Figure 3.1 shows a cabin tilting operation from its nominal position (a) to its fully raised position (c). Nowadays, the tilting mechanism usually consists of a hydraulic system to raise or lower the cabin, lock it in the down position, and release it when it tilts [168].

The CAD model of a COE suspension system produced by Beijing Foton Daimler Automotive Co., Ltd. (BFDA) is shown in Figure 3.2. As can be seen, there are four vertical airbag-damper units at each suspension corner, and two lateral dampers at the rear. The motions of the chassis frame are considered as the disturbance inputs to the cabin, which is known as a base excitation system, as shown in Figure 3.3. Specifically, z_f , φ_f , and θ_f represent the vertical displacement, roll angle, and pitch angle of the chassis frame, respectively. In the cabin front suspension, a planar linkage mechanism is used, which connects the cabin to the truck chassis frame to ensure the necessary motion constraints in longitudinal,

lateral, and yaw directions with respect to the chassis frame, while allowing three DOFs in the planes of chassis vertical, roll, and pitch motions.



Figure 3.1. Cabin tilting of a flat-nose truck

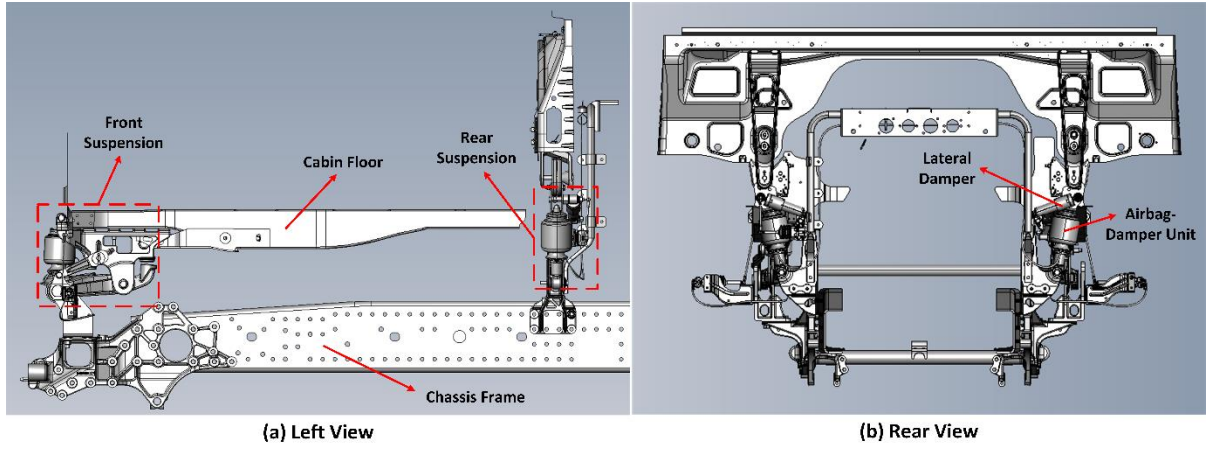


Figure 3.2. The CAD model of a cab-over-engine suspension system

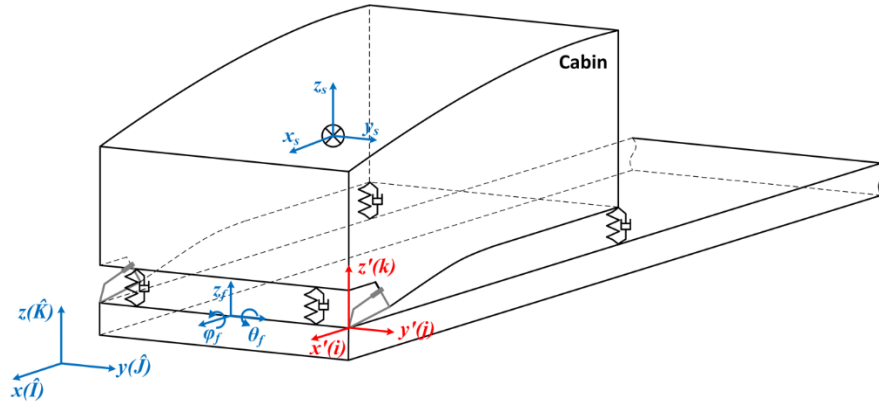


Figure 3.3. Schematic of truck cabin system

3.1.1 Passive COE suspension system

The planner linkage mechanism can be considered as a modified four bar linkage, in which one of the revolute joints has been replaced with a revolute-slide joint. As shown in Figure 3.4, the coordination of the linkage mechanism is located at the revolute joint A . The longitudinal axis x' points to the front, and the vertical axis z' points upward. The lengths of four links are r_1 , r_2 , r_3 , and r_4 , respectively. However, the length of r_3 is not fixed because of the slotted hole design, which allows the cabin moves independently in vertical and pitch directions. The cabin is in fact the extension of link BC which has a combined motions of translation and rotation, while Link AD is rigidly attached to the chassis frame. The angles θ_i ($i = 1, 2, 3, 4$) are defined accordingly for future analysis. Note that θ_4 represents the angle between link AD and the perpendicular axis which is a fixed angle. To further improve the ride comfort, two revolute joints A and B are replaced by two bushings, by which the cabin can roll independently with respect to the chassis frame. Since the planner linkage mechanisms on the front-left and front-right corners have the same configuration, the left one is chosen as the example and introduced in detail in this section.

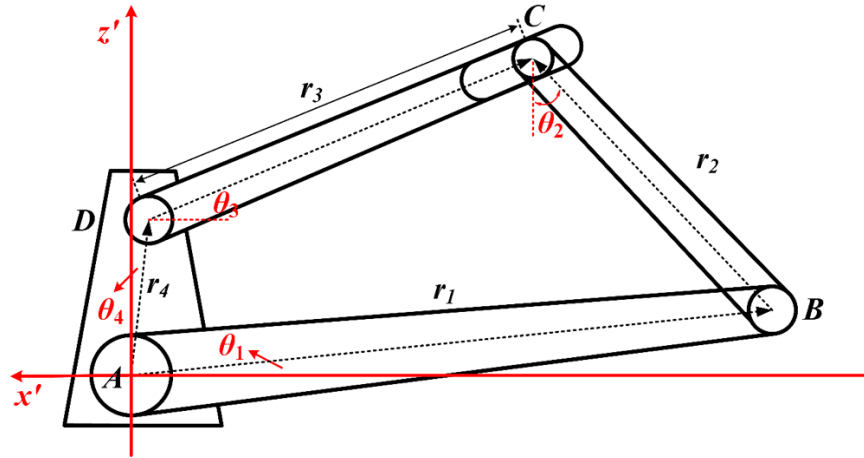


Figure 3.4. Drawing of the linkage mechanism

The purpose is to develop a mathematical model for the linkage mechanism that can be used to arrive at the cabin dynamic model based on the Lagrangian principle. Firstly, use the angles θ_i and the lengths r_i ($i = 1, 2, 3, 4$) to describe the geometry of the linkage mechanism. Note that θ_1 , θ_2 , and θ_3 are dependent and can be defined based on two generalized coordinates such as θ_2 and r_3 .

The vector loop-closure equation of the linkage mechanism is written as [169]:

$$\vec{r}_1 + \vec{r}_2 = \vec{r}_4 + \vec{r}_3 \quad (3.1)$$

which can be rewritten in terms of the lengths and angles as:

$$\begin{aligned} r_1 \cos \theta_1 - r_2 \sin \theta_2 &= r_3 \cos \theta_3 + r_4 \sin \theta_4 \\ r_1 \sin \theta_1 + r_2 \cos \theta_2 &= r_3 \sin \theta_3 + r_4 \cos \theta_4 \end{aligned} \quad (3.2)$$

Using small-angle approximation around the nominal angle θ_i^0 , we have:

$$\begin{aligned} \cos \theta_i &= \cos(\theta_i^0 + \Delta\theta_i) = \cos \theta_i^0 \cos \Delta\theta_i - \sin \theta_i^0 \sin \Delta\theta_i = \cos \theta_i^0 - \sin \theta_i^0 \Delta\theta_i \\ \sin \theta_i &= \sin(\theta_i^0 + \Delta\theta_i) = \sin \theta_i^0 \cos \Delta\theta_i + \cos \theta_i^0 \sin \Delta\theta_i = \sin \theta_i^0 + \cos \theta_i^0 \Delta\theta_i \end{aligned} \quad (3.3)$$

in which θ_i^0 is the nominal angle, and $\Delta\theta_i$ represents the change of the angle with respect to its nominal value.

Besides, the variable r_3 can be represented by

$$r_3 = r_3^0 - \Delta r_3 \quad (3.4)$$

According to equations (3.3) and (3.4), the Eq.(3.2) can be fully linearized, so that the variable $\Delta\theta_3$ can be derived as:

$$\Delta\theta_3 = \underbrace{\frac{r_2 \cos(\theta_2^0 - \theta_1^0)}{r_3^0 \sin(\theta_3^0 - \theta_1^0)}}_b \Delta\theta_2 - \underbrace{\frac{\cos(\theta_1^0 - \theta_3^0)}{r_3^0 \sin(\theta_3^0 - \theta_1^0)}}_c \Delta r_3 = b\Delta\theta_2 - c\Delta r_3 \quad (3.5)$$

Similarly, the variable $\Delta\theta_1$ can also be represented in terms of $\Delta\theta_2$ and Δr_3 as follows:

$$\Delta\theta_1 = \underbrace{\frac{r_2 \cos(\theta_2^0 - \theta_3^0)}{r_1 \sin(\theta_3^0 - \theta_1^0)}}_a \Delta\theta_2 - \underbrace{\frac{1}{r_1 \sin(\theta_3^0 - \theta_1^0)}}_d \Delta r_3 = a\Delta\theta_2 - d\Delta r_3 \quad (3.6)$$

The previous kinematic equations are written based on the non-inertial coordinate $x'y'z'$, in which the origin is located at the revolute joint A that is rigidly attached to the chassis frame. It is necessary to transfer the component movements from the non-inertial coordinate to the inertial (global) coordinate xyz , as shown in Figure 3.3. Denoting unit vectors i, j, k as the longitudinal, lateral, and vertical

directional vectors of the non-inertial coordinate $x'y'z'$. They can be expressed in the global coordinate xyz in terms of unit vectors $\hat{I}, \hat{J}, \hat{K}$ as follows:

$$\begin{aligned} \text{Longitudinal: } i &= \cos \theta_f \hat{I} - \sin \theta_f \hat{K} \\ \text{Lateral: } j &= \cos \varphi_f \hat{J} + \sin \varphi_f \hat{K} \\ \text{Vertical: } k &= \sin \theta_f \hat{I} - \sin \varphi_f \hat{J} + \cos \theta_f \cos \varphi_f \hat{K} \end{aligned} \quad (3.7)$$

Due to the mechanical constraint of the linkage mechanism, it is assumed that the cabin follows the chassis' longitudinal, lateral, and yaw motions, so that the cabin has three degrees of freedom, i.e., vertical, pitch, and roll. In the pitch plane shown in Figure 3.5, the following parameters are defined in order to describe the motion of the cabin's CG location S . As shown in Figure 3.7, l_{BS} represents the longitudinal distance from the bushing B to the cabin's CG location S ; l_{BS1} represents the longitudinal distance from the bushing B to the upper point of the front air spring S_1 ; l_{BS2} represents the longitudinal distance from the bushing B to the upper point of the rear air spring S_2 . Since link BC is rigidly connected to the cabin, its extended line CC' and perpendicular line BB' are used as a reference coordinate. Besides, α_1 is the angle between CC' and BS_1 , α_2 is the angle between BB' and BS_2 , α_3 is the angle between BB' and BS . Since the cabin is assumed as a rigid body, the above parameters are fixed values which can be calculated according to the cabin dimensions. In the roll plane shown in Figure 3.6, half of the lateral distance between the left and right suspension struts is represented by l , and the cabin CG height with respect to the floor is named as h . Since the use of the two bushings at locations B and C provides more flexibility for the cabin's roll motion, its roll angle φ_s is considered as the third generalized coordinate.

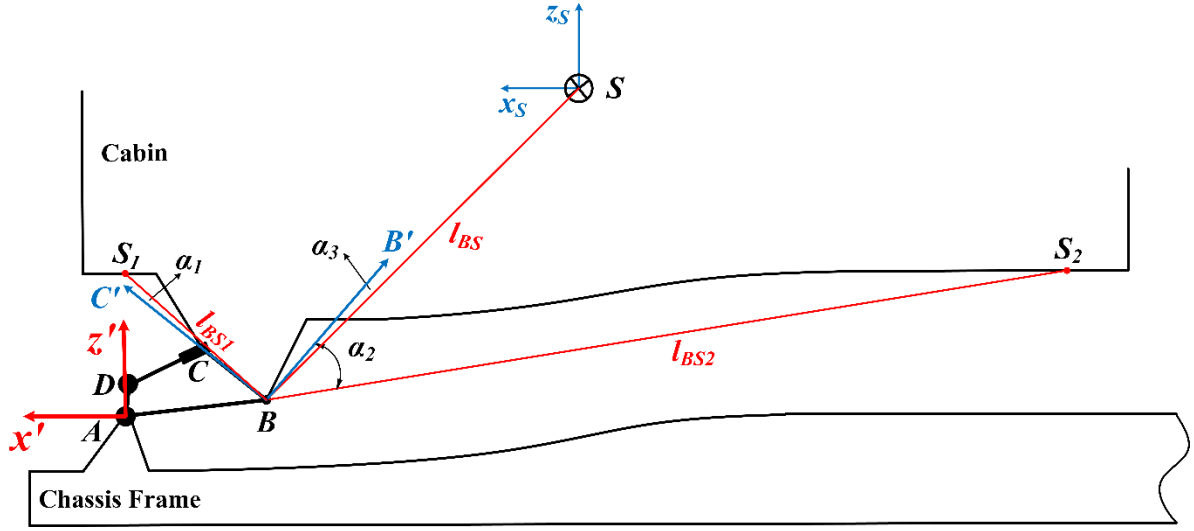


Figure 3.7. Cabin coordinate located at the revolute joint B

Hence, the vector $\overrightarrow{OS}$ can be written as

$$\begin{aligned}
 \overrightarrow{OS} &= \overrightarrow{OA} + \overrightarrow{AB} + \overrightarrow{BS} \\
 &= x_A \hat{i} + y_A \hat{j} + z_A \hat{k} - [r_1 \cos \theta_1 + l_{BS} \cos(\theta_2 - \alpha_3)]i \\
 &\quad - [h \sin(\varphi_S - \varphi_f) + l \cos(\varphi_S - \varphi_f)]j \\
 &\quad + [r_1 \sin \theta_1 + l_{BS} \sin(\theta_2 - \alpha_3)]k
 \end{aligned} \tag{3.8}$$

in which z_A represent the height at the revolute joint A which is the origin of the non-inertial coordinate, x_A and y_A represent the longitudinal and lateral distances at the revolute joint A with respect to the origin O of the inertial coordinate.

Combining Eq.(3.7) and (3.8), the vertical displacement at the cabin CG location S can be linearized based on the small-angle approximation as described in Eq.(3.3):

$$\begin{aligned}
 z_S &= z_f + \underbrace{[r_1 a \cos \theta_1^0 + l_{BS} \cos(\theta_2^0 - \alpha_3)]}_{p_1} \Delta \theta_2 - \underbrace{r_1 d \cos \theta_1^0}_{p_2} \Delta r_3 \\
 &\quad + \underbrace{[l_{BS} \cos(\theta_2^0 - \alpha_3) + r_1 \cos \theta_1^0]}_{p_3} \theta_f
 \end{aligned} \tag{3.9}$$

in which z_f represents the vertical displacement of the chassis frame at the middle of the front cabin suspension, as shown in Figure 3.6.

Similarly, the longitudinal x_S and lateral y_S displacements at the cabin's CG location can be represented as

$$x_S = x_f + \underbrace{[l_{BS} \sin(\theta_2^0 - \alpha_3) + r_1 a \sin \theta_1^0]}_{p_4} \Delta\theta_2 - \underbrace{r_1 d \sin \theta_1^0}_{p_5} \Delta r_3 + [r_1 \sin \theta_1^0 + l_{BS} \sin(\theta_2^0 - \alpha_3)] \theta_f \quad (3.10)$$

$$y_S = y_f - [r_1 \sin \theta_1^0 + l_{BS} \sin(\theta_2^0 - \alpha_3)] \varphi_f - h \varphi_S \quad (3.11)$$

in which x_f and y_f represent the longitudinal and lateral displacement of the chassis frame at the middle of the front cabin suspension.

Similarly, the suspension relative displacements Δx_i with respect to the nominal positions at the four corners can be derived as follows:

$$\begin{aligned} \Delta x_1 &= \underbrace{[r_1 a \cos \theta_1^0 - l_{BS1} \sin(\theta_2^0 - \alpha_1)]}_{p_6} \Delta\theta_2 - \underbrace{r_1 d \cos \theta_1^0}_{p_2} \Delta r_3 + l(\varphi_S - \varphi_f) \\ \Delta x_2 &= \underbrace{[r_1 a \cos \theta_1^0 + l_{BS2} \cos(\theta_2^0 - \alpha_2)]}_{p_7} \Delta\theta_2 - \underbrace{r_1 d \cos \theta_1^0}_{p_2} \Delta r_3 + l(\varphi_S - \varphi_f) \\ \Delta x_3 &= p_6 \Delta\theta_2 - p_2 \Delta r_3 - l(\varphi_S - \varphi_f) \\ \Delta x_4 &= p_7 \Delta\theta_2 - p_2 \Delta r_3 - l(\varphi_S - \varphi_f) \end{aligned} \quad (3.12)$$

▪ Lagrangian Formulation

With the three generalized coordinates $\Delta\theta_2$, Δr_3 , and φ_S that describes the configuration of the COE system, the following Euler-Lagrange equations can be derived:

$$\begin{aligned} \frac{d}{dt} \left(\frac{\partial T}{\partial \dot{\theta}_2} \right) - \frac{\partial T}{\partial \Delta\theta_2} + \frac{\partial V}{\partial \Delta\theta_2} + \frac{\partial D}{\partial \dot{\theta}_2} &= 0 \\ \frac{d}{dt} \left(\frac{\partial T}{\partial \dot{r}_3} \right) - \frac{\partial T}{\partial \Delta r_3} + \frac{\partial V}{\partial \Delta r_3} + \frac{\partial D}{\partial \dot{r}_3} &= 0 \\ \frac{d}{dt} \left(\frac{\partial T}{\partial \dot{\varphi}_S} \right) - \frac{\partial T}{\partial \varphi_S} + \frac{\partial V}{\partial \varphi_S} + \frac{\partial D}{\partial \dot{\varphi}_S} &= 0 \end{aligned} \quad (3.13)$$

in which T is the kinetic energy, V is the potential energy, and D is the dissipation energy.

Compared to the cabin's mass and moments of inertia, the inertia of the linkages can be neglected. Hence, the total kinetic energy of the system can be simplified as

$$T = \frac{1}{2} [m_S \dot{x}_S^2 + m_S \dot{y}_S^2 + m_S \dot{z}_S^2 + I_x \dot{\phi}_S^2 + I_y \dot{\theta}_S^2 + I_z \dot{\psi}_S^2] \quad (3.14)$$

in which I_x , I_y , and I_z are the cabin's roll, pitch, and yaw moments of inertia, respectively.

Similarly, the potential energy of the system can be simplified as

$$V = m_S g z_S + \frac{1}{2} k_1 (\Delta x_1^0 - \Delta x_1)^2 + \frac{1}{2} k_2 (\Delta x_2^0 - \Delta x_2)^2 + \frac{1}{2} k_3 (\Delta x_3^0 - \Delta x_3)^2 + \frac{1}{2} k_4 (\Delta x_4^0 - \Delta x_4)^2 + \frac{1}{2} k_\phi (\varphi_S - \varphi_f)^2 \quad (3.15)$$

On the right-hand side, the first term represents the potential energy due to gravity, the following four terms represent the elastic potential energy stored by the air springs, and the last term represents elastic potential energy stored by the bushings. Note that g is the gravitational acceleration, k_i ($i = 1, 2, 3, 4$) represents the air spring stiffness, k_ϕ represents the suspension equivalent roll stiffness, and Δx_i^0 are the compressed lengths of the four air springs at the vehicle rest position due to the cabin mass.

The dissipation energy of the system due to the dampers is

$$D = \frac{1}{2} c_1 \Delta \dot{x}_1^2 + \frac{1}{2} c_2 \Delta \dot{x}_2^2 + \frac{1}{2} c_3 \Delta \dot{x}_3^2 + \frac{1}{2} c_4 \Delta \dot{x}_4^2 \quad (3.16)$$

By substituting equations (3.14)-(3.16) into (3.13), three differential equations can be obtained based on three generalized coordinates, i.e., $\Delta \theta_2$, Δr_3 , and φ_S . Note that the cabin vertical displacement z_S and pitch angle θ_S are dependent on the movements of the suspension linkage mechanism. Thus, the following three dynamic equations of the COE cabin system can be derived as:

$$\begin{aligned}
m_S \ddot{z}_S = & -(k_1 + k_2 + k_3 + k_4)z_S \\
& + [k_1(p_1 - p_6) + k_2(p_1 - p_7) + k_3(p_1 - p_6) + k_4(p_1 - p_7)]\theta_S \\
& - l(k_1 + k_2 - k_3 - k_4)\varphi_S - (c_1 + c_2 + c_3 + c_4)\dot{z}_S \\
& + [c_1(p_1 - p_6) + c_2(p_1 - p_7) + c_3(p_1 - p_6) + c_4(p_1 - p_7)]\dot{\theta}_S \\
& - l(c_1 + c_2 - c_3 - c_4)\dot{\varphi}_S + (k_1 + k_2 + k_3 + k_4)z_f \\
& + (c_1 + c_2 + c_3 + c_4)\dot{z}_f \\
& + [k_1(p_6 + p_3 - p_1) + k_2(p_7 + p_3 - p_1) + k_3(p_6 + p_3 - p_1) \\
& + k_4(p_7 + p_3 - p_1)]\theta_f \\
& + [c_1(p_6 + p_3 - p_1) + c_2(p_7 + p_3 - p_1) + c_3(p_6 + p_3 - p_1) \\
& + c_4(p_7 + p_3 - p_1)]\dot{\theta}_f - \frac{m_S p_5}{p_2} \ddot{x}_S
\end{aligned} \tag{3.17}$$

$$\begin{aligned}
I_y \ddot{\theta}_S = & [k_1(p_1 - p_6) + k_2(p_1 - p_7) + k_3(p_1 - p_6) + k_4(p_1 - p_7)]z_S \\
& - [k_1(p_1 - p_6)^2 + k_2(p_1 - p_7)^2 + k_3(p_1 - p_6)^2 + k_4(p_1 - p_7)^2]\theta_S \\
& - l[k_1(p_6 - p_1) + k_2(p_7 - p_1) - k_3(p_6 - p_1) - k_4(p_7 - p_1)]\varphi_S \\
& + [c_1(p_1 - p_6) + c_2(p_1 - p_7) + c_3(p_1 - p_6) + c_4(p_1 - p_7)]\dot{z}_S \\
& - [c_1(p_1 - p_6)^2 + c_2(p_1 - p_7)^2 + c_3(p_1 - p_6)^2 + c_4(p_1 - p_7)^2]\dot{\theta}_S \\
& - l[c_1(p_6 - p_1) + c_2(p_7 - p_1) - c_3(p_6 - p_1) - c_4(p_7 - p_1)]\dot{\varphi}_S \\
& - [k_1(p_1 - p_6) + k_2(p_1 - p_7) + k_3(p_1 - p_6) + k_4(p_1 - p_7)]z_f \\
& - [c_1(p_1 - p_6) + c_2(p_1 - p_7) + c_3(p_1 - p_6) + c_4(p_1 - p_7)]\dot{z}_f \\
& + [k_1(p_1 - p_6)(p_1 - p_6 - p_3) + k_2(p_1 - p_7)(p_1 - p_7 - p_3) \\
& + k_3(p_1 - p_6)(p_1 - p_6 - p_3) + k_4(p_1 - p_7)(p_1 - p_7 - p_3)]\theta_f \\
& + [c_1(p_1 - p_6)(p_1 - p_6 - p_3) + c_2(p_1 - p_7)(p_1 - p_7 - p_3) \\
& + c_3(p_1 - p_6)(p_1 - p_6 - p_3) + c_4(p_1 - p_7)(p_1 - p_7 - p_3)]\dot{\theta}_f \\
& + \frac{m_S(p_1 p_5 - p_2 p_4)}{p_2} \ddot{x}_S
\end{aligned} \tag{3.18}$$

$$\begin{aligned}
I_x \ddot{\phi}_S = & -l(k_1 + k_2 - k_3 - k_4)z_S \\
& + l[k_1(p_1 - p_6) + k_2(p_1 - p_7) - k_3(p_1 - p_6) - k_4(p_1 - p_7)]\theta_S \\
& - (k_1 l^2 + k_2 l^2 + k_3 l^2 + k_4 l^2 + k_\phi)\phi_S - l(c_1 + c_2 - c_3 - c_4)\dot{z}_S \\
& + l[c_1(p_1 - p_6) + c_2(p_1 - p_7) - c_3(p_1 - p_6) - c_4(p_1 - p_7)]\dot{\theta}_S \\
& - (c_1 l^2 + c_2 l^2 + c_3 l^2 + c_4 l^2)\dot{\phi}_S + l(k_1 + k_2 - k_3 - k_4)z_f \\
& + l(c_1 + c_2 - c_3 - c_4)\dot{z}_f \\
& + l[k_1(p_6 + p_3 - p_1) + k_2(p_7 + p_3 - p_1) - k_3(p_6 + p_3 - p_1) \\
& - k_4(p_7 + p_3 - p_1)]\theta_f \\
& + l[c_1(p_6 + p_3 - p_1) + c_2(p_7 + p_3 - p_1) - c_3(p_6 + p_3 - p_1) \\
& - c_4(p_7 + p_3 - p_1)]\dot{\theta}_f + (k_1 l^2 + k_2 l^2 - k_3 l^2 - k_4 l^2 + k_\phi)\phi_f \\
& + (c_1 l^2 + c_2 l^2 - c_3 l^2 - c_4 l^2)\dot{\phi}_f + m_S h \ddot{y}_S
\end{aligned} \tag{3.19}$$

The above three equations (3.17), (3.18), and (3.19) can be rewritten into the linear-time-invariant (LTI) state-space form as follows:

$$\dot{X} = AX + BU \tag{3.20}$$

in which the state vector X is

$$X = [z_S \quad \theta_S \quad \phi_S \quad \dot{z}_S \quad \dot{\theta}_S \quad \dot{\phi}_S]^T$$

and the input vector U is

$$U = [z_f \quad \dot{z}_f \quad \theta_f \quad \dot{\theta}_f \quad \phi_f \quad \dot{\phi}_f \quad \ddot{x}_S \quad \ddot{y}_S]^T$$

and A is the state transition matrix, B is the input matrix.

The COE's parameters used in this study are listed in Table 3.1, which are obtained from the mass-product cabin suspension shown in Figure 3.2.

Table 3.1. Parameters of the COE system

Symbol	Description	Value
m_S	Cabin weight	1250 kg
I_x	Cabin roll moment of inertia	1468 kgm ²
I_y	Cabin pitch moment of inertia	1229 kgm ²

k_1, k_3	Front air spring stiffness	23658 N/m
k_2, k_4	Rear air spring stiffness	14854 N/m
k_ϕ	Equivalent roll stiffness of the bushings	155 kNm/rad
h	Height of the cabin CG	542 mm
l	Half of the distance between the left and right suspension strut	630 mm
l_{BS}	Longitudinal distance between B and CG	1221 mm
l_{BS1}	Longitudinal distance between B and S_1	364 mm
l_{BS2}	Longitudinal distance between B and S_2	1776 mm
α_1	Angle between CC' and BS_1	2.5°
α_2	Angle between BB' and BS_2	35.2°
α_3	Angle between BB' and BS	3.4°
θ_1^0	Nominal value of angle θ_1	6.0°
θ_2^0	Nominal value of angle θ_1	44.0°
θ_3^0	Nominal value of angle θ_1	22.9°
r_1	Length of linkage 1	265 mm
r_2	Length of linkage 2	144 mm
r_3^0	Nominal length of linkage 3	156 mm
r_4	Length of linkage 4	65 mm

3.1.2 Adaptive COE suspension system

Based on the discussion above, the dynamic model of the adaptive COE suspension system is developed in this section, in which the passive dampers are replaced by controllable dampers. As shown in Figure 3.8, the control forces are represented by F_1, F_2, F_3 , and F_4 . Different from Eq.(3.15), the potential energy with the adaptive dampers is rewritten as

$$\begin{aligned}
 V = m_s g z_s + \frac{1}{2} k_1 (\Delta x_1^0 - \Delta x_1)^2 + \frac{1}{2} k_2 (\Delta x_2^0 - \Delta x_2)^2 + \frac{1}{2} k_3 (\Delta x_3^0 - \Delta x_3)^2 \\
 + \frac{1}{2} k_4 (\Delta x_4^0 - \Delta x_4)^2 + \frac{1}{2} k_\phi (\varphi_s - \varphi_f)^2 - F_1 \Delta x_1 - F_2 \Delta x_2 - F_3 \Delta x_3 \\
 - F_4 \Delta x_4
 \end{aligned} \tag{3.21}$$

in which the last four terms on the right-hand side represent the potential energies provided by the damping forces which are the control objectives.

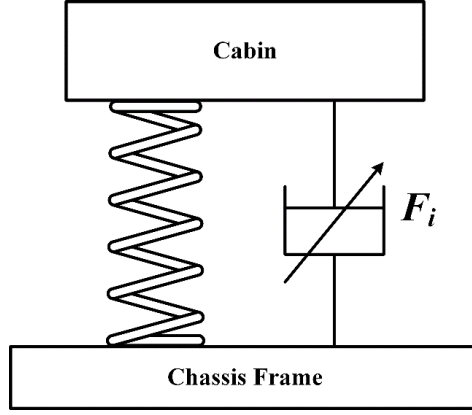


Figure 3.8. Equivalent adaptive cabin suspension (quarter)

In this way, the three dynamic equations of the adaptive COE suspension can be rewritten as

$$\begin{aligned}
 m_s \ddot{z}_S = & -(k_1 + k_2 + k_3 + k_4)z_S \\
 & + [k_1(p_1 - p_6) + k_2(p_1 - p_7) + k_3(p_1 - p_6) + k_4(p_1 - p_7)]\theta_S \\
 & - l(k_1 + k_2 - k_3 - k_4)\varphi_S + (k_1 + k_2 + k_3 + k_4)z_f \\
 & + [k_1(p_6 + p_3 - p_1) + k_2(p_7 + p_3 - p_1) + k_3(p_6 + p_3 - p_1) \\
 & + k_4(p_7 + p_3 - p_1)]\theta_f - \frac{m_s p_5}{p_2} \ddot{x}_S + F_1 + F_2 + F_3 + F_4
 \end{aligned} \tag{3.22}$$

$$\begin{aligned}
 I_y \ddot{\theta}_S = & [k_1(p_1 - p_6) + k_2(p_1 - p_7) + k_3(p_1 - p_6) + k_4(p_1 - p_7)]z_S \\
 & - [k_1(p_1 - p_6)^2 + k_2(p_1 - p_7)^2 + k_3(p_1 - p_6)^2 + k_4(p_1 - p_7)^2]\theta_S \\
 & - l[k_1(p_6 - p_1) + k_2(p_7 - p_1) - k_3(p_6 - p_1) - k_4(p_7 - p_1)]\varphi_S \\
 & - [k_1(p_1 - p_6) + k_2(p_1 - p_7) + k_3(p_1 - p_6) + k_4(p_1 - p_7)]z_f \\
 & + [k_1(p_1 - p_6)(p_1 - p_6 - p_3) + k_2(p_1 - p_7)(p_1 - p_7 - p_3) \\
 & + k_3(p_1 - p_6)(p_1 - p_6 - p_3) + k_4(p_1 - p_7)(p_1 - p_7 - p_3)]\theta_f \\
 & + \frac{m_s(p_1 p_5 - p_2 p_4)}{p_2} \ddot{x}_S + (p_6 - p_1)F_1 + (p_7 - p_1)F_2 + (p_6 - p_1)F_3 \\
 & + (p_7 - p_1)F_4
 \end{aligned} \tag{3.23}$$

$$\begin{aligned}
I_x \ddot{\phi}_S = & -l(k_1 + k_2 - k_3 - k_4)z_S \\
& + l[k_1(p_1 - p_6) + k_2(p_1 - p_7) - k_3(p_1 - p_6) - k_4(p_1 - p_7)]\theta_S \\
& - (k_1 l^2 + k_2 l^2 + k_3 l^2 + k_4 l^2 + k_\phi)\varphi_S + l(k_1 + k_2 - k_3 - k_4)z_f \\
& + l[k_1(p_6 + p_3 - p_1) + k_2(p_7 + p_3 - p_1) - k_3(p_6 + p_3 - p_1) \\
& - k_4(p_7 + p_3 - p_1)]\theta_f + (k_1 l^2 + k_2 l^2 - k_3 l^2 - k_4 l^2 + k_\phi)\varphi_f \\
& + m_S h \ddot{y}_S + l(F_1 + F_2 - F_3 - F_4)
\end{aligned} \tag{3.24}$$

Meanwhile, its state-space form is organized as:

$$\dot{X} = AX + B_w W + B_u U \tag{3.25}$$

in which the state vector is

$$X = [z_S \quad \theta_S \quad \varphi_S \quad \dot{z}_S \quad \dot{\theta}_S \quad \dot{\varphi}_S]^T$$

and the disturbance input vector is

$$W = [z_f \quad \theta_f \quad \varphi_f \quad \dot{z}_f \quad \dot{\theta}_f \quad \dot{\varphi}_f \quad \ddot{x}_S \quad \ddot{y}_S]^T$$

and the control vector is

$$U = [F_1 \quad F_2 \quad F_3 \quad F_4]^T$$

and A is the state transition matrix, B_w is the disturbance input matrix, B_u is the control input matrix.

3.2 Suspension Coordinated Control Strategies

In this chapter, two control strategies are introduced, namely Model Predictive Control (MPC) and integrated Skyhook-LQR. Both controllers are developed based on the cabin dynamics model derived in Section 3.1.2. After studying the existing works, it is noticeable that MPC can properly handle the constrained multi-objective optimization problems, so it has great potential in vehicle suspension control task that is constrained by damping dissipative characteristic. However, its complicated algorithm requires a more powerful processor due to the high computational cost, which makes MPC expensive to be implemented on an automotive-grade microcontroller. To solve this issue, a novel control strategy named integrated Skyhook-LQR is introduced in this thesis. As the name implies, the proposed algorithm integrates the popular Skyhook approach to attenuate vertical vibration and the LQR approach to further optimize rotational dynamics. Additionally, two concise gain-adaptive

algorithms are developed for the integrated Skyhook-LQR to optimize the overall control performance in various driving conditions.

3.2.1 Model Predictive Control for semi-active suspension systems

MPC is naturally capable of handling multi-variable systems, which simplifies the controller development when different control objectives need to be fulfilled. Furthermore, the state and input constraints can be easily formulated into the optimization problem. Besides, MPC has inherent local robustness to disturbances and uncertainties, as it can be designed to locally recover the LQR behavior [170]. As a model-based dynamic optimizer, the system mathematical model is used to predict the future states over a certain horizon, and the control action sequence is achieved by repeatedly solving finite-time optimization problems in a receding horizon fashion [171]. Specifically, the optimization is formulated as a constrained quadratic programming problem.

The proposed MPC controller is developed based on the adaptive COE model described in Section 3.1.2. The core of MPC is to utilize the receding horizon idea, its principle is presented as follows [30]:

- 1) At time instant k , based on current states of the control plant, the controller uses a mathematical model to predict the model output over a finite prediction horizon N_p .
- 2) The optimization problem is solved over the control horizon N_c based on a predefined cost function subject to the system constraints. An optimal control sequence is generated in this step.
- 3) The first element in the control sequence is applied to the system at the current time instant, and the rest elements are discarded.
- 4) Return to step 1 for the next time instant $k+1$.

The MPC controller is designed to predict the cabin's vertical, roll, and pitch motions with the purpose of improving its ride comfort and stability. The prediction model's accuracy and complexity have a critical impact on the closed-loop optimization, i.e., computational cost and control performance. The prediction model of the truck cabin system is explicitly given in a state-space formulation as introduced in Eq.(3.25). The proposed linear-time-invariant (LTI) mathematical model is discretized by using the Euler method with a sampling period T_s . Then, the discretized matrices can be written as follows:

$$\begin{aligned} A_d &= I + T_s A \\ B_{ud} &= T_s B_u \end{aligned} \tag{3.26}$$

$$B_{wd} = T_s B_w$$

$$C_d = C$$

$$D_{ud} = D_u$$

$$D_{wd} = D_w$$

and the discretized plant model can be formed as:

$$\begin{aligned} X(k+1) &= A_d X(k) + B_{ud} U(k) + B_{wd} W(k) \\ Y(k) &= C_d X(k) + D_{ud} U(k) + D_{wd} W(k) \end{aligned} \quad (3.27)$$

where A_d is the state transition matrix, B_{ud} is the control input matrix, B_{wd} is the disturbance input matrix, C_d is the output matrix, and D_{ud} and D_{wd} are feedforward matrices. The subscript d is used to clarify that the matrices are discrete, which are from now on neglected and the matrices are assumed to be discrete.

The state predictions over the prediction horizon are:

$$\begin{aligned} \hat{X}(k+1|k) &= A\hat{X}(k|k) + B_u U(k|k) + B_w W(k|k) \\ \hat{X}(k+2|k) &= A\hat{X}(k+1|k) + B_u U(k+1|k) + B_w W(k+1|k) \\ &= A[A\hat{X}(k|k) + B_u U(k|k) + B_w W(k|k)] + B_u U(k+1|k) + B_w W(k+1|k) \\ &= A^2\hat{X}(k|k) + AB_u U(k|k) + AB_w W(k|k) + B_u U(k+1|k) \\ &\quad + B_w W(k+1|k) \\ &\quad \vdots \\ \hat{X}(k+N_p|k) &= A\hat{X}(k+N_p-1|k) + B_u U(k+N_c-1|k) + B_w W(k+N_c-1|k) \\ &= A^{N_p}\hat{X}(k|k) + A^{N_c-1}B_u U(k|k) + A^{N_c-1}B_w W(k|k) + A^{N_c-2}B_u U(k+1|k) \\ &\quad + A^{N_c-2}B_w W(k+1|k) + \dots + B_u U(k+N_c-1|k) + B_w W(k+N_c-1|k) \end{aligned}$$

in which N_p is the prediction horizon, and N_c is the control horizon.

The output predictions over the prediction horizon are:

$$\hat{Y}(k|k) = C\hat{X}(k|k) + D_u U(k|k) + D_w W(k|k)$$

$$\begin{aligned}
\hat{Y}(k+1|k) &= C[A\hat{X}(k|k) + B_u U(k|k) + B_w W(k|k)] + D_u U(k+1|k) + D_w W(k+1|k) \\
&= CA\hat{X}(k|k) + CB_u U(k|k) + CB_w W(k|k) + D_u U(k+1|k) \\
&\quad + D_w W(k+1|k) \\
&\quad \vdots \\
\hat{Y}(k+N_p|k) &= C\hat{X}(k+N_p|k) + D_u U(k+N_p|k) + D_w W(k+N_p|k) \\
&= CA^{N_p}\hat{X}(k|k) + CA^{N_p-1}B_u U(k|k) + CA^{N_p-1}B_w W(k|k) \\
&\quad + CA^{N_p-2}B_u U(k+1|k) + CA^{N_p-2}B_w W(k+1|k) + \dots \\
&\quad + CA^{N_p-N_c}B_u U(k+N_c-1|k) + CA^{N_p-N_c}B_w W(k+N_c-1|k) \\
&\quad + CB_u U(k+N_c|k) + CB_w W(k+N_c|k) + D_u U(k+N_c|k) \\
&\quad + D_w W(k+N_c|k)
\end{aligned}$$

Hence, the state predictions can be rewritten into the matrix form as

$$\begin{aligned}
\begin{bmatrix} \hat{X}(k+1|k) \\ \hat{X}(k+2|k) \\ \vdots \\ \hat{X}(k+N_p-1|k) \\ \hat{X}(k+N_p|k) \end{bmatrix} &= \begin{bmatrix} A \\ A^2 \\ \vdots \\ A^{N_p-1} \\ A^{N_p} \end{bmatrix} \hat{X}(k|k) \\
&\quad + \begin{bmatrix} B_u & 0 & 0 & \dots & 0 \\ AB_u & B_u & 0 & \ddots & 0 \\ \vdots & \ddots & \ddots & \ddots & 0 \\ A^{N_p-2}B_u & A^{N_p-3}B_u & \ddots & A^{N_p-N_c-2}B_u & 0 \\ A^{N_p-1}B_u & A^{N_p-2}B_u & \dots & A^{N_p-N_c-1}B_u & A^{N_p-N_c}B_u \end{bmatrix} \begin{bmatrix} U(k|k) \\ U(k+1|k) \\ \vdots \\ U(k+N_c-2|k) \\ U(k+N_c-1|k) \end{bmatrix} \\
&\quad + \begin{bmatrix} B_w & 0 & 0 & \dots & 0 \\ B_w & B_w & 0 & \ddots & 0 \\ \vdots & \ddots & \ddots & \ddots & 0 \\ B_w & B_w & \ddots & B_w & 0 \\ B_w & B_w & \dots & B_w & B_w \end{bmatrix} \begin{bmatrix} W(k|k) \\ W(k+1|k) \\ \vdots \\ W(k+N_c-2|k) \\ W(k+N_c-1|k) \end{bmatrix}
\end{aligned} \tag{3.28}$$

The output predictions can be rewritten into matrix form as

$$\begin{aligned}
\underbrace{\begin{bmatrix} \hat{Y}(k|k) \\ \hat{Y}(k+1|k) \\ \vdots \\ \hat{Y}(k+N_p-2|k) \\ \hat{Y}(k+N_p-1|k) \end{bmatrix}}_{\hat{Y}} &= \underbrace{\begin{bmatrix} C \\ CA \\ \vdots \\ CA^{N_p} \\ CA^{N_p-1} \end{bmatrix}}_{\Phi} \hat{X}(k|k) \\
&+ \underbrace{\begin{bmatrix} D_u & 0 & 0 & \cdots & 0 \\ CB_u & D_u & 0 & \ddots & 0 \\ \vdots & \ddots & \ddots & \ddots & 0 \\ CA^{N_p-3}B_u & CA^{N_p-4}B_u & \ddots & D_u & 0 \\ CA^{N_p-2}B_u & CA^{N_p-3}B_u & \cdots & CA^{N_p-N_c}B_u & D_u \end{bmatrix}}_{\Gamma_u} \underbrace{\begin{bmatrix} U(k|k) \\ U(k+1|k) \\ \vdots \\ U(k+N_c-2|k) \\ U(k+N_c-1|k) \end{bmatrix}}_{\hat{U}} \\
&+ \underbrace{\begin{bmatrix} D_w & 0 & 0 & \cdots & 0 \\ CB_w & D_w & 0 & \ddots & 0 \\ \vdots & \ddots & \ddots & \ddots & 0 \\ CA^{N_p-3}B_w & CA^{N_p-4}B_w & \ddots & D_w & 0 \\ CA^{N_p-2}B_w & CA^{N_p-3}B_w & \cdots & CB_w & D_w \end{bmatrix}}_{\Gamma_w} \underbrace{\begin{bmatrix} W(k|k) \\ W(k+1|k) \\ \vdots \\ W(k+N_c-2|k) \\ W(k+N_c-1|k) \end{bmatrix}}_{\hat{W}}
\end{aligned} \tag{3.29}$$

The above equation can be written short as

$$\hat{Y} = \Phi \hat{X}(k) + \Gamma_u \hat{U} + \Gamma_w \hat{W} \tag{3.30}$$

In order to minimize the output of the system, a quadratic optimization problem is formulated. The objectives are minimizing cabin's vertical, roll, and pitch accelerations to improve ride quality. The objective function J is written in the quadratic form as follows:

$$J = \min_{\hat{U}} \hat{Y}^T Q \hat{Y} + \hat{U}^T R \hat{U} \tag{3.31}$$

where Q is the output penalty matrix and R is the control signal penalty matrix.

By substituting output prediction $\hat{Y}$ into J , the following objective function is derived as:

$$\begin{aligned}
\hat{Y}^T Q \hat{Y} + \hat{U}^T R \hat{U} &= [\Phi \hat{X}(k) + \Gamma_u \hat{U} + \Gamma_w \hat{W}]^T Q [\Phi \hat{X}(k) + \Gamma_u \hat{U} + \Gamma_w \hat{W}] + \hat{U}^T R \hat{U} \\
&= \hat{X}(k)^T \Phi^T Q \Phi \hat{X}(k) + \hat{X}(k)^T \Phi^T Q \Gamma_u \hat{U} + \hat{X}(k)^T \Phi^T Q \Gamma_w \hat{W} + \hat{U}^T \Gamma_u^T Q \Phi \hat{X}(k) \\
&\quad + \hat{U}^T \Gamma_u^T Q \Gamma_u \hat{U} + \hat{U}^T \Gamma_u^T Q \Gamma_w \hat{W} + \hat{W}^T \Gamma_w^T Q \Phi \hat{X}(k) + \hat{W}^T \Gamma_w^T Q \Gamma_u \hat{U} \\
&\quad + \hat{W}^T \Gamma_w^T Q \Gamma_w \hat{W} + \hat{U}^T R \hat{U}
\end{aligned}$$

Omitting the terms that are not dependent on the optimization variable $\hat{U}$ gives:

$$\begin{aligned}
& \hat{X}(k)^T \Phi^T Q \Gamma_u \hat{U} + \hat{U}^T \Gamma_u^T Q \Phi \hat{X}(k) + \hat{U}^T \Gamma_u^T Q \Gamma_u \hat{U} + \hat{U}^T \Gamma_u^T Q \Gamma_w \hat{W} + \hat{W}^T \Gamma_w^T Q \Gamma_u \hat{U} + \hat{U}^T R \hat{U} \\
& = \hat{U}^T \left(\underbrace{\Gamma_u^T Q \Gamma_u + R}_H \right) \hat{U} + 2 \left(\underbrace{\hat{X}(k)^T \Phi^T Q \Gamma_u + \hat{W}^T \Gamma_w^T Q \Gamma_u}_f \right) \hat{U}
\end{aligned}$$

Thus, the objective function can be rewritten as:

$$J = \min_{\hat{U}} \hat{U}^T H \hat{U} + 2f\hat{U} \quad (3.32)$$

Now the problem is formulated in such a way that a quadratic programming (QP) solver can be used, where the Hessian matrix (H) needs to be positive definite, f is the gradient vector. The typical optimization problem to solve in an MPC problem is formulated as

$$\begin{aligned}
J &= \min_{\hat{U}} \frac{1}{2} \hat{U}^T H \hat{U} + f\hat{U} \\
s.t. \quad & A\hat{U} \leq b \\
& lb \leq \hat{U} \leq ub
\end{aligned} \quad (3.33)$$

where the constraints can be formulated as a linear combination of the optimization variables and also by upper and lower bounds on the optimization variables.

To ensure the dissipativity constraint of the adaptive dampers, the following constraint must be satisfied:

$$0 \leq c_{min} \leq c_{adaptive} \leq c_{max} \quad (3.34)$$

The above dissipativity conditions can be transformed into input constraints.

$$\begin{cases} c_{min}(\dot{z}_{fi} - \dot{z}_{si}) \leq F_i \leq c_{max}(\dot{z}_{fi} - \dot{z}_{si}), & \text{if } (\dot{z}_{fi} - \dot{z}_{si}) \geq 0 \\ c_{max}(\dot{z}_{fi} - \dot{z}_{si}) \leq F_i \leq c_{min}(\dot{z}_{fi} - \dot{z}_{si}), & \text{if } (\dot{z}_{fi} - \dot{z}_{si}) < 0 \end{cases} \quad (3.35)$$

Then, the dissipativity constraint is expressed as a set of linear inequalities between control input and state variables. Let $\Delta v_i = \dot{z}_{fi} - \dot{z}_{ci}$, the dissipativity constraint over the prediction horizon N_c can be written as

$$\text{If } \Delta v_i(k) \geq 0, \quad (3.36)$$

$$0 \leq \underbrace{\begin{bmatrix} c_{min}\Delta v_i(k) \\ c_{min}\Delta v_i(k+1) \\ c_{min}\Delta v_i(k+2) \\ \vdots \\ c_{min}\Delta v_i(k+N_c-1) \end{bmatrix}}_{lb} \leq \underbrace{\begin{bmatrix} F_i(k) \\ F_i(k+1) \\ F_i(k+2) \\ \vdots \\ F_i(k+N_c-1) \end{bmatrix}}_{\bar{0}} \leq \underbrace{\begin{bmatrix} c_{max}\Delta v_i(k) \\ c_{max}\Delta v_i(k+1) \\ c_{max}\Delta v_i(k+2) \\ \vdots \\ c_{max}\Delta v_i(k+N_c-1) \end{bmatrix}}_{ub}$$

If $\Delta v_i(k) < 0$,

$$\underbrace{\begin{bmatrix} c_{max}\Delta v_i(k) \\ c_{max}\Delta v_i(k+1) \\ c_{max}\Delta v_i(k+2) \\ \vdots \\ c_{max}\Delta v_i(k+N_c-1) \end{bmatrix}}_{lb} \leq \underbrace{\begin{bmatrix} F_i(k) \\ F_i(k+1) \\ F_i(k+2) \\ \vdots \\ F_i(k+N_c-1) \end{bmatrix}}_{\bar{0}} \leq \underbrace{\begin{bmatrix} c_{min}\Delta v_i(k) \\ c_{min}\Delta v_i(k+1) \\ c_{min}\Delta v_i(k+2) \\ \vdots \\ c_{min}\Delta v_i(k+N_c-1) \end{bmatrix}}_{ub} \leq 0$$

It can be seen that the constraints on the damping forces exerted by the semi-active dampers dependent on the directions of suspension deflection velocities that are parts of the system states, which is hard to be solved in real-time. There are a few solutions introduced in the literature to handle the constraint problems. In [26], three different approaches are introduced. First, relative damper velocities are predicted as a function of the initial values, road profile over the prediction horizon, and the damping force which is the optimization variable. In this expression, the relative velocities were calculated as outputs. Using the pre-measured, nonlinear damper characteristic which represented by a polygonal line, the constraint on the damping force were obtained. This resulted in a linear quadratic optimization problem with nonlinear constraints. The optimization was solved at each time step by using the MATLAB function *fmincon*. Second, an appropriate linear damper force was chosen and then the relative damper velocities could be approximated. Constraints for the damping force were calculated over the prediction horizon which led to a linear optimization problem. In the third approach, optimal control signals were calculated without considering input constraints. If the obtained damping force was outside the feasible region, the force would be clipped and the maximum or minimum force would be applied. In this way, the resulting forces were no longer the optimal one as desired. In [28], based on a nonlinear half-car model, the control signal was written as a function of the state variables using a set membership approach, which was obtained off-line (could not be calculated online due to the long computational time). Then the only online calculation was a simple evaluation of the approximated off-line calculated function. However, a model with only four states was used in [28]. The necessary number of offline calculations would be much more difficult to cover with a more complex model. Another approach by Ahmed and Svaricek [172] was to apply the road classification strategies together

with MPC algorithm. They used a lidar to measure the road profile in front of the vehicle and the Fast Fourier Transform (FFT) was applied to analyze the frequency content of the road. Then road profiles were divided into different classes such as potholes, bumps, and random road, etc. The MPC optimization problem was solved offline for all road classes and the results were stored in lookup tables. During real-time operation, the control signals were recalled from pre-defined lookup tables based on the current state values and the road class. This method had been validated by using a half-car model, which showed that the MPC developed in [172] provided a better ride comfort than Skyhook and H_∞ control. There is another popular approach which has been employed for several times in the literature. The system can be modelled as a Mixed Logical Dynamical (MLD) hybrid model by using *Hybrid Systems Description Language* (HYSDEL) included in Multi-Parametric Toolbox (MPT). The non-linear and non-convex force constraint is approximated by a piecewise affine approximated with three segments. The results are obtained with the aid of software packages MATLAB and *Yalmip*. *HYSDEL* was developed by Torrisi and Bemporad to obtain MLD models from a high-level textual description of the hybrid dynamics. *HYSDEL* models are used in the hybrid toolbox of MATLAB for modeling, simulating, and verifying the safety properties of hybrid systems. Besides, it also used for designing MPC controllers for linear systems with constraints and hybrid systems and determining equivalent piecewise affine control functions that can be immediately prototyped on hardware. By using the MPT, one can design, analyze, and deploy optimal controllers for constrained linear, nonlinear and hybrid systems. *Yalmip* is a modeling language for advanced description and solution of convex and non-convex optimization problems, allowing the user to concentrate on the high-level model. Based on this method, Cseko, Kvasnica, and Lantos [29] developed an explicit MPC for semi-active suspension system. While Morato and Dugard [32,173] considered a linear parameter varying (LPV) MPC approach. Through simulation, the online computation of the MPC based controller proposed in [25] was tested by Nguyen et. al, with the use of MATLAB, *Yalmip* toolbox, and *Gurobi* solver. Giorgetti [27] proposed MPC hybrid control tools as a way for obtaining control laws with different degrees of optimality. Moreover, Kjellberg and Sundell [30] introduced a new approach to deal with the semi-active dissipative constraint. In their work, a full-car model with the damping coefficient as the control signal was proposed. This resulted in a nonlinear model since the control signal was multiplied with some of the states. The benefit of formulating in this way was that the constraint on damping coefficients were linear (only had a lower and an upper bound). Hence, the MIQP problem no longer existed, but this approach resulted in a nonlinear MPC problem. *ACADO* Toolkit was employed to

handle this NMPC problem, which is an open-source framework for MPC and NMPC problems that is utilizing *qpOASES* as its main low-level solver [35]. The proposed approach had been validated against a realistic vehicle model in *CarMaker*, which was able to run in real-time and the ride quality had been significantly improved compared to a passive suspension system.

In this thesis, an approach that is similar with one of the solutions mentioned in [26], has been used to deal with the semi-active dissipative constraint. For each time step, the previous optimal control signal $U^*(k-1)$ has been collected and then used for further system states. In order to linearize the dissipativity constraint problem, it was assumed that the damping forces over the prediction and control horizons were the same. Under this assumption, further vehicle states could be obtained, which were then used for calculating the boundaries of the optimal control signals $U^*(k)$ for the next time step. Based on above description, the proposed MPC control algorithm is concluded as follows:

Algorithm 1: MPC algorithm

Input: state vector $X(k)$, disturbance input vector $W(k)$, control signal vector $U^*(k-1)$.

Output: optimal control signal $U^*(k)$.

Assume that $U(k|k) = U(k+1|k) = \dots = U(k+N_c-1|k) = U^*(k-1)$,

and $W(k|k) = W(k+1|k) = \dots = W(k+N_c-1|k) = W(k-1)$.

