



2014

L. RAY BUCKENDALE LECTURE

2014-01-2408

Integrating Electromechanical Systems in Commercial Vehicles for Improved Handling, Stability, and Comfort

Mehdi Ahmadian

Virginia Polytechnic Institute and State University

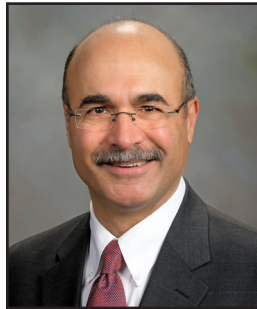
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L. RAY BUCKENDALE LECTURE

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Mehdi Ahmadian

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Dr. Ahmadian's research interests include: vehicle system dynamics and control, advanced automotive systems, Smart Materials and Systems, Advanced materials for improving ground vehicle dynamic performance and control, energy harvesting systems, intelligent suspensions, magneto-rheological fluids and biodynamics.

59TH ANNUAL L. RAY BUCKENDALE LECTURE

Integrating Electromechanical Systems in Commercial Vehicles for Improved Handling, Stability, and Comfort

Mehdi Ahmadian
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2014 SAE BUCKENDALE LECTURE

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ABSTRACT	1
CHAPTER 1 BACKGROUND	2
1.1. <i>A Historical Perspective on Vehicle Handling and Stability</i>	2
1.2. <i>Vehicle Stability and Control History</i>	3
1.3. <i>Commercial Vehicle Application of Active Safety Systems</i>	4
1.4. <i>Active and Semiactive Suspensions for Improved Ride Comfort and Safety</i>	4
1.5. <i>Outline</i>	5
CHAPTER 2 ADVANCED CONTROLLABLE SUSPENSIONS	7
2.1. <i>Vehicle Primary Suspensions</i>	7
2.1.1. <i>Passive Suspensions</i>	7
2.1.2. <i>Active Suspensions</i>	8
2.1.3. <i>Adjustable Suspensions</i>	9
2.1.4. <i>Semiactive Suspensions</i>	9
<i>Skyhook Control</i>	9
<i>Groundhook Control</i>	11
<i>Hybrid Control</i>	12
2.2. <i>Passive vs. Semiactive Dampers</i>	13
2.3. <i>Actual Passive Representation of Semiactive Suspensions</i>	14
2.4. <i>Magneto-rheological (MR) Suspension Basics</i>	14
2.4.1. <i>Types of MR Dampers</i>	14
2.4.2. <i>Mathematical Fundamentals of MR Dampers</i>	15
2.4.3. <i>Magnetic Circuit Design</i>	17
2.5. <i>MR Dampers for Vehicle Applications</i>	18
2.5.1. <i>MR Damper Performance</i>	20
2.5.2. <i>Damper Dynamic Testing</i>	22
2.6. <i>Controller Suspension System Configuration</i>	22
2.7. <i>Performance Benefits of Controllable Suspensions</i>	24
2.7.1. <i>Mathematical Model</i>	25
2.7.2. <i>H₂ Optimization Method</i>	26
2.7.3. <i>Relationship between vibration isolation, suspension deflection, and road holding</i>	27
2.8. <i>Final Remarks</i>	32

CHAPTER 3 ELECTRONIC ROLL AND YAW CONTROL SYSTEMS	34
3.1. Introduction	34
3.2. Physics of Rollover	36
3.3. How Does ESC Work?	37
3.4. Benefits of Electronic Stability Control Systems	39
3.5. Standard used for assessing ESC effectiveness	39
3.5.1. Slowly Increasing Steer Maneuver (SIS)	39
3.5.2. Sine with Dwell Maneuver (SWD)	40
3.6. Control Algorithm	41
3.6.1. Direct Yaw Control	41
3.6.2. Emergency Roll Control	44
3.6.3. Implementation of DYC and ERC	44
3.6.4. Performance Simulation	44
3.6.5. DYC and ERC Off-Road Performance	47
3.6.6. Evaluation of DYC and ERC Control Effort	50
3.7. Final Remarks	50
CHAPTER 4 CONCLUSIONS	52
ACKNOWLEDGMENT	52

Dedicated to

Azin, Rustin, and Neema

and

The memory of my parents

for their endless love!

Integrating Electromechanical Systems in Commercial Vehicles for Improved Handling, Stability, and Comfort

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Virginia Tech.

ABSTRACT

The 2014 SAE Buckendale Lecture will address the past developments and challenges of electromechanical “smart” systems for improving commercial vehicles’ functionality. Electromechanical systems combine traditional mechanical devices with electrical components to provide far higher degree of functionality and adaptability for improved vehicle performance. The significant advances in microprocessors and their widespread use in consumer products have promoted their implementation in various classes of vehicles, resulting in “smart” devices that can sense their operating environment and command an appropriate action for improved handling, stability, and comfort.

The chassis and suspension application of electromechanical devices mostly relate to controllable suspensions and vehicle dynamic management systems, such as Electronic Stability Control. Controllable suspensions include an active or semiactive element—most commonly, damper—that enables changing the dynamic characteristics of the suspension in real time, to adapt to the instantaneous driving dynamics of the vehicle. Such suspensions most often use an electromagnetic actuator, mechanically adjustable damper, or magneto-rheological fluid to change the damping force at the suspension with sufficiently high dynamic bandwidth to respond to every bump and each steering maneuver of the driver. Various analytical and experimental studies for automobiles and heavy trucks have shown significant improvements in suspension performance, substantially beyond those achievable with a conventional passive suspension, for both highway driving and off-highway operations.

Active safety systems broadly refer to devices that can improve the rollover dynamics and directional stability of vehicles. Such devices include Roll Stability Control (RSC), Electronic Stability Control (ESC), and Vehicle Dynamics Integrated Management (VDIM) systems, among other trade names that are commonly used. Active safety systems continuously monitor a series of stability indicators and, when needed, reduce the vehicle speed and implement a corrective steering moment (through applying the foundation brakes at selected wheels) to reduce the over turning accelerations in a turn and maintain the directional stability of the vehicle. It has been shown that such systems can significantly reduce the likelihood of rollovers and understeer, oversteer, or jackknifing during a steering maneuver, particularly in adverse road conditions or emergency steering maneuvers.

Chapter 1 provides a historical perspective on vehicle handling and stability, along with the application of active safety systems and controllable suspensions for commercial trucks. The concept of controllable suspensions for vehicles, particularly as it relates to magneto-rheological dampers, is introduced in Chapter 2, in addition to the common control methods for semiactive suspensions. The critical design aspects of magneto-rheological dampers along with the results of a simulation study that highlights their benefits in improving vehicle ride, handling, and stability are also introduced in this chapter. Chapter 3 is dedicated to highlighting the benefits of Electronic Stability Control (ESC) and Roll Stability Control (RSC) systems, and their effect on reducing rollovers in heavy trucks. It also includes the physics of tripped and untripped rollovers, the working principle of RSC and ESC systems, the government rulemaking efforts as documented in FMVSS 136, standard used for assessing ESC effectiveness, and the effectiveness of a control algorithm called Direct Yaw Control (DYC).

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CHAPTER 1 BACKGROUND

This chapter provides a historical perspective on vehicle handling and stability, as well as an overview of some of the past studies on vehicle stability and control. It also includes background discussions on active safety systems and controllable suspensions for commercial trucks. The information in this chapter is from various sources, including reference [1], the PhD dissertation of one of my past students.

