POLITECNICO DI TORINO

Master's Degree in Automotive Engineering

Master's Degree Thesis

SUBJECTIVE EVALUATION OF VEHICLE SEMI-ACTIVE SUSPENSION FOR IMPROVED RIDE AND HANDLING

Supervisor

Prof. NICOLA AMATI

Candidate

ZACHARY SINASAC

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DECLARATION OF ORIGINALITY

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ABSTRACT

The number of passenger cars currently equipped with semi-active suspensions has been steadily increasing in recent decades. These suspension systems provide an improvement in ride and handling when compared to passive suspensions. Currently, the approach to evaluating and tuning semi-active suspensions has been limited to objective methods or time-consuming alterations made on physical components. To alleviate the time and costs and improve the fidelity of such methods, a novel solution to subjectively evaluating vehicle semi-active suspensions is presented. The subjective evaluation method herein involves the use of a state-of-the-art dynamic driving simulator with drivers to subjectively evaluate and tune virtual semi-active suspensions.

To consider the results of the proposed evaluation method accurate, high-fidelity vehicle models supplied by an OEM are studied. These vehicle models have previously been validated with objective and subjective performance data by an OEM's expert drivers. First, offline co-simulations between VI-grade's CarRealTime vehicle simulation software and several versions of a Simulink semi-active suspension controller are completed to objectively evaluate ride and handling. The semi-active suspension controller is based on several well-known control strategies and incorporates the vehicle's passive suspension settings as one of the suspension modes. This feature permits a comparison between the passive and semi-active suspensions in terms of ride and handling.

For the subjective evaluation, the vehicle and controller models are uploaded in a driver-in-the-loop environment. Expert drivers then execute a series of maneuvers and provide subjective feedback on the ride and handling of the different suspension modes. A questionnaire is implemented involving a list of subjective metrics tailored for ride and handling of semi-active suspensions. Furthermore, a correlation between changes in objective and subjective metrics is made to determine where correlation exists and to suggest predictive methods for future subjective ratings. A specific evaluation procedure is presented to ensure a bias among drivers is removed.

The results of the subjective evaluation method prove that the method is effective at capturing relatively small changes in ride and handling, in a timely manner. The subjective ratings from the drivers showed acceptable agreement and considered many ride and handling improvements as major differences according to SAE standards. The correlation study identified a list of strong correlations between objective and subjective metrics. These results can be used to predict subjective performance when implementing offline changes to suspensions.

DEDICATION

Never underestimate the support you're given in life and surround yourself with individuals who inspire you to push forward and achieve your goals. This thesis is dedicated to the beings who ran alongside me since the first day of undergrad. This includes my family, close friends, colleagues, professors, and canine companion who always patiently listened to my struggles and answered my abundant list of questions. Without their support, I could never have reached my goals.

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Mohammed Malik; Technical Supervisor; Stellantis Automotive Research and Development Centre (ARDC)

Mohammed ("M0") was the backbone of the project. With his years of knowledge, experience, and expertise he always ensured the project remained on a path consistent with industry practice. Without his vision, critiquing, and providing of the opportunity to work with his team in the simulator lab, this project would never have been completed. This project has been the best part of my academic engineering career.

Bharath Kaimal; Former Engineer at Stellantis' ARDC

For a majority of the project, Bharath was the primary source of providing me with industry knowledge and practices. He explained practical approaches to evaluating ride and handling, many aspects of vehicle dynamics, and continuously explained the working principles behind the software and hardware involved with the driver-in-the-loop environment. His commitment and patience with the project was outstanding.

Stellantis ARDC Engineers

Several engineers including Marie Mills, Harry Zheng, Arnalld Kurusumuthu and others were crucial to the continued progress of the project. They aided in understanding theoretical concepts, completing the subjective evaluations, controller debugging, along with the organization and communication between the many individuals involved in the project. Considering the global circumstances, they strived to ensure the completion of the final evaluations on the simulator. Thank you to them for being a part of this project.

Global Stellantis Engineers

Elena Salino was another technical supervisor for the project who provided key insight on OEM evaluation practices as well as the secondary vehicle model for the project. Ryan Haveman and Prabhu Madabhusi-Raman also gave insight on industry practices for objective and subjective evaluations. In part, the developed evaluation methodologies used herein are a result of their input.

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LIST OF ABBREVIATIONS

ARDC – Automotive Research and Development Centre
OEM – Original Equipment Manufacturer
ISO – International Organization for Standardization
NVH – Noise, Vibration, and Harshness
CAE – Computer Aided Engineering
MBS – Multibody Systems
DOF – Degrees of Freedom
FEM – Finite Element Method
COS – Continuous System
CRT – CarRealTime
DiM – Driver-in-Motion
NADS-1 – National Advanced Driving Simulator
ADAS – Advanced Driver Assistance Systems
ESC – Electronic Stability Control
HMI – Human Machine Interfaces
V2X – Vehicle to Everything
DIL – Driver-in-the-Loop
MCA – Motion Cueing Algorithm
MR – Magnetorheological
K&C – Kinematics and Compliance
MF – Magic Formula
RMS – Root-mean-square
P2P – Peak-to-peak
SWS – Suspension Working Space
DTL – Dynamic Tire Loading
SWA – Steering Wheel Angle
VSC – Variable structure control
GM – General Motors
DLC – Double Lane Change
HPG – Hällered Proving Ground
CES – Controlled Electronic Suspension
RTDBs – Real time databases
SimWB – SIMulation Workbench
MF – Magic Formula
MF-SWIFT – MF Short Wavelength Intermediate Frequency Tire Model
SUV – Sport Utility Vehicle
CPG – Chelsea Proving Grounds
DCC – Dynamic Chassis Control

NOMENCLATURE

Symbol	Description
M_s	Sprung mass
M_u	Unsprung mass
K_s	Suspension stiffness, primarily due to the spring
C_s	Suspension damping, primarily due to the dampers
K_t	Tire stiffness
z_s	Sprung mass vertical displacement
z_u	Unsprung mass vertical displacement
z_r	Road vertical profile; input road vertical displacement
F_b	Force on the sprung mass, due to the loading of the vehicle
m	Bicycle model vehicle mass
v	Bicycle model centre of gravity velocity
ψ_B	Bicycle model yaw displacement (heading)
β	Bicycle model side slip angle
$F_{A,y}$	Bicycle model rear tire lateral force
$F_{B,y}$	Bicycle model front tire lateral force
l_A	Bicycle model distance from rear tire centre to centre of gravity
l_B	Bicycle model distance from front tire centre to centre of gravity
$\theta_{b.m.}$	Bicycle model moment of inertia
δ	Bicycle model front tire steering angle
m_f	Vibrating bicycle model front unsprung mass
m_r	Vibrating bicycle model rear unsprung mass
I_y	Vibrating bicycle model sprung mass moment of inertia
C_f	Vibrating bicycle model front damping rate
C_r	Vibrating bicycle model rear damping rate
a_1	Vibrating bicycle model distance from front to centre of gravity
a_2	Vibrating bicycle model distance from rear to centre of gravity
K_f	Vibrating bicycle model front spring stiffness
K_r	Vibrating bicycle model rear spring stiffness
$K_{t,f}$	Vibrating bicycle model front tire stiffness
$K_{t,r}$	Vibrating bicycle model rear tire stiffness
CG	Centre of gravity
φ	Sprung mass roll angle
K_r	Anti-roll bar stiffness
b_1	Half-car model distance from right wheel to centre of gravity

b_2	Half-car model distance from left wheel to centre of gravity
I_x	Half-car sprung mass moment of inertia
ρ	Twin track model wheel rotation
δ	Twin track model front wheels' steering angle
V_S	Sprung mass vertical velocity
V_U	Unsprung mass vertical velocity
V_D	Damper relative velocity
F_D	Damper force
$t_{90\%}$	Time for a system response to reach 90% of its steady state
$t_{\pm 5\%\, steady\ state}$	Time for a system response to remain within 5% of its steady state
$t_{10\%\, steady\ state}$	Time for a system to reach 10% of its steady state value
δ_{steer}	Steering angle
L	Vehicle wheelbase
R	radius of steady-state cornering or turn
A_y	Vehicle lateral acceleration
U	Input for semi-active suspension
$C_{passive}$	Passive suspension damping coefficient
C_{sky}	Skyhook semi-active suspension damping coefficient
C_{ground}	Groundhook semi-active suspension damping coefficient
F_{sky}	Skyhook semi-active suspension damping force
F_{ground}	Groundhook semi-active suspension damping force
α	Hybrid semi-active control strategy weighting parameter
F_{hybrid}	Hybrid semi-active suspension damper force

CHAPTER 1. Introduction

The evaluation of vehicle performance has been a subject of study in the automotive industry for decades. Vehicle dynamics experts, professional drivers, university research groups, and many more have spent countless hours developing their products with a goal to improve the ride and handling experience for the driver and passengers, among other aspects. Alongside passenger vehicles, the development process has also transformed into a complex structure of engineering and design. During the design stage of a vehicle, before a physical prototype is produced, virtual engineering tools such as simulation and modelling software with driving simulators are used to create mathematical representations of vehicle subsystems and evaluate their performance.

Evaluating the performance of vehicles is comprised of a combination of simulation and physical testing. When simulating a vehicle model, dynamic variables can be recorded and post-processed to quantify its performance. This can also be done during physical testing of a vehicle with the addition of sensors and instrumentation. Moreover, a subjective study can be done where the driver of the vehicle uses their knowledge to evaluate and "rate" the driving experience. Physical testing has the drawback of requiring favourable weather conditions, human drivers in many cases, and an abundance of time if changes to a vehicle are required in between evaluations.

In 2019, Stellantis' Automotive Research and Development Centre (ARDC) in Windsor, Ontario started using a dynamic driving simulator, which allows the OEM to evaluate and rapidly apply changes to vehicle models before the need for prototypes or physical testing. The simulator allows engineers and designers alike to subjectively evaluate the driving experience throughout the development of a vehicle. As an example, the dampers of the front and rear suspensions can be altered to study the effect on a vehicle's ride and handling performance. The novel research herein focuses on studying a semi-active suspension while considering objective and subjective evaluation metrics for vehicle ride and handling. The use of ARDC's dynamic driving simulator provides a powerful tool for evaluating ride and handling for different damper settings, determining the sensitivity of the simulator to such changes, and developing procedures for subjective evaluation methods.

The remainder of this chapter presents concepts and information to introduce the reader to several subjects relevant to the research in this project. General concepts of vehicle performance, tools, and applications utilized in the automotive industry, as well as some terminology will be discussed. Finally, the contribution, novelty, and outline of the thesis is presented.

1.1 Vehicle Performance – Ride and Handling

Differing definitions of ride and handling exist amongst literature. For the sake of the reader and for the remainder of this work, the generic definition of ride is taken from an ISO standard. Ride is considered as the motion environment involving the vehicle's vibration, shock, and translational and rotational accelerations in response to road excitations [1]. As a result, ride is referring to the translational and angular accelerations felt by the driver in the vertical, lateral, longitudinal, as well as the pitch and roll directions. These accelerations originate from road irregularities which displace the tires of the vehicle. Eventually, the vibrations are transmitted to the driver. Depending on the level of damping characteristics of a given vehicle, these vibrations can be attenuated, significantly due to the damping characteristics of the suspension. Ride can be broken down into subcategories based on the frequency of motion such as primary and secondary ride [2], [3]. Primary ride consists of motion in the frequency range from 0.5 to 3 Hz containing high amplitude and low frequency heaving, pitching, and rolling motions [4]. Secondary ride encompasses motions in the frequency range of 5 to 20 Hz, involving suspension and body shake [4]. Vehicle motion with higher frequency content is considered Noise, Vibration, and Harshness (NVH), which is not considered in this work.

Handling is another area of study regarding vehicle performance. According to [5], vehicle handling considers the response of a vehicle to inputs from the driver. Driver inputs are the steering wheel angle, the position of the accelerator or throttle, and the position of the brake pedal. The responsiveness of the vehicle generally consists of the magnitude of the vehicle's response characteristics and the delay of the response to driver inputs. For instance, given a steering wheel input from the driver, the vehicle will yaw accordingly. Depending on how much and how soon the vehicle yaws is one aspect pertaining to vehicle handling. Other sources such as [6], [7] expand on handling to describe the lateral behaviour of a vehicle, the road holding ability pertaining to tire grip, and the agility and preciseness of the vehicle's response during maneuvering.

1.1.1 What is Better for the Consumer

After a vehicle has been produced and shipped to a dealership, the consumer's perception of the vehicle's ride and handling plays a role in the decision to buy a vehicle. If a vehicle dissipates road irregularities smoothly and responds to driver inputs in a way that promotes the driver's confidence, then the driver could perceive the vehicle as more attractive than other vehicles with lesser performance. The ability of a vehicle's tires to maintain grip while driving also contributes to the driver's perception of ride and handling, as well as passenger safety. For vehicle ride, the ideal vehicle would attenuate all road irregularities so that the driver does not feel bumps, potholes, cracks, and other obstacles. Thus, it is advantageous to the seller, the consumer, and the Original Equipment Manufacturer (OEM) to produce a vehicle with good ride and handling performance. Typically, OEMs rigorously evaluate physical vehicles through universally standardized and internal testing to ensure these characteristics exist in their vehicles.

From a development point-of-view, it would be advantageous if OEMs could have a subjective method to evaluate the ride and handling of vehicle designs before having to manufacture the vehicle. In this way, experienced or everyday drivers could give their opinions of a vehicle before money and time is spent on the final product. A dynamic driving simulator is an application which supports this method of development and evaluation [8]. The consumer would benefit from such methods since vehicle would be produced in a timely fashion and with a lower development cost.

1.1.2 Ride and Handling: Traditional Damper Tuning and Compromise

Conventional suspension systems have been designed to provide a compromise in vehicle ride and handling [9]. There exists a high number of possible configurations of driving maneuvers, vehicle speed, vehicle loading, and road profiles that make it difficult to tune a suspension for high performance in both ride and handling in all cases. Dampers are one component of vehicle suspension that are tuned for ride and handling. For better secondary ride, lower damping rates of the dampers have been found advantageous whereas higher rates promote better handling and primary ride [7]. Choosing an appropriate compromise between the two trends depends on the type and trim of the vehicle, among other aspects. Typically, OEMs have internal standardized laboratory procedures which outline a set of maneuvers to evaluate a vehicle's ride and handling performance objectively, or subjectively. When physically testing the performance of different dampers, onsite damper suppliers will exchange dampers with different characteristics on a given vehicle and a driver will evaluate the vehicle performance with said dampers. Some dampers have manual knob adjusters to adjust their damping characteristics, meaning the dampers do not have to be removed and others re-installed [10]. Figure 1 shows an example of a manually adjustable damper. These dampers contain adjustable valves for the compression and extension characteristics of the damper. These valves are meant for fine-tuning adjustments. In either case, time is consumed when making such adjustments to the vehicle suspension. The work presented herein will show that a dynamic driving simulator can reduce the time for tuning vehicle suspension while subjectively evaluating the ride and handling performance of vehicles in development or production.

Figure 1: Bump adjustment knob on the bottom of a conventional fluid damper [10]

1.2 Introduction to Modelling

Modelling real systems using computer-aided engineering (CAE) provides another powerful tool in product development. Through modelling, a real system can be simplified and broken down into mathematical relationships that describe its kinetics and kinematics. In this way, the performance of systems can be observed virtually while avoiding the time and cost of creating physical models. Modelling software aid in the generation of such models. There are three common approaches in modelling technical systems as discussed in [11]. The first approach considers Multibody Systems (MBS) where a real system is modelled as a group of rigid bodies connected by joints or links. The rigid bodies have mass and inertia properties. Forces and moments in such systems are concentrated and act at discrete points on the bodies. Upon completion of modelling, MBS are described by ordinary differential systems, differential algebraic systems, or a combination of the two. These systems typically have lower degrees of freedom (DOF) [11]. The second approach considers the Finite Element Method (FEM). With

FEM, relatively simple geometries are represented by a mesh of discretized elements having mass and stiffness properties. For such systems, the mass and elasticity are continuously distributed [11]. FEM modelling results in a set of ordinary differential equations with many DOF, and it is typically used to study the stress and deformation of bodies under an applied load. The third modelling approach is for Continuous Systems (COS) having a continuous distribution of mass, elasticity, and plasticity [11]. These systems are described by partial differential equations having an infinite number of DOF. In the automotive industry, MBS modelling is the most common approach since it is best for modelling vehicles and describing vehicle dynamics.

1.2.1 Automotive Modelling Applications

Modelling vehicles and their dynamics as MBS is a common application in the automotive industry. The moving components of a vehicle are grouped into subsystems which define several major operations in a vehicle. The subsystems are comprised of the drivetrain, chassis or body, front and rear suspensions, braking system, steering system, and wheels. Each of these systems has a unique function to permit the vehicle to move and handle inputs from a driver model and road irregularities. Driver models act as one input to the dynamics of a vehicle, the road profile being another. The accelerator position, brake position, and steering wheel angle are inputs to the vehicle coming from the driver model. Focusing on the suspension subsystems, disturbances originating from the road causes the springs and dampers also to be deformed. In MBS, the springs and dampers are represented as applied forces where the spring force is proportional to spring deformation and the damper force is proportional to the rate of the damper deformation. Conveniently, in many software applications, components of subsystems such as dampers can be interchanged with control models. This process will be discussed in detail in Chapter 2. Returning to MBS, each of the vehicle subsystems are connected by joints which can constrain the motion of adjacent bodies and transmit forces or moments between them. Therefore, a vehicle is represented as a virtual model containing a group of unique subsystems which function together to describe the vehicles dynamics. The model can be based on a real vehicle in production, or a design iteration in vehicle development.

Adams Car is one highly reputable example of a MBS vehicle dynamics modeling software, created by MSC Software [12]. Adams Car allows engineers to develop virtual models, perform analyses on and tune the vehicle’s subsystems, and execute virtual driving maneuvers to mimic real world testing. These virtual tests can be done in a fraction of the time it takes engineering teams to complete physical testing [13]. Figure 2 presents an example of what an Adams Car model looks like in the software interface. The standard reference for vehicle modelling has also been added to the image. Note that this reference frame is common among vehicle dynamics analysists and will be used throughout the work herein.

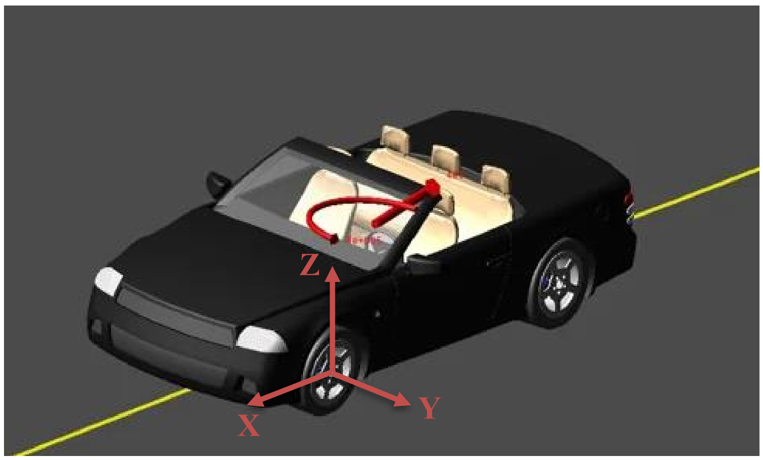


Figure 2: Example ADAMS Car vehicle model [13]

Model simplicity is a factor that affects simulation efficiency and performance. Adams Car provides an environment to perform real time computations, but the models are complex and require significant computing power. Users can study the kinematic behaviour of individual components with Adams Car at the cost of increased computation time. An example of a different vehicle dynamics software with preprocessing, event and model building, and a dynamics solver is VI-grade's CarRealTime (CRT) [14]. CarRealTime provides real time vehicle dynamics solvers for vehicle models requiring less information for generating, compared to Adams Car. This feature is advantageous when developing a vehicle if the physical version does not yet exist. The vehicle models from CRT that are based on already existing vehicles have been validated against Adams Car models to show that CRT models have similar fidelity to the real vehicle [15]. Regarding component-based iterations, CRT does not permit the study of the kinematics of individual components since the software uses a simplified, conceptual approach. However, this feature reduces the computation power and time delay for the vehicle dynamics solver for real time simulations with a dynamic simulator. With similar model performance, reduced DOF, and fast simulation times, CRT provides a way to develop and evaluate design iterations quickly on a dynamic driving simulator. How vehicle models are created for CRT and their structure will be discussed in detail in Chapter 2.

1.2.2 Offline and Simulator Environments

In virtual simulation, the environments can be described in two different settings. The first consists of an offline environment where the virtual simulations can be run on a computer without input from a human driver. Instead, a model of a driver is used to replicate how a driver would anticipate and react when driving. Driver models can replicate a steering robot used in some maneuvers for physical testing or a human driver, depending on the vehicle dynamics software. The second virtual simulation environment consists of a simulator comprised of hardware, software, and programming to produce a replication of a real vehicle's dynamics. Depending on the type of simulator, the seat or cabin in which the driver is placed can have motion. In both the offline and simulator environments, a virtual driver model can control the inputs to the vehicle model such as those discussed in Section 1.2.1. Only in the simulator environment, a human can replace the virtual driver model and act as a feedback controller to drive the simulator. This application is known as driver-in-the-loop. Another major difference between the two simulation environments is the type of ride and handling evaluations that can be completed. In the offline sense, only objective evaluations can be performed. Objective and subjective evaluations with driving simulators can be completed where the driver rates the vehicle's performance. This is a relatively new trend in the automotive industry where several OEMs have purchased and installed state-of-the-art motion-based dynamic driving simulators for evaluating and developing vehicle models and production vehicles.

1.3 Dynamic Driving Simulators

A major contribution to the research completed herein is attributed to the technologies that makeup dynamic driving simulators. These simulators are suitable for imitating realistic driving experiences through replicating the exchange of information between the driver, the environment, and the vehicle in a virtual setting [11]. Dynamic driving simulators provide motion, audio, visual, and haptic information to replicate a real driving activity such as a driving maneuver, including weather and traffic conditions. Since the simulation must occur in real time, vehicle models typically require a reduced number of DOF compared to that of an offline simulation. This aspect is the result of avoiding heavy computations and time lags between the vehicle dynamics solvers and the motion of the platform. There are different architectures of dynamic driving simulators depending on the number of DOF. Dynamic simulators with lower DOF can provide a good compromise between an accurate driving experience, cost, and requiring less accommodation space. On the contrary, dynamic driving simulators with more DOF can provide a higher fidelity driving experience with larger working spaces, longer sustained accelerations, and a larger selection of testing capabilities. Static simulators also exist where the cabin does not move. Such simulators can still provide haptic or force feedback through active seats and belts to the driver. These simulators can come in the form of a desktop simulator or a full vehicle simulator in which the driver sits in a real vehicle cabin without motion.

The dynamic driving simulator used for evaluations in this research can be found in Figure 3. The simulator is called the Driver-in-Motion (DiM) 250 dynamic driving simulator. This simulator was designed and installed by VI-grade and contains the typical six DOF of a hexapod with an additional three DOF. These additional DOF consists of the base or tripod's x- and y-translation as well as yaw. The tripod's DOF extend the longitudinal, lateral, and yaw motion limits of the hexapod. The combination of the hexapod, sometimes referred to as the Stewart Platform [8], and the tripod allows the simulator to separate high and low frequency motions of a vehicle. More details on VI-grade's simulator and the associated technologies can be found in Chapters 3.4 and 4.2.

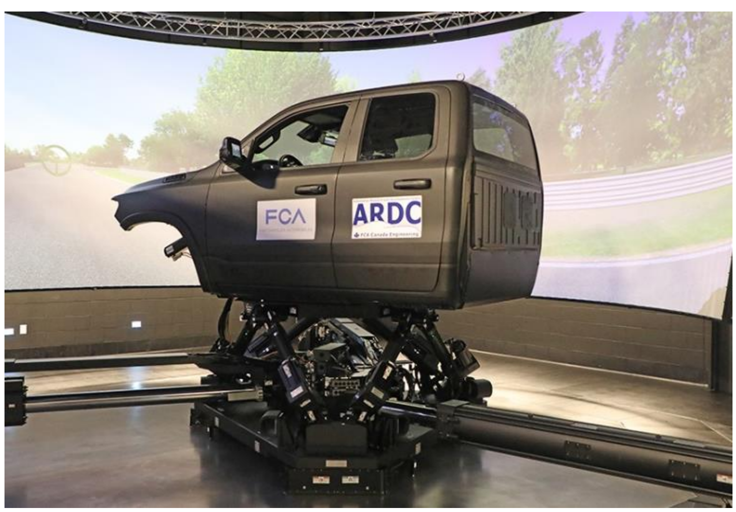


Figure 3: VI-grade DiM 250 dynamic driving simulator

There are several requirements for driving simulators to be considered useful when replicating real driving experiences [11]. The first requirement is that the simulation models of the vehicle must be precise, accurate, and validated to ensure the model is representative of the real vehicle. The second requires that the simulator paired with its software and hardware provides the possibility to simulate all the vehicle subsystems and components in real time. Thirdly, the driver must be immersed in a realistic driving experience where the vehicle cabin and the vehicle's surrounding geographical environment are included through physical or virtual representation. The geographical environment is virtual in most cases. The final requirement states that the driving activities and maneuvers should be executed in the closest way possible to a real driving experience. Through satisfying these requirements, dynamic driving simulators can achieve the following advantages over physical testing:

- Providing a safe driving environment while avoiding harm to the driver during dangerous vehicle maneuvers
- High repeatability in driving scenarios
- Reduced cost associated with creating physical prototypes and design iterations
- Increased efficiency during vehicle development when no physical prototype exists, and many design iterations can be tested through timely model alterations

There are also limitations which must be considered when working with dynamic driving simulators. For instance, the safe environment of the simulator can promote more risk-free driving behaviour since the driver is aware that they cannot be physically harmed. Furthermore, the driving time of many drivers is limited by the onset of motion sickness, otherwise known as kinetosis [11]. Also, different drivers can have different sensitivity to motion sickness. Selecting the appropriate driver(s) can therefore affect the amount of testing to be done in a specified time. Finally, depending on the type of simulator and audio or visual software being used, the level of fidelity of the perceived driving experience can differ. Thus, it is important to consider the requirements and limitations of driving simulators when completing vehicle performance studies.

1.3.1 A Brief History

In the beginning of the twentieth century, the first simulators for replicating a real mechanism's motion were used in the aerospace field for safely training pilots. It was not until the 1970s when the automotive industry adopted the technology [16]. The simulators in this era only had three DOF. Unfortunately, computing power was not sufficient to provide real time calculation of more complex vehicle dynamics. During the 1970s, improvements in graphics technology such as curved screens and projectors provided adequate resolution with simulator displays [8]. At that time, researchers had discussed the use of head-mounted visuals, but again this application was not possible due to computer technology limitations. The Stewart platform was designed previously in the 1960s, having six DOF made possible through six actuators and two platforms. However, due to the technological limitations at the time, the Stewart platform was not created successfully until 1985 by Daimler-Benz while also implementing a vehicle dome to immerse the driver [17]. Through the 1990s and 2000s, advancements in computing power, visual graphics, and improvements in numerical solution algorithms made it possible to provide a high-fidelity, real-time simulator driving experience [11]. A recent advancement in 2020 for driving simulator technologies is the use of cables instead of linear actuators to pull an air-suspended platform over a large flat plat. This feature increases the working space and provides longer exposures to steady-state accelerations, compared to alternative dynamic simulators with linear actuators for such motion [18]. This technology was invented by VI-grade and is currently being implemented in the automotive industry.

1.3.2 Simulators in the Automotive Industry

Driving simulators can been used to complete many different studies on both driver and vehicle performance. As with Stellantis, several other OEMs and automotive companies have also adopted the use of VI-grade's DiM simulators. These include, but are not limited to Ferrari, Volvo Cars, Porsche Motorsports, NIO, Honda R&D, Audi Motorsports, Maserati, and Mercedes AMG [19]. The driving simulators have been used for vehicle model tuning, the development of new electric vehicles, both objective and subjective evaluations, NVH, vehicle steering and handling, and other applications. Furthermore, other driving simulators can be used to study several Human Machine Interfaces (HMI) to determine the potential distraction they have on the driver, as studied in [20]. The HMI study used static simulators from Jaguar Land Rover and the University of Nottingham in the United Kingdom. Another example is the National Advanced Driving Simulator (NADS-1) at the University of Iowa. It has been considered as one driving simulator with the highest fidelity to a real vehicle, although it requires a large space for installation [8]. The NADS-1 has been used for developing or testing Advanced Driver Assistance Systems (ADAS), Electronic Stability Control (ESC), crash avoidance, the performance of young drivers, and in several other research areas [21]. Clearly, driving simulators are used extensively in both industry as well as academia. Other notable areas of study that can use the potential of driving simulators include adaptive cruise control, vehicle ride and comfort, powertrain development, and the effect of substances on the driver's performance as all discussed in [8]. In the future, the adoption of Vehicle to Everything (V2X) technology along with fully autonomous vehicles will likely inspire new studies to be completed with driving simulators in a safe and repeatable environment.

1.3.3 Driver-in-the-Loop Basics

When working with driving simulators, the concept of Driver-in-the-Loop (DIL) is an important part of the virtual simulation and subjective evaluation environment. It brings the virtual simulation and evaluation of vehicle models one step closer to evaluating a real vehicle. Common to all DIL simulations, there are three major components which make up the system. One of the components is the simulator which replicates the motion of a real vehicle. The simulator is similar to the role of a plant in a control system where it receives input from a controller and outputs motion and information. Note that in this case, the simulator receives input from multiple sources – a human controller and a virtual environment including the vehicle model provided by computer software. The design and type of the simulator depends on the needs of the research, but for this research a moving-base dynamic driving simulator is utilized. The DiM 250 has a full vehicle cabin and interior with multiple HMI including a steering wheel, accelerator, and brake pedal. The second component of the DIL environment is the driver which interacts with the HMI to provide input to the simulator. The driver acts as the feedback controller in the control system where the

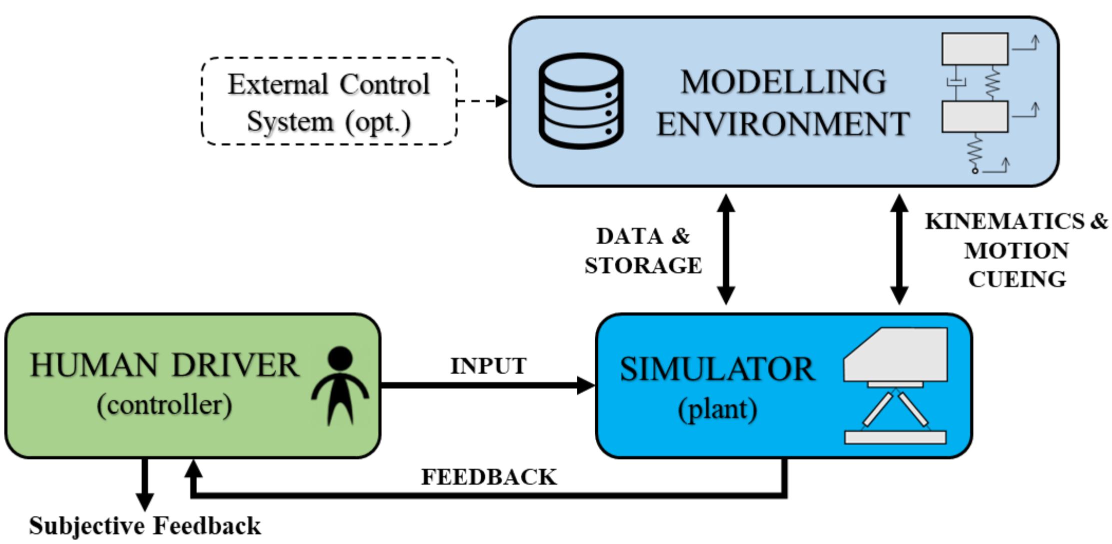


Figure 4: DIL interaction between the driver and simulator

feedback is the audio, visual, haptic, and motion coming from the simulator. The driver uses this information to make decisions on steering, braking, and accelerating. This loop of information exchange occurs in real time while the simulator is being driven. Figure 4 contains a schematic to depict this exchange of information between the major components of DIL simulations.

The final component of DIL simulations is the collection of software acting in the background. There is a graphics computer to provide the visual feedback to the driver in the form of a large, curved projector screen, a rear-view mirror in the cabin, and two side-view mirrors on the exterior of the cabin. Scenery, road topology, road signs and markers, obstacles, and even weather can be displayed to the driver. Another computer calculates motion cues to provide inertial motion of the simulator. This process is done using a Motion Cuing Algorithm (MCA), which generates actuator commands for the simulator based on the vehicle's response, simulator kinematics and working space, and a model of the human vestibular system. As a result, the MCA provides motion cues to make the driver perceive they are driving in a real vehicle. Additionally, the modelling environment connects other software that contain necessary information regarding the road profile, the vehicle model, and external models for vehicle subsystems. As depicted in Figure 4, an external control system can replace one of the vehicle subsystems. It is also possible that input from a simulator observer can change parameters of such systems while the driver provides feedback on the vehicles performance. One example would be to replace the model of the dampers with an external control strategy to replicate a different type of damper. Finally, objective data regarding the vehicle's dynamics can also be displayed and exported from the DIL simulation for post processing.

1.4 Objective and Subjective Testing

When developing a vehicle, there are two distinct areas of testing and evaluating vehicle ride and handling. The two areas are objective and subjective testing. Each method has its advantages and drawbacks, but both are necessary at different stages of vehicle design and development [22].

1.4.1 The Difference Between Objective and Subjective Testing

For objective testing, either a real vehicle or a vehicle model can be evaluated. When performing tests with a real vehicle, instrumentation and sensors record the response of various vehicle systems to driver and road inputs in order to quantify its performance. For instance, accelerometers can be placed at different locations on the driver's seat to measure the accelerations felt by the driver. This approach is one way to quantify ride. The same can be done with virtual sensors in offline simulations. Objective metrics, which are performance indicators, are the quantitative parameters that are calculated from the recorded data. On the other hand, subjective testing involves using a human driver to rate a real or virtual vehicle based on their perception of the vehicles performance. In this case, a rating scale comprised of numbers and corresponding word descriptors is used to evaluate vehicle ride and handling [23]. The evaluators provide subjective feedback to quantify the vehicle performance, rather than recording vehicle dynamics data. As a result, the vehicle's perceived performance is a result of the driver's level of knowledge of vehicle ride and handling and driving skills.