Obtain $\hat{X}(k+1|k), \dots, \hat{X}(k+N_p|k)$ according to Equation (3.28).

Calculate boundaries of the control signals according to Equation (3.36).

Obtain optimal solution $U^*(k)$ according to Equation (3.33).

return the optimal control signal $U^*(k)$ for next iteration.

3.2.2 Integrated Skyhook-LQR controller for semi-active suspension systems

The MPC approach is difficult to be implemented in practical applications due to its high computational costs. Alternatively, the LQR is another option for realizing coordinated suspension control. It is a popular control technique used in both industry and academia to control linear dynamic systems because of its simplicity, optimality, integrability, and design flexibility. However, it cannot properly handle the damping constraints. Thus, developing a computationally efficient, robustness, and effective suspension controller for practical applications becomes a key task in the automotive industry. In this thesis, the Skyhook and LQR strategies are integrated to attenuate the undesired vibration in vertical, roll, and pitch directions. The reasons of choosing these two control approaches are listed as follows:

- Skyhook is the most widely used control method for vehicle suspension systems because of its simplicity, robustness, and efficiency. Using Skyhook as the base control algorithm can guarantee the satisfactory ride quality and robustness. In addition, this approach considers the damping constraint in terms of the force applied direction, but not the force boundary.
- However, Skyhook cannot handle the multi-objective optimization problem, such as improving the cabin dynamics in vertical, roll, and pitch directions at the same time. But LQR naturally seeks global optimum solution for multi-objective problems. Thus, LQR is suitable for further optimizing the control performances.
- LQR request less computational power than the MPC algorithm, so that it is easier and cheaper for practical applications, especially the suspension control needs to be processed within the range of 100Hz to 200Hz.

The schematic of the proposed Skyhook-LQR approach is shown in Figure 3.9. In this algorithm, the control signal F_i is determined by both Skyhook and LQR modules. The methodology of the proposed control system and the gain-adaptive algorithm are introduced in the following sections.

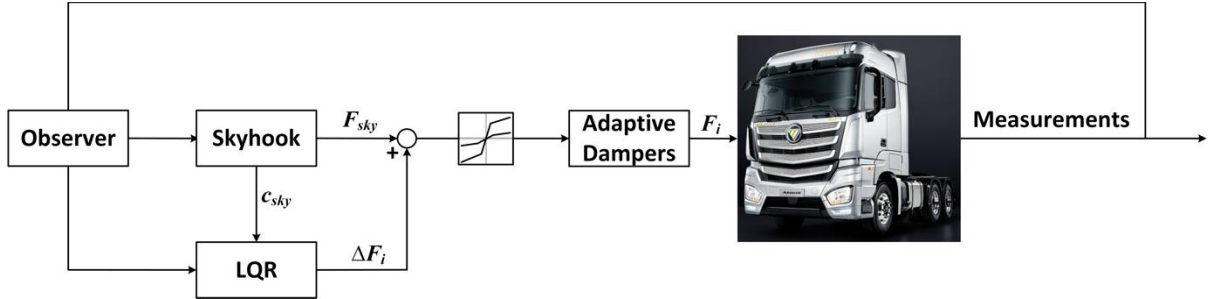


Figure 3.9. Integrated Skyhook-LQR control

3.2.2.1 Skyhook control

The Skyhook algorithm is a classical control strategy for vehicle suspension systems, which mainly focuses on the vertical vibration attenuation. The concept of the Skyhook control is that the sprung mass connects to an imaginary sky through a damper, and a spring installs between the cabin (sprung mass) and the chassis frame (unsprung mass), as shown in Figure 3.10.

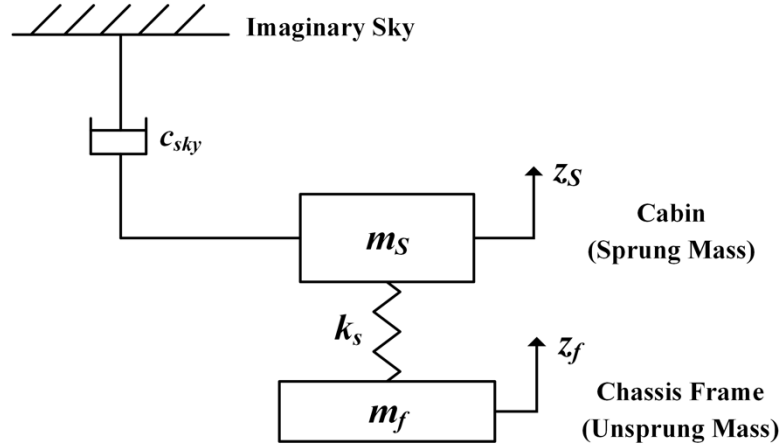


Figure 3.10. Skyhook control approach

This approach in the literature consists of two control states, i.e., low-damped and high-damped. The damping coefficient c_{sky} changes according to the sign of the product of the sprung mass and unsprung mass velocities as follows:

$$\begin{cases} c_{sky} = c_{max}, & \text{if } \dot{z}_s(\dot{z}_s - \dot{z}_f) \geq 0 \\ c_{sky} = c_{min}, & \text{if } \dot{z}_s(\dot{z}_s - \dot{z}_f) < 0 \end{cases} \quad (3.37)$$

It can be seen that the damping coefficient c_{sky} switches between the maximum value c_{max} and the minimum value c_{min} . In the active continuous skyhook control strategy, if the product $\dot{z}_s(\dot{z}_s - \dot{z}_f)$ is greater than or equal to zero, the high state is applied. If the product is negative, the damper is adjusted to the low state. After determining the damping coefficient, the corresponding control signal F_{sky} is calculated by

$$F_{sky} = -c_{sky}\dot{z}_s \quad (3.38)$$

This comfort-oriented control law focuses on reducing the absolute vertical acceleration of the sprung mass. Hence, the Skyhook approach has been widely used in minimizing the vertical vibrations [174]. But the classic Skyhook control is not adequate for optimizing the cabin's roll and pitch dynamics, so that it is not able to handle the coordinated control problems. To solve this multi-objective optimization problem, the LQR approach is chosen to adjust the Skyhook control signals F_{sky} in real-time with a focus on optimizing the cabin's pitch and roll dynamics.

3.2.2.2 Linear Quadratic Regulator (LQR)

For the proposed linear time-invariant system, the cost function J is firstly defined in a quadratic form shown in Eq.(3.39), which is very similar to the MPC's shown in Eq.(3.31). It aims to minimize the cabin's pitch and roll accelerations.

$$J = \int_0^{\infty} Y^T Q_y Y + U^T R_y U dt \quad (3.39)$$

in which the output penalty matrix $Q_y \geq 0$, the control signal weighting matrix $R_y > 0$, and the output vector Y is

$$Y = \dot{X} = [\dot{z}_s \quad \dot{\theta}_s \quad \dot{\phi}_s \quad \ddot{z}_s \quad \ddot{\theta}_s \quad \ddot{\phi}_s]^T \quad (3.40)$$

The above cost function needs to be transformed into the standard form as follows:

$$J = \int_0^{\infty} X^T \underbrace{C^T Q_y C}_{\bar{Q}} X + U^T \underbrace{(R_y + D^T Q_y D)}_{\bar{R}} U dt \quad (3.41)$$

The feedback control law that minimizes the above cost function is defined as

$$U = -KX \quad (3.42)$$

in which the feedback gain K can be calculated by

$$K = R^{-1} B^T P \quad (3.43)$$

where P is solved from the continuous time algebraic Riccati equation (ARE) as follows:

$$A^T P + PA - PBR^{-1}B^T P + Q = 0 \quad (3.44)$$

As shown in Figure 3.9, based on the Skyhook control signal F_{sky} , the LQR module adjusts the final control signal F_i by the increment term ΔF_i . In this way, these two controllers work together to provide excellent coordinated control performances in vertical, pitch, and roll directions.

3.2.3 Gain-adaptive algorithms

By choosing different gain values for the output penalty matrix Q_y , the control objectives can be easily adjusted. Hence, a proper weighting matrix can compromise the motions among vertical, pitch, and roll

directions to provide a better overall performance. However, it is not possible to use only one set of gains to achieve the optimum performance on various road conditions. For example, when tuning a suspension controller, higher weights are needed to attenuate the significant disturbances, while lower weights are preferred when driving on better road conditions. This conflict makes the gain-adaptive technique necessary to continuously isolate the undesired vibrations as effective as possible.

Technically, one of the most commonly used solutions is to intelligently adjust the weighting matrices online based on the available measurements. With the updated output penalty matrix, the gain-adaptive Skyhook-LQR controller can attenuate the vibrations in the human-sensitive range to the most extent. In this study, two 6-axis inertial measurement unit (IMU) are installed on the chassis frame and the cabin floor to detect their motions in real-time. Thus, the sensor measurements can be processed and analyzed for adjusting the control gains. Two gain-adaptive algorithms are introduced in this section. Specifically, the algorithm A is processed according to the motions of the chassis frame (disturbance-based), while the algorithm B focuses on the dynamics of the cabin (state-based).

3.2.3.1 Algorithm A: disturbance-based

The structure of the first gain-adaptive algorithm is presented in Figure 3.11, which aims to select the optimal gains according to the result of its vibration level grading algorithm. As an example, the gains and weightings used in this study are listed in Table 4.1, which are tuned by trial-and-error based on the co-simulation introduced in Chapter 4. Specifically, the non-adaptive weighting matrices are tuned for bumpy roads that are not ideal for paved road surfaces. The gain-adaptive algorithm is utilized to adjust the LQR weights for rapid changed disturbances.

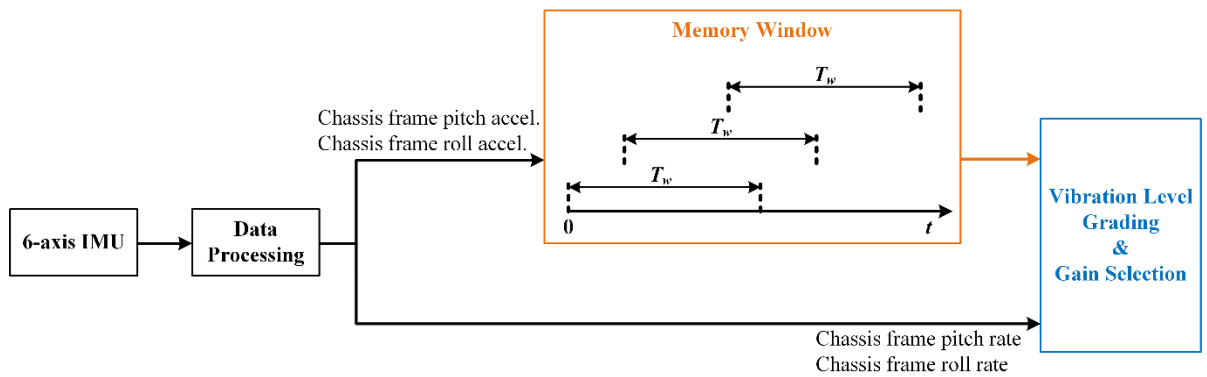


Figure 3.11. Gain-adaptive algorithm

Based on the real-time IMU measurements, the vibration levels of the chassis frame in the pitch $\bar{\theta}_f$ and roll $\bar{\varphi}_f$ directions are graded based on the angular rates and the angular accelerations as follows:

$$\begin{aligned}\bar{\theta}_f &= 5[abs(\dot{\theta}_f) + rms(a_{\theta_w})] \\ \bar{\varphi}_f &= 10[abs(\dot{\varphi}_f) + rms(a_{\varphi_w})]\end{aligned}\tag{3.45}$$

in which the absolute values of the pitch rate $\dot{\theta}_f$ and roll rate $\dot{\varphi}_f$ of the chassis frame are directly measured by the 6-axis IMU; the frequency-weighted root-mean-square (RMS) values of the pitch acceleration a_{θ_w} and roll acceleration a_{φ_w} are calculated over a certain period of time T_w that is named as the memory window. In detail, the measured angular rates are processed to get the corresponding angular accelerations, and then the frequency-weighted RMS values are calculated according to the definitions in ISO 2631-1 [87].

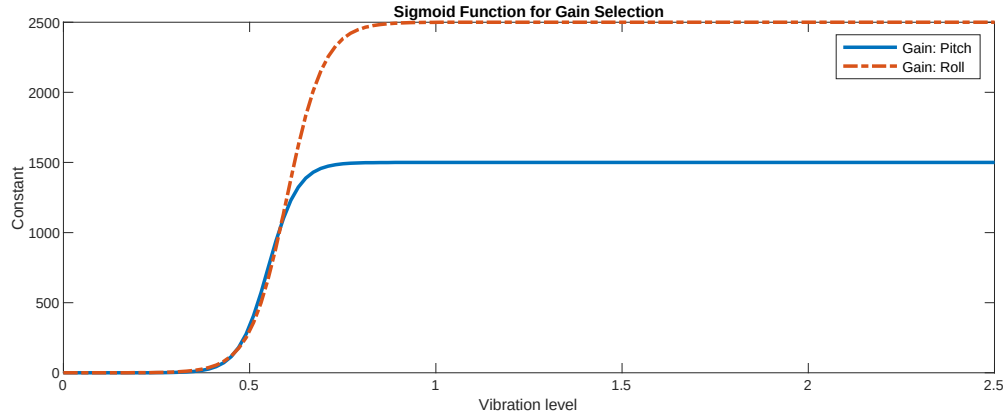


Figure 3.12. The sigmoid function for gains selection

The essential step of the gain-adaptive technique is to select the optimal gain values based on the vibration level grading results. In this study, the sigmoid function is used to schedule the gains of pitch q_5 and roll q_6 which guarantees a relatively smooth transition, as shown in Figure 3.12.

$$\begin{aligned}q_5 &= \frac{1500}{1 + e^{-25*(\bar{\theta}_f - 0.55)}} \\ q_6 &= \frac{2500}{1 + e^{-20*(\bar{\varphi}_f - 0.6)}}\end{aligned}\tag{3.46}$$

3.2.3.2 Algorithm B: state-based

Different from the above algorithm A, algorithm B selects the control gains based on the IMU measurement of the cabin motions, i.e., the cabin pitch rate $\dot{\theta}_s$ and its roll rate $\dot{\phi}_s$. As shown in Figure 3.13, the trends of cabin's roll rate and pitch rate are firstly defined as follows:

$$\begin{aligned}\dot{\theta}_{slope}(k) &= \text{sign}[\dot{\theta}_s(k) - \dot{\theta}_s(k-1)] \\ \dot{\theta}_{dir}(k) &= \dot{\theta}_{slope}(k) + \dot{\theta}_{slope}(k-1)\end{aligned}\tag{3.47}$$

$$\begin{aligned}\dot{\phi}_{slope}(k) &= \text{sign}[\dot{\phi}_s(k) - \dot{\phi}_s(k-1)] \\ \dot{\phi}_{dir}(k) &= \dot{\phi}_{slope}(k) + \dot{\phi}_{slope}(k-1)\end{aligned}\tag{3.48}$$

If sign of $\dot{\theta}_{dir}(k)$ equals to 1 or -1, it means that the trend of the pitch rate maintains the same. Otherwise, if $\text{sign}[\dot{\theta}_{dir}(k)]$ equals to 0, it means that the trend of the pitch rate changes in this time step and reaches its local maximum value.

As a result, a coefficient is determined for pitch direction:

$$\begin{aligned}\dot{\theta}_{coef}(k) &= \text{abs}[\dot{\theta}_s(k-1)], \text{ if } \text{sign}[\dot{\theta}_{dir}(k)] = 1 \text{ or } -1 \\ \dot{\theta}_{coef}(k) &= \text{abs}[\dot{\theta}_s(k)], \text{ if } \text{sign}[\dot{\theta}_{dir}(k)] = 0\end{aligned}\tag{3.49}$$

Same logic is used for the roll direction as follows:

$$\begin{aligned}\dot{\phi}_{coef}(k) &= \text{abs}[\dot{\phi}_s(k-1)], \text{ if } \text{sign}[\dot{\phi}_{dir}(k)] = 1 \text{ or } -1 \\ \dot{\phi}_{coef}(k) &= \text{abs}[\dot{\phi}_s(k)], \text{ if } \text{sign}[\dot{\phi}_{dir}(k)] = 0\end{aligned}\tag{3.50}$$

Thus, the resulted control gain-adaptive algorithm for pitch $q_5(k)$ and roll $q_6(k)$ directions are

$$\begin{aligned}q_5(k) &= \frac{1500}{1 + e^{-30[\dot{\theta}_{coef}(k) - 0.05]}} \\ q_6(k) &= \frac{2500}{1 + e^{-30[\dot{\phi}_{coef}(k) - 0.1]}}\end{aligned}\tag{3.51}$$

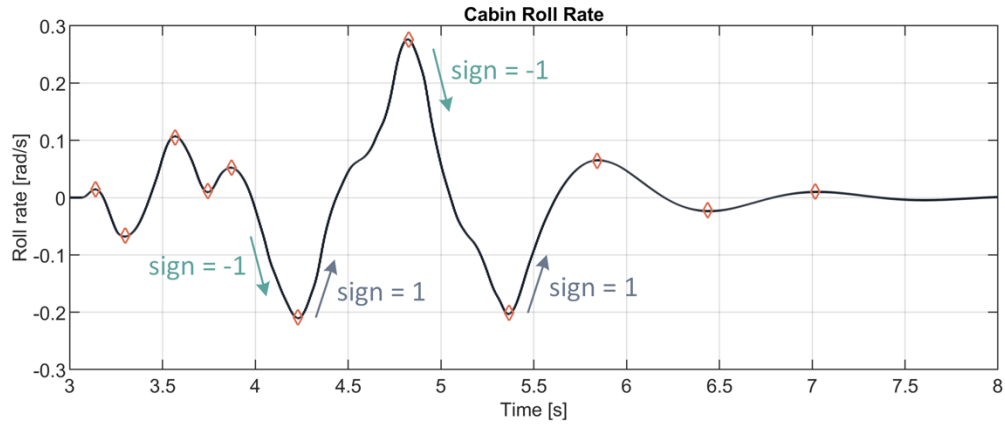


Figure 3.13. Cabin roll rate response

The expected performances after implementing the proposed gain-adaptive algorithms on the adaptive suspension system are:

- When driving on bumpy roads, the gain-adaptive Skyhook-LQR controller should be able to provide competitive performances as the non-adaptive Skyhook-LQR controller. Since the gains of the non-adaptive controller are tuned for harsh road conditions.
- When driving on paved road, the gain-adaptive Skyhook-LQR controller should provide better ride quality than the non-adaptive Skyhook-LQR controller, in which the vertical vibration is usually dominate.

It can be seen that Algorithm A calculates the frequency-weighted acceleration in real-time for grading the disturbance level, which spends more computational effort but fulfills the comfort criteria defined in the ISO 2631. Differently, Algorithm B utilizes the available state measurements, requiring less computational cost. In addition, the gain values generated by Algorithm A are more sensitive to the disturbance level, which scarifies the system robustness to some extent. Note that a comprehensive comparison of these two gain-adaptive algorithms can be found in Chapter 4.

Overall, the integrated Skyhook-LQR controller with gain-adaptive algorithm can intelligently adjusting its control objective based on the available sensor measurements to achieve the optimum overall performances. For example, when the vertical vibration is dominant, the controller puts more efforts into minimizing the cabin's vertical acceleration; when the roll and/or pitch motions become larger, the corresponding gains of LQR increase to reduce the cabin's rotational motions.

3.3 State Estimation

The Kalman Filter is a state estimator that is widely used in practical applications. As introduced in Section 3.1.2, the continuous time state-space form of the adaptive cabin suspension system is:

$$\underbrace{\begin{bmatrix} \dot{z}_S \\ \dot{\theta}_S \\ \dot{\phi}_S \\ \ddot{z}_S \\ \ddot{\theta}_S \\ \ddot{\phi}_S \end{bmatrix}}_{\dot{\underline{x}}} = A \underbrace{\begin{bmatrix} z_S \\ \theta_S \\ \phi_S \\ \dot{z}_S \\ \dot{\theta}_S \\ \dot{\phi}_S \end{bmatrix}}_{\underline{x}} + B_u \underbrace{\begin{bmatrix} F_1 \\ F_2 \\ F_3 \\ F_4 \end{bmatrix}}_U + B_w \underbrace{\begin{bmatrix} z_f \\ \theta_f \\ \phi_f \\ \dot{z}_f \\ \dot{\theta}_f \\ \dot{\phi}_f \\ \ddot{x}_S \\ \ddot{y}_S \end{bmatrix}}_W \quad (3.52)$$

The procedure of developing a Kalman Filter estimation algorithm can be concluded as follows:

- 1) Give initial state information: The initial values of the above system states are all zeros.
- 2) Calculate the measurable variables based on the predicted states: As shown in Figure 3.14, two 6-axis IMUs and four height sensors were installed on cabin, chassis frame, and four suspension corners, respectively. The available measurements are the vertical accelerations, longitudinal accelerations, lateral accelerations, roll rates, and pitch rates of the cabin and chassis frame, as well as the suspension displacements at four corners. Thus, the measurable vector y_m can be written as

$$y_m = [\dot{\theta}_S \quad \dot{\phi}_S \quad \ddot{z}_S] \quad (3.53)$$

- 3) Calculate the estimation error: It is the difference between the measurement y_m and the predicted value y_p which is usually formed as

$$e(k) = y_m(k) - y_p(k) \quad (3.54)$$

$$y_p(k) = C_o X_o(k) + \underbrace{[D_u \quad D_w]}_{D_o} \begin{bmatrix} U(k) \\ W(k) \end{bmatrix}$$

in which C_{od} is the discretized output matrix of the observer,

- 4) Calculate the corrected state estimation by adding the corrective term $Ke(k)$ to the estimated state $X_o(k)$:

$$X_o(k) = X_p(k) + Ke(k) \quad (3.55)$$

in which K is the Kalman Gain which minimizes the posteriori error covariance that can be calculated by

$$K = P_k C_{od}^T (C_{od} P_k C_{od}^T + R_y)^{-1}$$

$$P_k = (1 - KC_{od}) P_{k-1} \quad (3.56)$$

$$P_{k+1} = A_{od}(1 - KC_{od}) P_k A_{od}^T + Q_y$$

in which P_k is the state error covariance, Q_y is the state noise covariance, and R_y is the measurement noise covariance.

- 5) Calculate the predicted states for the next time step based on the present state estimation:

$$X_o(k+1) = A_{od}X_o(k) + B_{od}U(k) + K[y_m(k) - y_p(k)] \quad (3.57)$$

in which $X_o(k)$ is the observed state vector at time k , A_{od} is the discretized state transition matrix of the observer, and B_{od} is the discretized input matrix of the observer.

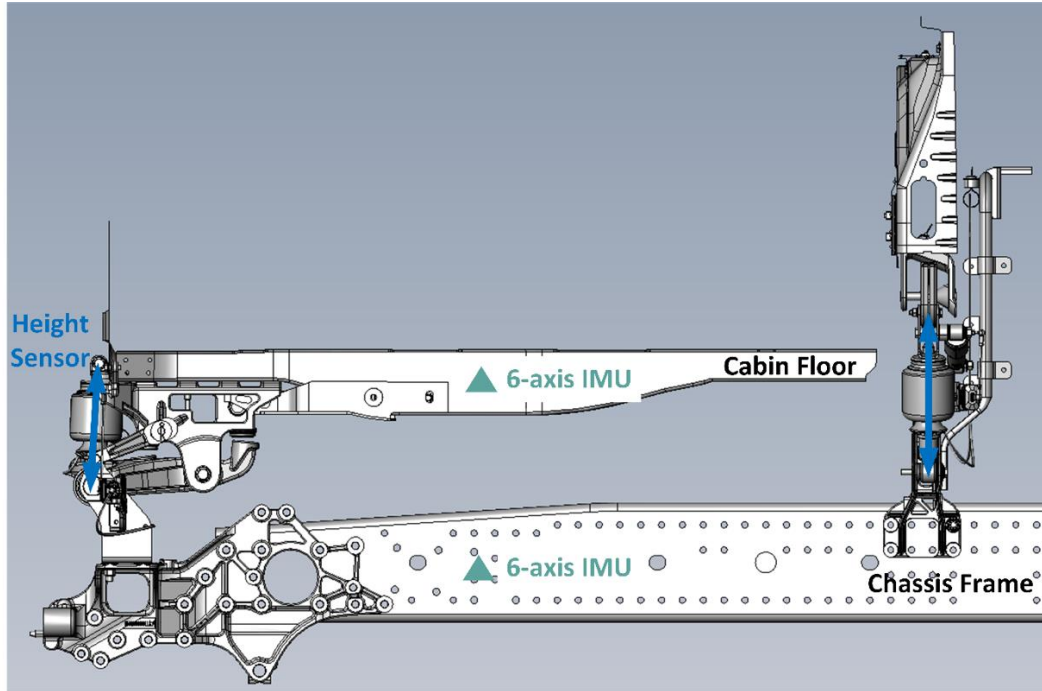


Figure 3.14. Sensor location

3.4 Summary

The study presented in this chapter can be summarized as follows:

- Two coordinated control systems were developed for semi-active suspension systems, namely MPC and integrated Skyhook-LQR. The two coordinated controllers aim to improve vehicle dynamics in vertical, roll, and pitch directions simultaneously. The MPC approach has great potential in this kind of multi-objective optimization problem with known constraints, however, it is difficult to be implemented in practical applications due to the high computational cost. To solve this issue, the integrated Skyhook-LQR controller was introduced. As mentioned in many studies, the Skyhook approach can provide satisfactory improvement in minimizing the body's vertical acceleration, and the LQR is capable of handling multi-objective control problems. Besides, these two algorithms require much less computational cost than MPC. Thus, the integrated Skyhook-LQR is recommended for practical suspension control tasks.
- Two gain-adaptive algorithms were developed for the integrated Skyhook-LQR controller to achieve the optimum performance in different driving conditions, namely disturbance-based gain-adaptive algorithm (A) and state-based gain-adaptive algorithm (B). As the name implies, the disturbance-based algorithm intelligently adjusts the weighting matrix of the LQR algorithm according to the disturbance measurements. While the state-based approach generates the LQR gains based on the measurements of the vehicle states. In this way, the gain-adaptive Skyhook-LQR adjust its control objective according to the current driving condition. For example, when the vehicle is driven on the paved road, the Skyhook algorithm is mainly used because the vertical vibration is dominant compared with rotational motions. When the vehicle is driven on rough roads, the LQR impact in the suspension control becomes more dominant because the disturbances in roll and pitch direction increase.
- A flat-nose truck's cabin suspension system was considered in this study. In detail, the dynamic equations in vertical, roll, and pitch directions of the COE system were derived to represent the cabin's motions with a high level of accuracy. Different from the mathematical models of passenger cars, the COE model considers the special tilting mechanism which affects the cabin's vertical and pitch dynamics that should not be neglected.
- An observer based on the Kalman Filter approach was developed for the cabin system to estimate the unmeasurable states based on the mathematical model mentioned above.

Chapter 4

Simulation and Experimental Studies of Coordinated Semi-Active Suspensions

This chapter presents the performance examination of the proposed adaptive suspension systems through numerical and experimental studies. The discussion and analysis of the simulation and experimental results provide a comprehensive evaluation of the proposed suspension systems and offer insights into the design and implementation of these systems for future applications.

4.1 Co-simulation

A high-fidelity truck model was developed in ADAMS/Car to evaluate the performances of the cabin semi-active suspension system. The truck model was built based on a flat-nose truck produced by Foton Daimler, which had a cabin and primary suspensions. The cabin suspension was installed between the cabin and the chassis frame. The three-dimensional drawing in SolidWorks is shown in Figure 3.2. In the three-axle primary suspension system shown in Figure 4.1, two leaf springs were installed at the front, together with two shock absorbers. A two-axle air suspension system was used at the rear of the primary suspension, which contained four shock absorbers, eight air springs, and two anti-roll bars. Besides, the Fiala tire model was utilized since it is accurate enough to predict tire characteristics within the linear range and is very good in ride comfort analysis [175]. Note that more details of the proposed truck model can be found in Appendix A.

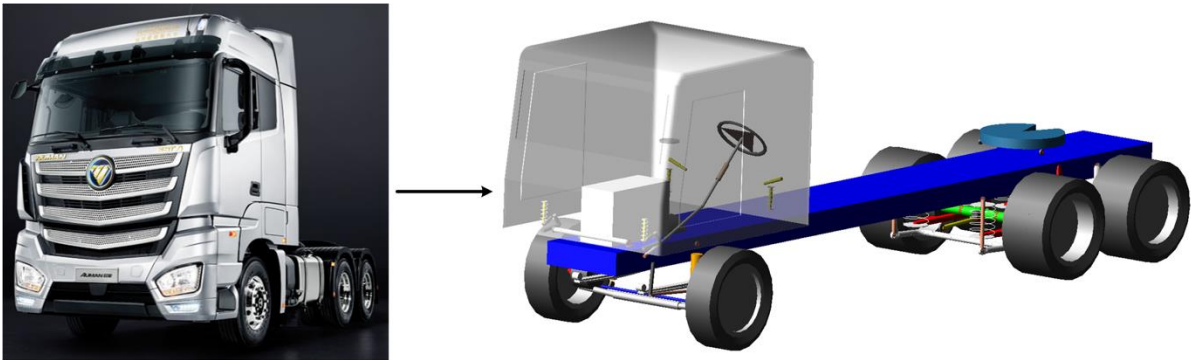


Figure 4.1. High-fidelity flat-nose truck model in ADAMS/Car

The control performance of the adaptive COE suspension system was first evaluated by running co-simulations between MATLAB/Simulink and ADAMS/Car, as shown in Figure 4.2. In this process,

the two software programs communicated with each other during run-time to simulate the dynamic responses of the truck. In this study, the control systems were implemented in MATLAB/Simulink, and the high-fidelity truck model was executed in ADAMS/Car.

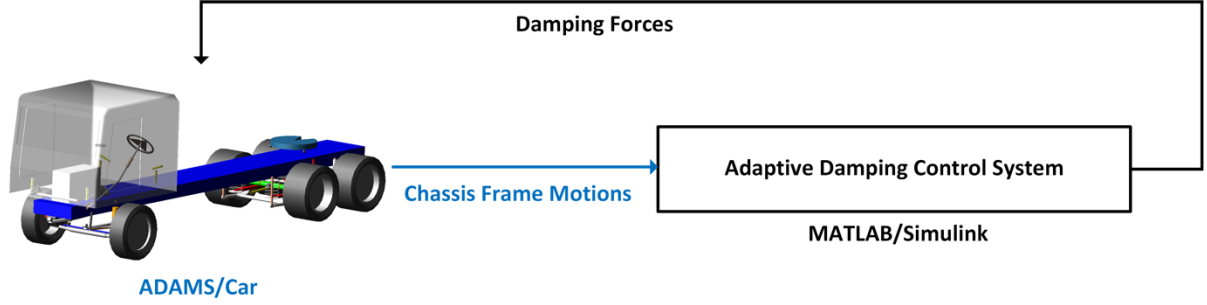


Figure 4.2. Co-simulation between ADAMS/Car and MATLAB/Simulink

The parameters of the COE system used in the co-simulation are listed in Table 3.1, and the control gains are listed in Table 4.1. The *qpOASES* was used for solving the MPC quadratic programming problem in MATLAB, which provides an interface for many programming languages, including MATLAB, Python, and C/C++. It offers a variety of options for solving QP problems, including primal-dual interior-point methods and active-set methods.

Table 4.1. The control gains of cabin suspension controllers

Controller	Gains and weighting matrices
Model Predictive Control (MPC)	State cost matrix: $R = 10^{-3} \text{diag}(1, 1, 1, 1)$ Control cost matrix: $Q = \text{diag}(0, 0, 0, 3000, 1500, 2500)$
Skyhook	Skyhook gain: $c_{sky} = 5000 \text{ Ns/m}$
Integrated Skyhook-LQR (Non-adpt. S-LQR)	Skyhook gain: $c_{sky} = 5000 \text{ Ns/m}$ Control signal penalty matrix: $R_y = 10^{-3} \text{diag}(1, 1, 1, 1)$ Output penalty matrix: $Q_y = \text{diag}(0, 0, 0, 0, 1500, 2500)$
Gain-adaptive Skyhook-LQR with algorithm A or B (Gain-adpt. S-LQR A or B)	Skyhook gain: $c_{sky} = 5000 \text{ Ns/m}$ Control signal penalty matrix: $R_y = 10^{-3} \text{diag}(1, 1, 1, 1)$ Output penalty matrix: $Q_y = \text{diag}(0, 0, 0, 0, q_5, q_6)$ Note: the selection of q_5 and q_6 can be found in Section 3.2.3.

4.2 Vibration Assessment

At present, there are two standard methods to evaluate driver discomfort in vehicles, i.e., BS 6841 and ISO 2631-1. This thesis considers the translational and rotational frequency-weighted accelerations, velocities, and displacements to evaluate the suspension performance. In detail, the vibration exposure is usually expressed in terms of frequency-weighted root-mean-square (RMS) acceleration $\bar{a}$, given by:

$$\bar{a} = \sqrt{\frac{1}{T} \int_0^T a_w^2 dt} \quad (4.1)$$

where T is the exposure duration considered, and a_w is the frequency-weighted acceleration along a given axis.

The frequency-weighted acceleration along the different axis can be calculated in the frequency or time domain. However, since the frequency domain analysis is used for linear models, the time domain formulation is used for multi-body dynamic models. In the time domain, the frequency-weighted acceleration along the different axis can be calculated in the following order:

- 1) Along each axis, the corresponding weighting function is applied to the acceleration response of the vehicle in order to calculate the weighted acceleration response, as shown in the following chart. Specifically, a_z , a_y , a_x , a_ϕ , a_θ are the un-weighted vertical, lateral, longitudinal, roll and pitch acceleration, respectively. W_k , W_d , W_e are the corresponding transfer function of the weighting factor along vertical, horizontal and rotational axes, respectively.

Un-weighted acceleration		Weighting transfer function		Weighted acceleration
$a_z(t)$	→	W_k	→	$a_{wz}(t)$
$a_x(t)$	→	W_d	→	$a_{wx}(t)$
$a_y(t)$	→	W_d	→	$a_{wy}(t)$
$a_\phi(t)$	→	W_e	→	$a_{w\phi}(t)$
$a_\theta(t)$	→	W_e	→	$a_{w\theta}(t)$

- 2) a_{wz} , a_{wy} , a_{wx} , $a_{w\phi}$ and $a_{w\theta}$ are the corresponding weighted accelerations. The RMS of the weighted accelerations are calculated as weighted rms acceleration that is used for vibration assessment propose:

$$\begin{aligned}\bar{a}_z &= RMS(a_{wz}(t)) = \sqrt{\frac{1}{T} \int_0^T a_{wz}(t)^2 dt} \\ \bar{a}_y &= RMS(a_{wy}(t)) = \sqrt{\frac{1}{T} \int_0^T a_{wy}(t)^2 dt} \\ \bar{a}_x &= RMS(a_{wx}(t)) = \sqrt{\frac{1}{T} \int_0^T a_{wx}(t)^2 dt} \\ \bar{a}_\phi &= RMS(a_{w\phi}(t)) = \sqrt{\frac{1}{T} \int_0^T a_{w\phi}(t)^2 dt} \\ \bar{a}_\theta &= RMS(a_{w\theta}(t)) = \sqrt{\frac{1}{T} \int_0^T a_{w\theta}(t)^2 dt}\end{aligned}\tag{4.2}$$

where $\bar{a}_z$, $\bar{a}_y$, $\bar{a}_x$, $\bar{a}_\phi$, and $\bar{a}_\theta$, are the weighted RMS acceleration in vertical, lateral, longitudinal, roll and pitch direction, respectively. Noted that most of the studies have only considered the weighted vertical RMS value as the vehicle ride measured since the vertical vibration is dominant.

- 3) When the effect of the horizontal vibration is also considerable particularly during driving over rough road, the standard also requires the evaluation of total whole body vibration (WBV) exposure through vector summation of the translational weighted accelerations, such that:

$$a_v = \sqrt{(k_x^2 \bar{a}_x^2 + k_y^2 \bar{a}_y^2 + k_z^2 \bar{a}_z^2)}\tag{4.3}$$

where a_v is the vector summation of the weighted accelerations along the three translational axes. The constants k_x , k_y , and k_z are additional weightings used to account for human sensitivity to horizontal and vertical vibration, as recommended in ISO 2631-1 based on the evaluation purpose, as bellow:

- For health evaluation purpose:

$$k_x = k_y = 1.4$$

$$k_z = 1$$

- For comfort and perception purpose:

$$k_x = k_y = k_z = 1$$

It should be noted that acceptable values of vibration magnitude depend on many factors that vary with each application. It means that the reactions at various magnitudes depend on passenger expectations concerning the trip duration, the type of activities passengers expect to accomplish (sleeping, reading, eating, etc.), and many other factors (acoustic noise, temperature, etc.). Therefore, there is no limitation defined in the related standards.

4.3 Simulation Results

In this section, the co-simulation results are presented and discussed, focusing on evaluating the Kalman Filter observer and the semi-active suspension systems. The tested truck was driven on different road surfaces at various speeds, such as dual bumps roads at 25 km/h, offset-bump road at 20 km/h, and paved roads at different speeds.

4.3.1 Observer evaluation

The Kalman Filter state observer introduced in Section 3.3 is examined and discussed in the following paragraphs. In the first scenario, the simulation was based on the speedbumps road profile designed to reduce vehicle speed and create a safer environment for pedestrians, cyclists, and other road users. They are particularly useful in areas with heavy traffic that needs to reduce vehicle speed for safety reasons, such as residential areas, school zones, shopping areas, etc. Speedbumps are typically made from asphalt, concrete, or rubber, and are installed permanently or temporarily on the road. They are usually about 3 to 4 inches in height and are placed across the road in a manner that requires vehicles to slow down to traverse them safely. However, speedbumps can also cause discomfort for drivers and

passengers and may damage vehicles. As a result, it is essential to consider multiple factors when selecting and installing speedbumps, such as the road condition, vehicle type, speed limitation, traffic volume, etc. The dual bumps road profile in Figure 4.3 was used in this test, on which the truck was driven at 25 km/h. Since the two front wheels hit the speedbump simultaneously, the resulting motions of the vehicle were mainly in the vertical and pitch directions.

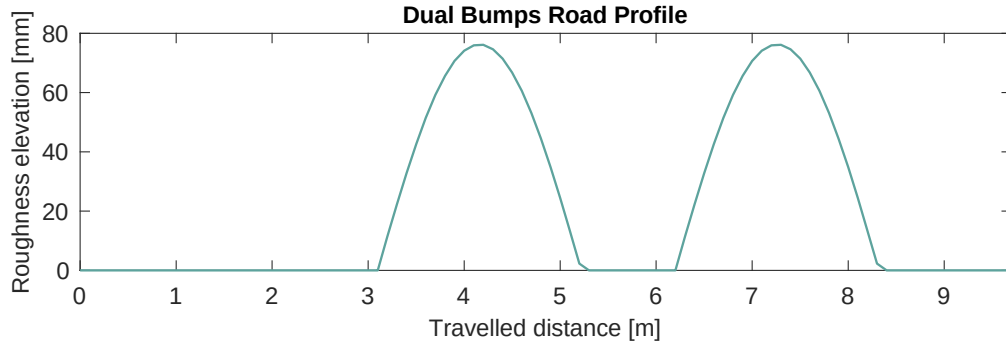


Figure 4.3. Dual bumps road profile

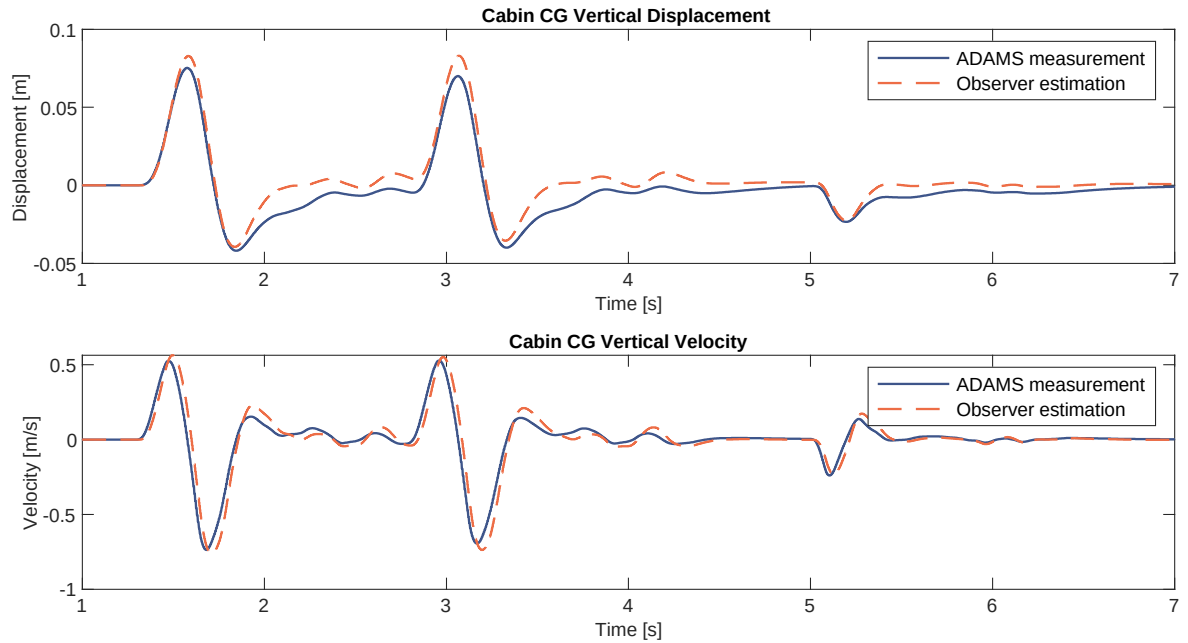


Figure 4.4. Observer estimation of cabin vertical motions (dual bumps at 25 km/h)

The cabin responses in vertical and pitch directions are presented in Figure 4.4 and Figure 4.5, respectively. The measurements were obtained from the high-fidelity truck model with air suspensions

developed in ADAMS/Car, which were considered as the benchmark in this study. As can be seen, the ADAMS measurement and the observer estimation showed a high level of similarity in vertical and pitch directions.

An offset-bump road profile was utilized in the second scenario, as shown in Figure 4.6. It consists of two offset rounded speedbumps with 3 inches in height. In this scenario, the vehicle motions in vertical, pitch, and roll directions were excited, and the truck passed the bumps at 20 km/h.

The cabin responses in vertical, pitch, and roll directions are presented in Figure 4.7, Figure 4.8, and Figure 4.9, respectively. As can be seen, the estimation of the vertical displacement showed acceptable errors, and the observed vertical velocity provided satisfactory accuracy. The measured and estimated differences in the pitch angle were slightly large, mainly due to the small movements in the pitch direction. Meanwhile, the ADAMS measurement and observer estimation showed a high level of similarity in the roll direction.

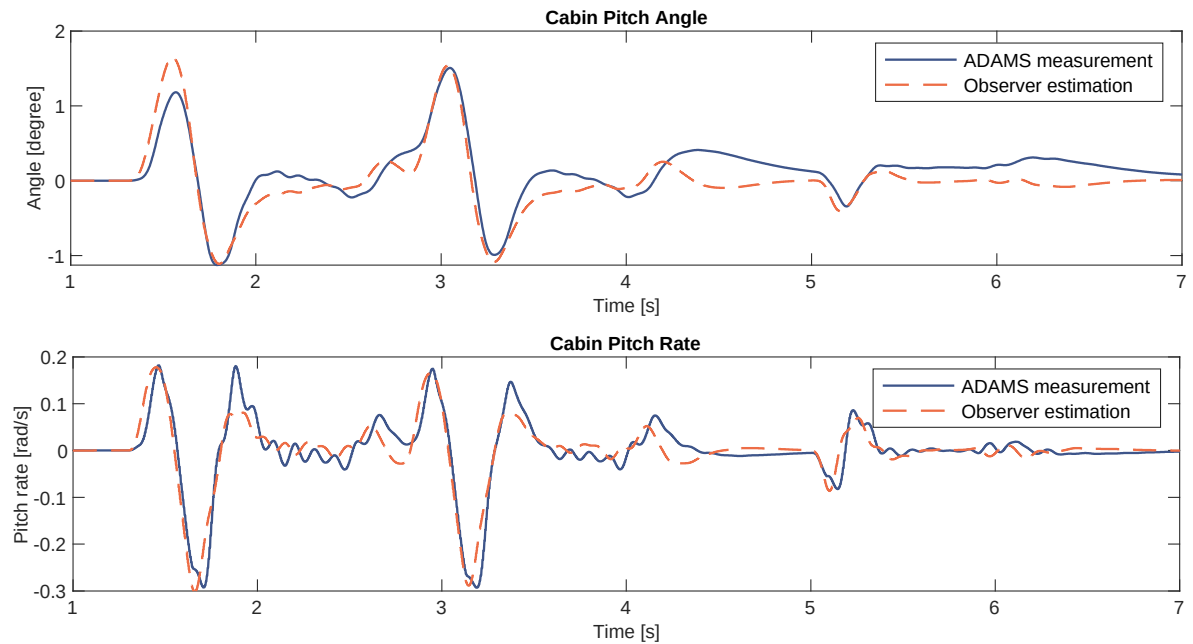


Figure 4.5. Observer estimation of cabin pitch motions (dual bumps at 25 km/h)

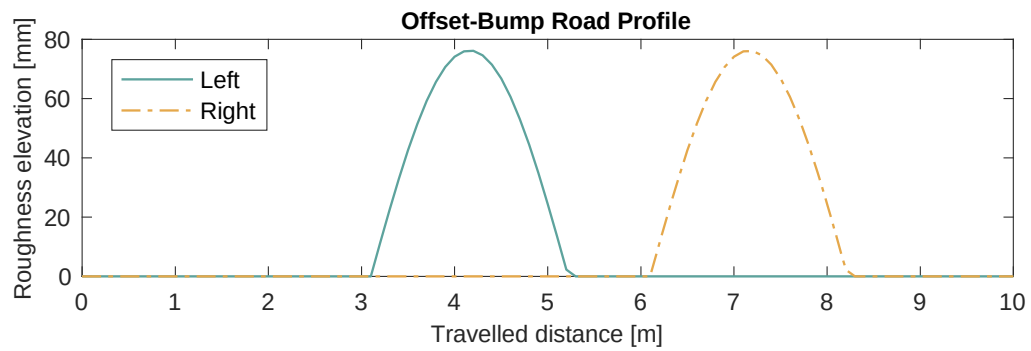


Figure 4.6. Offset-bump road profile

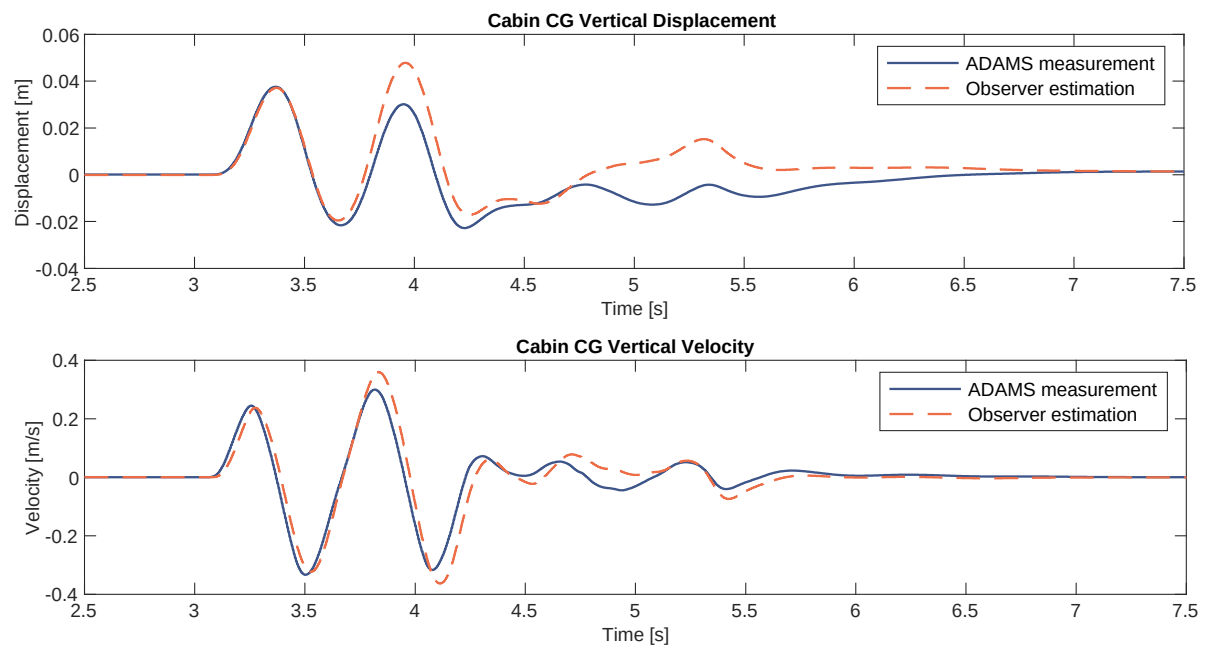


Figure 4.7. Observer estimation of cabin vertical motions (offset-bump at 20 km/h)

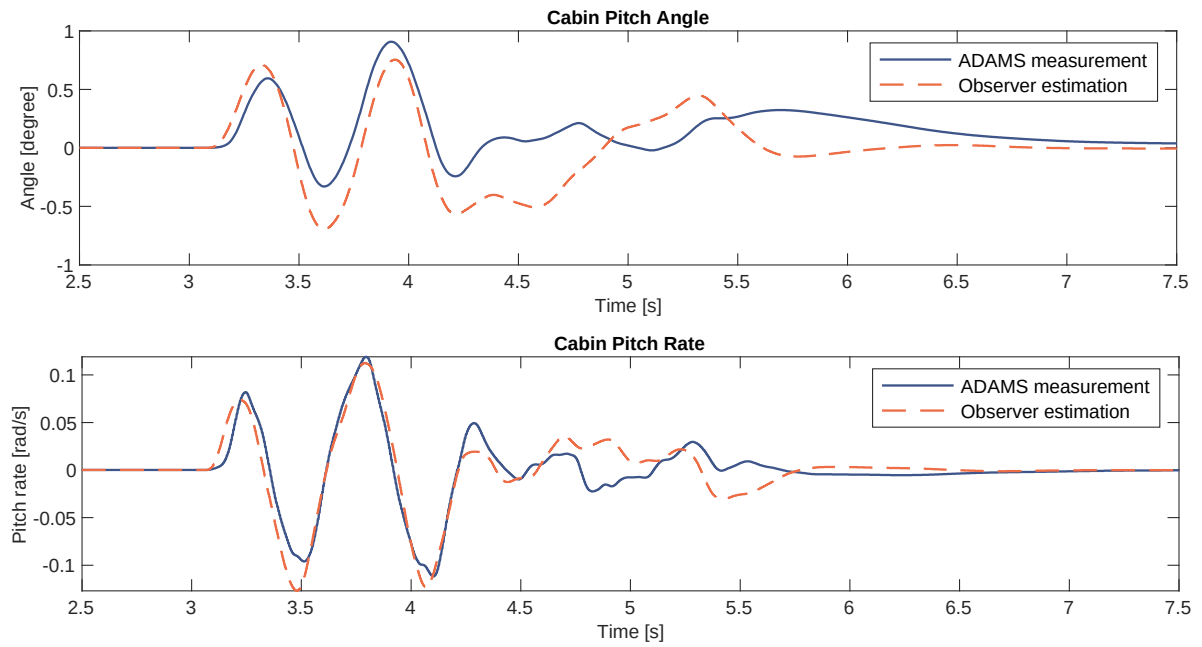


Figure 4.8. Observer estimation of cabin pitch motions (offset-bump at 20 km/h)

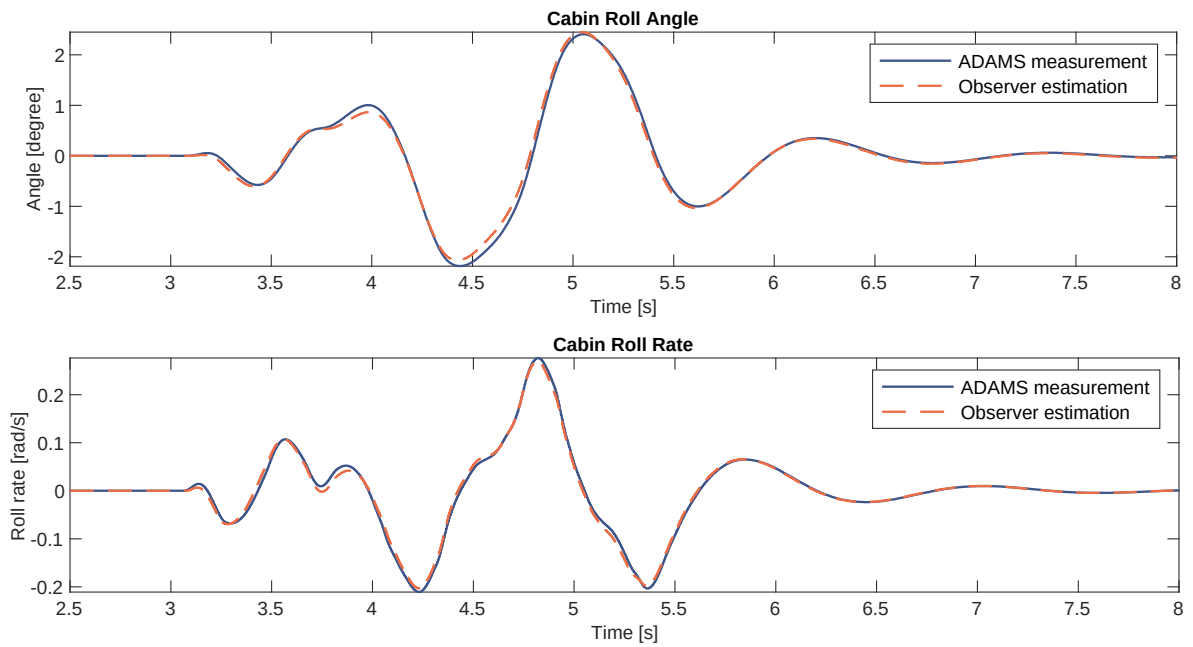


Figure 4.9. Observer estimation of cabin roll motions (offset-bump at 20 km/h)

4.3.2 Cabin: dual bumps at 25 km/h

The dual bumps road profile shown in Figure 4.3 was utilized again in this scenario. The frequency-weighted vertical acceleration, vertical velocity and vertical displacement at the cabin's CG location are shown in Figure 4.10, Figure 4.11, and Figure 4.12, respectively. Compared to the passive cabin suspension, MPC reduced the vertical frequency-weighted accelerations by 19%. Similarly, the integrated Skyhook-LQR also attenuated the vertical vibration by 19%. Besides, the gain-adaptive Skyhook-LQR approaches reduced 20% (A) and 21% (B) of the undesired vibrations in the vertical direction, slightly better than MPC and the non-adaptive Skyhook-LQR. Regarding the cabin's vertical velocity, MPC, integrated Skyhook-LQR, and the gain-adaptive Skyhook-LQR controllers provided similar reductions of around 22%. However, MPC reduced the RMS of the cabin's vertical displacement by 17%, which was better than the integrated and gain-adaptive Skyhook-LQR approaches (9%).