1.1. A Historical Perspective on Vehicle Handling and Stability

Vehicles (ground, water, and air alike) are interesting and complex dynamic systems that require careful analysis and design to make sure they perform properly. The performance of vehicles is often judged by their safety and ride comfort. Safety commonly relates to the ability to maneuver the vehicle such that it can perform its intended function. For ground vehicles, safety is characterized by stability and handling of the vehicle. Stability assures that the vehicle maintains the needed contact with the ground so that it can remain upright. Stability also includes the ability to operate the vehicle such that it does not pose any risks to other vehicles around it. Handling refers to the ability of the operator-or a controller that is used in place of the operator, in the case of autonomous vehicles-to guide the vehicle along its intended path with sufficient agility, while maintaining stability.

During the period between 1900 and 1930, the emphasis on improving the ability of the driver to control the vehicle led to kinematic studies of suspension and steering geometries, resulting in improved designs including the Ackermann steering geometry. During that period, vehicles were plagued with the steering shimmy phenomenon. Broulheit introduced the basic concepts of sideslip and slip angle for the first time to better explain this phenomenon [2]. Following Broulheit's work, Becker, Fromm, and Maruhn published a text on the role of the tire in steering vibrations and further developed the field of tire mechanics [3].

In the 1930s, Olley and his colleagues found that certain steering geometries led to a condition called roll oversteer that was later recognized to be unsafe at high speeds [4, 5]. In the mid-1930s, an unpublished report by Olley stated the proposition of oversteer/understeer and the idea of critical speed [4]. In 1935 Goodyear Tire and Rubber Company began rolling drum tests to determine tire characteristics. The results of lateral tire characteristics including data on cornering force and self-aligning torque were published by Evans in 1935 [6].

During 1940s very little progress was made due to World War II.

The 1950s witnessed the resurgence of vehicle dynamics research and led to findings that serve as the backbone of our modern vehicle engineering. In 1950, Lind Walker introduced the concept of the neutral steer line and the stability margin [7]. Although these concepts had already been established for

aeronautical vehicles, Walker for the first time suggested them as criteria for steady state directional stability in automobiles. By the mid-1950s a basic understanding of the tire enabled the creation of reasonably accurate mathematical tire models. In 1956 William F. Milliken, David W. Whitcomb, and Leonard Segel of the Cornell Aeronautical Laboratory published the first major quantitative and theoretical analysis of vehicle handling in a series of papers [8, 9, 10, 11].

Milliken recognized that the major effort in handling research to date had been in the recognition of individual effects, their isolation, and examination as separate entities [8]. There was no universally accepted set of reference axes and measured tire data of the period was typically confined to two or three of the possible six force/moments. This made translation of the data from one set of axes to another difficult, if not impossible. Milliken emphasized the need to concentrate on the objective analysis of car stability and control. He made the important between stability and control, performance and ride. In general, an automobile has six-degrees-of-motion freedom. Stability and control may be thought of as those lateral motions out of the plane of symmetry involving rolling, yawing and sideslipping. Performance often includes fore-and-aft mobility in the plane of symmetry, such as acceleration, cruising, and braking; whereas ride commonly relates to the vertical and pitch dynamics of the vehicle.

Leonard Segel's work was devoted to derivation of a set of non-dimensionalized, linearized three degree of freedom equations for lateral and directional motion [10]. Segel ignored the bounce and pitch degrees of freedom of the chassis and used a fixed longitudinal roll axis parallel to the ground. The other simplifying assumptions included constant forward velocity, fixed driving thrust divided equally between the rear wheels, decoupled tire lateral properties from longitudinal properties, and a single non-rolling lumped mass as the unsprung mass. An experimental validation of the Segel's suggested model was performed on a four-door sedan at 32 mph in a series of frequency response curves.

A study by Whitcomb introduced the concept of "static margin," [11]. That study discussed automobile stability and control, using a two-degree-of-freedom model, commonly known as the "bicycle model," which includes the vehicle's yaw and sideslip.

In 1961 Martin Goland and Frederick Jindra added the roll degree of freedom as a quasi-coordinate to the two-degree-of-freedom (yaw and sideslip) vehicle model of Segel in order to calculate the vertical load on the tires. The effects of load transfer and the variation of the cornering performance of the tires with vertical loading were presented by Goland in 1961 [12].

In 1965 Walter Bergman published his findings of the nature of vehicle understeer and oversteer. He noted that understeer and oversteer could be recognized by considering the change in the yaw velocity induced by a change in lateral acceleration. This definition forms the foundation for the standardized

definitions of oversteer and understeer in SAE's Recommended Practice, J670e, Vehicle Dynamics Terminology, first issued in 1952 and last updated in 1976.

In 1967 Bundorf of General Motors published a study of the relationship between vehicle design parameters and characteristic speed and understeer [13]. Bundorf argued that under most normal driving conditions, which he characterized as having lateral accelerations below $1/3$ g, a vehicle could be accurately modelled by a linear model.

During the 1970s several timesaving methods were published, such as Bernard's digital simulation code for Highway Safety Department [14], Hybrid Computer Vehicle Handling Program [15], and Ford Motor Company's model containing detailed three-degree-of-freedom model of the front suspension [16]. In 1973, Frank H. Speckhart developed a vehicle model containing 14 degrees of freedom, using Lagrangian approach [17]. Six degrees of freedom were assigned to the sprung mass, four degrees of freedom were associated with the suspension movement at the four corners of the vehicle, and four rotational degrees of freedom were assigned to the wheels. Nalecz tested 25 different suspensions in order to understand the effect of suspension design on the stability of vehicles [18].

In 1992, Nalecz developed another model with eight degrees of freedom [19]. The model consisted of a three-degree-of-freedom lateral dynamics model coupled to a five-degree-of-freedom planar rollover model. The lateral dynamics model was derived in the same manner as Segel's original model. The rollover model consisted of sprung and unsprung masses connected through the suspension elements. It also included the aerodynamic forces and moments, and lateral and longitudinal weight transfer.

1.2. Vehicle Stability and Control History

In order to improve vehicle safety and comfort when the vehicle is at the physical limits of its performance, various active control systems such as Anti-lock Braking System (ABS), Four Wheel Steering (4WS), Electronic Stability Control (ESC), and semi-active/active suspensions were developed for vehicles since the late 1970s.

In 1929 the Anti-lock Braking System (ABS) was first developed for aircraft application by Gabriel Voisin, a French automobile and aircraft pioneer. In the 1950s, Dunlop's Maxaret system introduced an early ABS system with 30% improvement in braking performance in some aircraft models [20]. The experiments on the Maxaret anti-lock brake in 1958 demonstrated that anti-lock brakes can be of great value to motorcycles, for which skidding is involved in a high proportion of accidents. With anti-lock brakes, stopping distances were reduced in most of the tests compared with locked wheel braking, particularly on slippery surfaces.

In 1971, a computerized, three-channel, all-wheel anti-skid system with four channels called "Sure Brake" was introduced for passenger cars by Chrysler and the Bendix Corporation [21]. In the same year, General Motors (GM) introduced the "Trackmaster" rear-wheel-only ABS as an option on their rear-wheel drive Cadillac models [22]. In the same year, Nissan offered an EAL (Electro Anti-lock System) as an option on the Nissan President, which became Japan's first electronic antilock braking system [23].

In 1986, Yuhara proposed a design methodology for an Adaptive Rear Wheel Steering Control System (ARWSCS) that maintains desirable vehicle response through computer control regardless of changes in vehicle dynamics [24]. The system controls the rear wheels based on a Self-Tuning Controller (STC) in such a manner that the vehicle follows the prescribed reference model that presents the desired response to driver's input. In the same year, another passive four-wheel steering (4WS) control scheme was introduced by Takiguchi based on a gain-schedule control system in which the rear-to-front steer angle ratio varies as a function of vehicle speed [25]. During very low speed driving, this control system steers the rear wheels in the opposite direction of the front wheels (out-of-phase steering) at the maximum steer angle ratio, whereas at high speed it steers the rear wheels in the same direction as the front wheels (in-phase steering). The out-of-phase steering reducing the turning radius of the vehicle, while in-phase steering at higher speeds provides improved vehicle stability.