1.4.2 The Purpose of Objective and Subjective Testing

Objective testing provides an unbiased method to quantify and compare the performance of one vehicle to another, especially when a robot is used to drive the vehicle. In this case, human sources of error do not affect the objective performance of a vehicle or the repeatability of testing. This method allows a good estimate of predicting the ride and handling of a vehicle in early development stages before the final product is completed. On the other hand, humans make the final decision to purchase vehicles, thus the subjective evaluation of a vehicle's performance can play a dominant role in the vehicle's acceptance by consumers. Using a dynamic driving simulator for subjective evaluations of vehicle models allows OEMs to evaluate their vehicles before the need for a real prototype. Care must be taken when selecting the drivers for subjective testing as many physical and psychological factors can affect a driver's perception of the vehicle's performance. Correlation between objective and subjective metrics has been a topic of study for several decades in the automotive industry [24]. Here, objective metrics are correlated to subjective ratings for OEMs to predict the subjective performance of their vehicles. As a result, both objective and subjective testing methods are required throughout the development of vehicles. Both methods expedite the process and can aid engineers with providing accurate predictions of future vehicle ride and handling.

1.5 Project Description

The research conducted is focused on the development of a subjective method to evaluate the ride and handling of vehicle semi-active suspension. This method combines the conventional use of subjective questionnaires and experienced drivers with a state-of-the-art dynamic driving simulator. Several validated vehicle models from Stellantis are studied with the implementation of a virtual controller model for semi-active suspension, replacing their passive counterparts. Several different controller strategies are studied both objectively and subjectively. A correlation between objective metrics and subjective ratings is completed at the end of the research to evaluate the sensitivity of the simulator and to identify which objective metrics have the most significant effect on the subjective performance of vehicles equipped with a certain semi-active suspension control strategy.

1.5.1 Project Novelty

The number of OEMs that have been using suspensions with controllable dampers has been steadily increasing [25]. At the same time, evaluating the performance of controllable dampers has been limited to objective methods or time-consuming tuning in a physical environment. With recent developments in driving simulator technologies, it has become possible to subjectively evaluate accurate vehicle models equipped with semi-active suspension in a safe and timely manner. The evaluation methods developed consider a full set of vehicle dynamics and driving maneuvers, which have not been considered in previous work regarding the ride and handling evaluation of a virtual semi-active suspension model. Furthermore, a subjective evaluation method utilizing a dynamic driving simulator and a questionnaire with subjective metrics and ratings for each maneuver has yet to be developed for this application. The completion of the project also provides a method to reduce prototype development time and reduce associated costs of intermediate physical prototypes.

1.5.2 Project Goals and Objectives

The overall objective of the research is to develop a method to subjectively evaluate the ride and handling performance of Stellantis’ vehicle suspension systems with a dynamic driving simulator. As a baseline, validated vehicle models equipped with their production passive suspensions are evaluated objectively and subjectively to quantify their ride and handling performance. At the same time, a semi-active suspension controller model is implemented with the vehicle models to evaluate objectively and subjectively the ride and handling improvements. The use of the dynamic driving simulator for subjective evaluations will validate performance benefits of the semi-active suspension, expedite the suspension tuning practices, and allow engineers at Stellantis to determine if the simulator is sufficient in capturing the performance improvements. Correlation between the changes in objective metrics and subjective metrics is also to be completed to determine which objective metrics significantly affect the subjective ratings. The project is divided into multiple sub-objectives which outline the intermediate steps to complete the research. The sub-objectives of the project are outlined as follows:

I. Select the validated vehicle models and maneuvers for ride and handling evaluations
II. Simulate the vehicle models with their passive suspension on the maneuvers chosen in (I)
III. Develop a semi-active suspension controller model and connect it to each vehicle model
IV. Perform offline co-simulations between the virtual suspension controller and vehicle models to quantify their objective ride and handling on the maneuvers chosen in (I)
V. Use the dynamic driving simulator to subjectively evaluate the vehicle models with their passive and semi-active suspensions to study changes in ride and handling, on the maneuvers chosen in (I)
VI. Perform a correlation study between changes in objective ride and handling metrics and changes in subjective ratings to determine where correlation exists and simulator sensitivity

1.6 Thesis Organization

Chapter 2 of this thesis explores concepts and theory related to vehicle dynamics and modeling, ride and handling, basics on control theory, and how to correlate data. Chapter 3 presents a review of the published work in the areas of this research such as evaluation of ride and handling, semi-active controllers, and the use of dynamics driving simulators. Chapter 4 contains a preliminary study to determine which objective metrics and maneuvers are viable for the evaluation of ride and handling of solely changing damper characteristics. Chapter 6 presents the work completed relating to the selection of the semi-active suspension controller and its architecture. Chapters 7 and 8 outline the objective and subjective evaluation methods, respectively. Chapter 9 contains the results of the evaluations and a discussion on the performance of the controller and the subjective evaluation method. Finally, conclusions and recommendations are made in Chapter 10.

CHAPTER 2. Theory

The theory presented and discussed in this chapter will ensure that the reader has the mathematical tools and understanding for the implementation of the research. The concepts will provide a foundation on the understanding of how the vehicle responds to inputs from the road and driver. These concepts were utilized when developing and tuning the final semi-active suspension controller to save time and improve the ride and handling performance. An emphasis is placed on the suspension subsystem, as it is the part of the vehicle being studied in this research while being related to the goals of the project from Section 1.5.2. Finally, modelling of the vehicles and some of their components, descriptions of objective performance metrics, and control theory fundamentals for modelling semi-active suspension are explored later in the chapter.

2.1 Vehicle Dynamics and Models

The subject of vehicle dynamics is concerned with studying the forces imposed on a vehicle while it is performing some motion or maneuver. The reasoning behind how and why the vehicle responds to these forces is equally as important as the imposing forces [5]. The contact area or "patch" between the tires of a vehicle and the road is the location where the forces are developed when a vehicle is in motion. Whether it be a vertical force acting on the tires as a result of road elevation changes or a lateral force due to transient steering inputs causing lateral acceleration, vehicle dynamics allows engineers to understand how the forces are transmitted through the tires and into the cabin. A vehicle subsystem which plays a vital role in the vehicle's response to all forms of inputs is the front and rear suspension systems.

2.1.1 Vehicle Suspension

The suspension systems of a vehicle play two major roles when a vehicle is in motion. The first pertains to the isolation of severe disturbances from the road. In this way, the components of the suspension attenuate disturbances induced by impacts on the road such as potholes and bumps. The second role is to maintain contact between the tires and the road to permit a stable and controllable vehicle for the driver [9]. Vehicle suspensions connect the wheels to the body of a vehicle through a variety of systems generally comprised of a spring, damper, and connecting elements such as rods, control arms, and other linkages. The spring's stiffness allows flexibility in the vehicles motion and is a major determinant in a vehicles static ground clearance, whereas the dampers literally provide damping to the vehicle's motion. Dampers are typically a hydraulic element that constricts the flow of a fluid through a set of tubes. The constricting motion of the fluid always provides a resistance to the vehicle body motion. As a result, dampers play a significant role in the vehicle's transient response to inputs and energy dissipation. Without dampers, vehicles would roll faster, disturbances would take significantly longer to dissipate, and vehicle ride and handling would be considerably poor by today's standards.

Vehicle suspension dampers can be classified into three categories – passive, semi-active, and active [26]. The feature of the dampers that differentiates them is the damper force-velocity relationship. This relationship is represented by a two-dimensional curve called a damper characteristic curve. The slope of the curves represented the damping rate of the dampers. Thus, a steeper damper curve corresponds to more damping. The curves for each damper type can be generalized as in Figure 5, similar to [27].

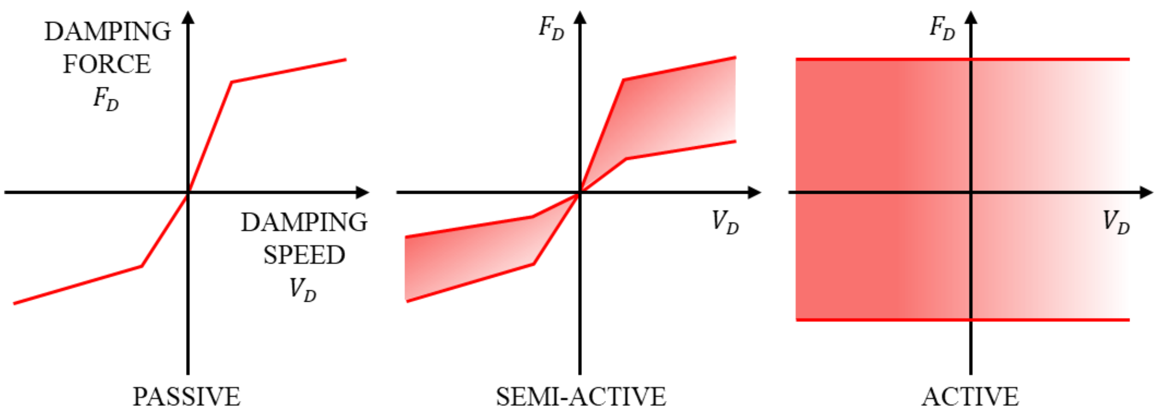


Figure 5: Damper types in vehicle suspension

Most vehicles on the road have a passive suspension with the damper curves represented by a single shape, where the dampers can only dissipate energy from the vehicle and provide damping forces in the opposite direction of wheel vertical motion. Vehicles with such suspensions are typically tuned for a compromise between ride and handling. Fully active dampers utilize a control strategy to completely control the response of the dampers where the suspension systems act like actuators. The damping curves become planes that cover the complete cartesian plane as in Figure 5. As a result, an active suspension can input energy to the vehicle and force a desired response to road irregularities and driver inputs. Furthermore, both ride and handling can be significantly improved at the cost of system complexity, high power requirements, and higher price [9], [28], [29]. In between the two extremes lies semi-active suspension. Semi-active dampers are controllable in the sense that the damping rates can be varied in real time, but the dampers can only resist the motion of the vehicle like a passive suspension. In Figure 5, this is represented by a range of damping curves between the physical upper and lower damping capabilities of the damper. Figure 6 contains a breakdown of the damping characteristic curve and its features. An example of a semi-active damper is the magnetorheological (MR) damper where changing the electric current passing through a MR fluid will alter the viscosity of the fluid and thus its damping characteristics. Another example is a solenoid-valve controllable damper produced by Thyssenkrupp where hard and soft settings can be programmed into the dampers and selected by the driver [30]. The performance of semi-active dampers is superior to a passive suspension, but inferior to active suspensions while avoiding the aforementioned higher costs.

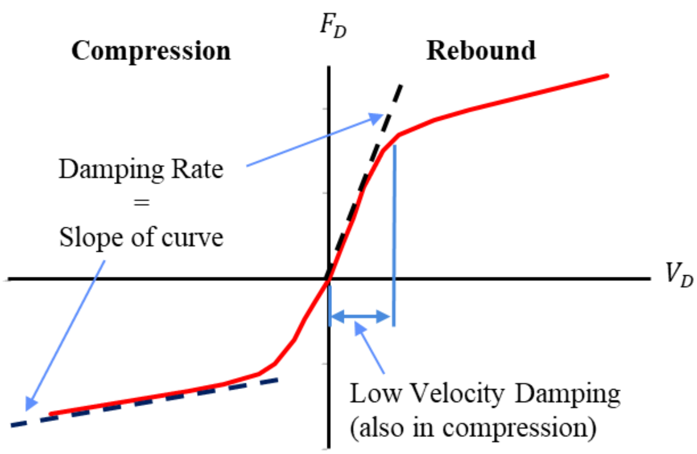


Figure 6: Damper characteristic curve features

Damper curves have rebound and compression regions, sometimes called the rebound and bump, respectively. Note that damping curves have a low-velocity damping region defined by the seemingly steeper, linear regions of the damper curve near the origin. These damping rates are a result of the fluid dampers orifice geometry in the main tube or piston and affects the primary and secondary ride of the vehicle [6], [9]. The shallower slopes of the upper portion of the curves are a result of the spring-loaded valve inside fluid dampers which affects higher frequency motion of the vehicle's response. The transition between the two velocity regions is a result of pressure buildup in the damper before the spring-loaded valve is opened [6]. Moreover, the feature of high damping at low velocities and lower damping at high velocities promotes low transmissibility of the vehicle body for primary and secondary ride frequencies [6], [31]. The transmissibility of a vehicle is simply the ratio of input disturbance amplitude to the resulting vehicle body displacement, velocity, or acceleration. Road disturbances tend to create larger compression velocities in dampers and therefore larger forces. Rebound forces on the other hand are due to the suspensions returning to its equilibrium position. As a result, the compression damping is lower than rebound to avoid significantly higher compression forces [6].

2.1.2 Responses of the Vehicle

To better understand how a vehicle responds during different types of maneuvering, it is possible to isolate and study different directional responses of the vehicle. The vehicle's vertical, lateral, yaw, pitch, and roll responses are presented to understand which parameters of the vehicle contribute significantly to the vehicle's response to driver and road inputs. Figure 7 contains the sign convention typically used in the automotive field for reference to the vehicle's directional responses.

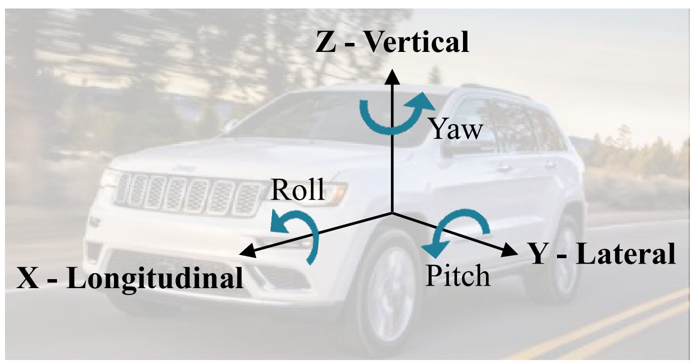


Figure 7: Direction of vehicle response

The vertical response of the vehicle is excited by elevation changes or disturbances in the road. These disturbances displace the wheels in the vertical direction which result in the suspension springs and dampers being compressed or extended. When the springs and dampers are deformed from their equilibrium position, spring and damping forces are generated. In vehicle dynamics, these forces act on the sprung and unsprung masses of each corner of the vehicle. Consider the quarter-car model displayed in Figure 8. The sprung mass represents part of the vehicle body mass, and the unsprung mass represents the mass of the wheel, wheel carrier, and some of the suspension components. The quarter-car model represents a lumped mass model containing only the essential dynamics of the vehicle's vertical response [5]. Roll, pitch, yaw, longitudinal, and lateral dynamics are not considered here. This model has two DOF which are the vertical motions of the two masses.

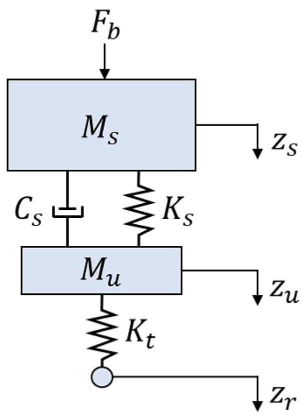


Figure 8: Quarter-car model

Applying Newton's second law to the free body diagrams of the two masses in the quarter-car model above results in equations (1) and (2), assuming $z_r > z_u > z_s$ and steady-state vibration.

$$M_s\ddot{z}_s + C_s\dot{z}_s + K_s z_s = C_s\dot{z}_u + K_s z_u + F_b \tag{1}$$

$$M_u\ddot{z}_u + C_s\dot{z}_u + (K_s + K_t)z_u = C_s\dot{z}_s + K_s z_s + K_t z_r \tag{2}$$

Adding a spring element in series with the spring and damper elements in Figure 8 can be done to include the effects of a top mount typically found in strut assemblies with shock absorbers, to increase model fidelity. The transmissibility of the quarter-car model can be plotted to study two phenomena that are important for understanding the role of dampers in vehicle suspension and the vehicles response to road disturbances. Figure 9 adapted from [32] presents the transmissibility of the two masses in a quarter-car model similar to Figure 8 with (solid line) and without a top mount (dashed line). The input to the system response in Figure 9 is harmonic motion.

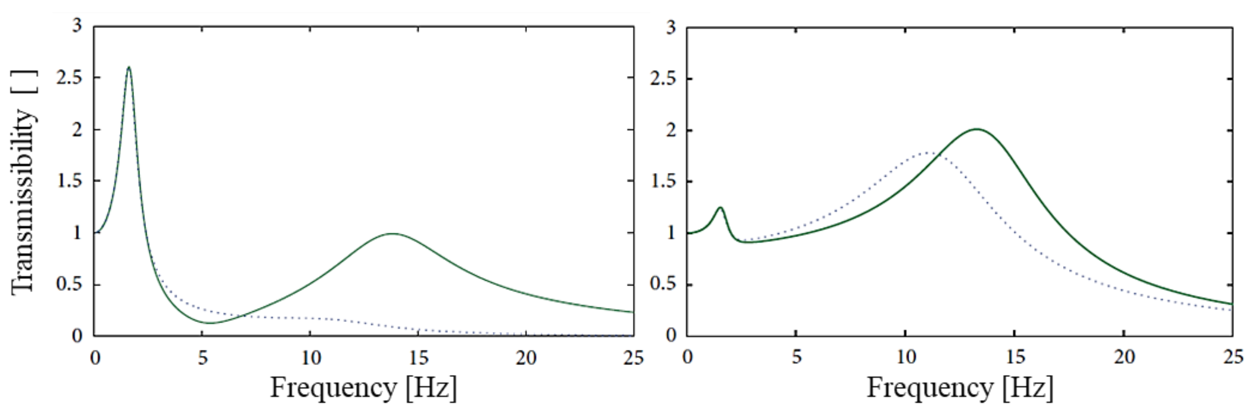


Figure 9: Quarter-car model transmissibility of sprung mass (left) and unsprung mass (right)

In Figure 9, there is a peak transmissibility near 1 Hz and another around 13 Hz which reside in the primary and secondary ride frequency ranges, respectively. These are also the typical values for the damped natural frequency of vibrations for the sprung and unsprung masses, respectively [5]. The role of dampers is to attenuate the acceleration transmissibility at these two locations, thus improving ride. By increasing the damping coefficient in equations (1) and (2), the acceleration transmissibility of the sprung mass can be reduced at the peak near 1 Hz, whereas the transmissibility of the unsprung mass will increase near 13 Hz [33] until a certain damping rate depending on the mass and stiffness of the model. Furthermore, the position of the transmissibility peaks will shift slightly due to changes in damping. A detailed discussion of other effects on transmissibility due to spring rates, road excitations, and mass can be found in [5], [32]–[34]. Note the significant difference between the dashed and solid curves in Figure 9, where neglecting the presence of the strut mount can have a significant difference on the system's response to road inputs. This aspect is why models encompassing more of the real features of vehicle architecture are important when studying vehicle dynamics.

Concerning the fundamentals of a vehicle's lateral and yaw response, the linear single track or "bicycle" model can be studied for lateral accelerations up to 0.4g [11]. Figure 10 presents a schematic of the bicycle model with the important variables, as well as the model's general input-output relationship. For the complete mathematical model, see [11]. See Appendix A for the complete list of assumptions and simplifications with the linear bicycle model.

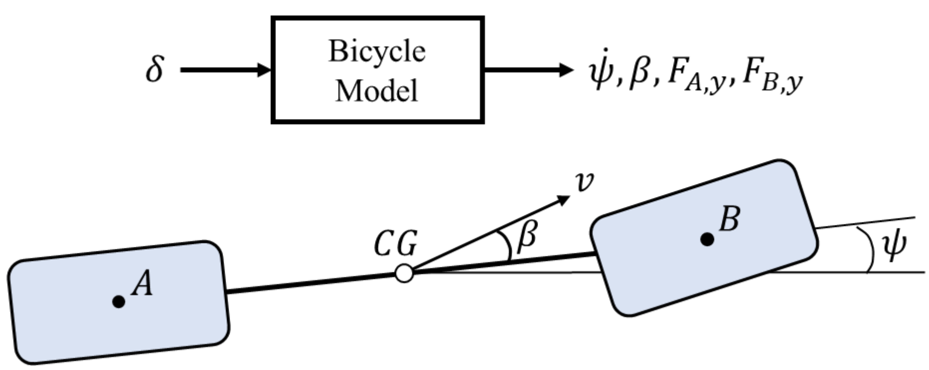


Figure 10: Bicycle model

The input to the bicycle model is the steering angle of the front tires and the output is the yaw rate, vehicle sideslip angle, and the front and rear tire lateral forces. This model only has two DOF represented by the yawing of the vehicle and the sideslip angle of the body. Applying the conservation of linear momentum in the vehicle's lateral direction and the conservation of angular momentum about the vehicle's vertical axis, equations (3) and (4) can be developed. See [11] for the complete derivation of these equations and further discussion.

$$mv(\dot{\psi}_B + \dot{\beta})cos\beta = cos\delta\, F_{B,y} + F_{A,y} \tag{3}$$

$$\theta_{b.m.}\ddot{\psi}_B = F_{B,y}\, cos\delta\, l_B - F_{A,y} l_A \tag{4}$$

The effect of the vehicle's suspension is not captured by the bicycle model, but it is important to note the dependance of the bicycle model's dynamics on the tire lateral forces, resulting from the vehicle's lateral acceleration. In general, the lateral tire forces are proportional to the tires' cornering stiffness, which is a function of the tires vertical loading as described in Chapter 6 of [5]. Dampers play a significant role in the dynamic behaviour of tire vertical loading where an increase in damping rates will reduce the fluctuation of the vertical tire loading and permit a more static tire lateral force compared to softer damping settings. As a result, the damping rates of vehicle suspension have an effect on the vehicle's road holding, or grip capabilities during dynamic steering inputs. The bicycle model can be paired with the quarter-car model to study this affect. For instance, Figure 11 compares the vertical loading between a relatively low and a relatively high front suspension damper's damping rate. The relationship can be extended to the rear suspension as well. However, the magnitude of the effect for reducing the vertical load fluctuation increases with lower spring rates and larger rear sprung mass.

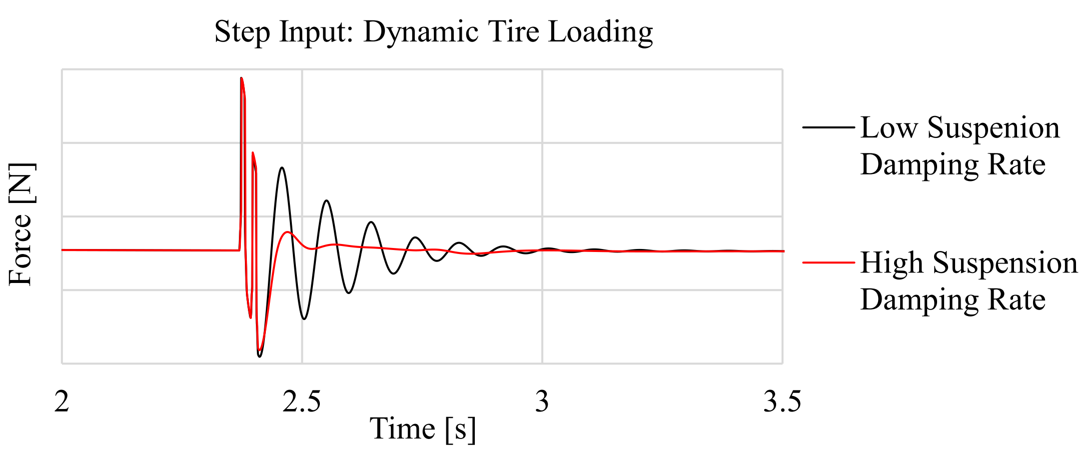


Figure 11: Effect of damping rates on vertical tire loading

Concerning the side slip angle of the bicycle model, one can look deeper into the model and observe the front and rear tire slip angles. In general, if the two slip angles are equal for a given steering angle, then the vehicle exhibits a neutral steering behaviour. If the front tire slip is larger than the rear, then the vehicle exhibits an under-steering behaviour. If the rear tire slip is larger, then the vehicle over-steers. Depending on which behaviour a given vehicle exhibits, the yaw rate of the vehicle is affected. This phenomenon can be identified with the bicycle model. For instance, a vehicle with a larger rear tire slip angle than the front tire will exhibit over-steering and result in higher yaw rates [5], [11]. At high speeds, over-steering can become dangerous for the driver.

When the bicycle model is paired with two quarter-car models, the result is a bicycle vibrating model. This model can be used to study the pitch response of the vehicle. In this case, there are different road displacement inputs from the front and rear quarter-car models. Moreover, the front and rear quarter-car models can have different spring stiffnesses and damping rates, depending on the vehicle. Figure 12 presents the lumped mass representation of the vibrating bicycle model. Furthermore, depending on the geometry of the vehicle being modelled, the centre of gravity, CG, will lie towards the rear quarter-car for a rear heavy vehicle or towards the front for a front heavy vehicle. For many passenger vehicles, the engine resides in front of the vehicle cabin and thus the vehicles are front-heavy. The position of the centre of gravity affects the steering behaviour of the vehicle where front-heavy vehicles result in higher lateral acceleration experienced by the front tires, thus higher tire slip angles [5]. This phenomenon contributes to the understeering behaviour found with front-heavy passenger vehicles.

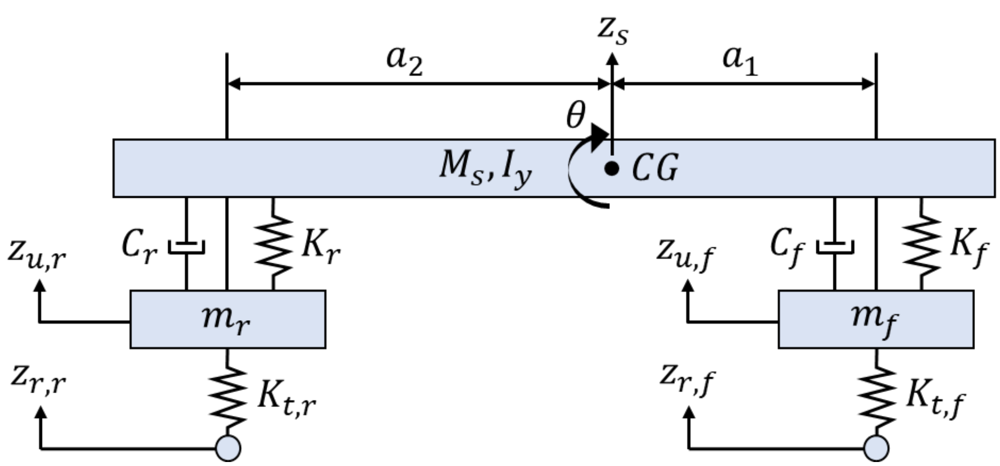


Figure 12: Vibrating bicycle vehicle model

The vibrating bicycle model has four DOF consisting of the sprung mass's vertical and pitching modes and the vertical motion of each unsprung mass. For this model, the lateral, yaw, roll, and longitudinal responses of the vehicle are not studied. In this case, Lagrange's method is applied to quickly obtain the equations of motion through defining the Lagrangian as the subtraction of the kinetic, potential, and dissipated energies of the bodies in the vibrating bicycle model. Equations (5), (6), (7), (8), (9), and (10) are the equations of motion for this model by applying Lagrange's method and using matrix form. For a detailed derivation of these equations, see [35].

$$[m]\ddot{z} + [c]\dot{z} + [k]z = F \tag{5}$$

$$z = [z_s \quad \theta \quad z_{u.f} \quad z_{u,r}]' \tag{6}$$

$$[m] = \begin{bmatrix} M_s & 0 & 0 & 0 \\ 0 & I_y & 0 & 0 \\ 0 & 0 & m_f & 0 \\ 0 & 0 & 0 & m_r \end{bmatrix} \tag{7}$$

$$[c] = \begin{bmatrix} C_f + C_r & a_2C_r - a_1C_f & -C_f & -C_r \\ a_2C_r - a_1C_f & C_f a_1^2 + C_r a_2^2 & a_1C_f & -a_2C_r \\ -C_f & a_1C_f & C_f & 0 \\ -C_r & -a_2C_r & 0 & C_r \end{bmatrix} \tag{8}$$

$$[k] = \begin{bmatrix} K_f + K_r & a_2K_r - a_1K_f & -K_f & -K_r \\ a_2K_r - a_1K_f & K_f a_1^2 + K_r a_2^2 & a_1K_f & -a_2K_r \\ -K_f & a_1K_f & K_f + K_{t,f} & 0 \\ -K_r & -a_2K_r & 0 & K_r + K_{t,r} \end{bmatrix} \tag{9}$$

$$F = [0 \quad 0 \quad z_{r,f}K_{t,f} \quad z_{r,r}K_{t,r}]' \tag{10}$$

For this model, the matric F is the input to the system, implicitly being the front and rear tire vertical displacements. Furthermore, there are now four natural frequencies or modes of the system, namely one for the sprung mass's vertical motion, one for its pitch motion, and two modes corresponding to the front and rear unsprung masses' vertical motions. Similar to the quarter-car model, the role of the damping coefficients C_f and C_r is to attenuate the vertical acceleration of the masses and the pitching acceleration of the sprung mass. Still, the natural frequency magnitudes of the sprung and unsprung masses are similar to those discussed with the quarter-car [35]. When the frequency of excitation to the vibrating bicycle model is not exactly one of the natural frequencies, the response of the system is a combination of the four modes.

The final directional response of a vehicle to be discussed is the roll response. The half-car model, consisting of one sprung mass connected to two quarter-car models, is used for studying the roll response. The model is shown in Figure 13 and the resulting equations of motion are in Appendix A. Vehicle yaw, lateral, longitudinal, and pitch motions are not considered with here.

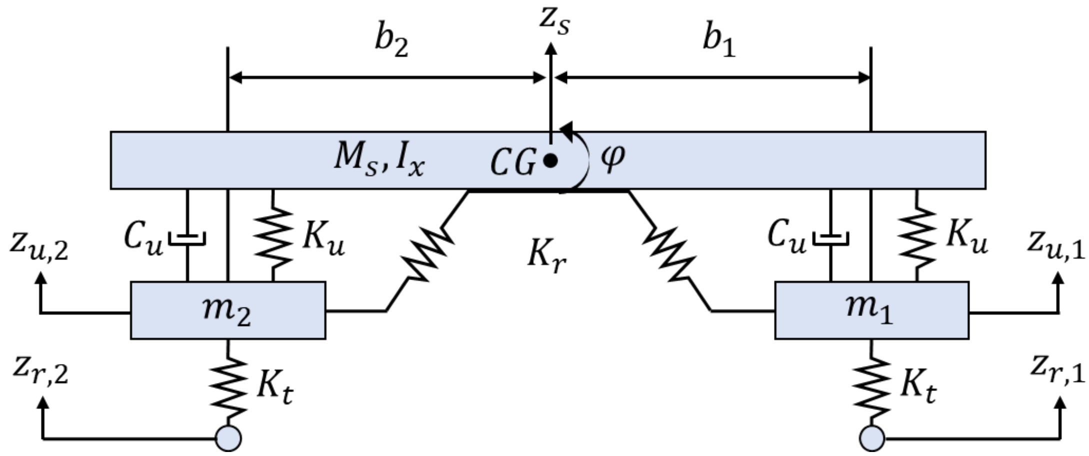


Figure 13: Half-car vehicle model

Note that an anti-roll bar element with stiffness K_r has been added to the model, as many passenger vehicles on the road contain this feature. For the half-car model, there are four DOF pertaining to the vertical motion of each mass and the roll of the sprung mass. As a result, there are four natural modes of the vehicles response, with similar magnitudes of natural frequency as the vibrating bicycle model. In this case, the natural frequency of the roll motion for the half-car model lies in the primary frequency range near the body's vertical natural frequency. When the excitation coming from the road does not have the same frequency as one of the systems natural frequencies, the response of the half-car model is a combination of the four modes. The role of the dampers is to attenuate the amplitude of vibrations mainly at the natural frequencies. As with the previous models, attenuating the amplitude of acceleration for only one mode can have detrimental effects on the other modes if not tuned properly. Finally, since the vertical acceleration of the unsprung masses is proportional to the vertical forces on the unsprung masses, dampers can also attenuate the rate of weight transfer. Referring back to Figure 11, a higher damping coefficient can reduce the time and fluctuations in weight transfer between axles or left and right tires before a vehicle reaches its steady state roll or pitch responses. Therefore, the damping rates attenuate the change in tire loading and help maintain the tires' contact with the road. The next step in vehicle dynamics and modelling is to combine the aforementioned models into a full car model encompassing all of the vehicle's motion.

2.1.3 Twin Track Model

The use of a dynamic driving simulator, such as the DiM 250, requires a certain threshold of model DOF for the driving experience to be realistic. The Twin Track vehicle model is the simplest vehicle model containing the necessary vehicle dynamics and DOF for a full motion dynamic driving simulator [11]. The twin track model consists of five bodies, namely the sprung mass and four unsprung masses. Each unsprung mass consists of the corresponding tire and wheel hub carrier at each corner of the vehicle. Each wheel has rotation about its lateral axis and displacement in the vehicle's vertical axis. The sprung mass has six DOF pertaining to translation in and rotation about the vehicles x-, y-, and z-axes. In total, there are fourteen DOF. The model can be expanded to contain additional DOF such as the spatial motion of the wheels and suspension kinematics. Here, the focus will be on the basic twin track model without suspension kinematics. Furthermore, models for the drivetrain, brakes, tires, and driver can be added to the system as well as anti-roll bars, and applied forces to model external effects on the vehicle. The discussion here is limited to the vehicle model for the chassis and suspension. See [11] for the description of the drivetrain, brakes, tires, and driver mathematical models. Figure 14 presents the twin track model and a visual description of the connections between the sprung and unsprung masses.