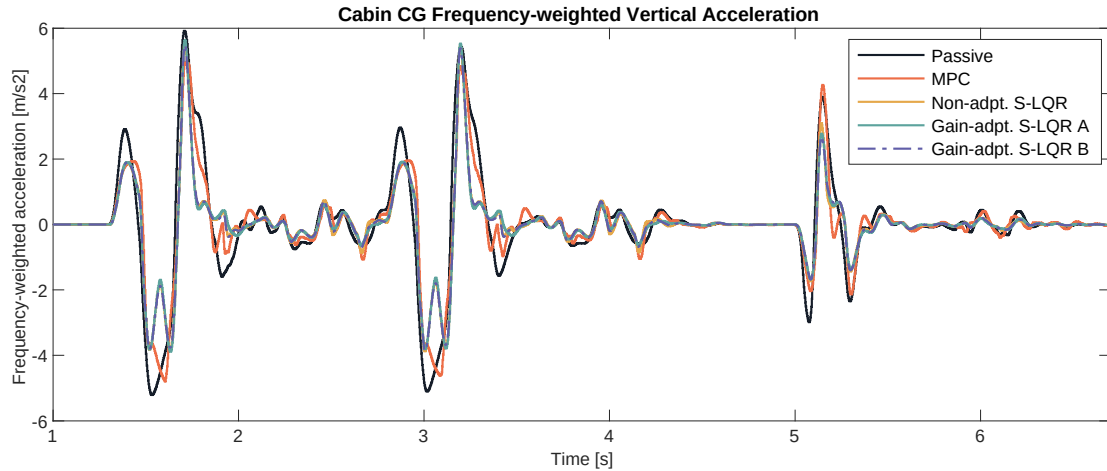


Figure 4.10. Cabin frequency-weighted vertical acceleration (dual bumps at 25 km/h)

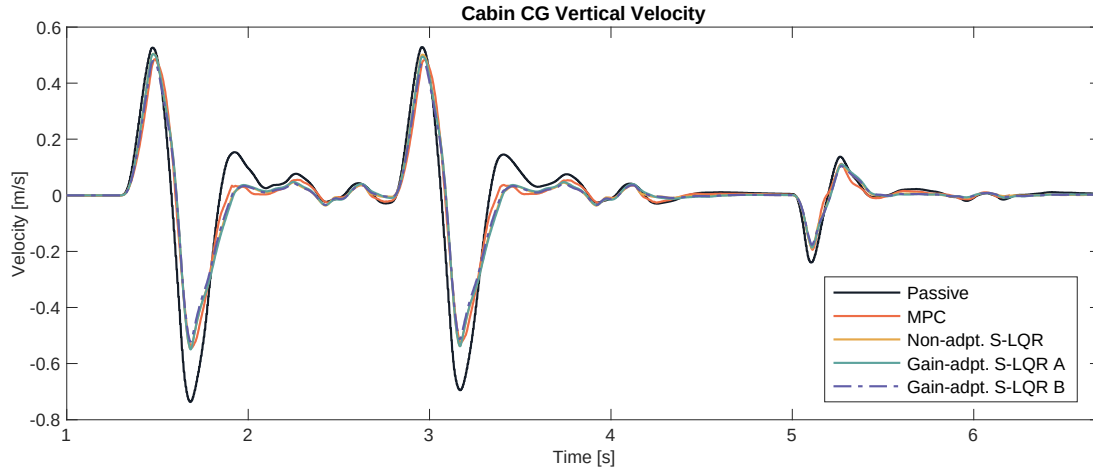


Figure 4.11. Cabin vertical velocity (dual bumps at 25 km/h)

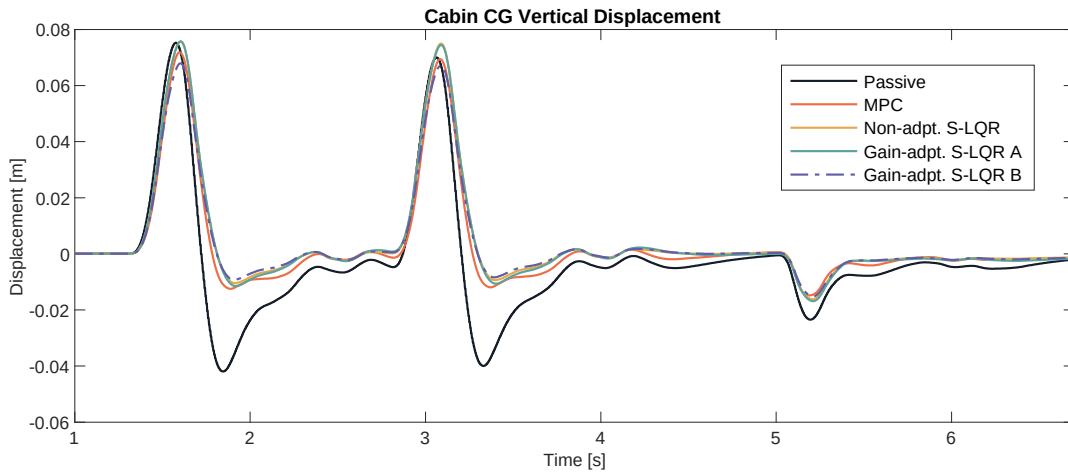


Figure 4.12. Cabin vertical displacement (dual bumps at 25 km/h)

The cabin's frequency-weighted pitch acceleration, pitch rate and pitch angle are presented in Figure 4.13, Figure 4.14, and Figure 4.15, respectively. Compared to the passive suspension, MPC reduced the RMS of the frequency-weighted accelerations in pitch direction by 17%. The gain-adaptive Skyhook-LQR approaches attenuated 25% (A) and 26% (B) of the undesired pitch acceleration, respectively. Besides, the non-adaptive Skyhook-LQR improved the pitch dynamic by 26%, which was better than MPC and the gain-adaptive Skyhook-LQR ones. In terms of the cabin's pitch rate, MPC provided a 17% reduction, which was worse than the non-adaptive (26%) and gain-adaptive (25%) Skyhook-LQR controllers. Moreover, these three controllers gave comparable reductions in the cabin's

pitch angle, i.e., 25% (MPC), 27% (non-adaptive Skyhook-LQR), and 21% (gain-adaptive Skyhook-LQR), respectively.

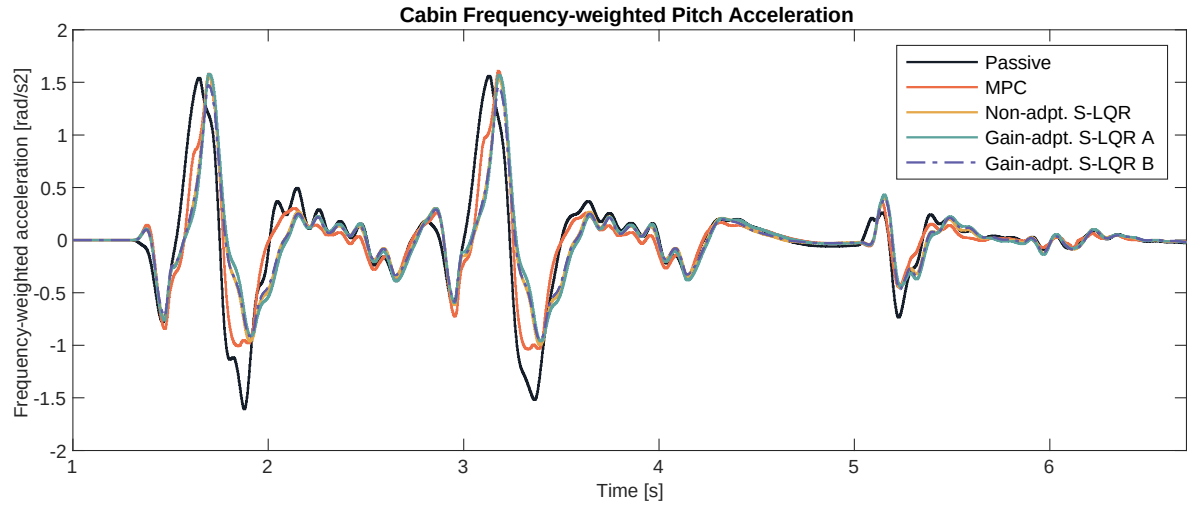


Figure 4.13. Cabin frequency-weighted pitch acceleration (dual bumps at 25 km/h)

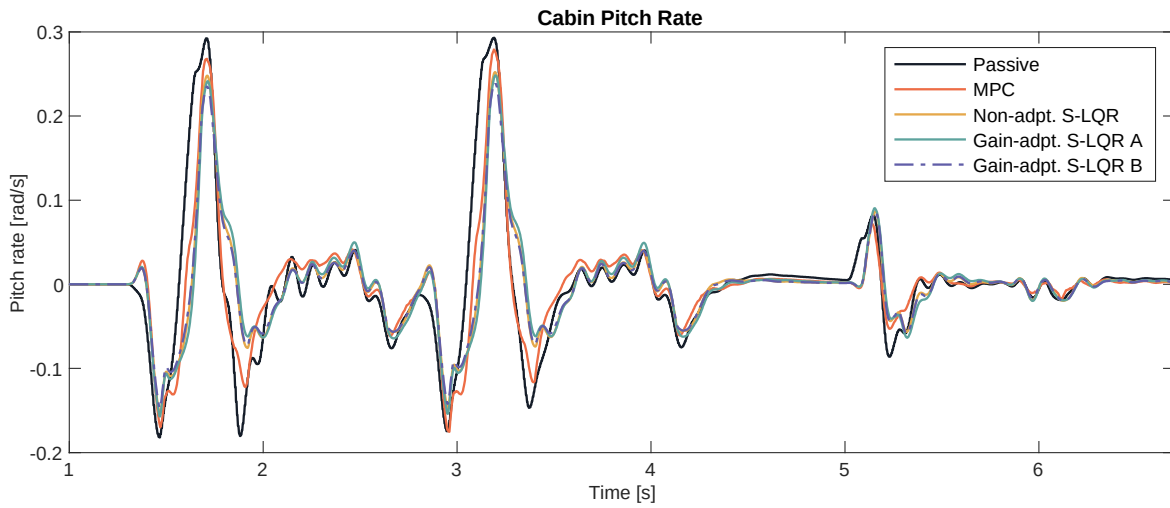


Figure 4.14. Cabin pitch rate (dual bumps at 25 km/h)

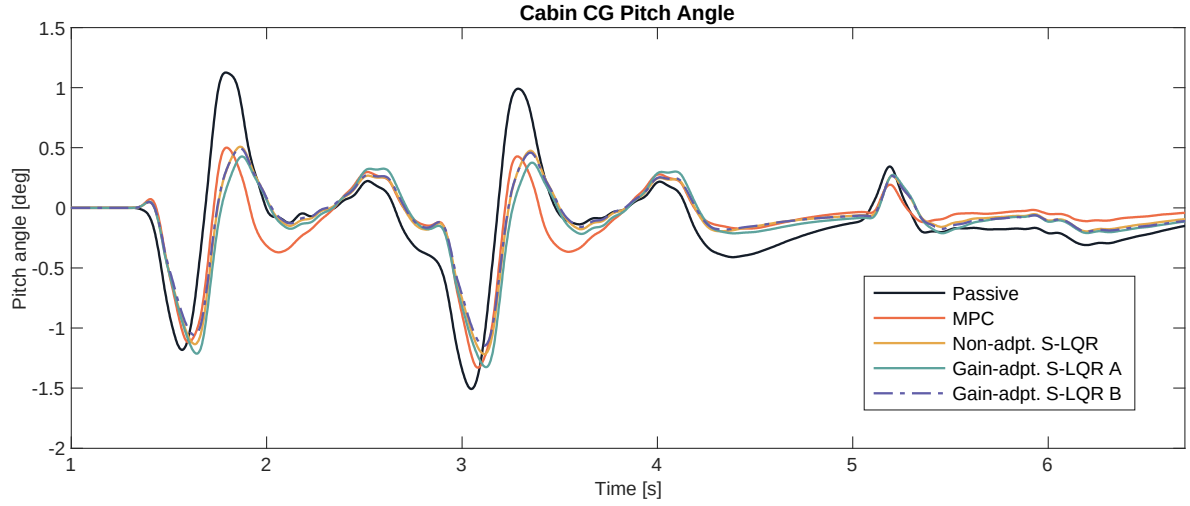


Figure 4.15. Cabin pitch angle (dual bumps at 25 km/h)

Figure 4.16 shows the gain scheduling process in this scenario. The gain of pitch q_5 was adjusted between the pre-defined boundaries from 0 to its maximum value of 1500. Since the disturbances in the roll direction were minimal, the gain of roll q_6 remained at a low level under 60. It is noticeable that the algorithm B switched the gain value more moderately than algorithm A, which was beneficial to the system robustness.

Figure 4.17 compares the front left damping forces generated by the three controllers. It can be seen that the control forces of the integrated Skyhook-LQR and gain-adaptive Skyhook-LQR were very similar in this scenario, while MPC requested higher peak values.

Overall, the ride quality improvements in vertical and pitch directions after implementing the proposed controllers were assessed based on the cabin's frequency-weighted accelerations. Their performances were very similar in the vertical direction, but the gain-adaptive Skyhook-LQR controllers were slightly better than the other two approaches. In the pitch direction, both non-adaptive and gain-adaptive Skyhook-LQR overperformed MPC with nearly 10% increments. Overall, all controllers improved the ride quality to a large extent, and the novel Skyhook-LQR approach performed better than MPC because the vertical vibration was dominant in this scenario. Note that their control performances can be further optimized by tuning the control gains, but how to find the best combination of multiple gains is out of this thesis's scope.

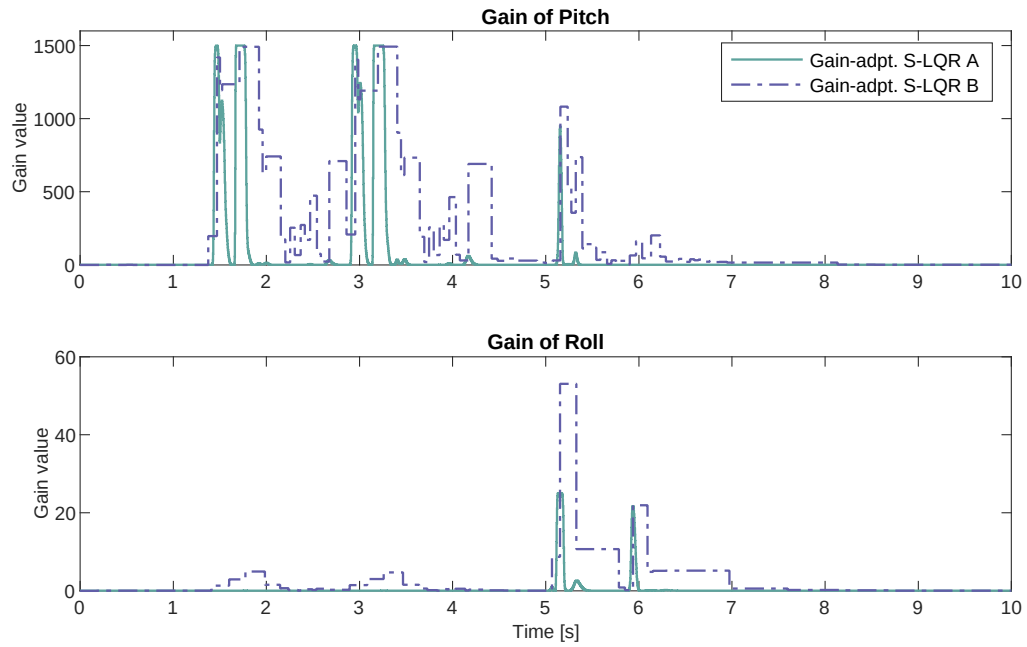


Figure 4.16. Adaptive gain scheduling (dual bumps at 25 km/h)

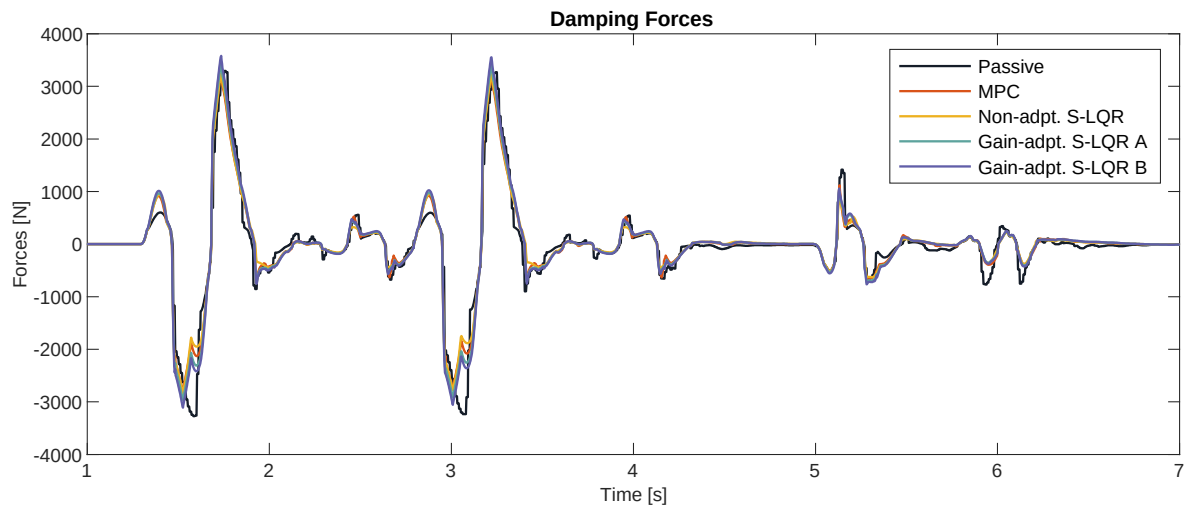


Figure 4.17. Comparison of damping forces (dual bumps at 25 km/h)

4.3.3 Cabin: offset-bump at 20 km/h

The offset-bump road profile is shown in Figure 4.6. It consists of two offset rounded speedbumps with 3 inches in height. In this scenario, the vehicle motions in vertical, pitch, and roll directions were excited, and the truck passed the bumps at 20 km/h.

The frequency-weighted vertical acceleration, vertical velocity and vertical displacement at the cabin's CG location are shown in Figure 4.18, Figure 4.19, and Figure 4.20. Compared to the passive suspension, MPC reduced the RMS of the frequency-weighted accelerations in the vertical direction by 11%. The integrated and gain-adaptive Skyhook-LQR attenuated 16% to 17% of the undesired vibrations in the vertical direction, better than MPC. Regarding the cabin's vertical velocity, MPC, integrated Skyhook-LQR, and the gain-adaptive Skyhook-LQR provided similar reductions of 20% to 22%. However, MPC reduced the RMS of the cabin's vertical displacement by 24%, which was better than the integrated and gain-adaptive Skyhook-LQR (15%).

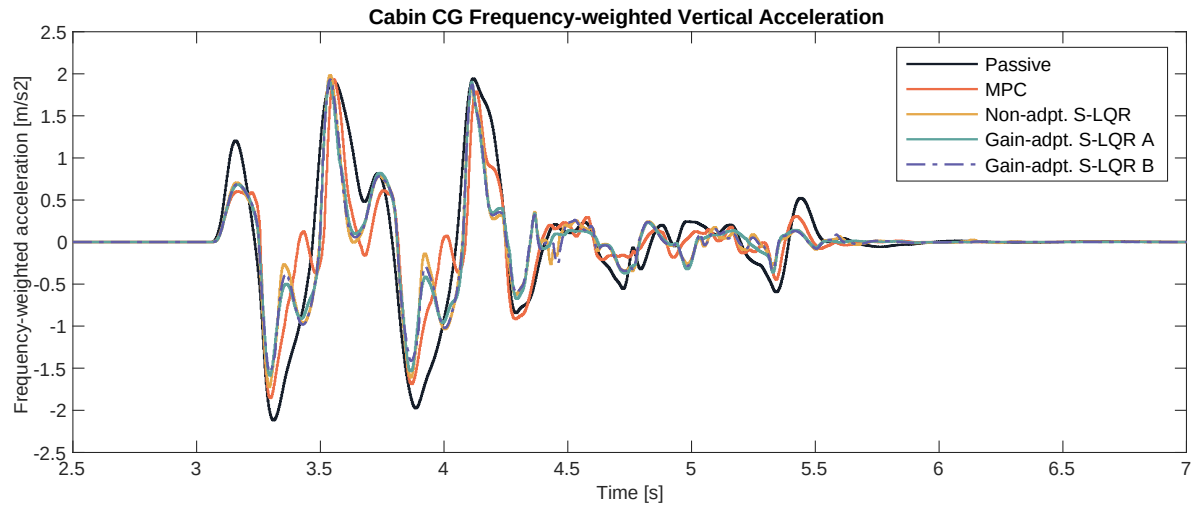


Figure 4.18. Cabin frequency-weighted vertical acceleration (offset-bump at 20 km/h)

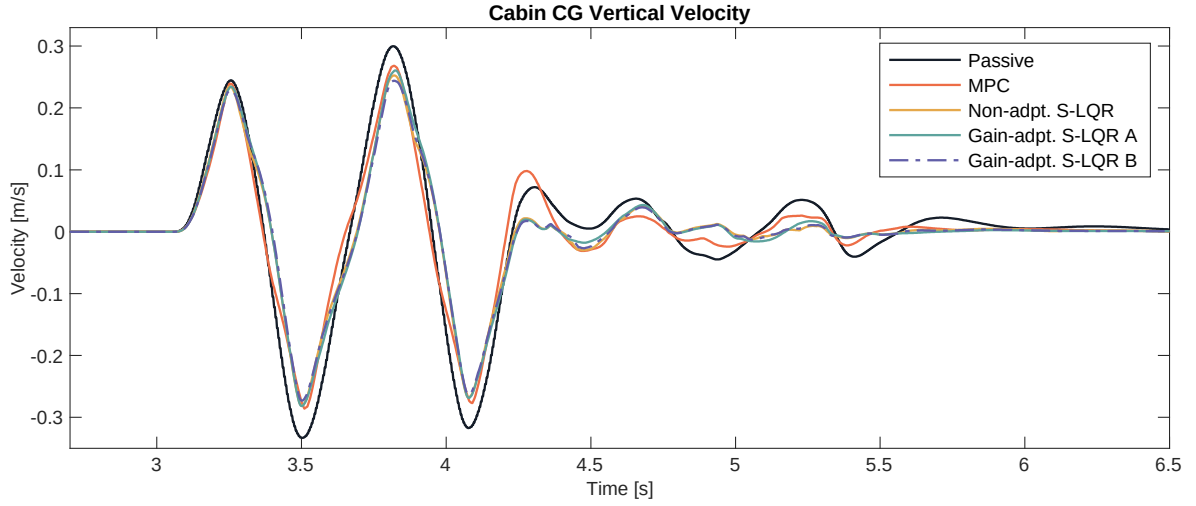


Figure 4.19. Cabin vertical velocity (offset-bump at 20 km/h)

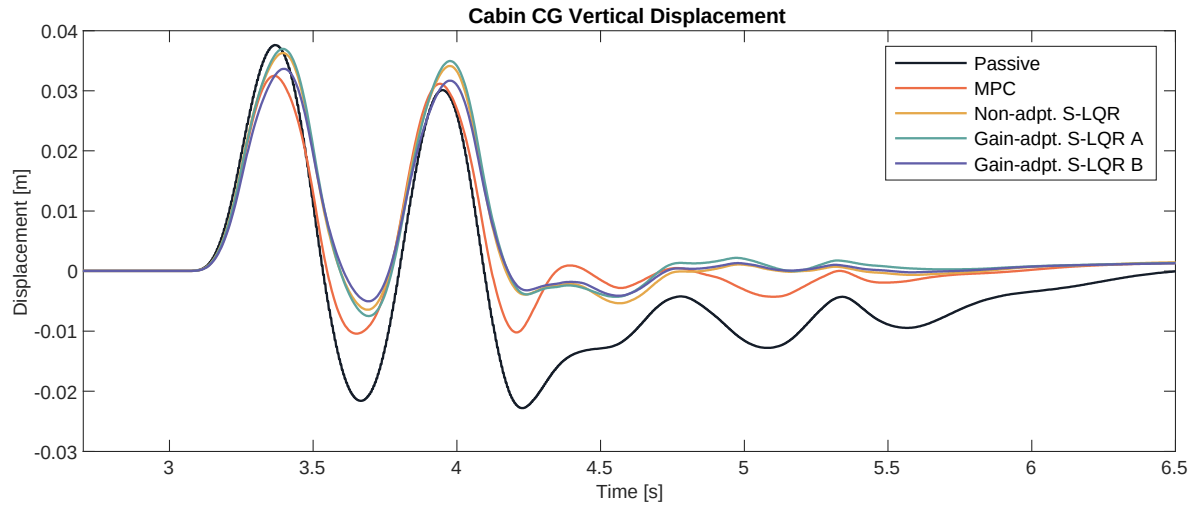


Figure 4.20. Cabin vertical displacement (offset-bump at 20 km/h)

The cabin's frequency-weighted roll acceleration, roll rate, and roll angle are shown in Figure 4.21, Figure 4.22, and Figure 4.23, respectively. MPC reduced the RMS of the frequency-weighted roll acceleration by 41%, which was slightly better than both non-adaptive (20%) and gain-adaptive Skyhook-LQR (A: 21%, B: 24%). In terms of the roll rate and roll angle, MPC provided the best performance with improvements of 47% and 53%, which was significantly better than the integrated Skyhook-LQR (25% and 29%) and the gain-adaptive Skyhook-LQR (24% and 28%).

The cabin's frequency-weighted pitch acceleration and pitch rate are presented in Figure 4.24 and Figure 4.25, respectively. Compared to the passive suspension, MPC reduced the RMS of the frequency-weighted accelerations in pitch direction by 25%, while the non-adaptive and the two gain-adaptive Skyhook-LQR attenuated the undesired motions in pitch direction by 30%, 25% (A), and 28% (B), respectively. Regarding the cabin's pitch rate, MPC provided a 27% reduction, which was slightly worse than the integrated Skyhook-LQR (30%), but better than the gain-adaptive Skyhook-LQR (25%).

Figure 4.26 shows the gain scheduling process in this scenario. Since the roll excitations were significant, the corresponding gain q_6 was adjusted from 0 to the largest pre-defined value 2500. The pitch disturbance was relatively low, so the gain of pitch q_5 did not reach its maximum value. Similar to the previous test, algorithm B switched the gain value more moderately than algorithm A, which was beneficial to the system robustness.

After implementing the proposed controllers, the ride quality improvements in vertical, roll, and pitch directions were assessed based on the cabin's frequency-weighted accelerations. In the vertical and pitch directions, the performances of the three controllers were very similar, but the integrated Skyhook-LQR was slightly better than the MPC approach. In the roll direction, MPC overperformed the integrated and gain-adaptive Skyhook-LQR with nearly 20% increments. Overall, the novel Skyhook-LQR approach performed better than MPC when vertical and/or pitch vibrations were dominant, while MPC improved the roll dynamics to a large extent.

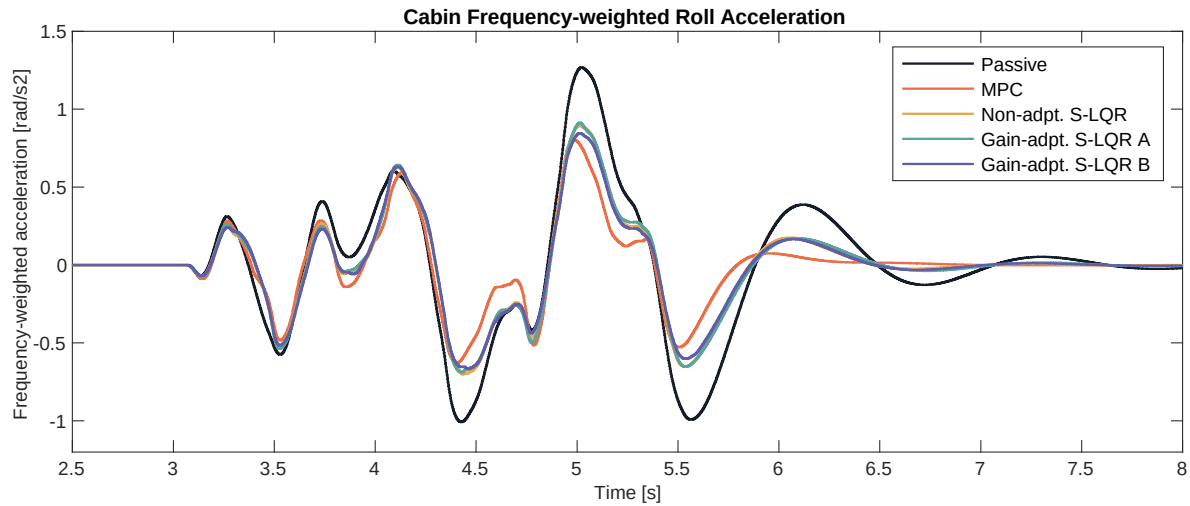


Figure 4.21. Cabin frequency-weighted roll acceleration (offset-bump at 20 km/h)

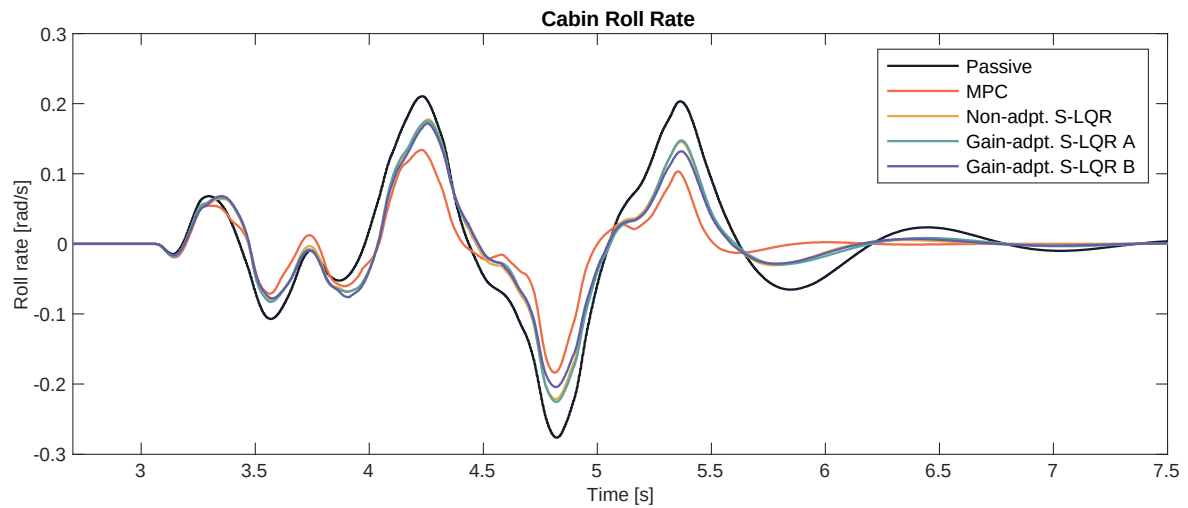


Figure 4.22. Cabin roll rate (offset-bump at 20 km/h)

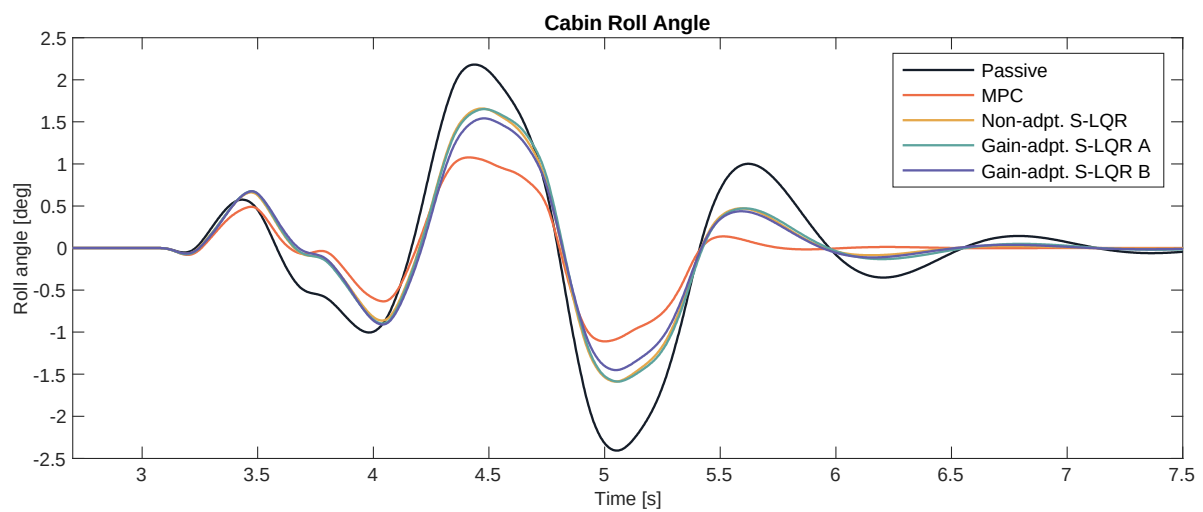


Figure 4.23. Cabin roll angle (offset-bump at 20 km/h)

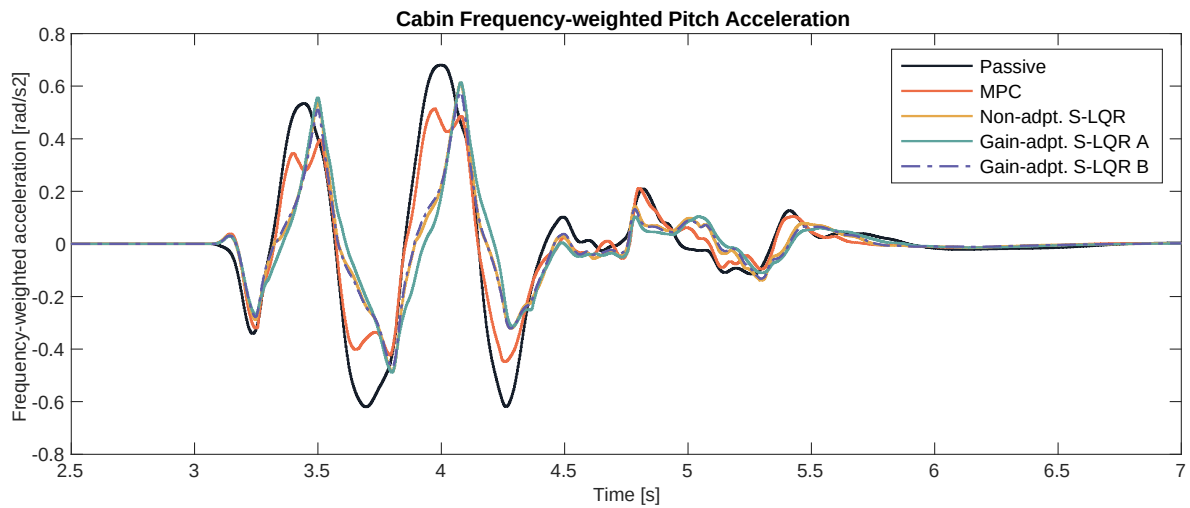


Figure 4.24. Cabin frequency-weighted pitch acceleration (offset-bump at 20 km/h)

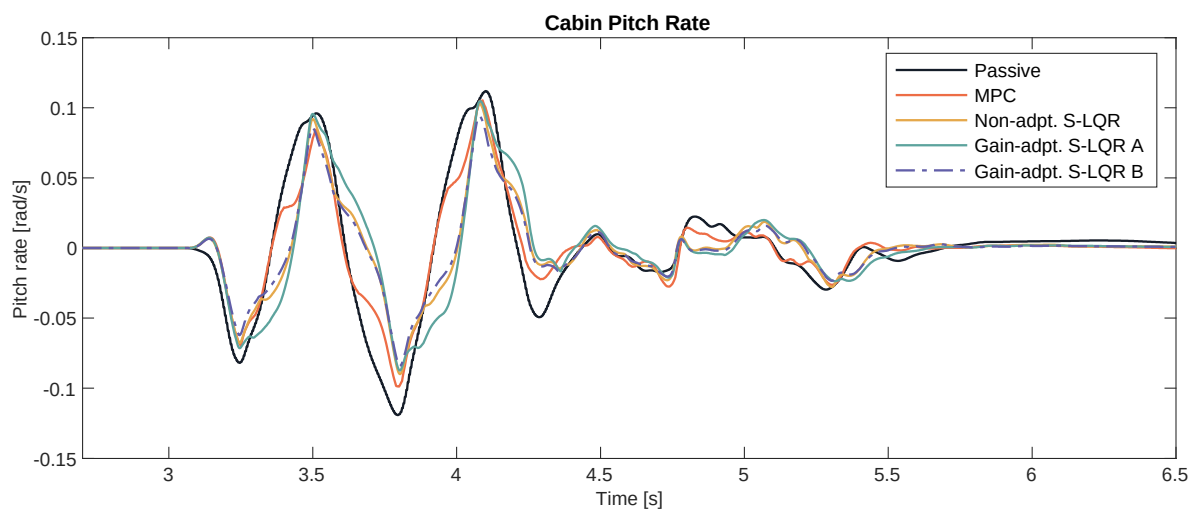


Figure 4.25. Cabin pitch rate (offset-bump at 20 km/h)

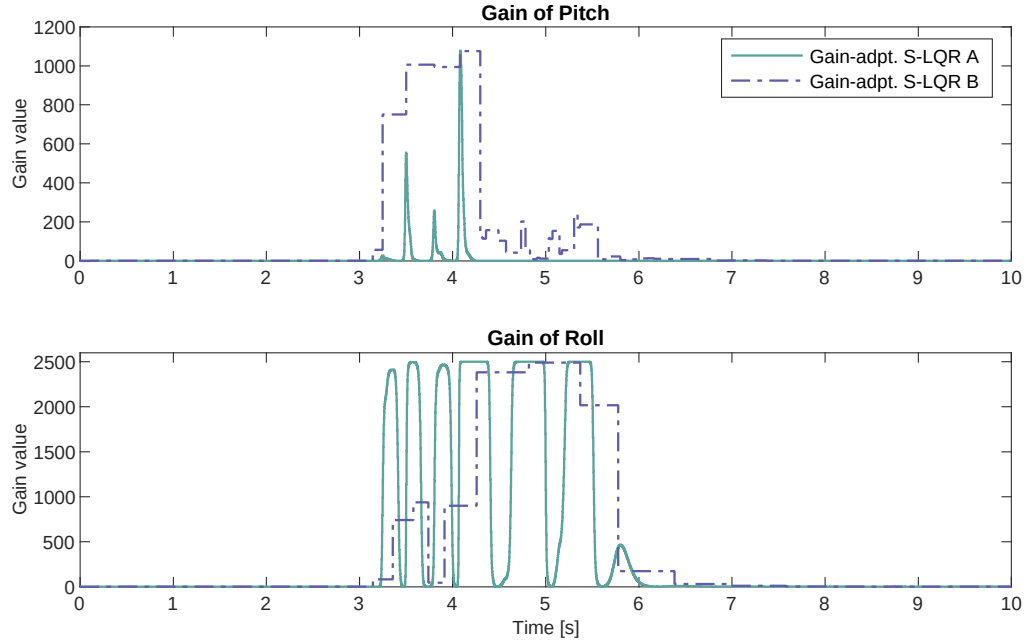


Figure 4.26. Adaptive gain scheduling (offset-bump at 20 km/h)

4.3.4 Cabin: paved road test

The paved road is a surface constructed using asphalt or concrete. The purpose of paving a road is to provide a durable, smooth, and stable surface for vehicles to travel on. Paved roads are used in various environments, from urban to rural areas. They provide many benefits over unpaved roads, including improved safety, reduced road maintenance costs, and increased durability. In this test, the paved road profile contains three kinds of ISO standard road conditions (i.e., class A, class B, and class C) shown in Figure 4.27, and different vehicle speeds (i.e., 90 km/h, 60 km/h, 30 km/h). This is one of the majority driving scenarios of a transportation vehicle.

The performance evaluation factors are listed in Table 4.2. The truck was driven on the standard ISO A road at 90 km/h in the first scenario. The gain-adaptive Skyhook-LQR significantly reduced the RMS of the frequency-weighted vertical acceleration by 23% (A) and 26% (B), which was better than that of the non-adaptive (16%) and MPC (15%). Regarding the overall assessment, the gain-adaptive Skyhook-LQR showed 21% (A) and 25% (B) improvements, while the non-adaptive Skyhook-LQR and MPC performed slightly worse, with 15% and 14%, respectively. The rotational responses were not listed here since the disturbances in roll and pitch directions were not dominant when driving on

paved roads. The same observations can be seen in the second and the third scenarios, which were driving on the ISO B road at 60 km/h and the ISO C road at 30 km/h. In terms of the RMS of the frequency-weighted vertical acceleration, the gain-adaptive Skyhook-LQR A (24% and 28%) presented better performance than the non-adaptive Skyhook-LQR (18% and 21%) and MPC (16% and 15%) as expected. Surprisingly, the gain-adaptive Skyhook-LQR B performed the best, which reduced the frequency-weighted vertical accelerations by 27% and 33%. In terms of the overall assessment, both gain-adaptive Skyhook-LQR maintained their good performance with improvements of 21%-31%, better than those of the non-adaptive Skyhook-LQR (15%-20%) and MPC (12%-14%).

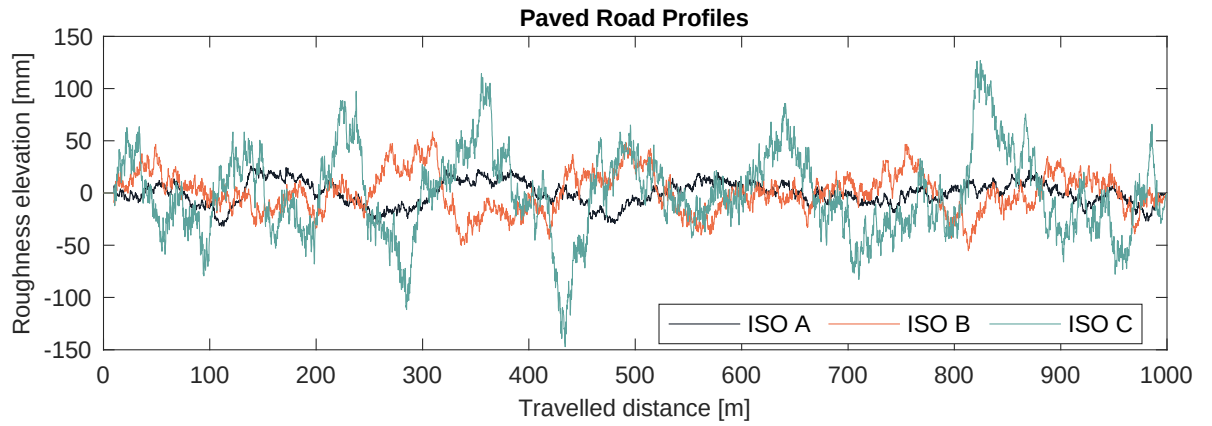


Figure 4.27. Paved road profiles

In addition, a combined paved road shown in Figure 4.28 was used, which contained the above three road classes, and the truck was driven at 30 km/h. In this scenario, the integrated Skyhook-LQR approach showed great protentional when the vertical vibration was dominant, which provided 26% and 22% in the frequency-weighted vertical acceleration and the overall assessment. Besides, the gain-adaptive Skyhook-LQR A performed better, which gives 27% and 24% improvements, while the gain-adaptive Skyhook-LQR B provided the best performance (30% and 28%). However, MPC only gave 14% and 11% reductions since its control gains were tuned for harsh conditions, which was not ideal for paved roads. However, this issue can be solved by implementing the proposed gain-adaptive algorithm introduced in Section 3.2.3 on the MPC controller to intelligently adjust its control gains according to the disturbance level.

As expected, both MPC and Skyhook-LQR controllers could provide excellent performances in minimizing the undesired vibration and shock on rough roads. However, when the disturbances in the

rotational directions were not dominant, such as driving on a paved road, the gain-adaptive one performed better because it paid more attention to the vertical vibrations. Overall, the proposed gain-adaptive Skyhook-LQR approach took advantage of the Skyhook control when the vertical disturbance was dominant and utilized the benefits of the LQR algorithm when all modes of vibrations (pitch, roll, and vertical) were not neglectable.

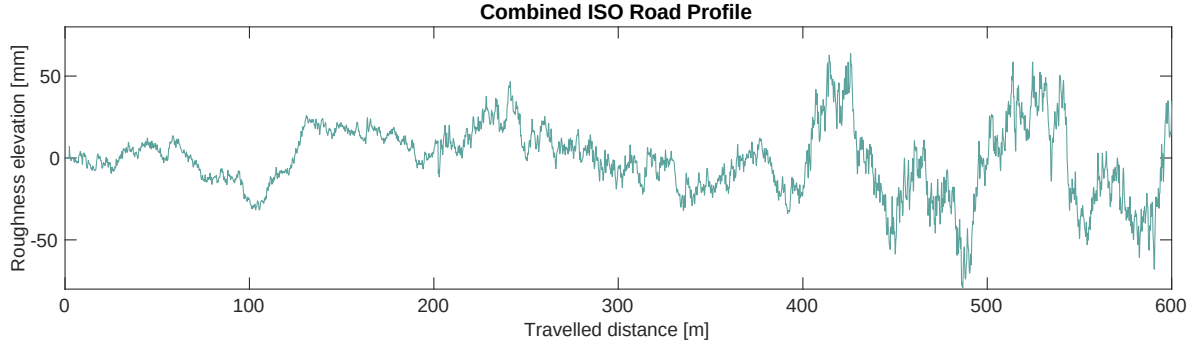


Figure 4.28. Combined ISO road profile

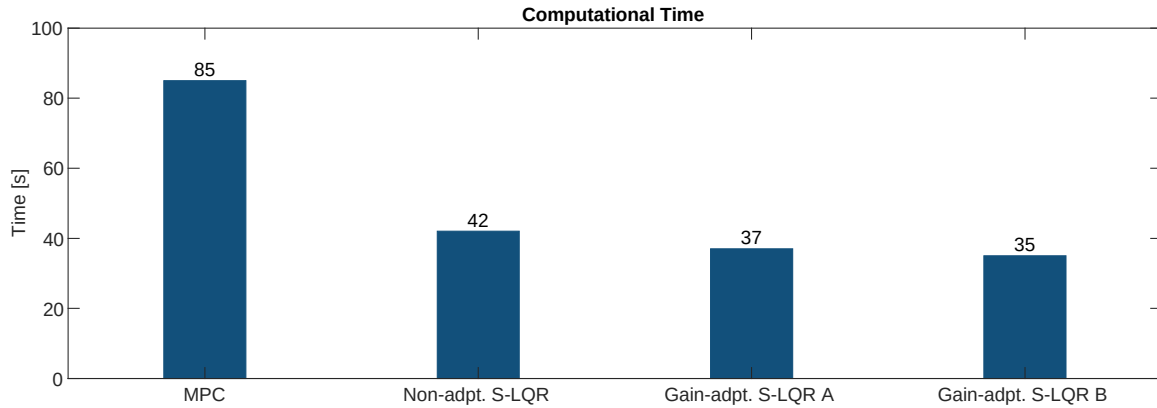


Figure 4.29. Comparison of computational time (combined ISO road)

Moreover, the computational time of MPC, integrated Skyhook-LQR, and gain-adaptive Skyhook-LQR were compared in this scenario. The total simulation time was 45 seconds, and the control algorithm was running in MATLAB/Simulink on a desktop computer with Intel Core i7-8700 CPU @ 3.20GHz and RAM 16GB. As presented in Table 4.2 and Figure 4.29, the MPC took 85s to finish this 40-second simulation task, which was much higher than the integrated Skyhook-LQR (42s) and the two gain-adaptive Skyhook-LQR (37s and 35s). In general, the high computational cost of MPC is mainly

due to its complicated prediction algorithm and the external QP solver, which makes it challenging to be implemented in practical applications.

Table 4.2. Performance evaluation (paved road test)

ISO-A at 90 km/h					
	Passive	MPC	Non-adpt. S-LQR	Gain-adpt. S-LQR A	Gain-adpt. S-LQR B
RMS of Freq.-weighted accel. (m/s ²)	0.50	0.43 (-15%)	0.42 (-16%)	0.39 (-23%)	0.37 (-26%)
Overall assessment	0.56	0.48 (-14%)	0.47 (-15%)	0.44 (-21%)	0.42 (-25%)
ISO-B at 60 km/h					
RMS of Freq.-weighted accel. (m/s ²)	0.88	0.74 (-16%)	0.72 (-18%)	0.67 (-24%)	0.64 (-27%)
Overall assessment	0.94	0.81 (-13%)	0.79 (-16%)	0.73 (-22%)	0.69 (-26%)
ISO-C at 30 km/h					
RMS of Freq.-weighted accel. (m/s ²)	1.48	1.25 (-15%)	1.17 (-21%)	1.09 (-28%)	0.99 (-33%)
Overall assessment	1.60	1.40 (-12%)	1.28 (-20%)	1.20 (-25%)	1.10 (-31%)
Combined ISO road					
RMS of Freq.-weighted accel. (m/s ²)	1.10	0.95 (-14%)	0.82 (-26%)	0.80 (-27%)	0.77 (-30%)
Overall assessment	1.18	1.05 (-11%)	0.92 (-22%)	0.90 (-24%)	0.85 (-28%)
Computational time (sec)	N/A	85	42	37	35

4.4 Experimental Results

The performance of vehicle suspensions is critical to drivers' and passengers' overall comfort and safety. To evaluate the effectiveness of the proposed semi-active suspension systems and their gain-adaptive algorithms, the experimental studies were conducted based on a scaled-down cabin model mounted on a Stewart Platform to simulate a truck chassis motion. The experimental study aimed to assess the effectiveness of the adaptive damping control algorithms in improving ride comfort. The results of these experiments can guide the further development and optimization of adaptive suspension systems that ultimately lead to improved driving experiences and safety.

4.4.1 Experimental setup

Instead of implementing the proposed semi-active suspension system on a real truck, a scaled-down cabin model was first built and used in the experimental studies. Generally, scaled models offer several advantages over full-size models [176], such as low cost, high repeatability, time efficiency, safety, etc. Note that a scaled-down model should have similar dynamic characteristics as the full-size system [177]. Based on the cabin dynamic equations (3.17), (3.18), and (3.19), the non-dimensional analysis was used to investigate the static and dynamic characteristics [178,179]. The scaled ratio used in this study can be found in Appendix C. As shown in Figure 4.30, the scaled-down cabin contains scaled sprung mass (cabin body), tilting mechanism, air springs (Firestone W02-358-7036), MR dampers (Lord RD-8041-1), and lateral dampers.

After fabricating the scaled-down cabin, it was installed on top of a Stewart Platform, as shown in Figure 4.31. The Stewart Platform, a hexapod or six-legged parallel robot, is a parallel mechanism used for simulation and testing in various fields such as aerospace, robotics, and automotive engineering. It was first analyzed by Stewart in 1965 [180], who wanted to create a simple platform with six DOFs that applies to a variety of fields. In general, a Stewart Platform consists of a fixed base, a mobile platform, and six variable legs, so the Platform has six DOFs with respect to the Base. The legs act as actuators to move the upper platform, allowing it to simulate various motions in a controlled environment. It is designed to provide high accuracy, repeatability, and stability, making it an ideal tool for experimentation in fields requiring precise and controlled movement. Initially, it was designed to simulate flight conditions and earthquakes or waves. Nowadays, it is commonly used for testing the performance of sensors, electronic systems, and other components under simulated dynamic loads and conditions. In the automotive industry, it is used to test the durability and reliability of vehicle

components such as suspension systems, engines, and transmissions. Overall, the Stewart Platform is a versatile and valuable tool for testing and evaluation, providing researchers and engineers with an accurate and controlled environment for conducting experiments and evaluating the performance of various systems.

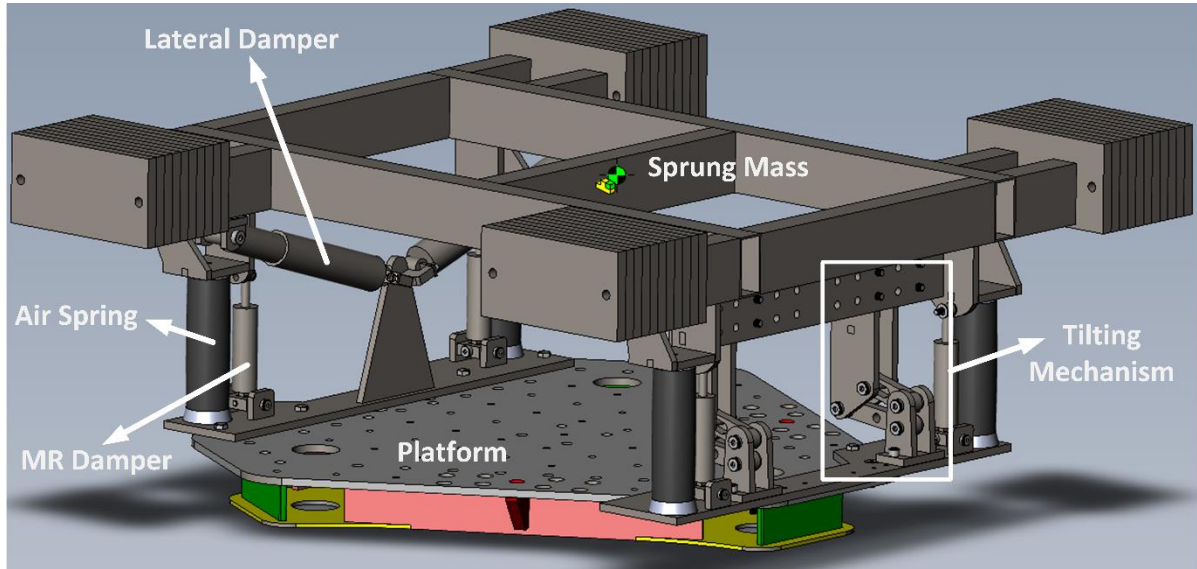


Figure 4.30. CAD drawing of the scaled-down cabin model

The control of a Stewart Platform can be achieved through inverse kinematics [181][182][183], which is a method of determining the required actuator positions to achieve a desired end-effector position. The inverse kinematics solution calculates the needed actuator angles or lengths to move the Platform to a desired position and orientation. The Stewart Platform can be accurately and precisely controlled using inverse kinematics. The basic process can be concluded as follows:

- 1) Model the Stewart Platform: A mathematical model of the Stewart Platform needs to be developed, which describes geometry, kinematics, and dynamics.
- 2) Define the desired pose: The desired end-effector position and orientation of the Platform are defined in terms of the x , y , and z coordinates and the roll, pitch, and yaw angles.
- 3) Solve for actuator positions: Using the mathematical model of the Stewart Platform, the inverse kinematics solution can be obtained to find the corresponding actuator displacements to achieve the desired position and orientation of the Platform.