In 1987, the earliest innovators of Electronic Stability Controllers (ESC), Mercedes-Benz and BMW, introduced their first traction control systems. Traction control (TCL) works by applying individual wheel braking and throttle to keep traction while accelerating. The system has since evolved into Mitsubishi's modern Active Skid and Traction Control (ASTC) system that was developed to help the driver maintain the intended path through a corner using an on-board computer to monitor several vehicle-operating parameters using various sensors. When too much throttle has been used while taking a curve, engine output and braking are automatically regulated to ensure the proper path through a curve and to provide the proper amount of traction under various road surface conditions. While conventional traction control systems at the time featured only a slip control function, Mitsubishi developed a traction control system that had a preventive (active) safety feature. This improved the course tracing performance by automatically adjusting the traction force, thereby restraining the development of excessive lateral acceleration, while turning. The TCL system's standard wheel slip control function improves traction on slippery surfaces or during cornering. In addition to the TCL's traction control feature, it also works together with Diamante's electronic controlled suspension and four-wheel steering that Mitsubishi had equipped to improve total handling and performance [26].

In 1989, Kanai applied the design theory of model reference adaptive control to the four-wheel steering system and proposed a design method for a system that would adaptively control yaw rate and lateral acceleration [27].

During 1987 and 1992, Mercedes-Benz and Robert Bosch GmbH co-developed a lateral slippage control system called Electronic Stability Control (ESC) [28].

In 1992, BMW, working with Robert Bosch GmbH and Continental Automotive Systems, developed a system to reduce engine torque to prevent loss of control and applied it to the entire 1992 BMW model line [29].

In 1995, automobile manufacturers commercially introduced ESC systems. Mercedes-Benz supplied by Bosch was the first to implement this with their W140 S-Class model. That same year, BMW and Volvo Cars began to offer ESC on some of their models while Toyota's own Vehicle Stability Control system appeared on the Crown Majesta. General Motors (GM) worked with Delphi Corporation and introduced its version of ESC called "StabiliTrak" in 1997 for select Cadillac models [30]. StabiliTrak was made standard equipment on all GM SUVs and vans sold in U.S. and Canada by 2007 except for certain commercial and fleet vehicles.

1.3. Commercial Vehicle Application of Active Safety Systems

The application of active safety systems in commercial vehicles has followed the smaller vehicles. The existence of airbrakes, significantly larger gross vehicle weights (GVWTs), and trailers introduce additional challenges in applying active safety systems in commercial vehicles. Although they use the same general control design philosophy, the specifics of such systems are significantly different than passenger cars. The ABS mandate on Heavy Duty trucks was put in place in the early-1990s, concurrent with an extensive amount of system development work performed independently by suppliers such as Eaton and Wabco on tractor and trailer ABS/TCS systems. Eaton developed a 4- and 6-channel ABS that could operate at both 12 and 24 Volts. A 4-channel trailer ABS unit was available as well. During this period, great strides were made in speed sensors, tone rings, modulator valves, vehicle harnessing, and connectors. Improvements to SAE Recommend Practices J1587 and J1939 also emerged during this period.

The progression of active systems has included simpler systems that take advantage of the same sensors and actuators that are used for anti-lock brakes, and more sophisticated systems that integrate other safety systems. Roll Stability Control (RSC) continually monitors conditions that can lead to rollover (such as excessive speed in curves) and automatically de-throttles the engine, applies the engine brake, and activates the drive and trailer axle foundation brakes to reduce tractor-trailer speed when lateral acceleration exceeds its safe limit. The RSC concept is advanced further in

Electronic Stability Control (ESC) systems by adding the capability of yaw or rotational control. Released in 2005 for commercial vehicles, ESC can reduce the risk of vehicle instability on slippery surfaces during steering, such as in curves and during evasive maneuvers. Through selective braking of the tractor, application of trailer brakes, and de-throttling the engine power, ESC attempts to prevent jack-knife and drift-out conditions.

1.4. Active and Semiactive Suspensions for Improved Ride Comfort and Safety

Active and semiactive suspension refers to systems that can control the motion and dynamics of the sprung and unsprung masses to which the suspension is connected. Although primarily developed for passenger cars, they have had applications in commercial vehicles during the past two decades. The early studies on active suspensions are included in a comprehensive review by Hrovat that was published in 1997 [31]. The studies include well-known approaches such as modal analysis, eigenvalue assignment, model order reduction, nonlinear programming, multi-criteria optimization, and optimal control. Classic methods have also been considered, such as root locus, Bode diagrams, and Nichols plots. A common prerequisite for these methods is access to a well-defined, linear model for the system. For nonlinear systems, it is required to linearize the system about an operating point for designing the controller. The nonlinearities inherent in suspension systems could not be effectively addressed by such approaches. More recent studies suggest using control methods such as sliding mode control [32] and constrained H_{∞} control [33]. The results show that these approaches are able to deal with the negative effects such as force saturation, force nonlinearity, and model uncertainty, although the applications that have been considered are limited to single suspension models, also referred to as quarter-car models. Linear Matrix Inequality (LMI) optimization methods have shown promising results for dealing with these challenges, when full-state feedback systems are possible [34].

The main obstacles for commercialization of active suspensions are their significant power requirements and the fact that most of their benefits can be realized by semiactive suspensions. One of the production systems is the Bose active suspension that is based on linear electromagnetic actuators (motors). For primary suspensions, the main challenge associated with the Bose system is the significant weight that it adds to the unsprung mass. A heavier unsprung mass is considered undesirable in view of vehicle dynamics and stability control. The Bose active suspension can be implemented on seat and cab suspensions for commercial vehicles, offering ride improvements over passive suspensions.

The concept of a semiactive suspension system was first introduced by Karnopp and Crosby in the early-1970s, based on the well-known skyhook control [35, 36]. Semiactive systems have a number of advantages over fully active systems, including significantly lower power requirements

system simplicity, ease of implementation, fail-safe operation, and lower cost. Another important factor is that skyhook control and various offshoots proposed over the years, such as groundhook and hybrid control [37], are model-free control algorithms, where prior knowledge of the system parameters and excitations is not essential for their implementation. These factors have resulted in wide acceptability and implementation of semiactive systems—mostly as semiactive dampers—for vehicle suspensions during the past four decades. A survey of the semiactive suspension efforts through the mid-1990s can be found in the literature [38, 39]. A review paper by Cao et al. provides a more recent review of semiactive systems [40].

The application of semiactive systems for commercial vehicles dates back to the late 1980s when Ahmadian evaluated their suitability for truck suspensions [41, 42]. These systems included mechanically adjustable dampers that could react to the vehicle and suspension dynamics in real time, to adjust the damping force for improved ride comfort. A decade later, studies by Simon and Ahmadian considered the application of magneto-rheological fluids for changing damping force in real time [43, 44, 45, 46]. Collectively, these studies showed that the application of semiactive systems would provide better ride and to some extent, improve vehicle handling.

Other applications of semiactive suspension include cab and seat suspension. Dating back to mid-1990s, a novel seat suspension that uses magneto-rheological dampers with skyhook control was successfully developed and marketed for truck seats. This system, marketed as MotionMaster™, has proven to significantly improve truck ride by reducing vibration transmission through the seat suspension and controlling the vertical motion of the seat. The improved seat motion also results in maintaining better positioning of the driver relative to the vehicle controls, therefore increasing vehicle safety.