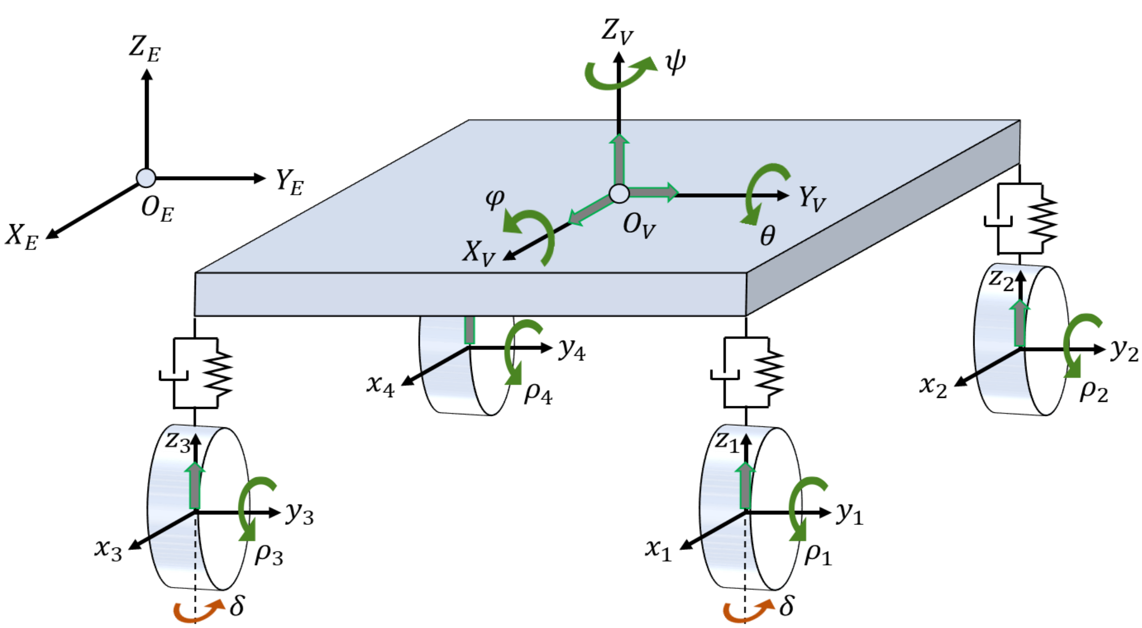


Figure 14: Twin track model without suspension kinematics

With the twin track model, there are additional coordinate systems for the global reference frame, denoted with "O", and the wheel coordinate reference frame, denoted with a number corresponding to each wheel. Transformation matrices are required to transform velocity and force vectors between the three reference frames. The steering angle of the front wheels is assumed to be constant, proportional to the steering wheel angle and steering system's steering ratio, and not a DOF since it is an input to the system [11]. Moreover, tire mathematical models are used in parallel with the twin track model to calculate the tire slip angles and forces resulting from the twin track model's wheel loading and lateral acceleration. The wheel loads come from the twin track model, which are transmitted through the springs and dampers. The connection between the sprung and unsprung masses of the twin track model are applied forces. These applied forces act at discrete points and are produced by the springs and dampers when they are deformed from their equilibrium. A transformation matrix is used to convert the spring and damper forces from the chassis to the wheel reference frames, and vice versa. Finally, the fourteen DOF of the twin track model results in fourteen modes with corresponding natural frequencies. As with previous models discussed, the dampers play a similar role in attenuating the vertical and rotational accelerations of the vehicle, as well as attenuating fluctuations in wheel loading. For a comprehensive derivation of the equations of motion for the twin track model with and without suspension kinematics, see Chapter 11 in [11]. See Appendix A for more information on the twin track model's simplifications and assumptions. As these vehicle models become more complex, the relationship between the damping rates and the response modes of the vehicle become increasingly complex, but they encompass more aspects of the real vehicles behaviour compared to the quarter and half-car models. Additionally, the computational power required to solve the equations of motion increases. For dynamic driving simulators, it is important to use vehicle models with the minimum necessary complexity to maintain a realistic driving experience. A simpler and faster vehicle modelling approach than the twin track model with kinematic wheel suspensions is discussed in the next section.

If kinematic wheel suspensions are added to the twin track model, a set of generalized coordinates are used to describe the spatial motion of the wheels. The camber of the wheels is the rotation of the wheels about their longitudinal axis, which is included in this type of model. However, the vertical motion of the wheels are removed, while the DOF of the model remains the same as the twin track model without kinematic wheel suspension. Finally, the wheels and wheel carriers are considered separate bodies where the kinematics describing the translation and rotation between the two bodies at each suspension are considered. This model allows vehicles with different types of suspension to be studied as opposed to the generalized suspensions in each of the models previously discussed. However, the model's complexity and computation requirement increase further.

2.1.4 CarRealTime Modelling

Vehicle models in CRT are not considered purely MBD systems. Instead, CRT combines the simplified vehicle chassis from the twin track model and uses a functional or parametric description of the suspensions. With CRT, the vehicle model is split into a set of subsystems pertaining to the chassis, front and rear suspensions, front and rear wheels, powertrain, brakes, and the steering system. Each of these subsystems are connected through inputs and outputs in a continuous loop during simulations inside CRT. Additional "auxiliary subsystems" can be added to the models for considering additional vehicle features.

The vehicle models in CRT have the same fourteen DOF as the twin track model without suspension kinematics. However, if the torsional stiffness of a vehicle chassis is known either through physical testing or from data of an Adams model, then translational and rotational stiffnesses can be added to the chassis subsystem in CRT. This feature would add an additional six DOF to the model and allow for the consideration of chassis twist under asymmetric loading. Aerodynamic forces such as drag and downforces can be applied at discrete points in the vehicle reference frame, similar to the twin track model.

For the suspension subsystems in CRT, a major time-saving aspect of the models compared to purely MBD suspension models is the use of Kinematics and Compliance (K&C) data. Kinematics data describes the motion of the vehicle's wheels as a function of suspension component geometry and orientation. As a result, the camber, caster, and toe angle behaviour of the wheels for the front and rear are included in CRT models. Compliance data describes the vehicle's wheel motion as a function of the stiffness of suspension components such as strut mounts, bushings, and linkages. In essence, when the parts of the vehicle's suspension are under loading when the vehicle is maneuvering, the elastic behaviour of the components are captured in CRT simulations. K&C data comes from a set of testing in which a vehicle suspension is loaded in different directions, and the translation and rotation of many of the suspension components is recorded. The result of the testing is a set of lookup tables where the independent variable is either the wheel vertical displacement, a tire force, or an aligning torque and the output variable can be the wheel toe angle, camber angle, caster angle, or the additional x and y translation of the wheel. Figure 15 is one example of a lookup table from CRT with the scales hidden. The different curves describe the toe angle to longitudinal force relation for differences in wheel jounce. In CRT, there are as many as 250 lookup tables to describe the motion of the front and rear suspensions from K&C data.

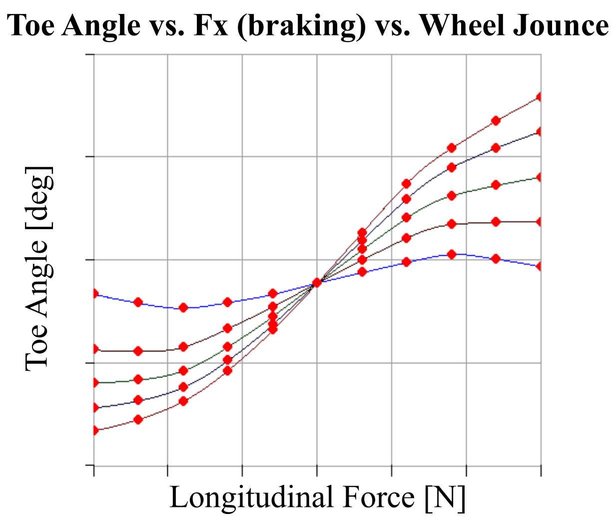


Figure 15: CarRealTime lookup table example

The additional motion considered in the K&C data are not to be confused with additional DOF. The lookup tables are a conceptual approach to model the additional rotations of the wheel without having to explicitly solve additional equations of motion. Rebound and bump stops can also be added to the suspension subsystems to model the behavior of the suspension at the limits of the working space. Finally, dampers are modelled by property files containing the data points for damper characteristic curves as well as preload forces required to install the dampers on the real vehicle. The connection between the suspension and wheel subsystems is a simple three step process, as follows:

1. Transform the wheel displacement into the spring, damper, and other suspension components' deformations
2. Use the lookup tables (i.e., damper curve) to generate the forces for the springs, dampers, and suspension components

3. Transform the force from the suspension components to the equivalent force acting on the wheel centre.

The steering subsystem in CRT is modeled in a similar fashion as the suspension subsystems. Instead of wheel vertical displacement or “jounce”, the steering rack travel and the driver model steering wheel angle are the independent variables of the lookup tables. Furthermore, feedback maps are added to the steering subsystem to describe the relationship between the torque acting about the wheel steering axis and the resulting force applied back on the steering rack. On the dynamic driving simulator, this aspect translates to the haptic feedback torque on the steering wheel when turning a vehicle. This feedback is not considered in the two track or other models discussed. To conclude, this conceptual approach of lookup tables to help describe the dynamics of the vehicle is much faster than more complex MBD models and reduces the DOF compared to similar MBD models.

2.1.5 Tire Modelling Basics

Tires are a significant part of the suspension subsystem. The first role of the tires is to transmit the forces and moments between the vehicle and the road. This transmission is done at the point or plane of contact between the tires and the road, known as the contact patch. The second is to absorb the wheel carrier loadings and protect the vehicle occupants from severe road impact loads. These functions help provide better ride and handling of the vehicle. The outer part of the tires consists of a viscoelastic material which permits two types of friction that help maintain grip between the vehicle and road. The first is adhesion where intermolecular bonds form between the tire material and the road material. The second is hysteresis friction resulting from the different loading and unloading regions of the contact patch causing a moment that resists the rotation of the tires. Finally, the performance of tires is affected by several factors, including the tire inflation pressure, the temperature of the rubber and road, tire size, tread shape and depth, the material and construction of the tire, as well as the history of the tires use.

There are many models that have been developed to describe the behaviour of tires. Some are purely mathematical, physical, or a combination of both. In many cases, the desire for a specific model depends on the application. For vehicle dynamics, mathematical tire models combined with empirical data (semi-empirical models) are used which capture low frequency tires characteristics in the range of 0-20 Hz [11]. This frequency range encompasses the behaviour of tires that affects vehicle ride and handling and is thus suitable for most vehicle dynamics-related applications. For higher frequency ranges, physical models such as finite element models are used for comfort and vehicle crash scenarios. Semi-empirical models partly rely on experimental data that is specific to the tire under study. Purely physical models are significantly complex and require higher computation power. For a dynamic driving simulator, it is thus advantageous to use a semi-empirical tire model when studying ride and handling. For semi-empirical models, the input and output of tire models in general is discussed in Chapter 7 of [11] and recreated in Figure 16.

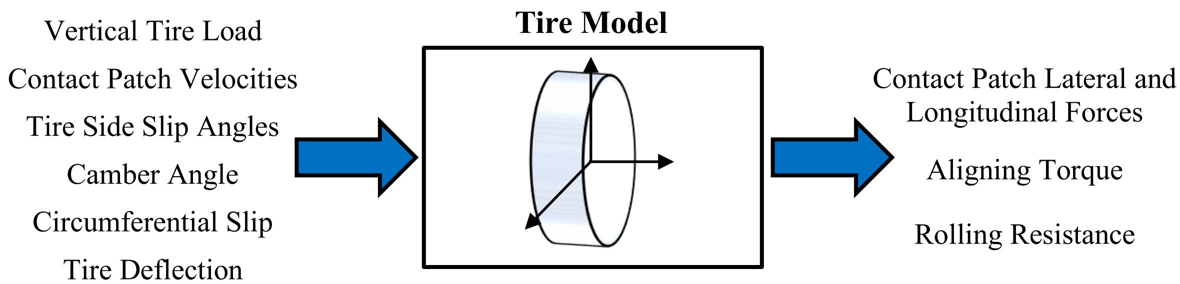


Figure 16: Tire models inputs and outputs

The brief discussion here is focused on the fundamentals of the widely used Magic Formula (MF) developed by Hans Pacejka, a semi-empirical model [36]. Equations (11), (12), and (13) are the general structure of the MF.

$$y(x) = D\sin[C\tan^{-1}\{Bx - E(BX - \tan^{-1}Bx)\}] \tag{11}$$

$$Y(X) = y(x) + S_V \tag{12}$$

$$x = X + S_H \tag{13}$$

In (11) and (12), Y is the desired output and X is an input. For the MF and (11), three relationships are described. The first is the contact patch longitudinal force as a function of longitudinal slip. The second is the contact patch lateral force as a function of side slip angle. The third is the aligning torque as a function of the tire side slip angle. The MF can accurately describe these relationships for steady-state maneuvering [36]. The constants $B, C, D,$ and E are interdependent, describe the shape of the curves resulting from (11), and are fitted with experimental data from physical tire testing. In (12) and (13), S_V and S_H are constants that allow the curves from (11) to be shifted from the origin. The MF is an efficient model for accurately describing the behaviour of real tires being modeled in vehicle dynamics applications. See [36] for a complete description of the relationships between the constants in the MF and a thorough discussion of the modifications made to the MF over the past twenty years. In short, modifications to the MF have been made to consider tire phenomena including combined lateral and longitudinal slip, pneumatic trail, higher frequencies of tire dynamics, rigid ring modelling, large camber angles, turn slip, and others. A recent version of the MF is discussed in Section 4.1.3.

2.2 Vehicle Ride and Handling Metrics

This section lists a handful of objective metrics presented in vehicle dynamics textbooks for quantifying ride and handling. Many of the metrics described here have been used in academic institutions and automotive OEMs for internal evaluations. The metrics are presented with their formulas and a brief description to understand which trends in the metrics correspond to better ride and handling.

2.2.1 Ride Performance Metrics

The following ride metrics are taken and summarized from [5], [9], and [35]. Recall that ride deals with the response of the vehicle to road disturbances or excitations.

1. Vehicle Body Amplitude Ratios

$$\frac{\ddot{z}_s}{\ddot{z}_r} \tag{14}$$

$$\frac{\ddot{z}_s}{F_b/M_s} \tag{15}$$

Equations (14) and (15) quantify the vertical acceleration of the vehicle chassis in response to road excitations or vertical dynamic loading. Both of these metrics are typically recorded over a frequency spectrum including primary and secondary ride. The power spectral density of the sprung mass is another way to address these amplitude ratios. Lower ratios correspond to improved ride.

2. Root Mean Square (RMS) Accelerations

$$RMS(\,) = \sqrt{\frac{1}{n}\sum_{i=1}^{n} x_i^2} \tag{16}$$

In Equation (16), n is the total number of data points and x is replaced by the sprung mass vertical acceleration, chassis pitch or roll acceleration, or the driver's seat and head vertical, lateral, or longitudinal acceleration. Under dynamic loading, the acceleration signals pass over the zero-horizontal axis many times. The RMS value of these accelerations is used to capture both positive and negative accelerations and compare one vehicle's suppression of motion to another's. The RMS accelerations are measured in either the time or frequency domain (primary and secondary ride ranges), depending on the road excitation and application.

3. Suspension Working Space (SWS)

$$SWS = \max(suspenion\ extension) - \min(suspension\ compression) \quad (17)$$

The SWS simply measures how much space is required for the suspension to operate in when a vehicle is maneuvering. Suspension extension is considered positive whereas suspension compression is considered negative. A lower SWS means the vehicle can be lower to the ground, promoting better aerodynamics and a lower centre of gravity resulting in reduced roll and pitch.

4. Dynamic Tire Loading (DTL)

$$DTL = Fluctuation\ of\ vertical\ tire\ loading\ (F_{z,tire}) \quad (18)$$

This metric can also be considered a handling metric if the DTL is measured in response to steering inputs. The fluctuations are sometimes measured using the RMS of the tire vertical loading or as the RMS of the tire vertical deflection. Lower DTL corresponds to improved dissipation of road disturbances and better road holding ability, or grip.

5. Transient Response Characteristics

$$Overshoot = Peak\ Value - Steady\ State\ Value \quad (19)$$

$$Response\ Time = t_{90\%\ steady\ state} - t_0 \quad (20)$$

$$Settling\ Time = t_{\pm 5\%\ steady\ state} - t_0 \quad (21)$$

$$Rise\ Time = t_{90\%\ steady\ state} - t_{10\%\ steady\ state} \quad (22)$$

$$Peak\ Value = maximum\ value\ of\ the\ response \quad (23)$$

These metrics are computed when road step inputs or "cleats" are imposed on the vehicle. The sprung mass or vehicle chassis' vertical displacement, velocity, or acceleration can be considered here. In general, a quick response having a short settling time, low peak value, and low overshoot is beneficial for dissipating road disturbances. However, there is typically a tradeoff between each of the characteristics. Note that these metrics are also used to study the roll angle and yaw rate responses of the vehicle as handling metrics.

2.2.2 Handling Performance Metrics

The following handling metrics are taken and summarized from [5] and [35]. Recall that handling deals with the vehicle's response to driver inputs.

1. Understeer Gradient, K

$$\delta_{steer} = \frac{L}{R} + KA_y \tag{24}$$

For a positive K, the vehicle exhibits understeering behaviour. For a negative K, the vehicle oversteers and for zero understeer gradients, the vehicle exhibits neutral steering. The understeer gradient is a function of vehicle speed and is evaluated during steady state cornering [5]. Typically, passenger vehicles exhibit understeering for safety and improved stability at high speeds.

2. Lateral Acceleration Gain, L_{gain}

$$L_{gain} = \frac{Chassis\ Lateral\ Acceleration}{Steering\ Wheel\ Angle\ (SWA)} \tag{25}$$

This metric corresponds to the cornering capacity of a vehicle and its ability to produce lateral acceleration when cornering. For a relatively higher lateral acceleration gain, a vehicle can produce higher cornering (lateral) tire forces and promote better road holding.

3. Yaw Velocity Gain, Y

$$Y = \frac{\dot{\psi}}{SWA} \tag{26}$$

The yaw velocity gain describes how fast the vehicle yaws for a given SWA. For higher gains, the vehicle responds more to steering inputs. For sport and race cars with a target for better handling, a higher yaw velocity gain is typically desired. This aspect also results in reduced steering inputs required by the driver for a given yaw rate.

4. Roll Gain, R

$$R = \frac{\varphi}{SWA} \tag{27}$$

The roll gain metric describes how much the vehicle rolls in response to steering inputs. For improved handling, lower roll gains are advantageous. Reduced roll can correspond to reduced camber angle changes when maneuvering, promoting a larger contact patch between the tire and road. The rollover limit of a vehicle is harder to reach if a vehicle has a lower roll gain.

The aforementioned ride and handling metrics are common metrics from a theoretical point-of-view. In the automotive industry, OEMs have internal standards and laboratory procedures that address other vehicle characteristics and metrics. Additional metrics will be discussed in the review of literature in Chapter 3 of this work. Subjective metrics will also be reviewed from published work on ride and handling.

2.3 Control Theory Basics

The groundwork for control theory is creating a mathematical model to describe a known physical or virtual system, analyzing the model, and simulating its response to varying inputs. The step input is one form of input where a constant value is switched on and off instantaneously; a cleat on a road is a practical example of a step input. Newton's Laws, conservation of energy, and simplifying a system with a free body diagram are common methods to mathematically model systems. The use of graphical simulation tools such as MathWorks' Simulink provide an even simpler way to describe dynamic systems. The response of a given system has two components: the steady state and the transient responses. The role of damping is only significant during the transient response or how a dynamic system reaches its steady state response. Contrarily, suspension springs and geometry dominate the characteristics of the steady state response.

2.3.1 Block Diagram Fundamentals

Block diagrams are a schematic of graphical blocks that represent the transfer functions of system components or entire subsystems [37]. Connected blocks each have one or multiple inputs and outputs that are transferred between the blocks. For instance, a block can represent the front right suspension of a passenger car and another block can represent the chassis. The connection between the two blocks is the applied forces from the springs, dampers, and transferred loads through the linkages. These applied forces and loads are the output of the suspension block and the input for the chassis block, or in the opposite sense depending on the application. In some cases, a single block can represent a constant value such as a damper coefficient, or a spring rate. Common to many control systems are plants, controllers, and sensors connected in a feedback loop. Figure 17 is a schematic representing the block diagram of a simple closed loop-feedback automotive controller system adapted from [37]. Note that a block can also contain its own subsystem, meaning an entire block diagram can reside within a block from a higher level. In essence, block diagrams can have a hierarchy structure where each level breaks down a system or subsystem into a collection of simpler blocks.

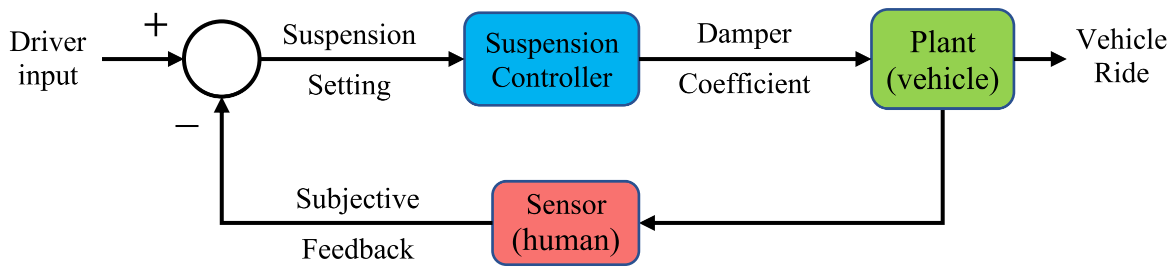


Figure 17: Automotive feedback control system example

In Figure 17, the input to the system is the driver's choice for improved ride or improved handling from the vehicle, or the plant of the system. The plant block would have many layers and vehicle subsystems each represented by their own block diagrams. The driver acts as a sensor to monitor or sense the actual ride and handling they perceive from the vehicle and use it to decide a suspension setting for the vehicle. Sensors can also be in the form of hardware such as an accelerometer measuring the RMS vertical acceleration of a seat or the vertical velocity of the chassis. Note that this system is applicable for a vehicle with semi-active or active suspension, but not possible for a vehicle with passive suspension. Finally, the controller accepts the selected suspension setting from the driver and computes the appropriate damper settings to improve the ride or handling performance of the vehicle. The controller block would have a deeper level with a block diagram representing the four vehicle wheel suspensions and control algorithms to compute the damping coefficient. In general, a feedback controller converts the input feedback and desired signal into a control signal through structured logic or rules [37]. This example illustrates a closed-loop feedback system.

2.3.2 Modelling Semi-Active Suspensions

For semi-active suspensions, the damping rates or coefficients of dampers are computed. This damping rate is the control signal produced by a suspension controller as in Figure 17. Modelling semi-active suspension controllers involves the implementation of an algorithm to compute the suspension damping. Block diagrams are used to represent the controller, as it a simple, visual tool that can be easily altered or tuned by users of the software. For simple vehicle models such as the quarter-car, the damper component is replaced by a controllable damper with input, as presented in Figure 18. The input is the damping coefficient, which can change either continuously in real time, or discreetly as the driver desires. In some passenger vehicles, a dial can be found on the dashboard with selectable driving settings.

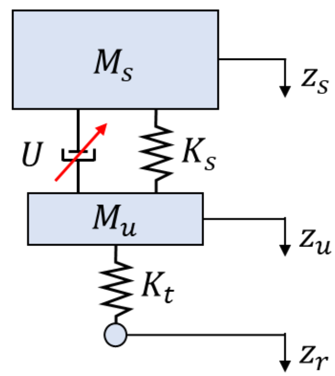


Figure 18: Quarter-car model with a semi-active damper

For vehicle models from CRT, the suspension is not modelled with the quarter-car, but instead with lookup tables. For CRT, modelling a semi-active suspension controller requires computation of the damper forces as well as the damping rates. As a result, the controller replaces the entire damper in the CRT model's suspension subsystems. The methodology for modelling the final semi-active controller and its architecture is discussed in detail in Chapter 6.

2.3.3 Controller Types and Working Principles

There are several groups of semi-active damper control algorithms for suspension and damping applications. They can be grouped into proportional-integral-derivative, adaptive, and robust control where the latter provides the best performance in scenarios of uncertainty [26]. A subset of robust control, called variable structure control (VSC), provides multiple methods to change the structure of control depending on the environmental conditions, such as a change in road type. VSC algorithms allow the system to switch between controller settings to maintain a desired performance, especially when uncertainty arises. This feature renders VSC advantageous for vehicle dynamics applications when untested driving conditions or driver inputs arise. The theory behind four well-established VSC algorithms for semi-active suspension are presented, namely the switchable controller, skyhook, groundhook, and hybrid controllers.

The switchable controller involves the switching between discrete damper characteristic curves and selecting a damping coefficient, depending on a switching boundary control law (or manually). Each curve represents a "soft", "stiff", or intermediate damping coefficient for vehicle suspension. Moreover, the damper curves can be linear, or nonlinear such as the one presented in Figure 6, although nonlinear asymmetric damper curves are more practical for vehicle suspension. Figure 19 presents the working principle of a switchable semi-active controller.

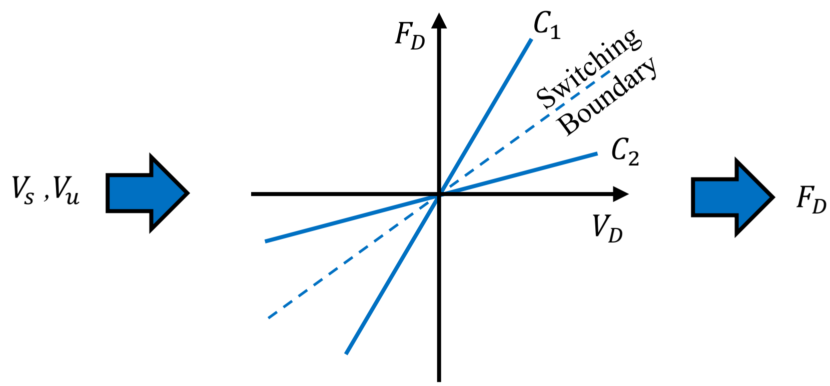


Figure 19: Switchable control strategy

In Figure 19, the vertical velocities of the sprung and unsprung masses are recorded from a sensor to determine which damper curves to set, depending on the switching boundary. Equations (28) and (29) represents Karnopp's strategy used for the switching boundary [38]. Once the damper curve is selected, the velocity of the damper is then mapped to a damper force according to the damper curve. Equation (30) is the relationship for the damper velocity.

$$C_1 \; if \; V_S(V_S - V_U) > 0 \tag{28}$$

$$C_2 \; if \; V_S(V_S - V_U) < 0 \tag{29}$$

$$V_D = V_S - V_U \tag{30}$$

With the switchable strategy, the goal is to avoid the ride and handling tradeoff found with passive suspensions by providing two or more damper curve settings. Each setting or curve is designed for improved ride, improved handling, or a better compromise between the two compared to passive. Skyhook VSC was also developed by Karnopp in which the vehicle chassis is connected to a virtual damper fixed to the "sky". The damper coefficient of the suspension damper is switched between a minimum and maximum value, depending on the same velocity products as in equations (28) and (29). Skyhook logic is defined in Equations (31) and (32). An adaptation was made by [39] to define the maximum and minimum practical damping coefficients for the skyhook and subsequent strategies [39]. The practical limits are considered in Equation (32).

$$C_{sky} = \begin{Bmatrix} C_{max} \; if \; V_s V_D \geq 0 \\ C_{min} \; if \; V_s V_D < 0 \end{Bmatrix} \; ; \; F_{sky} = C_{sky} V_S \tag{31}$$

$$\{C_{max}, C_{min}\} = \{2.2C_{passive}, 0.2C_{passive}\} \tag{32}$$

The goal of skyhook control is to add damping to the vehicle chassis, or to improve ride. This goal is reached by maximizing the damper force when the suspension damper applies a force in the opposite sense as the conceptual sky damper would. Contrarily, the Groundhook VSC adds damping to the unsprung mass to improve the vehicle's road holding ability. As a result, the wheel vertical motion is suppressed to reduce the DTL or tire deflection. Equation (33) summarizes the control logic for the Groundhook strategy.

$$C_{Ground} = \begin{Bmatrix} C_{max}\ if\ -V_U V_D \geq 0 \\ C_{min}\ if\ -V_U V_D < 0 \end{Bmatrix}\ ;\ F_{gnd} = C_{ground} V_U \tag{33}$$

Finally, the Hybrid VSC strategy is a combination of Skyhook and Groundhook strategies. Conceptual dampers attach the sprung mass to the sky and the unsprung mass to the ground. Depending on a weighted term, α, the strategy places more importance on dampening the sprung mass, unsprung mass, or a compromise between the two. Equations (34), (35), and (36) summarize the logic behind the Hybrid VSC strategy.

$$F_{hybrid} = G\left[\alpha * \sigma_{sky} + (1-\alpha) * \sigma_{ground}\right];\ \ G = 2.2C_{passive} \tag{34}$$

$$\sigma_{sky} = \begin{Bmatrix} V_S\ \ if\ \ V_S V_D \geq 0 \\ 0\ \ if\ \ V_S V_D < 0 \end{Bmatrix} \tag{35}$$

$$\sigma_{ground} = \begin{Bmatrix} V_U\ \ if\ \ -V_U V_D \geq 0 \\ 0\ \ if\ \ \ -V_U V_D < 0 \end{Bmatrix} \tag{36}$$

When α is set to a value of zero, the hybrid strategy becomes the Groundhook strategy and when α is set to 1, the Hybrid strategy becomes the Skyhook strategy. Intermediate values of α result in a compromise between the Skyhook and Groundhook performance benefits.

2.4 Correlating Data Sets

For the research conducted, theory for the method of linear correlation and regression is required to identify correlation between single objective metrics and subjective ratings. Linear correlation relates to identifying the direction and strength of the relationship between two variables. The result of linear correlation is a single value, positive or negative, known as the "R-value". This linear correlation method was developed by Karl Pearson and is formally called the Pearson product moment correlation coefficient [40]. The value ranges from -1 to 1, where an absolute value of one corresponds to a strong correlation between two variables. A Positive R-value means that increasing the independent variable, the dependent variable also increases linearly. The same is true in the opposite sense when the independent variable is decreased. Equation (37) is the relation for determining the R-value between two variables, x and y.

$$r = \frac{n(\sum xy) - (\sum x)(\sum y)}{\sqrt{[n(\sum x^2) - (\sum x)^2][n(\sum y^2) - (\sum y)^2]}} \tag{37}$$

In Equation (37), n represents the total number of data points. Linear regression is an extended study of linear correlation where one can define a line of best fit for the data set under study. If a strong linear correlation exists, then linear regression is applied to find the slope and y-intercept of a line representing the relationship. With linear regression, the sum of the squares of the vertical distances from each data point to the line of best fit is a minimum [40]. Equation (38) represents the standard form for the equation of a line. Equations (39) and (40) present the relations for the y-intercept and slope of the line of best fit for data that has a linear correlation, respectively.

$$y = a + bx \tag{38}$$

$$a = \frac{(\sum y)(\sum x^2) - (\sum x)(\sum xy)}{n(\sum x^2) - (\sum x)^2} \tag{39}$$

$$b = \frac{n(\sum xy) - (\sum x)(\sum y)}{n(\sum x^2) - (\sum x)^2} \tag{40}$$

In linear regression, it is not appropriate to extend the line of best fit outside the data set. Therefore, the line of best fit can be used for predicting the output of a variable in between the lowest and highest data points; data extrapolation is not appropriate with linear regression [40].

CHAPTER 3. Literature Review

The following chapter presents a review of the literature and published research on several areas of the project. First, the reader will be informed on what work has already been done regarding objective and subjective evaluation of vehicle ride and handling. Next, a comparison of semi-active suspension controllers is made to determine which control strategies have the best ide and handling performance. Lastly, research with dynamic simulators and correlating objective and subjective data is reviewed. Within each section of the chapter, gaps in the research will be identified and highlighted to ensure the reader is aware of what research remains to be completed. Later chapters will outline how these gaps are filled with the research carried out in this project.

3.1 Objective Evaluation of Ride and Handling

Passenger vehicles have been studied for decades in the automotive industry to objectively quantify their ride and handling performance. By measuring certain signals, engineers can measure in real time the responses of physical vehicles and virtual models. In many cases, driving maneuvers or obstacles are standardized for testing specific aspects of a vehicle's response. OEM's, suppliers, and academic institutions have tested real vehicles and virtual models over several maneuvers to objectively evaluate vehicle ride and handling. This section of the chapter will highlight the methods, metrics, and maneuvers commonly used for the objective evaluation of vehicle performance.

3.1.1 Methods for Objective Evaluation – Physical

Several OEMs such as the General Motors Company, the Ford Motor Company, and the Hyundai Motor Company have published papers documenting their procedures for objectively evaluating vehicle ride performance. In [41], engineers at General Motors (GM) highlighted the difference between vehicle primary ride or "motion smoothness" and secondary ride or "shake" and the necessity to evaluate both ride attributes, objectively. The tradeoff between improving primary ride and worsening secondary ride when tuning passive dampers on production vehicles is the focus in [41]. A highlight of GM's objective ride evaluation procedure was the activity of tuning the different regions of damper curves and measuring vehicle vertical, pitch, and roll accelerations at different frequencies [41]. At Stellantis, these ride frequencies are measured with accelerometers on various locations on a vehicle's chassis, the driver seat, and near the suspensions [2]. Engineers at Hyundai also objectively evaluated an SUV's ride by measuring similar accelerations when driving a vehicle over different road profiles, exciting primary and secondary ride frequencies [42]. Different suspension attributes such as shock absorber damping characteristics and several bushings' stiffness were altered, where the vehicle was driven over each road profile before another suspension change was made. At Ford, a real vehicle with accelerometers was mounted on a four-post testing rig to measure vertical accelerations at the driver seat and head-level lateral accelerations [43]. A road profile was simulated with the four-post testing rig by encompassing ride frequencies from 0-20 Hz, thus incorporating primary and secondary ride, similar to GM. Similarly, a four-post testing rig was run at MTS Systems Corporation to record various accelerations at the driver seat, floorboard, and headrest at primary and secondary ride frequencies [44]. As was completed by all of the OEMs' research aforementioned, the ride performance of a vehicle is typically compared to a reference vehicle to determine the relative improvement or worsening ride as a result of tuning. In general, there is a clear pattern among OEMs to objectively evaluate vehicle ride based on acceleration measurements near the driver's location. This methodology will be adopted in the objective evaluation of the semi-active suspension controller discussed in later chapters. This objective evaluation method of ride provides an opportunity to tune the controller before implementing it on a dynamic driving simulator.

Similar to the objective evaluation of ride, objective evaluation of handling has historically consisted of driving a vehicle through several driving maneuvers. GM and Stellantis have documented objective handling evaluations studying the tire slip angles in response to transient steering inputs [41], [45]. Both OEMs mention the objective evaluation of real vehicles on roads by driving vehicles through a set of maneuvers, one at a time and measuring certain dynamic variables. Stellantis had measured the driver SWA, and the chassis' lateral acceleration, roll angle, and yaw rate [45]. Hyundai has completed similar objective handling evaluations on real vehicles, where several vehicles with different spring stiffnesses, antiroll bar stiffnesses, and damper characteristics were tuned [42]. The measured signals were used to compute objective metrics such as response gains and delays. The metrics are then compared between each vehicle configuration relative to a baseline vehicle, to determine which vehicle has better relative performance. This comparison allows the OEMs to benchmark their vehicles with other production vehicles.

Two types of maneuvers are commonly tested in handling objective evaluations – open loop and closed loop steering tests. Open loop steering considers applying specific SWAs and driver inputs to the vehicle whereas closed loop maneuvers allow a driver to use feedback to freely control the SWA, throttle, and brake. The studies in [45] involved both types of maneuvers. When completing evaluations on physical vehicles, it is important to consider the repeatability and driving conditions of the test procedure. The National Highway Traffic Safety Administration noted that during their handling objective evaluations, they ensure the road is flat and dry, and the condition of the tires is constantly monitored to ensure repeatability between evaluations [46]. Despite this methodology, repeatability issues can still occur due to the human driver's inability to perfectly replicate their steering and acceleration commands between driving tests. For virtual objective methods involving offline simulations, these issues are not present due to the controlled environment. Finally, a driving simulator provides the possibility to combine the benefits of physical and virtual evaluations by driving a high-fidelity vehicle model, validated with physical test data, in a controlled environment.