Implement the actuator control algorithm: In this experiment, the Stewart Platform's control algorithm was processed on the Beckhoff system, by which the actuators were moved to the desired positions. The feedback from the displacement sensors was used to continuously monitor and adjust the actuator positions to ensure the Platform remains in the desired pose.

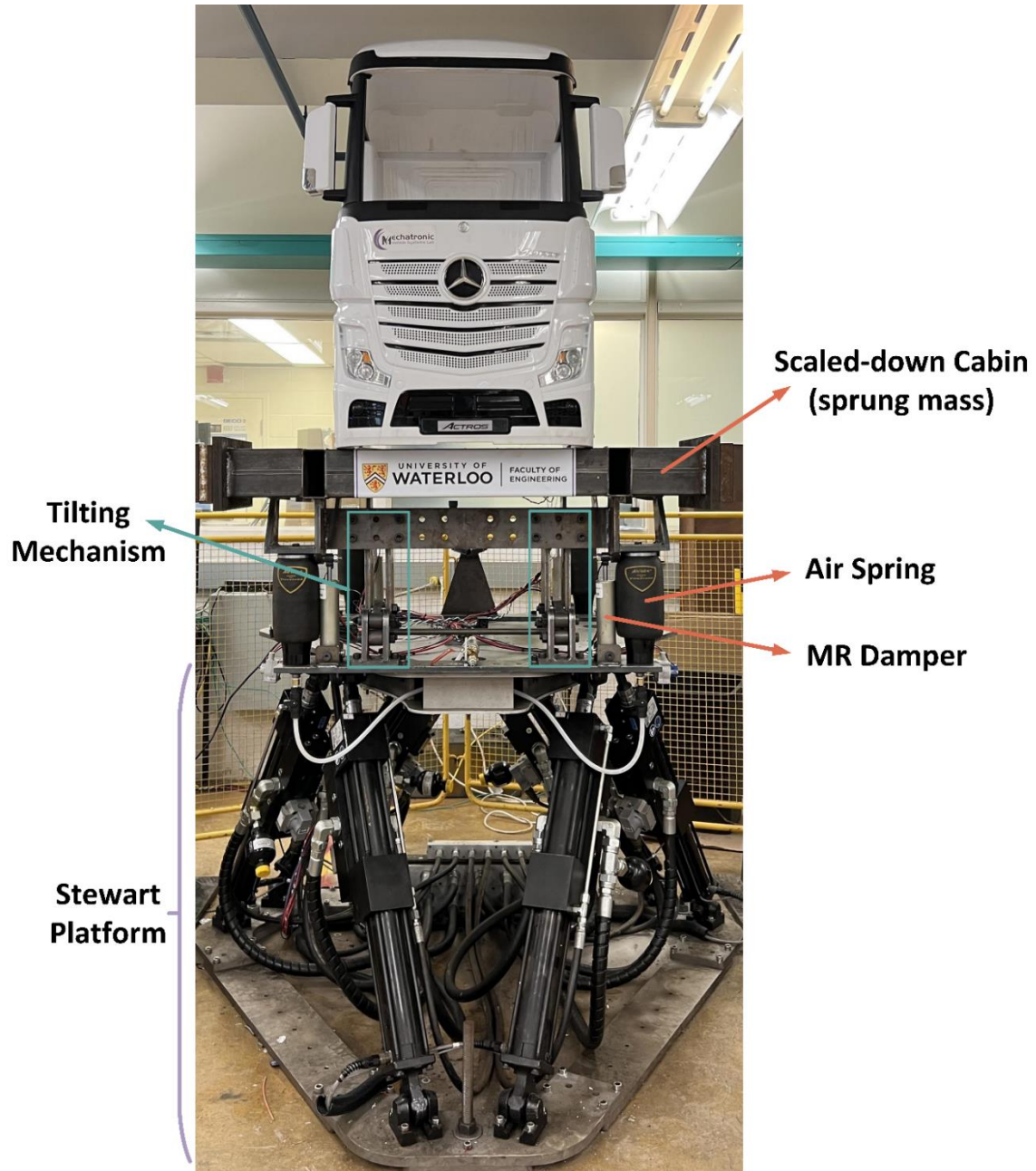


Figure 4.31. Test rig for the adaptive suspension systems

In detail, the Base is considered as the reference plate with orthogonal axes x , y , and z . The Platform has its orthogonal coordinates, x' , y' , and z' . The origin of the Platform coordinates can be defined by three translational displacements and three angular displacements with respect to the Base. A set of Euler angles are used in the following sequence:

- 1) Rotate an angle ψ (yaw) around the z -axis
- 2) Rotate an angle θ (pitch) around the y -axis
- 3) Rotate an angle φ (roll) around the x -axis

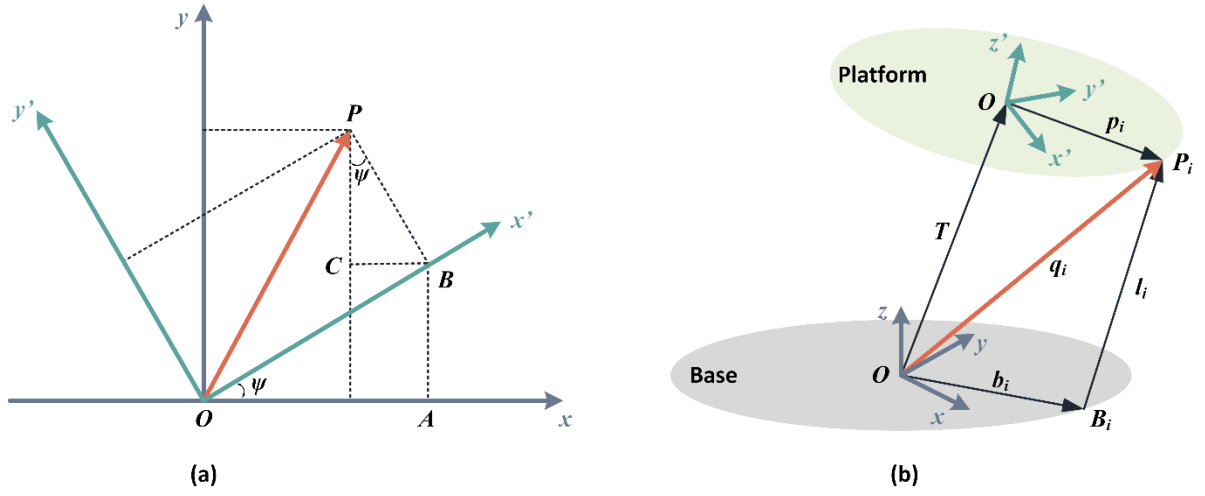


Figure 4.32. Schematic of the Stewart Platform coordinates

When having pure yaw rotation ψ around the z -axis, as shown in Figure 4.32(a), the following coordinates can be obtained:

$$P = i'x' + j'y' + k'z' = ix + jy + kz \quad (4.4)$$

$$x = \overrightarrow{OA} - \overrightarrow{BC} = x' \cos \psi - y' \sin \psi \quad (4.5)$$

$$y = \overrightarrow{AB} + \overrightarrow{PC} = x' \sin \psi + y' \cos \psi \quad (4.6)$$

$$z = z' \quad (4.7)$$

Hence, the yaw rotational matrix $R_z(\psi)$ can be written as

$$\begin{bmatrix} x \\ y \\ z \end{bmatrix} = R_z(\psi) \begin{bmatrix} x' \\ y' \\ z' \end{bmatrix} \quad (4.8)$$

in which

$$R_z(\psi) = \begin{bmatrix} \cos \psi & -\sin \psi & 0 \\ \sin \psi & \cos \psi & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad (4.9)$$

Similarly, the pitch rotational matrix $R_y(\theta)$ can be written as

$$R_y(\theta) = \begin{bmatrix} \cos \theta & 0 & \sin \theta \\ 0 & 1 & 0 \\ -\sin \theta & 0 & \cos \theta \end{bmatrix} \quad (4.10)$$

The roll rotational matrix $R_x(\varphi)$ can be written as

$$R_x(\varphi) = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \varphi & -\sin \varphi \\ 0 & \sin \varphi & \cos \varphi \end{bmatrix} \quad (4.11)$$

Thus, the full rotation matrix of the upper Platform with respect to the Base is

$$\begin{aligned} R_P &= R_z(\psi)R_y(\theta)R_x(\varphi) \\ &= \begin{bmatrix} \cos \psi \cos \theta & -\sin \psi \cos \varphi + \cos \psi \sin \theta \sin \varphi & \sin \psi \sin \varphi + \cos \psi \sin \theta \cos \varphi \\ \sin \psi \cos \theta & \cos \psi \cos \varphi + \sin \psi \sin \theta \sin \varphi & -\cos \psi \sin \varphi + \sin \psi \sin \theta \cos \varphi \\ -\sin \theta & \cos \theta \sin \psi & \cos \theta \cos \varphi \end{bmatrix} \end{aligned} \quad (4.12)$$

Considering the i^{th} leg of the Stewart Platform shown in Figure 4.32(b), the coordinates q_i of the anchor point P_i with respect to the Base coordinate can be written as

$$q_i = \vec{T} + R_P \vec{p}_i \quad (4.13)$$

in which $\vec{T}$ is the translation vector which represents the positional linear displacement of the Platform's origin with respect to the Base coordinate, and $\vec{p}_i$ is the vector that defines the location of the anchor point P_i with respect to the Platform coordinate.

Similarly, the length of the i^{th} leg can be obtained by

$$l_i = \vec{T} + R_P \vec{p}_i - b_i$$

where $\vec{b}_i$ is the vector that defines the location of the lower anchor point B_i .

For the Stewart Platform with linear actuators used in this experiment, the length l_i is defined as the Euclidean L_2 Norm that can be used as the input for the Platform orientation controller.

4.4.2 Observer evaluation and model validation

The cabin observer introduced in Section 3.3 was examined through experimental studies, as well as the model validation of the COE mathematical model. In this study, the dynamic responses of the mathematical model were compared to the measured movements of the scaled-down cabin prototype under very similar conditions to assess its accuracy.

As for the benchmark, the passive cabin suspension was well-tuned to achieve satisfactory ride quality. The cabin vertical acceleration measurements with five different damping levels are presented in Figure 4.33, in which number 1 represents the lowest damping, and number 5 represents the highest level. As can be seen in the figure, the cabin vertical acceleration dropped with the increase of the suspension damping. However, levels 4 and 5 showed similar responses because the suspension system reached the over-damped range. After trying multiple damping settings, the bold red curve (damping level 4) was chosen as the baseline in this experiment.

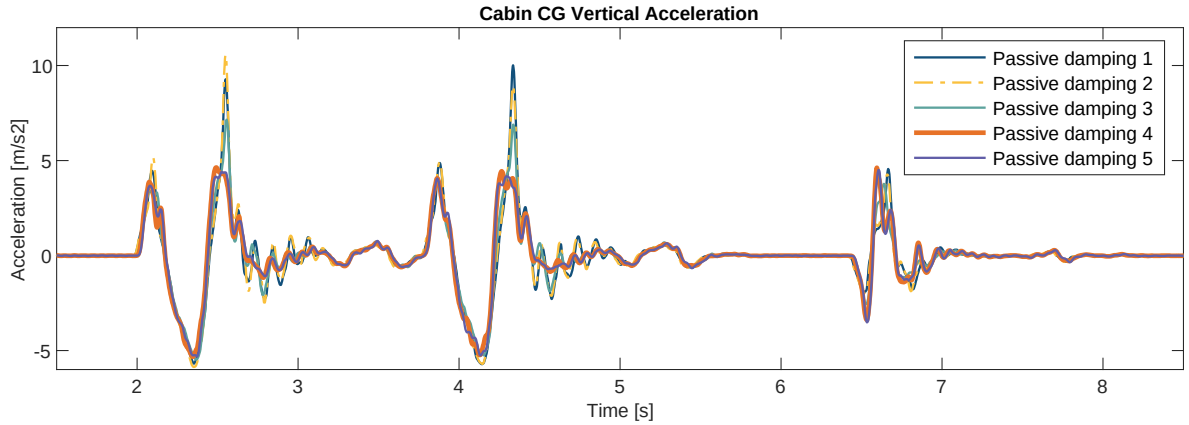


Figure 4.33. Passive damper tuning

To examine if the observer could accurately estimate the unmeasurable cabin states in real-time, the Stewart Platform was actuated in three directions to simulate the body motions when driving over a pair of offset-bump at 25 km/h. Note that the ‘measured’ vertical velocity, vertical displacement, roll

angle, and pitch angle used in the following discussion were obtained based on the known Stewart Platform's motions and the measured suspension relative displacements.

The vertical velocity and displacement at the cabin CG location are shown in Figure 4.34 and Figure 4.35, respectively. The navy curve shows the measured responses, and the orange curve presents the estimated values by the proposed observer. The measurements and the estimated results showed a perfect level of similarity in the vertical direction, although there were few acceptable differences at peak values. Because it was assumed that the cabin's longitudinal and lateral motions with respect to the chassis frame were very small due to the mechanical constraints, and these were neglected in the dynamic equations. Meanwhile, the relative yaw angle was assumed to be zero. However, using the bushings in the suspension planar linkage mechanism resulted in system nonlinearity and realized a certain amount of relative movements between the cabin and the Stewart Platform. Hence, the assumptions, linearization of the dynamic equations, and some uncertain parameters of the test rig caused acceptable estimation errors.

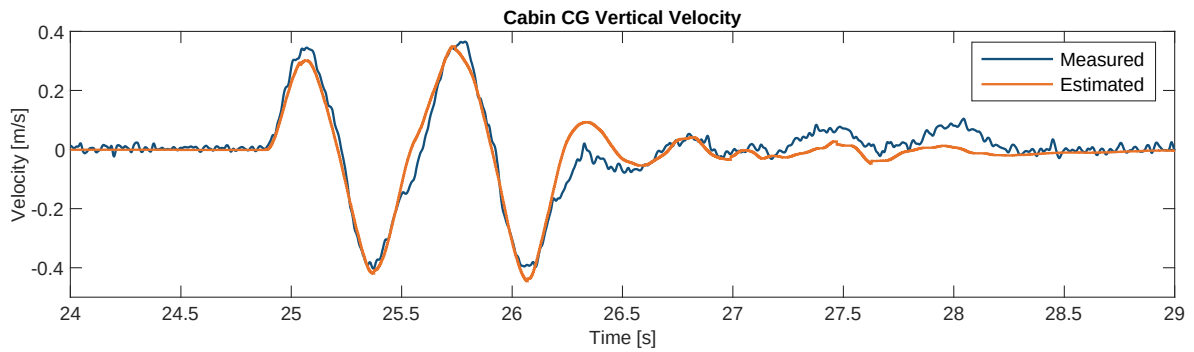


Figure 4.34. Observer evaluation: body vertical velocity

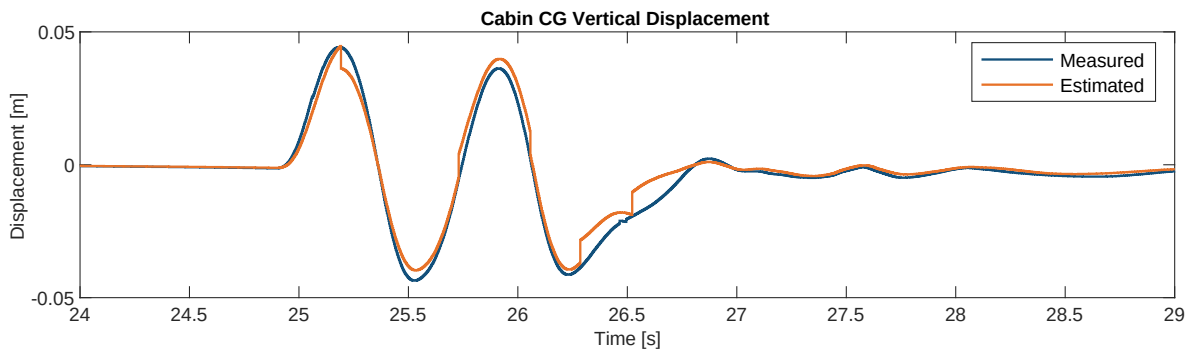


Figure 4.35. Observer evaluation: body vertical displacement

The cabin's roll and pitch angles are presented in Figure 4.36 and Figure 4.37, respectively. As expected, the estimated results provided satisfactory accuracy in both directions. Compared to the estimations in the vertical and roll directions, the geometry and nonlinearity of the linkage mechanism were affected more in the pitch direction. However, the proposed observer captured all the necessary pitch motions that met the dynamic analysis and control system development requirements.

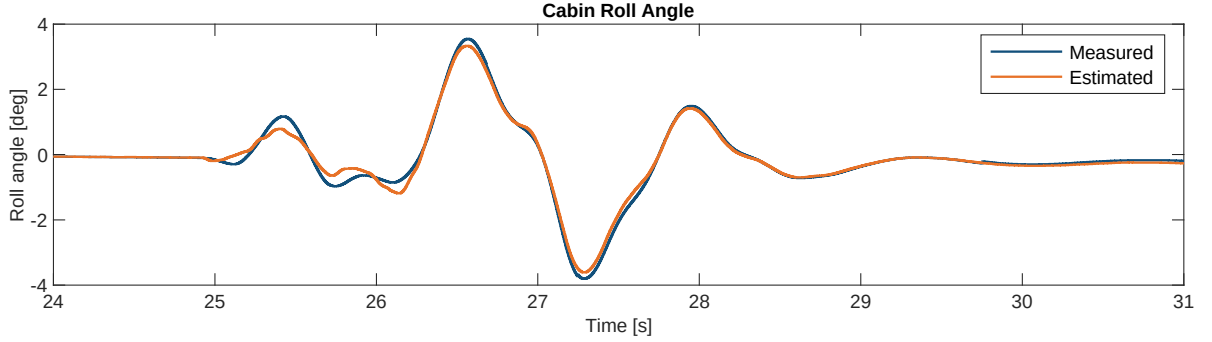


Figure 4.36. Observer evaluation: body roll angle

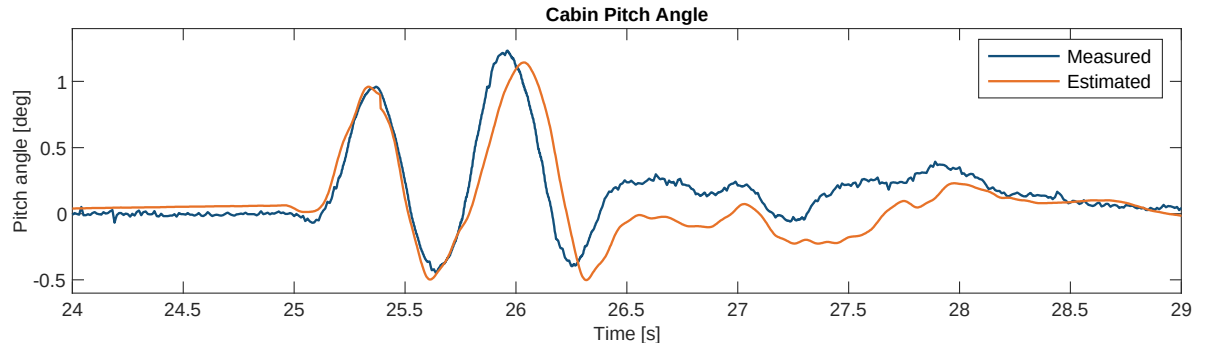


Figure 4.37. Observer evaluation: body pitch angle

4.4.3 Performance evaluation

This section aims to assess the performance of the proposed adaptive damping control algorithms. The experimental results regarding ride quality, stability, computational cost, and robustness are presented and discussed.

4.4.3.1 Paved road

In this scenario, the cabin responses were examined when the truck was driven on a rough paved road. The vertical motion of the Stewart Platform is shown in Figure 4.38, representing a majority driving

scenario. The RMS of the frequency-weighted vertical acceleration and the overall comfort assessment were listed and compared in Table 4.3. Compared to the passive suspension, the Skyhook minimized the frequency-weighted vertical acceleration by 22%, better than LQR (17%). The non-adaptive Skyhook-LQR provided a comparable performance with a 20% reduction. As expected, both gain-adaptive Skyhook-LQR approaches significantly improved the ride quality by 23% (A) and 27% (B), respectively. In terms of the overall assessment, the two gain-adaptive Skyhook-LQR controllers also provided the best performance (23% and 27%) compared to the non-adaptive Skyhook-LQR (19%), LQR (17%), Skyhook (21%), and passive modes.

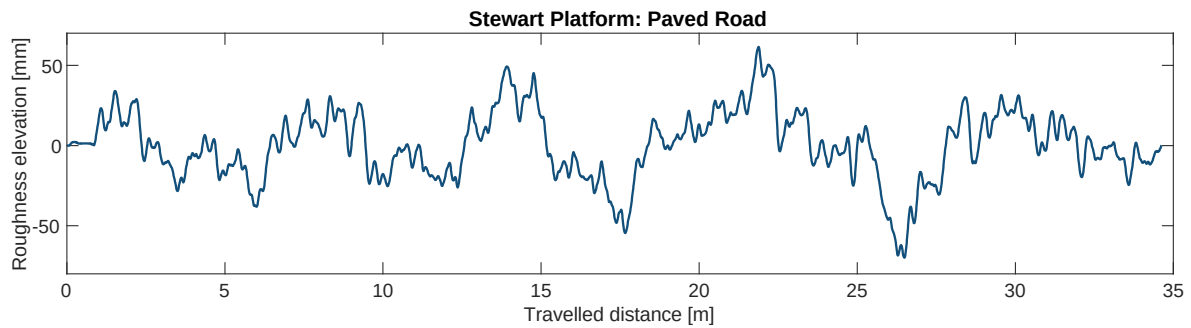


Figure 4.38. Stewart Platform movement: paved road

Overall, both gain-adaptive algorithms improved the vibration attenuation performance to a large extent, especially in the vertical direction, which coincided with the intention introduced in Section 3.2.3. When the rotational motions were slight, the gain-adaptive Skyhook-LQR controller focused more on minimizing the vertical disturbance. This resulted in a better control performance and reduced the computational cost.

Table 4.3. Experimental study: paved road test

600-meter paved road at 60 km/h						
	Passive	Skyhook	LQR	Non-adpt. S-LQR	Gain-adpt. S-LQR A	Gain-adpt. S-LQR B
RMS of Freq.-weighted accel. (m/s^2)	0.83	0.65 (-22%)	0.69 (-17%)	0.66 (-20%)	0.63 (-23%)	0.61 (-27%)

Overall assessment	0.83	0.66 (-21%)	0.69 (-17%)	0.67 (-19%)	0.63 (-23%)	0.61 (-27%)
--------------------	------	----------------	----------------	----------------	----------------	----------------

4.4.3.2 Pure roll motion

In this scenario, the Stewart Platform was moved only in the roll direction. The cabin's roll rate was measured by the 6-axis IMU, and its roll angle and acceleration were calculated accordingly. In Figure 4.39, Figure 4.41 and Figure 4.43, the cabin's roll responses of passive, Skyhook, LQR, and non-adaptive Skyhook-LQR were compared. In terms of the roll acceleration, Skyhook and LQR showed similar improvements within a range of 24% to 26% compared to the passive mode, while the integrated Skyhook-LQR provided the best performance with a 36% reduction. Moreover, Skyhook reduced the scaled-down cabin's roll rate by 4% compared to the passive mode, worse than LQR (14%) and non-adaptive Skyhook-LQR (19%). Regarding the roll angle, LQR decreased it by 15%, slightly better than Skyhook-LQR (13%). However, Skyhook made the roll angle even worse with an increment of 6%.

In Figure 4.40, Figure 4.42 and Figure 4.44, the cabin's roll responses of non-adaptive Skyhook-LQR, gain-adaptive Skyhook-LQR with algorithm A, and gain-adaptive Skyhook-LQR with algorithm B were compared. By implementing the two gain-adaptive algorithms, the cabin's roll acceleration was reduced by 33% (A) and 38% (B), respectively. The cabin's roll rate was reduced by 8% with gain-adaptive algorithm A, worse than algorithm B (15%) and the non-adaptive one (19%). Regarding the roll angle, algorithm A dropped it by 8%, while algorithm B reduced it by 14%.

Figure 4.45 shows the gain scheduling process in this scenario. Since the roll excitations were dominant, the corresponding gain q_6 was adjusted from 0 to the maximum pre-defined value of 700. It is noticeable that algorithm B switched the gain value more moderately than algorithm A, which was beneficial to the system robustness. Overall, the gain-adaptive algorithm further improved the performance of the integrated Skyhook-LQR controller to some extent, and algorithm B performed better than algorithm A. The percentages of improvements by each control approach are listed and compared in Table 4.4.

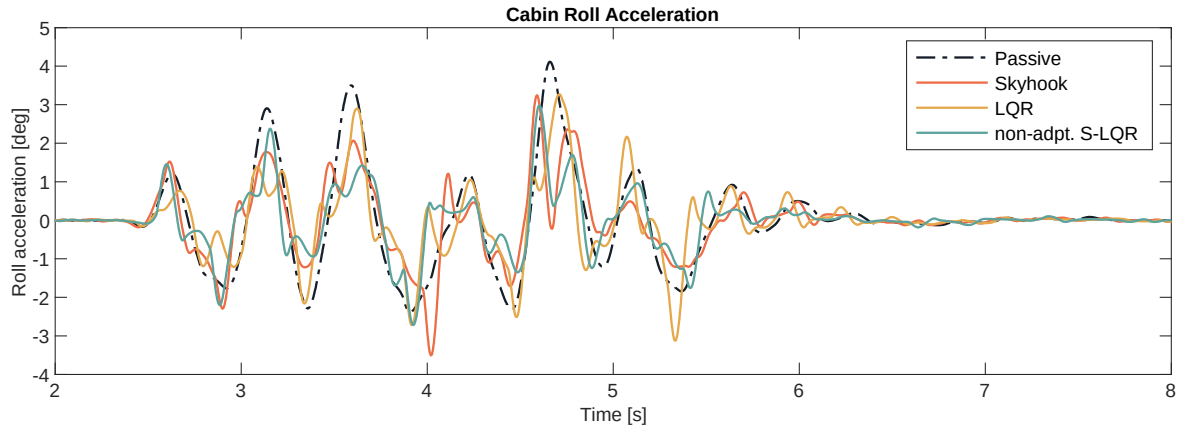


Figure 4.39. Scaled-down cabin roll acceleration (pure roll motion)

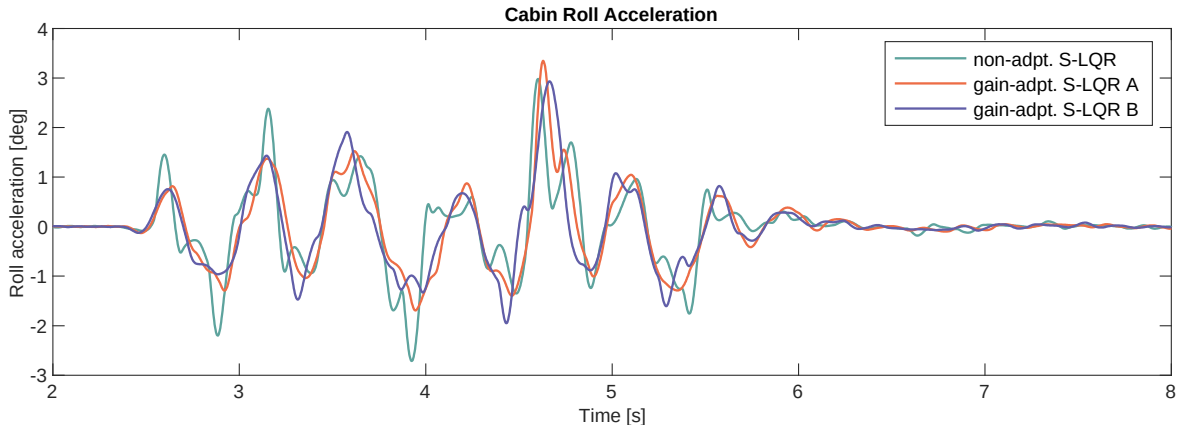


Figure 4.40. Scaled-down cabin roll acceleration (pure roll motion)

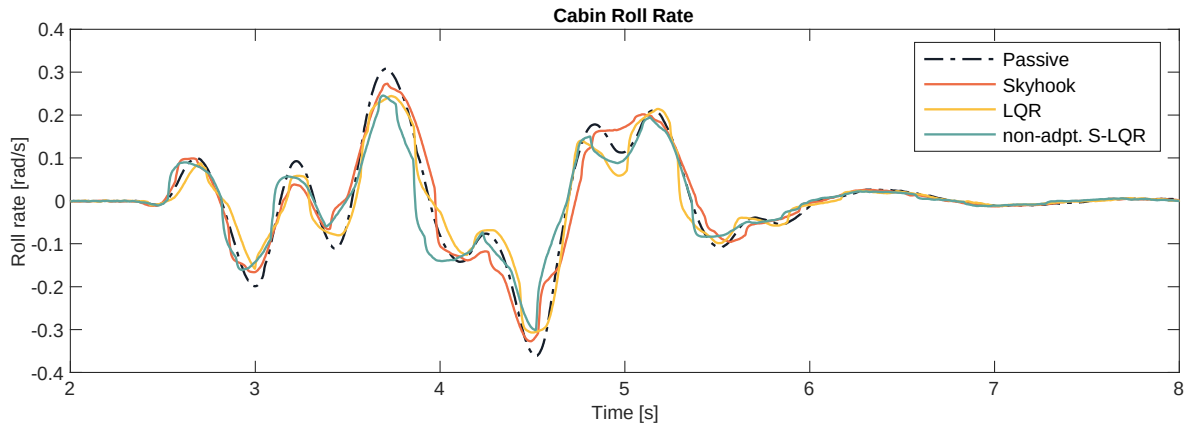


Figure 4.41. Scaled-down cabin roll rate (pure roll motion)

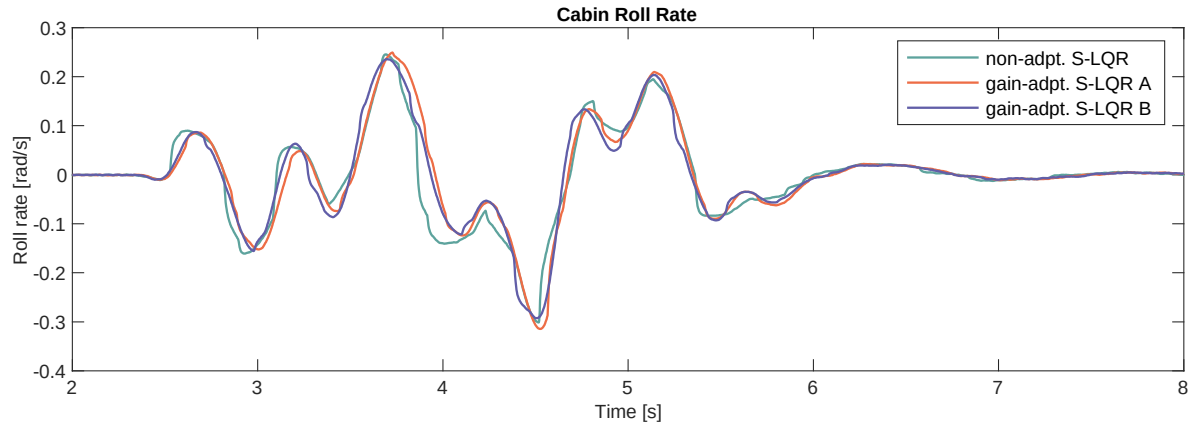


Figure 4.42. Scaled-down cabin roll rate (pure roll motion)

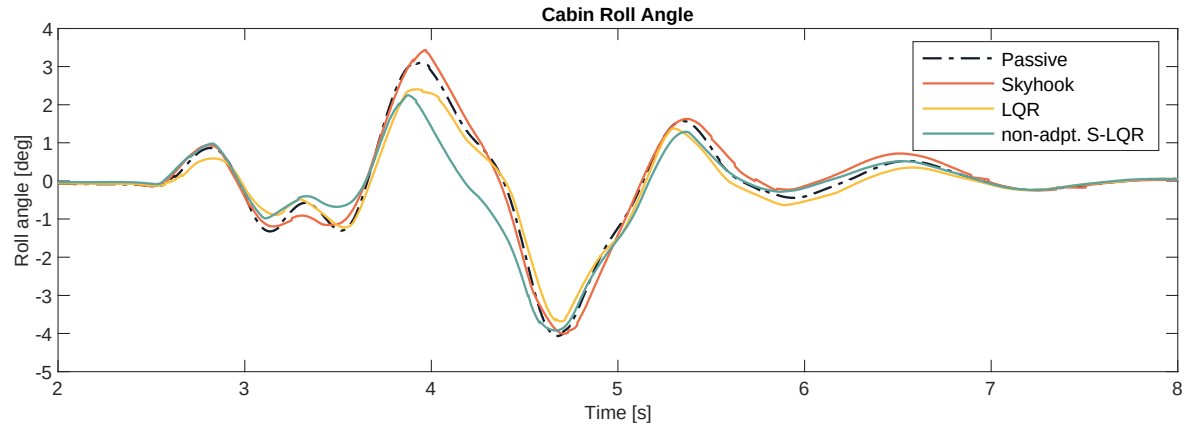


Figure 4.43. Scaled-down cabin roll angle (pure roll motion)

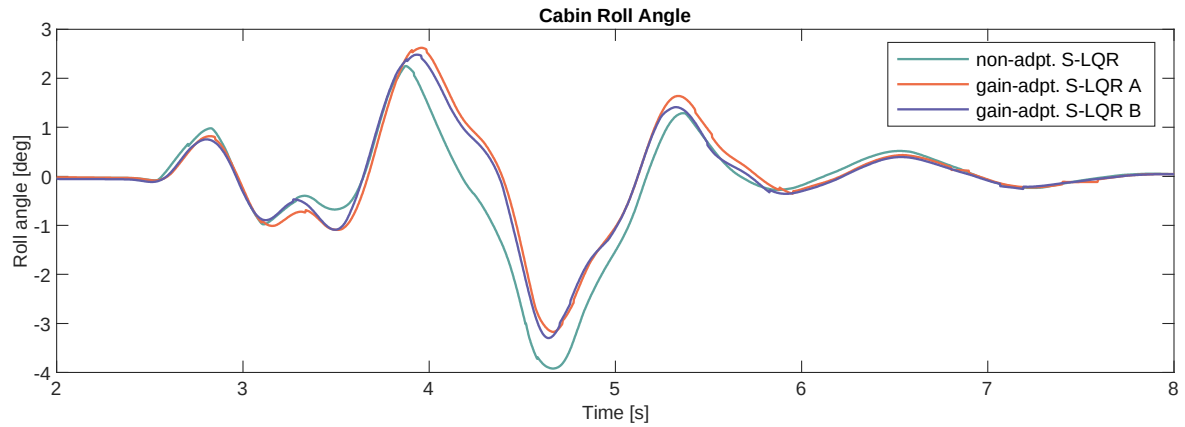


Figure 4.44. Scaled-down cabin roll angle (pure roll motion)

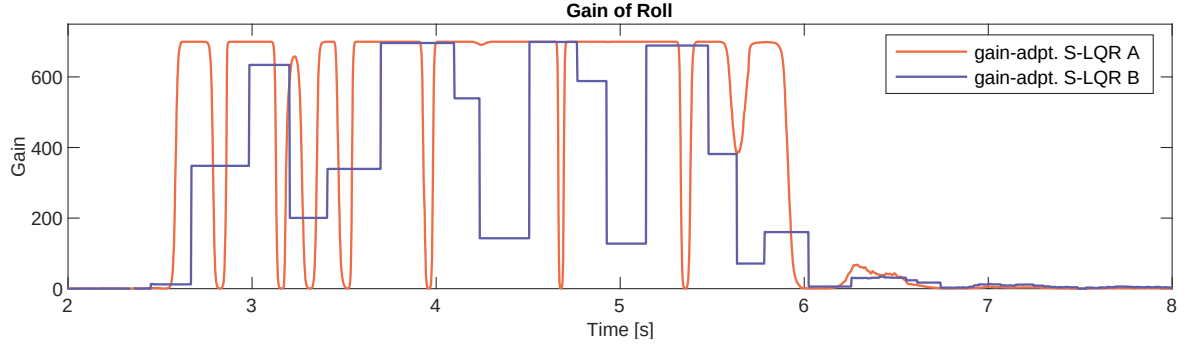


Figure 4.45. Adaptive gain of roll (pure roll motion)

Table 4.4. Percentage of improvement compared to passive suspension (pure roll motion)

	Skyhook	LQR	Integrated S-LQR	Gain-adapt. S-LQR A	Gain-adapt. S-LQR B
Roll acceleration	24%	26%	36%	33%	38%
Roll angle	-6%	15%	13%	8%	14%

4.4.3.3 Emergency brake and acceleration

In this scenario, the Stewart Platform was moved only in the pitch direction to simulate the emergency brake and aggressive acceleration scenarios. The cabin's pitch rate was measured by the 6-axis IMU attached to the cabin floor.

In Figure 4.46, Figure 4.48, and Figure 4.50, the cabin's pitch responses of passive, Skyhook, LQR, and non-adaptive Skyhook-LQR were compared. Skyhook reduced the cabin's pitch acceleration by 20% compared to the passive suspension, while LQR and non-adaptive Skyhook-LQR provided 34% and 39% reductions, as expected. In addition, Skyhook increased the pitch rate by 1% compared to the passive mode, while both LQR (25%) and non-adaptive Skyhook-LQR (29%) minimized the pitch rate to a large extent. Regarding the cabin's pitch angle, the non-adaptive Skyhook-LQR provided a 42% improvement compared to the passive mode, which was much better than LQR (31%). However, Skyhook gave an unsatisfactory result by a 1% increment.

Figure 4.47, Figure 4.49, and Figure 4.51 compare the cabin's pitch responses of non-adaptive Skyhook-LQR, gain-adaptive Skyhook-LQR with algorithm A, and gain-adaptive Skyhook-LQR with

algorithm B. By implementing the two gain-adaptive algorithms, the cabin's pitch acceleration was reduced by 40% (A) and 44% (B), respectively. The cabin's pitch rate was reduced by 20% with gain-adaptive algorithm A, slightly better than algorithm B's (18%). Regarding the pitch angle, algorithm A dropped the roll angle by 26%, while algorithm B reduced it by 30%.

Figure 4.52 shows the gain scheduling process in this scenario. Since the pitch motions were dominant, the corresponding gain q_5 was adjusted from 0 to the maximum pre-defined value of 200. Apparently, algorithm B switched the gain value more moderately than algorithm A, which was beneficial to the system robustness. Overall, the gain-adaptive algorithm further improved the performance of the integrated Skyhook-LQR controller to some extent, and algorithm B performed better than algorithm A. The percentages of improvements by each control approach are listed and compared in Table 4.5.

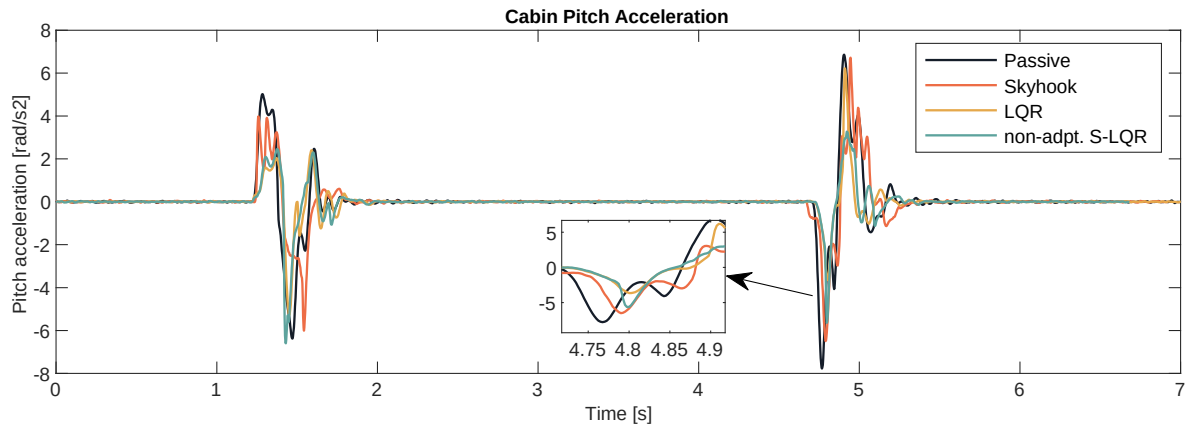


Figure 4.46. Scaled-down cabin pitch acceleration (emergency brake and acceleration)

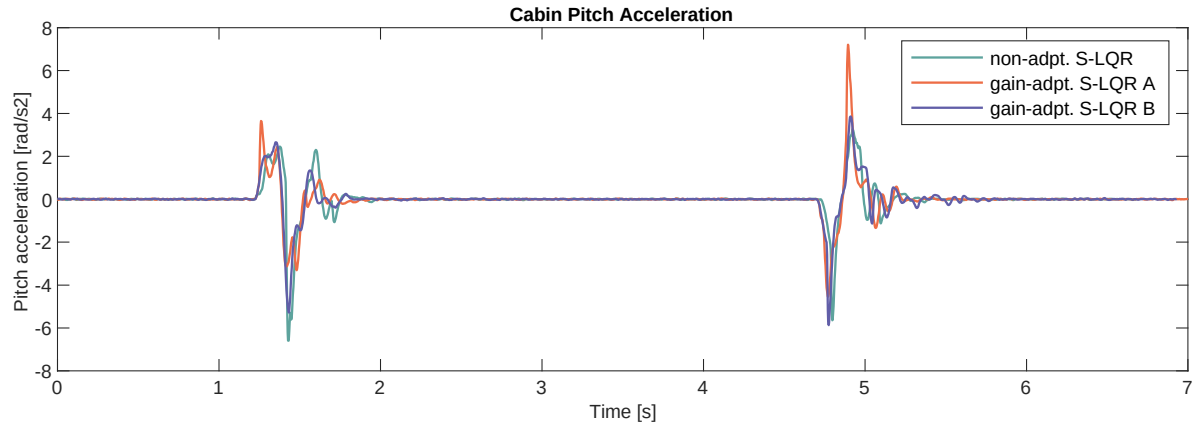


Figure 4.47. Scaled-down cabin pitch acceleration (emergency brake and acceleration)

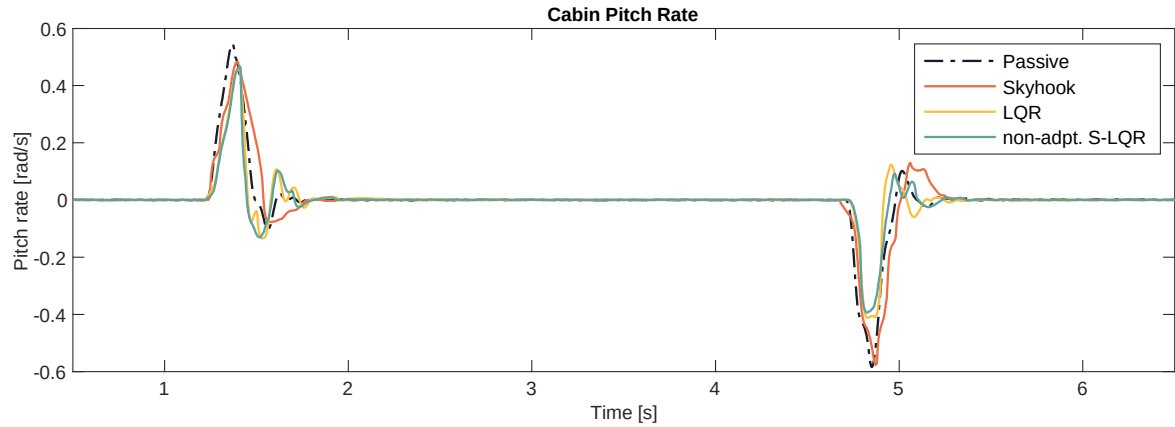


Figure 4.48. Scaled-down cabin pitch rate (emergency brake and acceleration)

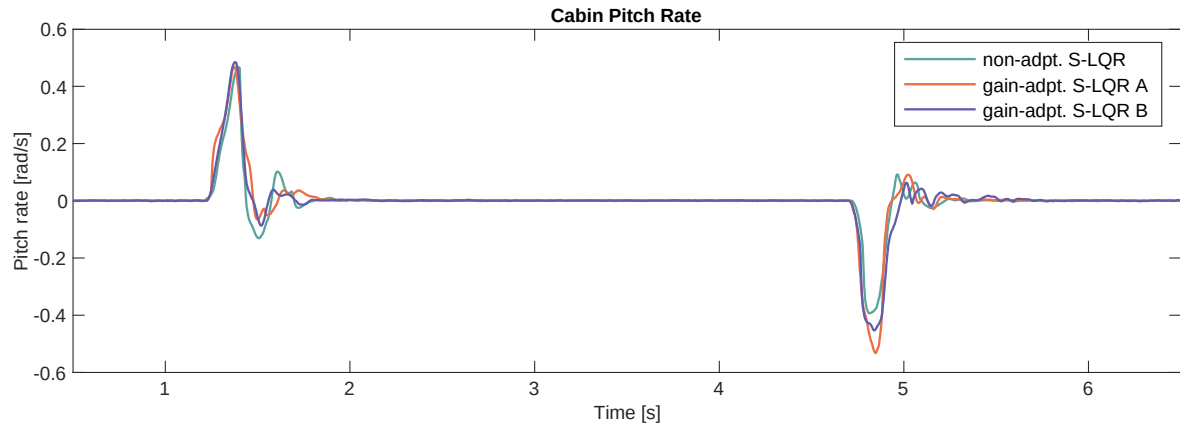


Figure 4.49. Scaled-down cabin pitch rate (emergency brake and acceleration)

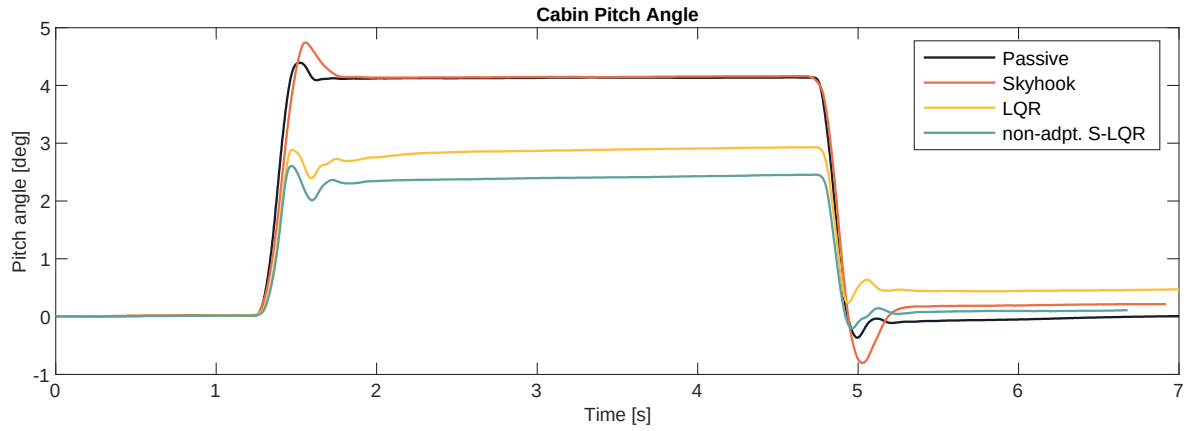


Figure 4.50. Scaled-down cabin pitch angle (emergency brake and acceleration)

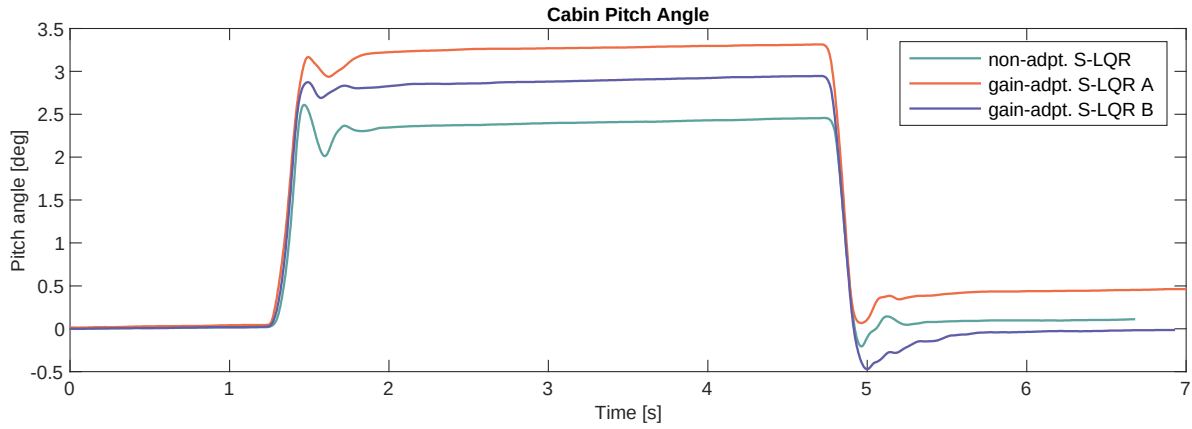


Figure 4.51. Scaled-down cabin pitch angle (emergency brake and acceleration)

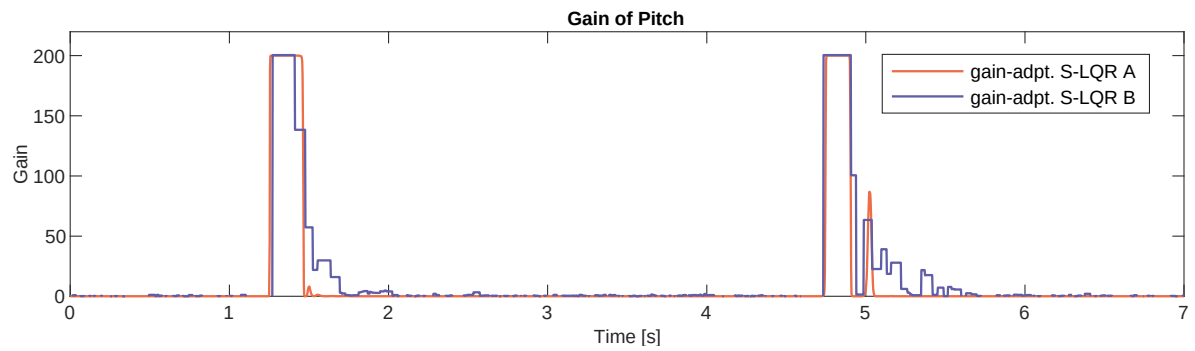


Figure 4.52. Adaptive gain of pitch (emergency brake and acceleration)

Table 4.5. Percentage of improvement compared to passive suspension (emergency brake)

	Skyhook	LQR	Integrated S-LQR	Gain-adapt. S-LQR A	Gain-adapt. S-LQR B
Pitch acceleration	20%	34%	39%	40%	44%
Pitch angle	-1%	31%	42%	26%	30%

4.4.3.4 Offset-bump at 20 km/h

In this scenario, the Stewart Platform was moved in the vertical, pitch, and roll directions to simulate when the truck was driven on an offset-bump road. The 6-axis IMU measured the cabin's vertical acceleration and angular rates.

Figure 4.53 compares the cabin's vertical acceleration responses of passive, Skyhook, LQR, and non-adaptive Skyhook-LQR. Skyhook significantly reduced the vertical disturbance by 35% compared to the passive mode, which was better than both LQR (24%) and non-adaptive Skyhook-LQR (29%). However, an undesired peak shock occurred at 3.7s, which was because the roll motion became large at this time, which affected the Skyhook control signal to a large extent. Figure 4.54 compares the cabin's vertical acceleration of non-adaptive Skyhook-LQR, gain-adaptive Skyhook-LQR with algorithm A, and gain-adaptive Skyhook-LQR with algorithm B. By implementing the two gain-adaptive algorithms, the cabin's vertical acceleration was reduced by 23% (A) and 22% (B), which were slightly worse than the non-adaptive mode (29%).

In Figure 4.55, Figure 4.57, and Figure 4.59, the cabin's roll responses of passive, Skyhook, LQR, and non-adaptive Skyhook-LQR were represented. Skyhook reduced the scaled-down cabin's roll acceleration by 11% compared to the passive suspension, which was worse than both LQR (19 %) and non-adaptive Skyhook-LQR (27%). Regarding the cabin's roll rate, LQR and non-adaptive Skyhook-LQR maintained their performance by providing 25% and 21% improvements, respectively. While Skyhook only reduced the roll rate by 3%. Additionally, LQR and non-adaptive Skyhook-LQR significantly decreased the cabin's roll angle by 37% and 41%, which were much better than Skyhook (11%).

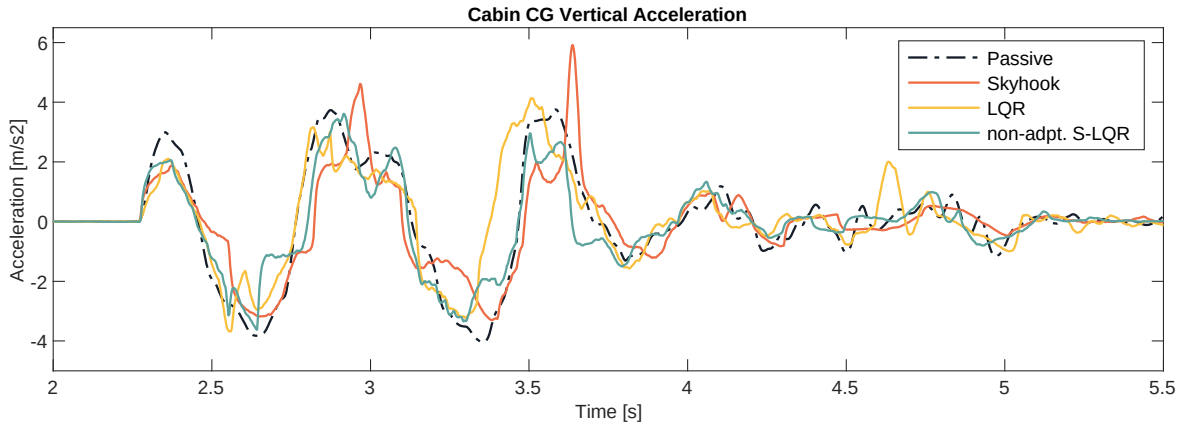


Figure 4.53. Scaled-down cabin vertical acceleration (offset-bump at 20 km/h)

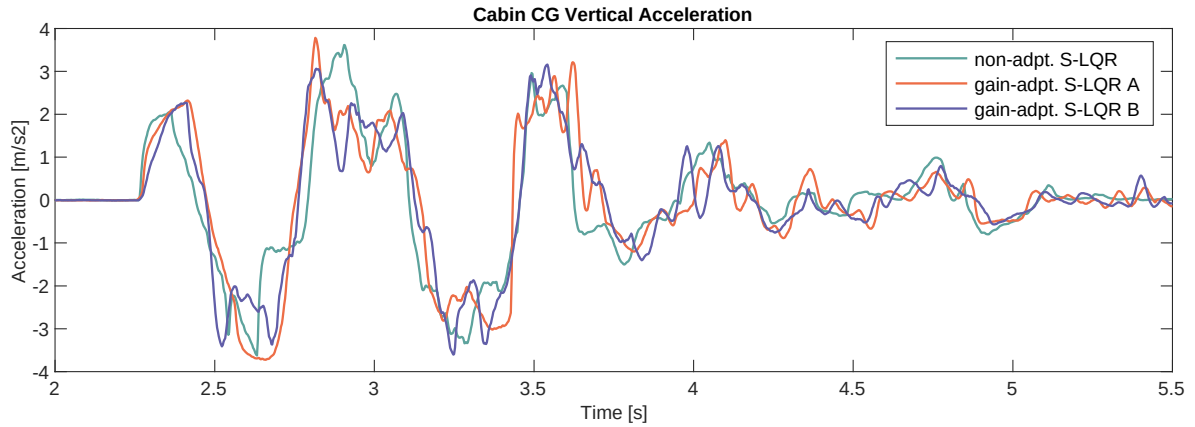


Figure 4.54. Scaled-down cabin vertical acceleration (offset-bump at 20 km/h)

In Figure 4.56, Figure 4.58, and Figure 4.60, the cabin's roll responses of non-adaptive Skyhook-LQR, gain-adaptive Skyhook-LQR with algorithm A, and gain-adaptive Skyhook-LQR with algorithm B were compared. By implementing the two gain-adaptive algorithms, the cabin's roll acceleration was reduced by 41% (A) and 43% (B), which were better than the non-adaptive one (27%) to a large extent. The cabin's roll rate was reduced by 28% with gain-adaptive algorithm A, which was almost the same as algorithm B (29%) but better than the non-adaptive one (21%). Regarding the roll angle, algorithm A dropped the roll angle by 25%, and algorithm B reduced it by 28%, which was worse than the non-adaptive mode (41%).