In a more recent effort, the researchers at the Center for Vehicle Systems and Safety (CVeSS) at Virginia Tech considered the application of semiactive systems for cab suspensions [47]. Their study showed that such systems could significantly improve truck ride and result in a better synergy between the cab and seat motion, therefore enabling the vehicle operator to have better access to vehicle controls. Several original equipment manufacturers are considering the applications of such systems for on- and off-highway vehicles, mainly for improved ride.

1.5. Outline

The following chapters provide a detailed discussion of controllable suspensions as embodied by active and semiactive systems, and active systems that include RSC and ESC. [Chapter 2](#) provides the concept of passive, semiactive, and active suspensions. It also details the control concepts that commonly are used for semiactive systems. The critical design issues related to semiactive suspensions that use magneto-rheological fluids are addressed. The chapter concludes by

presenting the results of a comprehensive study that shows the improvements achieved by semiactive suspensions in vehicle ride, handling, and stability.

[Chapter 3](#) concentrates on the development of active safety control, namely RSC and ESC, for commercial vehicles. After providing background information on rollover dynamics of vehicles, this chapter provides a detailed discussion of RSC and ESC systems and their similarities and differences. It also provides some of the recent developments related to such systems, including a combined rollover and yaw control system that incorporates sensory and actuation means that are used for conventional ESC systems with suspension control to reduce vehicle body roll.

Although discussing vastly different system concepts, both of the following chapters include discussions on how the vehicle safety, handling, and ride can be improved through controllable systems. [Chapter 2](#) provides the means for doing so through advanced suspensions; [Chapter 3](#) uses active safety control to achieve the same goal.

[Chapter 4](#) provides a summary of the document and discussions on potential future trends for the application of advanced electro-mechanical systems for heavy trucks.

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CHAPTER 2 ADVANCED CONTROLLABLE SUSPENSIONS

This chapter provides the concept of controllable suspensions for vehicles, particularly as it relates to magneto-rheological dampers that have found increased popularity for primary suspensions in the past two decades. After introducing the concept of passive, semiactive, and active systems, the discussions will provide an in-depth analysis of skyhook control and its derivatives for various classes of vehicles. The use of magneto-rheological (MR) fluids as an effective means for changing damping in semiactive suspension is introduced, along with the critical design aspects of MR dampers. The result of a simulation study that provides the trade-off among vehicle safety, handling, and ride comfort is also presented, for various suspension damping and stiffness settings. A comparison between semiactive suspensions with hybrid control and passive suspensions indicates far superior performance for semiactive suspensions. The results highlight the reason for the adaptation of MR suspensions in different passenger vehicle platforms, and their availability for seat and cab suspensions for commercial vehicles.

2.1. Vehicle Primary Suspensions

Primary suspension is the term used to designate those suspension components connecting the axle and wheel assemblies of a vehicle to the frame of the vehicle. This is in contrast to the suspension components connecting the frame and body of the vehicle, or those components located directly at the vehicle's seat, commonly called the secondary suspension. A typical primary suspension consists of a spring and damper. The suspension spring, which is designed to support the static weight of the vehicle, is chosen based on the weight and ride height of the vehicle. The role of the damper is to dissipate the energy at the suspension, to reduce the dynamic peaks that can occur due to the road surface irregularities and vehicle maneuvers, such as body roll and pitch during steering, braking, and acceleration.

2.1.1. Passive Suspensions

Vehicle suspensions can commonly be categorized as passive, semiactive, and active, as illustrated schematically for a single-degree-of-freedom (SDOF) system in Figure 2.1.

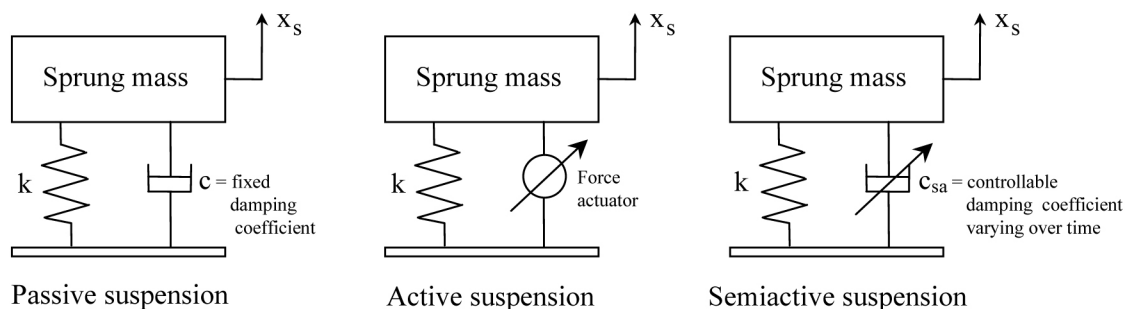
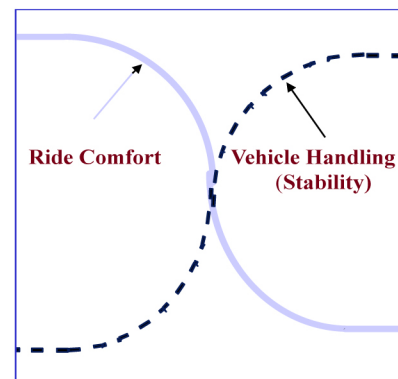


Figure 2.1. Passive, Active, and Semiactive Suspensions

Passive dampers represent the conventional shock absorbers that are used in most vehicles. Their damping characteristics cannot be changed while operating the vehicle, and, as such, they are designed to achieve the most suitable trade-off between ride, handling, and stability, as illustrated in Figure 2.2. A low damping coefficient will result in a ride that is less harsh, but it will reduce the vehicle roll motion stability of the vehicle. A vehicle with a lightly damped suspension will not be able to hold the road as well as with a one more damped suspension. On the other hand, a highly damped suspension provides better road-holding ability at the expense of a harsher ride. The dynamic performance of both a lightly-and highly damped suspension will be discussed further in later sections.



Low Damping

High Damping

Figure 2.2. Passive Suspension Trade-off

A heavily damped suspension will yield good vehicle handling, but also transfers much of the road input to the vehicle body. When the vehicle is traveling at low speed on a rough road or at high speed in a straight line, this will be perceived as a harsh ride. The vehicle operators may find the harsh ride objectionable, or it may damage cargo. A lightly-damped suspension will yield a more comfortable ride, but can significantly reduce the stability of the vehicle in turns, lane change maneuvers, or in negotiating an exit ramp. Good design of a passive suspension can optimize ride and stability to some extent, but cannot eliminate this trade-off.

2.1.2. Active Suspensions

In an active suspension, the passive damper or both the passive damper and spring are replaced with a force actuator, as shown in [Figure 2.1](#). The force actuator is able to both add and dissipate energy from the system, unlike a passive damper, which can only dissipate energy. With an active suspension, the force actuator can apply force independent of the relative displacement or velocity across the suspension. Given the correct control strategy, this results in a better compromise between ride comfort and vehicle stability as compared to a passive system, as shown in [Figure 2.3](#) for a quarter-car model.