3.1.2 Methods for Objective Evaluation – Virtual

With virtual objective evaluations, the procedure is relatively similar to physical testing. The obvious difference between the two methods is the use of a virtual vehicle model in a simulated environment. The accuracy of the results from virtual evaluations are heavily affected by the fidelity of the models being testes. Thus, it is necessary that users of vehicle models validate the model behaviour with physical test data. With offline simulations, repeatability is significantly increased over human-driven vehicles on real roads where variations in road conditions, weather, and driver inputs can exist [44], [47]. GM has used virtual simulation tools to objectively evaluate vehicle primary and secondary ride on a track with different road types and obstacles at different sections [47]. In each section of the track, the size of the obstacle and vehicle speed are controlled to ensure isolation of primary and secondary ride frequencies [47]. Similar to physical objective evaluation, measurements of accelerations experienced by the driver are recorded to compute objective metrics. These metrics are presented in the next section. Stellantis also simulates their vehicle models on several virtually replicated roads based on their proving grounds. After a simulation, vehicle dynamic signals are extracted with a post-processing software to compute the objective metrics and compare the ride performance between different vehicle models.

Regarding virtual handling objective evaluation, Stellantis has measured the driver SWA, the chassis' lateral acceleration, roll angle, and yaw rate on several open and closed loop steering maneuvers [48]. Volvo Cars and Chalmers University of Technology have also completed objective evaluations of vehicle models' handling through applying changes to damper curves and simulating several open loop steering maneuvers [49]. Here, the objective handling of two track vehicle models equipped with the MF were validated with real test vehicles. Similarly, Stellantis engineers [50] stated that objective handling evaluations are completed whereby MBD vehicle models are simulated through open loop steering maneuvers to study the vehicle's yaw, roll, and lateral response dynamics in addition to the signals recorded in [48]. For the handling studies completed by Stellantis [45], [48], [50], the vehicle models studied were validated with measured data during experimental testing on real vehicles. This validation is the link between physical and virtual objective evaluations where virtual vehicle models are typically based on an existing production vehicle. The physical vehicle must be evaluated to validate the model's behaviour to ensure accurate objective metrics are being analyzed from the evaluations. With this research, validated and high-fidelity vehicle models are provided by Stellantis, whereby a dynamic driving simulator provides a controlled environment to subjectively evaluate the models. The objective evaluation methods mentioned will be adopted and adapted so suite the evaluation of a semi-active suspension and to help tune the controller before implementing it on the driving simulator.

3.1.3 Objective Evaluation Metrics

After objective evaluations are completed for vehicle ride and handling, a collection of objective metrics are computed from the data collected from simulations and experimental testing. Table 1 is a summary of the objective metrics from a collection of published work on vehicle ride and handling, in addition to the metrics presented in Section 2.2. The studies from which these metrics originated are concerned with evaluating vehicles with tuning a passive suspension. Through an initial study on the semi-active suspension controller presented in Chapter 6, several of the metrics in Table 1 are adopted in the objective evaluation of the controller. The maneuvers corresponding to each metric and reference have also been recorded in Table 1.

Table 1: Objective Metrics from Published Literature

Metric Type	Metric Name / Description	Maneuver Details	Published Work
Ride	Tire Normal Force Loading and Fluctuation	Rough roads	[41]
	Peak-to-Peak Accelerations (x, y, z) at Seat	Impact bar road (cleat) and high-speed proving grounds	[42]

Ride (cont.)	RMS, Power Spectral Density, or Frequency-Weighted Accelerations (x, y, z) at Driver Seat	Impact bar road (cleat) and high-speed proving grounds	[42], [43] [44]
	Dissipation Lasting Time at Seat	Impact bar road (cleat)	[42]
	Driver Seat Vertical Acceleration	Belgium Blocks and a road with many potholes	[47]
	Chassis Pitch and Roll Accelerations	Cross Ditches, a road with cats' eyes on one side, and local road profiles	[44], [47]
	Driver Head-Level Lateral Acceleration, Power Spectral Density	Test Track exciting primary and secondary frequencies	[43]
	Vertical Acceleration at Chassis Centre of Gravity	Sinusoidal open loop steering input	[49]
	RMS Sprung Mass Vertical Acceleration	Step and sinusoidal road input	[51]
Handling	Agility – Min. Rear Tire Slip Angle Overshoot	Transient steering	[41]
	Stability - Min. Front Tire Slip Angle Overshoot	Transient steering	[41]
	Understeer	Step steering input	[42]
	Side slip angle	Step steering input	[42], [45]
	Roll Angle or Velocity	Step steering input	[42], [52]
	Steering Wheel Torque	Step steering input	[42], [45], [52]
	Lateral Acceleration vs. SWA	Sine sweep steering input	[42]
	Yaw Rate vs. SWA	Sine sweep steering input	[42]
	Yaw Rate Time Delay	Sine sweep steering input	[42]
	Roll Angle vs. Lateral Acceleration	Sine sweep steering input	[42]
	SWA vs. Steering Wheel Torque	Sine sweep steering input	[42]
	Steering ratio	Steady state circular test	[45]
	Time Lag between SWA and Yaw Rate	Step steering input	[45]
	Gain between Roll Angle and Lateral Acceleration	Step steering input	[45]
	Time lag between SWA and roll angle	Step steering input	[45]
	Gain between Yaw rate and SWA	Double lane change	[45], [48]
	Time Delay Yaw Rate and Lateral Acceleration	Double lane change	[45], [48]
	Gain between roll velocity and jerk of lateral acceleration	Double lane change	[45], [48]
	Gain between roll velocity and SWA	Double lane change	[45], [48]
	Magnitude of Roll and Pitch Acceleration	Sinusoidal open loop steering input	[49]
	Magnitude of Roll Acceleration	Step steering input	[49]
	Magnitude of Chassis Lateral Acceleration and Yaw Rate	Sinusoidal open loop steering input	[49]
	Yaw Rate Response Time	Step steering input	[46], [52]
	Yaw Rate Peak Response Time	Step steering input	[46]
	Yaw Rate Overshoot	Step steering input	[46]

Handling (cont.)	Road Holding – RMS of tire vertical loading	Step and sinusoidal road input	[51]
	Lateral Acceleration Gain and Phase	Frequency Response	[52]
	Yaw Rate Gain and Phase	Frequency Response	[52]
	Slope of SWA vs. Lateral Acceleration	Steady state circular test	[22], [52]
	Slope of Front Slip vs. Lateral Acceleration	Steady state circular test	[22], [52]
	Slope of Side Slip vs. Lateral Acceleration	Steady state circular test	[22], [52]
	Slope of Steering Wheel Torque vs. Lateral Acceleration	Steady state circular test	[22], [52]
	Maximum Roll Angle	Steady state circular test	[22], [52]
	Peak Yaw Rate and Response Time	Step steering input	[22], [52]
	Lateral Stability – side slip angle and chassis yaw rate and roll angle	Slalom	[53]

In this research, only the dampers of a vehicle model are being altered. As a result, not all of the metrics and maneuvers listed in Table 1 will be relevant in this study. The important point to consider from Table 1 is that accelerations experienced by the driver at the head rest and seat location as well as pitch and roll accelerations of the vehicle chassis are recorded for objective ride evaluation. For handling, the vehicle's yaw, roll, pitch, and tire response characteristics are recorded to determine the timing and overall behaviour of the vehicle when exposed to driver inputs. The maneuvers which excite these ride and handling aspects will be chosen for the objective and subjective evaluation methods outlined in Chapters 7 and 8.

3.1.4 Maneuvers and Standards for Objective Evaluation

This section contains descriptions of most of the maneuvers referenced in Table 1. Corresponding standards can also be found with certain maneuvers, such as ISO standards. The description of some maneuvers is not presented here as the published work did not provide specific details. Recall that OEMs retain much of their internal procedures and details for confidentiality from competitors.

1. Rough Roads: Road profiles with changes in elevation that excite primary, secondary, and even higher frequencies of acceleration. Real roads have been laser-scanned to measure the road profile for virtual simulations. Rough roads are often used for evaluating ride. The vehicle speed is typically held constant to ensure the excitation of certain frequencies.
2. Impact Bar – Cleat: A straight bar is placed on a flat road, perpendicular to the vehicle's direction of travel. The vehicle travels straight over the cleat. The size of the cleat and

speed of the vehicle depend on which characteristic of ride is studied, although it is often used for secondary ride [4].

3. Belgium Blocks: Similar to a cobblestone road, a vehicle is driven in a straight line over the blocks where secondary ride and higher frequency accelerations are studied.
4. Proving Grounds and Test Tracks: Often specific to an OEM, proving grounds can be a collection of maneuvers and obstacles placed in different parts of a test track for both ride and handling evaluations.
5. Step Steer: Part of the standard ISO 7401 for studying the lateral transient response of road vehicles where a step SWA is applied to the vehicle for several seconds to allow the vehicle to reach equilibrium [54]. The vehicle is driven in a straight line until the SWA is applied as fast as possible. The magnitude of the SWA and vehicle speed are determined to produce a certain vehicle lateral acceleration. This maneuver is considered open loop steering.
6. Sinusoidal Steer: Part of ISO 8725 and complementary to ISO 7401, where one period of a sinusoidal SWA signal is applied to the vehicle to study its lateral transient response. The throttle input is not changed during this open loop test [55].
7. Sine Sweep or Frequency Response: Part of ISO 7401, a vehicle is first driven in a straight line where a specified sinusoidal SWA is gradually applied at an increasing frequency up until a specified maximum frequency. The vehicle speed and SWA are set to achieve a certain lateral acceleration, determined during a step steer test first. This maneuver is also open loop.
8. Steady State Circular Test: This test is for studying the steady state characteristics of a vehicle. Part of ISO 4138, the vehicle is driven to negotiate a turn at constant speed, constant driver SWA, or on a constant radius track [55].
9. Double Lane Change (DLC): Part of ISO 3888-1 [56], this maneuver involves driving a vehicle through a set of cones that replicate a severe obstacle avoidance maneuver. The vehicle exits one set of cones, enters a second lane, and then immediately returns to the first lane. The vehicle cannot contact a single cone. The maneuver is repeated while the speed of the vehicle is increased after each successful execution. The width of the lanes is dependent on the vehicle track. This maneuver is considered closed loop.
10. Slalom: Similar to ISO 13674-1, a vehicle is driven at nearly constant speed through a set of cones aligned in a straight path. The spacing between the cones is constant and the

vehicle speed is increased after each successful run. This maneuver is not representative of real driving conditions, but it is used to study the vehicle's lateral dynamics at lower lateral accelerations than the DLC .

Concerning the published literature and research presented herein, it can be identified that none of the works completely encompass all aspects of vehicle dynamics, ride, and handling in response to suspension damper alterations, or specifically semi-active dampers. Part of the novelty of this research is to incorporate a significant amount of the vehicle's dynamics and driving scenarios to ensure the behaviour of the semi-active suspension modelled is studied completely. Thus, a larger envelope of vehicle behaviour is studied, concerning the vehicle's lateral, longitudinal, vertical, yaw, pitch, and roll dynamics. The next section of Chapter 3 presents a review of previous published work on vehicle subjective ride and handling evaluations. This review provides insight on what ride and handling subjective metrics or question should be considered when developing a new subjective evaluation method for semi-active suspensions.

3.2 Subjective Evaluation of Ride and Handling

For subjective evaluations, an experienced driver is often used to evaluate vehicle ride and handling. Rather than objective metrics that quantify the vehicle's performance, subjective rating scales have been used by drivers to rate the quality of a vehicle's performance. With subjective evaluations, there are additional factors that affect the driver's perception of a vehicle's performance, compared to objective evaluations. Such factors are highlighted in the proceeding chapter sections. The goal of this chapter is to identify the details of the methods for ride and handling subjective evaluations of vehicles or virtual models in the case of a driving simulator.

3.2.1 Methods for Subjective Evaluation

To date, subjective evaluations for ride and handling do not have a standardized method or procedure to be followed by OEMs and researchers [24], [57], [58]. This aspect is true despite the fact that subjective evaluations of vehicles are required before the final product is released on the market [24]. Moreover, OEMs and carmakers tend to keep their procedures hidden, resulting in a scarce amount of industry-published literature on vehicle ride and handling subjective evaluation, compared to the work published on objective ride and handling evaluation. To date, most of the literature available pertains to ride and handling subjective evaluations of real vehicles, since high-fidelity virtual models and simulation tools have only recently become more popular among OEMs. Finally, subjective evaluations on vehicles where only changes to a semi-active suspension are made are also scarce. This research gap presents a need for a subjective evaluation method tailored specifically for the case of developing or tuning semi-active suspensions.

As for the work published to date on subjective evaluation methods for ride and handling, the generally accepted approach has been to have expert drivers drive a vehicle and evaluate its performance using some form of a questionnaire [24]. The questionnaire involves a set of subjective metrics or questions with a corresponding rating to be either selected or written by the expert drivers [57], [58]. Expert drivers are discussed later in Section 3.2.4. In [24], two general types of questionnaire evaluations are emphasized – a rating method and a ranking method. The later of the two corresponds to expert drivers determining the relative performance of different vehicle configurations to a baseline, where the absolute performance of the vehicle is not considered. This method also requires the drivers to recall all vehicle configurations being studied at the same time. As a result, the rating method has been the dominant method since the early 1970s with subjective ride and handling evaluations since different vehicle configurations are evaluated on an absolute scale. Such a method allows the data collected from several rating-based subjective evaluations to be combined for a larger database [24], [57], [58]. There is a split in the opinion on whether to let the drivers have the freedom to choose certain maneuvers, or to follow a set of predefined circuits and obstacles for a more controlled environment. In either case, it is common for closed loop driving maneuvers to be executed in subjective ride and handling evaluations. Finally, the drivers are typically kept unaware of the specific changes that were made to the vehicle in which they are evaluating. This approach avoids a bias or any expectations for the driver which could negatively affect the accuracy of their subjective ratings. Therefore, these general practices will be adopted for the subjective evaluation method on semi-active suspensions.

At the University of Leeds, several subjective studies have been completed on vehicle handling. In [22], eight expert drivers evaluated sixteen different vehicle configurations. The differences in the configurations were related to front and rear roll stiffness, tire characteristics, damping characteristics, and other suspension aspects. No predefined maneuvers were forced upon the drivers to execute. Instead, the drivers completed the subjective evaluations on the Motor Industry Research Association's proving grounds consisting of a steering pad, closed handling circuit with a sudden braking area, a ride and handling circuit, and a high-speed oval circuit. The questionnaire had forty-nine subjective metrics and used a rating scale from one (worst) to seven (best). A similar study was completed in [59] where a feedback section was added to the questionnaire to assess the quality of the evaluation method and drivers' verbal comments were recorded for additional feedback on the vehicles' performances. This aspect will be considered when developing the questionnaire on semi-active suspension. Due to the limited affect that dampers have on vehicle ride and handling, the questionnaire for semi-active suspension will be shorter and focus on only significant maneuvers and metrics which excite the dampers and capture their performance impacts. Furthermore, an absolute scale will be utilized, rather than the relative scale in [22].

Volvo Cars Corporation in partnership with the Royal Institute of Technology (KTH) has completed two extensive subjective evaluation studies on the handling when altering a physical vehicle's suspension, powertrain, and wheel subsystems [57], [58]. Adding semi-active dampers was among one of the alterations made to the vehicle being studied. However, details regarding the damper control strategy and what individual contribution of the semi-active dampers had on the vehicle performance were not stated. Seven expert drivers were used for these studies. The general procedure was to have the drivers drive each vehicle configuration at Volvo's proving grounds and fill out a questionnaire with subjective metrics. The questionnaire had several levels, depending on the complexity of the subjective metric and used the SAE J1441 one to ten absolute subjective rating scale [57]. At the beginning of each testing day, the drivers completed a first impression test to keep familiarity with the vehicles being studied. For the subjective evaluation method developed in this research, experienced drivers from Stellantis are already familiar with the vehicle's modes being studied, thus a first impression test is redundant. However, the drivers will have the option to briefly test drive the models to become comfortable with the simulator environment. The questionnaire will have one level of metrics so that is concise, easy to read and understand, and to be consistent with Stellantis' already proven subjective evaluation techniques. A final common aspect of subjective handling evaluations is the use of the closed loop ISO single or double lane change maneuvers. These maneuvers have been used extensively for evaluating the lateral, roll, and yaw dynamics of vehicles in at least ten published studies before 2002 [24]. Dampers play a role in suppressing vehicle accelerations during such maneuvers; thus, they will be considered herein.

Subjective evaluations of vehicle ride are inherently complex, since human physical and psychological factors have an impact on the driver's perception of the accelerations they are experiencing [60]. Driver physique, fatigue, and threshold sensitivity of acceleration can all affect the driver's perception of vehicle ride [24]. Expert drivers at OEMs are used for subjective ride evaluations due to their knowledge of such factors, as they are also the ones to evidently sign off a vehicle during the development stage [24]. Additionally, the accelerations felt by the driver are a combination of several accelerations in different directions, such as roll, pitch, and vertical accelerations when driving straight over a rough, asymmetric road. Thus, it is sometimes difficult to evaluate a single ride characteristic of a vehicle. This is one reason why it is important to isolate the different accelerations experienced by a driver during ride subjective evaluations to tune the vehicle for different types of excitations. Therefore, the developed subjective evaluation method will incorporate road profiles from Stellantis' proving grounds which isolate certain ride aspects of a vehicle.

Regarding the methods for ride subjective evaluation, the process is similar to handling evaluations in respect to the use of different maneuvers for different ride characteristics and a questionnaire with subjective metrics. The subjective evaluation of ride in [49] involved evaluating primary and secondary ride in the vertical, pitch, and roll directions of physical vehicles. They found that thirty percent changes in damper curve characteristics produced noticeable differences by the expert drivers at Volvos proving grounds. As with the handling subjective evaluations, a one to ten SAE rating scale was used, and a baseline vehicle was evaluated first as a reference. Work by GM also emphasized the use of expert drivers, proving grounds with a variety of different road profiles, and a similar rating scale for subjective ride evaluations [41], [47]. In [47], damping was varied from ten percent softer to fifteen percent harder than a baseline sport utility vehicle and found that this was noticeable by an evaluation team of two expert drivers. To conclude, many subjective evaluations with OEMs and institutions conducting research on subjective evaluations of vehicle ride and handling utilize a small group of expert drivers evaluating several vehicle configurations with a questionnaire and list of subjective metrics. Although this is not a standard, it is a well-accepted and successful procedure for such evaluations.

A common issue with subjective evaluations on physical vehicles is the time and resources required to execute them. Vehicles must be adapted to include measurement equipment, allow for the alteration of certain components to be varied in a study, and maintain tire performance over several days of driver. The use of a driving simulator to replace the subjective evaluations of intermediate design changes of such vehicles could promote a reduction in time and cost of resources. In this case, the subjective evaluation method can replicate the practices used with physical vehicles in a more controlled, safer environment. Since changes to a virtual model require a matter of seconds, the evaluation process can be done in a timelier manner as well.

3.2.2 Subjective Evaluation Metrics and Maneuvers

Unlike objective metrics, subjective metrics consider qualitative aspects of a vehicle's dynamics and do not require computation from measured variables. Subjective metrics are qualities that address different aspects of vehicle ride and handling. However, the subjective ratings chosen by drivers are quantitative and the ratings are what allow OEMs to subjectively quantify and compare their vehicle's performance with others. Much like objective evaluations, subjective evaluations involve the execution of several maneuvers to isolate a vehicle's many response characteristics. Table 2 summarizes the available information on maneuvers and corresponding subjective metrics used for evaluations on ride and handling.

Table 2: Subjective Metrics and Maneuvers from Published Literature

Metric Type	Metric Name / Description	Maneuver / Obstacle Details	Published Work
Ride	Body Motion Smoothness – Primary ride in the vertical, roll, and pitch directions	Normal, non-severe road disturbances	[41]
	Body Motion Control – Secondary ride	More severe undulating road profiles at higher vehicle speeds	[41]
	Shake – Secondary ride	Road profiles exciting unsprung mass natural frequencies	[41], [49]
	End of Travel Performance – Impact felt on severe disturbances	Severe events causing the limits of the SWS to be encountered	[41]
	Absorption Capability – Absorption and dissipation of secondary ride disturbances	Road profiles with small stone-sized disturbances	[47]
	Jounce Bumper – Bump impact felt by driver	Rough road profiles with obstacles such as potholes	[47]
	Ride Balance – Pitch stability	Cross ditch followed by a flat road	[47]
	Primary Ride Control	Hällered Proving Ground (HPG), Volvo	[49]
	Primary Ride Comfort – Accelerations felt by the driver	HPG, Volvo	[49]
	Choppiness – Secondary Ride	HPG, Volvo	[49]
	Rolling Feel – Secondary Ride	HPG, Volvo	[49]

Handling	Turn-in Response – Body Roll Rate	Transient cornering	[22], [24], [59]
	Recovery from Obstacle Avoidance	Single lane change, DLC	[22], [24], [59]
	Controllability – Tire slip	Single lane change, DLC	[22], [24], [59]
	Limiting Behaviour	Single lane change, DLC	[22], [24], [59]
	Steering Wheel Activity, Quickness in Car Response, Roll Motion and Roll Motion Velocity, Turn-In	ISO DLC	[48]
	Stability, Controllability, Capacity Feel	Handling circuit or proving grounds with handling circuit, slalom, and other features	[57], [58]
	Maneuverability, Steering Effort, Cornering Stability	Lane change maneuvers	[61]
	Controllability, Steering Effort	Slalom (18m x 11 cones)	[62]
	Controllability, Rollover Stability, Steering Effort	ISO DLC	[62]

In general, most handling subjective metrics focus on the driver's perception of the vehicle lateral acceleration, yaw rate, roll rate, roll angle, and pitching motion [24]. The ride metrics focus on the acceleration felt by the driver. Many of the vehicles tested in the studies references in Table 2 had alterations made to their passive damping characteristics. Thus, several of the metrics in Table 2 are adopted and tested for their significance in the subjective evaluation method being developed in this research dealing with semi-active dampers. For an example of an SAE subjective rating scale, see [63]. A gap in the literature has been identified regarding the subjective evaluation of purely semi-active suspension and its individual impact on passenger vehicle ride and handling. Later sections of Chapter 3 expand this research gap to include the use of a dynamic driving simulator.

3.2.4 Drivers for Subjective Evaluation

The group of drivers for subjective evaluations is often called a jury in published literature. The goal of the jury is to provide an accurate subjective evaluation of a vehicle or product which benefits the general consumer population. The size of the jury has varied between projects carried out, but a minimum of seven to ten expert drivers has been considered acceptable for evaluating attributes of vehicle performance even if the drivers are not completely representative of the entire population [22], [24], [49], [57-59], [64], [65]. However, examples of studies with a smaller group of expert drivers for GM and other associations can be found in [47], [66]. An "expert" driver is considered as an individual having experience and knowledge of ride and handling performance indicators, vehicle dynamics, driving vehicles, and evaluating vehicle performance [57], [58].

These drivers have the capacity to complete demanding closed loop maneuvers and evaluate the vehicle at the same time. According to [24], these “trained drivers” are sometimes accepted by OEMs to represent the opinions of the consumers. Furthermore, the use of expert drivers avoids the issue of random customer drivers being inconsistent in their evaluations and difficulty with understanding the subjective metrics [67].

For Stellantis, a smaller group of expert drivers otherwise known as performance engineers are used for studies on certain vehicle platforms and projects. At Stellantis, one or two performance engineers already familiar with a certain vehicle will carry out the subjective evaluations on that vehicle. Part of these engineers’ occupation is to drive and tune a vehicle model to match a production vehicle. When implementing new products, such as a semi-active suspension controller, the performance engineers visit the ARDC simulator lab and drive the DiM 250 while subjectively evaluating the product’s ride and handling. At Stellantis, the term “jury” refers to a group of drivers simply representing the general population, including managers and other employees, which typically drive a vehicle at the end of a project once it has already been evaluating by the performance engineers. In the end, OEMs will first develop, subjectively evaluate, and produce vehicles with one or two performance engineers before a jury representing the general population drives the OEM’s product.

3.2.5 Correlating Objective Metrics and Subjective Performance

In a majority of the referenced literature on subjective evaluations in the previous section, a correlation study between objective metrics and subjective ratings has been done. Linear regression, or method of least squares, was used in [22], [57], [58], and [64] to determine the relationship between handling objective metrics and driver ratings on handling. It was considered that a correlation coefficient above 0.7 resulted in well-correlated results [57], [58]. Neural networks have also been used to combine data from several subjective studies and for non-linear correlations, such as in [24] and [58], but this approach is only possible if data is available for training and a large data set results from the evaluations. There has not been a subjective evaluation study with purely semi-active suspension changes to a passenger vehicle, so this is not a logical choice for this research. However, linear correlation will be implemented in this research. To improve the accuracy of the correlation studies, the maneuvers completed during objective and subjective evaluations should be done as close as possible with high repeatability [68]. A correlation study in this research will provide guidelines for determining which objective metrics can be used to predict the subjective performance of a semi-active suspension. The DiM 250 dynamic driving simulator provides a more controlled environment than subjective evaluations on a physical vehicle. As a result, the correlation between the objective metrics and subjective ratings from this research provide a more accurate representation of the objective-subjective relationship between metrics for evaluating semi-active suspension.

3.3 Semi-Active Suspension Controller Evaluations

This section explores the research published on the development of semi-active suspensions, the methods used to evaluate their ride and handling, and which products are commercially available. Note that the concept of semi-active dampers has been studied since the first paper was published on "semi-active isolators" in 1974 [38], at least objectively. Since then, many types of semi-active control strategies, virtual controller models, and physical products have been created. As will be discussed at the end of this chapter subsection, the practical implementation of semi-active dampers by suppliers and OEMs is usually based on the robust VSC 'classical' skyhook control strategy [25].

3.3.1 Types of Controllers Modelled and Evaluated

A significant amount of research has focused on several well-established semi-active control strategies that have yet to be subjectively evaluated. The switchable strategy, which consists of a controller switching between different damper curves was studied in [69] and [70] in response to random road vertical excitations. In both studies, only the vertical dynamics of a quarter-car or half-car model were studied, while focusing on the objective ride performance of the vehicle model with the switchable damper controller replacing the conventional damper in the model. A gain scheduling technique was utilized to choose predefined values for a set of gains for the damper forces, depending on the magnitude of the SWS. These gains amplify the damping force and add damping during excessive damper deflections. Both studies also considered the DTL to address some handling characteristics [69][70]. As a result, both studies proved that the ride of a passenger vehicle could be improved with this strategy. However, these studies neglected the use of nonlinear damper curves, which has a significant impact on the frequency response of the vehicle. In this research, non-linear damper curves are used with the studied vehicle models.

The classical Skyhook semi-active control strategy has been studied by [39] with a 7-DOF full vehicle model, incorporating the vehicle body's roll, pitch, and vertical dynamics. Both discrete and sinusoidal road excitations were applied to the vehicle suspension. The objective ride study in [39] showed that the Skyhook strategy could improve control of the sprung mass's motion over a reference passive suspension. Two universities also studied the performance of a skyhook damper controller, with a quarter-car model excited by vertical step inputs [71], [72]. In both cases, the skyhook control strategy was used to model a semi-active damper and the results indicated that the vehicle model's objective ride performance was improved over a baseline passive suspension. MathWorks's Simulink was used to model the controllers. Another study presented in [73] expanded the use of road excitations to include white Gaussian noise input, and found that the Skyhook control strategy for semi-active suspension still improves control of the sprung mass over passive suspension, objectively. Finally, the study conducted in [74] modified the Skyhook strategy to apply a continuous change in the damping rates, rather than switching between a maximum and minimum damping setting. A half-car vehicle model was excited by a 0-20 Hz sinusoidal signal with an amplitude of 1 cm and found the modified strategy to retain the improvement of sprung mass control over a passive damper, but lost control of the unsprung mass resulting in higher DTL [74]. In each of these studies, a virtual controller for Skyhook semi-active dampers was implemented in a vehicle model to replace the passive dampers. All of these studies have focused on objectively evaluating the ride performance of the vehicle mode. Thus, it is evident that the Skyhook strategy has yet to be subjectively evaluated on a simulator. Additionally, the objective evaluations were focused only on vertical road excitation while neglecting the possibility of driver inputs to steering and accelerating. These aspects are considered in this research.

The research conducted in [39] and [72] also objectively evaluated the performance of the Groundhook and Hybrid semi-active control strategies. In [39], the ride performance of the Groundhook strategy deteriorated compared to a passive suspension, but significantly improved control of the unsprung mass motion. An optimization technique in [39], called h-infinity control, was used to find the optimal value for the α tuning parameter (see Chapter 2.3.3) to optimize the tradeoff between the sprung mass control of the Skyhook strategy and the unsprung mass control of the Groundhook strategy. An α-value of 0.5 was found to give the best overall performance over a passive suspension for the single excitation studied. The Groundhook and Hybrid semi-active control strategies were also objectively evaluated in [75] while analyzing the response of a quarter-car model to sinusoidal and step road excitations. The simulation results were validated with an MR damper on a quarter-car test rig. The results supported the overall improvement of the Hybrid control strategy with an α tuning parameter equal to 0.5 [75]. A very recent study of the Groundhook strategy was completed by a Formula SAE university team which co-simulated an Adams MBS vehicle model and a Simulink Groundhook-modified semi-active suspension controller [76]. A single lane change maneuver and a swept-sine maneuver were simulated to objectively evaluate the handling performance of the vehicle model. The DTL was improved with the modified Groundhook strategy, but a significant amount of high-frequency content was generated in the sprung mass's response, thus deteriorating the vehicle's ride. This study emphasizes the need to consider multiple maneuvers which excite as much of the vehicle's dynamics as possible to avoid unprecedented performances.

Since the early 2010s, there has been a shift in academic research and development of virtual semi-active control strategies, focusing on fuzzy-logic controllers. Studies presented in [28], [29], and [77-79] developed fuzzy logic controllers for a semi-active suspension on vehicle models ranging from quarter-car models to full 11 DOF vehicle models. Essentially, a fuzzy logic controller uses a large set of "if-then" logical statements to determine the best damping rates for the damper based on the vehicle's dynamics. The creation of these logical statements requires expert knowledge [77]. The full vehicle models in [29], [77], and [78] were also validated with experimental tests with a four-post shaker machine. In [28] and [29], preview control algorithms were developed to allow the vehicle model to "see" the oncoming road profile in their offline, objective simulations. These studies presented significant improvements of primary ride performance over a passive suspension, but the controller computational demand is significantly increased due to the additional control steps for the preview control, optimization strategies, and the fuzzification and defuzzification steps in the fuzzy logic controller. Not to mention, the studies assumed the vehicle was capable of measuring the oncoming road profile, where practical implementation of hardware and measurement precision were neglected. A notable addition to a fuzzy-logic controller was created in [78] to better reflect the architecture of modern semi-active suspension systems in the automotive industry – a "Triple-Mode-Controller". This controller had a normal, sport, and comfort modes which could theoretically be selected by the driver while driving a real vehicle. In this case, each mode was tuned for maximizing either ride or handling.

However, maneuvers simulated were restricted to straight line driving and only vertical excitations on the vehicle model's four tires. The performance of the fuzzy logic controller in [79] was similar to and in some cases inferior to the performance of the Skyhook strategy. Note there was no subjective evaluation of such controllers

A few more recent studies have focused on optimal control strategies such as sliding mode in [7], clipped optimal control in [80] and [81], and a multiple-objective evolutionary algorithm in [82]. The objective studies of vehicle models with the sliding mode control resulted in performance improvements similar to that of the Skyhook control strategy [7]. McLaren spent years developing a clipped optimal control strategy to improve its vehicles ride and handling [80]. An experimental vehicle was equipped with their semi-active suspension system and objectively evaluated on a bumpy roundabout [81]. The results indicated a "better performance compromise" compared to two fixed damper settings [81]. In [82], a multi-objective algorithm was set at finding a group of damper settings which produced improvements ride and handling over a reference passive suspension. The findings suggested that tradeoffs between ride and handling still exist with semi-active suspensions based on discrete damper curve settings.

In all of the semi-active control strategies objectively evaluated, there has yet to be a published project dedicated to subjectively evaluating a known semi-active control strategy on a dynamic driving simulator or a physical vehicle. This feature is the major contribution of the project, where a semi-active control strategy with noticeable improvements over a passive suspension is subjectively evaluated on a simulator.

3.3.2 Ride and Handling Metrics Considered

A summary of the objective metrics and maneuvers used to evaluate the different semi-active control strategies has been summarized in Table 3. Note that any maneuver with an asterisk next to it represents a maneuver for which steering inputs were applied to the vehicle model.

Table 3: Metrics and Maneuvers for Evaluating Semi-Active Control Strategies from Literature

Metric Type	Metric Name / Description	Maneuver (* for steering inputs)	Published Work
Ride	Sprung Mass Vertical Acceleration (RMS or PSD)	Random road excitation , Sine sweep,	[29], [39], [69], [70], [73], [74], [77], [78], [82]

Ride (cont.)	Sprung Mass Vertical Acceleration (P2P)	Bump, Cleat/step, Sinusoidal input, *Bumpy roundabout	[28], [71], [72], [75], [80], [81]
	Sprung Mass Vertical Displacement (P2P)	Bump, Cleat/Step, Sinusoidal, Sine sweep	[71], [72], [75], [79]
	Sprung Mass Roll Acceleration (RMS or PSD)	Sinusoidal, 'Chuck hole', Sine sweep, Measured road profile	[29], [39], [77], [78]
	Sprung Mass Pitch Acceleration (RMS or PSD)	Sinusoidal, 'Chuck hole', Sine sweep, Measured road profile	[29], [39], [77], [78], [82]
	Sprung Mass Vertical Velocity	*Bumpy Roundabout	[80], [81]
	Unsprung Mass Vertical Displacement (P2P)	Bump, Cleat/Step, Sinusoidal, Sine sweep,	[72], [75], [79]
	Unsprung Mass Vertical Acceleration (P2P)	Step, Sinusoidal	[75]
	Roll Angle	Sine sweep, *Bumpy roundabout, Chuck hole	[39], [74], [80], [81], [82]
	Pitch Angle	Chuck hole (trapezoidal pothole)	[39]
Handling	Dynamic Tire Loading	Bump, Random road excitation, *Single lane change, *Frequency response	[28], [39], [69], [70], [76]
	Dynamic Tire Deflection	Bump, Cleat, Sine sweep, Sinusoidal, *Single lane change, *Frequency response	[71], [72], [73], [74], [76], [79]
	SWS	Random road excitation, Bump, Cleat, Sinusoidal, Sine sweep	[28], [29], [39], [69], [70-73], [77], [78], [79]
	SWA	*Bumpy roundabout	[80], [81]

The only references that considered steering inputs are for the studies in which a modified Groundhook strategy showed unsatisfactory performance [76] and a clipped optimal control strategy created a better compromise over two fixed damper settings [80], [81]. There is a disconnect between objective evaluation of ride and handling between vehicles in general and the semi-active control strategies previously mentioned. Note that a significant majority of evaluations on semi-active suspension has been limited to evaluating ride objectively, straight line driving without driver input considerations, and vertical excitations for the vehicle. A braking event, a step steer, a slalom, or a full-loop test track for evaluations have not been considered to date. A comprehensive objective and subjective evaluation including ride and handling maneuvers which excite vertical, lateral, longitudinal, yaw, pitch, and roll motion would allow engineers to study the complete effect of semi-active suspension on OEMs' vehicles. Such a study has yet to be done subjectively with a dynamic driving simulator. This project fills this research gap by considering such aspects of evaluating the ride and handling of a semi-active suspension.