Figure 4.61, Figure 4.63, and Figure 4.65 show the cabin's pitch responses of passive, Skyhook, LQR, and non-adaptive Skyhook-LQR. Compared to the passive suspension, Skyhook made the cabin's

pitch acceleration even worse with a 20% increment, while LQR and non-adaptive Skyhook-LQR provided a 14% and 18% reduction, as expected. Regarding the cabin's pitch rate, LQR maintained its performance by providing a 16% improvement, but non-adaptive Skyhook-LQR performed better with a 37% reduction. However, Skyhook increased the pitch rate by 16%, which was unsatisfactory.

In Figure 4.62, Figure 4.64, and Figure 4.66, the cabin's pitch responses of non-adaptive Skyhook-LQR, gain-adaptive Skyhook-LQR with algorithm A, and gain-adaptive Skyhook-LQR with algorithm B were compared. By implementing the two gain-adaptive algorithms, the cabin's pitch acceleration was reduced by 7% (A) and 32% (B), which were significantly better than the non-adaptive one (18%). The cabin's pitch rate was reduced by 20% with gain-adaptive algorithm A, worse than algorithm B (44%) and the non-adaptive one (37%).

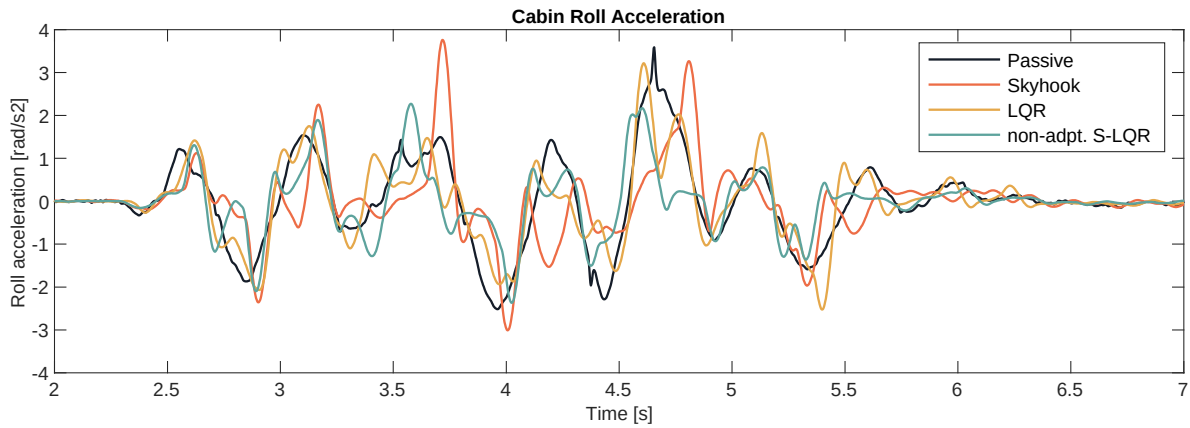


Figure 4.55. Scaled-down cabin roll acceleration (offset-bump at 20 km/h)

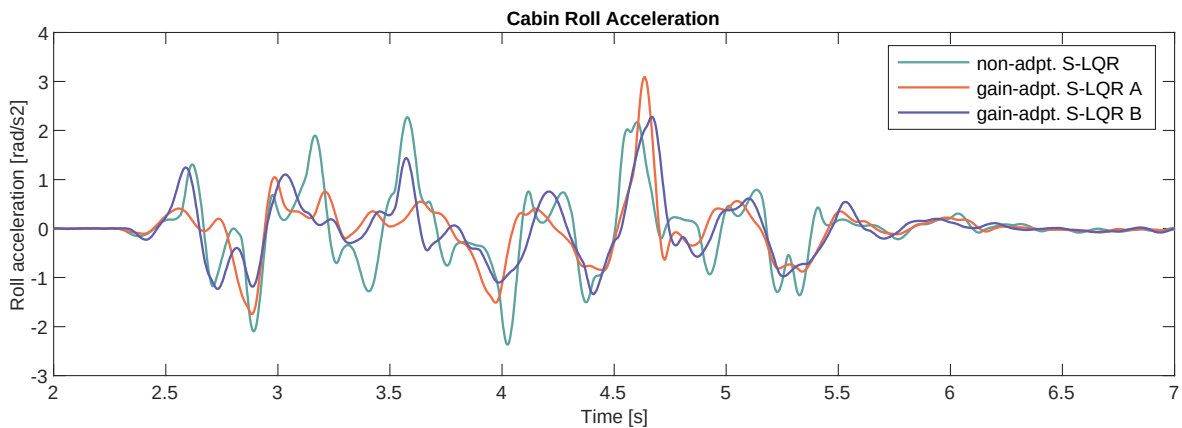


Figure 4.56. Scaled-down cabin roll acceleration (offset-bump at 20 km/h)

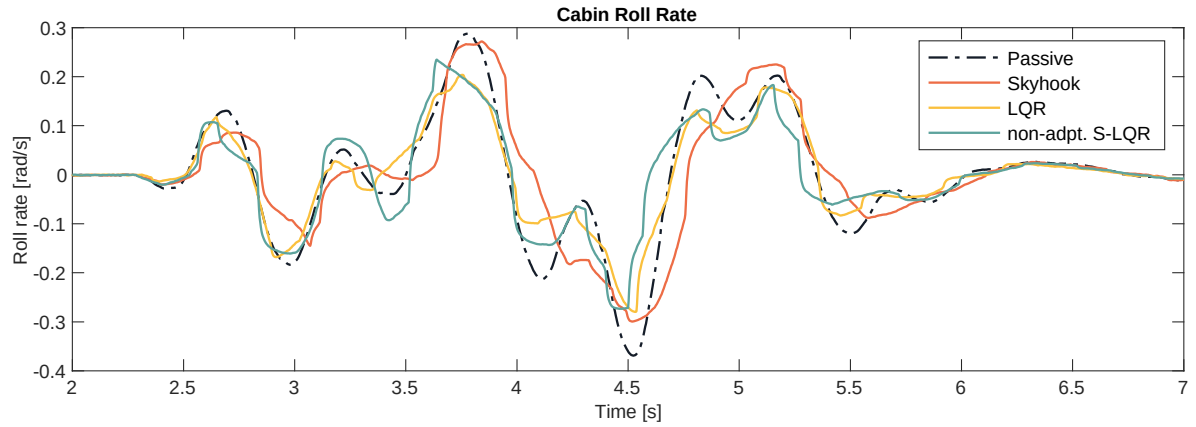


Figure 4.57. Scaled-down cabin roll rate (offset-bump at 20 km/h)

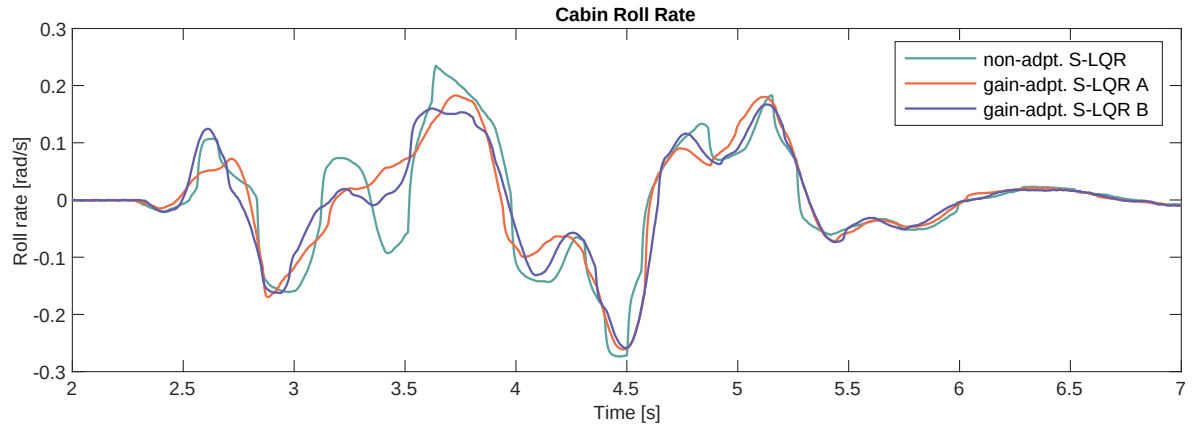


Figure 4.58. Scaled-down cabin roll rate (offset-bump at 20 km/h)

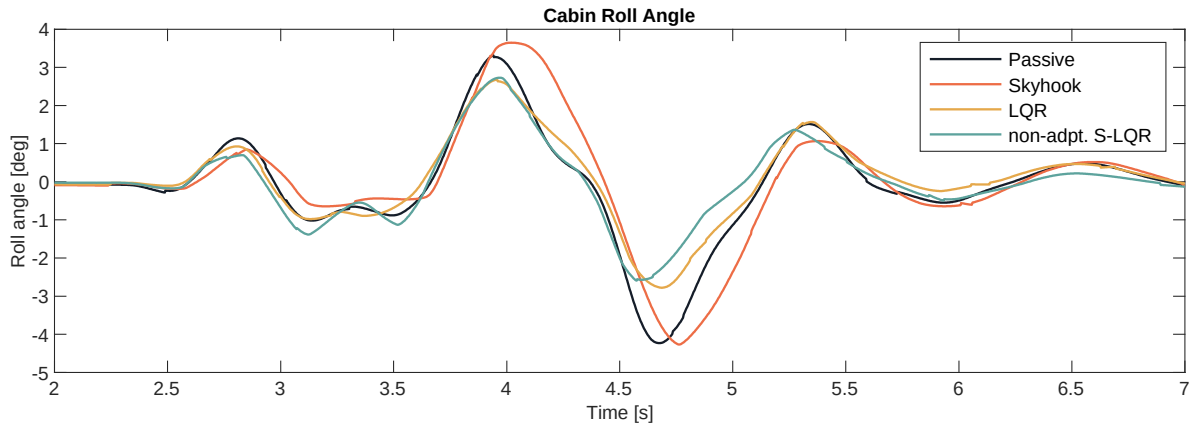


Figure 4.59. Scaled-down cabin roll angle (offset-bump at 20 km/h)

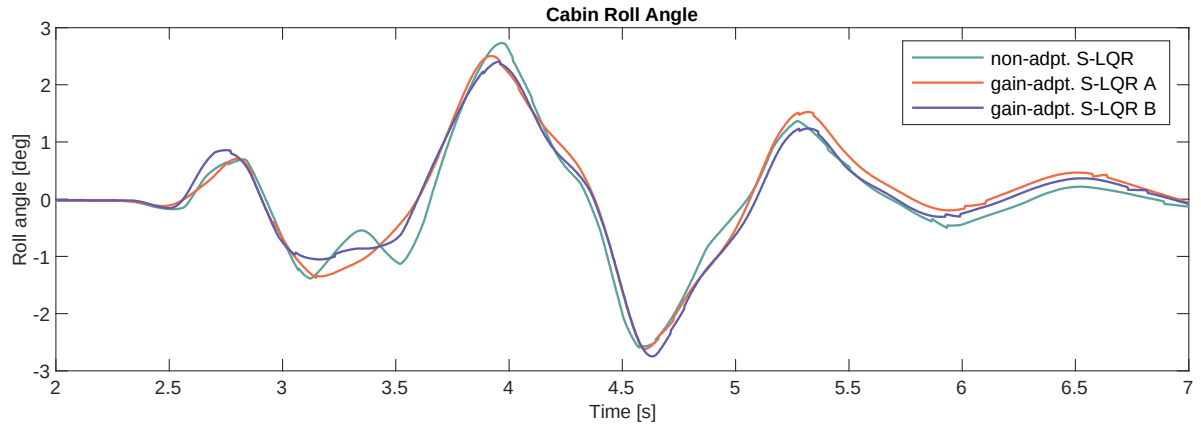


Figure 4.60. Scaled-down cabin roll angle (offset-bump at 20 km/h)

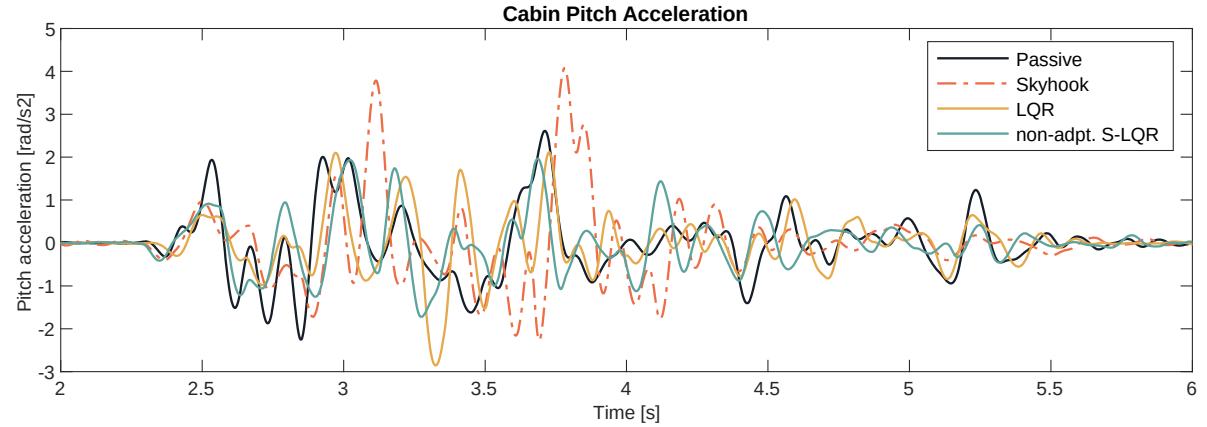


Figure 4.61. Scaled-down cabin pitch acceleration (offset-bump at 20 km/h)

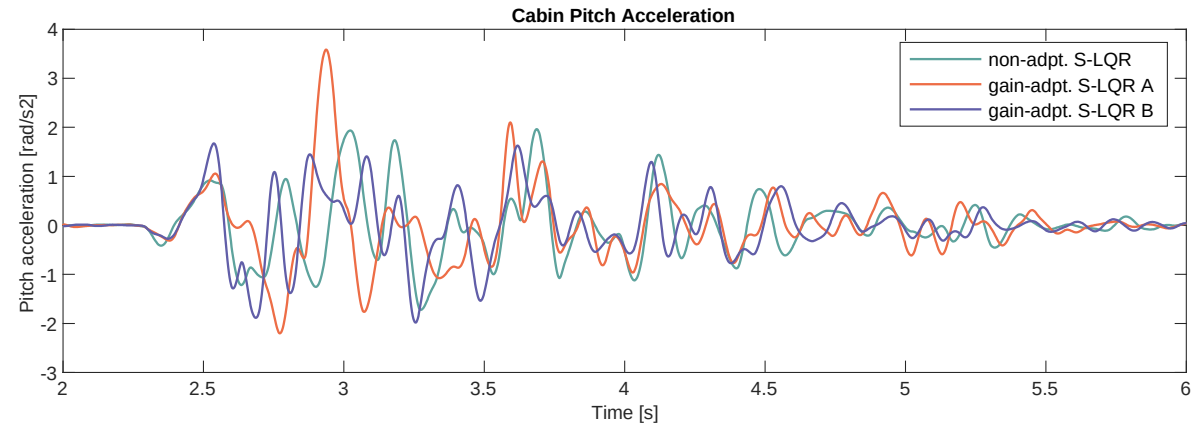


Figure 4.62. Scaled-down cabin pitch acceleration (offset-bump at 20 km/h)

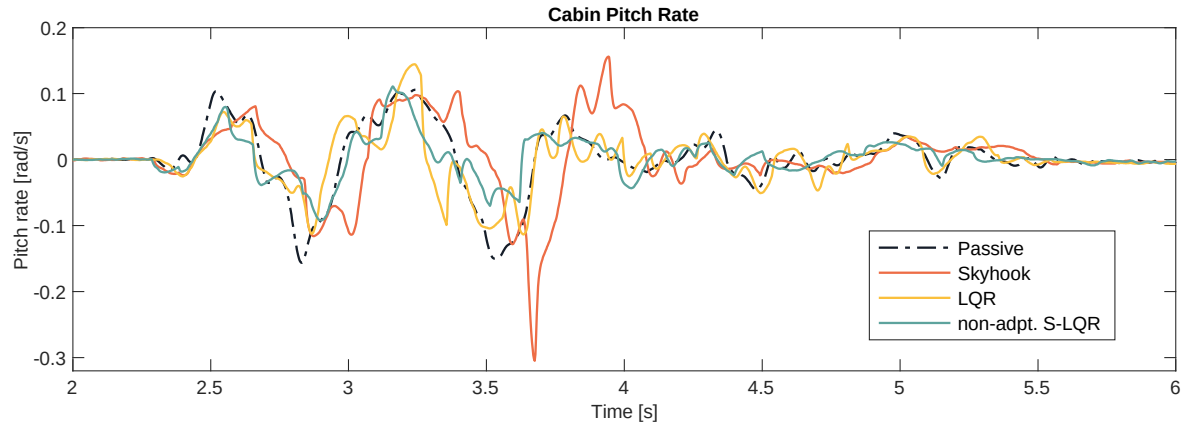


Figure 4.63. Scaled-down cabin pitch rate (offset-bump at 20 km/h)

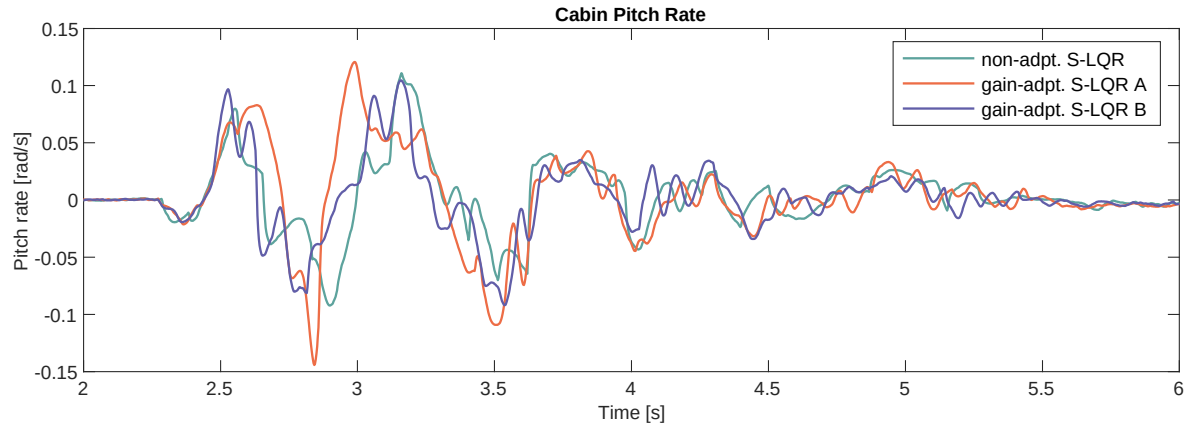


Figure 4.64. Scaled-down cabin pitch rate (offset-bump at 20 km/h)

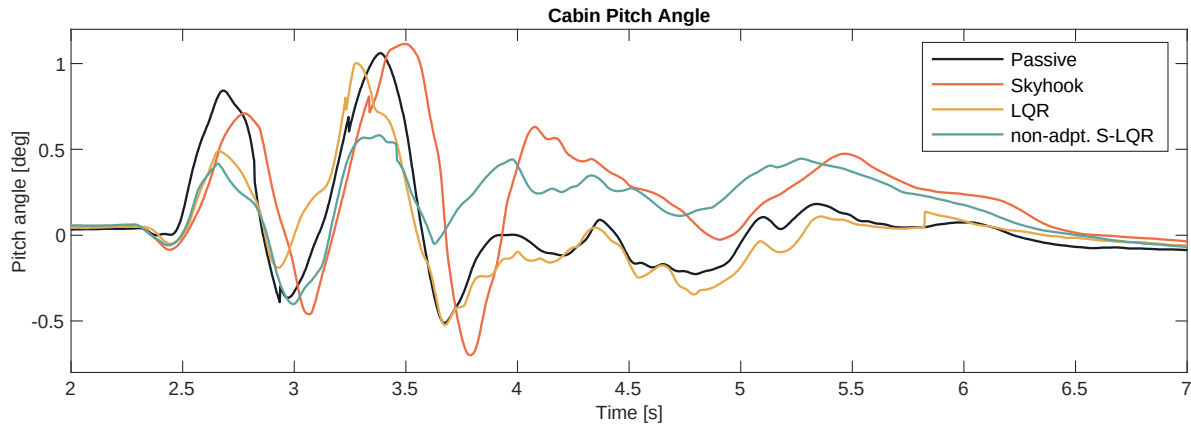


Figure 4.65. Scaled-down cabin pitch angle (offset-bump at 20 km/h)

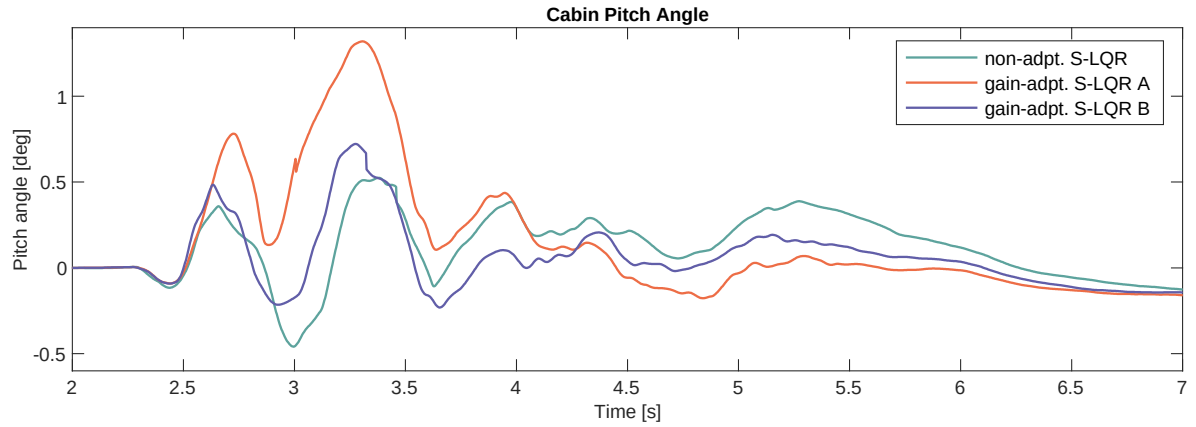


Figure 4.66. Scaled-down cabin pitch angle (offset-bump at 20 km/h)

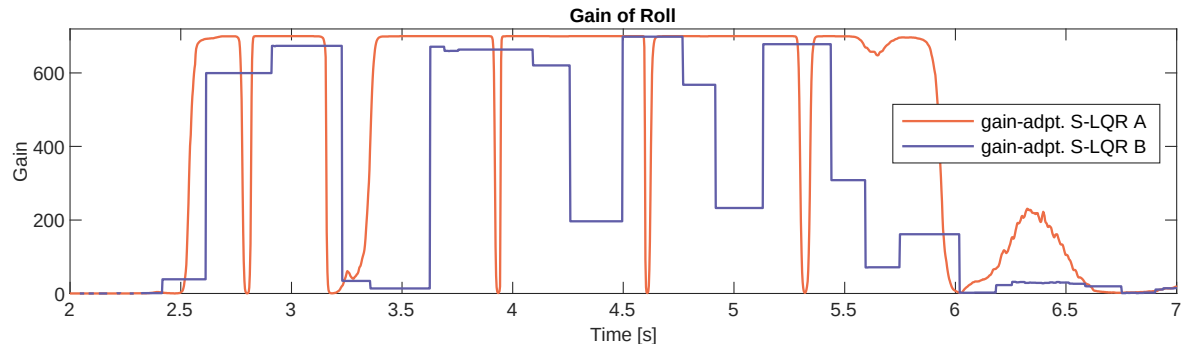


Figure 4.67. Adaptive gain of roll (offset-bump at 20 km/h)

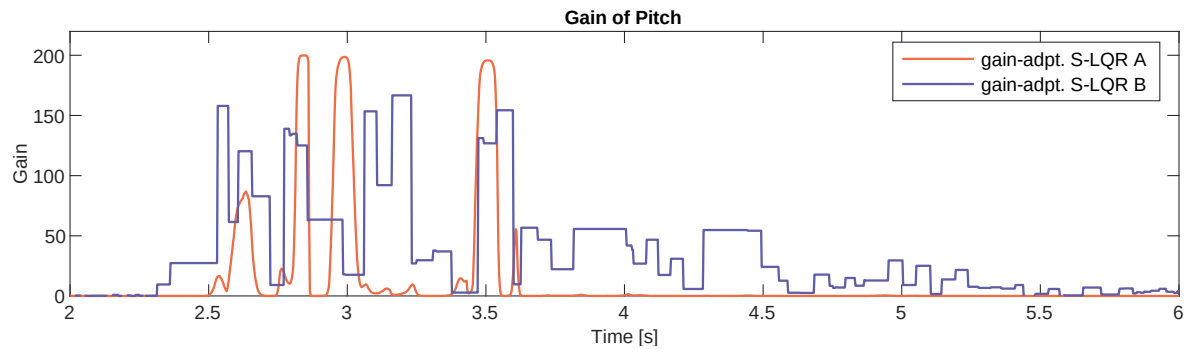


Figure 4.68. Adaptive gain of pitch (offset-bump at 20 km/h)

Figure 4.67 and Figure 4.68 show the gain scheduling process in this scenario. Since the roll motions were dominant most of the time, the corresponding gain q_5 was gradually adjusted from 0 to the maximum pre-defined value of 700 by algorithm B, which was a benefit to the system robustness, while

algorithm A generated some sudden changes. In addition, the pitch motion was significant between 2.5s to 3.7s, so the gain of pitch reached its high range during this period. Similarly, algorithm B's gain scheduling was smoother than algorithm A's. Overall, the gain-adaptive algorithm further improved the performance of the integrated Skyhook-LQR controller to some extent, and algorithm B performed better than algorithm A in most cases. The percentages of improvements by each control approach are listed and compared in Table 4.6.

Table 4.6. Percentage of improvement compared to passive suspension (offset-bump)

	Skyhook	LQR	Integrated S-LQR	Gain-adapt. S-LQR A	Gain-adapt. S-LQR B
Vertical acceleration	35%	24%	29%	23%	22%
Pitch acceleration	-20%	14%	18%	7%	32%
Roll acceleration	11%	19%	27%	41%	43%

4.5 Summary

The chapter presents and discusses the simulation and experimental studies on semi-active suspension and their adaptive damping control systems. The examined control systems include Skyhook, LQR, MPC, integrated Skyhook-LQR, and gain-adaptive Skyhook-LQR. The above control algorithms were evaluated through simulation and experimental studies. The simulation results are compared and summarized in Table 4.4, Table 4.5, and Table 4.6. A summary of the experimental studies is presented in Table 4.7. Generally, the Skyhook approach is widely used in the automotive industry because of its excellent vertical vibration attenuation performance. The Skyhook algorithm's ability to deliver exceptional ride comfort while demanding minimal computational power has made it a popular choice for implementation on cost-effective automotive-grade microcontrollers. However, it is imperative to acknowledge that the Skyhook approach encounters limitations regarding vehicle roll and pitch control tasks. Thus, an LQR controller, a multi-objective optimal control approach, was developed to achieve coordinated suspension control. As expected, it improved the vehicle's vertical, roll, and pitch dynamics at the same time. Nonetheless, the control performance of the semi-active suspension system faced constraints imposed by the boundaries of the adaptive dampers that were not been adequately considered so far. Another multi-objective optimizer called MPC was utilized to solve

this issue. It effectively addressed the limitations by incorporating proper modelling and consideration of damping dissipative constraints. Meanwhile, the MPC algorithm showcased its predictive capabilities by estimating the future motions of the sprung mass over a prediction horizon. However, it is worth noting that the complexity of the MPC algorithm significantly escalated the computational costs, rendering it an expensive and challenging choice for practical applications. In response to this computational issue, a novel control approach named integrated Skyhook-LQR was introduced in this study. It significantly optimized the vehicle dynamics in all three directions (vertical, roll, and pitch) and, most importantly, could be processed in real-time and implemented in semi-active suspension systems. Furthermore, two gain-adaptive algorithms were developed to intelligently adjust the control gains (weighting matrix) of the LQR algorithm based on available sensor measurements in real time since it is impossible to find one optimum set of gains for various scenarios. Algorithm A dynamically selected the weighting matrix Q in response to the recent disturbance inputs, whereas algorithm B computed the gains based on the current vehicle states.

Table 4.7. Summary of percentage of improvement compared to passive suspension

	Skyhook	LQR	Integrated S-LQR	Gain-adapt. S-LQR A	Gain-adapt. S-LQR B
Experiment: pure roll					
Roll acceleration	24%	26%	36%	33%	38%
Roll angle	-6%	15%	13%	8%	14%
Experiment: emergency brake					
Pitch acceleration	20%	34%	39%	40%	44%
Pitch angle	-1%	31%	42%	26%	30%
Experiment: offset-bump at 20 km/h					
Vertical acceleration	35%	24%	29%	23%	22%
Pitch acceleration	-20%	14%	18%	7%	32%
Roll acceleration	11%	19%	27%	41%	43%

Chapter 5

Adaptive Interconnected Suspension Systems with Adjustable Rotational Stiffness

This chapter introduces two novel interconnected suspension topologies, namely AIS-ARS and AIS-ARPS, which have been designed carefully considering various hydraulic components. Their mathematical models, damping characteristics, and stiffness properties have been thoroughly derived and analyzed to understand their functionality comprehensively. For example, the AIS-ARS topology is implemented on the primary suspension of a heavy truck, while the AIS-ARPS is equipped on the cabin suspension system. Note that both topologies can be easily adjusted and fit for different vehicle models or axle types, which will be introduced in detail in the following sections.

5.1 AIS-ARS

This section proposes a novel adaptive interconnected suspension with adjustable roll stiffness (AIS-ARS) to address the existing designs' shortcomings. The AIS-ARS system uses a unique design that avoids expensive proportional flow control valves, which has been a significant barrier to widespread implementation. Instead, the system employs a simple and cost-effective control strategy that uses two solenoid valves to achieve adaptive damping. The mathematical model for the AIS-ARS system is also straightforward, requiring less computational cost and making it more practical for real-time applications. Moreover, the proposed system has improved ride quality and roll stability. These functionalities include adjustable roll stiffness, enhancing the vehicle's handling performance to adapt to different vehicle types and applications. Overall, the AIS-ARS system represents a significant advancement in the design of interconnected suspension systems, offering a more cost-effective, practical, and versatile solution.

5.1.1 Topology of the AIS-ARS

As shown in Figure 5.1, all the hydraulic components used in the AIS-ARS system are connected through hydraulic hoses, such as four hydraulic cylinders, two flow control valves (manifolds), two accumulators, and one Hydraulic Power Unit (HPU). More information on these components can be found in Appendix B. In detail, four hydraulic cylinders are located at each corner of the suspension and connected through hydraulic hoses. In the first hydraulic circuit, the lower chamber of the front-

left hydraulic cylinder is connected with the upper chamber of the front-right hydraulic cylinder, the lower chamber of the rear-left hydraulic cylinder, and the upper chamber of the rear-right hydraulic cylinder. In the second hydraulic circuit, the upper chamber of the front-left hydraulic cylinder is connected with the lower chamber of the front-right hydraulic cylinder, the upper chamber of the rear-left hydraulic cylinder, and the lower chamber of the rear-right hydraulic cylinder. Besides, one accumulator is assembled along each hydraulic circuit.

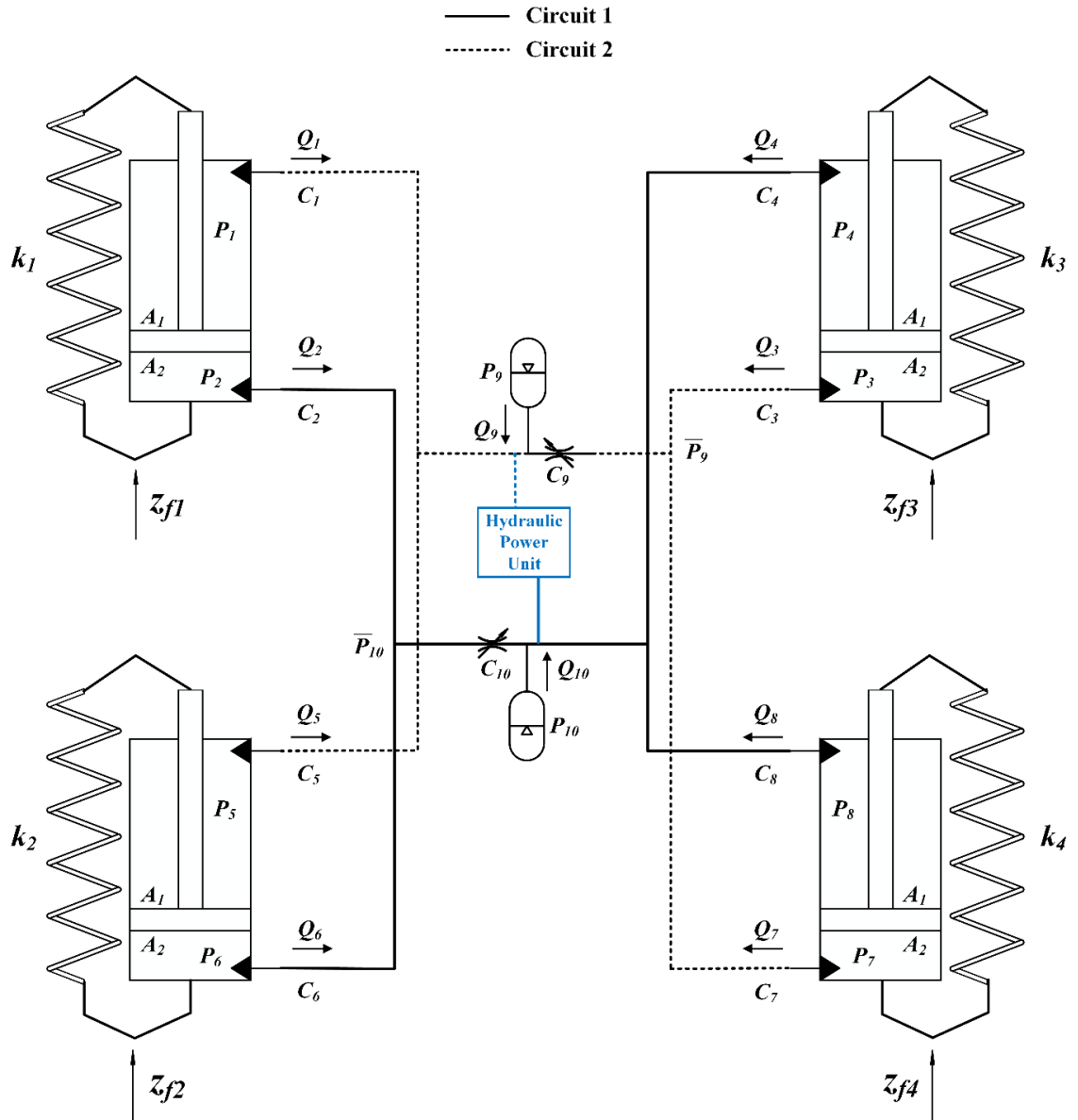


Figure 5.1. Schematic of the roll-plane AIS-ARS

When the suspension is compressed in the vertical direction, the oil flows from the lower chambers to the upper chambers. Due to the unequal cross-sectional areas of the upper and lower chambers, a small amount of oil flows into the accumulator. When the suspension rotates in the roll direction, a large amount of oil flows into the accumulator, which causes gas compression. The increased pressure in the hydraulic circuit results in a certain amount of roll stiffness. Two electro-proportional flow control valves are installed beside the two accumulators' inlets to achieve adaptive damping properties.

Moreover, an HPU is connected to both hydraulic circuits. It usually contains motor, tank, solenoid valves, and other essential hydraulic components. By pumping or draining fluids into the interconnected suspension system, the hydraulic pressure can be easily changed, which results in adjustable roll stiffness. After deriving the mathematical model of the proposed roll-plane IS-ARS, its stiffness and damping properties are thoroughly analyzed. Note that the model accuracy is validated through experimental tests introduced in Section 6.2.

5.1.2 Mathematical modelling

The major assumptions made in formulating the interconnected struts include 1) polytropic process of the gas in accumulators; 2) ideal friction between the floating and main pistons seals and the cylinder; 3) negligible leakage across the pistons; 4) negligible thermal expansion of the cylinder, piston, and oil.

Orifice and valve: The orifice usually possess nonlinear pressure-flow characteristic, which plays a vital role in the hydraulic system. This study uses a simplified orifice model, which assumes that the damper orifice has negligible fluid volume and involves linear pressure loss. Hence, the flow through an orifice can be calculated as

$$Q = \frac{\Delta P}{C} \quad (5.1)$$

where C is the flow resistance coefficient that will be further introduced in Section 5.1.5.

Cylinder chamber: The volume of the chamber varies as the sprung and unsprung masses move relative to one another. Considering the compressibility of the fluid in the chamber, the continuity equation for the cylinder can be written as

$$A\Delta z - \frac{V\Delta P}{\beta} - \int Q dt = 0 \quad (5.2)$$

where A is the area of the cylinder chamber, Δz is the relative vertical displacement between cylinder and piston rod with respect to the equivalent position, V is the volume of the chamber, ΔP is the pressure drop in the chamber, β is the bulk modulus, and Q is the corresponding flow rate.

Accumulators: The accumulator is modelled by assuming an adiabatic process, and the oil compressibility in the accumulator is neglected since the oil stiffness is much greater than that of the nitrogen. The gas pressure and volume at any time in the accumulator gas chamber have the following relationship based on Ideal Gas Law and Boyle's Law:

$$P_0 V_0^n = P_1 V_1^n \quad (5.3)$$

where P_0 and V_0 are the nominal gas pressure and the nominal gas volume in the accumulator, n is the polytropic coefficient, and P_1 and V_1 are the working pressure and gas volume in the accumulator.

The mathematical model of the proposed roll-plane hydro-pneumatic interconnected suspension system has been formulated to investigate its vertical and roll characteristics. It comprises the cabin, four double-acting hydraulic cylinders that are connected through hydraulic hoses, accumulators, and flow control valves. This model considers four disturbance inputs with vertical DOFs (z_{f1} , z_{f2} , z_{f3} , z_{f4}), and the sprung mass m_s with vertical z_s , roll ϕ_s , and pitch θ_s DOFs. The right-handed Cartesian coordinate system defined in ISO-8855 is placed to describe the vehicle system. The x -axis is substantially horizontal and forwards. The y -axis is perpendicular to the cabin's longitudinal plane of symmetry and points to the driver's left hand. The z -axis points upward along the vertical direction. It is worth mentioning that four air springs are installed in parallel with the hydraulic cylinders for carrying the huge load and providing necessary vertical stiffness.

The mathematical modelling of the roll-plane AIS-ARS is derived based on the hydraulic components and their characteristics introduced above. Firstly, the chamber volumes of hydraulic cylinders can be calculated based on the suspension relative displacement. Assuming the strokes of the hydraulic cylinder at its nominal position are S_1 and S_2 , the eight chamber volumes can be represented by

$$\begin{aligned} V_1 &= -A_1 \Delta z_1 + A_1 S_1 \\ V_2 &= A_2 \Delta z_1 + A_2 S_2 \end{aligned} \quad (5.4)$$

$$V_3 = A_2 \Delta z_3 + A_2 S_2$$

$$V_4 = -A_1 \Delta z_3 + A_1 S_1$$

$$V_5 = -A_1 \Delta z_2 + A_1 S_1$$

$$V_6 = A_2 \Delta z_2 + A_2 S_2$$

$$V_7 = A_2 \Delta z_4 + A_2 S_2$$

$$V_8 = -A_1 \Delta z_4 + A_1 S_1$$

Note that the front left (FL), rear left (RL), front right (FR), and rear right (RR) corners are labeled as 1, 2, 3, and 4, respectively. Besides, the suspension relative displacements at the four corners can be written as

$$\begin{aligned}\Delta z_1 &= z_s + l\varphi_s - L_1\theta_s - z_{f1} \\ \Delta z_2 &= z_s + l\varphi_s + L_2\theta_s - z_{f2} \\ \Delta z_3 &= z_s - l\varphi_s - L_1\theta_s - z_{f3} \\ \Delta z_4 &= z_s - l\varphi_s + L_2\theta_s - z_{f4}\end{aligned}\tag{5.5}$$

in which l represents half of the distance between the left and right hydraulic cylinders, L_1 represents the longitudinal distance from the CG location to the front suspension, and L_2 represents the longitudinal distance from the CG location to the rear suspension.

Secondly, the time derivative of fluid pressures in the hydraulic cylinders can be calculated according to equations (5.1) and (5.2) as follows:

$$\begin{aligned}\dot{P}_1 &= \frac{\beta}{V_1} (A_1 \Delta \dot{z}_1 - Q_1), \text{ in which } Q_1 = \frac{1}{c_1} (P_1 - P_9) \\ \dot{P}_2 &= \frac{\beta}{V_2} (-A_2 \Delta \dot{z}_1 - Q_2), \text{ in which } Q_2 = \frac{1}{c_2} (P_2 - \bar{P}_{10}) \\ \dot{P}_3 &= \frac{\beta}{V_3} (-A_2 \Delta \dot{z}_3 - Q_3), \text{ in which } Q_3 = \frac{1}{c_3} (P_3 - \bar{P}_9) \\ \dot{P}_4 &= \frac{\beta}{V_4} (A_1 \Delta \dot{z}_3 - Q_4), \text{ in which } Q_4 = \frac{1}{c_4} (P_4 - P_{10})\end{aligned}\tag{5.6}$$

$$\dot{P}_5 = \frac{\beta}{V_5} (A_1 \Delta \dot{z}_2 - Q_5), \text{ in which } Q_5 = \frac{1}{C_5} (P_5 - P_9)$$

$$\dot{P}_6 = \frac{\beta}{V_6} (-A_2 \Delta \dot{z}_2 - Q_6), \text{ in which } Q_6 = \frac{1}{C_6} (P_6 - \bar{P}_{10})$$

$$\dot{P}_7 = \frac{\beta}{V_7} (-A_2 \Delta \dot{z}_4 - Q_7), \text{ in which } Q_7 = \frac{1}{C_7} (P_7 - \bar{P}_9)$$

$$\dot{P}_8 = \frac{\beta}{V_8} (A_1 \Delta \dot{z}_4 - Q_8), \text{ in which } Q_8 = \frac{1}{C_8} (P_8 - P_{10})$$

in which $C_1 \sim C_8$ represent the equivalent flow resistance parameters of the hydraulic hoses which are constant values. Besides, the pressure drop at the two flow control valves are

$$\bar{P}_9 - P_9 = C_9 (Q_3 + Q_7) \quad (5.7)$$

$$\bar{P}_{10} - P_{10} = C_{10} (Q_2 + Q_6)$$

Note that two flow control valves with the resistance parameters C_9 and C_{10} determines the expected suspension damping characteristics.

Thus, the fluid pressures $\bar{P}_9$ and $\bar{P}_{10}$ can be calculated as follows:

$$\begin{aligned} \left(\frac{1}{C_3} + \frac{1}{C_7} + \frac{1}{C_9} \right) \bar{P}_9 &= \frac{1}{C_3} P_3 + \frac{1}{C_7} P_7 + \frac{1}{C_9} P_9 \\ \left(\frac{1}{C_2} + \frac{1}{C_6} + \frac{1}{C_{10}} \right) \bar{P}_{10} &= \frac{1}{C_2} P_2 + \frac{1}{C_6} P_6 + \frac{1}{C_{10}} P_{10} \end{aligned} \quad (5.8)$$

Thirdly, based on the nominal gas volume, the two accumulator gas volumes can be iteratively calculated:

$$\begin{aligned} V_9 &= V_0 - \int (Q_1 + Q_3 + Q_5 + Q_7) dt \\ V_{10} &= V_0 - \int (Q_2 + Q_4 + Q_6 + Q_8) dt \end{aligned} \quad (5.9)$$

Fourthly, the time derivative of the gas pressures in the accumulators can be calculated according to Eq.(5.3).

$$\begin{aligned}
\dot{P}_9 &= -\frac{nP_0V_0^n}{V_9^{n+1}}\dot{V}_9 = \frac{nP_0V_0^n}{V_9^{n+1}}(Q_1 + Q_3 + Q_5 + Q_7) \\
\dot{P}_{10} &= -\frac{nP_0V_0^n}{V_{10}^{n+1}}\dot{V}_{10} = \frac{nP_0V_0^n}{V_{10}^{n+1}}(Q_2 + Q_4 + Q_6 + Q_8)
\end{aligned} \tag{5.10}$$

Finally, the pressures in hydraulic cylinders $P_1 \sim P_8$ and accumulators $P_9 \sim P_{10}$ can be found based on their time derivatives $\dot{P}_i$ and the system nominal pressure P_0 .

5.1.3 Roll stiffness analysis

Suspension roll stiffness represents the vehicle's rolling resistance about its longitudinal axis. In other words, it determines how much a vehicle's body leans in turn. A higher roll stiffness results in less body roll, which can lead to improved stability. In the conventional suspension, its roll stiffness is usually achieved through anti-roll bars and bushings so that the roll stiffness can be adjusted by changing the diameter or material of the anti-roll bar or the bushing stiffness. Using the proposed AIS-ARS, the roll stiffness adjustment becomes more convenient and cheaper for vehicle manufacturers.

In the suspension steady status, its roll moment can be written as

$$\begin{aligned}
M_\varphi &= -(k_1 + k_2 + k_3 + k_4)l^2\varphi_s - A_1lP_1 + A_2lP_2 - A_2lP_3 + A_1lP_4 - A_1lP_5 + A_2lP_6 \\
&\quad - A_2lP_7 + A_1lP_8 \\
&= -(k_1 + k_2 + k_3 + k_4)l^2\varphi_s + 2l(A_1 + A_2)(P_{10} - P_9)
\end{aligned} \tag{5.11}$$

The roll stiffness can be derived from the restoring roll moment subject to the relative roll deflection as

$$K_\varphi = -\frac{dM_\varphi}{d\varphi} = (k_1 + k_2 + k_3 + k_4)l^2 - 2l(A_1 + A_2)\frac{d}{d\varphi}(P_{10} - P_9) \tag{5.12}$$

The corresponding derivative of the gas pressures in the accumulators can be expressed as

$$\begin{aligned}
\frac{dP_9}{d\varphi} &= \frac{2n(A_1 + A_2)lP_0V_0^n}{V_9^{n+1}} = \frac{2n(A_1 + A_2)lP_0V_0^n}{\gamma_1} \\
\frac{dP_{10}}{d\varphi} &= \frac{-2n(A_1 + A_2)lP_0V_0^n}{V_{10}^{n+1}} = \frac{-2n(A_1 + A_2)lP_0V_0^n}{\gamma_2}
\end{aligned} \tag{5.13}$$

in which

$$\gamma_1 = [V_0 + 2(A_2 - A_1)z_s - 2l(A_1 + A_2)\varphi_s - (A_1 - A_2)(L_2 - L_1)\theta_s]^{n+1}$$

$$\gamma_2 = [V_0 + 2(A_2 - A_1)z_s + 2l(A_1 + A_2)\varphi_s - (A_1 - A_2)(L_2 - L_1)\theta_s]^{n+1}$$

Hence, the roll stiffness is derived as

$$\begin{aligned} K_\varphi &= (k_1 + k_2 + k_3 + k_4)l^2 - 2l(A_1 + A_2)\frac{d}{d\varphi}(P_{10} - P_9) \\ &= (k_1 + k_2 + k_3 + k_4)l^2 + 4n(A_1 + A_2)^2l^2P_0V_0^n\left(\frac{1}{\gamma_1} + \frac{1}{\gamma_2}\right) \end{aligned} \quad (5.14)$$

Besides, the above formulation can be simplified when $n = 1$:

$$K_\varphi = (k_1 + k_2 + k_3 + k_4)l^2 + 8(A_1 + A_2)^2l^2\frac{P_0}{V_0} \quad (5.15)$$

The relevant parameters are selected after a comprehensive study and sensitivity analysis based on a mass-produced dual-axle truck suspension system, which can be found in Table 5.1. Based on Eq.(5.14), the estimated roll stiffness of the AIS-ARS system is shown in Figure 5.2, which is compared to two anti-roll bars. The estimated results of the four example settings are plotted in different colors, i.e., $V_0 = 2.4L$ & $P_0 = 1730psi$, $V_0 = 2.6L$ & $P_0 = 1597psi$, $V_0 = 2.8L$ & $P_0 = 1483psi$, and $V_0 = 3L$ & $P_0 = 1384psi$. It can be seen that the roll stiffness can be adjusted by charging or discharging the hydraulic system. In other words, the roll stiffness K_φ increases with the increase of the system nominal pressure P_0 . Attributed to the air compression in the accumulator, the roll stiffness tends to increase with the increase of the roll angle. Thus, a lower roll stiffness can be realized when the suspension relative roll angle is small, while a higher roll stiffness can be achieved when the angle increases. Overall, the anti-roll property of the IS-ARS has considerable potential for a better compromise between ride comfort and roll stability without equipping the anti-roll bars. Note that more than 84% of the total roll stiffness of the conventional dual-axle suspension comes from the anti-roll bars, and only a small portion generates by the air springs, while the interconnected configuration can provide enough roll stiffness to compensate for the elimination of the anti-roll bars.

Note that the parameters in Table 5.1 may not be the best option for the proposed truck suspension system. These values are chosen based on the available off-the-shelf hydraulic components at this time.

Similar to conventional air suspension systems, the suspension parameters must be fine-tuned to achieve desired properties.

Table 5.1. Parameters of the AIS-ARS system

Symbol	Description	Value
k_1, k_3	Front air spring stiffness	250 kN/m
k_2, k_4	Rear air spring stiffness	250 kN/m
m_s	Sprung mass weight (sum of two drive-axle loads)	22 tons
k_ϕ	Torsion stiffness of the anti-roll bar	9.5 kN·m/deg
I_x	Chassis frame roll moment of inertia	31892 kg/m ²
I_y	Chassis frame pitch moment of inertia	351 kg/m ²
l	Lateral distance from damper to CG	0.525 m
L_1	Distance from CG to front axle	0.3573 m
L_2	Distance from CG to rear axle	1.7073 m
A_1	Rod side chamber cross-sectional area (rod diameter is 1.5 inches)	0.00342 m ²
A_2	Bore side chamber cross-sectional area (bore diameter is 3.5 inches)	0.00456 m ²
S_1, S_2	Half of the full stroke length	4 inches
n	Polytropic coefficient	1.38
β	Fluid bulk modulus	1400 MPa
C_i	Resistance coefficient of hydraulic hose ($i = 1, 2, 3, \dots, 8$)	5×10^7 Pa·s/m ³

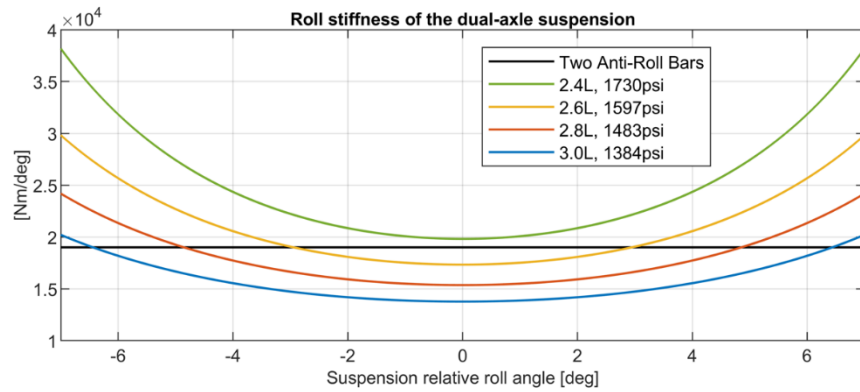


Figure 5.2. Roll stiffness comparison of the roll-plane AIS-ARS

5.1.4 Vertical stiffness analysis

Suspension vertical stiffness refers to the resistance effects when the suspension is retracting or extending, affecting overall ride quality and stability. The vertical stiffness of the proposed AIS-ARS is analyzed in this section, which is useful for suspension tuning and air spring selection. Specifically, the area difference between the cylinder's bore side and rod side results in a certain amount of additional vertical stiffness so that a smaller air spring can be used with AIS-ARS. The suspension force at steady status can be written as

$$\begin{aligned} F_z = & -(k_1 + k_2 + k_3 + k_4)z_s + (k_1L_1 - k_2L_2 + k_3L_1 - k_4L_2)\theta_s + k_1z_{f1} + k_2z_{f2} \\ & + k_3z_{f3} + k_4z_{f4} - A_1P_1 + A_2P_2 + A_2P_3 - A_1P_4 - A_1P_5 + A_2P_6 \\ & + A_2P_7 - A_1P_8 = -(k_1 + k_2 + k_3 + k_4)z_s + 2(A_2 - A_1)(P_9 + P_{10}) \end{aligned} \quad (5.16)$$

The vertical stiffness (suspension rate) K_z can be obtained by taking the first derivative of the suspension force with respect to the relative displacement as

$$K_z = -\frac{dF_z}{dz} = k_1 + k_2 + k_3 + k_4 - 2(A_2 - A_1)\frac{d}{dz}(P_9 + P_{10}) \quad (5.17)$$

According to the vehicle motion, the gas volumes in two accumulators can be calculated as follows:

$$\begin{aligned} V_9 = & V_0 + 2(A_2 - A_1)z_s - 2l(A_1 + A_2)\varphi_s + (A_2 - A_1)(L_2 - L_1)\theta_s + A_1z_{f1} + A_1z_{f2} \\ & - A_2z_{f3} - A_2z_{f4} \\ V_{10} = & V_0 + 2(A_2 - A_1)z_s + 2l(A_1 + A_2)\varphi_s + (A_2 - A_1)(L_2 - L_1)\theta_s - A_2z_{f1} - A_2z_{f2} \\ & + A_1z_{f3} + A_1z_{f4} \end{aligned} \quad (5.18)$$

Hence, the corresponding derivative of the gas pressures in the accumulators can be expressed as

$$\begin{aligned} \frac{dP_9}{dz} &= \frac{-2n(A_2 - A_1)P_0V_0^n}{V_9^{n+1}} = \frac{-2n(A_2 - A_1)P_0V_0^n}{\gamma_3} \\ \frac{dP_{10}}{dz} &= \frac{-2n(A_2 - A_1)P_0V_0^n}{V_{10}^{n+1}} = \frac{-2n(A_2 - A_1)P_0V_0^n}{\gamma_4} \end{aligned} \quad (5.19)$$

in which

$$\gamma_3 = [V_0 + 2(A_2 - A_1)z_s - 2l(A_1 + A_2)\varphi_s + (A_2 - A_1)(L_2 - L_1)\theta_s]^{n+1}$$

$$\gamma_4 = [V_0 + 2(A_2 - A_1)z_s + 2l(A_1 + A_2)\varphi_s + (A_2 - A_1)(L_2 - L_1)\theta_s]^{n+1}$$

The vertical stiffness K_z is derived as

$$\begin{aligned} K_z &= k_1 + k_2 + k_3 + k_4 + 4n(A_2 - A_1)^2 P_0 V_0^n \left(\frac{1}{V_9^{n+1}} + \frac{1}{V_{10}^{n+1}} \right) \\ &= k_1 + k_2 + k_3 + k_4 + 4n(A_2 - A_1)^2 P_0 V_0^n \left(\frac{1}{\gamma_3} + \frac{1}{\gamma_4} \right) \end{aligned} \quad (5.20)$$

Besides, the above formulation can be simplified when $n = 1$:

$$K_z = k_1 + k_2 + k_3 + k_4 + 8(A_2 - A_1)^2 \frac{P_0}{V_0} \quad (5.21)$$

It can be seen that the suspension rate of the interconnected suspension system at the nominal position is proportional to the nominal gas pressure and squared rod area and inversely proportional to the nominal gas volume. In other words, the effective area that affects vertical stiffness is the rod area, which allows designers to tune vertical stiffness by adjusting the piston area ratio. Note that the suspension rate is mainly decided by springs with a smaller rod size, while the interconnection configuration provides additional stiffness with a larger rod size due to the gas compression in the accumulators.