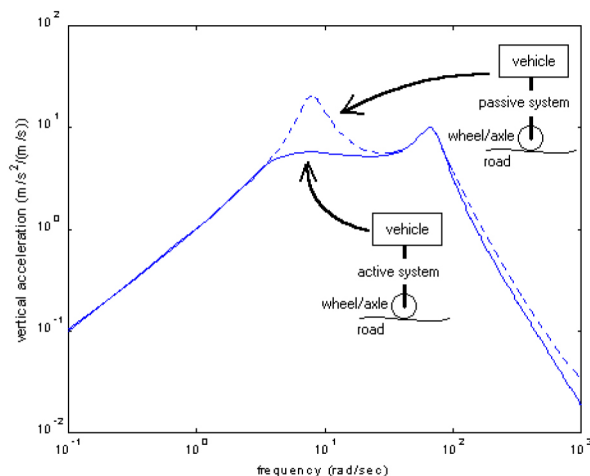


Figure 2.3. Passive and Active Suspension Comparison (adapted from reference [48], p. 201)

The early studies on active suspensions are included in a comprehensive review by Hrovat that was published in 1997 [49]. The studies include well-known approaches such as modal analysis, eigenvalue assignment, model order reduction, nonlinear programming, multi-criteria optimization, and optimal control. Classic methods have also been considered, such as root locus, Bode diagrams, and Nichols plots. A common prerequisite for these methods is access to a well-defined, linear model for the system. For nonlinear systems, it is required to linearize the system about an operating point for designing the controller. The nonlinearities inherent in suspension systems could not be effectively addressed by such approaches. More recent studies suggest using control methods such as sliding mode control [50] and constrained H_∞ control [51]. The results show that these approaches are able to deal with the negative effects such as force saturation, force nonlinearity, and model uncertainty, although the applications that have been considered are limited to quarter-car models. Linear Matrix Inequality (LMI) optimization methods have shown promising results for dealing with these challenges, when full-state feedback systems are possible [52].

The main detractor for commercialization of active suspensions is the significant power requirement. In order to reduce the cost associated with the required power, practical active suspension

designs generally function as a low-bandwidth system that requires a modest 4 peak kw of power for road vehicle applications. One of the production systems is the active body control (ABC) from Mercedes-Benz. It uses a skyhook control approach that works in the primary ride mode. The ABC system, however, does not include an energy-regeneration function. Other systems that have been developed are the Bose active suspension that is based on linear electromagnetic actuators (motors). The main challenge associated with the Bose system is the significant weight that it adds to the unsprung mass, when compared to the ABC system. A heavier unsprung mass is considered undesirable in view of vehicle dynamics and stability control. Another promising system is the suspension developed by Visteon Corp., which includes compressible fluid struts and a digital displacement pump motor [53, 54]. This system takes advantage of the fluid flow to achieve power regeneration for energy harvesting. It uses a frequency-domain control approach to synthesize established suspension controls to achieve the desired dynamic performance.

In a study by Chalasani [48], the single suspension model in [Figure 2.4](#) is used to investigate the performance gains possible with an active suspension system. In this study, the road input is modeled as white-noise velocity input. The study found that within practical design limitations, an active suspension reduces the root mean square (RMS) acceleration of the sprung mass by 20%. This suspension configuration exhibited approximately the same amount of suspension travel and wheel-hop damping ratio as a lightly damped, soft-passive suspension. In a later study [55], similar simulations and analysis were performed for a full-car model. This study estimated that active suspensions could reduce the RMS acceleration of the sprung mass by 15%.

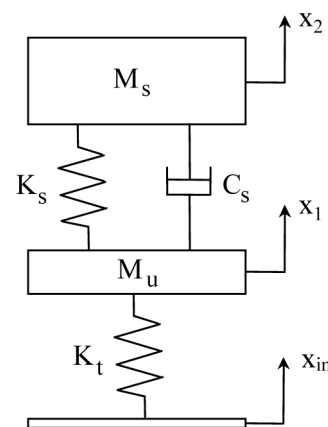


Figure 2.4. Single Suspension (Quarter-car) Model.

Active suspension systems have the added advantage of controlling the attitude of a vehicle. They can reduce the effects of braking, which causes a vehicle to nose-dive, or acceleration, which causes a vehicle to squat. They also reduce the vehicle roll during cornering maneuvers.

A comprehensive study of active suspensions for improving vehicle ride and handling can be found in a study by Williams, et al. [56]. This study evaluates the benefits of the Lotus active suspension, which was successfully implemented in Formula 1 race cars and a few prototype passenger cars in early- to mid-1990, for passenger vehicles.

2.1.3. Adjustable Suspensions

An adjustable suspension combines the passive spring element found in a passive suspension, with a damper element whose characteristics can be adjusted by the operator. As shown in Figure 2.5, the vehicle operator can use a selector device to set the desired level of damping based on their subjective feel.

This system has the advantage of allowing the operator to occasionally adjust the dampers according to the road characteristics and driving conditions. It is, however, unrealistic to expect the operator to adjust the suspension system to respond to time inputs such as potholes, turns, or other common road inputs.

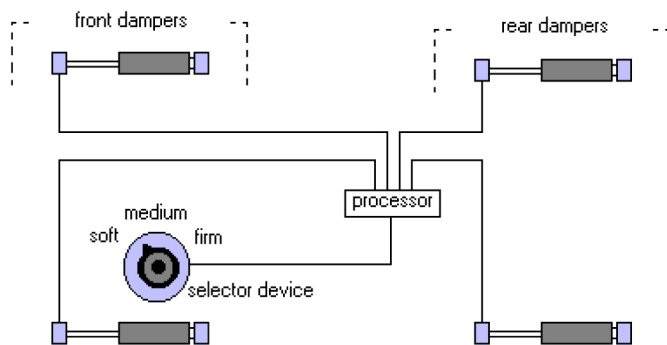


Figure 2.5. Adjustable Suspension (adapted from reference [57], p. 107)

2.1.4. Semiactive Suspensions

Semiactive suspension systems, first proposed in the early-1970's by Crosby and Karnopp, retain the spring element and replace the damper with a controllable damper [58, 59].

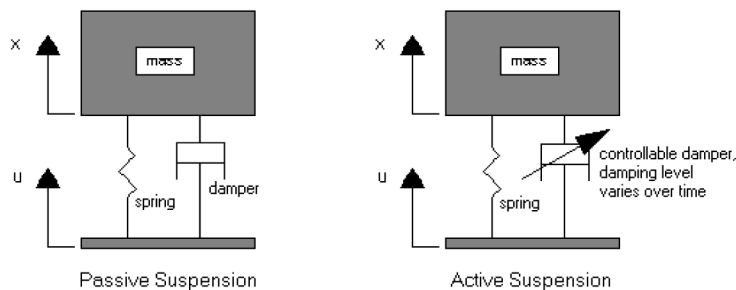


Figure 2.6. Passive and Semiactive Suspensions

Semiactive suspensions include dampers that can change their damping characteristics in real time, with sufficiently high dynamic bandwidth to favorably influence the suspension

dynamics. Semiactive dampers use a minimal amount of energy, sufficient to change the damping characteristics. It is important to keep in mind that even though semiactive dampers are "actively controlled," at each instant in time, the direction of the force exerted by the damper still depends on the relative velocity across the suspension. This is in contrast to a fully active system in which the suspension forces can be made completely independent of the suspension's relative motion. One can think of a semiactive damper as a passive damper, with the ability to change its damping characteristics in real time, using a small amount of external power. The change in damping force is determined by a control algorithm based on the information received from the sensors. Although semiactive suspensions are more expensive than passive suspensions, they are far less expensive than active suspensions; as such, they are considered as a more commercially viable option for all vehicles including commercial trucks. Various studies have shown that semiactive suspensions have because of their significantly better dynamic performance.

Skyhook Control

One of the most common semiactive control policies is skyhook control that adjusts the damping force to emulate the damping force of a passive damper in its optimal configuration, connected between the sprung mass and a stationary reference, i.e., the sky, as shown in Figure 2.7.