3.3.3 Comparison of Controller Types

Regarding the ride and handling performance of the semi-active control strategies, the control strategies discussed in Section 3.3.1 have different performance improvements over a reference passive suspension. Table 4 is a summary of the most notable performance improvements or worsening of the control strategies found to date. It is important to note that the passive suspension or passive dampers are not the same for all literature. Furthermore, the type of road excitations and driving parameters are not the same, please see the referenced literature for such details. Thus, the results between each study cannot be directly compared, but they do give insight on which control strategies have significant performance advantages.

Table 4: Notable Objective Performance Improvements of Semi-Active Control Strategies

Control Strategy	**Objective Metric Improvement or Worsening** (Relative to a passive suspension)
Switchable	A. 8% reduced SWS at front and rear [69] B. 3% and 12% reduced DTL at front and rear, respectively [69] C. 14% reduced body vertical acceleration [70]
Skyhook	A. 50% reduced seat vertical acceleration [39] B. 7% increased SWS [39] C. 40% reduced P2P roll angle [39] D. 35% reduced P2P pitch angle [39] E. 67% reduced body vertical acceleration at 10Hz [39] F. 27% reduced body P2P vertical displacement [72] G. 36% reduced SWS [72]
Groundhook	A. 300% increased body vertical acceleration at 10Hz [39] B. 27% increased P2P roll angle [39] C. 22% increased P2P pitch angle [39] D. 46% reduced SWS [72] E. 23% increased body vertical P2P acceleration [72]
Hybrid	A. 50% reduced body vertical acceleration at 10Hz [39] B. 20% reduced P2P roll angle [39] C. 20% reduced P2P pitch angle [39] D. 25% reduced SWS [72] E. 22% reduced body vertical P2P acceleration [72]
Fuzzy Logic	A. 11%, 1.5%, and 14% reduced body vertical , pitch, and roll RMS accelerations, respectively [77] B. 16% reduced front left SWS [77] C. 40% increased and 30% reduced body vertical acceleration at 10Hz for sport and ride modes, respectively [78]

Others	A. 91% reduced body vertical P2P acceleration (fuzzy logic with preview) [28] B. 88% and 83% reduced SWS and DTL, respectively (fuzzy logic with preview) [28] C. 19%, 9%, and 7% reduced body vertical , pitch, and roll RMS accelerations, respectively (fuzzy logic with preview) [77] D. 33% reduced body vertical RMS acceleration (sliding mode control) [28] E. 30% reduced body vertical P2P acceleration (linear quadratic regulator) [28] F. 35% reduced SWS (linear quadratic regulator) [28]

The most popular control strategies for semi-active suspension appear to be the Skyhook, Groundhook, and Hybrid control strategies. The Skyhook strategy shows significant improvements in damping the sprung mass accelerations, but loses control of the unsprung mass. In [39], it was found that mid-to-high frequency content was generated in the unsprung mass's vertical motion when the vehicle was excited by road inputs. On the other hand, the Groundhook results in the opposite performance improvements, where control over the sprung mass deteriorates when improving control over the unsprung mass. The Hybrid strategy shows a good compromise of the Skyhook and Groundhook strategies, improving the damping or control of the sprung and unsprung masses compared to a passive suspension. The fuzzy logic strategies, being more complex and requiring more computing steps, show similar and sometimes inferior performance to the Skyhook or Hybrid strategies. When preview control is added to the fuzzy logic strategies, the performance improvements are the most impressive, but published work on the details of creating fuzzy logic and the actual measuring of an oncoming road profile are scarce. Not to mention, the tuning capabilities of the Hybrid strategy and the Triple-Mode-Controller would allow the driver to choose between independently improving handling or ride performance. Considering the goal of this research to create a subjective evaluation method for vehicle semi-active suspension and the performance improvements of the well-established control strategies, the Skyhook, Groundhook, and Hybrid strategies were chosen to be evaluated. Furthermore, the option for the driver to choose between different suspension modes will also be implemented with these strategies, thus incorporating a simplified and manual switchable logic.

3.3.4 Semi-Active Suspension in Production Vehicles

OEMs have recently begun to increase the use of semi-active suspension technologies, such as semi-active dampers, with their passenger cars [25]. Several systems have been developed, such as controlled electronic suspension, continuous damping control, dynamic chassis control (DCC), and continuously variable damping [25]. Most of these systems use sensory information from chassis accelerometers, ride height sensors, driver inputs, road conditions, and other information, but the details behind the control strategies or algorithms are kept secretive by OEMs, suppliers, and other institutions. One aspect that has been noted in [25] is that the semi-active control algorithms developed by OEMs or suppliers are usually based on the Skyhook strategy. In some cases, such as the DCC system by Volkswagen, the semi-active systems provide different damping modes (normal, comfort, or sport) for the driver to select by the push of a button. A recent study on Renault's "Multi-Sense" system discovered objectively that the sport mode is aimed at improving vertical primary ride, higher yaw rates versus SWAs, and higher lateral acceleration versus SWAs while providing better road holding or grip [83]. The comfort mode was found to improve ride quality overall at the expensive of increased SWS. It was also noted that with many vehicles having such driving modes, semi-active dampers are not the only components of the vehicle being controlled. Aspects of the engine, transmission, electronic stability control, and steering systems are also continuously controlled to further improve the vehicle's ride and handling. Thus, the use of semi-active dampers is one of many electronically-controller aspects of modern vehicles for improving ride and handling. The individual contribution from the dampers is one fraction of the overall ride and handling improvements.

3.4 Evaluation of Vehicle Ride and Handling with Dynamic Driving Simulators

This final section of the literature review explores the published research on evaluating vehicles with the use of dynamic driving simulators. Dynamic or motion-based driving simulators avoid issues such as repeatability, inefficient time use, and higher testing costs associated with physically testing vehicles [8], [68]. Dynamic driving simulators can also be used during all stages of vehicle development before a physical prototype is fabricated [8]. This approach permits an opportunity for developing vehicle applications quicker and safer than in the past. The research discussed in the following section focuses on the use of such simulators to evaluate vehicle performance, where research on other simulator applications can be found in [8] and [16]. Firstly, the simulator used in this research is compared to other simulators in the automotive industry. Next, a brief discussion of motion sickness with driving simulators is presented, followed by a review of the research done on vehicle ride and handling with dynamic driving simulators. Note there is a lack of research in the area of ride and handling on dynamic simulators and no significant research in the area of semi-active suspension on dynamic simulators.

3.4.1 Comparing VI-grade's DiM 250 Simulator with Others

A recent literature review on dynamic driving simulators was published in 2015 which focused on classifying and comparing high-, mid-, and low-level simulators [84]. The classification is based on simulator fidelity, useability, complexity, and cost. High-level dynamic simulators are classified as providing full vehicle dynamics, a wide field of view, a complete cabin with significant functionality, and a minimum of 6 DOF [84]. The DiM 250, see Figure 3 in section 1.3, could also be classified as high-level with these requirements. The DiM 250 has 9 DOF as a result of a Stewart Platform and sliding tripod combined architecture, 50Hz maximum vertical frequency, 1000kg maximum payload, a real vehicle cabin mounted on top, active seats and seat belts, shakers for NVH, over 1 m of travel in the x- and y-directions, only requires 12x12x5 metres of space, and several other features as presented in [85]. The sliding table of the DiM 250 levitates on several microns of air over a flat metal platform, avoiding the friction induced on other driving simulators which use a rail system instead. The combination of the Stewart platform, tripod, and shakers in the cabin separate high and low frequency motions of the vehicle. Essentially, the driver is immersed in a virtual environment which replicates the real driving experience including visual, audio, haptic, and motion feedback to the driver of the DiM 250. This simulator has been used for steering and handling subjective evaluations when changing a vehicle's anti-roll bar configuration [68]. As described in [84], mid-level simulators have less DOF, lower fidelity cabin and motion platform structure, and produce visual, audio, haptic, and motion cues. Low-level simulators are described as compact or static simulators consisting of a PC, steering wheel, and a monitor or screen for visual information.

Other high-level dynamic driving simulators such as the Swedish National Road Transport Research Institute's SIM IV or Daimler's dynamic simulator created in 2010 have 8 DOF. Both of these simulators combine a hexapod with a rail system for increased lateral or longitudinal travel. The SIM IV is mainly used for evaluating heavy-duty vehicles, has significantly lower maximum translational accelerations, but provides more translational travel than the DiM 250 [86], [87]. Daimler's dynamic simulator can provide up to 12 m of lateral travel for simulating a vehicle driving across several lanes during lane change maneuvers and has been used for studying driver assistance control systems [88]. Two other similar, high-level simulators that are used for evaluating vehicle ride or handling are FKFS's Stuttgart driving simulator and the University of Leeds's driving simulator. Both of these simulators also have 8 DOF produced by a hexapod and rail system architecture [8], [89]. Each of these 8 DOF simulators do not have the same yaw motion capacity as the DiM 250 due to the absence of the additional DOF, but do have more translational travel permitting longer sustained lateral and longitudinal accelerations. Lastly, the NADS-1 at the University of Iowa has the same 9 DOF as the DiM 250, with an additional 4 DOF through the addition of four high-frequency vibration actuators, but requires significantly more space and costs compared to the DiM 250 [8], [21], [84]. A dome encompassing a cabin with 360-degree view for the driver is mounted on top of a hexapod and rail system for providing the first 9 DOF.

Although the simulators discussed have varying performance, each of them has been proven advantageous for evaluating vehicle and driver performance. One topic that can heavily impact the accuracy of studies completed with dynamic driving simulators is motion sickness. It can affect both the accuracy of the driving experience and the total time drivers can perform studies on the simulators.

3.4.2 Motion Sickness

Dynamic driving simulators use MCAs to provide inertial motion to drivers and adjust the dynamics from simulation software to account for the simulator's travel limits [90]. Motion cues are provided by the algorithms to produce a realistic driving experience. Motion cues can take the form of audio, haptic, inertial, and visual feedback to the driver. MCAs convert vehicle translational accelerations and angular velocities into admissible driver seat motions [91]. Depending on the algorithm, a model of the human vestibular system is used to make this conversion and help create the motion cues. These cues are required to ensure the vestibular system of the driver perceives realistic motions while driving and that they correspond to what the driver sees, hears, and feels while driving the simulator. This process helps avoid motion sickness or discomfort during subjective evaluations on driving simulators. Motion sickness mainly occurs when there is poor synchronization between visual and inertial cues, causing discomfort for the driver [92]. Motion sickness has been an issue with simulator subjective evaluation in the past, as in [93] where simulator discomfort caused by inaccurate motion cueing made it difficult for drivers to maintain a certain driving speed when evaluating the ride of a vehicle. Furthermore, as found during the evaluations with a simulator on vehicle handling in [68], one driver had to stop driving due to the discomfort of the vehicle's tendency to oversteer. It was not confirmed if the root cause of the discomfort was inaccurate motion cueing, but it was clear that different drivers can be naturally prone to different levels of motion sickness. Motion cueing is already done by engineers at Stellantis, and the cueing is tailored to different types of maneuvers. Therefore, tuning of the DiM 250 motion cueing is not a part of the research conducted, but the threat of motion sickness on the simulator subjective evaluations will be considered and avoided at all costs.

3.4.3 Evaluation of Ride and Handling with Dynamic Driving Simulators

Most of the research published on vehicle performance subjective evaluations with mid- and high-level dynamic driving simulators is limited to the last five to ten years. In 2012, a model of a US military Stryker vehicle was driven using a 6 DOF ride simulator in Michigan to test the ride quality of the vehicle with and without MR semi-active dampers [93]. No formal test with a subjective rating scale or questionnaire was completed, but several engineers from the Tank Automotive Research Development and Engineering Centre, an OEM, and a supplier were asked to drive the validated Stryker model over two road profiles having differently sized bumps and Belgium blocks. The result of the study found that the engineers noticed an improvement in vehicle ride over the bump course, but due to simulator discomfort, the repeatability of the Belgium block course was poor. Moreover, the details of the control strategy for the semi-active MR dampers were only known to the supplier, where a "black box" model was used by the US military. This is the only source found in which a semi-active suspension is subjectively evaluated, only for ride of a non-passenger vehicle, and with less-than-satisfactory results. Therefore, this gap should be filled to develop a timelier and cost-saving method to subjectively evaluate vehicle ride and handling of a semi-active suspension with a dynamic driving simulator. The method should clearly indicate improvements in absolute ride and handling performance through the use of a questionnaire and rating system as suggested in Section 3.2.

In 2015, the DiM 250 dynamic driving simulator at Volvo was used to study the steering and handling performance of a passenger vehicle with different anti-roll bar configurations [68]. Objective data from both the offline simulations and physical tests were overlayed to ensure the same vehicle performance was captured by the model and vehicle. Moreover, subjective ratings recorded during the physical tests were used to validate the vehicle model subjective performance on the DiM 250. The study conducted in [68] used two expert drivers from Volvo with a goal of validating the use of a dynamic driving simulator for subjective evaluation of steering and handling. The results indicated that the DiM 250 was able to capture the same performance trends that were found during the physical subjective evaluation on the real vehicle. The testing method with the DiM 250 was to drive the same maneuvers that were simulated offline, not allow the drivers to know what changes were made to the vehicle, interview the drivers for additional feedback, and have the drivers always complete three laps on the Hällard proving grounds handling track [68]. Details regarding the questionnaire and what rating scale was used were not published with the study by Volvo, as this area of research is a new and growing field. Although this study with the DiM 250 does not involve semi-active suspension or evaluating ride, the general process of the subjective method can be adopted. In the case of this project, the subjective metrics and maneuvers will be chosen based on their ability to excite ride and handling characteristics.

In 2018, the Swedish National Road Transport Research Institute and Volvo Group Trucks tested the ability of the SIM IV dynamic driving simulator to be used in evaluations of heavy vehicles [92]. The study was focused on determining if certain ride and handling differences could be noticed by ten Volvo expert drivers when changing the front and real roll stiffness, tire properties, and the vehicle models roll understeer. A validated baseline vehicle model was first driven by the drivers to get used to the environment, and then a first test session was conducted to determine if the drivers found the driving experience realistic. A second session was then conducted, where the drivers would complete a single lane change and test the lateral dynamics of the vehicle on a straight road. In this session, the driver evaluated the baseline and four different configurations of the vehicle model, focusing only on the vehicle handling. The results of the study indicated that the simulator captured the performance changes of the different vehicle configurations and the driving experience to be realistic. Volvo and the Swedish research institute stated that the driving simulator would be a helpful tool for heavy duty vehicle development [92]. For the subjective evaluation method developed in this project, a reference or baseline vehicle with a passive suspension will always be evaluated first and if a driver requests to test out the simulator to become acquainted with the driving environment before a complete subjective evaluation. This approach will ensure the drivers can focus only on noticing the differences between the semi-active suspension controller modes.

Most recently in 2020, the University of Leeds and Jaguar Land Rover asked six expert drivers to evaluate the ride and handling of a vehicle model on the university's dynamic driving simulator [94]. The ride height of the vehicle was varied to study the effect on the vehicle ride and handling. The vehicle primary ride, secondary ride, and several handling characteristics were evaluated. No rating scale was used; the drivers were asked to state whether certain characteristics were better or worse than a validated baseline ride height performance, or to select certain word descriptors from a list [94]. A rural road with straight and changing radii sections in the UK was lidar-scanned and uploaded for driving on the simulator. The result of the study indicated that the expert drivers noticed improved ride with higher ride height and improved handling with lower ride height. Once again, it was shown that dynamics driving simulators can be used for ride and handling subjective evaluations, but still no standard for a method exists [94]. Such a research gap can be filled with the development of a subjective evaluation method for semi-active suspension. The procedures for implementing said method can be found later in Chapter 8 and can be used as a basis for future subjective evaluation methods.

In addition to the research discussed, another study on powertrain development with dynamic driving simulators in [95] and one on accelerator pedal mapping for drivability in [96] have been published. With the growing implementation of VI-grade DiM simulator technology with OEMs and simulation centers (i.e., Multimatic's simulation centre in Detroit, Michigan), there will be an increase in vehicle ride and handling subjective evaluations with dynamic driving simulators.

This concludes the review of relevant literature on objective and subjective evaluations on vehicle ride and handling, semi-active suspensions, and dynamic driving simulators. It is clear that there is a gap in the research on the subjective evaluation of ride and handling of vehicle semi-active suspension using a dynamic driving simulator. This project is aimed at completing such research for providing a subjective evaluation method using the DiM 250 simulator and a virtual model of semi-active suspension dampers with Stellantis' vehicle models.

CHAPTER 4. VI-grade Technology

This chapter presents the structure of the vehicle models being studied as well as a deeper discussion of the DIL environment with the DiM 250. First, details regarding the organization of data for vehicle models in several VI-grade software are presented, as well as the methodology on how vehicle models are transferred from the offline environment to the simulator. Next, the software, hardware, and entities involved in the DIL environment for subjective evaluations are presented. Lastly, aspects of the two Stellantis vehicles models to be studied are presented along with a discussion of how certain vehicle characteristics can affect vehicle ride and handling. Throughout the chapter, the reader should develop an understanding of the process behind creating Stellantis' vehicle models and how they are simulated in a virtual environment.

4.1 Vehicle Modelling

In addition to the vehicle modelling theory and creation of vehicle models discussed in Sections 2.1.3 to 2.1.5, this section focuses on the structure of CRT models and the software involved in the offline and simulator environments. For the simulations run during this project, the road profiles over which the vehicle is simulated or driven deserve a few notes for the reader to consider. A software also created by VI-grade, called VI-Road, allows the generation and manipulation of road geometry and graphics. Engineers can create entire proving grounds, test tracks, road geometry, and graphical aspects of the road profiles to be used in the offline and simulator environments. Furthermore, real roads can be scanned to measure, with high resolution, the 3D geometry of the road profile. The data collected from the scanning is used to create a road data file, which stores the coordinates of each point on the real road. Users of VI-road also have the option to manually enter these coordinates and other information to create their own road profiles. For instance, the friction coefficient for a specific section of a road can be arbitrarily set by the user. In this project, VI-road was used to create one of the road profiles, and several other road profiles were provided by VI-grade from the scanning of real road profiles in Canada and the USA. Please refer to VI-grade's official help documentation for further capabilities and specifics on VI-road and other software discussed in this chapter.

4.1.1 Software and Tools for Vehicle and Suspension Modeling

For this research, VI-grade's CarRealTime software is used throughout building of the vehicle models to the vehicle dynamics solver during the DIL simulations. CRT has several functionalities. First, CRT is a pre-processing software in which vehicle models can be created from K&C data or adapted from generic vehicle class models, such as a compact car, pickup, race car, sedan car, etc. In CRT, vehicle models are essentially a large hierarchy database of subsystem and property files using .xml file format. Each vehicle model is a single database or folder at the highest level, next is a collection of folders containing information files of the vehicle subsystems, damper characteristic curves, graphics, tire models, aerodynamic force mapping, engine mapping, etc. In the subsystems folder, a file for each vehicle subsystem can be found. These .xml files contain all the information regarding the lookup tables for kinematic relationships, the properties for the components of a subsystem such as the characteristic curves for dampers or the force-deflection curves for bump stops, and other information. In CRT, this information is can be altered by the user through the editing of .xml files or by changing the parameters and tables in the software interface. See Figure 20 for a presentation of the CRT user interface.

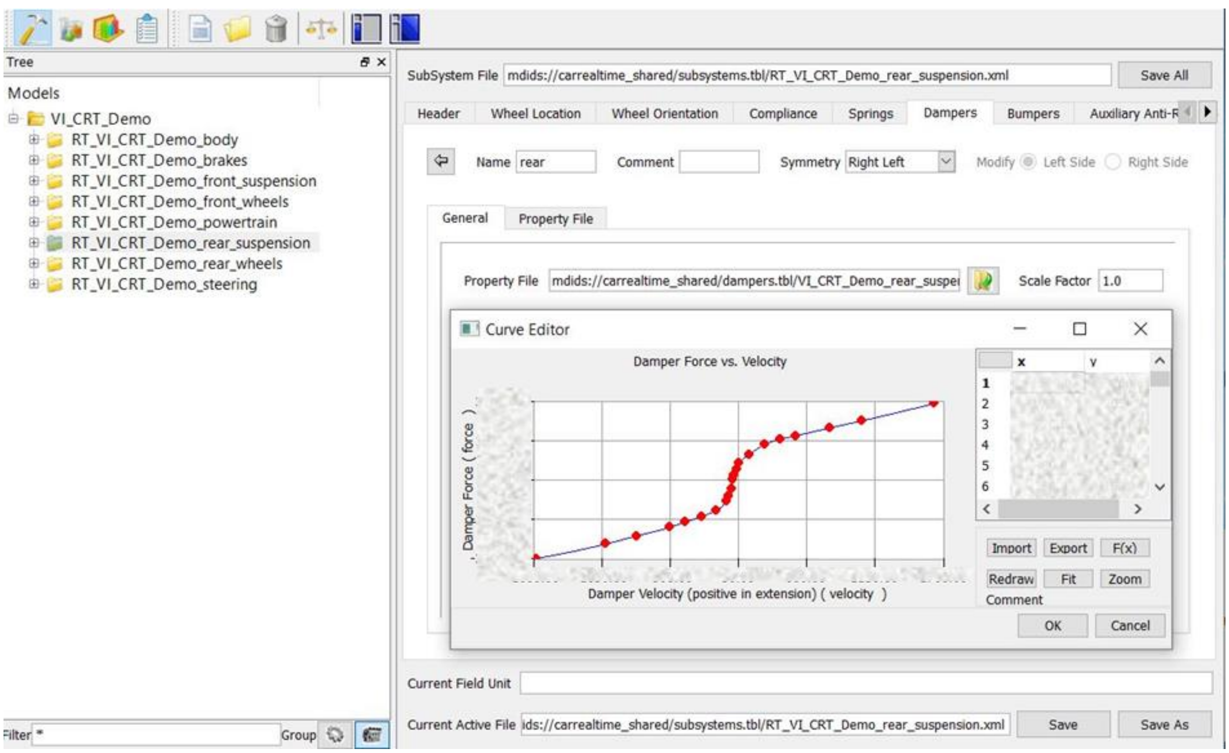


Figure 20: CarRealTime pre-processing user interface

As mentioned in Chapter 2, the dynamics of the vehicle models in CRT stem from a twin track model with a parametric description of the suspension subsystems. Lookup tables are used to describe the kinematics and compliance of the vehicle's suspension while accounting for the tire's caster, camber, and toe angles. The curves can be manually edited in the "build mode" of the CRT interface, as well as properties of the vehicle body, brakes, steering, powertrain, and wheel subsystems. Tire property files are text files used to describe the geometry and characteristics of the tire model being used for a specific vehicle. The tire property file can also be manually edited in a text-editing software to change the geometric or physical properties of the tire, as well as scaling factors and coefficients for the tire model relationships. More aspects of tire modelling are discussed in Section 4.1.3. Finally, CRT uses the VI-driver model developed by VI-grade to provide the inputs to the vehicle model. Driver parameters relating to how long a real driver would take to shift gears, how much a real driver can anticipate its own actions, and other features can be set in CRT. Each of these pre-processing aspects allows engineers at Stellantis to make timely changes to the vehicle models before building an event and running a simulation.

The second function of CRT is related to event or maneuver building. Templates for common maneuvers including open loop steering, stability maneuvers such as the fishhook, cornering, and straight-line driving can be adapted to the needs of the user for fast simulation building. Additionally, a genetic algorithm by VI-grade is implemented with the VI-driver model to test the limits of a vehicle in "press maneuver" events with cones or pylons such as a double lane change or slalom. The algorithm iteratively calculates target trajectories for the vehicle while promoting a constant vehicle speed throughout the cones. If a trajectory is completed without the vehicle hitting a single cone, then the vehicle speed is increased, and a new set of target trajectories are calculated and tested until an increase in speed does not allow for any new trajectory to be successfully completed. The user of CRT can set initial conditions for these press maneuver events. An accessory software to CRT is VI-Event Builder, where driving maneuvers can be manually created by a user. The type of driver (virtual human or robot), the vehicle path, the driver inputs, the style of driving, and other event-specific information is set by the user. This feature allows OEMs to implement their internal standard maneuvers with CRT. Default road data files are also provided by VI-grade for common driving maneuvers.

The third capability of CRT is its function as the vehicle dynamics solver in both the offline and simulator environments. In both environments, the vehicle dynamics are solved in real time. As explained in detail in Chapter 2, the equations of motion of the twin track model along with the parametric description of the suspension are solved in response to the inputs coming from the driver model used in CRT and the road. The equations of motion are solved at each time step of the simulations. Furthermore, the user can specify the resolution of the results to be recorded during a simulation and specify the mode of simulation. For instance, a live animation of the simulation can be displayed as the vehicle dynamics solver completes the computation of the vehicle's response. Once a simulation is completed, a results file is created containing the values of each dynamic variable at all timesteps of the simulations. The data in this file will be used for the objective evaluation of the semi-active suspension controller.

Co-simulations can also be run using CRT and external control system modelling software, such as MathWorks's Simulink. In this case, a control model of a component or entire subsystem can be modelled in Simulink with block diagrams and transfer functions. An .xml file is created first in CRT, which contains a reference to the vehicle model database, the road data file, mapping of the vehicle model inputs and outputs, channel or signal names, and the information which defines the type of maneuver to be simulated. In Simulink, an add-in that contains a library of CRT-specific Simulink blocks is required to configure the medium for transferring data or signals between the vehicle model in CRT and the external control system in Simulink. For this research, the dampers of the vehicle suspension are replaced by a Simulink control model of semi-active dampers. Details regarding the controller are presented in Chapter 6.

The last functionality of CRT is its post-processing ability. VI-Animator is another accessory to CRT which allows the plotting of vehicle dynamics to be displayed either during or after a simulation is run. Users can plot specific channels of the vehicle model. There is also the option to execute some mathematical operations on the results inside VI-Animator such as addition, subtraction, multiplication, and division of signals. Furthermore, more complex operations can be executed such as a Fast Fourier Transform to study the vehicle's response in the frequency domain. VI-Animator also allows the data from a simulation to be exported in a .csv file format to be further analyzed in Microsoft Excel, as one example. Other post-processing software such as Adams Postprocessor can also be used to take the results file from a CRT simulation and execute other operations on the results. More on post-processing of data from simulations is described in the objective evaluation of the semi-active suspension in Chapter 7.

As a result of CRT's pre-processing, model building, event building, vehicle dynamics solving, and post-processing capabilities, users can take advantage of this all-in-one simulation tool for objectively evaluating a vehicle or subsystem in real time. The vehicle models contain less DOF compared to models from other software, because of the parametric description of the vehicle models and K&C suspension behaviour. This feature translates to a lower computational power requirement for dynamics solver. When CRT is used in the simulator environment, this feature is advantageous as a smaller computation time delay is created. For this project, a recent model of the Jeep Grand Cherokee will be studied for the entire project. Another vehicle model, for the Jeep Renegade, will also be studied to test the robustness of the controller with a different vehicle class than the Grand Cherokee. Details on the models will be presented later in the thesis. The next section discusses how the vehicle models from CRT are brought into the simulator environment.

4.1.2 Transferring Models from CarRealTime to DriveSim

The vehicle model in CRT is the same model that is used in the simulator environment, however the communication between the vehicle model, external control systems, the simulator, and additional software is different than in the offline environment. VI-grade provides another software called DriveSim, which has multiple functions. DriveSim is a graphical user interface for running and configuring DIL simulations as well as a simulation manager for streaming data between software. Before a driving simulation can be executed on VI-grade's simulators, the vehicle model must be transferred from CRT to DriveSim. The following list contains the general steps for transferring a vehicle model. More specifics on how to complete these steps can be found in VI-grade official documentation. The generalized steps are as follows:

1. In CRT, an "Xternal VIDriveSim" event must be created. Once ran, this event will create the necessary input .xml file for the driving simulator. In the .xml file is a condensed version of the vehicle model readable by DriveSim. A configuration file is also created, which provides the name and location of the files in the vehicle model database for DriveSim, so that DriveSim can locate them.
2. The input .xml file, configuration file, and the vehicle model database must be placed in specifically registered folders on the computer which runs DriveSim, called the "concurrent" machine. This ensures that DriveSim knows where the vehicle model database and input .xml file are and can readily access them during simulations.

This process outlines how the models are transferred from the offline environment to the simulator environment. A major difference between the offline and simulator environments is the medium in which data is transferred between the software involved. In the simulator environment, real time databases (RTDBs) are used to transfer the vehicle signals between DriveSim, CRT, Simulink (if external control systems are used), and an additional software called SIMulation Workbench (SimWB) by Concurrent. SimWB is a modelling environment for developing the RTDBs and the setup of testing sessions on VI-grade's simulators. Further discussion on the creation, configuration, and communication with the RTDBs and SimWB is described in Section 4.2.1 where an in-depth discussion of DIL software for this research is presented.

4.1.3 Tire Model – MF-SWIFT Advantages

For the Jeep Grand Cherokee vehicle model, a different tire model than the MF is implemented for improving the fidelity of the offline and simulator evaluations. For the Jeep Renegade, the MF tire model is what has been historically used by Stellantis for studying the Renegade, thus it is also used in this project. The tire model used for the Grand Cherokee is called the MF Short Wavelength Intermediate Frequency Tire model (MF-SWIFT). This model is an extension of the MF tire model encompassing a larger frequency range of wheel vibrations [36], [97-99]. This model is generally accepted as an efficient, all-around good tire model for handling, ride comfort, durability, and testing the intervention of electronic control systems [98]. The model can be used for simulating a step steer, slalom, sine steer, j-turn, and tire behaviour on uneven road surfaces [98]. A rigid ring model is used to describe the belt dynamics of the tire. Both damping and spring properties are assigned to the tire wall or carcass, as well as residual stiffness and damping at the contact point [97], [98]. In this case, the model has 6 DOF for a tire, permitting a faster calculation time than higher DOF tire models while maintaining good accuracy [98]. The MF-SWIFT model also considers the tire tread width and turn slip. Turn slip is significant when a high amount of torque is applied around the vertical axis of the tire, such as during parking and aggressive turning at low speeds [36].

Both the MF and MF-SWIFT tire models use a single contact point with their slip models, however the MF-SWIFT model results in a higher fidelity of the contact patch dimensions. One major advantage of the MF-SWIFT model over the MF model is the 2D and 3D road enveloping capability. This envoloping process uses a grid of elipses or "cams" to envelope the input road surface during a simulation. The point of intersection between each cam and the road creates a point, where the collection of all the points of intersection between the cams and the road surface generate an effective road plane seen by the tire. This feature allows the MF-SWIFT model to consider the forward slope and camber of the road surface in which the tire is contacting during simulations [97-99]. Figure 21 presents a representation of the 2D enveloping method, which can be extended to consider the width (into the page) of the road surface in the 3D case.

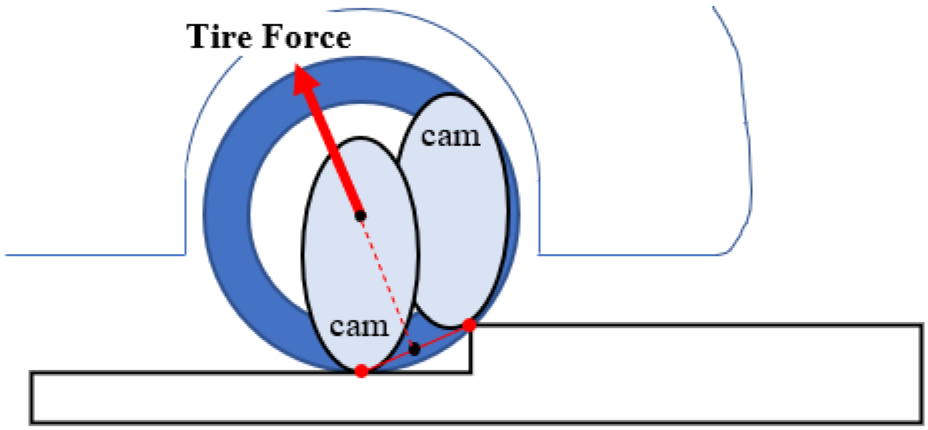


Figure 21: MF-SWIFT road surface enveloping

Finally, the MF-SWIFT model can capture the effects of suspension vibration in the 10-25 Hz frequency range, which affects the secondary ride of the vehicle [98]. In fact, the tire model can capture wheel vibration up to 100Hz, where the MF fails at this. Although this is considered NVH, it is still important during the subjective evaluations of the project to provide as realistic a feeling for the drivers as possible, compared to the driving the physical vehicle. The MF-SWIFT tire model has several use modes which alter the type of calculation being performed during simulations. For this project, the 114 mode is selected, which means a smooth road contact is considered, along with linear tire relaxation behaviour, and a combined force/moment calculation for x- and y- forces and x-, y-, and z-moments (in the wheel coordinate system). For details on other use modes, please refer to official documentation on the MF-SWIFT tire model. This tire model is used for the Grand Cherokee throughout the objective and subjective evaluations in the project.

4.2 VI-grade and Driver-in-the-Loop

In Section 1.3.3, the basics on the entities involved in DIL environments as well as several details on motion cueing and the interaction between the driver and simulator were explained. It was mentioned that one of the components of DIL environments for driving simulators is the collection of background software. This section of the thesis describes the roles and communication between the software involved in the subjective evaluations on the DiM 250 dynamic driving simulator. Moreover, a brief description of the advantages of the subjective evaluations with the DiM 250 over the offline objective evaluations is presented.

4.2.1 Software Description and Application

To complete the subjective evaluations on the DiM 250, it is necessary to configure the vehicle models and semi-active suspension controller with the RTDBs. For the co-simulations run between the Simulink semi-active suspension controller and the DriveSim/CRT vehicle models, a RTDB must be configured for each vehicle model. The RTDB must reference specific input and output channel names used for a given vehicle and controller model. For instance, the damper controller will require the relative velocity between the front right chassis and the front right wheel hub centre from the vehicle model, to calculate the front right damper force. Details on the controller architecture are discussed in Chapter 6. Once the configuration is done, a condensed version of the Simulink controller, readable by SimWB, is created and sent to the concurrent computer where DriveSim is running. DriveSim can then reference both the vehicle model database and the Simulink controller during a driving simulator evaluation.

VI-grade has created VI-GraphSim to configure and handle the graphical and audio information for the road, environment, and vehicle. Moreover, VI-MotionCueing is VI-grade's software for calculating the motion cues during driving simulations and for tuning their MCA. Motion cueing has already been tuned by engineers at Stellantis and VI-grade for the subjective evaluations in this project. During a driving simulation, SimWB, DriveSim, CRT, Simulink, MotionCueing, and GraphSim transfer data through the RTDB to replicate the driving experience of a real vehicle. Figure 22 contains a graphical representation of the data transfer routes and communication between the entities involved in a driving simulation.