5.1.5 Damping characteristics

Suspension damping characteristics refer to how the suspension system responds to motions and disturbances. By controlling the amount of energy absorbed and dissipated by the damping system, the proposed interconnected suspension system aims to provide excellent roll stability and a smooth ride. Specifically, the forces generated by each hydraulic cylinder can be calculated by

$$\begin{aligned} F_1 &= A_2 P_2 - A_1 P_1 = A_2 C_2 Q_2 + A_2 \bar{P}_{10} - A_1 C_1 Q_1 - A_1 P_9 \\ F_2 &= A_2 P_6 - A_1 P_5 = A_2 C_6 Q_6 + A_2 \bar{P}_{10} - A_1 C_5 Q_5 - A_1 P_9 \\ F_3 &= A_2 P_3 - A_1 P_4 = A_2 C_3 Q_3 + A_2 \bar{P}_9 - A_1 C_4 Q_4 - A_1 P_{10} \\ F_4 &= A_2 P_7 - A_1 P_8 = A_2 C_7 Q_7 + A_2 \bar{P}_9 - A_1 C_8 Q_8 - A_1 P_{10} \end{aligned} \quad (5.22)$$

in which the flow rates along the hydraulic circuits can be calculated as follows:

$$\begin{aligned}
Q_1 &= A_1 \Delta \dot{z}_1 \\
Q_2 &= -A_2 \Delta \dot{z}_1 \\
Q_3 &= -A_2 \Delta \dot{z}_3 \\
Q_4 &= A_1 \Delta \dot{z}_3 \\
Q_5 &= A_1 \Delta \dot{z}_2 \\
Q_6 &= -A_2 \Delta \dot{z}_2 \\
Q_7 &= -A_2 \Delta \dot{z}_4 \\
Q_8 &= A_1 \Delta \dot{z}_4 \\
Q_9 &= -Q_1 - Q_3 - Q_5 - Q_7 \\
Q_{10} &= -Q_2 - Q_4 - Q_6 - Q_8
\end{aligned} \tag{5.23}$$

Combining equations (5.7), (5.22), and (5.23), the cylinder forces can be simplified as

$$\begin{aligned}
F_1 &= -[A_1^2 C_1 + A_2^2 C_2 + 2A_2^2 C_{10}] \dot{z}_s - l[A_1^2 C_1 + A_2^2 C_2 + 2A_2^2 C_{10}] \dot{\phi}_s \\
&\quad + [A_1^2 C_1 L_1 + A_2^2 C_2 L_1 + A_2^2 C_{10} L_1 - A_2^2 C_{10} L_2] \dot{\theta}_s \\
&\quad + (A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_{10}) \dot{z}_{f1} + A_2^2 C_{10} \dot{z}_{f2} + \dots \\
F_2 &= -[A_1^2 C_5 + A_2^2 C_6 + 2A_2^2 C_{10}] \dot{z}_s - l[A_1^2 C_5 + A_2^2 C_6 + 2A_2^2 C_{10}] \dot{\phi}_s \\
&\quad - [A_1^2 C_5 L_2 + A_2^2 C_6 L_2 + A_2^2 C_{10} L_2 - A_2^2 C_{10} L_1] \dot{\theta}_s \\
&\quad + (A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_{10}) \dot{z}_{f2} + A_2^2 C_{10} \dot{z}_{f1} + \dots \\
F_3 &= -[A_2^2 C_3 + A_1^2 C_4 + 2A_2^2 C_9] \dot{z}_s + l[A_2^2 C_3 + A_1^2 C_4 + 2A_2^2 C_9] \dot{\phi}_s \\
&\quad + [A_2^2 C_3 L_1 + A_1^2 C_4 L_1 + A_2^2 C_9 L_1 - A_2^2 C_9 L_2] \dot{\theta}_s \\
&\quad + (A_2^2 C_3 + A_1^2 C_4 + A_2^2 C_9) \dot{z}_{f3} + A_2^2 C_9 \dot{z}_{f4} + \dots \\
F_4 &= -[A_2^2 C_7 + A_1^2 C_8 + 2A_2^2 C_9] \dot{z}_s + l[A_2^2 C_7 + A_1^2 C_8 + 2A_2^2 C_9] \dot{\phi}_s \\
&\quad - [A_2^2 C_7 L_2 + A_1^2 C_8 L_2 + A_2^2 C_9 L_2 - A_2^2 C_9 L_1] \dot{\theta}_s \\
&\quad + (A_2^2 C_7 + A_1^2 C_8 + A_2^2 C_9) \dot{z}_{f4} + A_2^2 C_9 \dot{z}_{f3} + \dots
\end{aligned} \tag{5.24}$$

Thus, the total vertical force generated by the four hydraulic cylinders can be written as

$$\begin{aligned}
F_z &= F_1 + F_2 + F_3 + F_4 \\
&= -(A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8 \\
&\quad + 4A_2^2 C_9 + 4A_2^2 C_{10})\dot{z}_s \\
&\quad - l(A_1^2 C_1 + A_2^2 C_2 - A_2^2 C_3 - A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 - A_2^2 C_7 - A_1^2 C_8 \\
&\quad - 4A_2^2 C_9 + 4A_2^2 C_{10})\dot{\phi}_s \\
&\quad + [A_1^2 L_1 C_1 + A_2^2 L_1 C_2 + A_2^2 L_1 C_3 + A_1^2 L_1 C_4 - A_1^2 L_2 C_5 - A_2^2 L_2 C_6 \\
&\quad - A_2^2 L_2 C_7 - A_1^2 L_2 C_8 + 2(L_2 - L_1)A_2^2 C_9 + 2(L_2 - L_1)A_2^2 C_{10}]\dot{\theta}_s \\
&\quad + (A_1^2 C_1 + A_2^2 C_2 + 2A_2^2 C_{10})\dot{z}_{f1} + (A_1^2 C_5 + A_2^2 C_6 + 2A_2^2 C_{10})\dot{z}_{f2} \\
&\quad + (A_2^2 C_3 + A_1^2 C_4 + 2A_2^2 C_9)\dot{z}_{f3} + (A_2^2 C_7 + A_1^2 C_8 + 2A_2^2 C_9)\dot{z}_{f4} + \dots
\end{aligned} \tag{5.25}$$

Considering a pure roll movement, the roll moment M_ϕ can be written as

$$\begin{aligned}
M_\phi &= (F_1 + F_2 - F_3 - F_4)l \\
&= l(-A_1^2 C_1 - A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 - A_1^2 C_5 - A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8 \\
&\quad + 4A_2^2 C_9 - 4A_2^2 C_{10})\dot{z}_s \\
&\quad - l^2(A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8 \\
&\quad + 4A_2^2 C_9 + 4A_2^2 C_{10})\dot{\phi}_s \\
&\quad + l[A_1^2 L_1 C_1 + A_2^2 L_1 C_2 - A_2^2 L_1 C_3 - A_1^2 L_1 C_4 - A_1^2 L_2 C_5 - A_2^2 L_2 C_6 \\
&\quad + A_2^2 L_2 C_7 + A_1^2 L_2 C_8 + 2(L_2 - L_1)A_2^2 C_9 + 2(L_1 - L_2)A_2^2 C_{10}]\dot{\theta}_s \\
&\quad + l(A_1^2 C_1 + A_2^2 C_2 + 2A_2^2 C_{10})\dot{z}_{f1} + l(A_1^2 C_5 + A_2^2 C_6 + 2A_2^2 C_{10})\dot{z}_{f2} \\
&\quad - l(A_2^2 C_3 + A_1^2 C_4 + 2A_2^2 C_9)\dot{z}_{f3} - l(A_2^2 C_7 + A_1^2 C_8 + 2A_2^2 C_9)\dot{z}_{f4} + \dots
\end{aligned} \tag{5.26}$$

Considering a pure pitch movement, the corresponding pitch moment M_θ can be written as

$$\begin{aligned}
M_\theta &= -F_1 L_1 + F_2 L_2 - F_3 L_1 + F_4 L_2 \\
&= [-A_1^2 L_1^2 C_1 - A_2^2 L_1^2 C_2 - A_2^2 L_1^2 C_3 - A_1^2 L_1^2 C_4 - A_1^2 L_2^2 C_5 - A_2^2 L_2^2 C_6 \\
&\quad - A_2^2 L_2^2 C_7 - A_1^2 L_2^2 C_8 + (2A_2^2 L_1 L_2 - A_2^2 L_1^2 - A_2^2 L_2^2) C_9 \\
&\quad + (2A_2^2 L_1 L_2 - A_2^2 L_1^2 - A_2^2 L_2^2) C_{10}] \dot{\theta}_s \\
&\quad - [A_1^2 C_1 L_1 + A_2^2 C_2 L_1 + A_2^2 C_{10} (L_1 - L_2)] \dot{z}_{f1} \\
&\quad + [A_1^2 C_5 L_2 + A_2^2 C_6 L_2 + A_2^2 C_{10} (L_2 - L_1)] \dot{z}_{f2} \\
&\quad - [A_2^2 C_3 L_1 + A_1^2 C_4 L_1 + A_2^2 C_9 (L_1 - L_2)] \dot{z}_{f3} \\
&\quad + [A_2^2 C_7 L_2 + A_1^2 C_8 L_2 + A_2^2 C_9 (L_2 - L_1)] \dot{z}_{f4} + \dots
\end{aligned} \tag{5.27}$$

As the result, the equivalent vertical damping coefficient c_z is

$$\begin{aligned}
c_z &= A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8 + 4A_2^2 C_9 + 4A_2^2 C_{10} \\
&= 4(A_1^2 + A_2^2) C_{hose} + 4A_2^2 C_9 + 4A_2^2 C_{10}
\end{aligned} \tag{5.28}$$

Similarly, the equivalent roll damping coefficient c_φ is

$$\begin{aligned}
c_\varphi &= l[A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8 + 4A_2^2 C_9 \\
&\quad + 4A_2^2 C_{10}] = 4(A_1^2 + A_2^2) l C_{hose} + 4A_2^2 l C_9 + 4A_2^2 l C_{10}
\end{aligned} \tag{5.29}$$

And the equivalent pitch damping coefficient c_θ is

$$\begin{aligned}
c_\theta &= A_1^2 L_1^2 C_1 + A_2^2 L_1^2 C_2 + A_2^2 L_1^2 C_3 + A_1^2 L_1^2 C_4 + A_1^2 L_2^2 C_5 + A_2^2 L_2^2 C_6 + A_2^2 L_2^2 C_7 + A_1^2 L_2^2 C_8 \\
&\quad + A_2^2 (L_1 - L_2)^2 C_9 + A_2^2 (L_1 - L_2)^2 C_{10} \\
&= (A_1^2 L_1^2 + A_1^2 L_1^2 + 2A_1^2 L_2^2 + A_2^2 L_1^2 + A_2^2 L_1^2 + 2A_2^2 L_2^2) C_{hose} \\
&\quad + A_2^2 (L_1 - L_2)^2 C_9 + A_2^2 (L_1 - L_2)^2 C_{10}
\end{aligned} \tag{5.30}$$

It can be seen that the suspension vertical and roll damping properties are determined by the two flow control valves besides the accumulators. By controlling the valves' fluid resistance characteristics, the suspension damping properties in vertical and roll directions can be controlled in real-time according to road conditions or driver preferences.

5.1.6 Damping control system for AIS-ARS

The damping control system of the proposed AIS-ARS consists of six main components: one microcontroller, one 6-axis IMU, four height sensors, four pressure sensors, two electro-proportional

flow control valves, and one HPU. As shown in Figure 5.3, the measurements of available sensors are collected and sent the data to the state observer. The estimated vehicle motions are forwarded to the Skyhook controller introduced in Section 3.2.2.1. Both algorithms are compiled in the microcontroller in practice. The resulting control commands are applied to the electro-proportional flow control valves, usually in current or voltage. This way, the suspension damping characteristics can be controlled in real-time to isolate the undesired vibrations.

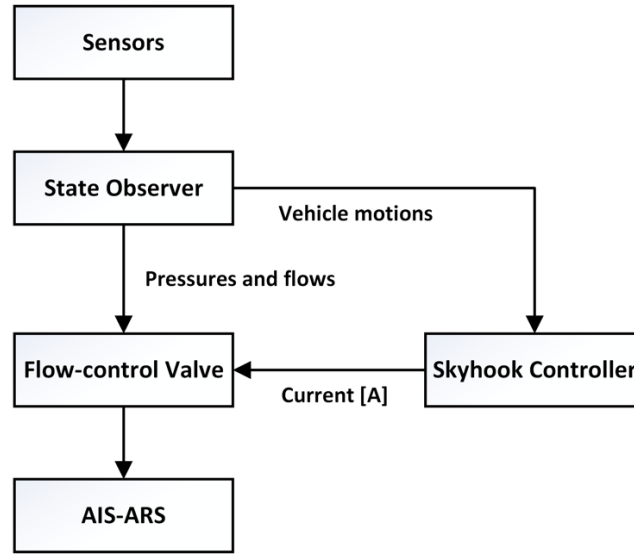


Figure 5.3. Damping control system for AIS-ARS

5.2 AIS-ARPS

The above roll-plane interconnected suspension can provide enhanced anti-roll performance and ride quality. However, the pitch motion of the truck cabin is usually harsh and uncomfortable due to its high CG location. Thus, developing a roll and pitch coupled interconnected suspension system with anti-roll and anti-pitch properties is necessary.

5.2.1 Topology of the AIS-ARPS

A novel interconnected topology is introduced in this section, as shown in Figure 5.4. In the first hydraulic circuit, the upper chamber of the front left cylinder is connected to the lower chamber of the rear right cylinder. In the second hydraulic circuit, the lower chamber of the front left cylinder is connected to the upper chamber of the rear right cylinder. In the third hydraulic circuit, the upper chamber of the front right cylinder is connected to the lower chamber of the rear left cylinder. In the

fourth hydraulic circuit, the lower chamber of the front right cylinder is connected to the upper chamber of the rear left cylinder. Four accumulators are assembled along the four hydraulic circuits to generate enough roll and pitch stiffness. In addition, this topology utilizes eight flow control valves to achieve real-time damping control.

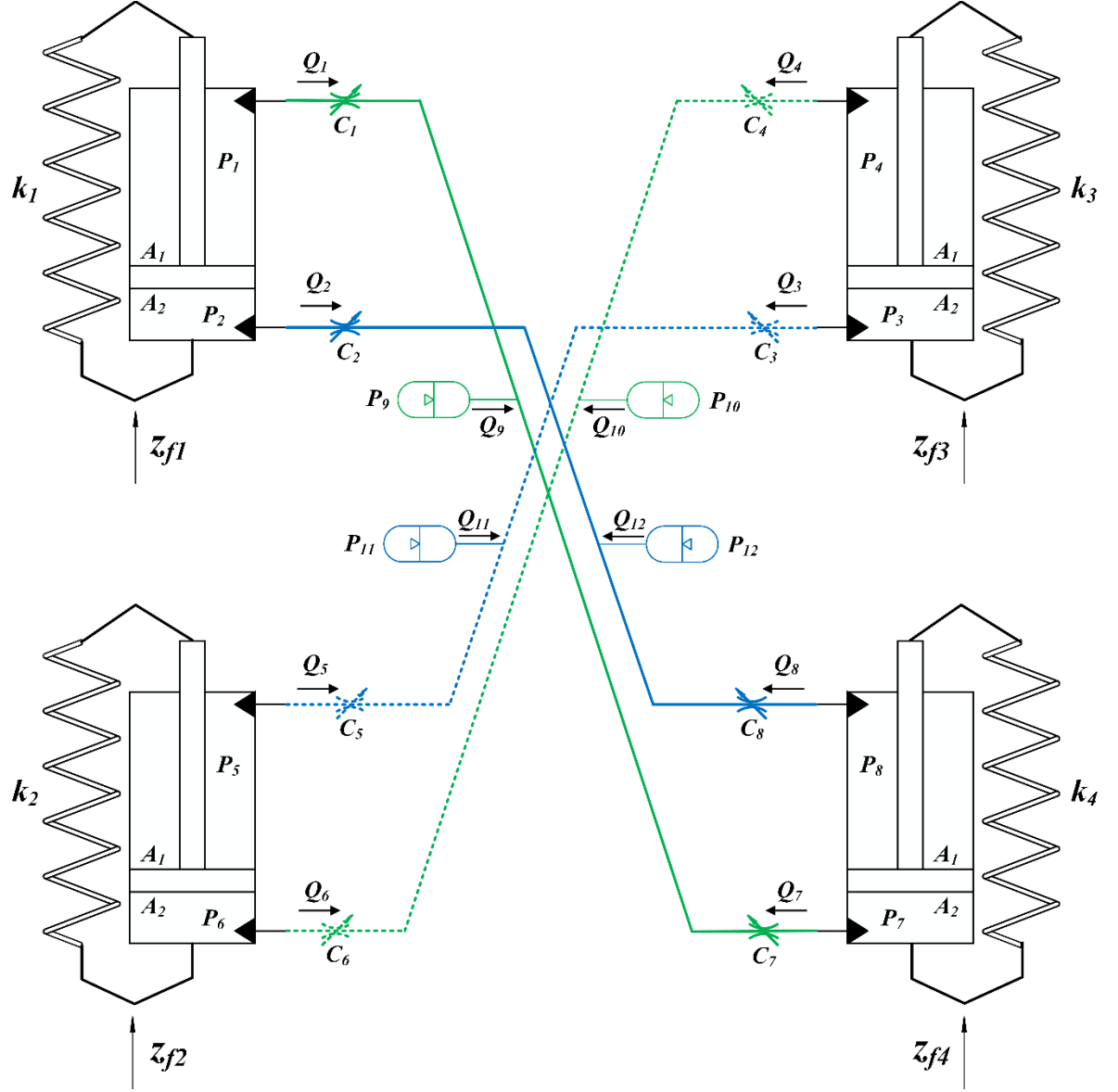


Figure 5.4. Schematic of the roll and pitch coupled interconnected suspension

Firstly, the chamber volumes of hydraulic cylinders can be calculated based on the suspension relative displacement. Assuming the strokes of the hydraulic cylinders at the nominal position are S_1 and S_2 , the eight chamber volumes can be represented by

$$\begin{aligned}
V_1 &= -A_1\Delta z_1 + A_1S_1 \\
V_2 &= A_2\Delta z_1 + A_2S_2 \\
V_3 &= A_2\Delta z_3 + A_2S_2 \\
V_4 &= -A_1\Delta z_3 + A_1S_1 \\
V_5 &= -A_1\Delta z_2 + A_1S_1 \\
V_6 &= A_2\Delta z_2 + A_2S_2 \\
V_7 &= A_2\Delta z_4 + A_2S_2 \\
V_8 &= -A_1\Delta z_4 + A_1S_1
\end{aligned} \tag{5.31}$$

in which the suspension relative displacements at the four corners are

$$\begin{aligned}
\Delta z_1 &= z_s + l\varphi_s - L_1\theta_s - z_{f1} \\
\Delta z_2 &= z_s + l\varphi_s + L_2\theta_s - z_{f2} \\
\Delta z_3 &= z_s - l\varphi_s - L_1\theta_s - z_{f3} \\
\Delta z_4 &= z_s - l\varphi_s + L_2\theta_s - z_{f4}
\end{aligned} \tag{5.32}$$

in which l represents half of the distance between the left and right hydraulic cylinders, L_1 represents the longitudinal distance from cabin CG location to the front cabin suspension, and L_2 represents the longitudinal distance from cabin CG location to the rear cabin suspension.

Secondly, the time derivative of oil pressures in hydraulic cylinders can be calculated according to equations (5.1) and (5.2).

$$\begin{aligned}
\dot{P}_1 &= \frac{\beta}{v_1} (A_1\Delta \dot{z}_1 - Q_1), \text{ in which } Q_1 = \frac{1}{c_1} (P_1 - P_9) \\
\dot{P}_2 &= \frac{\beta}{v_2} (-A_2\Delta \dot{z}_1 - Q_2), \text{ in which } Q_2 = \frac{1}{c_2} (P_2 - P_{12})
\end{aligned} \tag{5.33}$$

$$\dot{P}_3 = \frac{\beta}{V_3} (-A_2 \Delta \dot{z}_3 - Q_3), \text{ in which } Q_3 = \frac{1}{C_3} (P_3 - P_{11})$$

$$\dot{P}_4 = \frac{\beta}{V_4} (A_1 \Delta \dot{z}_3 - Q_4), \text{ in which } Q_4 = \frac{1}{C_4} (P_4 - P_{10})$$

$$\dot{P}_5 = \frac{\beta}{V_5} (A_1 \Delta \dot{z}_2 - Q_5), \text{ in which } Q_5 = \frac{1}{C_5} (P_5 - P_{11})$$

$$\dot{P}_6 = \frac{\beta}{V_6} (-A_2 \Delta \dot{z}_2 - Q_6), \text{ in which } Q_6 = \frac{1}{C_6} (P_6 - P_{10})$$

$$\dot{P}_7 = \frac{\beta}{V_7} (-A_2 \Delta \dot{z}_4 - Q_7), \text{ in which } Q_7 = \frac{1}{C_7} (P_7 - P_9)$$

$$\dot{P}_8 = \frac{\beta}{V_8} (A_1 \Delta \dot{z}_4 - Q_8), \text{ in which } Q_8 = \frac{1}{C_8} (P_8 - P_{12})$$

in which $C_1 \sim C_8$ represent the equivalent flow resistance parameters of the hydraulic hoses and flow control valves.

Thirdly, based on the nominal gas volume, the gas volumes in the four accumulators can be iteratively calculated as

$$\begin{aligned} V_9 &= V_0 - \int (Q_1 + Q_7) dt \\ V_{10} &= V_0 - \int (Q_4 + Q_6) dt \\ V_{11} &= V_0 - \int (Q_3 + Q_5) dt \\ V_{12} &= V_0 - \int (Q_2 + Q_8) dt \end{aligned} \tag{5.34}$$

Fourthly, the time derivative of the gas pressures in the accumulators can be calculated according to equations (5.1) and (5.3).

$$\begin{aligned} \dot{P}_9 &= -\frac{n P_0 V_0^n}{V_9^{n+1}} \dot{V}_9 = \frac{n P_0 V_0^n}{V_9^{n+1}} (Q_1 + Q_7) \\ \dot{P}_{10} &= -\frac{n P_0 V_0^n}{V_{10}^{n+1}} \dot{V}_{10} = \frac{n P_0 V_0^n}{V_{10}^{n+1}} (Q_4 + Q_6) \end{aligned} \tag{5.35}$$

$$\dot{P}_{11} = -\frac{nP_0V_0^n}{V_{11}^{n+1}}\dot{V}_{11} = \frac{nP_0V_0^n}{V_{11}^{n+1}}(Q_3 + Q_5)$$

$$\dot{P}_{12} = -\frac{nP_0V_0^n}{V_{12}^{n+1}}\dot{V}_{12} = \frac{nP_0V_0^n}{V_{12}^{n+1}}(Q_2 + Q_8)$$

Finally, the pressures in hydraulic cylinders $P_1 \sim P_8$ and accumulators $P_9 \sim P_{12}$ can be found based on their time derivatives $\dot{P}_i$ and the system initial pressure P_0 .

5.2.2 Roll stiffness analysis

At the suspension steady status, its roll moment can be written as

$$\begin{aligned} M_\theta &= -(k_1 + k_2 + k_3 + k_4)l^2\varphi_s + (F_1 + F_2 - F_3 - F_4)l \\ &= -(k_1 + k_2 + k_3 + k_4)l^2\varphi_s - A_1lP_1 + A_2lP_2 - A_2lP_3 + A_1lP_4 \\ &\quad - A_1lP_5 + A_2lP_6 - A_2lP_7 + A_1lP_8 \\ &= -(k_1 + k_2 + k_3 + k_4)l^2\varphi_s - l(A_1 + A_2)(P_9 - P_{10} + P_{11} - P_{12}) \end{aligned} \quad (5.36)$$

The roll stiffness can be derived from the restoring roll moment subject to the relative roll deflection as

$$K_\varphi = -\frac{dM}{d\varphi} = (k_1 + k_2 + k_3 + k_4)l^2 + l(A_1 + A_2)\frac{d}{d\varphi}(P_9 - P_{10} + P_{11} - P_{12}) \quad (5.37)$$

The corresponding derivative of the gas pressures in the accumulators can be calculated as

$$\begin{aligned} \frac{dP_9}{d\varphi} &= \frac{n(A_1 + A_2)lP_0V_0^n}{V_9^{n+1}} = \frac{n(A_1 + A_2)lP_0V_0^n}{\gamma_5} \\ \frac{dP_{10}}{d\varphi} &= \frac{-n(A_1 + A_2)lP_0V_0^n}{V_{10}^{n+1}} = \frac{-n(A_1 + A_2)lP_0V_0^n}{\gamma_6} \\ \frac{dP_{11}}{d\varphi} &= \frac{n(A_1 + A_2)lP_0V_0^n}{V_{11}^{n+1}} = \frac{n(A_1 + A_2)lP_0V_0^n}{\gamma_7} \\ \frac{dP_{12}}{d\varphi} &= \frac{-n(A_1 + A_2)lP_0V_0^n}{V_{12}^{n+1}} = \frac{-n(A_1 + A_2)lP_0V_0^n}{\gamma_8} \end{aligned} \quad (5.38)$$

in which

$$\gamma_5 = [V_0 + (A_2 - A_1)z_s - (A_1 + A_2)l\varphi_s + (A_1L_1 + A_2L_2)\theta_s]^{n+1}$$

$$\gamma_6 = [V_0 + (A_2 - A_1)z_s + (A_1 + A_2)l\varphi_s + (A_1L_1 + A_2L_2)\theta_s]^{n+1}$$

$$\gamma_7 = [V_0 + (A_2 - A_1)z_s - (A_1 + A_2)l\varphi_s - (A_1L_2 + A_2L_1)\theta_s]^{n+1}$$

$$\gamma_8 = [V_0 + (A_2 - A_1)z_s + (A_1 + A_2)l\varphi_s - (A_1L_2 + A_2L_1)\theta_s]^{n+1}$$

Hence, the roll stiffness of the proposed interconnected suspension can be represented as

$$K_\varphi = (k_1 + k_2 + k_3 + k_4)l^2 + nP_0V_0^n(A_1 + A_2)^2l^2 \left[\frac{1}{\gamma_5} + \frac{1}{\gamma_6} + \frac{1}{\gamma_7} + \frac{1}{\gamma_8} \right] \quad (5.39)$$

When the Polytropic coefficient $n = 1$, the roll stiffness formulation can be simplified as

$$K_\varphi = (k_1 + k_2 + k_3 + k_4)l^2 + 4(A_1 + A_2)^2l^2 \frac{P_0}{V_0} \quad (5.40)$$

Based on Eq.(5.39) and the parameters listed in Table 5.2, the estimated roll stiffness of the roll and pitch coupled AIS-ARPS system is shown in Figure 5.5. Specifically, the estimated results of the four example settings are plotted in different colors, i.e., $V_0 = 1.63L$ & $P_0 = 245psi$, $V_0 = 1.6L$ & $P_0 = 250psi$, $V_0 = 1.57L$ & $P_0 = 255psi$, and $V_0 = 1.54L$ & $P_0 = 260psi$. Noticeably, the roll stiffness can be adjusted by charging or discharging the hydraulic system. In other words, the roll stiffness K_φ increases with the increase of the system nominal pressure P_0 . Attributed to the air compression in the accumulator, the roll stiffness tends to increase with the increase of the roll angle. Thus, a lower roll stiffness can be realized when the suspension relative roll angle is small, while a higher roll stiffness can be achieved when the angle increases.

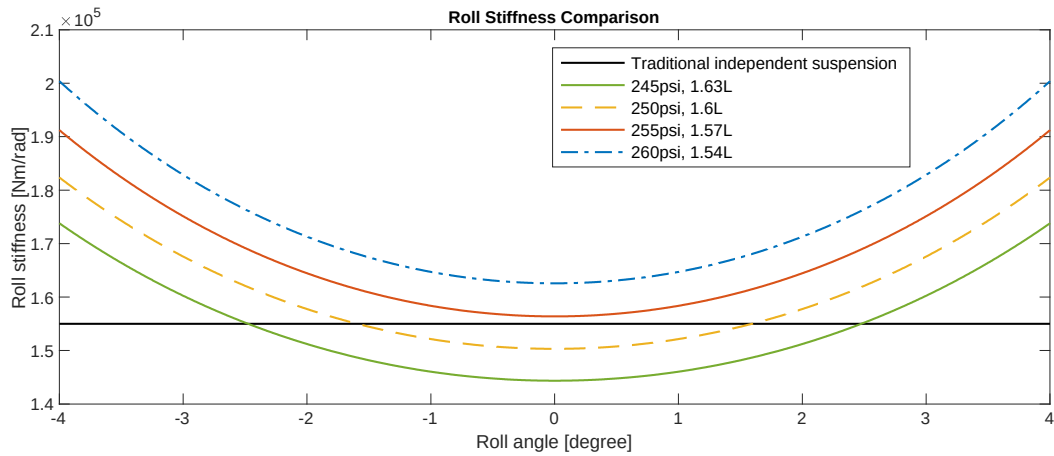


Figure 5.5. Comparison of the dual-axle suspension roll stiffness

Table 5.2. Parameters of the AIS-ARPS system

Symbol	Description	Value
k_1, k_3	Front air spring stiffness	23658 N/m
k_2, k_4	Rear air spring stiffness	14854 N/m
m_s	Cabin weight	1250 kg
k_ϕ	Equivalent roll stiffness of the bushings	155 kN·m/rad
I_x	Cabin roll moment of inertia	1468 kgm ²
I_y	Cabin pitch moment of inertia	1229 kgm ²
l	Lateral distance from damper to CG	0.63 m
L_1	Distance from CG to front suspension	0.936 m
L_2	Distance from CG to rear suspension	1.055 m
A_1	Rod side chamber cross-sectional area (rod diameter is 1.5 inches)	0.00342 m ²
A_2	Bore side chamber cross-sectional area (bore diameter is 3.5 inches)	0.00456 m ²
S_1, S_2	Half of the full stroke length	4 inches
n	Polytropic coefficient	1.38
β	Fluid bulk modulus	1400 MPa
C_i	Resistance coefficient of hydraulic hose ($i = 1, 2, 3, \dots, 8$)	5×10^7 Pa·s/m ³

5.2.3 Pitch stiffness analysis

Suspension pitch stiffness is a measure of the tilting resistance about its transverse axis. It determines how much the cabin will tilt forward or backward in response to longitudinal acceleration or road disturbance. A higher pitch stiffness results in less body pitch, which leads to better stability. In the conventional suspension, the pitch stiffness is mainly provided by springs, while it is hard to be independently adjusted since the springs affect many other suspension design factors. Using the proposed AIS-ARPS, an additional amount of pitch stiffness is added to the suspension system. Meanwhile, the rotational stiffness can be easily adjusted for different vehicle models or applications, which is analyzed in the following section.

The pitch moment can be written as

$$\begin{aligned}
M_\theta &= (k_1L_1 - k_2L_2 + k_3L_1 - k_4L_2)z_s - (k_1L_1^2 + k_2L_2^2 + k_3L_1^2 + k_4L_2^2)\theta_s \\
&\quad + (-L_1F_1 + L_2F_2 - L_1F_3 + L_2F_4) \\
&= (k_1L_1 - k_2L_2 + k_3L_1 - k_4L_2)z_s - (k_1L_1^2 + k_2L_2^2 + k_3L_1^2 + k_4L_2^2)\theta_s \\
&\quad - L_1(A_2P_2 - A_1P_1) - L_1(A_2P_3 - A_1P_4) + L_2(A_2P_6 - A_1P_5) \\
&\quad + L_2(A_2P_7 - A_1P_8) \\
&= (k_1L_1 - k_2L_2 + k_3L_1 - k_4L_2)z_s - (k_1L_1^2 + k_2L_2^2 + k_3L_1^2 + k_4L_2^2)\theta_s \\
&\quad + (A_1L_1 + A_2L_2)(P_9 + P_{10}) - (A_1L_2 + A_2L_1)(P_{11} + P_{12})
\end{aligned} \tag{5.41}$$

The pitch stiffness can be derived from the restoring pitch moment subject to the relative pitch deflection as

$$\begin{aligned}
K_\theta &= -\frac{dM_\theta}{d\theta_s} = k_1L_1^2 + k_2L_2^2 + k_3L_1^2 + k_4L_2^2 - (A_1L_1 + A_2L_2)\frac{d}{d\theta_s}(P_9 + P_{10}) \\
&\quad + (A_1L_2 + A_2L_1)\frac{d}{d\theta_s}(P_{11} + P_{12})
\end{aligned} \tag{5.42}$$

The corresponding derivative of the gas pressures in the accumulators can be expressed as

$$\begin{aligned}
\frac{dP_9}{d\theta_s} &= \frac{-n(A_2L_2 - A_1L_1)P_0V_0^n}{V_9^{n+1}} = \frac{-n(A_1L_1 + A_2L_2)P_0V_0^n}{\gamma_5} \\
\frac{dP_{10}}{d\theta_s} &= \frac{-n(A_2L_2 - A_1L_1)P_0V_0^n}{V_{10}^{n+1}} = \frac{-n(A_1L_1 + A_2L_2)P_0V_0^n}{\gamma_6} \\
\frac{dP_{11}}{d\theta_s} &= \frac{-n(A_2L_1 - A_1L_2)P_0V_0^n}{V_{11}^{n+1}} = \frac{n(A_1L_2 + A_2L_1)P_0V_0^n}{\gamma_7} \\
\frac{dP_{12}}{d\theta_s} &= \frac{-n(A_2L_1 - A_1L_2)P_0V_0^n}{V_{12}^{n+1}} = \frac{n(A_1L_2 + A_2L_1)P_0V_0^n}{\gamma_8}
\end{aligned} \tag{5.43}$$

Hence, the pitch stiffness of the proposed interconnected suspension can be represented as

$$\begin{aligned}
K_\theta &= k_1L_1^2 + k_2L_2^2 + k_3L_1^2 + k_4L_2^2 + nP_0V_0^n(A_1L_1 + A_2L_2)^2 \left[\frac{1}{\gamma_5} + \frac{1}{\gamma_6} \right] \\
&\quad + nP_0V_0^n(A_1L_2 + A_2L_1)^2 \left[\frac{1}{\gamma_7} + \frac{1}{\gamma_8} \right]
\end{aligned} \tag{5.44}$$

When the Polytropic coefficient $n = 1$, the pitch stiffness formulation can be simplified as

$$K_{\theta} = k_1 L_1^2 + k_2 L_2^2 + k_3 L_1^2 + k_4 L_2^2 + 2[(A_1 L_1 + A_2 L_2)^2 + (A_1 L_2 + A_2 L_1)^2] \frac{P_0}{V_0} \quad (5.45)$$

As shown in Figure 5.6, the estimated results of the four settings are plotted in different colors, i.e., $V_0 = 1.63L$ & $P_0 = 245psi$, $V_0 = 1.6L$ & $P_0 = 250psi$, $V_0 = 1.57L$ & $P_0 = 255psi$, and $V_0 = 1.54L$ & $P_0 = 260psi$. The pitch stiffness changes with the charging or discharging process of the hydraulic system. In other words, the pitch stiffness K_{θ} increases with the increase of the system nominal pressure P_0 . Attributed to the air compression in the accumulators, the pitch stiffness tends to increase with the increase of the pitch angle. Compared to the conventional suspension, the proposed AIS-ARPS provides additional pitch stiffness to prevent harsh pitch motions. Besides, it is noticeable that the pitch stiffness is not as sensitive as the roll stiffness when the system's nominal pressure changes.

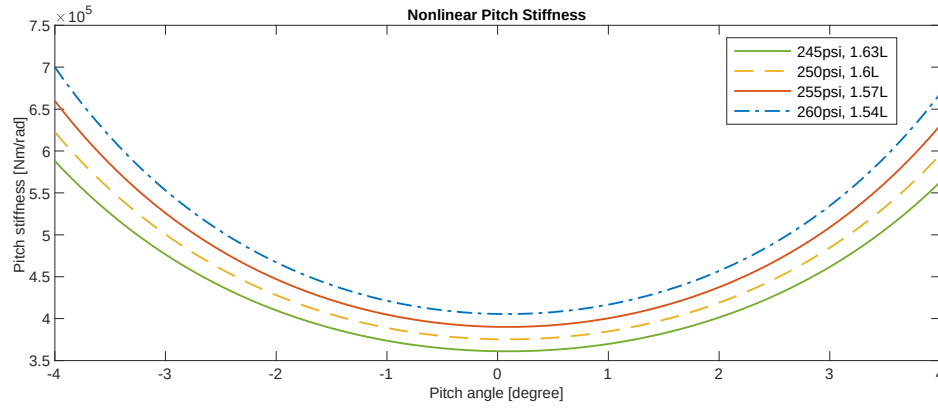


Figure 5.6. Nonlinear pitch stiffness

5.2.4 Damping characteristics

The following derivation method is the same as the one introduced in Section 5.1.5. In detail, the forces generated by each hydraulic cylinder can be written as

$$\begin{aligned} F_1 &= A_2 P_2 - A_1 P_1 = A_2 C_2 Q_2 + A_2 P_{12} - A_1 C_1 Q_1 - A_1 P_9 \\ F_2 &= A_2 P_6 - A_1 P_5 = A_2 C_6 Q_6 + A_2 P_{10} - A_1 C_5 Q_5 - A_1 P_{11} \\ F_3 &= A_2 P_3 - A_1 P_4 = A_2 C_3 Q_3 + A_2 P_{11} - A_1 C_4 Q_4 - A_1 P_{10} \\ F_4 &= A_2 P_7 - A_1 P_8 = A_2 C_7 Q_7 + A_2 P_9 - A_1 C_8 Q_8 - A_1 P_{12} \end{aligned} \quad (5.46)$$

in which the flow rates along the hydraulic circuits can be calculated as follows:

$$\begin{aligned}
Q_1 &= A_1 \Delta \dot{z}_1 \\
Q_2 &= -A_2 \Delta \dot{z}_1 \\
Q_3 &= -A_2 \Delta \dot{z}_3 \\
Q_4 &= A_1 \Delta \dot{z}_3 \\
Q_5 &= A_1 \Delta \dot{z}_2 \\
Q_6 &= -A_2 \Delta \dot{z}_2 \\
Q_7 &= -A_2 \Delta \dot{z}_4 \\
Q_8 &= A_1 \Delta \dot{z}_4 \\
Q_9 &= -Q_1 - Q_3 - Q_5 - Q_7 \\
Q_{10} &= -Q_2 - Q_4 - Q_6 - Q_8
\end{aligned} \tag{5.47}$$

Combining equations (5.32), (5.46), and (5.47), the cylinder forces can be simplified as

$$\begin{aligned}
F_1 &= -(A_1^2 C_1 + A_2^2 C_2) \dot{z}_s - l(A_1^2 C_1 + A_2^2 C_2) \dot{\phi}_s + L_1(A_1^2 C_1 + A_2^2 C_2) \dot{\theta}_s \\
&\quad + (A_1^2 C_1 + A_2^2 C_2) \dot{z}_{f1} - A_1 P_9 + A_2 P_{12} \\
F_2 &= -(A_1^2 C_5 + A_2^2 C_6) \dot{z}_s - l(A_1^2 C_5 + A_2^2 C_6) \dot{\phi}_s - L_2(A_1^2 C_5 + A_2^2 C_6) \dot{\theta}_s \\
&\quad + (A_1^2 C_5 + A_2^2 C_6) \dot{z}_{f2} - A_1 P_{11} + A_2 P_{10} \\
F_3 &= -(A_2^2 C_3 + A_1^2 C_4) \dot{z}_s + l(A_2^2 C_3 + A_1^2 C_4) \dot{\phi}_s + L_1(A_2^2 C_3 + A_1^2 C_4) \dot{\theta}_s \\
&\quad + (A_2^2 C_3 + A_1^2 C_4) \dot{z}_{f3} - A_1 P_{10} + A_2 P_{11} \\
F_4 &= -(A_2^2 C_7 + A_1^2 C_8) \dot{z}_s + l(A_2^2 C_7 + A_1^2 C_8) \dot{\phi}_s - L_2(A_2^2 C_7 + A_1^2 C_8 L_2) \dot{\theta}_s \\
&\quad + (A_2^2 C_7 + A_1^2 C_8) \dot{z}_{f4} - A_1 P_{12} + A_2 P_9
\end{aligned} \tag{5.48}$$

Thus, the total vertical force generated by the four hydraulic cylinders can be written as

$$\begin{aligned}
F_z &= F_1 + F_2 + F_3 + F_4 \\
&= -(A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8) \dot{z}_s \\
&\quad - l(A_1^2 C_1 + A_2^2 C_2 - A_2^2 C_3 - A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 - A_2^2 C_7 - A_1^2 C_8) \dot{\phi}_s \\
&\quad + (A_1^2 L_1 C_1 + A_2^2 L_1 C_2 + A_2^2 L_1 C_3 + A_1^2 L_2 C_4 - A_1^2 L_2 C_5 - A_2^2 L_2 C_6 \\
&\quad - A_2^2 L_2 C_7 - A_1^2 L_2 C_8) \dot{\theta}_s + (A_1^2 C_1 + A_2^2 C_2) \dot{z}_{f1} + (A_1^2 C_5 + A_2^2 C_6) \dot{z}_{f2} \\
&\quad + (A_2^2 C_3 + A_1^2 C_4) \dot{z}_{f3} + (A_2^2 C_7 + A_1^2 C_8) \dot{z}_{f4} \\
&\quad + (A_2 - A_1)(P_9 + P_{10} + P_{11} + P_{12})
\end{aligned} \tag{5.49}$$

Considering a pure roll movement, the roll moment M_ϕ can be written as

$$\begin{aligned}
M_\phi &= (F_1 + F_2 - F_3 - F_4)l \\
&= l(-A_1^2 C_1 - A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 - A_1^2 C_5 - A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8) \dot{z}_s \\
&\quad - l^2(A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8) \dot{\phi}_s \\
&\quad + l(A_1^2 L_1 C_1 + A_2^2 L_1 C_2 - A_2^2 L_1 C_3 - A_1^2 L_1 C_4 - A_1^2 L_2 C_5 - A_2^2 L_2 C_6 \\
&\quad + A_2^2 L_2 C_7 + A_1^2 L_2 C_8) \dot{\theta}_s + l(A_1^2 C_1 + A_2^2 C_2) \dot{z}_{f1} + l(A_1^2 C_5 + A_2^2 C_6) \dot{z}_{f2} \\
&\quad - l(A_2^2 C_3 + A_1^2 C_4) \dot{z}_{f3} - l(A_2^2 C_7 + A_1^2 C_8) \dot{z}_{f4} \\
&\quad + l(A_1 + A_2)(-P_9 + P_{10} - P_{11} + P_{12})
\end{aligned} \tag{5.50}$$

Considering a pure pitch movement, the corresponding pitch moment M_θ can be written as

$$\begin{aligned}
M_\theta &= -F_1 L_1 + F_2 L_2 - F_3 L_1 + F_4 L_2 \\
&= -[L_1^2(A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4) \\
&\quad + L_2^2(A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8)] \dot{\theta}_s - L_1(A_1^2 C_1 + A_2^2 C_2) \dot{z}_{f1} \\
&\quad + L_2(A_1^2 C_5 + A_2^2 C_6) \dot{z}_{f2} - L_1(A_2^2 C_3 + A_1^2 C_4) \dot{z}_{f3} + L_2(A_2^2 C_7 + A_1^2 C_8) \dot{z}_{f4} \\
&\quad + (L_1 A_1 + L_2 A_2) P_9 + (L_1 A_1 + L_2 A_2) P_{10} - (L_2 A_1 + L_1 A_2) P_{11} \\
&\quad - (L_2 A_1 + L_1 A_2) P_{12} + \dots
\end{aligned} \tag{5.51}$$

As the result, the equivalent vertical damping coefficient c_z is

$$c_z = A_1^2 C_1 + A_2^2 C_2 + A_2^2 C_3 + A_1^2 C_4 + A_1^2 C_5 + A_2^2 C_6 + A_2^2 C_7 + A_1^2 C_8 \tag{5.52}$$

Similarly, the equivalent roll damping coefficient c_ϕ is

$$c_\varphi = l^2(A_1^2C_1 + A_2^2C_2 + A_2^2C_3 + A_1^2C_4 + A_1^2C_5 + A_2^2C_6 + A_2^2C_7 + A_1^2C_8) \quad (5.53)$$

And the equivalent pitch damping coefficient c_θ is

$$c_\theta = L_1^2(A_1^2C_1 + A_2^2C_2 + A_2^2C_3 + A_1^2C_4) + L_2^2(A_1^2C_5 + A_2^2C_6 + A_2^2C_7 + A_1^2C_8) \quad (5.54)$$

It can be seen that the vertical, roll, and pitch damping characteristics are determined by the eight flow control valves at the cylinder inlets. With electro-proportional flow control valves, these damping characteristics can be controlled in real time by adjusting the valves' fluid resistance characteristics according to road conditions or driver preferences. Similar to conventional suspension systems, the vertical, roll, and pitch damping characteristics are coupled, which need to be fine-tuned to achieve the optimum overall performance.

5.3 Summary

This chapter introduces two novel hydro-pneumatic interconnected suspension systems with adjustable rotational stiffness. Their topologies, stiffness characteristics, damping properties, and control techniques were thoroughly studied. Compared to the existing designs in the literature, the proposed interconnected suspension systems have the following advantages:

- **Enhanced roll and pitch stability:** The proposed interconnected topologies have nonlinear rotational stiffness characteristics, i.e., the rotational stiffness increases with the increase of the suspension relative rotational angle. Hence, it can enhance the vehicle's roll and/or pitch stability without compromising ride comfort.
- **Adjustable rotational stiffness:** A method of adjusting suspension rotational stiffness was developed, by which the suspension rotational characteristics can be conveniently adjusted according to the driving scenario, road condition, vehicle load, etc. In detail, a compact HPU was utilized to charge or discharge the hydraulic system, which makes the system easy to be operated.
- **Feasibility for various applications:** The proposed interconnected suspension systems can be easily tuned and fitted to different car models. Various rotational stiffness levels can be achieved by adjusting the system's nominal pressure, resulting in one topology for any vehicle suspension. Besides, the proposed topologies can be easily extended to multi-axle suspension systems.

- Additional vertical stiffness: The area difference between the upper and lower chambers of the hydraulic cylinder results in an upward vertical force. Thus, the original air springs can be replaced by smaller ones to further decrease the manufacturing costs and the installation space.

Chapter 6

Experimental and Simulation Studies of Adaptive Interconnected Suspension Systems with Adjustable Rotational Stiffness

This chapter presents a detailed evaluation of the two novel interconnected suspension systems, AIS-ARS and AIS-ARPS. The simulation and experimental results comprehensively evaluate the suspension systems and their corresponding mathematical models. The prototype of the AIS-ARS was assembled using the available off-the-shelf products, which validates the feasibility and practicality of the proposed topologies for future applications.

6.1 Simulation Results

The development of a high-fidelity truck model in ADAMS/Car was a crucial aspect of this study, which served as a valuable tool for evaluating the performance of the adaptive interconnected suspension systems. The model shown in Figure 6.1(a) was designed to accurately represent the truck's various components and systems, including the suspension, chassis, steering, etc. The co-simulation between MATLAB/Simulink and ADAMS/Car enables the evaluation of various control strategies for the adaptive interconnected suspension systems, as shown in Figure 6.1(b). The co-simulation environment ensured that the simulated results were reliable and could be used to gain insights into the system's behavior under different operating conditions, such as steady turn, articulation test, speedbumps, double-lane-change, paved road test, and emergency braking. The vehicle parameters utilized in this section were chosen based on the available off-the-shelf hydraulic components. Note that the fully loaded truck specifications are listed in Table 5.1, and the cabin parameters are presented in Table 5.2.

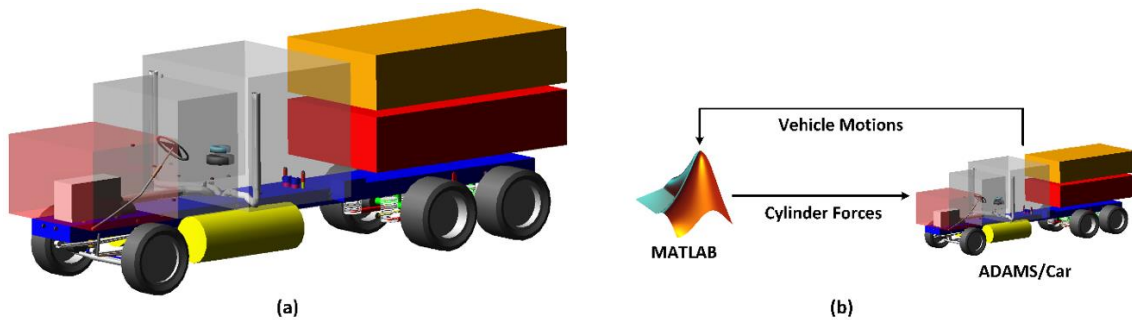


Figure 6.1. Fully loaded truck built in ADAMS/Car.

6.1.1 Heavy truck: steady turn at 45 km/h

In this scenario, the truck was equipped with the passive IS-ARS. The vehicle turned at a constant speed of 45 km/h on an even road and was about to roll over. Two stiffness levels were simulated and compared to the conventional air suspension, i.e., $V_0 = 2.4L$ & $P_0 = 1730psi$ and $V_0 = 2.8L$ & $P_0 = 1483psi$.

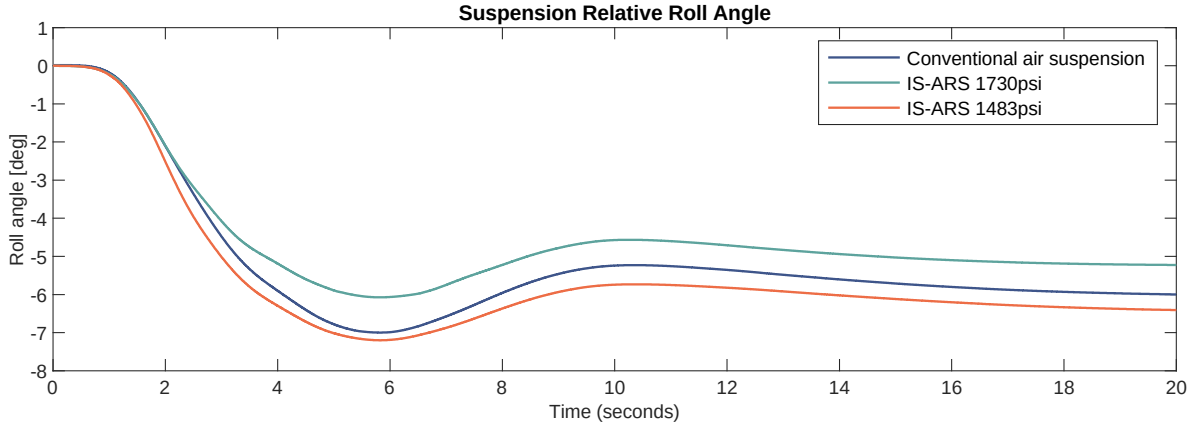


Figure 6.2. Truck suspension relative roll angle (steady-turn at 45 km/h)

As can be seen in Figure 6.2, the suspension relative roll angle was reduced by nearly 20% after implementing the passive IS-ARS when $V_0 = 2.4L$ & $P_0 = 1730psi$, which significantly improved the truck roll stability. Besides, it is noticeable that the suspension relative roll angle of the passive IS-ARS became even larger than that of the conventional air suspension when $V_0 = 2.8L$ & $P_0 = 1483psi$, which presented that the roll stiffness of the passive IS-ARS can be adjusted by changing the nominal pressure P_0 . With the functionality of adjustable roll stiffness, a wide range of suspension roll stiffness can be achieved without needing a mechanical anti-roll bar. This way, the roll stiffness characteristics can be conveniently tuned according to the vehicle load, driver preference, driving condition, etc. Besides, the stiffness range can be further adjusted for different truck models using the proposed embodiment and method.