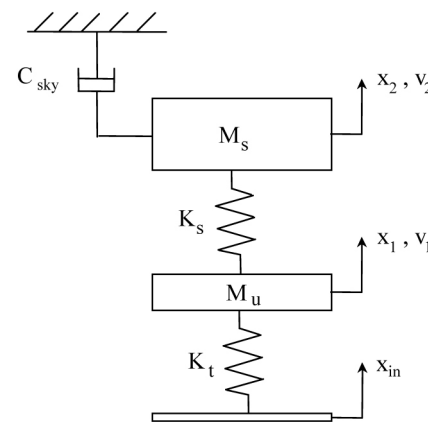


Figure 2.7. Skyhook Damper Configuration

In its simplest form, skyhook control can be described as a two-stage damper (on or off) that is controlled according to a switching control such as:

$$\left\{ \begin{array}{ll} v_2 v_{21} \geq 0 & F_{sa} = C_{\max} v_{12} \\ v_2 v_{21} < 0 & F_{sa} = 0 \end{array} \right\}$$

(2.1)

In this equation $\dot{x}_1$ is the velocity of the sprung mass and $\dot{x}_2$ is the velocity of the unsprung mass. Equation (2.1) indicates that when the sprung mass is moving up, and the two masses are getting closer, the damping constant should ideally be zero.

Due to the physical limitations of a practical damper, a damping constant of zero is not practical and a low damping constant is used. When the sprung mass is moving down and the two masses are getting closer, the skyhook control policy ideally calls for the largest damping force possible. The skyhook control force concept is further illustrated in Figure 2.8. Dynamically, skyhook control attempts to minimize the absolute velocity of the sprung mass.

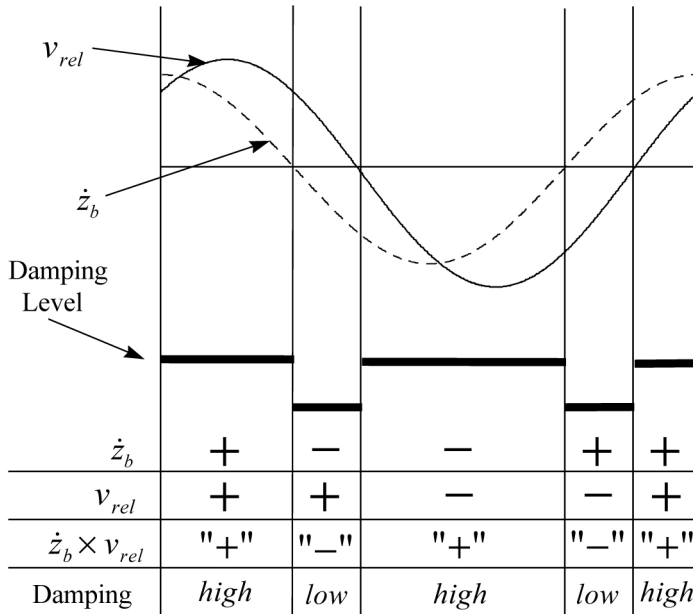


Figure 2.8. On-off Skyhook Control Illustration

It has been shown that a continuously variable semiactive suspension system is able to achieve performance comparable to that of an optimized passive damper, hooked to the sky and the sprung mass. In a continuously variable damper, the damping force is not limited to the maximum and minimum forces. It can assume any force in between the two, as shown by the shaded damping range in Figure 2.9

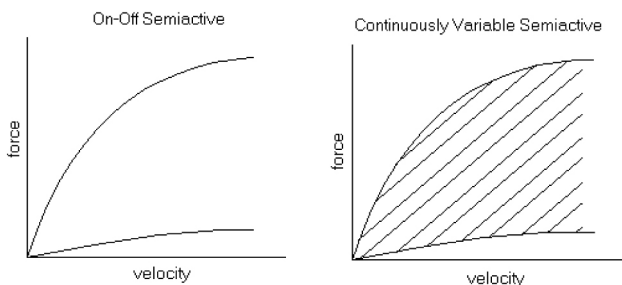


Figure 2.9. On-off and Continuously Variable Semiactive Damping

For a continuously variable damper (shown in Figure 2.9), the skyhook control in Eq. (2.1) can be re-written as:

$$\left\{ \begin{array}{ll} v_2 v_{21} \geq 0 & F_{sa} = C_{sky} v_2 \\ v_2 v_{21} < 0 & F_{sa} = C_{min} v_2 \end{array} \right\} \quad (2.2)$$

Equation (2.2) indicates that when the product of the sprung mass velocity and suspension relative velocity is positive, then the semiactive damping force is proportional to the velocity of the sprung mass. Otherwise, the semiactive damping force is at a minimum. Simulating the two velocities with a sinusoidal function, the semiactive skyhook control policy of Equation (2.2) closely follows the "ideal skyhook" damping shown in Figure 2.10, except at the zero crossings of the relative velocity.

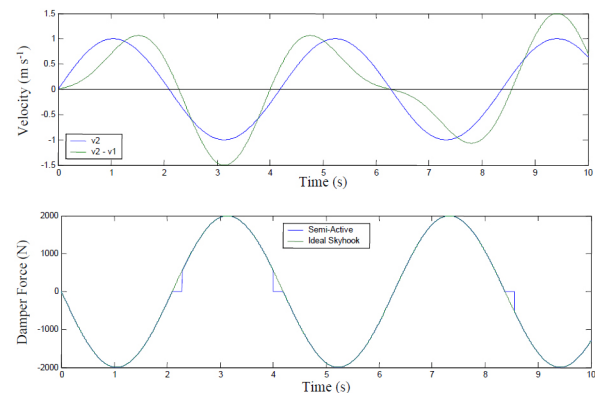


Figure 2.10. Comparison between semiactive skyhook control and "ideal skyhook" implementation

Using the commonly-used single suspension model (also known as "quarter-car" model) in Figure 2.4, one can describe the dynamics of the sprung and unsprung mass according to:

$$\begin{bmatrix} M_s & 0 \\ 0 & M_u \end{bmatrix} \begin{Bmatrix} \ddot{x}_2 \\ \ddot{x}_1 \end{Bmatrix} + \begin{bmatrix} C_s & -C_s \\ -C_s & C_s \end{bmatrix} \begin{Bmatrix} \dot{x}_2 \\ \dot{x}_1 \end{Bmatrix} + \begin{bmatrix} K_s & -K_s \\ -K_s & K_s + K_t \end{bmatrix} \begin{Bmatrix} x_2 \\ x_1 \end{Bmatrix} = \begin{bmatrix} 0 \\ K_t \end{bmatrix} x_{in} \quad (2.3)$$

The system input, x_{in} , emulates the road profile. The input excites the unsprung mass, which represents the wheel, tire, and some suspension components, through a spring element that models the tire stiffness. The unsprung mass is connected to the sprung mass, representing the portion of the vehicle supported on the suspension, through the primary suspension that is modeled as a spring and damper.

The transmissibility of the system is shown in Figure 2.11, for various passive damping coefficients. The first plot shows the displacement of the sprung mass (x_2) with respect to the road input (x_{in}), while the second plot shows the displacement of the unsprung mass (x_1) with respect to the input (x_{in}).