Once a simulation is started in DriveSim and the driver in the DiM has begun driving, engineers in the simulator control room can interact with the Simulink controller in SimWB, through the RTDB. Since the Simulink controller was configured with the RTDB, each of the variables, parameters, and vehicle model signals being used by the controller can be displayed in real time in SimWB. Additionally, the parameters in the semi-active control strategy can be changed in SimWB. This feature allows the mode of the semi-active dampers to be switched in real time and permits tuning of the semi-active controller while the driver is driving the simulator. The users of SimWB in the simulator control room can display certain vehicle signals to see if the change to the controller settings has an effect on the vehicle's behaviour. This change can be done without the driver knowing or having to stop driving. Lastly, the interaction between the software and hardware blocks in Figure 22 is the visual and audio presentation to the driver and the computation of motion cues for the DiM 250 by VI-MotionCueing. As discussed in Section 1.3.3, the driver of the DiM 250 interacts with the simulator through the HMIs. The steering, throttling, and braking inputs come from the driver, rather than a driver model as in the offline co-simulations between CRT and Simulink. The next section lists the major advantages of DIL evaluations over offline evaluations.

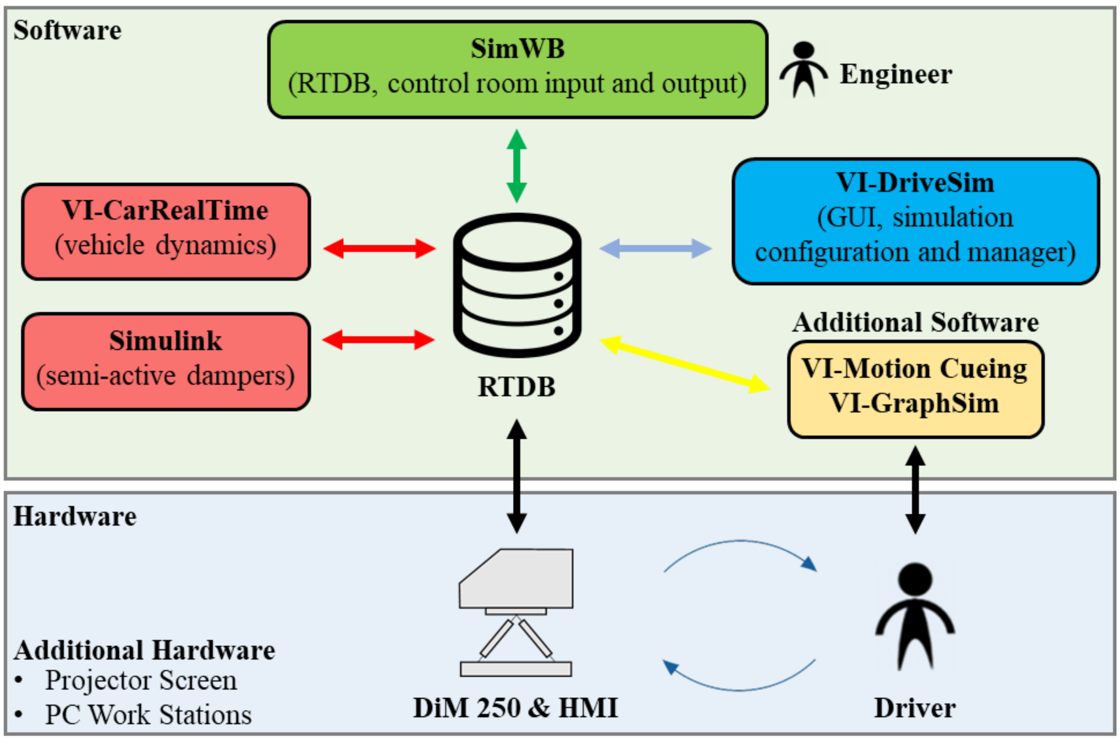


Figure 22: DIL software and hardware communication

4.2.2 Advantages of Simulator Evaluation over Offline Evaluation

Although both simulator and offline testing are used by OEMs to evaluate and tune their vehicle models, the simulator poses several advantages in terms of its capabilities. Please refer to Section 1.3 for the advantages of using simulators over physical testing on vehicles. The following list consists of advantages stemming from driving simulator evaluations over offline testing:

1. DIL testing can involve the recording of data from a vehicle model and the collection of ratings from driver questionnaires, permitting both objective and subjective evaluations

2. The DiM 250 can also use driver models to control the vehicle while a human remains in the driver seat, thus permitting the driver to understand what an offline simulation would feel like if a robot driver were used
3. Subjective evaluations with a simulator capture the psychological, physical, and emotional behaviour of human drivers, resulting in more realistic inputs to and perception of the vehicle model as opposed to a simplified driver model
4. Driving simulators permit performance engineers or expert drivers to use their skills and knowledge to freely evaluate a vehicle's ride and handling performance

This concludes the discussion behind the connection between software and hardware for the implementation of the DIL environment for the subjective evaluations on the DiM 250. In conclusion, the DIL setting is a complex collaboration of software and hardware communication through RTDBs and drivers to replicate a real driving experience. This collaboration allows dynamic driving simulators to provide accurate subjective evaluations of Stellantis' vehicles and their response characteristics. The final section of this chapter is focused on presenting details for the Stellantis vehicle models studied in this project.

4.3 Studied Vehicle Models

Vehicle models for Stellantis' Jeep Grand Cherokee and Jeep Renegade are focused on in this research. The methods for evaluating and connecting the vehicle models and the semi-active suspension controller are the same for both vehicle models. The Grand Cherokee is the primary vehicle model, whereas the Renegade is an additional model used to test the controller's robustness and capabilities with a different vehicle class. Both models are currently in production and offer several different trim levels. For this research, recent versions of the base trim vehicle models are being studied.

4.3.1 The Jeep Grand Cherokee and Jeep Renegade

An image of the vehicle's being studied is presented in Figure 23, courtesy of Stellantis media. Both vehicles are considered sport utility vehicles (SUVs), where the Grand Cherokee is a midsize SUV, and the Renegade is a subcompact SUV. The Grand Cherokee has a 30 cm longer wheelbase and a 10 cm longer trackwidth (front and rear) than the Renegade. The Grand Cherokee is more than 600 kg heavier than the Renegade. Both SUVs are front heavy, although the front-to-rear weight distribution is higher for the Renegade. The Grand Cherokee has all-wheel drive whereas the Renegade is front wheel drive. At the base trim level, the Grand Cherokee has 5 cm more ground clearance and the centre of gravity height 10 cm higher.

Figure 23: Jeep Grand Cherokee (left) and Jeep Renegade (right)

As for the suspensions of the two models, both use gas-filled passive dampers and front and rear anti-roll bars. The dampers curves for the models match the curves for the physical dampers on the real vehicles. For the Grand Cherokee, the rear spring rates and low velocity damping rates are around 2.7x and 1.5x higher than the front spring and dampers, respectively. For the Renegade, the front and rear spring rates are the same and the front dampers have approximately 1.5x higher damping rates in the low velocity region of the damping curves. In general, the Grand Cherokee has higher damping and spring rates all around. Finally, the Grand Cherokee has 18-inch rims whereas the Renegade uses 16-inch rims, and the vertical damping in the Renegade's tire models is 10x higher than the Grand Cherokee's. This brief discussion is intended to highlight a few major differences between the vehicle models being studied and to emphasize the fact that the same controller type is evaluated in two different vehicle classes.

4.3.2 Validation of the Vehicle Models for Offline and Simulator Environments

When completing objective or subjective evaluations of vehicle models, a validation between the models and physical vehicles must be completed. This validation is done to ensure the models are replicating the physical vehicle's behaviour. Stellantis has previously completed this validation of the Grand Cherokee and Renegade vehicle models with their passive suspensions, both objectively and subjectively. Objective data has been recorded during driving maneuvers with the physical vehicles and objective data from simulations with the vehicle models. Figure 24 is one example of some of the validation plots containing objective data from the CRT model (CarRealTime Model in green) and the physical vehicle (labelled Physical Testing in red). This data is from a frequency response maneuver at 0.3g. More validation data can be found in

Appendix B. Objectively, the CRT model is similar to the physical Grand Cherokee vehicle for the entire frequency range of the maneuver.

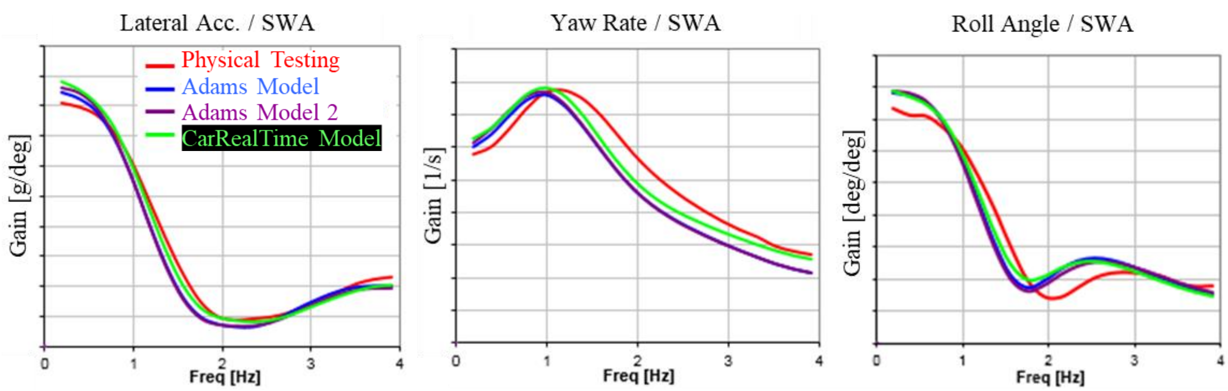


Figure 24: Example objective validation of Grand Cherokee Model

Regarding the validation of the vehicle models with the passive suspension driven on the simulator, this is completed subjectively by performance engineers from Stellantis proving grounds. Drivers visit the simulator lab at ARDC having already been familiar with the physical vehicles through extensive driving on test tracks. The drivers drive the models on the DiM 250 and work with the vehicle dynamics engineers at ARDC to tune the vehicle models and the motion cuing algorithm. This tuning is done until the performance engineers are satisfied with the model's replication of the physical vehicle's behaviour and ride and handling performance, among other aspects. Note that these validation procedures are for the vehicle models with the passive suspensions. When evaluating the different modes of the semi-active suspension controller, the ride and handling performance of each mode will be relative to the validated "default" mode of the suspension. For continuity, the default mode of the semi-active suspension controller refers to the passive suspension with which the Grand Cherokee or Renegade physical vehicles are currently equipped. The next chapter section discusses several performance differences that will affect the ride and handling of the Jeep vehicle models of focus.

4.3.3 Vehicle Configuration Effect on Ride and Handling

Properties of a vehicle such as centre of gravity height, weight and its distribution, spring rates, shock absorber damping rates, as well as wheelbase and track width impact the vehicle's ride and handling. OEMs tune their vehicle's ride and handling while considering these aspects as well as many other parameters. A higher centre of gravity increases the risk of rollover due to higher roll angles and accelerations as well as weight transfer when the vehicle is cornering. However, a higher ground clearance permits more SWS, allowing the dampers to absorb more energy transmitting from larger-amplitude road disturbances. The heavier a vehicle, the more inertia the vehicle has when turning and the more the suspension dampers and springs are compressed to their equilibrium positions under static loading. In reality, a heavier vehicle will require higher spring rates to compensate for this, which will affect the ride performance since the damper will have to dissipate more energy to control the springs. More vehicle weight also translates to more weight being transferred when accelerating, braking, or turning. If the dampers and springs are underrated for a given vehicle weight, aggressive turning, accelerating, or braking can result in tire lift, causing loss of grip and stability. Furthermore, the vehicle's weight distribution will affect its stability and steering behaviour under accelerating, braking, and turning. For instance, when a vehicle is entering a turn, the vehicle's moment of inertia pushes the vehicle opposite to the direction of the turn. The front tires generate a torque about the vehicle's vertical axis that causes it to turn while the rear wheels produce lateral forces that oppose this torque. A rearward weight distribution produces a larger distance between the lateral force on the front tires and the vehicle's centre of gravity, resulting in higher yawing or less understeer of a vehicle during entry. The opposite case results with a front heavy vehicle.

As discussed in Chapter 2, higher damping rates in a vehicle's suspension can reduce the pitch, roll, and vertical accelerations at the vehicle body's natural frequency, but degrade the accelerations at the wheels' natural frequencies. The vehicle's track width and wheelbase also affect the ride and handling. A wider track width reduces the risk of rollover in contrast to higher centre of gravities or roll centers. Wide tracks also reduce weight transfer and help the tires maintain grip on the road when turning. Longer wheelbases have similar effects under accelerating or braking. The wheelbase also contributes to the turning radius of a vehicle, where a longer wheelbase increases the turning radius requiring the vehicle to make wider turns. On the other hand, a longer wheelbase can improve the isolation between the front and rear disturbance dissipations. The longer wheelbase increases the time between the impact of a road disturbance on the front and rear tires, permitting more time for the front suspension to dissipate a disturbance before the rear's impact. As a result, a better balance between the front and rear ride can be determined. In the end, harmonious tuning of all the aforementioned vehicle parameters play a major role in the ride and handling performance of a vehicle. Keeping in mind the differences between the Grand Cherokee and Renegade models as described in Section 4.3.1, the final damping settings for the semi-active suspension controller modes will be different for each vehicle, but the overall architecture will be the same. The next chapter will address which characteristics of the dampers are most significant when tuning a vehicle for ride and handling. The results will lead to identifying the significant ride and handling aspects to be addressed later subjective evaluation method.

CHAPTER 5. Preliminary Damper Studies

The focus of this chapter is to establish the significant objective metrics and maneuvers for evaluating the virtual semi-active suspension with the Grand Cherokee and Renegade vehicle models. This study is done by analyzing how significant the effect of altering dampers is on the vehicle's ride and handling. Certain aspects of damper characteristic curves are altered to study the effect. The semi-active suspension controllers discussed in the literature review replace the damper force calculation of the vehicle models. Similarly, altering the damper curves changes the magnitude of the damper forces for a given damper velocity. As a result, this preliminary study on the effect of damper curve tuning provides insight on which ride and handling aspects on certain maneuvers are significantly affected by dampers, whether passive or semi-active. This approach will avoid redundant evaluations on maneuvers which do not excite the dampers enough to result in a change in ride or handling performance. Thus, not all the metrics and maneuvers mentioned in Chapter 3 are used in this project. At the end of the chapter, the maneuvers chosen for the evaluations are presented and discussed. These maneuvers will be implemented for the objective and subjective evaluation methods presented in Chapters 7 and 8.

5.1 Objective Evaluation of Damper Curves

The objective study was completed offline to study the effect of changing several different aspects of the damper curves of the Grand Cherokee vehicle model. The aspects that were altered where the low-velocity damping region, high-velocity damping region, rebound and compression damping rates, and the front to rear relative damping rates of the damper curves. The primary maneuvers simulated for this study included a step steer maneuver, cleat, frequency response, and a rough road profile known as the "body twist" section of one of Stellantis' proving grounds. These short and concise maneuvers cover most of the excitations discussed in the objective evaluation literature review in Chapter 3. Other maneuvers considered include steady state cornering and a sine-with-dwell, which are both steady state maneuvers. Specific details on the maneuvers are presented in Section 5.2. For each maneuver simulated in the offline environment, the focus was on the damper aspects resulting in noticeable ride and handling performance changes. For ride evaluation, the focus was on the vertical accelerations located near the driver seat of the vehicle. In this case, two virtual accelerometer sensors were added to the vehicle model. Adding the sensors can be done in CRT by setting the coordinates and reference frame for each sensor on the chassis subsystem. The virtual accelerometers measure the accelerations near the driver seat rail and the driver's head location and are used for addressing primary and secondary ride characteristics. For the handling evaluation, the focus was on the vehicle yaw response and delay, tire slip angles or grip, vehicle roll response and delay, and steering behaviour (under-, over-, or neutral steer).

The results of the study identify trends and guidelines for selecting maneuvers and metrics for ride and handling evaluation when only tuning dampers. These trends are used later in Section 6.3.3 to help tune the semi-active suspension controller modes for the Grand Cherokee and Renegade vehicle modes. The following section highlights the general findings of this initial study.

5.1.1 Effect of Damper Curve Tuning on Ride and Handling Metrics

The first step of the study is to define the changes made to the Grand Cherokee damper curves. Certain practical limitations of altering the damping curves were considered. For instance, the damping rates could not be reduced too much to avoid deterioration of the vehicle's ability to dissipate disturbances and primary ride control. Furthermore, the damping rates could not be increased too much as this would deteriorate secondary ride and generate an extremely stiff suspension. In the case of severely over-damped suspensions, the dampers act like rigid connections between the wheels and chassis. This setting would result in vehicle yaw instability due to poor tire grip as well as severely poor ride and handling. Keeping these points in mind, two recent projects at Volvo studied the effects of dampers on ride and handling [49], [100]. The first study involved applying changes the rebound or compression damping rates by only ±30%. The results presented did not focus on objective metrics, but instead the authors analyzed only the dynamic signals [49]. The current research focuses on larger changes to different damping regions and specific objective metrics. In [100], the results depicted the necessity to study non-linear asymmetric dampers as opposed to the linear curves (or a constant damping coefficient) commonly used with quarter-car models. At best, a half car model was used for a roll analysis, which assumed the front and rear damping was the same. Moreover, the authors concluded that one of the main drivers for ride and handling performance is the low-velocity damping region of the curves. This aspect will be the first focus of the preliminary damper curve study herein. Also, higher fidelity vehicle models are analyzed to capture a more comprehensive ride and handling performance.

The first damper curve alterations consisted of changing the low-velocity damping rates, or slopes, of the damper curves in the Grand Cherokee's low-velocity damping region. Engineers at Stellantis aided in the choice of the percentage changes from the default damper's low-velocity damping rates. Figure 25 contains the changes made to the low-velocity region of the dampers. The vertical scale representing the damper forces has been removed for confidentiality of the OEM's damper properties; however, the scale is the same for the front and rear damper curves.

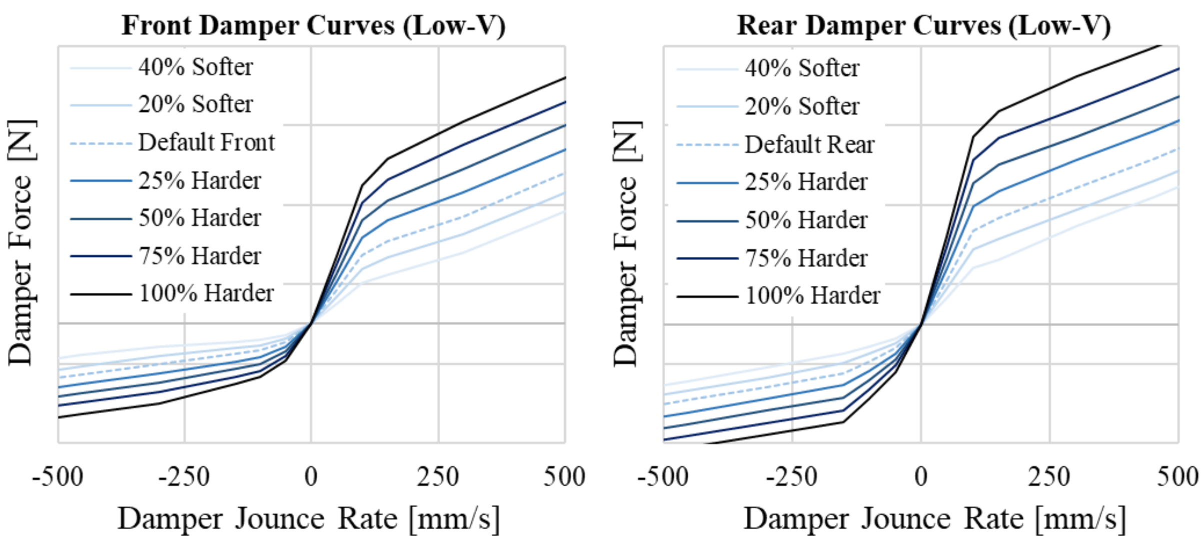


Figure 25: Low-velocity damper curve alterations

In Figure 25, the term "harder" corresponds to higher damping rates whereas the term "softer" corresponding to lower damping rates. The percentages represent the percentage change in slope from the default damper curves. Note that the figure is a concentrated image of the low-velocity region of the damper curves as well as the beginning of the high-velocity region. Moreover, the slopes of the high-velocity region are the same for each damper configuration, but the magnitudes of the forces have been slightly altered as a consequence of changing the low-velocity region slopes. The second set of alterations corresponded to percentage changes in the slopes of the high-velocity regions of the damper curves. Figure 26 illustrates these changes created for the study. In reality, the damper curve velocity regions extend further to approximately 3000 mm/s for the Grand Cherokee. However, at damper velocities above 1000 mm/s, the impact on vehicle ride and handling is nearly zero. High-velocity damping rates become more important for NVH and durability simulations, which are not a focus of the current research.

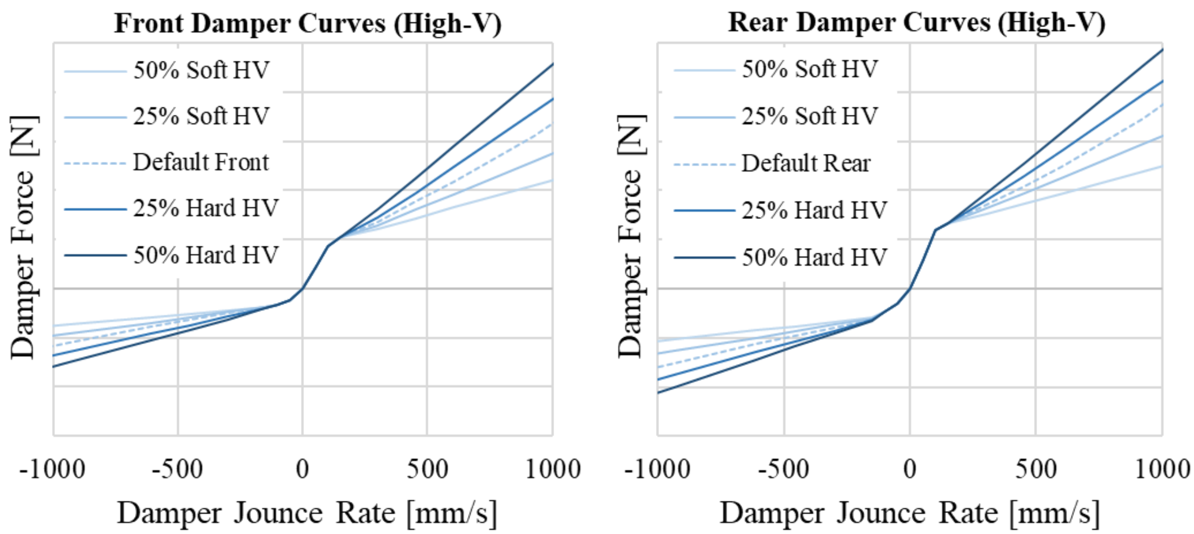


Figure 26: High-velocity damper curve alterations

In Figure 26, the abbreviation "HV" stands for high velocity. As with the low-velocity changes, the legends in Figure 26 describe the percentage changes of the slopes for the high-velocity regions of the damper curves. The third set of alterations consisted of increasing and decreasing the rebound and compression rates, one at a time. The damping rates for the entire damper curves were amplified by the same amount to keep the ratio of high-velocity to low-velocity damping rates the same. Figure 27 presents these alterations.

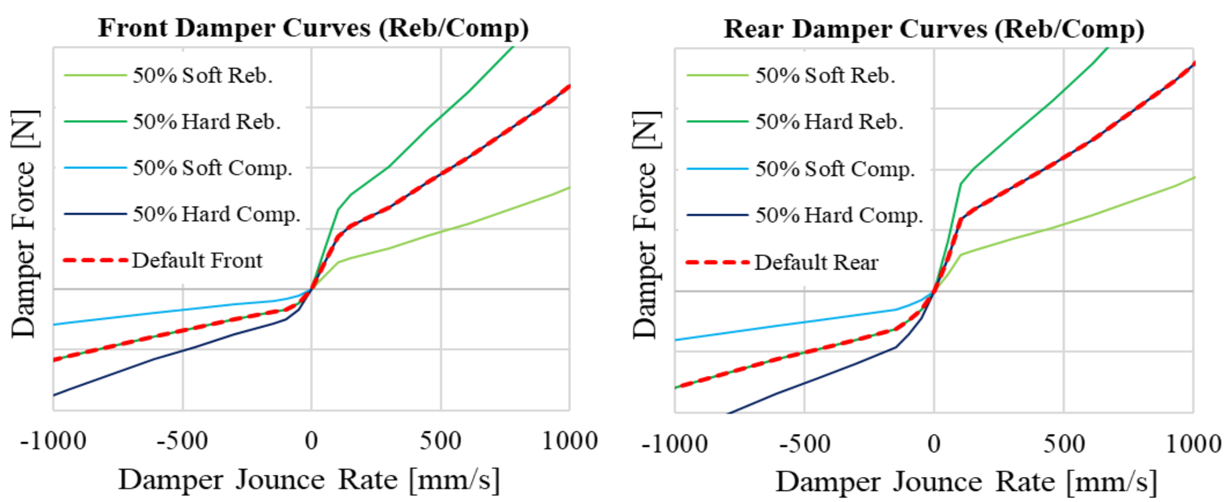


Figure 27: Rebound and compression damper curve alterations

In Figure 27, the ratio of front-to-rear damping rates are always the same as the default setting since the changes made affected the front and rear damper curves in the same manner. The final alterations made to the damper curves consisted of increasing and reducing the entire front damping rates by 50% to study the impact of changing the front-to-rear damping ratio. As a result, the rear damper curves were the same for both configurations. Damper curves can be viewed in Appendix C. The goal was to affect the yaw response of the vehicle by changing the front tire grip. The front damping rates influence the camber angle changes of the front wheels. Thus, the size of the contact patch can change as the front damping rates are altered. As the Grand Cherokee's front wheels have a slightly negative camber angle on installation, the goal was to control the front camber angle during the step steer maneuver to permit a zero-camber angle on the outside (loaded) tires while turning. This behaviour would generate the largest contact patch and highest cornering force for better grip. The next step of the study was to simulate each of the aforementioned damper curve changes and note the significant changes in vehicle ride and handling.

The first major conclusion of the study was that dampers to do not affect the steady state response of the vehicle. This result was previously stated in Section 2.3, but the decision was made to reaffirm this fact at the beginning of this study. As a result, the vehicle's roll, yaw, and steering behaviour on steady state cornering and sine-with-dwell were not affected. See Appendix C for some of the results from a constant radius, steady state cornering maneuver.

The low-velocity damping rates had the most significant impact on ride and handling. For the body twist maneuver, the harder damping settings resulted in 15% improvements in primary vertical ride and SWS but worsened the head toss or lateral acceleration of the driver's head by nearly 15%. On the cleat maneuver, the vertical secondary ride of the hardest damping setting was worsened by approximately 45% compared to the default Grand Cherokee. Moreover, the settling time of the cleat disturbance was significantly improved with the harder damper settings by as much as 45% compared to the default setting. Figures 28 and 29 contain the major findings from the body twist and cleat maneuvers, respectively.

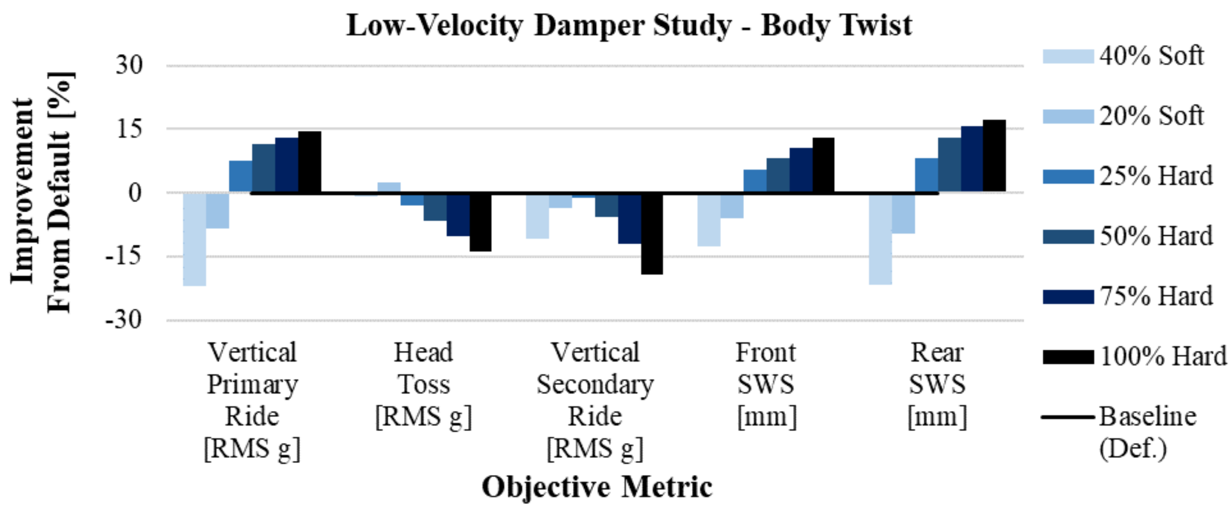


Figure 28: Low velocity objective ride on body twist

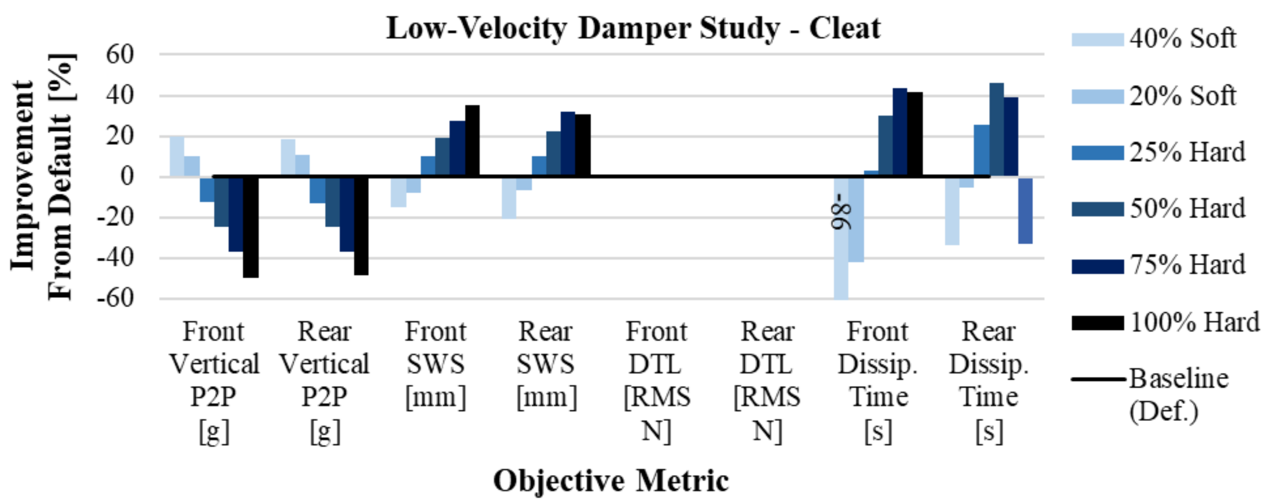


Figure 29: Low velocity objective ride on cleat

In Figure 29, it appears that the DTL objective metric is not capturing the difference in disturbance dissipation between the different damping settings. Instead, the dissipation or settling time metric does capture the difference. For the rear tire, the hardest setting seems overdamped, see Appendix C. As a result, another way to measure the DTL was discovered regarding the standard deviation rather than the RMS. This concept will be discussed in Chapter 7. As discussed in the theory section 2.2.1, the DTL represents the RMS of the vertical tire loading or tire deflection. When this metric was computed for each setting on the cleat maneuver, the results showed no change in DTL for each setting. However, when plotting the vertical tire force over time, it is clear there is a significant difference in the response of the tire to the cleat disturbance. Figure 30 presents the vertical tire loading for the front right tire of the Grand Cherokee for each low-velocity damping setting. Clearly, the DTL is different between each damping setting.

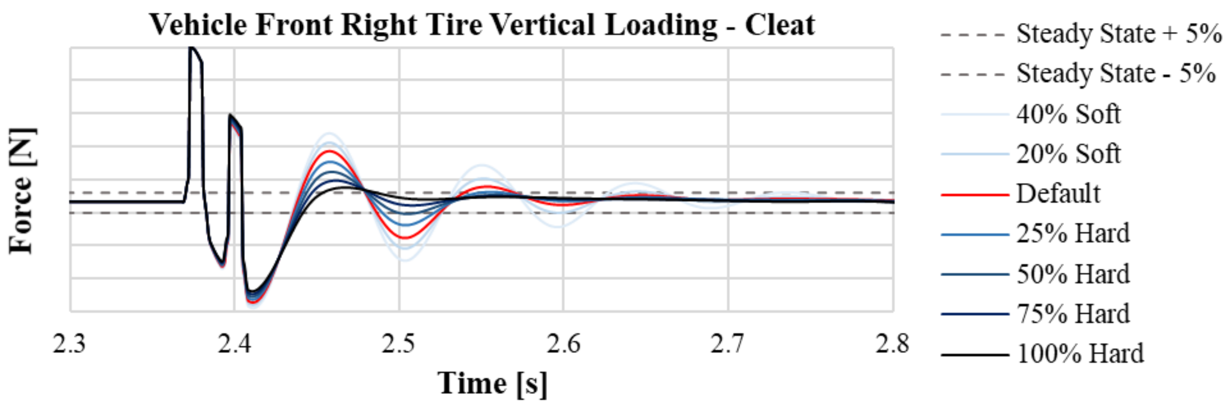


Figure 30: Low velocity DTL on cleat

Regarding the handling performance of the low-velocity damper curve alterations, the results are less significant on the frequency response and step steer maneuvers. Both maneuvers were simulated for producing a maximum lateral acceleration of 0.5g. The most significant objective handling results for each maneuver are presented in Figures 31 and 32. The step steer maneuver did not show significant differences in the transient characteristics of the vehicle response for any of the damping settings.