6.1.2 Heavy truck: articulation test

Another benefit of the IS-ARS topology is the enhanced road-holding ability by maintaining similar tire load on each side of the vehicle, especially when driving on uneven roads or off-road, which sometimes causes suspension articulation issues. In this scenario, a pair of 20cm offset-bump shown in Figure 6.3 was utilized, and the truck was slowly driven over it at 10 km/h.

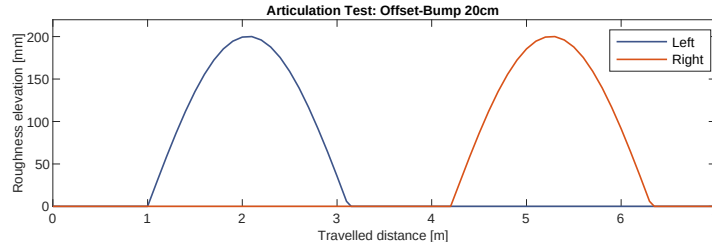


Figure 6.3. Articulation test road profile

Figure 6.4 shows the front drive-axle's lateral load transfer (LLT) when the truck was fully loaded (22 tons of cargo). It can be seen that the maximum value of the LLT value dropped significantly by 41% compared to that of the conventional air suspension. Figure 6.5 presents the simulated result of the half-loaded truck (11 tons of cargo). Similarly, the maximum value of the LLT was reduced by 25%. Moreover, a higher nominal pressure resulted in slightly larger LLT values; however, the difference was within 5% and could not be recognized in normal driving scenarios.

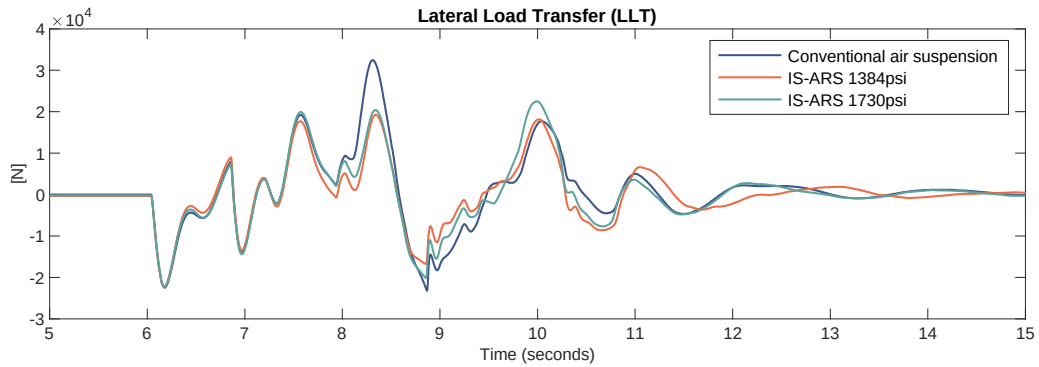


Figure 6.4. Lateral load transfer (articulation test - fully loaded)

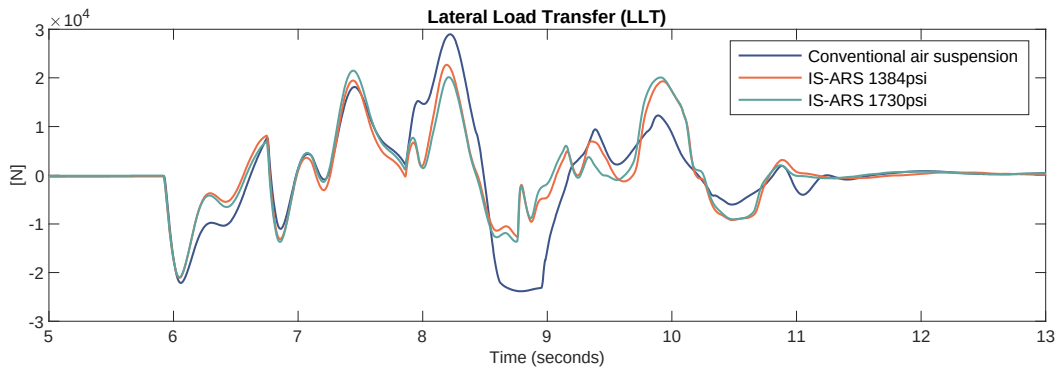


Figure 6.5. Lateral load transfer (articulation test - half loaded)

6.1.3 Heavy truck: speedbumps at 20 km/h

Figure 6.6 shows the road profile of the multiple speedbumps test. Each speedbump was spaced by 1 meter, 1.5 meters, and 2 meters, respectively. The truck chassis frame's pitch motion was insignificant because of its long wheelbase.

Figure 6.7 compares the vertical acceleration responses at the chassis frame CG location. With the passive IS-ARS, the undesired shock was effectively reduced by 12% compared to the conventional air suspension. By upgrading to the AIS-ARS, the vertical acceleration was significantly minimized by 49%. Meanwhile, the suspension travelled within the available deflection range, as shown in Figure 6.8.

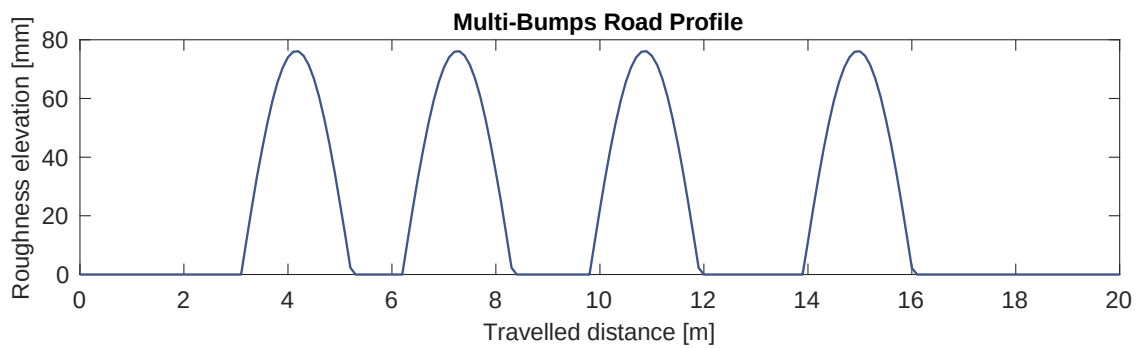


Figure 6.6. Multiple speedbumps road profile

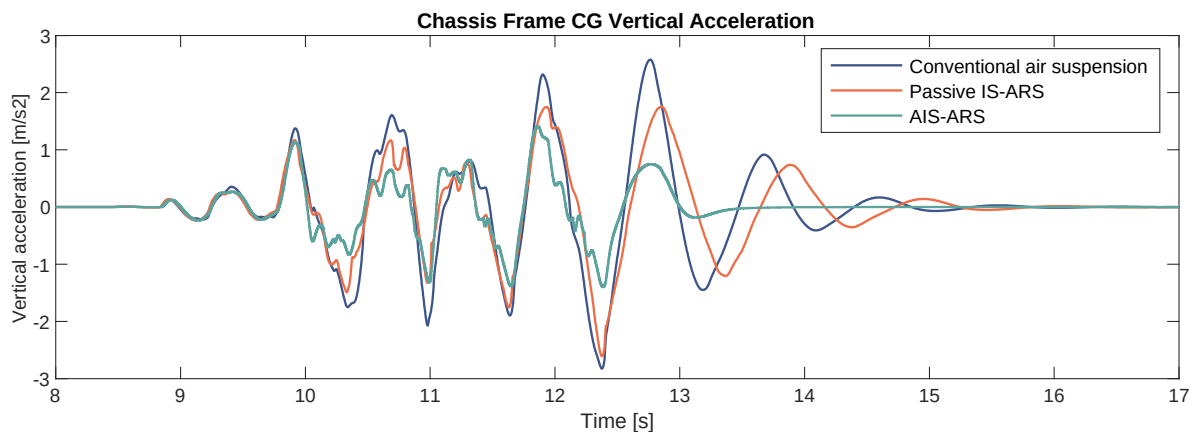


Figure 6.7. Truck chassis frame vertical acceleration (speedbumps at 20 km/h)

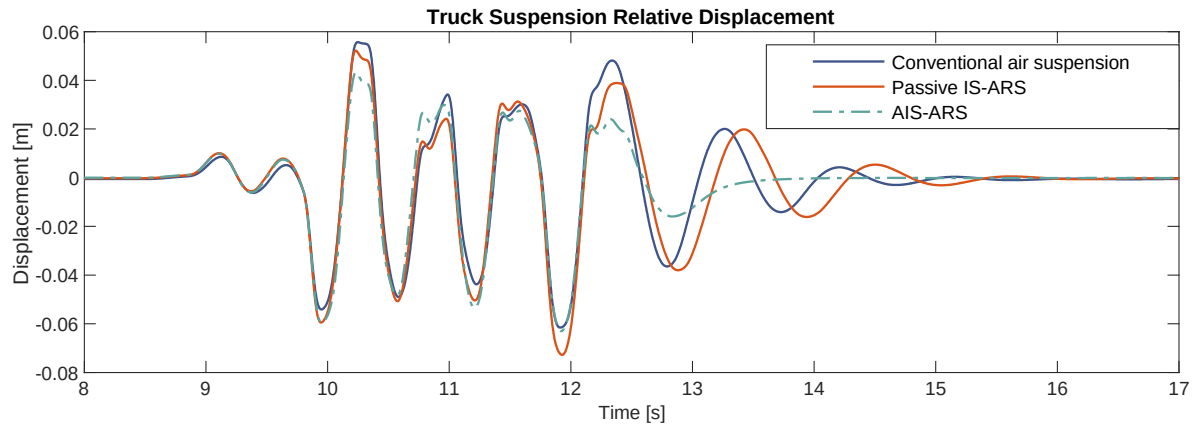


Figure 6.8. Truck suspension relative displacement (speedbumps at 20 km/h)

6.1.4 Heavy truck: paved road test

The paved road profile used in this scenario is shown in Figure 4.28, which contains three segments of ISO-A, ISO-B, and ISO-C roads. The evaluation criteria are listed in Table 6.1. As can be seen, the AIS-ARS provided 26% to 44% improvements, and the passive IS-ARS reduced the vibration by 2% to 24%. Notably, the optimization of the overall comfort assessment was 44% compared to the passive air suspension. It should be noted that the passive IS-ARS had been tuned to be as close as possible to the damping characteristics of the baseline. However, it can be well-tuned to achieve better performance.

Table 6.1. Performance evaluation (paved road test)

Paved road test			
	Passive Ind.	Passive IS-ARS	Adaptive IS-ARS
RMS of vertical accel. (m/s^2)	0.456	0.447 (-2%)	0.339 (-26%)
RMS of Freq.-weighted accel.	0.505	0.455 (-10%)	0.307 (-39%)
Overall assessment	1.192	0.910 (-24%)	0.670 (-44%)

6.1.5 Cabin: double lane change

This test examined the passive IS-ARS and the IS-ARPS systems equipped on a flat-nose cabin system. A double lane change maneuver was operated at 60 km/h, in which the vehicle changed lanes twice in

quick succession. It is commonly used when a driver needs to quickly move from one lane to another to avoid an obstacle, merge onto a highway, or get around slower-moving traffic. The vehicle responses were compared when the cabin was equipped with the conventional air suspension, IS-ARS, and IS-ARPS (charged at 300 psi).

Figure 6.9 presents the suspension relative roll angles. It can be seen that the maximum relative roll angle achieved approximately 6 degrees when using the independent cabin suspension. With the IS-ARS system, the maximum relative roll angle decreased to 3 degrees, better than the IS-ARPS (4 degrees). Regarding the RMS values of the cabin suspension relative roll angle, the two interconnected suspension systems improved the cabin roll stability by 59% (IS-ARS) and 42% (IS-ARPS), respectively. Meanwhile, the cabin roll rate was reduced by 39% (IS-ARS) and 30% (IS-ARPS), as shown in Figure 6.10.

One of the benefits of the IS-ARS is the adjustable roll stiffness. As shown in Figure 6.11, the anti-roll characteristics of the IS-ARS were changed by adjusting the nominal pressure. In detail, a higher nominal pressure resulted in a smaller suspension relative roll angle and vice versa. With the functionality of adjustable roll stiffness, a wide range of roll stiffness could be achieved without the need for mechanical anti-roll bars. This way, the roll stiffness characteristics can be conveniently tuned according to load, driver preference, driving condition, etc. Besides, the stiffness range can be further modified for different vehicle models using the proposed embodiment and method.

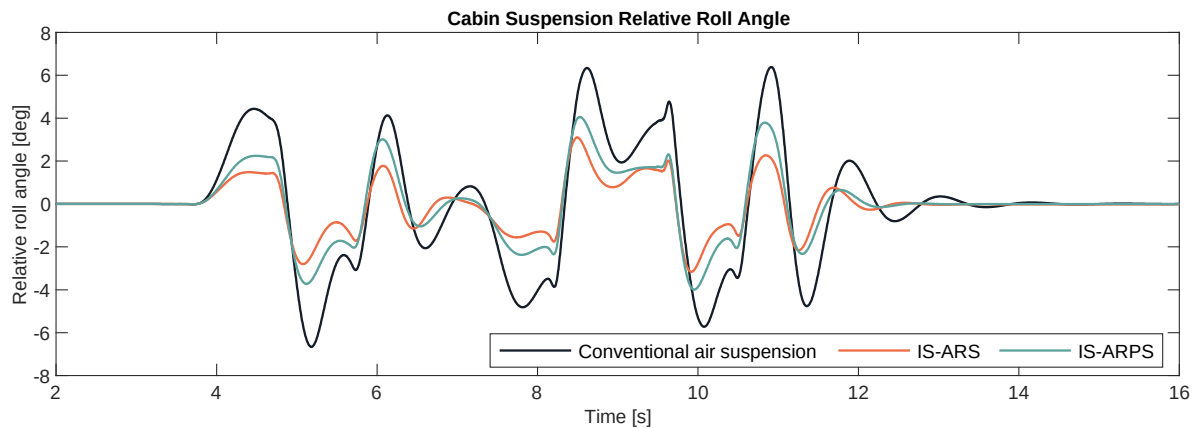


Figure 6.9. Cabin suspension relative roll angle (DLC at 60 km/h)

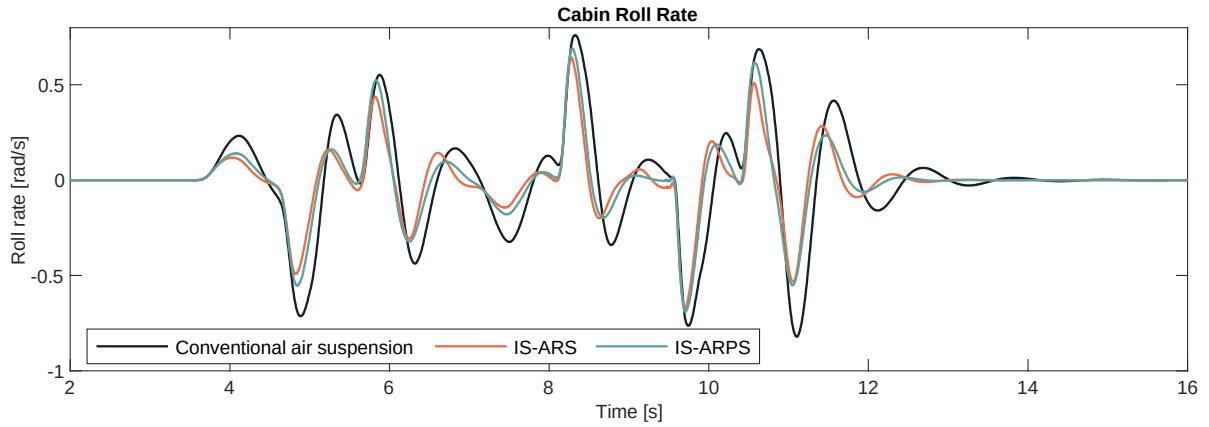


Figure 6.10. Cabin roll rate (DLC at 60 km/h)

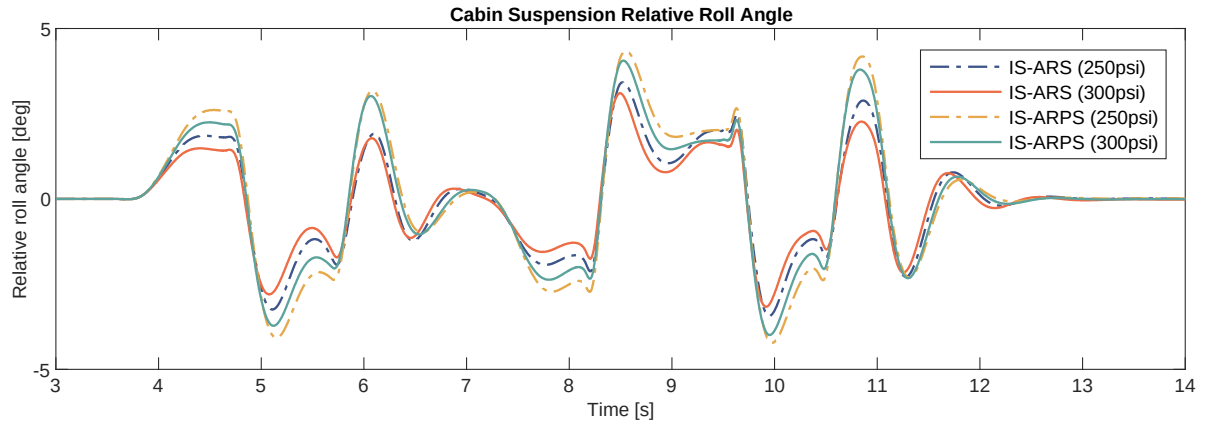


Figure 6.11. Cabin suspension relative roll angle at different pressures (DLC at 60 km/h)

6.1.6 Cabin: steady turn at 60 km/h

A steady turn maneuver is when a vehicle makes a controlled turn at a constant speed. This type of turn is commonly used when driving through curved highway ramps or making turns at intersections. Figure 6.12 shows the vehicle trajectory in this scenario, and the truck turned at 60 km/h.

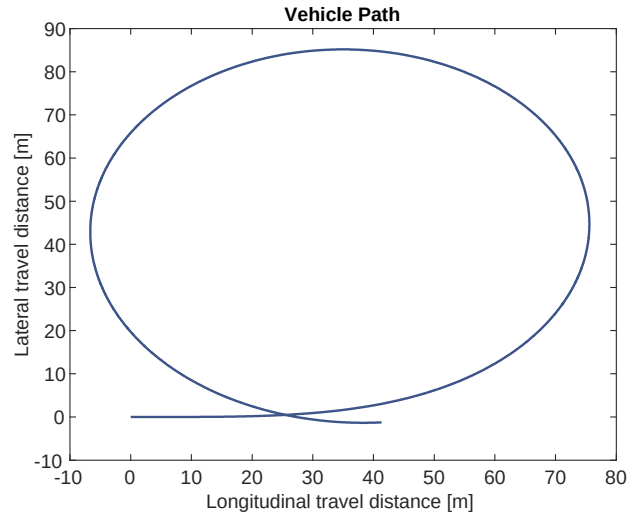


Figure 6.12. Vehicle trajectory (steady turn at 60 km/h)

Figure 6.13 shows the cabin suspension relative roll angles. It can be seen that the maximum roll angle was achieved by approximately 3.5 degrees by equipping the conventional air suspension. With the IS-ARPS, the maximum roll angle decreased to 2 degrees. Regarding the RMS values of the cabin suspension relative roll angle, the IS-ARPS improved the cabin roll stability by 45%. Meanwhile, the cabin roll rate was reduced by 42%, as shown in Figure 6.14. Moreover, the cabin roll accelerations are presented in Figure 6.15, demonstrating that the interconnected suspensions significantly reduced the rotational acceleration as expected.

One of the benefits of the proposed IS-ARPS is the adjustable roll stiffness. As shown in Figure 6.16, the anti-roll characteristics of the interconnected suspensions were changed by adjusting the nominal pressure. A higher nominal pressure resulted in less suspension relative roll angle and vice versa. With the functionality of adjustable roll stiffness, a wide range of suspension roll stiffness could be achieved without the mechanical anti-roll bars. This way, the roll stiffness characteristics can be conveniently adjusted according to load, driver preference, driving condition, etc. Besides, the stiffness range can be further adjusted for different cabin models using the proposed embodiment and method.

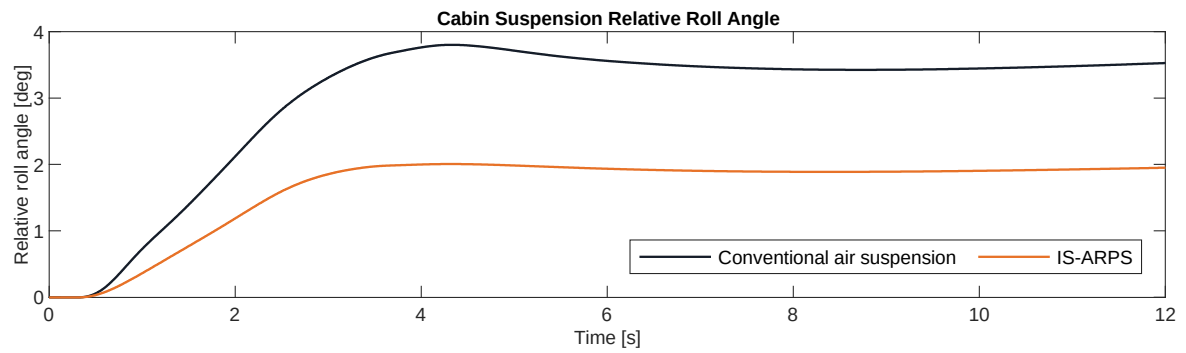


Figure 6.13. Cabin suspension relative roll angle (steady turn at 60 km/h)

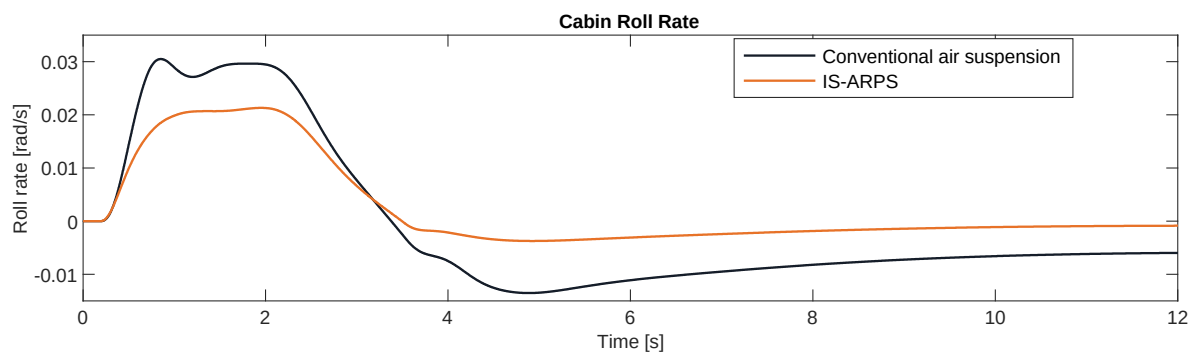


Figure 6.14. Cabin roll rate (steady turn at 60 km/h)

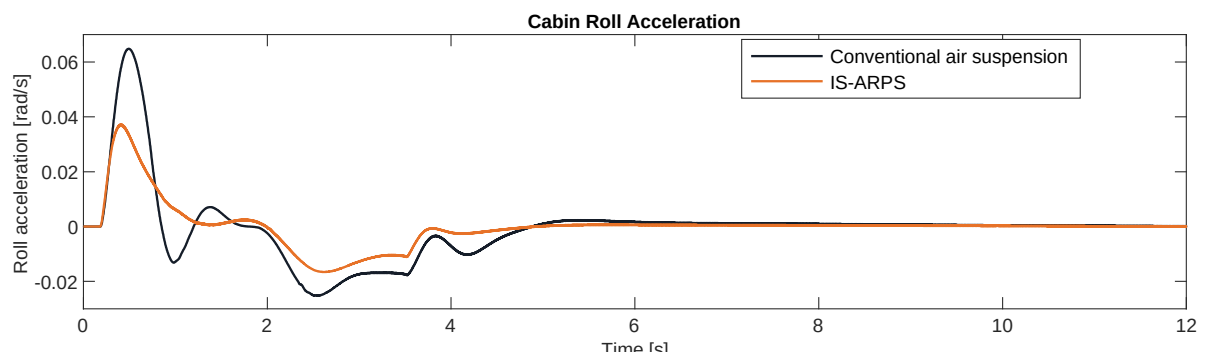


Figure 6.15. Cabin roll acceleration (steady turn at 60 km/h)

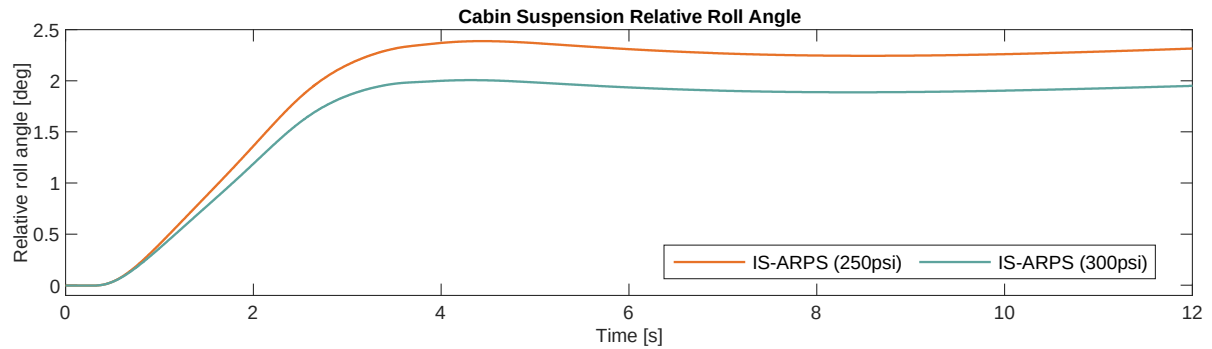


Figure 6.16. Cabin suspension relative roll angle at different pressures (steady turn at 60 km/h)

6.1.7 Cabin: emergency braking

On average, a fully loaded truck can have a maximum braking acceleration of around 0.5g to 0.7g. In this scenario, the truck was driven at 80 km/h and braked under non-constant accelerations within 0.45g. The corresponding speed and longitudinal acceleration are plotted in Figure 6.17.

Figure 6.18 shows the cabin suspension's relative pitch angles. It can be seen that the maximum pitch angle was approximately 3.7 degrees by using the conventional air suspension. When the nominal pressure equals 250 psi, the maximum pitch angle decreased to 1.3 degrees with the IS-ARPS system. Meanwhile, the cabin pitch rate was reduced by 52% in Figure 6.19, which significantly improved the cabin pitch stability as expected. Moreover, the anti-roll and anti-pitch characteristics of the IS-ARPS were adjustable by changing the nominal pressure. When the nominal pressure increased to 300 psi, the maximum pitch angle decreased to 0.9 degrees, and the cabin pitch rate was reduced by 59%. A higher nominal pressure generally resulted in a minor suspension relative pitch angle. With the functionality of adjustable stiffness, a wide range of suspension rotational stiffness can be achieved. This way, the rotational stiffness characteristics can be conveniently adjusted according to load, driver preference, driving condition, etc. Besides, the stiffness range can be further adjusted for different cabin models using the proposed embodiment and method.

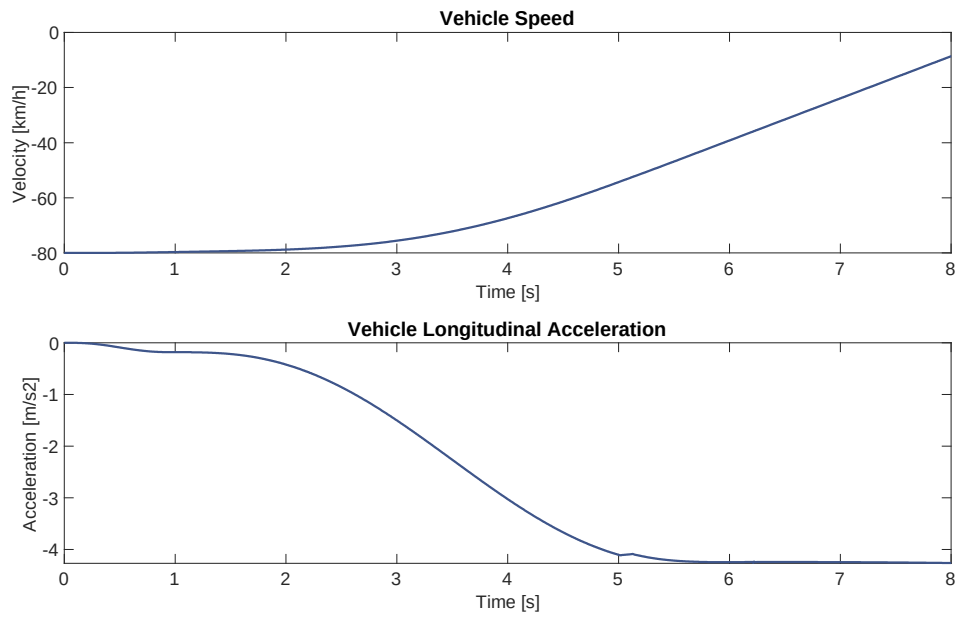


Figure 6.17. Vehicle longitudinal motion (emergency brake)

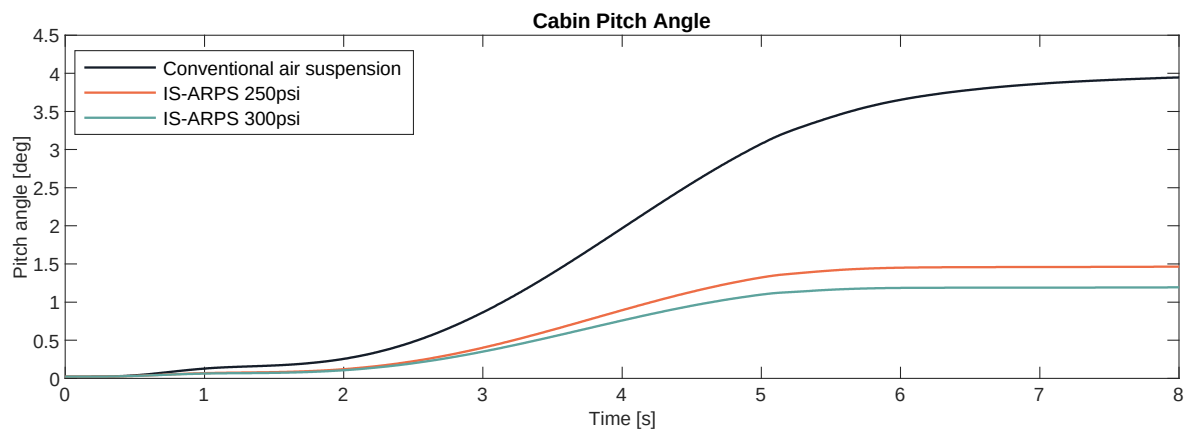


Figure 6.18. Cabin suspension relative pitch angle (emergency brake)

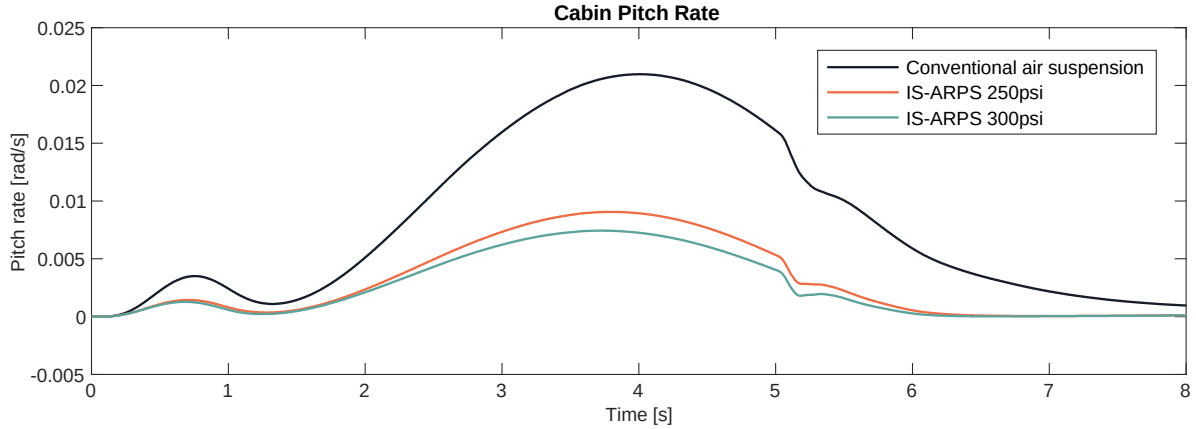


Figure 6.19. Cabin pitch rate (emergency brake)

6.2 Experimental Results

This laboratory experimental study aims to examine the mathematical model and the adjustable roll stiffness functionality of the AIS-ARS prototype, as shown in Figure 6.20. Considering the high fluid pressure and heavy parts, a horizontal platform was developed for safety, as shown in Figure 6.21. In detail, the four drive cylinders were rigidly bolted to the plate, and the rod sides were connected to the four hydraulic cylinders through the fork connectors. They were used to push the hydraulic cylinders back and forth to simulate the suspension's relative motions. The end sides of the four hydraulic cylinders were rigidly connected to the plate through L connectors, while the cylinders' rod sides were fixed to the plate by U bolts. The accumulators, manifold assemblies (hydraulic flow control valves), and one HPU were connected through hydraulic hoses. Besides, there were two pressure sensors on each manifold assembly to measure the fluid pressures at the manifold's inlet and outlet. Meanwhile, the hydraulic cylinders' travel distances were measured by four displacement sensors.

The final arrangement of the proposed test rig is presented in Figure 6.22(a), together with a diagram for describing the hardware communications in Figure 6.22(b). The motion-control program for the four drive cylinders was implemented in the Beckhoff TwinCAT platform, which was also connected to MATLAB through a Vector CAN Bus Transceiver. Meanwhile, the sensor measurements were processed and saved in MATLAB/Simulink for further analysis.

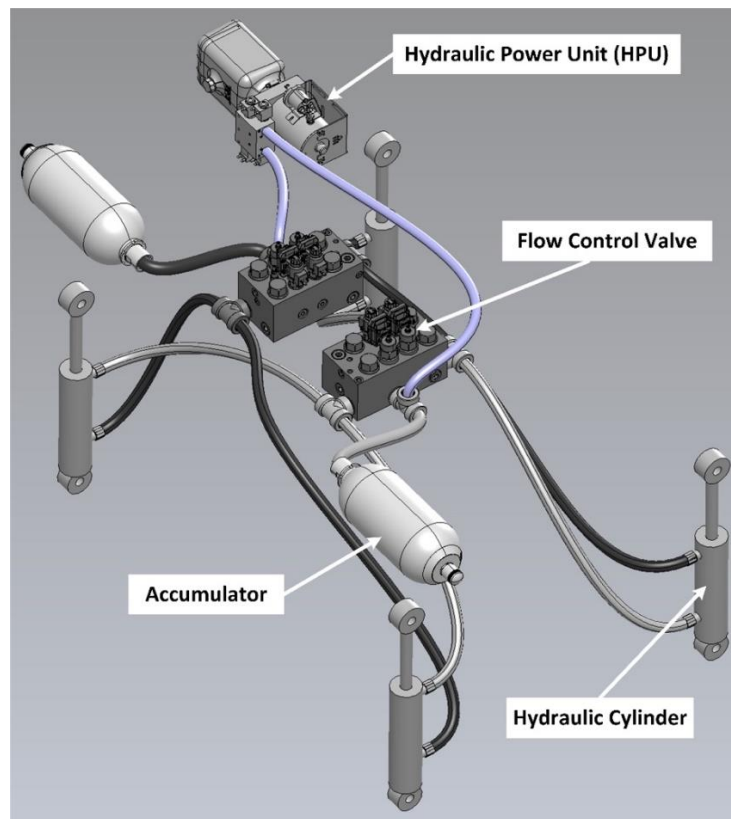


Figure 6.20. The CAD drawing of the roll-plane interconnected suspension

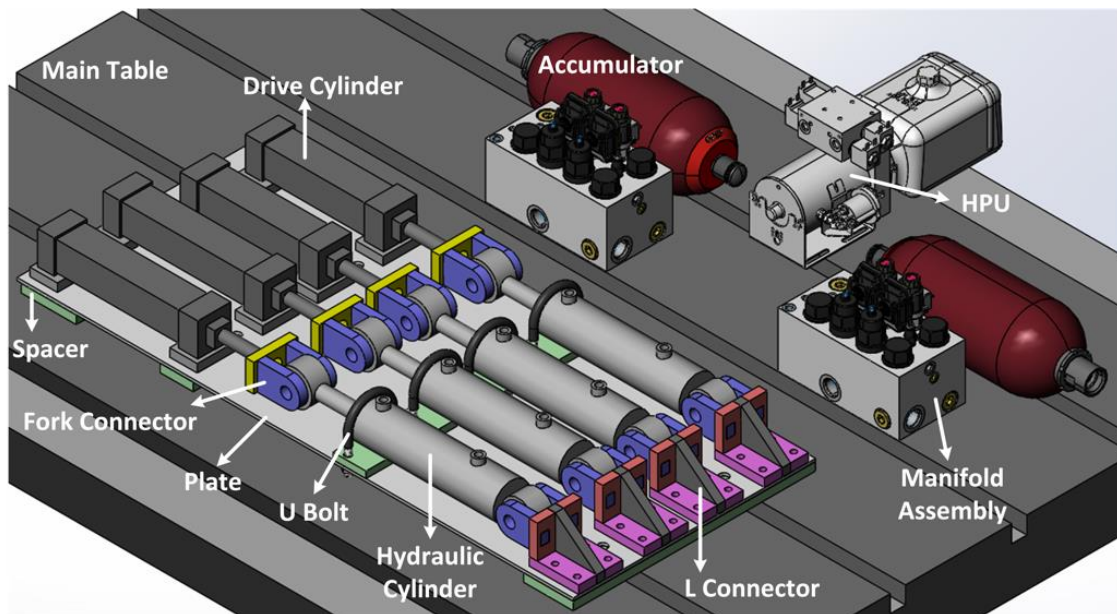


Figure 6.21 CAD drawing of the test-rig arrangement and main components

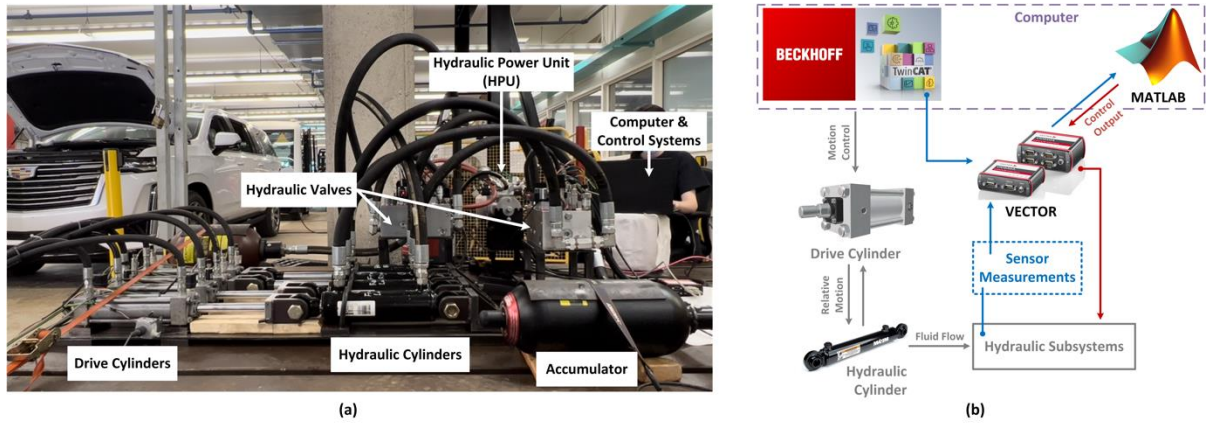


Figure 6.22. Laboratory experimental test rig

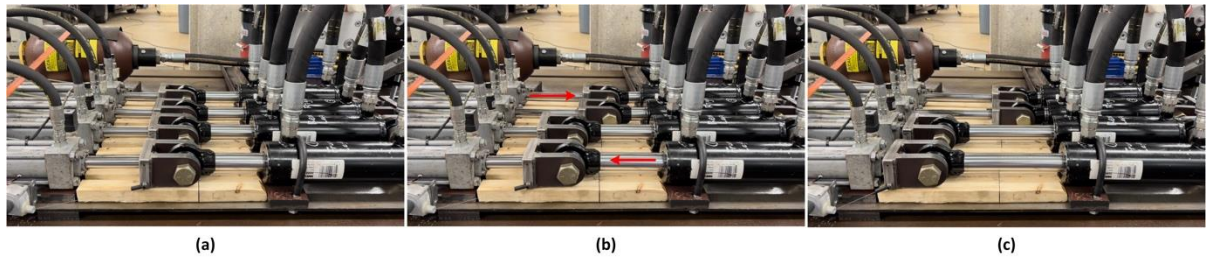


Figure 6.23. Hydraulic cylinder movements: from nominal position to maximum roll angle

As shown in Figure 6.23, the four hydraulic cylinders started from their nominal position, and the two left cylinders were slowly extended while the two right cylinders were retracted simultaneously. It meant that the suspension's relative roll angle started from 0 and slowly reached its maximum value at the end. Figure 6.24(a) presents the measured cylinder movements, and the equivalent suspension relative roll angle is calculated and plotted in Figure 6.24(b).

In this scenario, two pressure settings were tested and compared, i.e., $V_0 = 3L$ & $P_0 = 250psi$ and $V_0 = 2.5L$ & $P_0 = 300psi$. By increasing the suspension relative roll angle, the fluid pressure increased simultaneously because a large amount of oil flowed into an accumulator and caused air compression. In Figure 6.25, the tested roll moments and their fitted curves are presented and compared. It can be seen that the roll moment nonlinearly increased with the increases in the relative roll angle, and a higher nominal pressure resulted in a higher roll moment.

Figure 6.26 compares the estimated roll stiffness and its experimental results. Note that the estimated roll stiffness was calculated according to Eq.(5.15), and the tested roll stiffness was obtained by finding

the curvature slope of the roll moment plot in Figure 6.25. As expected, the estimated value and the experimental result showed a high level of agreement, which validated the accuracy of the proposed mathematical model. Moreover, the roll stiffness nonlinearly increased with the increase of the relative roll angle, which coincided with the proposed design. Besides, the roll stiffness increased by rising the nominal pressure, which validated the feasibility of the roll stiffness adjustment by utilizing an HPU.

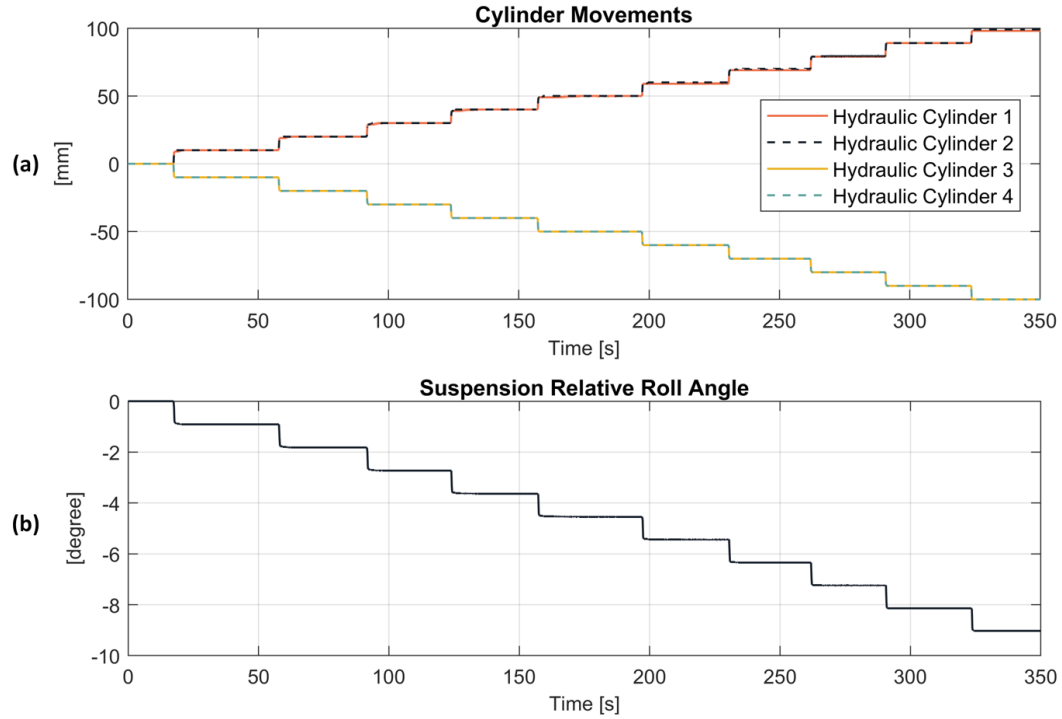


Figure 6.24. Hydraulic cylinder displacements (a), and equivalent roll angle (b)

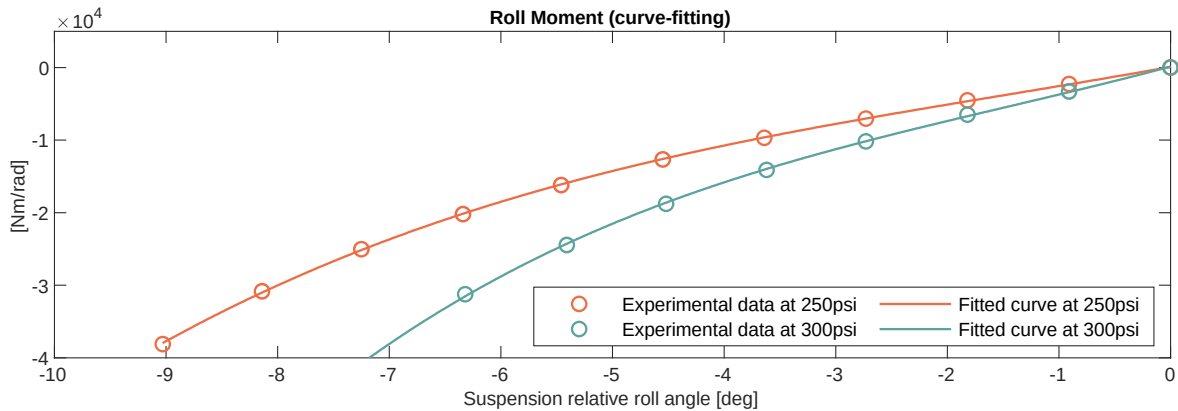


Figure 6.25. IS-ARS roll moment

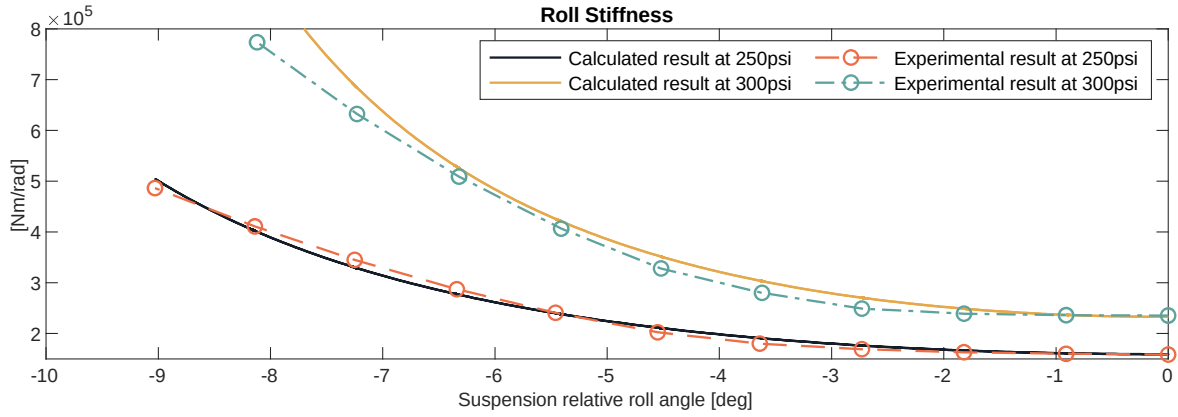


Figure 6.26. IS-ARS roll stiffness

6.3 Summary

Throughout the extensive simulation and experimental studies in this chapter, the proposed hydro-pneumatic interconnected suspension systems, AIS-ARS and AIS-ARPS, were found to be highly effective and could significantly improve suspension performance. The mathematical models and the stiffness adjustment method were tested through laboratory experiments, ensuring their reliability and feasibility for practical applications. The experimental results showed a clear correlation between the system nominal pressure and suspension rotational stiffness, demonstrating that the suspension rotational characteristics could be conveniently adjusted by operating the HPU. The overall performance of the two suspension systems was comprehensively evaluated through co-simulations between ADAMS/Car and MATLAB/Simulink under various operating conditions, such as steady turn, articulation test, speedbumps, paved road, emergency braking, etc. The simulation results revealed that the AIS-ARS not only significantly attenuated undesired vibrations but also greatly improved the vehicle's roll stability, while the AIS-ARPS demonstrated exceptional anti-pitch properties. These findings indicate that the two interconnected suspension systems offer superior performance and significant advantages compared to existing designs in the literature.

Chapter 7

Conclusions and Future Work

In this thesis, new suspensions and suspension control systems were developed to improve the ride, handling, and stability of trucks and truck cabins. The thesis outcomes are concluded in the following, and future works are proposed.

7.1 Conclusions

Development of coordinated semi-active suspension systems: In this thesis, the mathematical model of the COE cabin system was introduced in Chapter 3. The COE system posed a challenge as most past studies have used simple quarter-car, half-car, or full-car models for analyzing its dynamics. These car models failed to properly consider the unique cabin tilting mechanism, which significantly impacts the cabin's vertical and pitch dynamics. A linearized COE mathematical model was derived and implemented to overcome this limitation in MATLAB/Simulink. This model provided a precise estimation of the cabin motions, which made it an ideal tool for real-time suspension control tasks and performance analysis.

In addition, high-fidelity vehicle models are helpful in examining their performance in the virtual environment. In this regard, a high-fidelity truck model was developed in ADAMS/Car, which simulated the full vehicle dynamics with high accuracy. The real-time communication between MATLAB/Simulink and ADAMS/Car generated a realistic environment for suspension system development and performance evaluation. The truck model not only provided a comprehensive understanding of the full vehicle behavior but also assisted in optimizing the suspension design to meet the desired performance criteria.

In order to gain a comprehensive evaluation of the proposed semi-active suspension systems, a scaled-down COE prototype was designed and fabricated to provide a practical and manageable model for laboratory testing. As introduced in Section 4.4, a Stewart platform was utilized to generate the motions of the chassis frame in six directions, i.e., vertical, lateral, longitudinal, yaw, pitch, and roll. The scaled-down cabin prototype was then installed on the Stewart Platform with essential sensory systems. By thoroughly analyzing the motions of the scaled-down cabin and the Stewart Platform, the proposed COE mathematical model was examined, which validated that it could estimate cabin dynamics with a high level of accuracy and affordable computational cost. Based on this mathematical

model, a Kalman Filter observer was developed in Section 3.3 for estimating the unmeasurable cabin states, which played a vital role in the adaptive damping control systems.

To improve the ride quality in all three directions (vertical, roll, and pitch), a multi-objective control approach named Model Predictive Control (MPC) was introduced in Section 3.2.1 to develop semi-active suspension systems. In Chapter 4, its performance was examined through co-simulation and experimental studies. Both simulation and experimental results demonstrated that the proposed MPC controller improved the ride quality and cabin stability to a large extent. However, its computational time was too long for practical applications. Thus, a novel integrated Skyhook-LQR control algorithm was introduced in Section 3.2.2 to solve the high computational cost issue. The simulation and experimental studies showed that the integrated Skyhook-LQR approach took advantage of both Skyhook and LQR strategies and significantly optimized the vehicle dynamics in vertical, roll, and pitch directions simultaneously. Most importantly, the integrated Skyhook-LQR algorithm required affordable computational power, which could be implemented on automotive-grade microcontrollers for real-time processing.

Two gain-adaptive algorithms were introduced in Section 3.2.3 to further improve the performance of the integrated Skyhook-LQR controller, namely the disturbance-based algorithm and state-based algorithm. Since finding one optimum set of gains for various disturbance inputs is impossible, the two gain-adaptive algorithms aimed to intelligently adjust the LQR weighting matrix according to the rapidly changed road conditions. The disturbance-based algorithm utilized the measurements of disturbance inputs (motions of the chassis frame) to select the proper gains, while the state-based one was operated based on the measurement of the states (motions of the cabin body). One can choose the suitable gain-adaptive algorithm for optimal damping control tasks depending on the available onboard sensors. The experimental and simulation studies validated that the overall ride quality could be further improved by equipping the proposed gain-adaptive algorithms, especially when the vehicle was driven on paved roads.

Development of adaptive interconnected suspension systems: The above findings validated that the adaptive damping control can significantly attenuate the undesired vibration in multiple directions. However, it becomes ineffective during slow or steady motions, such as braking, lane changing, turning, etc. Considering the truck's high CG location and heavy load, its harsh rotational motions are

another source of discomfort and unsafety. Hence, developing a suspension system with enhanced rotational stability and safety is necessary.

To improve the vehicle's anti-roll characteristic, an adaptive interconnected suspension system with adjustable roll stiffness (AIS-ARS) was introduced in Section 5.1. Using the proposed interconnected suspension system, the conventional anti-roll bar(s) could be removed as the interconnected topology provided enhanced roll stiffness with nonlinear characteristics, i.e., the roll stiffness increased with increasing the suspension relative roll angle. Thus, the AIS-ARS was softer when the roll motion was minor to provide a comfortable ride, while it became stiffer when the roll motion started affecting the vehicle stability. Moreover, the ride quality was further improved by implementing an adaptive damping control strategy. Additionally, the adjustable roll stiffness functionality allowed the user to easily change the roll stiffness of the AIS-ARS according to driver preferences, load, vehicle model, etc. Meanwhile, the proposed AIS-ARS topology can be tuned to fit different vehicle models and applications.

Anti-pitch bars are mainly used in racing cars or high-performance vehicles to optimize the vehicle's pitch motion (nose-diving or tail-diving) during hard braking or acceleration. Considering that the anti-pitch bars increase the suspension complexity and manufacturing cost, it is not ideal for most passenger cars. However, a truck suffers harsher pitch movements because of its high CG location. Thus, it is necessary to develop a suspension system that can enhance both roll and pitch stability at the same time. In this thesis, a novel adaptive interconnected suspension system with adjustable roll and pitch stiffness (AIS-ARPS) was introduced in Section 5.2 to fill the gap. Specifically, the AIS-ARPS was equipped on a flat-nose cabin system, by which the cabin stability in pitch direction was significantly enhanced.