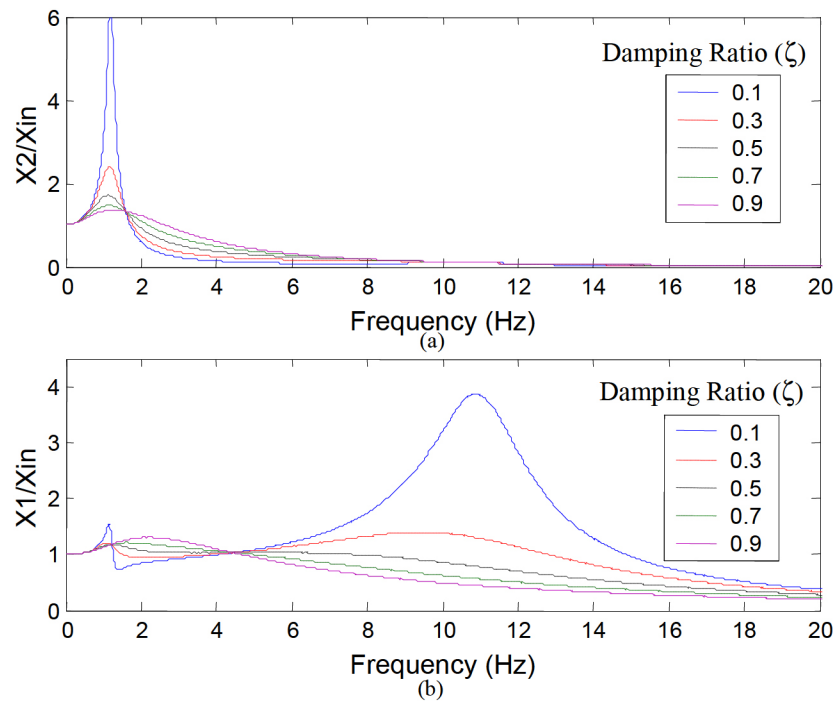


Figure 2.11. Passive Suspension Transmissibility; (a) Sprung Mass Transmissibility; (b) Unsprung Mass Transmissibility

At low passive damping, the resonant transmissibility near $\omega=\omega_{n1}$ (or 1.5 Hz) and $\omega=\omega_{n2}$ (or 10.5Hz) is relatively large, while the transmissibility at higher frequencies is quite low. As damping increases, the resonant peaks are attenuated, providing lower vibrations at these frequencies. The vibration isolation at higher frequencies and in the frequency range in between the two peaks, however, is reduced as a result of higher damping. The amplitude reduction at the two natural frequencies results in less resonance of the sprung and unsprung mass, while the reduced isolation at higher frequencies will result in a harsher vehicle ride. The plots graphically illustrate the inherent trade-off between handling (resonance control) and ride comfort (high frequency isolation) inherent to passive suspensions.

Repeating the same transmissibility analysis for the skyhook control in Eq. (2.2) results in Figure 2.12, for different values of the skyhook-damping coefficient C_{sky} . As the skyhook damping ratio increases, the resonant transmissibility near ω_{n1} decreases, even to the point of isolation, but the transmissibility near ω_{n2} increases. The skyhook configuration is adding more damping to the sprung mass and taking away damping from the unsprung mass, making it ideal for a much-improved ride, while causing excessive unsprung mass motion [60].

Groundhook Control

First introduced by Ahmadian [61], groundhook control differs from skyhook in that the damper is now connected to the unsprung mass rather than the sprung mass, as shown in Figure 2.13.

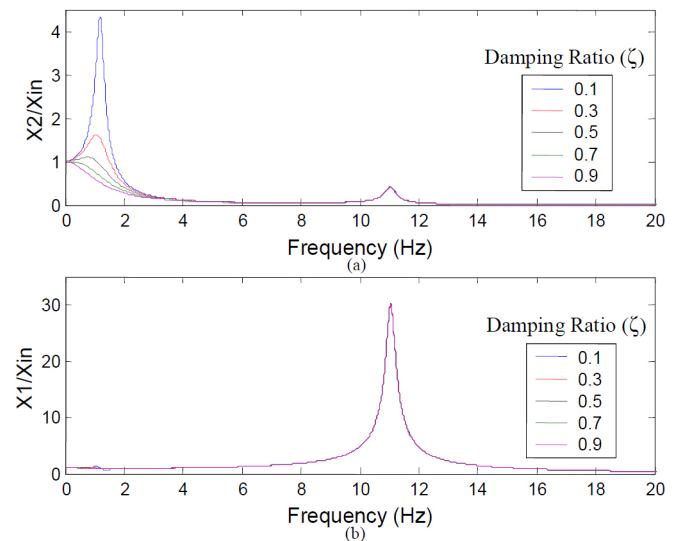


Figure 2.12. Skyhook Configuration Transmissibility; (a) Sprung Mass Transmissibility; (b) Unsprung Mass Transmissibility

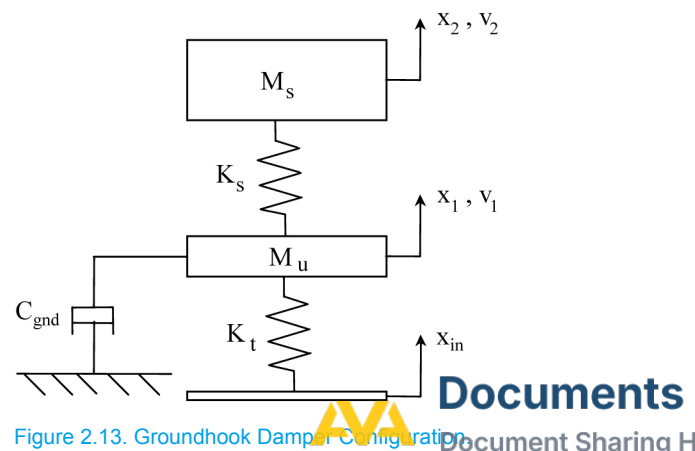


Figure 2.13. Groundhook Damper Configuration

Through the same reasoning used for skyhook, groundhook control can be expressed as:

$$\begin{cases} -v_1 v_{21} \geq 0 & F_{sa} = C_{gnd} v_1 \\ -v_1 v_{21} < 0 & F_{sa} = 0 \end{cases} \quad (2.4)$$

In this configuration, the focus shifts from the sprung mass to unsprung mass. Whereas skyhook performed well in controlling the sprung mass, groundhook is effective in controlling the unsprung mass excitations, as shown in transmissibility plots in [Figure 2.14](#).

Hybrid Control

An alternative semiactive control method known as hybrid control has been shown to take advantage of the benefits of both skyhook and groundhook control, as discovered by Ahmadian [62]. With hybrid control, shown in [Figure 2.15](#), one has the ability to specify how closely the controller emulates

skyhook or groundhook. In other words, hybrid control can divert the damping energy to the bodies in a manner that eliminates the compromise that is inherent in passive dampers.

Combining [equations \(2.3\)](#) and [\(2.4\)](#), we arrive at the semiactive hybrid control method:

$$\begin{cases} v_2 v_{21} \geq 0 & \sigma_{sky} = v_2 \\ v_2 v_{21} < 0 & \sigma_{sky} = 0 \end{cases} \quad \left\{ F_{sa} = G [\alpha \sigma_{sky} + (1 - \alpha) \sigma_{gnd}] \right\}$$

$$\begin{cases} -v_1 v_{21} \geq 0 & \sigma_{gnd} = v_1 \\ -v_1 v_{21} < 0 & \sigma_{gnd} = 0 \end{cases} \quad (2.5)$$

where σ_{sky} and σ_{gnd} are the skyhook and groundhook components of the damping force. The variable α is the relative ratio between the skyhook and groundhook control, and G is a constant gain. As the transmissibility plots in [Figure 2.16](#) shows for a damping ratio of 0.3, when $\alpha = 1$, the control method reduces to pure skyhook, whereas when $\alpha = 0$, the hybrid control becomes the same as groundhook.