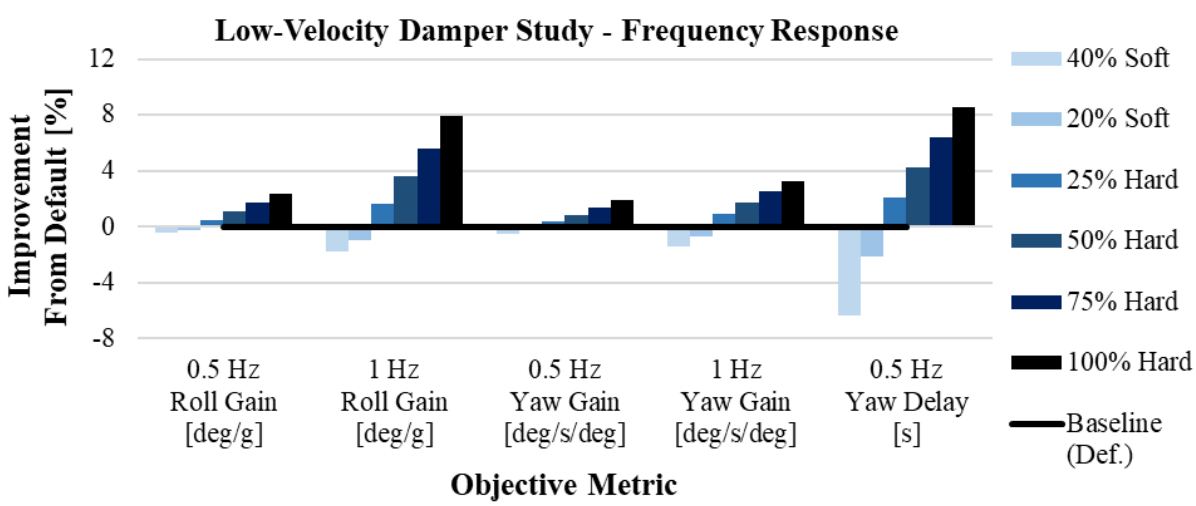


Figure 31: Low velocity objective handling on frequency response

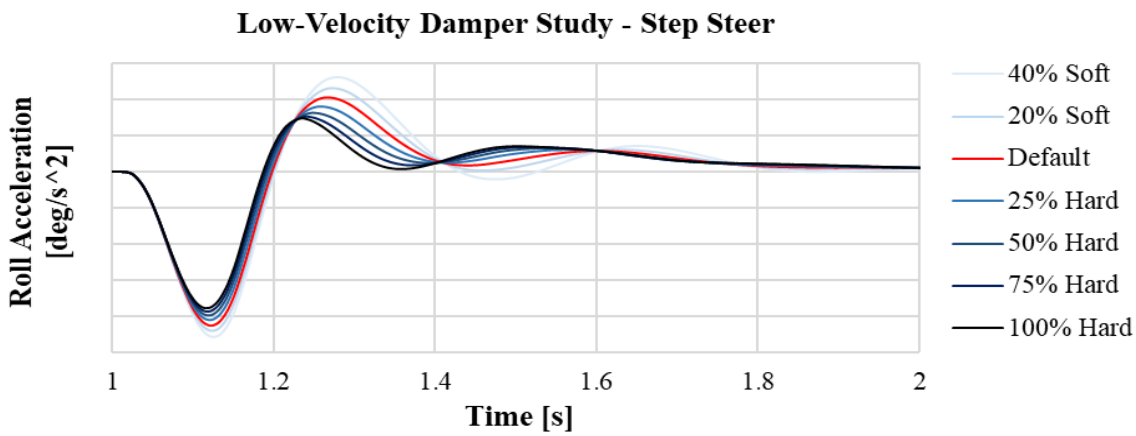


Figure 32: Low velocity P2P roll acceleration on step steer

For the frequency response maneuver, the increased low-velocity damping permitted a reduction in the roll gain and an increase in the yaw gain at low frequencies. This damping results in reduced body roll and increased heading rate in response to the driver input steering. At the same time, a slight improvement in the yaw delay (time between SWA and yaw rate response) at 0.5 Hz was found, permitting a quicker yaw response of the vehicle. One metric not shown in Figure 31 is the roll delay metric (time between roll angle and lateral acceleration) which resulted in large percent changes for the hardest damping settings. For instance, the 75% and 100% Hard damping settings resulted in increases in the roll delay at 0.5 and 1 Hz frequencies by more than 100%.

Finally, the step steer maneuver provided insignificant differences in vehicle ride and handling for the low-velocity damping alterations. The only noticeable difference in handling performance was the P2P roll acceleration shown previously in Figure 32. In the case of the hardest damping setting, the P2P roll acceleration was reduced by approximately 17%. The transient response characteristics outlined in Section 2.2.1 showed no difference when comparing the low-velocity damping settings. Furthermore, the tire slip angles of the outward loaded tires did not show differences. In general, when tuning the low-velocity region of the damper curves, the objective differences in handling are less significant than the objective differences in ride performance.

For the remaining damper curves alterations, less significant objective differences in ride and handling performance metrics were discovered. First, for the high-velocity curve alterations, no significant differences in handling were found on the step steer and frequency response maneuvers. As for objective ride, the maximum ride improvement on body twist was a 3% reduction of the RMS vertical acceleration at the driver seat ("Vertical Primary Ride") with the 50% Hard HV setting. On the cleat, the maximum difference computed was a 9% worsening of the front and rear P2P vertical accelerations at the driver seat. See Appendix C for the complete set of objective ride results. Second, the rebound and compression damping alterations also resulted in no noticeable changes in objective handling, other than the P2P roll acceleration metric on the step steer event. The results found that harder rebound or compression damping reduced the P2P roll acceleration (See Appendix C). As for objective ride, the harder compression and rebound settings resulted in the same trends as with the harder damping settings from the low-velocity damping alterations. The increased damping resulted in improved primary ride and worsened secondary ride. However, the percent differences were less significant than found with the low-velocity alterations. See Appendix C for the objective ride results of the rebound and compression alterations. Third and last, the alterations for increasing and reducing the entire front damper curves again resulted in no significant differences in objective handling. With the body twist and cleat maneuvers, the harder front settings improved vertical primary ride, worsened head toss and secondary ride, and improved dissipation times for the front suspension. The opposite was found with the softer front settings.

In general, it was discovered that the low-velocity region of the damper curves plays the most significant role in vehicle ride and handling among the alterations made. The vehicle's primary and secondary ride and the tradeoff between them was identified and found to be significant in response to the low-velocity alterations made. The vehicle's objective handling with regards to the yaw and roll responses was also found to be impacted by the low-velocity damper curve region. However, the handling was found to be less affected than the ride as shown in the results. As for steady state maneuvering, dampers do not affect the vehicle's response characteristics. As a result, tuning dampers and the semi-active suspension controller in this research should focus on the low-velocity damping settings of the controller modes during transient maneuvers, whether damper curves or a control strategy (or both) is used. Finally, an additional result which presents the maximum excited damper velocities found in each simulated maneuver and additional ones has been placed in Appendix C. It is important to note that for the maneuvers studied, the excited damper velocities primarily reside in the low-velocity region of the

Grand Cherokee's damper curves. The next subsection briefly outlines which maneuvers will be considered for the remainder of the project.

5.1.2 Significant Maneuvers for Evaluating Damper Changes

As presented in the results in the previous section, the body twist and cleat maneuvers capture the vehicle's primary and secondary ride performance. The frequency response maneuver is able to capture some handling differences in the vehicle's frequency response to driver inputs. The step steer maneuver has not resulted in highlighting differences in vehicle handling. As a result, the step steer maneuver will not be studied in this project. Moreover, the steady state maneuvers will not be studied since they do not excite vehicle dampers. To consider a wider set of vehicle dynamics including closed loop maneuvering and the vehicle's pitch response, several additional maneuvers were chosen in coordination with Stellantis' and other OEMs' testing standards. An ISO DLC, a slalom, a straight braking maneuver, and a complete proving grounds track are also considered in the objective and subjective evaluations of the virtual semi-active suspension controller. The addition of the closed loop maneuvers allows the drivers of the DiM 250 dynamic driving simulator to use their knowledge and expertise with driving and vehicle dynamics, rather than forcing them to apply specific driving inputs as with the frequency response maneuver. Moreover, the ISO DLC and slalom events excite the vehicle dampers more than the step steer maneuver as a result of more aggressive steering inputs (See Appendix C). These two maneuvers are considered press maneuvers in CRT which test the vehicle's limits to solve for the fastest execution time (See Section 4.1.1). Finally, the track event contains a combination of accelerating, braking, and cornering events with road disturbances to also test the limits of the vehicle models while exciting higher damper velocities than the frequency response and step steer maneuvers. The addition of these maneuvers is also to provide continuity with the discussion of capturing as much of the vehicle's ride and handling aspects as presented in Section 3.3.2. This research provides a more comprehensive study of ride and handling on vehicle semi-active suspension, both in the offline and simulator environments. The next and final section of Chapter 5 presents more details on the maneuvers considered for the remainder of the project.

5.2 Description of Maneuvers for Objective and Subjective Evaluations

As stated previously, the maneuvers considered in this research will be simulated in the offline environment for objective evaluation and driven through with the DiM 250 dynamic simulator for subjective evaluation in the simulator environment. Each maneuver is listed and briefly described in the following chapter sections. Note that certain details are not displayed as they are part of Stellantis' internal standards. For specific details on the objective metrics for each maneuver, please see Chapter 7. For subjective metrics, please see Chapter 8.

5.2.1 Maneuvers for Ride Evaluation

The following is a list of the maneuvers primarily used for objective and subjective ride evaluation.

1. Body Twist: A straight segment of Stellantis' Chelsea Proving Grounds (CPG) in Michigan, USA. The road section contains a series of asymmetric road disturbances to excite the primary ride of the vehicle. There are also disturbances which can excite the secondary ride of the vehicle, although the maneuver is mainly for primary ride evaluation. The positioning of the disturbances cause the chassis of the vehicle to twist due to asymmetric tire loading in the front and rear, hence the "body twist" name. The vehicle is driven in a straight line at a constant speed specified in Stellantis' laboratory procedures.
2. Cleat: Same maneuver as described in Section 3.1.3. This obstacle is also part of Stellantis' CPG; the road section at CPG contains several cleats with varying size. In this case, one of the cleats has been chosen for offline objective evaluation. For the subjective evaluations, all of the cleats on the CPG are driven over to consider more severe secondary ride events. This approach also permits the drivers of the simulator to test the dissipation of the vehicles in between each cleat.

5.2.2 Maneuvers for Handling Evaluation

The following is a list of the maneuvers primarily used for objective and subjective handling evaluation.

1. Frequency Response: Same maneuver as described in Section 3.1.3. The frequency of the driver's steering input considers practical limits for human drivers, as real drivers are used for the subjective evaluations. Steering frequencies at and below 2 Hz are considered.
2. Straight Braking: The vehicle is driven in a straight line at a constant speed specified by Stellantis' laboratory procedures. After several seconds, the brakes are applied as fast as possible to study the pitch response of the vehicle. Under certain vehicle conditions, rear

wheel lift can occur and thus it is also studied in this maneuver to observe the vehicle's stability.

3. ISO DLC: Same maneuver as described in Section 3.1.3. Figure 33 presents the dimensions of the cones as specified by ISO 3888-1 [56]. The black and red dots represent cones.

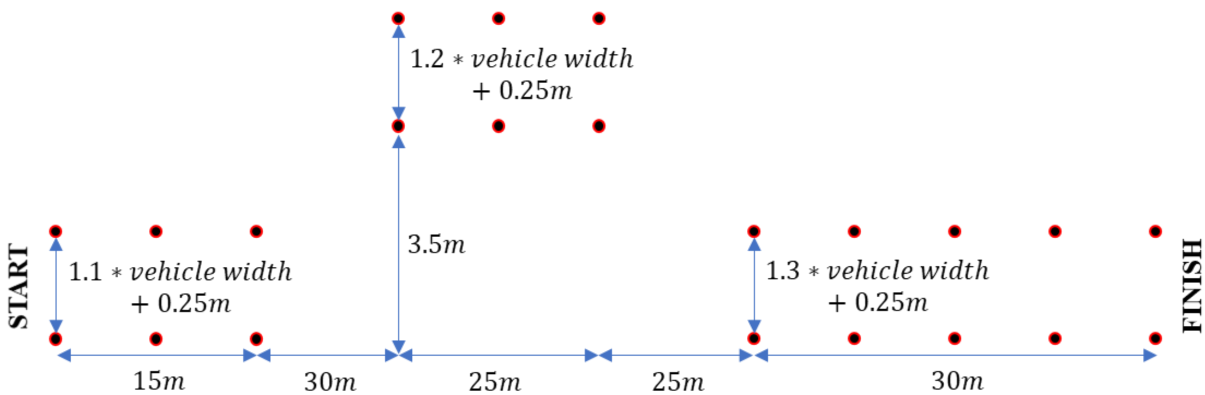


Figure 33: ISO double lane change specifications

4. Slalom: Same maneuver as described in Section 3.1.3. In this case seven cones with 30.5m of space between each cone is considered. This setting considers the smallest number of cones with an intermediate spacing as one of three default options in CRT. The driver is meant to apply consistent and smooth SWAs while driving through the cones the fastest possible speed. Each successful execution without contacting a cone permits the driver to increase the vehicle speed on the next trial run. This procedure is done until the maneuver cannot be completed at a faster speed.

5. Max Performance Track Event: This event involves driving a vehicle around one or more laps of a complete test track or proving grounds. For this project, the road data file for the Calabogie track in Ontario, Canada has been supplied by VI-grade. Another track provided by VI-grade, called the Grattan Raceway, can only be driven on the simulator, as Stellantis does not have the road data file for offline simulations. The Grattan track involves more aggressive maneuvering; thus, it will be used for the subjective evaluations. The Calabogie track will be simulated in the offline environment. In CRT, this type of event is known as a maximum performance event, where the CRT solver iteratively solves for the fastest lap time or velocity profile while simultaneously considering only feasible velocity profiles at discrete intervals of the track.

This concludes the discussion of the preliminary damper study and the maneuvers used throughout the rest of the project. The next chapter presents all details regarding the virtual semi-active suspension controller.

CHAPTER 6. Semi-Active Suspension Controller

This chapter outlines the development of the semi-active suspension controller starting from its architecture of suspension modes and ending with fine-tuning on the DiM 250 dynamic simulator. The architecture is split into high and low levels pertaining to the selection of suspension modes and the working principles of the semi-active suspension control strategies, respectively. Each of the control strategies selected in the literature review are modelled and explained herein. Furthermore, the parameters, variables, and input-output relationships are described for each control strategy being considered. Next, the criteria for selecting which semi-active suspension control strategies are best suited for the suspension modes are defined. At the end of the chapter, a discussion of how the controller is connected to the simulator is made, followed by fine-tuning of the suspension modes for improving the controllers ride and handling performance. The outcome of the chapter is the definition of the vehicle semi-active suspension controller designed for improved ride and handling.

6.1 Overall Controller Architecture

MathWorks's Simulink software is used to model the semi-active suspension controller. The highest level of the controller architecture contains the inputs and outputs between CRT and Simulink as well as several controller parameters. In general, the controller receives the velocity of each vehicle damper from CarRealTime as input and computes the damper force for each damper. The controller output is the damper force along with an activity flag which notifies CRT that the damper forces are being calculated in the Simulink controller instead. The activity flag is a Boolean value which is defined as a constant in the Simulink controller model. Depending on the control strategy implemented for modelling a semi-active suspension, additional control parameters and vehicle dynamic signals are required as input to the Simulink controller. Such inputs are described alongside their respective control strategy later in this chapter. Figure 34 presents the high-level architecture of the controller which contains the common inputs and outputs for all control strategies modelled in this project. Note that this architecture is for the controller used in offline co-simulations between CRT and Simulink. For co-simulations in the simulator environment, the architecture is slightly different.

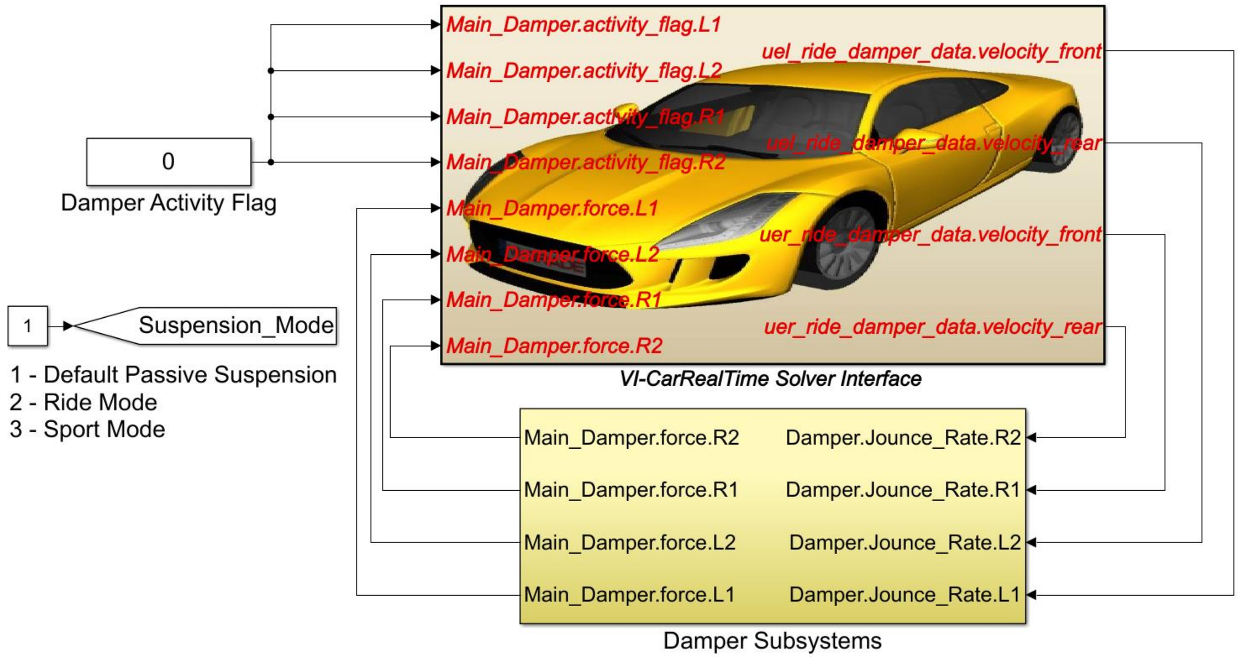


Figure 34: Semi-active controller high-level architecture

In Figure 34, the block labelled "Damper Activity Flag" is the Boolean value which is sent to CRT to disable the dampers in the vehicle model. To the right of the activity flag is the "VI-CarRealTime Solver Interface" block which is specific to a block library provided by VI-grade. This block is where the user can define the input and output channels to be used by the controller. The block is required to pass data between Simulink and CRT. In the case of a press maneuver or max performance event being simulated, additional interface blocks from this library are also needed to ensure VI-grade's solver algorithms are implemented. Below the activity flag is a block containing a constant which selects the suspension mode for the simulation. For an offline simulation, this "Suspension_Mode" parameter cannot be changed once the simulation starts, instead it is preset at the beginning of each offline co-simulation. The block titled "Damper Subsystems" in the lower part of Figure 34 represents a subsystem block containing four separate subsystems inside it - one for each vehicle damper. The next level of the controller architecture is a set of the four aforementioned damper subsystems, each having its respective damper velocity from CRT as input and its damper force as output. The dampers are named using a "L" or "R" for left and right, respectively. Additionally, a "1" denotes a front damper whereas a "2" denotes a rear damper. Therefore a "R1" is used to name the front right vehicle damper.

Once a co-simulation is run, the damper velocities are passed from CRT to a damper subsystem where the damper force is calculated using a semi-active control strategy or damper curve. The damper force is then passed back to CRT along with the damper activity flag. This process occurs in real time at every time step of the simulation. In addition to evaluating the controller's impact on ride and handling, engineers can evaluate the possibility of time delays in the damper force calculation if a certain controller strategy requires high computational demand. At the end of a simulation, Simulink displays a simulation efficiency parameter which represents the ratio of real-world time to simulation time required for a given simulation. This value was never below one for any of the control strategies studied. The next subsection discusses the suspension modes of the controller.

6.1.1 Controller Suspension Modes

The lowest level of the controller architecture contains a switch which determines which control strategy or damper curve is used for the damper force calculation. As concluded in Section 3.3.3, three suspension modes are implemented in the semi-active suspension controller. The modes are named default, sport, and ride. The names were chosen based on the suspension technologies in production vehicles, discussed in Section 3.3.4. The default mode contains the damper curves already validated in the vehicle model being studied. This mode is used as a baseline or reference for the other two modes. The sport mode is targeted at improving handling of the vehicle, where the goal is to provide better road holding capability, a quicker vehicle response, suppressed roll and pitch motion, and provide a sportier feeling for the driver. The ride mode is targeted for improving the primary and secondary ride of the vehicle. The name "ride" was chosen instead of the common term "comfort" as found in several production vehicles, since this research is aimed at evaluating an improvement in ride rather than comfort. Comfort is more related to the exposure time of and human sensitivity to accelerations with the driver, which is more of a concern in NVH. Furthermore, an "auto" mode is not modelled since it is expected that the driver uses their knowledge and familiarity with their vehicle to choose between improved handling or ride. For instance, the driver could choose the sport mode while driving on well-kept highways or urban roads where road disturbances are less of a concern to the driver. On the other hand, the ride mode could be chosen for poorly maintained roads with many road disturbances. Finally, it is important to keep in mind that semi-active suspensions still result in a tradeoff in ride and handling [82]. This aspect must also be considered when tuning the controller modes to avoid undesirable tradeoffs.

6.2 Modelling Semi-Active Suspension Control Strategies

Based on the comparison presented in Section 3.3.3 which discussed the major performance improvements of several semi-active suspension control strategies, three of the strategies were chosen to be studied. The strategies are the Skyhook, Groundhook, and Hybrid control strategies. In addition to these, a modified Switchable strategy is considered where different damping curves are chosen manually by a driver instead of automatically based on a switching border and criterion. In essence, the driver can choose between different damper curves, similar to the DampTronic-Select semi-active damper from Bilstein where the driver can select between a soft or hard by pressing a button [25]. This concept is called the Switchable Damper Curves strategy for the remainder of the thesis.

Common to all strategies modelled, the damper velocity channel for each of the four vehicle model dampers are required. Recall that the damper velocity is the relative velocity between the sprung and unsprung masses at each corner of the vehicle. Also, a damper curve is required for each strategy. Recall from Equations (31), (33), and (34) from Section 2.3.3 which represent the damper force for the Skyhook, Groundhook, and Hybrid strategies, respectively. In each equation, the damper force is a damping coefficient multiplied by a velocity term, for a quarter-car model. With the two track vehicle models with parametric suspensions in CRT, damper curves are used rather than single damping coefficient. Therefore, for each of the control strategies modelled using Simulink, the damping coefficient (C_{sky}, $C_{ground}, or\ G$) from Equations (31), (33), and (34) are replaced by a lookup table multiplied by 2.2. The lookup tables in Simulink represent the damper curves from the CRT vehicle models. The following subsections present and describe the Simulink models for each of the semi-active suspension control strategies studied in this project. Note that the final semi-active controller can contain any one of the strategies for a given suspension mode.

6.2.1 Switchable Damper Curves

The Switchable Damper Curves strategy is the simplest of the strategies considered as it is only a collection of different damper curves that can by chosen by the driver in this case, rather than implementing a switching boundary. For this project, the strategy is aimed at providing different low-velocity damping rates to improve ride or handling, based on the performance trends identified in Section 5.1.1. Figure 35 presents a generic Simulink diagram for modelling such a control strategy.

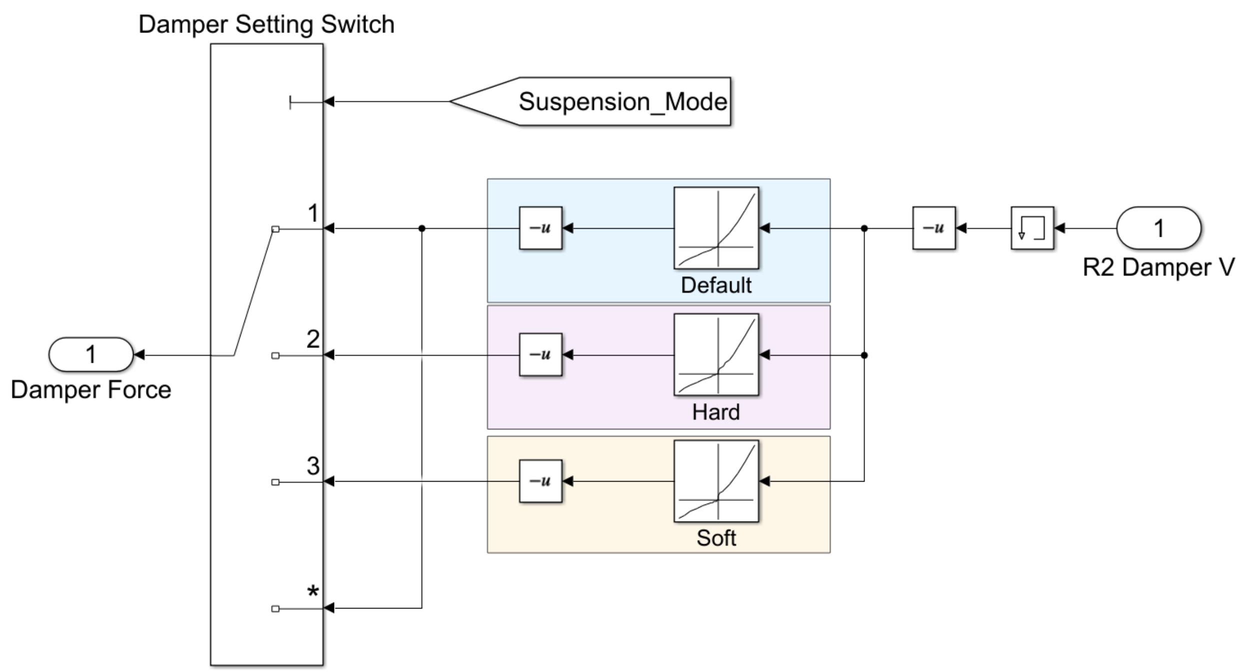


Figure 35: Switchable damper curves control Simulink model

The Simulink diagram in Figure 35 also represents the lowest level controller architecture where there is a switch dictating the working suspension mode. This diagram would be found inside each damper subsystem block. Three coloured blocks are presented in the figure to represent three different damping curve settings. The "Suspension_Mode" parameter is the same as in Figure 34, which represents the chosen mode by the driver. In any mode, a damper velocity is mapped to a damper force using one of the lookup tables. The lookup tables are labelled as "Default", Hard", and "Soft" only as an example in Figure 35. Note the unary minus blocks found before and after each lookup table in Figure 35. These blocks are necessary to transform the damper velocity sign convention from the CRT damper curve files to the damper velocity sign convention in the CRT output channels. This convention is inherently specific to VI-grade's software. Finally, the memory block located to the left of the "R2 Damper V" (the damper velocity for the rear right damper) is used to hold and delay the input damper velocity by one timestep. This block was recommended by VI-grade to ensure the solver does not skip an iteration step during the simulations. When simulating this control strategy, the main objective is to implement a variety of lookup tables representing soft and hard damper curve settings (as in Section 5.1.1) to determine the best improvement in ride and handling for the ride and sport modes while avoiding a significant tradeoff in the other.

6.2.2 The Skyhook Strategy

The Skyhook strategy contains an additional input compared to the Switchable Damper Curves control scheme. The vertical velocity of the sprung mass (chassis) directly above each of the respective dampers is required for this strategy. To acquire these velocities, four virtual sensors are added to the body subsystem of the CRT vehicle model to measure the velocity with respect to the road surface. Furthermore, these signals must be added as CRT outputs in the "VI-CarRealTime Solver Interface" block from Figure 34. Figure 36 presents the Simulink diagram for implementing the Skyhook strategy.

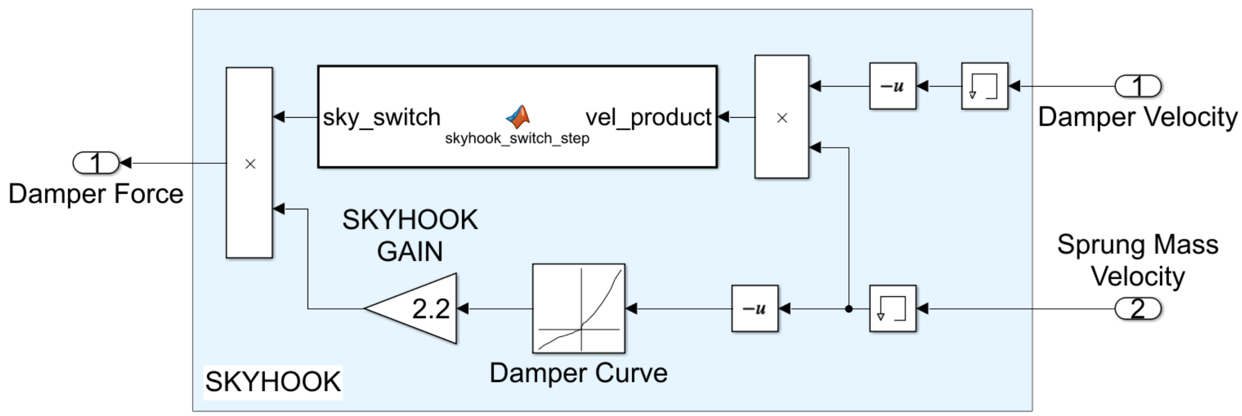


Figure 36: Skyhook control Simulink model

The Simulink diagram in Figure 36 is a graphical representation of Equation (31). Starting from the right of the diagram, the damper and sprung mass velocities are the input to the strategy. The product of these two velocities is then computed and sent to a function block. The Matlab function block in Figure 36 implements the if-then statement also found in Equation (31). This function block is required to determine if the maximum or minimum damping is to be applied to the sprung mass. At the end on the left of the diagram, the output is the Skyhook damper force. Only one unary minus block before the damper curve lookup table is required for the Skyhook strategy since the damper must apply a force in the opposite direction of the sprung mass velocity. For instance, if the chassis is moving upwards while the wheel hub is fixed, the damper must apply a force that acts downward on the chassis. Otherwise, the damper would be adding energy to the system, which is not possible for a semi-active suspension. Note that the CRT solver will run into errors if these unary minus blocks are not present to account for this. A similar case is found with the Groundhook strategy in the next section. As stated in Section 2.3.3, the goal of the Skyhook strategy is to add damping to the vehicle sprung mass to improve the velocity and acceleration response of the sprung mass. At the same time, the unsprung mass is neglected with this control strategy which can worsen handling. This behaviour is addressed in Section 6.3.1 when the performance of the controller strategies are compared.

6.2.3 The Groundhook Strategy

The Groundhook control strategy also requires an additional input compared to the Switchable Damper Curves strategy. Here, the vertical velocity of each unsprung mass is required. Virtual velocity sensors must be added to the CRT body subsystem to record the wheel centre vertical velocities in the front or rear suspension subsystems, relative to the road surface. As found with the Skyhook strategy, these additional vehicle channels must be added to the "VI-CarRealTime Solver Interface" block from Figure 34. Figure 37 presents the Simulink diagram for implementing the Groundhook control strategy.

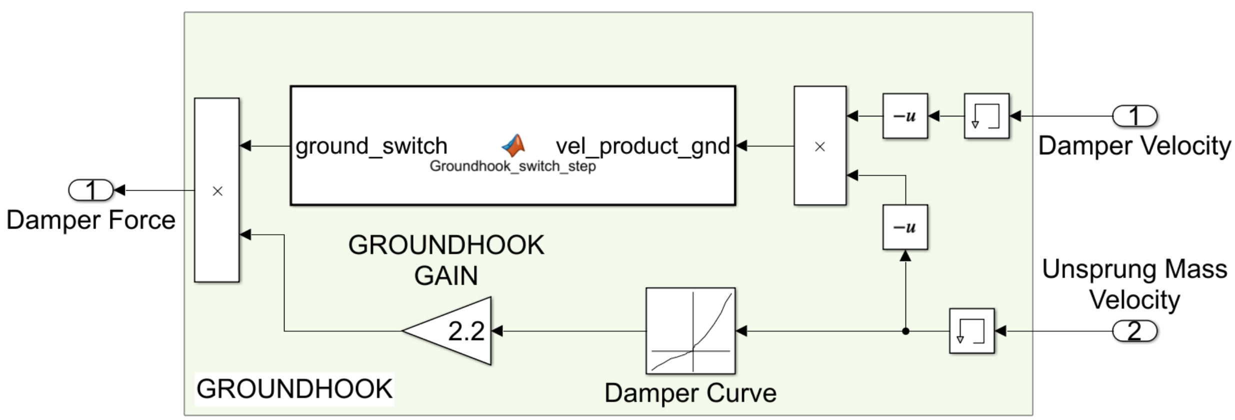


Figure 37: Groundhook control Simulink model

The Simulink diagram in Figure 37 follows the same procedures as outlined in the previous section. The difference with the Groundhook strategy is that the damping is added to the unsprung mass instead of the sprung mass. Also, the unsprung mass velocity is mapped to a damper force using the damper curves. In the Groundhook strategy diagram, a unary minus block is not required directly before the damper curve lookup table since the unsprung mass vertical velocity uses the same sign convention as the damper curve files in CRT. For instance, when the wheel centre moves upwards relative to the chassis, the dampers will be compressed. In other words, an upwards unsprung mass velocity is mapped to a positive (rebound force). Thus, the dampers provide a force that opposes the unsprung mass vertical velocity. Opposite to the Skyhook strategy, the sprung mass is neglected here. This neglection can lead to poor sprung mass control and poor ride performance. The goal of the Groundhook strategy is only concerned with improving unsprung mass control and road holding.

6.2.4 The Hybrid Strategy

The final control strategy modelled is the Hybrid control strategy. This strategy is a combination of the Skyhook and Groundhook strategies where the vertical velocities of the four unsprung masses and corners of the chassis are required. These velocities are recorded with virtual sensors in the CRT model as described in the last two subsections. Figure 38 presents a Simulink model for implementing the Hybrid semi-active control strategy.