To study the characteristics of the AIS-ARS and AIS-ARPS, their mathematical models were derived and implemented in MATLAB/Simulink. Furthermore, the performance of these two systems was evaluated through co-simulations between MATLAB/Simulink and ADAMS/Car. The simulation results demonstrated that AIS-ARS significantly reduced the undesired roll motions of a fully loaded heavy truck and maintained its road-holding ability simultaneously. In addition, the AIS-ARPS reduced the cabin's undesired roll and pitch motions to a large extent.

To experimentally examine the interconnected topologies, a prototype of the AIS-ARS was designed and manufactured based on off-the-shelf products as introduced in Section 6.2, such as double-acting hydraulic cylinders, accumulators, hydraulic flow control valves, HPU, hydraulic hoses, etc. The

experimental study validated the accuracy of the proposed mathematical model. Moreover, the feasibility of the stiffness adjustment method was examined, which demonstrated that the suspension rotational stiffness of AIS-ARS and AIS-ARPS could be conveniently adjusted by operating the HPU, which provided manufacturers and users more flexibility in customizing suspension stiffness characteristics.

7.2 Future Work

The journey of this thesis has uncovered numerous fascinating insights and advancements in the field of vehicle suspension design and control. Amidst these accomplishments, it becomes evident that there remain a multitude of intriguing challenges that demand further investigation. As such, embracing these challenges as opportunities for future work is imperative. This section suggests some of the related topics for future studies.

Gain-adaptive algorithm: One aspect that warrants thorough investigation is the gain selection logic, which is pivotal in determining the overall control performance. In this study, the approach of using sigmoid functions to adjust the LQR gains was proved to be effective, albeit relying on a trial-and-error method for tuning. Developing advanced gain-tuning methods that can cater to multi-objective optimal control strategies would significantly improve the performance of the proposed suspensions. Real-time gain tuning can strike an optimal balance between competing objectives such as stability, responsiveness, and ride comfort. Future research can unlock the potential for even more robust and efficient control strategies by delving into the development of advanced gain-tuning methodologies.

Field testing: In this thesis, adaptive damping control algorithms were examined using a scaled-down cabin test rig, while the characteristics of interconnected suspension systems were evaluated based on a horizontal prototype. However, to ensure the robustness and applicability of these advancements in real-world scenarios, it is crucial to further validate their performance through comprehensive field tests. These field tests would provide an opportunity to assess the adaptability and effectiveness of the developed algorithms and evaluate the behaviour of interconnected suspension systems under diverse and dynamic road conditions. Through rigorous field testing, researchers can gather invaluable data that will enable them to fine-tune and optimize the algorithms and suspension systems, ultimately leading to more reliable and efficient performance in real-world driving scenarios.

References

- [1] Chris Winkler. Rollover of heavy commercial vehicles. *The UMTRI Research Review*, 31(4), 1-17. 2000.
- [2] H. E. Tseng, & Davor Hrovat. State of the art survey: active and semi-active suspension control. *Vehicle System Dynamics*, 53(7), 1034-1062. 2015.
- [3] Kentarou Ueno, Tomoaki Fujibayashi, Mitsuo Sasaki, Masaru Konishi, & Tooru Takahashi. Vehicle Control Techniques for Safety, Environmental Performance, and Ride Comfort. *Hitachi Review*, 67(1), 64. 2018.
- [4] Avesta Goodarzi, & Amir Khajepour. *Vehicle suspension system technology and design*, San Rafael, California. Morgan & Claypool, 2017.
- [5] Ahmad Mozaffari, Shojaeddin Chenouri, Yechen Qin, & Amir Khajepour. Learning-based vehicle suspension controller design: A review of the state-of-the-art and future research potentials. *eTransportation*, 2, 100024. 2019.
- [6] Fernando D. Goncalves, & Mehdi Ahmadian. A hybrid control policy for semi-active vehicle suspensions. *Shock and Vibration*, 10(1), 59-69. 2003.
- [7] Changning Liu, Long Chen, Heow P. Lee, Yi Yang, & Xiaoliang Zhang. Generalized Skyhook-Groundhook Hybrid Strategy and Control on Vehicle Suspension. *IEEE Transactions on Vehicular Technology*, 2022.
- [8] Sergio M. Savaresi, Enrico Silani, & Sergio Bittanti. Acceleration-Driven-Damper (ADD): An optimal control algorithm for comfort-oriented semiactive suspensions. 2005.
- [9] Sergio M. Savaresi, & Cristiano Spelta. Mixed sky-hook and ADD: Approaching the filtering limits of a semi-active suspension. 2007.
- [10] Shida Nie, Ye Zhuang, Weiping Liu, & Fan Chen. A semi-active suspension control algorithm for vehicle comprehensive vertical dynamics performance. *Vehicle System Dynamics*, 55(8), 1099-1122. 2017.
- [11] Johan Theunissen, Antonio Tota, Patrick Gruber, Miguel Dhaens, & Aldo Sorniotti. Preview-based techniques for vehicle suspension control: A state-of-the-art review. *Annual Reviews in Control*, 51, 206-235. 2021.
- [12] Hongyi Li, Xingjian Jing, & Hamid R. Karimi. Output-feedback-based H_∞ control for vehicle suspension systems with control delay. *IEEE Transactions on Industrial Electronics*, 61(1), 436-446. 2013.
- [13] Haiping Du, Nong Zhang, & Lifu Wang. Switched control of vehicle suspension based on motion-mode detection. *Vehicle System Dynamics*, 52(1), 142-165. 2014.
- [14] Panshuo Li, James Lam, & Kie C. Cheung. Control of vehicle suspension using an adaptive

- inertor. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 229(14), 1934-1943. 2015.
- [15] X. Zheng, H. Zhang, H. Yan, F. Yang, Z. Wang, & L. Vlacic. Active Full-Vehicle Suspension Control via Cloud-Aided Adaptive Backstepping Approach. 503113-3124, 2020.
- [16] Xin Tang, Haiping Du, Shuaishuai Sun, Donghong Ning, Zhiwei Xing, & Weihua Li. Takagi–Sugeno fuzzy control for semi-active vehicle suspension with a magnetorheological damper and experimental validation. *IEEE/ASME Transactions on Mechatronics*, 22(1), 291-300. 2016.
- [17] Menghua Zhang, Xingjian Jing, & Gang Wang. Bioinspired nonlinear dynamics-based adaptive neural network control for vehicle suspension systems with uncertain/unknown dynamics and input delay. *IEEE Transactions on Industrial Electronics*, 68(12), 12646-12656. 2020.
- [18] Sy D. Nguyen, Bao D. Lam, & Seung-Bok Choi. Smart dampers-based vibration control–Part 2: Fractional-order sliding control for vehicle suspension system. *Mechanical Systems and Signal Processing*, 148, 107145. 2021.
- [19] Sy D. Nguyen, Seung-Bok Choi, & Joo-Hyung Kim. Smart dampers-based vibration control–Part 1: Measurement data processing. *Mechanical Systems and Signal Processing*, 145, 106958. 2020.
- [20] Shao-Bo Lu, Yi-Nong Li, Seung-Bok Choi, Ling Zheng, & Min-Sang Seong. Integrated control on MR vehicle suspension system associated with braking and steering control. *Vehicle System Dynamics*, 49(1-2), 361-380. 2011.
- [21] Mirko Čorić, Joško Deur, Li Xu, H. E. Tseng, & Davor Hrovat. Optimisation of active suspension control inputs for improved vehicle ride performance. *Vehicle System Dynamics*, 54(7), 1004-1030. 2016.
- [22] Jing Na, Yingbo Huang, Xing Wu, Yan-Jun Liu, Yunpeng Li, & Guang Li. Active suspension control of quarter-car system with experimental validation. *IEEE Transactions on Systems, Man, and Cybernetics: Systems*, 2021.
- [23] G. Koch, & T. Kloiber. Driving State Adaptive Control of an Active Vehicle Suspension System. 2244-57, 2014.
- [24] Enrico Pellegrini. *Model-Based Damper Control for Semi-Active Suspension Systems*. Technical University of Munich. 2012.
- [25] M. Q. Nguyen, M. Canale, O. Sename, & L. Dugard. A Model Predictive Control approach for semi-active suspension control problem of a full car. 721-726, 2016.
- [26] Christoph Gohrle, Andreas Schindler, Andreas Wagner, & Oliver Sawodny. Model Predictive Control of semi-active and active suspension systems with available road preview. 1499-1504, 2013.
- [27] N. Giorgetti, A. Bemporad, H. E. Tseng, & D. Hrovat. Hybrid model predictive control

- application towards optimal semi-active suspension. *International Journal of Control*, 79(5), 521-533. 2006.
- [28] M. Canale, M. Milanese, & C. Novara. Semi-Active Suspension Control Using "Fast" Model-Predictive Techniques. *IEEE Transactions on Control Systems Technology*, 14(6), 1034-1046. 2006.
- [29] Lehel H. Cseko, Michal Kvasnica, & Bela Lantos. Explicit MPC-Based RBF Neural Network Controller Design With Discrete-Time Actual Kalman Filter for Semiactive Suspension. *IEEE Transactions on Control Systems Technology*, 23(5), 1736-1753. 2015.
- [30] Kjellberg Fredrik, & Sundell Simon. *Real-time nonlinear model predictive control for semi-active suspension with road preview*. Chalmers University of Technology. 2018.
- [31] Johan Theunissen, Aldo Sorniotti, Patrick Gruber, Saber Fallah, Marco Ricco, Michal Kvasnica, & Miguel Dhaens. Regionless Explicit Model Predictive Control of Active Suspension Systems With Preview. *IEEE Transactions on Industrial Electronics (1982)*, 67(6), 4877-4888. 2020.
- [32] Marcelo M. Morato, Manh Q. Nguyen, Olivier Sename, & Luc Dugard. Design of a fast real-time LPV model predictive control system for semi-active suspension control of a full vehicle. *Journal of the Franklin Institute*, 356(3), 1196-1224. 2019.
- [33] Jian Wu, Hongliang Zhou, Zhiyuan Liu, & Mingqin Gu. Ride comfort optimization via speed planning and preview semi-active suspension control for autonomous vehicles on uneven roads. *IEEE Transactions on Vehicular Technology*, 69(8), 8343-8355. 2020.
- [34] C. Gohrle, A. Wagner, A. Schindler, & O. Sawodny. Active suspension controller using MPC based on a full-car model with preview information. 497-502, 2012.
- [35] Hans J. Ferreau, Christian Kirches, Andreas Potschka, Hans G. Bock, & Moritz Diehl. qpOASES: a parametric active-set algorithm for quadratic programming. *Mathematical Programming Computation*, 6(4), 327-363. 2014.
- [36] G. Cimini, A. Bemporad, & D. Bernardini. ODYS QP Solver. 2017.
- [37] Kazem Cheshmi, Danny Kaufman, Shoaib Kamil, & Maryam Dehnavi. NASOQ: numerically accurate sparsity-oriented QP solver. *ACM Transactions on Graphics*, 39(4), 96:1-96:17. 2020.
- [38] Bartolomeo Stellato, Goran Banjac, Paul Goulart, Alberto Bemporad, & Stephen Boyd. OSQP: an operator splitting solver for quadratic programs. *Mathematical Programming Computation*, 12(4), 637-672. 2020.
- [39] Jiangtao Cao, Honghai Liu, Ping Li, & D. J. Brown. State of the Art in Vehicle Active Suspension Adaptive Control Systems Based on Intelligent Methodologies. *IEEE Transactions on Intelligent Transportation Systems*, 9(3), 392-405. 2008.
- [40] Le H. Nguyen, Keum-Shik Hong, & Seonghun Park. Road-frequency adaptive control for semi-active suspension systems. *International Journal of Control, Automation, and Systems*, 8(5),

1029-1038. 2010.

- [41] R. Kosut. Suboptimal control of linear time-invariant systems subject to control structure constraints. *IEEE Transactions on Automatic Control*, 15(5), 557-563. 1970.
- [42] Keum-Shik Hong, Hyun-Chul Sohn, & J. K. Hedrick. Modified Skyhook Control of Semi-Active Suspensions: A New Model, Gain Scheduling, and Hardware-in-the-Loop Tuning. *Journal of Dynamic Systems, Measurement, and Control*, 124(1), 158-167. 2002.
- [43] K. Yi, & B. S. Song. A new adaptive sky-hook control of vehicle semi-active suspensions. *Proceedings of the Institution of Mechanical Engineers. Part D, Journal of Automobile Engineering*, 213(3), 293-303. 1999.
- [44] Guido Koch, Klaus J. Diepold, & Boris Lohmann. Multi-Objective Road Adaptive Control of an Active Suspension System. Springer Netherlands. 189-200, 2008.
- [45] I. Fialho, & G. J. Balas. Road adaptive active suspension design using linear parameter-varying gain-scheduling. *IEEE Transactions on Control Systems Technology*, 10(1), 43-54. 2002.
- [46] Koen Deprez, Dimitrios Moshou, Jan Anthonis, Josse De Baerdemaeker, & Herman Ramon. Improvement of vibrational comfort on agricultural vehicles by passive and semi-active cabin suspensions. *Computers and Electronics in Agriculture*, 49(3), 431-440. 2005.
- [47] C. L. Giliomee, & P. S. Els. Semi-active hydro-pneumatic spring and damper system. *Journal of Terramechanics*, 35(2), 109-117. 1998.
- [48] I. Hostens, & H. Ramon. Descriptive analysis of combine cabin vibrations and their effect on the human body. *Journal of Sound and Vibration*, 266(3), 453-464. 2003.
- [49] M.J. Griffin. *Handbook of Human Vibration*, Elsevier Ltd, 27-42. 1990.
- [50] P. Donati. Survey of technical preventative measures to reduce whole-body vibration effects with designing mobile machinery. *Journal of Sound and Vibration*, 253(1), 169-183. 2002.
- [51] W. Evers. *Improving driver comfort in commercial vehicles: modeling and control of a low-power active cabin suspension system*. Eindhoven University of Technology. 2010.
- [52] Thomas D. Gillespie. *Heavy Truck Ride*. 1985.
- [53] National Transportation Safety Board. *Highway Special Investigation Report: Truck Parking Areas*, CreateSpace Publishing, 6, 2014.
- [54] Federica Caffaro, Christian Preti, Margherita Micheletti Cremasco, & Eugenio Cavallo. Three-axial evaluation of whole-body vibration in agricultural telehandlers: The effects of an active cab-suspension system. *Journal of Occupational and Environmental Hygiene; J Occup Environ Hyg*, 14(10), 758-770. 2017.
- [55] Rick T. Tong, & Farid Amirouche. Truck cab suspension design: optimisation and vibration control. *International Journal of Heavy Vehicle Systems*, 5(3-4), 236-260. 1998.

- [56] Roy Judhajit. *Design of Cab Suspensions and Semi-active Seat Damping Control Strategies for Tractor Semi-Trailers*. Clemson University. 2015.
- [57] Wallace Flower. Analytical and Subjective Ride Quality Comparison of Front and Rear Cab Isolation Systems on a COE Tractor. 1978.
- [58] Alexander Gross, & Van W. Roy. Development of a 4-point-Air Cab Suspension System for Conventional Heavy Trucks. 2001.
- [59] Judhajit Roy, & E. H. Law. Effect of Cab Suspension Configuration and Location on Tractor Semi-Trailer Driver Comfort. *SAE International Journal of Commercial Vehicles*, 9(2), 405-416. 2016.
- [60] Paul S. Patricio. *Effects of Frame Design and Cab Suspension on the Ride Quality of Heavy Trucks*. Virginia Polytechnic Institute and State University. 2002.
- [61] Albert W. Foster. A Heavy Truck Cab Suspension for Improved Ride. 1978.
- [62] N. Dadhania, Sankalp Bhatiya, Ayanesh Y. Joshi, D. Sharma, & A. Patel. Review of Cabin Tilting Mechanism for Earth-Moving Machinery. *International Journal of Advance Research and Innovative Ideas in Education*, 2, 510-514. 2016.
- [63] Chao Jiang, QianJun Zhu, & HuiBin Li. Simulation and Optimization of Truck Cab Suspension System Based on ADAMS. *IOP Conference Series: Earth and Environmental Science*, 242(3) 32039.
- [64] Alexander Groß, & Christophe Jolibert. Integrated levelling system for the cabin suspension of commercial vehicles. *ATZ Worldwide*, 103(9), 10-12. 2001.
- [65] Lin Lin, Tongyu Wang, Yuxin Huang, Limeng Wang, Hu Zhang, & Yanwen Zheng. The design and simulation experiment of full floating cab suspensions system for conventional heavy trucks. *International Conference on Industrial Mechatronics and Automation*, 854-858.
- [66] Mehdi Ahmadian, & Paul S. Patricio. Effect of Panhard Rod Cab Suspensions on Heavy Truck Ride Measurements. 2004.
- [67] Paras Jain. Design and Analysis of a Tractor-Trailer Cabin Suspension. 2007.
- [68] Krishna Achanta, Bhuvaneshwar Sharma, Subrat Bhanja, & Narendra Behera. Flexible Anti-Vibration Stability Cab Mount System for Commercial Vehicles. 2020.
- [69] Rogério Ribeiro, Fernando M. Sauer, Domingos da Silva, & Luís C. C. Costa. International Project Brazil/Germany with High-Level Teamwork: Development of a New Cab Suspension for an European Medium Duty Truck. 2013.
- [70] Yechen Qin, Ze Zhao, Zhenfeng Wang, & Guofa Li. Study of Longitudinal–Vertical Dynamics for In-Wheel Motor-Driven Electric Vehicles. *Automotive Innovation*, 4(2), 227-237. 2021.
- [71] Shuilong He, Keren Chen, Enyong Xu, Mingsong Ye, & Yanxue Wang. Coupling analysis and optimization of commercial vehicles cab comfort with multi-platform integrated calculation. *J*

Mech Sci Technol, 2021.

- [72] S. Bruni, J. P. Meijaard, G. Rill, & A. L. Schwab. State-of-the-art and challenges of railway and road vehicle dynamics with multibody dynamics approaches. *Multibody System Dynamics*, 49(1), 1-32. 2020.
- [73] Christine Ekberg, & Erik Hansson. *Design and simulation of active and semi-active cab suspensions with focus to improve ride comfort of a heavy truck*. Chalmers University of Technology. 2015.
- [74] Florin M. Marcu. *Semiactive cab suspension control for semitruck applications*. Virginia Tech. 2009.
- [75] S. S. Van Iersel. *Passive and semi-active truck cabin suspension systems for driver comfort improvement*. Eindhoven University of Technology. 2010.
- [76] A. M. A. Soliman. Improvement of the Truck Ride Comfort Via Cab Suspension. 2008.
- [77] P. A. Hansson. Control Algorithms for Active Cab Suspensions on Agricultural Tractors. *Vehicle System Dynamics*, 25(6), 431-461. 1996.
- [78] Yukun Lu, Amir Khajepour, Yegang Liu, & Ran Zhen. Adaptive cabin suspension systems of commercial vehicles: a review of the state-of-art and future trends. *International Journal of Heavy Vehicle Systems*, 29(1), 33-47. 2022.
- [79] Jihai Gu, Hui Wang, Chengwen Wang, Ming Pang, & Xiangyang Jin. The Parameter Optimization and Performance Analysis of the Suspension System in the Cab of a Heavy Truck. *International Conference on Computer and Computing Technologies in Agriculture*, 16-24.
- [80] Leilei Zhao, Changcheng Zhou, & Yuewei Yu. Damping Parameters Identification of Cabin Suspension System for Heavy Duty Truck Based on Curve Fitting. *Shock and Vibration*, 2016, 1-6. 2016.
- [81] Rick T. Tong, Farid Amirouche, & László Palkovics. Ride Control - A Two State Suspension Design for Cabs and Seats. *Vehicle System Dynamics*, 33, 578-589. 1999.
- [82] T. Hiromatsu, T. Inaba, & Y. Matsuo. Development of an electromechanical active-cab-suspension. *Proceedings of IECON '93 - 19th Annual Conference of IEEE Industrial Electronics*, 2132-2137 vol.3.
- [83] Jun Yan, Zheng Yin, Xuexun Guo, & Chang Fu. Fuzzy Control of Semi-active Air Suspension for Cab Based on Genetic Algorithms. 2008.
- [84] Dean Karnopp. Active Damping in Road Vehicle Suspension Systems. *Vehicle System Dynamics*, 12(6), 291-311. 1983.
- [85] Kimihiko Nakano, Yoshihiro Suda, Shigeyuki Nakadai, Hitoshi Tsunashima, & Takeshi Washizu. Self-powered active control applied to a truck cab suspension. *JSAE Review*, 20(4), 511-516. 1999.

- [86] C. Graf, & J. Maas. Commercial vehicle cabin with active air suspension for improved ride comfort. *IEEE/ASME International Conference on Advanced Intelligent Mechatronics*, 259-264.
- [87] ISO 2631-1:1997 Mechanical vibration and shock — Evaluation of human exposure to whole-body vibration — Part 1: General requirements. 1997.
- [88] Vanliem Nguyen, Renqiang Jiao, Vanquynh Le, & Peiling Wang. Study to control the cab shaking of vibratory rollers using the horizontal auxiliary damping mount. *Mathematical Models in Engineering*, 6(1), 57-65. 2020.
- [89] Vanliem Nguyen, & Vanquynh Le. Development of cab isolation systems of off-road vibratory rollers: review research. *Mathematical Models in Engineering*, 6(2), 93-102. 2020.
- [90] Vanliem Nguyen, Renqiang Jiao, Vanquynh Le, & Anhtan Hoang. Performance of PID-Fuzzy control for cab isolation mounts of soil compactors. *Journal of Mathematical Models in Engineering (MME)*, 5(4), 137-145. 2019.
- [91] Renqiang Jiao, & Vanliem Nguyen. Improving ride comfort for vibratory roller utilizing semi-active hydraulic cab mounts with control optimization. *Vibroengineering PROCEDIA*, Vol. 28, 2019, p. 75-80. 2019.
- [92] D. Hrovat. Survey of Advanced Suspension Developments and Related Optimal Control Applications. *Automatica (Oxford)*, 33(10), 1781-1817. 1997.
- [93] David Cebon. Vehicle-generated road damage: a review. *Vehicle System Dynamics*, 18(1-3), 107-150. 1989.
- [94] Emanuele Guglielmino, Tudor Sireteanu, Charles W. Stammers, Gheorghe Ghita, & Marius Giuclea. *Semi-active suspension control: improved vehicle ride and road friendliness*, Springer Science & Business Media, 2008.
- [95] Robert D. Ervin. The Influence of Size and Weight Variables on the Roll Stability of Heavy Duty Trucks. 1983.
- [96] M. El-Gindy. An overview of performance measures for heavy commercial vehicles in North America. *International Journal of Vehicle Design*, 16(4-5), 441-463. 1995.
- [97] Van T. Vu, Olivier Sename, Luc Dugard, & Peter Gaspar. Enhancing roll stability of heavy vehicle by LQR active anti-roll bar control using electronic servo-valve hydraulic actuators. *Vehicle System Dynamics*, 55(9), 1405-1429. 2017.
- [98] Guodong Wang, Yang Liu, Shaosong Li, Yunsheng Tian, Niaona Zhang, & Gaojian Cui. New Integrated Vehicle Stability Control of Active Front Steering and Electronic Stability Control Considering Tire Force Reserve Capability. *IEEE Transactions on Vehicular Technology*, 70(3), 2181-2195. 2021.
- [99] Li Zhai, Tianmin Sun, & Jie Wang. Electronic Stability Control Based on Motor Driving and Braking Torque Distribution for a Four In-Wheel Motor Drive Electric Vehicle. *IEEE*

- Transactions on Vehicular Technology*, 65(6), 4726-4739. 2016.
- [100] Santhosh Chandrasekharan, Dennis A. Guenther, Gary J. Heydinger, M. K. Salaani, & Scott B. Zagorski. Development of a roll stability control model for a tractor trailer vehicle. *SAE Technical Paper Series*, 670-679. 2009.
- [101] Yu C. Chen, & Hong Y. Zheng. Research on TTR and Roll Stability Control of Heavy Vehicle. *Applied Mechanics and Materials*, 380-384, 601-604. 2013.
- [102] Yang Chen, Mehdi Ahmadian, & Andrew Peterson. Pneumatically Balanced Heavy Truck Air Suspensions for Improved Roll Stability. 2015.
- [103] Lin Yang, Jun W. Zhang, & Si Z. Chen. Research on Hydro-Pneumatic Balanced Suspension of Multi-Spindled Vehicle. *Applied Mechanics and Materials*, 66-68, 855-861. 2011.
- [104] Alireza Pazooki, Dongpu Cao, Subhash Rakheja, & Paul-Émile Boileau. Ride dynamic evaluations and design optimisation of a torsio-elastic off-road vehicle suspension. *Vehicle System Dynamics*, 49(9), 1455-1476. 2011.
- [105] Dongpu Cao, Xubin Song, & Mehdi Ahmadian. Editors' perspectives: road vehicle suspension design, dynamics, and control. *Vehicle System Dynamics*, 49(1-2), 3-28. 2011.
- [106] Fei Ding, Xu Han, Xuhui Mo, & Nong Zhang. Design of Hydraulically Interconnected Suspension Systems for Tri-axle Straight Trucks with Rear Tandem Axle Bogie Suspensions. 2013.
- [107] Qijia Liu, Nong Zhang, Fuyong Feng, & Min Zhou. Handling performance of tractor-semitrailers equipped with hydraulically interconnected suspension. *Proceedings of the Institution of Mechanical Engineers. Part D, Journal of Automobile Engineering*, 233(12), 3098-3111. 2019.
- [108] Hengmin Qi, Bangji Zhang, Nong Zhang, Minyi Zheng, & Yuanchang Chen. Enhanced Lateral and Roll Stability Study for a Two-Axle Bus via Hydraulically Interconnected Suspension Tuning. *SAE International Journal of Vehicle Dynamics, Stability, and NVH*, 3(1), 5-18. 2018.
- [109] Jason Fenske. The McLaren 720S Suspension Is a Complex Masterpiece. 2019. Retrieved from <https://drivetribe.com/p/the-mclaren-720s-suspension-is-KIv3Q3zjS2SnSHx2hJUu2Q>
- [110] W. A. Smith. *An investigation into the dynamics of vehicles with hydraulically interconnected suspensions*. University of Technology, Sydney. 2009.
- [111] Bohuan Tan, Shaohua Wang, Bangji Zhang, Boyuan Li, Nong Zhang, & Hengmin Qi. Sensitivity stratification concept and hierarchical multi-objective optimisation for an ambulance with hydraulically interconnected suspension and stretcher-human body model. *Vehicle System Dynamics, ahead-of-print*, 1-29.
- [112] T. Fukuda, X. Zhang, Y. Hasegawa, T. Matsuno, & H. Hoshino. Preview posture control and impact load control of rough terrain vehicle with interconnected suspension. 1761-766 vol.1, 2004.

- [113] Jie Zhang, Xiao Chen, Bangji Zhang, Lifu Wang, Shengzhao Chen, & Nong Zhang. Design and Dynamic Analysis of Bounce and Pitch Plane Hydraulically Interconnected Suspension for Mining Vehicle to Improve Ride Comfort and Pitching Stiffness. 2015.
- [114] J. FÉLEZ, & C. VERA. Bond Graph Assisted Models for Hydro-Pneumatic Suspensions in Crane Vehicles. *Vehicle System Dynamics*, 16(5-6), 313-332. 1987.
- [115] Ren Luo, & Changdong Liu. Dynamics simulation of the high-speed train using interconnected hydro-pneumatic suspension as a self-steering system. *Vehicle System Dynamics*, ahead-of-print, 1-20.
- [116] Wade A. Smith, Wade A. Smith, Nong Zhang, & Nong Zhang. Recent developments in passive interconnected vehicle suspension. *Frontiers of Mechanical Engineering in China*, 5(1), 1-18. 2010.
- [117] Christopher B Heyring, & Michael J. Longman. Passive ride control for a vehicle suspension system. US10/890,451. 2008.
- [118] Wade A Smith. *An investigation into the dynamics of vehicles with hydraulically interconnected suspensions*. University of Technology, Sydney. 2009.
- [119] Hengmin Qi, Yuanchang Chen, Nong Zhang, Bangji Zhang, Dong Wang, & Bohuan Tan. Improvement of both handling stability and ride comfort of a vehicle via coupled hydraulically interconnected suspension and electronic controlled air spring. *Proceedings of the Institution of Mechanical Engineers.Part D, Journal of Automobile Engineering*, 234(2-3), 552-571. 2020.
- [120] Hongxue Li, Shiwu Li, & Wencai Sun. Vibration and handling stability analysis of articulated vehicle with hydraulically interconnected suspension. *Journal of Vibration and Control*, 25(13), 1899-1913. 2019.
- [121] W. A. Smith, & N. Zhang. Hydraulically Interconnected Vehicle Suspension: Optimization and Sensitivity Analysis. *Proceedings of the Institution of Mechanical Engineers.Part D, Journal of Automobile Engineering*, 224(11), 1335-1355. 2010.
- [122] Ming Wang, Bangji Zhang, Yuanchang Chen, Nong Zhang, & Jie Zhang. Frequency-Based Modeling of a Vehicle Fitted With Roll-Plane Hydraulically Interconnected Suspension for Ride Comfort and Experimental Validation. *IEEE Access*, 8, 1091-1104. 2020.
- [123] Bangji Zhang, Kaidong Tian, Wen Hu, Jie Zhang, & Nong Zhang. Investigation of the Influence of an Hydraulically Interconnected Suspension (HIS) on Steady-State Cornering. 2017.
- [124] Haohua Hong, Lifu Wang, Minyi Zheng, & Nong Zhang. Handling Analysis of a Vehicle Fitted with Roll-Plane Hydraulically Interconnected Suspension Using Motion-Mode Energy Method. 2014.
- [125] Lifu Wang, Song Jian, Hengmin Qi, & Nong Zhang. Lateral stability study of a vehicle fitted with hydraulically interconnected suspension in slalom maneuver.1702-1707, 2017.

- [126] Hengmin Qi, Minyi Zheng, Bangji Zhang, & Nong Zhang. Lateral Dynamics and Suspension Tuning for a Two-Axle Bus Fitted with Roll-Resistant Hydraulically Interconnected Suspension. 2018.
- [127] Wade A. Smith, Nong Zhang, & Jeku Jeyakumaran. Hydraulically interconnected vehicle suspension: theoretical and experimental ride analysis. *Vehicle System Dynamics*, 48(1), 41-64. 2010.
- [128] Sangzhi Zhu, Guangzhong Xu, Anton Tkachev, Lifu Wang, & Nong Zhang. Comparison of the road-holding abilities of a roll-plane hydraulically interconnected suspension system and an anti-roll bar system. *Proceedings of the Institution of Mechanical Engineers. Part D, Journal of Automobile Engineering*, 231(11), 1540-1557. 2017.
- [129] Lifu Wang, Guangzhong Xu, Nong Zhang, & Holger Roser. Experimental comparison of anti-roll bar with hydraulically interconnected suspension in articulation mode. *SAE Technical Papers*, 2013.
- [130] Hui Hua, Lifu Wang, Hengmin Qi, Jie Zhang, & Nong Zhang. Implementation and Experimental Study of a Novel Air Spring Combined with Hydraulically Interconnected Suspension to Enhance Roll Stiffness on Buses. 2015.
- [131] Hongxue Li, Shiwu Li, Wencai Sun, Linhong Wang, & Dongye Lv. The optimum matching control and dynamic analysis for air suspension of multi-axle vehicles with anti-roll hydraulically interconnected system. *Mechanical Systems and Signal Processing*, 139, 106605. 2020.
- [132] Fei Ding, Xu Han, Zhen Luo, & Nong Zhang. Modelling and characteristic analysis of tri-axle trucks with hydraulically interconnected suspensions. *Vehicle System Dynamics*, 50(12), 1877-1904. 2012.
- [133] Hong Su. *An investigation of vibration isolation systems using active, semi-active and tunable passive mechanisms, with applications to vehicle suspensions*. Concordia University. 1990.
- [134] Peijun Liu. *An analytical study of ride and handling performance of an interconnected vehicle suspension*. Concordia University. 1994.
- [135] Dongpu Cao, Subhash Rakheja, & Chun-Yi Su. Roll- and pitch-plane coupled hydro-pneumatic suspension: Part 1: Feasibility analysis and suspension properties. *Vehicle System Dynamics*, 48(3), 361-386. 2010.
- [136] Zhang Jie, Deng Yuanwang, Zhang Nong, Zhang Bangji, Qi Hengmin, & Zheng Minyi. Vibration Performance Analysis of a Mining Vehicle with Bounce and Pitch Tuned Hydraulically Interconnected Suspension. *Chinese Journal of Mechanical Engineering*, 32, 2019.
- [137] Guangzhong Xu, & Nong Zhang. Characteristic Analysis of Roll and Pitch Independently Controlled Hydraulically Interconnected Suspension. 2014.
- [138] J. K. HEDRICK, R. RAJAMANI, & K. YI. Observer Design for Electronic Suspension Applications. *Vehicle System Dynamics*, 23(1), 413-440. 1994.

- [139] Pauline Bernard, & Vincent Andrieu. Luenberger Observers for Nonautonomous Nonlinear Systems. *IEEE Transactions on Automatic Control*, 64(1), 270-281. 2019.
- [140] Antonio Rodríguez J., Emilio Sanjurjo, Roland Pastorino, & Miguel Á Naya. State, parameter and input observers based on multibody models and Kalman filters for vehicle dynamics. *Mechanical Systems and Signal Processing*, 155, 107544. 2021.
- [141] F. YU, & D. A. CROLLA. State Observer Design for an Adaptive Vehicle Suspension. *Vehicle System Dynamics*, 30(6), 457-471. 1998.
- [142] V. Sankaranarayanan, M. E. Emekli, Bilin A. Guvenc, Levent Guvenc, E. S. Ozturk, S. S. Ersolmaz, . . . Mustafa Sinal. Observer based semi-active suspension control applied to a light commercial vehicle.1-7, 2007.
- [143] Zheng Liu, Hongbin Ren, Sizhong Chen, Yong Chen, & Jianbo Feng. Feedback Linearization Kalman Observer Based Sliding Mode Control for Semi-Active Suspension Systems. *IEEE Access*, 8, 71721-71738. 2020.
- [144] Kuiyang Wang, Ren He, Heng Li, Jinhua Tang, Ruochen Liu, Yangmin Li, & Ping He. Observer-based control for active suspension system with time-varying delay and uncertainty. *Advances in Mechanical Engineering*, 11(11), 168781401988950. 2019.
- [145] Sunny Gupta, Divyesh Ginoya, P. D. Shendge, & Shrivijay B. Phadke. An inertial delay observer-based sliding mode control for active suspension systems. *Proceedings of the Institution of Mechanical Engineers.Part D, Journal of Automobile Engineering*, 230(3), 352-370. 2016.
- [146] E. Akbari, M. Farsadi, Intan Mat Darus Z., & R. Ghelichi. Observer design for active suspension system using sliding mode control. 207-212, 2010.
- [147] Simon Olma, Andreas Kohlstedt, Phillip Traphöner, Karl-Peter Jäker, & Ansgar Trächtler. Observer-based nonlinear control strategies for Hardware-in-the-Loop simulations of multiaxial suspension test rigs. *Mechatronics*, 50, 212-224. 2018.
- [148] Bin Lin, Xiaoyu Su, & Xiaohang Li. Fuzzy Sliding Mode Control for Active Suspension System with Proportional Differential Sliding Mode Observer. *Asian Journal of Control*, 21(1), 264-276. 2018.
- [149] Shi-Yuan Han, Jin Zhou, Yue-Hui Chen, Yi-Fan Zhang, Gong-You Tang, & Lin Wang. Active Fault-Tolerant Control for Discrete Vehicle Active Suspension via Reduced-Order Observer. *IEEE Transactions on Systems, Man, and Cybernetics.Systems*, 1-11. 2020.
- [150] Ming-Xing Cheng, & Xiao-Hong Jiao. Observer-Based Adaptive L2 Disturbance Attenuation Control of Semi-Active Suspension with MR Damper. *Asian Journal of Control*, 19(1), 346-355. 2017.
- [151] S. Aubouet, L. Dugard, & O. Sename. Experimental results of an H_∞ -observer for an industrial semi-active suspension. *IFAC Proceedings Volumes*, 43(7), 318-323. 2010.

- [152] S. Aubouet, L. Dugard, & O. Sename. H_∞ /LPV observer for an industrial semi-active suspension. 756-763, 2009.
- [153] Miaomiao Du, Dingxuan Zhao, Tao Ni, Lizhe Ma, & Song Du. Output Feedback Control for Active Suspension Electro-Hydraulic Actuator Systems With a Novel Sampled-Data Nonlinear Extended State Observer. *IEEE Access*, 8, 128741-128756. 2020.
- [154] Huihui Pan, Weichao Sun, Huijun Gao, Tasawar Hayat, & Fuad Alsaadi. Nonlinear tracking control based on extended state observer for vehicle active suspensions with performance constraints. *Mechatronics (Oxford)*, 30, 363-370. 2015.
- [155] M. Chadli, A. Rabhi, & A. El Hajjaji. Observer-based H_∞ fuzzy control for vehicle active suspension. 1393-1398, 2008.
- [156] Hamid Taghavifar, Aref Mardani, Chuan Hu, & Yechen Qin. Adaptive Robust Nonlinear Active Suspension Control Using an Observer-Based Modified Sliding Mode Interval Type-2 Fuzzy Neural Network. *IEEE Transactions on Intelligent Vehicles*, 5(1), 53-62. 2020.
- [157] Min Li, Yingchun Zhang, & Yunhai Geng. Fault-tolerant sliding mode control for uncertain active suspension systems against simultaneous actuator and sensor faults via a novel sliding mode observer. *Optimal Control Applications & Methods*, 39(5), 1728-1749. 2018.
- [158] Hamid D. Taghirad, & E. Esmailzadeh. Automobile Passenger Comfort Assured Through LQG/LQR Active Suspension. *Journal of Vibration and Control*, 4(5), 603-618. 1998.
- [159] Xue Liu, Hui Pang, & Yuting Shang. An Observer-Based Active Fault Tolerant Controller for Vehicle Suspension System. *Applied Sciences*, 8(12), 2568. 2018.
- [160] Miaomiao Du, Dingxuan Zhao, Mengke Yang, & Hao Chen. Nonlinear extended state observer-based output feedback stabilization control for uncertain nonlinear half-car active suspension systems. *Nonlinear Dynamics*, 100(3), 2483-2503. 2020.
- [161] Jie Zhang, Fei Ding, Nong Zhang, Shengzhao Chen, & Bangji Zhang. A New SSUKF Observer for Sliding Mode Force Tracking H_∞ Control of Electrohydraulic Active Suspension. *Asian Journal of Control*, 22(2), 761-778. 2020.
- [162] Hamidreza Bolandhemmat. *Distributed sensing and observer design for vehicles state estimation*. University of Waterloo. 2009.
- [163] Hongbin Ren, Sizhong Chen, Yuzhuang Zhao, Gang Liu, & Lin Yang. State observer-based sliding mode control for semi-active hydro-pneumatic suspension. *Vehicle System Dynamics*, 54(2), 168-190. 2016.
- [164] Gi-Woo Kim, Sun-Woo Kang, Jung-Sik Kim, & Jong-Seok Oh. Simultaneous estimation of state and unknown road roughness input for vehicle suspension control system based on discrete Kalman filter. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 234(6), 1610-1622. 2020.

- [165] L. Dugard, O. Sename, S. Aubouet, & B. Talon. Full vertical car observer design methodology for suspension control applications. *Control Engineering Practice*, 20(9), 832-845. 2012.
- [166] Haoping Wang, Lei Chang, & Yang Tian. Extended state observer-based backstepping fast terminal sliding mode control for active suspension vibration. *Journal of Vibration and Control*, 107754632095952. 2020.
- [167] Yingbo Huang, Jiande Wu, Jing Na, Shichang Han, & Guanbin Gao. Unknown System Dynamics Estimator for Active Vehicle Suspension Control Systems With Time-Varying Delay. *IEEE Transactions on Cybernetics*, 2021.
- [168] Power-Packer. Hydraulic Latch, Cab Tilt Cylinder & Pumps. Retrieved from <https://www.powerpackerus.com/hydraulic-cylinders/hydraulic-cab-tilt-pump-cylinder/>
- [169] J. P. Meijsaard, & V. van der Wijk. Principal vectors and equivalent mass descriptions for the equations of motion of planar four-bar mechanisms. *Multibody System Dynamics*, 48(3), 259-282. 2020.
- [170] Stefano Di Cairano, & Ilya V. Kolmanovsky. Real-time optimization and model predictive control for aerospace and automotive applications. 2392-2409, 2018.
- [171] Yubiao Zhang. Multi-axle Vehicle Modeling and Stability Control: A Reconfigurable Approach. 2019.
- [172] Mahmoud Ahmed, & Ferdinand Svaricek. Preview control of semi-active suspension based on a half-car model using Fast Fourier Transform. 1-6, 2013.
- [173] Marcelo M. Morato, Olivier Sename, & Luc Dugard. LPV-MPC Fault Tolerant Control of Automotive Suspension Dampers. *IFAC-PapersOnLine*, 51(26), 31-36. 2018.
- [174] Saad B. A. Kashem, Mehran Ektesabi, & Romesh Nagarajah. Comparison between different sets of suspension parameters and introduction of new modified skyhook control strategy incorporating varying road condition. *Vehicle System Dynamics*, 50(7), 1173-1190. 2012.
- [175] Ganesh, L. (2021). *Evaluation of Accurate Tire Models for Vehicle Handling and Ride Comfort Simulations*. doi:10.4271/2021-01-0935
- [176] Andrew Liburdi. Development of a scale vehicle dynamics test bed. 2010.
- [177] Kyle Johnson, Vivake Asnani, Jeff Polack, & Mark Plant. Experimental evaluation of the scale model method to simulate lunar vehicle dynamics. *International Society for Terrain-Vehicle Systems (ISTV) Americas Regional Conference*, (GRC-E-DAA-TN32423)
- [178] Bahman Zohuri. *Dimensional analysis and self-similarity methods for engineers and scientists*, Springer, 2015.
- [179] Abdullah Nammari, & Hamzeh Bardaweel. Design enhancement and non-dimensional analysis of magnetically-levitated nonlinear vibration energy harvesters. *Journal of Intelligent Material Systems and Structures*, 28(19), 2810-2822. 2017.

- [180] D. Stewart. A PLATFORM WITH SIX DEGREES OF FREEDOM (Reprinted from vol 180, 1965). *Proceedings of the Institution of Mechanical Engineers.Part C, Journal of Mechanical Engineering Science*, 223(1), 266-273. 2009.
- [181] Takeyuki Ono, Ryosuke Eto, Junya Yamakawa, & Hidenori Murakami. Analysis and control of a Stewart platform as base motion compensators - Part I: Kinematics using moving frames. *Nonlinear Dynamics*, 107(1), 51-76. 2022.
- [182] Sungwook Yang, Robert A. MacLachlan, & Cameron N. Riviere. Manipulator Design and Operation of a Six-Degree-of-Freedom Handheld Tremor-Canceling Microsurgical Instrument. *IEEE/ASME Transactions on Mechatronics; IEEE ASME Trans Mechatron*, 20(2), 761-772. 2015.
- [183] Vatsal Patel, Sanjay Krishnan, Aimee Goncalves, & Ken Goldberg. SPRK: A low-cost stewart platform for motion study in surgical robotics. 1-6.

Appendix A

High-fidelity models in ADAMS/Car

The modelling of the high-fidelity truck model and the cabin suspension system is introduced in this section. As shown in Figure A.1, a full truck template model named *msc_tractor_unit.asy* is provided in ADAMS/Car and this is used as a benchmark to develop the proposed truck system. Since this study is focusing on the cabin system, the subsystems of powertrain, steering system, rigid tractor, drive wheels, tandem drive axles, aux parts, barking system are directly borrowed, while the flat-nose cabin suspension system needs to be redesigned in ADAMS/Car Template Builder.

As shown in Figure A.2, a COE cabin body and its suspension system are developed in this study. There are four vertical spring-damper strut at each corner, two lateral dampers at the rear, one bending bar, and a pair of linkage mechanism at the front. Specifically, the bending bar is defined as a flexible body element, which requires a *.mnf* file to determine its elastic properties and assembly location. Note that the above *.mnf* file can be generated in ADAMS/View according to its dimension and material. Moreover, the modeling of the linkage mechanism is presented in Figure A.3. The linkages are modelled as general parts in ADAMS/Car Template Builder and are connected by multiple joints. In detail, the slotted hole is constrained by a translational joint and a revolute joint. Meanwhile, a fixed joint is used to rigidly connected the right-hand linkage to the cabin body. In this way, the motions of the linkage mechanism can be properly constrained for further simulation.

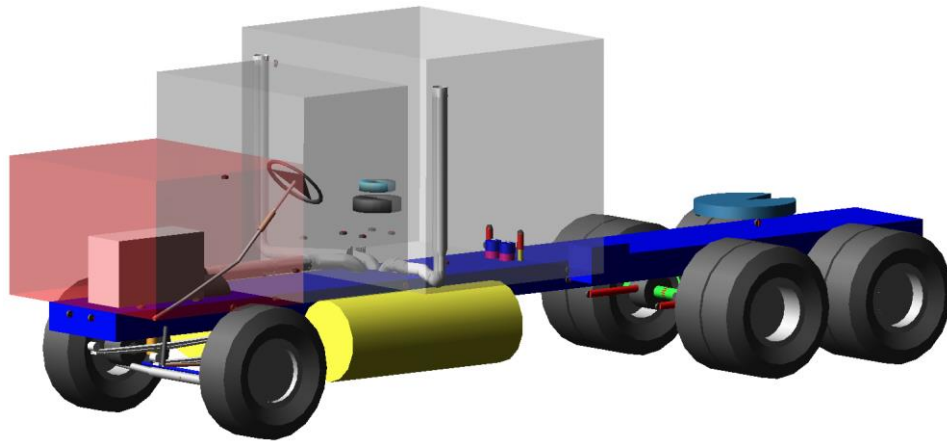


Figure A.1 The truck assembly example provided in ADAMS/Car.

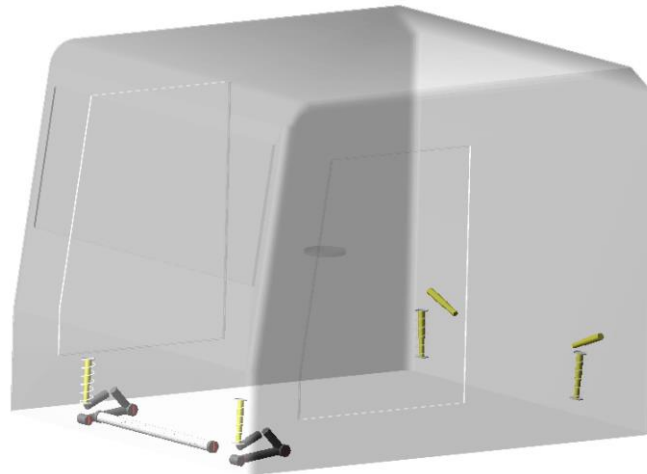


Figure A.2 The redesigned flat-nose cabin system.

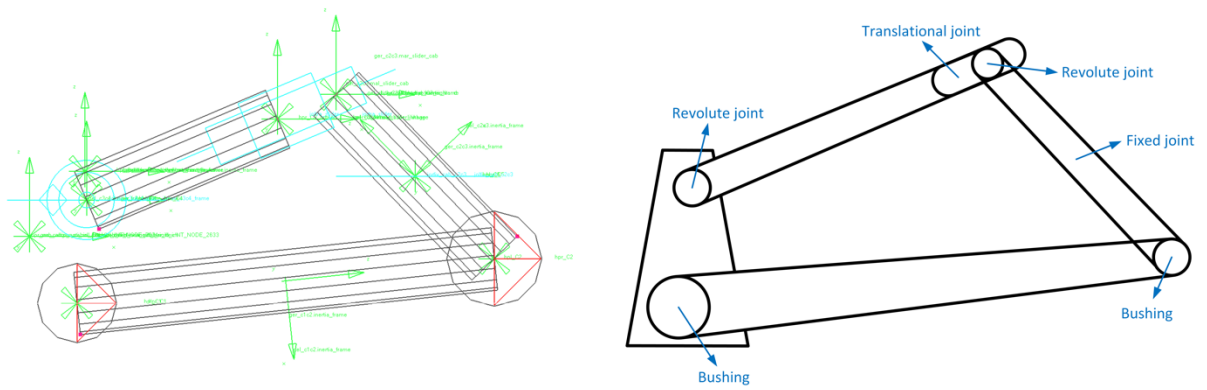


Figure A.3 The modelling of the linkage mechanism in ADAMS/Car.

Appendix B

Prototype of AIS-ARS

After investigating many relevant off-the-shelf products, the prototype's bill of materials is listed as follows:

Bill of Materials
1) Hydraulic Cylinder https://www.baileyhydraulics.com/Maxim-WSB-3000-PSI-Welded-Cylinders 28
2) Accumulator https://www.hydac.com/shop/en/hydraulic-accumulators/bladder-accumulators/standard-bladder-accumulators
3) Electro-proportional Flow Control Valve https://www.sunhydraulics.com/model/FPHK/XEN224NX01
4) Hydraulic Power Unit https://www.bucherhydraulics.com/en/products/power-units/hydraulic-power-packs--m-series/series-dc
5) Displacement sensor https://www.te.com/usa-en/product-SP1-12.html
6) Microcontroller https://www.audesseinc.com/product/flexcase/
7) Pressure Transducer https://www.omega.ca/en/pressure-measurement/pressure-transducers/px4200/p/PX4202-3KG5V
8) Temperature Transducer https://www.hydac-na.com/wp-content/uploads/ETS4100.pdf

In order to select the right size of the hydraulic pipes, it is required to investigate the pressure drop along the hydraulic pipeline. This would help to evaluate the overall circuit damping effect due to pipe resistance and damping valves. The main sections of the hydraulic lines that cause pressure drop are hydraulic pipes (flexible and rigid) and T-Junction connectors. The overall resistance and the resulted damping effects of the hydraulic system are simulated in MATLAB/SimScape, as shown in Figure B.1.

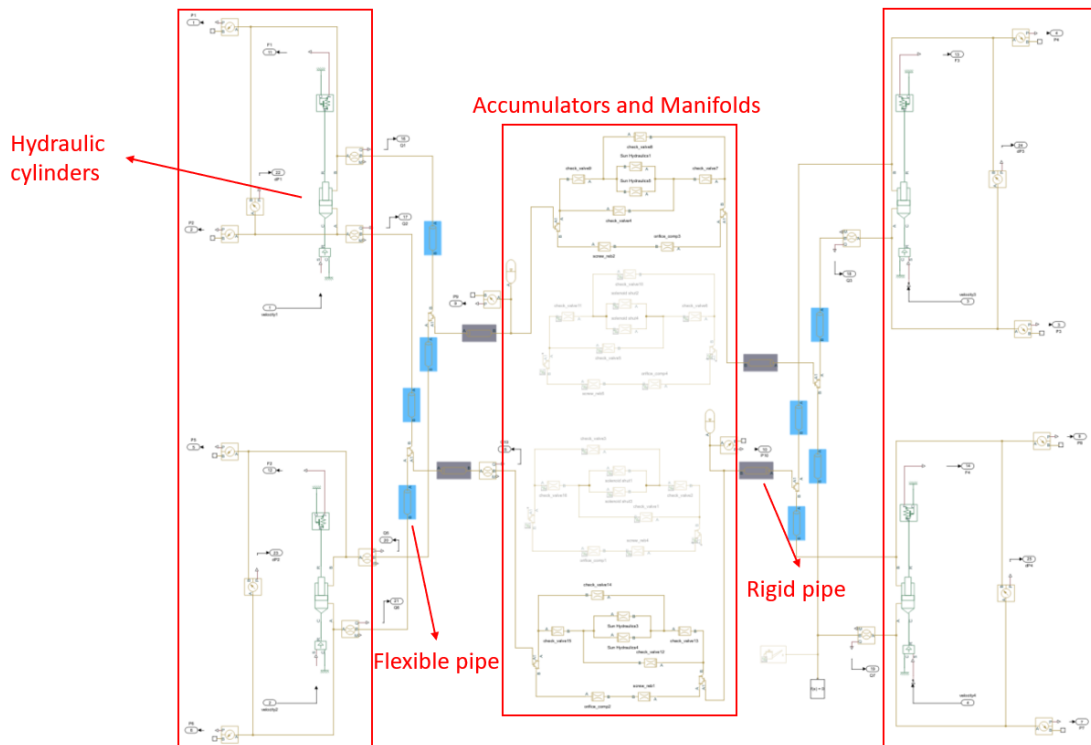


Figure B.1. Hydraulic system modelled in MATLAB/SimScape

Appendix C

Scaled-down cab-over-engine prototype

The design parameters of the scaled-down cabin prototype are listed as follows:

Parameters	Description	Full-size	Ratio	Scaled-down
m_s	Cabin mass	1250 kg	1/3	400 kg
I_x	Cabin roll moment of inertia	1468 kgm ²	1/12	122
I_y	Cabin pitch moment of inertia	1229 kgm ²	1/12	102
L_1	Distance from cabin CG to front cabin suspension	935.5 mm	1/2	467.75 mm
L_2	Distance from cabin CG to rear cabin suspension	1055 mm	1/2	527.5 mm
l	Half of the distance between the left and right airbag-damper strut	630 mm	1/2	315 mm
h	Height of the cabin CG location with respect to the cabin floor	542 mm	1/2	271 mm
f_{nz}	Vertical natural frequency	1.17 Hz	1	N/A
$f_{n\phi}$	Roll natural frequency	1.42 Hz	1	N/A
$f_{n\theta}$	Pitch natural frequency	1.23 Hz	1	N/A
k_1, k_3	Front air spring stiffness	23658 N/m	1/3	N/A
k_2, k_4	Rear air spring stiffness	14854 N/m	1/3	N/A
k_ϕ	Equivalent roll stiffness due to the bushings	86776 Nm/rad	1/12	N/A
Dimensions				
r_1	Length of linkage 1	0.265 m	1/2	132.5 mm
r_2	Length of linkage 2	0.144 m	1/2	72 mm

r_3^0	Nominal length of linkage 3	0.156 m	1/2	78 mm
r_4	Length of linkage 4	0.0653 m	1/2	32.65 mm
θ_1^0	Nominal value of angle θ_1	6.0°	1	6.0°
θ_2^0	Nominal value of angle θ_1	44.0°	1	44.0°
θ_3^0	Nominal value of angle θ_1	22.9°	1	22.9°
θ_4	Angle between linkage 4 and the vertical axis	5.3°	1	5.3°
l_{BS}	Longitudinal distance between joint B and CG location	1061.5 mm	1/2	530.75 mm
l_{BS1}	Longitudinal distance between joint B and S_1	362.5 mm	1/2	181.25 mm
l_{BS2}	Longitudinal distance between joint B and S_2	1762.8 mm	1/2	881.4 mm
α_1	Angle between CC' and BS_1	2.5°	1	2.5°
α_2	Angle between BB' and BS_2	36.09°	1	36.09°
α_3	Angle between BB' and BS	4.74°	1	4.74°