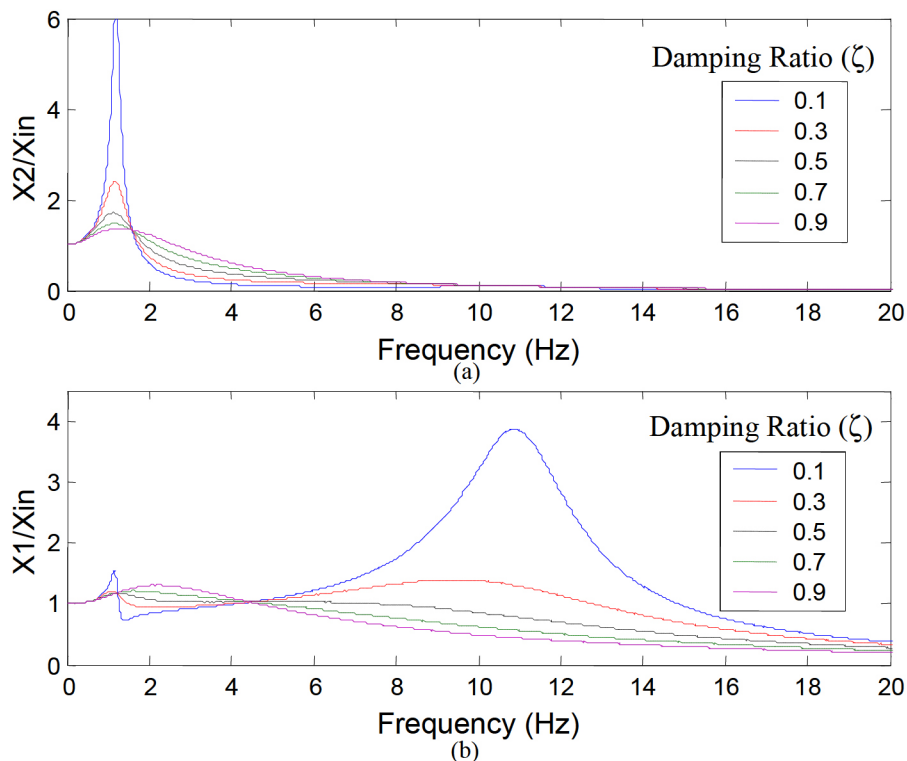


Figure 2.14. Groundhook Configuration Transmissibility; (a) Sprung Mass Transmissibility; (b) Unsprung Mass Transmissibility

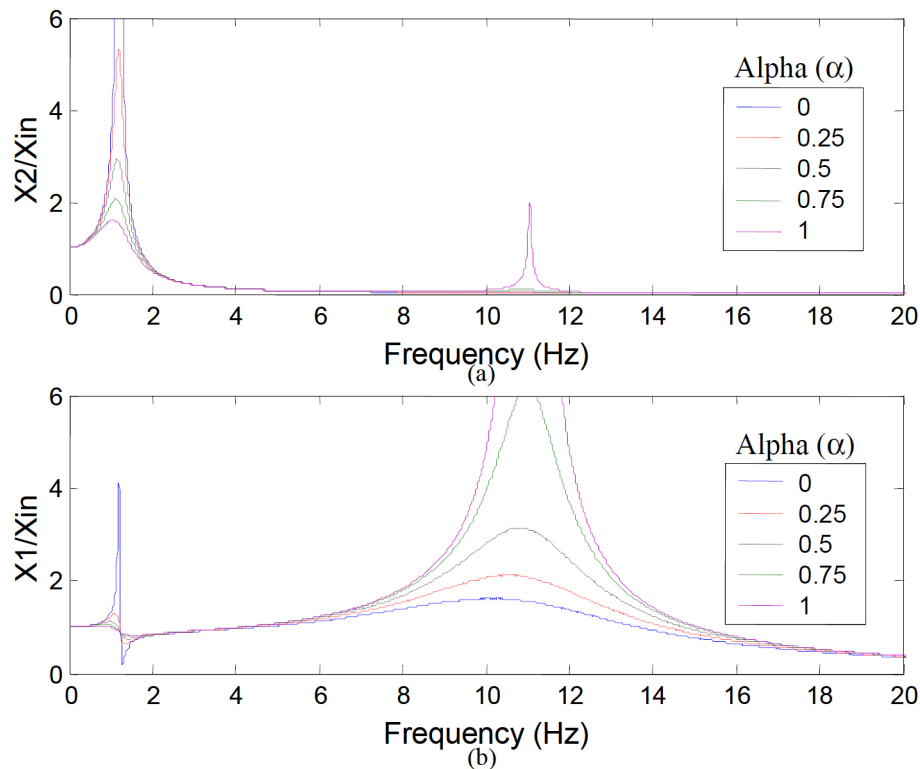


Figure 2.16. Hybrid Configuration Transmissibility; (a) Sprung Mass Transmissibility; (b) Unsprung Mass Transmissibility

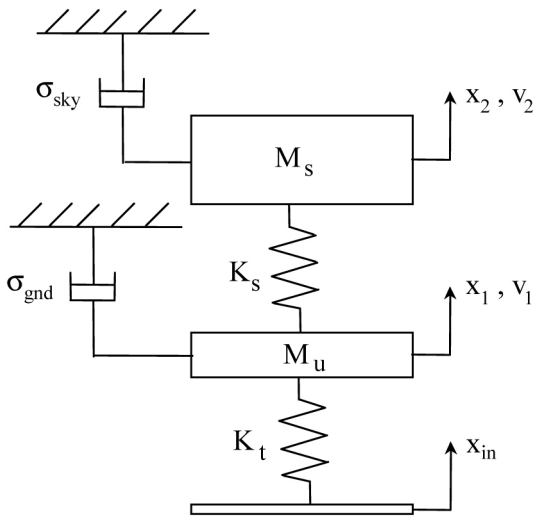


Figure 2.15. Hybrid Control Configuration

2.2. Passive vs. Semiactive Dampers

The previously-mentioned benefits of semiactive dampers over passive dampers are clearly evident if we compare the transmissibility plots for passive, skyhook, groundhook, and hybrid damping. Figure 2.17 shows the transmissibility of each at a damping ratio of 0.3. The hybrid control transmissibility is shown with an alpha of 0.5. As shown there, as compared with a passive suspension, skyhook control performs well in controlling the sprung mass resonance dynamics, while

groundhook control does the same at the unsprung mass resonance frequency. Hybrid control is able to perform well at both resonance frequencies, and as such provides a better option for semiactive suspensions, in comparison with other control methods.

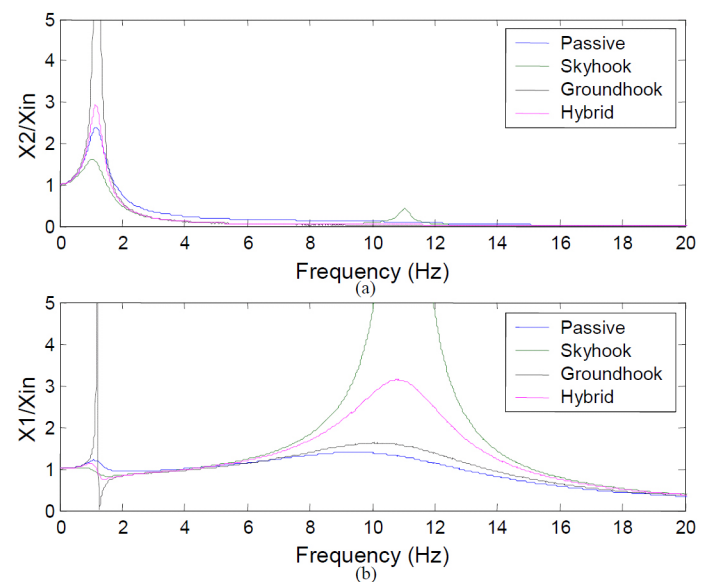


Figure 2.17. Transmissibility Comparison of Passive and Semiactive Dampers; (a) Sprung Mass Transmissibility; (b) Unsprung Mass Transmissibility