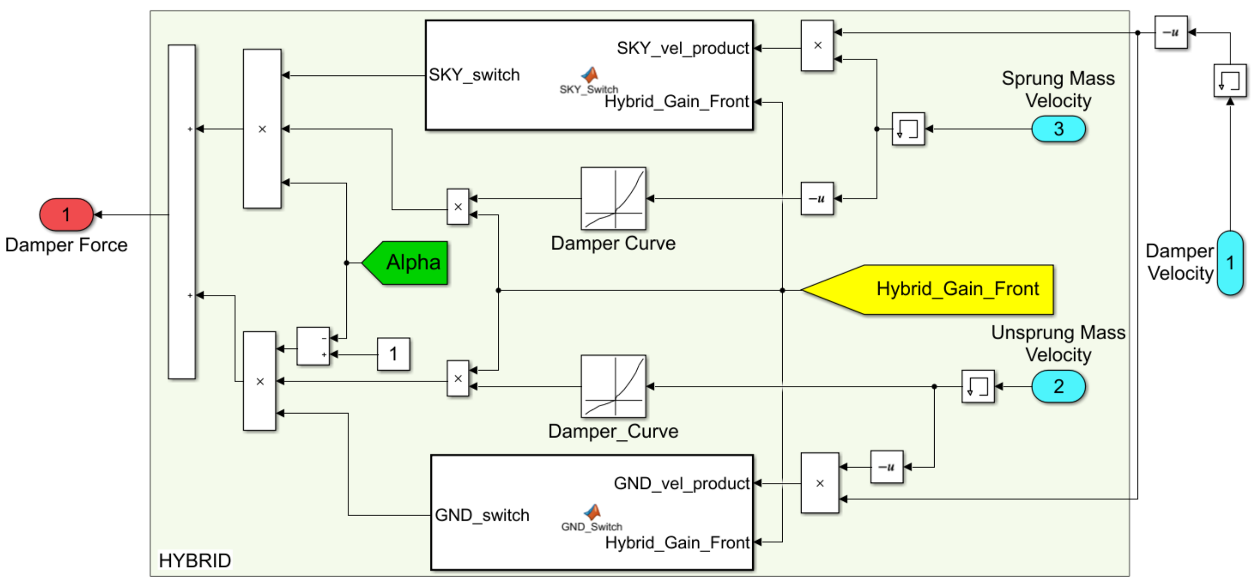


Figure 38: Hybrid control Simulink model

The two main workflows in the Simulink diagram in Figure 38 are nearly copied from the Skyhook and Groundhook control diagrams. Each workflow represents the Skyhook and Groundhook contribution to the Hybrid damper force calculation from Equations (34), (35), and (36). At the end of the control diagram on the left, the two contributions are summed to produce the Hybrid damper force. The additional alpha parameter is used to determine the relative weight of the Skyhook and Groundhook contributions to this damper force calculation. The alpha tuning parameter as well as "G" (labelled as the "Hybrid_Gain_Front") from Equation (34) can be found in the diagram in Figure 38. Note that "G" in equation (34) can lie between a maximum value of 2.2 and a minimum value of 0.2, just as the limits of the Skyhook or Groundhook gains. Instead of limiting one "G" parameter for all four dampers, it was decided to replace this parameter with an independent front and rear "hybrid gain" that can lie anywhere between 0.2 and 2.2. In this way, the front and rear dampers can be independently tuned in a timely manner by changing one constant in the Simulink diagram. The effect of changing the hybrid gains amplifies the slope of the damper curves, which is the same as changing the damping rates for the different regions of the damper curves. Therefore, tuning the hybrid gain parameters should produce a similar trend as found when tuning the low-velocity region of the damper curves (see Section 5.1.1). Moreover, the alpha parameter can be tuned by simply changing the constant in the Simulink diagram. Finally, the goal of the Hybrid strategy is to provide a balance between adding damping to the sprung and unsprung masses without losing control of either mass. This goal is aimed at improving ride and handling or generating an acceptable tradeoff superior to the ride and handling tradeoff found in the reference passive suspension (the default mode). The next subsections of Chapter 6 involves the selection of the strategies for the sport and ride modes.

6.3 Selecting the Best-Performing Control Strategies for Ride and Handling

Selection of the semi-active control strategies for each of the controller's suspension modes is based on several criteria. Referring to the results from the preliminary damper curve study in Section 5.1.1, it was made clear that dampers have a more significant effect on vehicle ride than handling. Here, more weight for evaluating the control strategy performances is thus placed on ride performance. Several primary and secondary ride metrics are evaluated on the body twist and cleat maneuvers. Furthermore, a collection of a few metrics from the ISO DLC, frequency response, and straight braking maneuvers are studied to evaluate the most significant handling impact the control strategies have on the vehicle. Note that the next chapter outlines the complete objective evaluation method used for evaluating the final settings for the controller modes. At this point, the goal was to find the best suspension mode candidates and eliminate controller settings which resulted in poor performance. The selected control strategies for the controller's sport and ride modes are to improve the ride mode's ride and the sport mode's handling over the default mode.

6.3.1 Objective Comparison of Control Strategy Performance

The goal of this section is to establish which control strategies are the best candidates for the initial semi-active suspension controller sport and ride modes. For the Switchable Damper Curves settings, a 200% Hard (named H. Curves) damper setting was created in an attempt to further improve the handling improvements found during the studies in Section 5.1.1. Moreover, a 50% Soft (named S. Curves) damper curve setting was added to further extend the secondary ride improvements from the same studies. The S. Curves setting contains damping rates that are 50% lower than those of the low-velocity region of the default dampers. For this initial evaluation, the Jeep Grand Cherokee model was used. For the Hybrid strategy, there are several parameters which can affect the strategy's performance. While modelling and testing the Hybrid strategy, it was discovered that a high alpha value above 0.8 generally improved ride and handling significantly, whereas values below 0.8 generally did not significantly improve ride and handling. For the initial objective comparison made here, the alpha value was set at 0.95 where most of the hybrid damper force is contributed by the Skyhook logic. Furthermore, the hybrid gain parameters were kept at their maximum values of 2.2 as this generally also provided the best ride and handling improvements. Investigation into the alpha parameter is presented later in section 6.3.3.

The first objective ride and handling aspects compared between the strategies were from the body twist and cleat maneuvers. Figure 39 presents the most significant findings. The first two metrics pertain to the body twist maneuver, whereas the last four originate from data recorded during the cleat maneuver simulations.

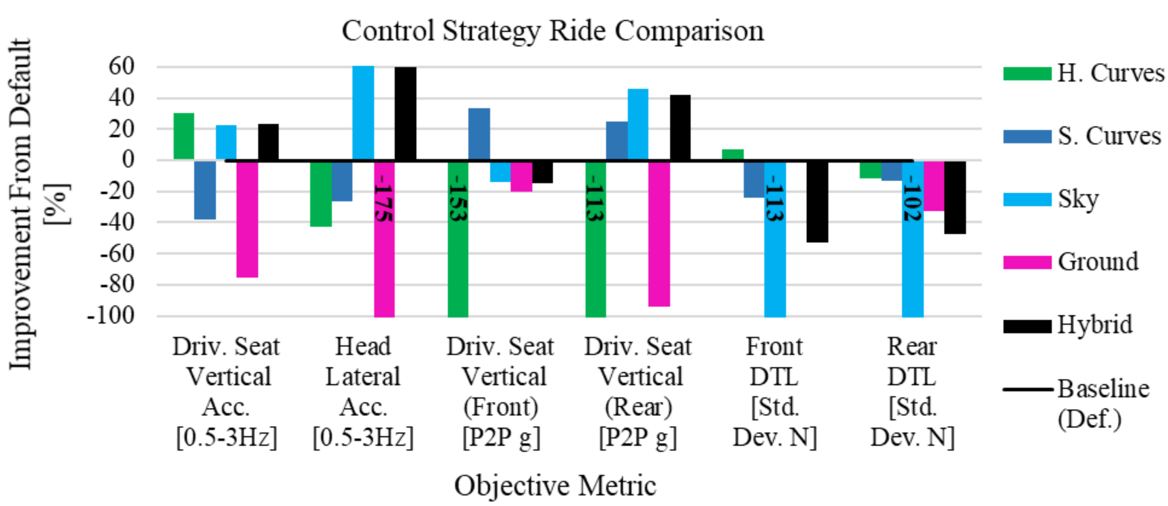


Figure 39: Ride performance comparison of semi-active control strategies

In Figure 39, the Skyhook (named Sky) and Hybrid strategies significantly improve the driver seat vertical acceleration and head lateral acceleration, both being measured in the primary ride frequency range, in RMS g's. These results indicate that the Skyhook and Hybrid strategies significantly improve control over the sprung mass. However, the H. Curves damper setting has the best improvement of suppressing the driver seat, primary ride vertical accelerations. This performance is contrasted by a large tradeoff in secondary ride when viewing the P2P driver seat vertical acceleration metrics, both of which are in the secondary ride frequency range. Regarding the DTL metrics on the cleat, the Skyhook strategy results in poor control of the unsprung mass motion as represented by the deterioration of the front and rear DTL. The variation in vertical tire force is more than doubled for the Skyhook strategy which suggests a worsening of road holding. For the Hybrid strategy with a high alpha parameter at 0.95 (95% weight on the Skyhook logic), this DTL deterioration is alleviated since the strategy places some weighting on the control of the unsprung mass. In the extreme case of the Groundhook strategy (named Ground), nearly all primary and secondary ride metrics are worsened. Furthermore, the DTL metrics are not improved. This result suggests that the Groundhook strategy is not an acceptable candidate for any semi-active suspension controller mode. See Appendix D for further discussion of the difference in sprung and unsprung mass control between the Skyhook, Groundhook, and Hybrid strategies. Finally, the S. Curves (50% soft) damper setting only improved the vehicle secondary ride on the cleat maneuver. Figure 40 presents the most significant improvements in handling discovered with the studied semi-active suspension control strategies.

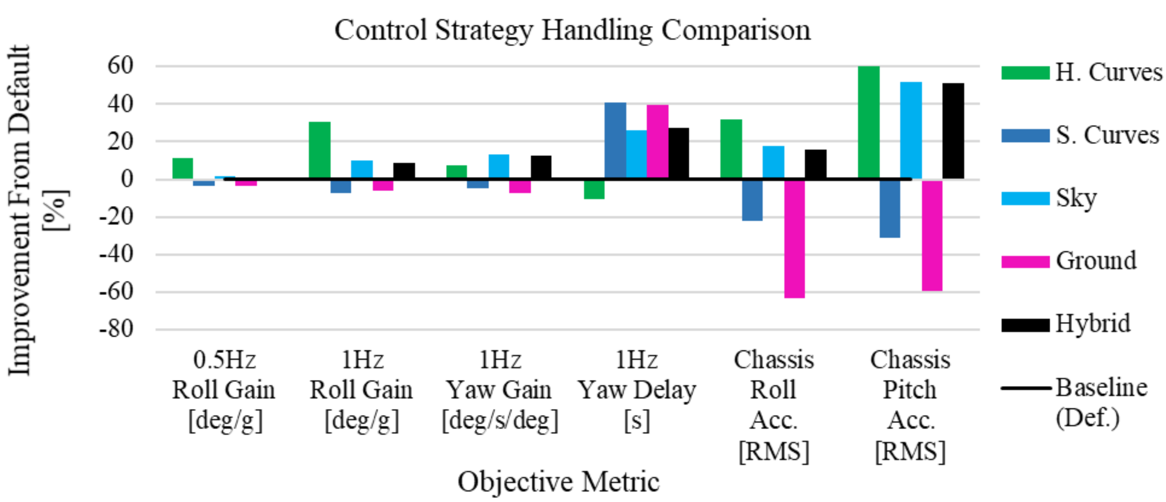


Figure 40: Handling performance comparison of semi-active control strategies

In Figure 40, the first four objective metrics were created from data recorded during the frequency response maneuver. The last two objective metrics are from the ISO DLC and braking maneuvers, respectively. Details on the list of all objective metrics for all maneuvers is presented in the next chapter. In general, the H. Curves (200% hard) damper setting suppresses the chassis roll and pitch motion in response to steering and braking inputs. The roll gain metrics on frequency response as well as the RMS chassis roll and pitch accelerations are objectively improved the most with this damper setting. Moreover, when analyzing the tire slip angles, there was no noticeable difference in tire slip characteristics between any of the controller settings on the frequency response and ISO DLC simulations. Regarding the yaw response of the vehicle, Figure 40 presents evidence that the Skyhook and Hybrid strategies can improve the handling of the vehicle as well as the ride performance over a passive (default) suspension. All metrics presented in Figure 40, with the exception of the first roll gain metric, are improved by at least 10% with the Skyhook and Hybrid control strategies. As found in the result of Figures 39 and 40, the Groundhook strategy did not show an improvement in vehicle ride or handling.

6.3.2 Initial Selection of Ride and Sport Mode Strategies

The initial selection of the semi-active controller modes were based on several criteria. For the ride mode, the goal is to improve both primary and secondary ride of the vehicle. The dissipation of the road disturbances, as predicted by the DTL objective metrics, should also be improved with the ride mode to avoid any long-lasting accelerations felt by the driver. At the same time, a significant tradeoff in handling cannot exist with the ride mode as this aspect could endanger the driver. The results in the previous section indicate that the Hybrid strategy significantly improves primary ride while also improving the secondary ride of the vehicle when the rear axle impacts a cleat. Additionally, the Hybrid strategy results in a significantly smaller tradeoff in the DTL objective metrics than the Skyhook strategy. Since the Hybrid strategy contains the tunable alpha parameter, this objective metric tradeoff could be further alleviated. Finally, the Hybrid strategy presented several noticeable improvements in handling with regards to the vehicle's roll, pitch, and yaw responses on several maneuvers. Thus, the hybrid strategy was chosen as the initial candidate for the controller ride mode.

The criteria for selecting the sport mode originated from the idea of improving the vehicle's roll, pitch, and yaw responses to driver inputs. The idea was to present a sport mode which responds quickly to the driver while providing a tight or sportier driving experience when executing turns. It was also a criterion to ensure the vehicle would not lose any road holding ability with the selection of the sport mode. The results of the study on objectively evaluating the semi-active control strategies found that the H. Curves damper setting improved the vehicles pitch and roll responses the most on the frequency response, ISO DLC, and braking maneuvers. On the cleat maneuver, the H. Curves avoided any significant tradeoff in road holding as predicted by the DTL objective metrics. Thus, the harder damper setting does not pose any disturbance dissipation worsening compared to the default suspension settings. However, this damper setting does result in a tradeoff in secondary ride performance, specifically on the cleat maneuver. In an attempt to keep the handling improvements found with this setting, it was decided to use the H. Curves as the initial selection of the sport mode. A potentially unacceptable tradeoff in secondary ride can be addressed on the simulator as the performance engineers at Stellantis are familiar with what the threshold is for acceptable ride in production vehicles. Upon completion of the subjective evaluation with the DiM 250, this threshold would present an objective target for engineers to use when tuning semi-active suspension controllers with a particular vehicle model in the future.

6.3.3 Offline Controller Tuning

This section highlights several tuning capabilities of the semi-active suspension controller modes that can be done quickly and offline. The most notable tuning aspect of the hybrid control strategy is the variation of the alpha parameter from 0 to 1. Recall that a value of zero places all weight on dampening the unsprung mass (Groundhook) whereas a value of one puts all weight on dampening the sprung mass (Skyhook). Figure 41 presents the variation in vehicle ride and an indication of road holding ability on the cleat maneuver when altering alpha parameter. The cleat maneuver provides a good prediction tool of the vehicle’s ability to dissipate road disturbances and for evaluating secondary ride. Thus, it was used to tune the Hybrid strategy. Upon studying the performance of a tuned setting during the cleat maneuver simulations, the objective performance on the other maneuvers must by checked as well.

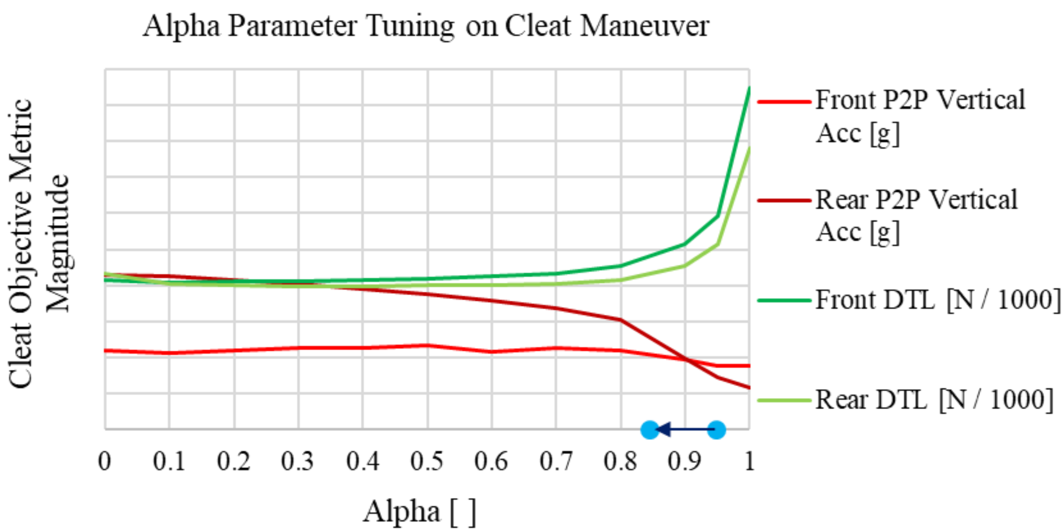


Figure 41: Offline effect of alpha parameter on Hybrid control strategy

Four metrics are presented in Figure 41 – the front and rear driver seat vertical P2P acceleration and DTL. The blue dots in the graph depict the change from the initial to final alpha value of the controller ride mode. The new value of alpha set to 0.85 significantly reduces the DTL whereas the rear P2P driver seat vertical acceleration is slightly worsened, but remains superior to the Grand Cherokee's passive suspension (default mode). For values of alpha below 0.7, the change in secondary ride and road holding abilities are not significantly affected. Note that the objective metric results in Figure 40 are for the Hybrid strategy with the front and gear hybrid gains set at 2.2.

Other parameters available for tuning the hybrid strategy consist of the hybrid front and rear gains as well as the reference damper curves as shown previously in Figure 38. The same trends as the results in the preliminary study in 5.1.1 were found when changing the damper curves in the hybrid strategy. Care should be taken when increasing both the steepness of the damper curves and the hybrid gain values as both increase the stiffness of the dampers. This aspect can lead to deterioration of the ride and handling if the damper forces become excessive.

6.4 The Controller and the Simulator

There were several steps involved in implementing the semi active suspension controller on the DiM 250 before the subjective evaluation method was developed and tested. This section highlights the major steps of testing and debugging the controller on a compact static driving simulator as well as fine-tuning adjustments for the initial controller modes on the DiM 250.

6.4.1 Connecting the Controller to the DiM 250 Dynamic Simulator

The first step of connecting the final semi-active controller involved building a simpler controller with the Switchable Damper Curve strategy. This simpler controller had the same architecture as the controller in Figure 35, having lower complexity than the final controller. This controller was used for verifying the connectivity between the CRT vehicle model, RTDBs, and Simulink. A compact simulator was used to test and debug this connectivity. Note that this is a static simulator housing a driver seat with active seat belts, a projector screen, a steering wheel with torque feedback, and other hardware. Once the connectivity of the models was established, the final controller with the Hybrid strategy and H. Curves damper settings could then be connected and driven on the compact simulator. Driving the vehicle model with the suspension controller on the compact simulator also allowed the stability of the controller to be tested. The yaw response and road holding on a proving grounds track were evaluated to ensure unprecedented handling performance was not caused by the final controller mode settings. Since the compact simulator does not provide seat motions, the ride was not tested. Figure 42, courtesy of Stellantis' media site, presents the compact simulator as well as the DiM 250 simulator that was driven once the semi-active suspension controller was debugged and tested.

Figure 42: Transition from the compact to the dynamic simulator

There is one alteration necessary when connecting a Simulink controller to the DIL environment with the simulator. To connect a controller's input and output signals with a vehicle model, certain RTDB input and output Simulink blocks replace the "VI-CarRealTime Solver Interface" block from Figure 34. This feature ensures the Simulink controller communicates correctly with SimWB and DriveSim in the DIL environment. A controller can then be compiled in Simulink and connected to the SimWB RTDB for the vehicle model being studied. Refer to

Section 4.1.2 for details regarding SimWB, RTDBs, and transferring a vehicle model from the offline to the simulator environment. With the semi-active suspension controller connected to the vehicle model and uploaded on the DiM 250, the tuning of the controller could then be completed.

6.4.2 Fine-Tuning the Controller

The architecture of the semi-active suspension controller permitted smooth transitions between suspension modes. When switching between modes on several virtual tracks, drivers noted there was no noise present. In fact, the drivers did not notice the switching between modes when driving on a flat road profile. The switching was only noticeable once an uneven road profile excited the dampers under a different controller setting. This switching was likely noticeable since the different suspension modes have different ride performances. The suspension modes required a few fine adjustments before they were fully subjectively evaluated by the expert drivers. Table 5 presents a summary of the adjustments made. A major advantage of using a dynamic driving simulator for tuning the controller was that changes to the controller could be made in real-time with the subjective feedback from the driver.

Table 5: Fine-Tuning of the Semi-Active Suspension Controller on the DiM 250

Controller Mode Affected	Adjustment Made (Or attempted)	Reason for Adjustment
Sport	Damper characteristic curve slope steepness reduction to 100% harder than default dampers	Secondary ride performance on body twist and cleat maneuvers was deemed unacceptable by expert drivers
Ride	Replace damper curves in Hybrid strategy with steeper curves (attempted)	Tried to improve primary ride performance, but yaw instability resulted where rear tires lost grip easily, and adjustment deemed unacceptable
Ride	Reduce front and rear "hybrid gains" values from 2.2 to 2 and 1, respectively	Expert drivers felt small vibrations while driving in ride mode, gains reduced until vibrations eliminated. This essentially softened the ride mode
Ride	Increase alpha parameter back to 0.95 (previously reduced to 0.85 during offline tuning)	Expert drivers felt the change in dissipation time between road disturbances was not affected between 0.85 and 0.95 alpha values. Thus, the value was set back to 0.85 value to retain original ride improvement

The adjustments presented in Table 5 refer to the controller settings for the Grand Cherokee model. This table highlights only a few of the adjustments that can be made with the use of a dynamic driving simulator when tuning a semi-active suspension. Due to the adjustments made to the sport and ride modes, the objective performance of the semi-active suspension had to be re-evaluated. The next chapter presents the details on the complete objective evaluation method. This method was used to aid in the decision of the initial controller modes and for the offline controller development and tuning.

CHAPTER 7. Objective Evaluation Method

This chapter is focused on the objective evaluation of ride and handling tailored for a semi-active suspension. This step in the project methodology was necessary to ensure the subjective evaluation method developed later on was implemented on a semi-active suspension that improved ride and handling. As a result, this step of the project is aimed at meeting the third sub-objective relating to help create and tune the semi-active suspension controller (See section 1.5.2). In reality, this step was completed in parallel with the development of the semi-active controller discussed in Chapter 6. First, the method is presented with regards to the objective metrics and simulated maneuvers. Next, how the controller modes' objective performances are compared as well as relative targets for the objective metrics are discussed. Finally, the validity of the sport and ride modes' objective performance is justified.

7.1 The Objective Method

It was highlighted in Chapter 3 that semi-active suspensions have not been comprehensively studied for ride and handling. Past studies had focused primarily on simulating a couple straight-line driving events and a few objective metrics. These metrics consisted of the SWS, the sprung mass vertical acceleration, and the DTL. The RMS roll and pitch accelerations as well as the pitch and roll angles were studied by a few other studies, but only road disturbances were studied (See section 3.3.2). Here, a more comprehensive evaluation method focuses on the vehicle vertical, lateral, longitudinal, yaw, pitch, and roll responses while considering road disturbances and driver inputs. The general procedure for the objective evaluation is as follows:

I. With the offline Simulink controller already setup with the necessary input .xml file from CRT (See section 4.1.1), the suspension mode is selected, and the offline co-simulation is started between Simulink and CRT

II. The results file of the co-simulation be opened in Adams Post-Processor. This software allows the user to define specific templates for post-processing certain maneuvers so that the necessary objective data can be automatically displayed

III. The results can then be exported in .csv format for additional post-processing. The objective metrics are extracted from a combination of MS Excel spreadsheet computations and manually from the plots in Adams Post-Processor

IV. Steps I through III are repeated for the remaining suspension modes to obtain their respective objective ride and handling metrics

The next section of the chapter presents the objective ride and handling metrics for each maneuver simulated.

7.1.1 Objective Metrics and Corresponding Maneuvers

In total there are seven maneuvers simulated in this project, each having several objective metrics. In general, individual maneuvers encourage certain vehicle responses. This feature allows engineers to evaluate specific aspects of a vehicle's ride and handling, as well to study individual contributions from changing components of vehicle subsystems. Here, the contributions from the different semi-active damper settings can be analyzed. Table 6 summarizes the maneuvers and objective metrics for the objective evaluation method.

Table 6: Semi-Active Suspension Objective Metrics and Corresponding Maneuvers

Maneuver	Highlighted Vehicle Response	Significant Objective Metrics
Body Twist	Chassis vertical motion in frequency domain	RMS Driver Seat Accelerations, Head Lateral Acceleration, SWS (front and rear)
Cleat	Chassis vertical motion and tire vertical response in time domain	P2P Driver Seat Accelerations, SWS (front and rear), DTL or Tire Force Settling Time (front and rear)
Frequency Response	Chassis yaw and roll response in frequency domain	Roll Gain and Delay, Yaw Gain and Delay, Lateral Acceleration Gain and Delay
Straight Braking	Chassis pitch response in time domain	Pitch Angle Settling Time, P2P Pitch Acceleration, Real Wheel Lift
ISO DLC	Chassis yaw and roll response in time domain during aggressive steering	Yaw Delay, Roll Delay, RMS Roll Acceleration
Slalom	Chassis yaw and roll response in time domain during moderate steering	Yaw Delay, Roll Delay, RMS Roll Acceleration
Max Performance Track Event	Vehicle lateral response and under/oversteer behaviour during transient cornering	Tire Maximum Slip Angles, Driver SWA Demand

In Table 6, it can be concluded that the wide range of objective metrics recorded across the seven maneuvers capture the vehicle's vertical, lateral, yaw, roll, and pitch responses. The objective metrics are able to capture the noticeable effects on ride and handling that the different semi-active suspension damper settings implement. Note also that Table 6 captures many of the objective metrics listed in Sections 3.1.3 and 3.3.2 where the list is tailored for evaluation of ride and handling of a semi-active suspensions. The objective metric results are presented in Chapter 9. Regarding the DTL metric on the cleat maneuver, it was discovered in the preliminary studies in Chapter 5 that the RMS of the vertical tire force did not capture any real difference in the tires' response to road disturbances (See Figures 29 and 30). The standard deviation, which measures the amount of variation in a variable, replaced the RMS value in this study since its results presented differences in the DTL by at least 10%. Thus, this approach was successful at objectively capturing the difference in the tire's response (see Figure 30) during the cleat simulations.

As for the vehicle's longitudinal response, it was discovered that all of the objective metrics evaluating aspects of the chassis longitudinal response to road disturbances and driver inputs did not present significant results. It was suggested that the suspension bushings would have more of an effect on the vehicle longitudinal response. Such bushings are not altered in this project. See Appendix E for a list of the objective metrics for each maneuver that did not identify noticeable changes in ride and handling results.

7.1.2 Post-Processing Objective Data and Comparing Controller Mode Results

For the post-processing of the results, all maneuvers simulated do not use the same method. For the cleat, straight braking, and max performance track event maneuvers, MS Excel templates are used to organize the data from the co-simulation results file. Several of the objective metrics are computed in these Excel templates whereas others such as RMS and P2P value metrics can be directly extracted from Adams Post-Processor using its built-in tools. For the ISO DLC and Slalom maneuver, computing the delay metrics requires the help of Matlab. Thus, the results from Adams Post-processor are exported in .csv format and read by a Matlab script which computes various delays between vehicle dynamic signals. For the body twist maneuver, all of the objective metrics can be extracted from Adams Post-Processor. The post-processing of the frequency response maneuver utilized the Adams Post-Processor add-on MB-Sharc to automatically generate necessary objective metrics. This template, which performs a Fast Fourier Transform to analyze the vehicle response in the frequency domain, was provided by Stellantis. Finally, additional post-processing was done in Excel to summarize the results of the objective evaluation method. Such results are presented in Chapter 9.

When comparing the performance of the semi-active suspension modes, the difference between a mode's objective metric and the default's metric is computed. In this way, the relative improvement or worsening of a given objective metric is displayed. Figure 43 presents the format for comparing the semi-active suspension controller mode performances.

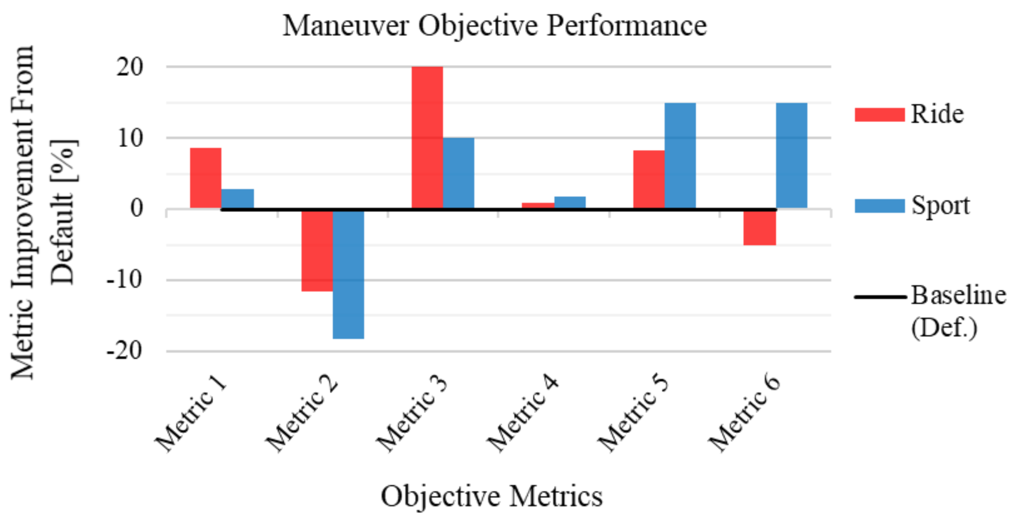


Figure 43: Comparing objective metrics between controller modes

The format in Figure 43 was chosen since the point of evaluating the semi-active suspension controller in this research is to establish an improvement of ride and handling over a reference passive suspension (the default controller mode). The purpose of the objective evaluation method is to ensure that an objective improvement in ride and handling exists with the sport and ride modes. In the following chapter, the subjective evaluation method developed is aimed at capturing these ride and handling improvements on a dynamic driving simulator. Therefore, it was necessary to implement a comprehensive objective evaluation before subjectively evaluating the new suspension. Alternatively, this step would be less important if a supplier semi-active suspension controller and its performance were already given to an OEM.

7.1.3 Objective Metric Targets

The targets for the semi-active suspension controller modes refer to which objective metrics are desired to be increased or decreased for a specific controller mode. Ideally, each metric should be improved as much as possible, but this is not always the case as there are still ride and handling tradeoffs with semi-active suspensions. Maximizing one objective metric improvement could deteriorate another. Moreover, absolute values for the objective metrics are not set since such values would have to be specific for each vehicle class and such absolute values were not available, since this is a new study. In general, the objective targets are considered achieved in this study if a suspension mode can improve the metric over the default mode by a significant percentage difference. Table 7 summarizes the targets for the objective metrics when evaluating a semi-active suspension's ride and handling.

Table 7: Objective Metric Targets for Ride and Handling of a Semi-Active Suspension

Maneuver	Objective Metric	Relation to Ride/Handling	Suspension Mode Target
Body Twist	(1) RMS Driver Seat Accelerations (2) Head Lateral Acceleration (3) SWS	(1) Primary Ride (2) Primary Ride (3) Handling – Ride Height	Sport – Reduce (3)
			Ride – Reduce (1/2)
Cleat	(1) P2P Driver Seat Accelerations (2) SWS (3) DTL or Tire Force Settling Time	(1) Secondary Ride (2) Handling – Ride Height (3) Ride/Handling - Dissipation	Sport – Reduce (2/3)
			Ride – Reduce (1)
Frequency Response	(1) Roll Gain and Delay (2) Yaw Gain (3) Yaw Delay (4) Lateral Acceleration Gain (5) Lateral Acceleration Delay	(1) Handling – Chassis roll (2) Handling – Chassis yaw (3) Handling – Response Time (4) Handling – Cornering Cap. (5) Handling – Response Time	Sport – Reduce (1/3/5) – Increase (2/4)
			Ride – N/A
Straight Braking	(1) Pitch Angle Response & Settling Times (2) P2P Pitch Acceleration (3) Real Wheel Lift	(1) Handling – Response Time (2) Primary Ride (3) Handling – Stability	Sport – Reduce (1/2/3)
			Ride – Reduce (2)
ISO DLC & Slalom	(1) Yaw and Lateral Acc. Delays (2) Roll Delay(s) (3) RMS Roll Acceleration	(1) Handling – Response Time (2) Handling – Response Time (3) Primary Ride	Sport – Reduce (1/2/3)
			Ride – Reduce (3)
Max Perf. Track Event	(1) Tire Maximum Slip Angles (2) Driver SWA	(1) Handling – Steering Behaviour (2) Handling – Drive Demand	Sport – Reduce (1/2)
			Ride – N/A

For the objective metrics in Table 7 pertaining to the driver seat accelerations, they all correspond to accelerations in the vertical direction. In the third column of Table 7, each objective metric contains a brief description of whether the metric evaluates ride or handling. In the fourth column of the table, the target metrics for the ride and sport modes are presented in parentheses. In general, the accelerations recorded at the driver seat and head level are targeted for a reduction when referring to the ride mode. The primary ride involving the chassis pitch and roll accelerations are also targeted for a reduction for the ride mode. Meeting these targets objectively improves the vehicle ride. For the sport more, the targets are set for improving the responsiveness of the vehicle to driver inputs, by reducing the response delays and times. Furthermore, the ability of the sport mode to execute turning maneuvers while maintaining grip, generating faster yaw rates, and higher cornering forces results in a better cornering capacity for the vehicle. For each semi-active suspension mode, as many of the targets as possible should be achieved to maximize the ride and handling benefits over the default passive suspension. This aspect is addressed in the results section. The next section of this chapter provides a discussion of the validity of the results for the sport and ride modes for the objective evaluation method.

7.2 Validity of Controller Objective Performance

In Section 4.3.2 a discussion was presented on the validation of the vehicle models for the offline simulations. The discussion focused on the vehicle models with their passive suspensions, already proven to replicate the physical vehicles in production. In this sense, the default modes of the semi-active suspension controllers have accurate ride and handling performance with respect to the physical vehicle. For this project, the objective evaluation method is evaluating the difference in objective ride and handling between the default and other suspension modes. This final subsection of Chapter 7 provides insight on how to interpret the results of the objective evaluation and their validity.

7.2.1 Performance Relative to Baseline Vehicle

The evaluation methods in this research are virtual, whether they are objective or subjective. Physical dampers with a control unit for implementing the semi-active suspension control from this research has yet to be fabricated and implemented on a vehicle. Therefore, there is no way to validate the sport and ride modes for this research with data from a physical vehicle. However, the objective evaluation method in this research is tailored for observing the relative percentage improvements in ride and handling over a passive suspension (the default mode). The default mode has been validated with objective data from testing a physical vehicle. Furthermore, the absolute values for the objective ride and handling metrics of the default mode are accurate and are used as a baseline in the evaluation method for this project. Thus, the relative ride and handling performance improvements computed for the sport and ride modes can be considered accurate and valid, since they are relative to the baseline performance. This is the reason why the objective targets and post-processing of the objective results only compares the relative changes between the default, sport, and ride modes. This methodology will allow future development of semi-active suspensions to be conducted in the same manner. OEMs and damper suppliers can use their objective data from a passive suspension to define a reference performance in which they can attempt to improve with their semi-active suspension technology. In the case an OEM is developing a new vehicle model and wishes to add a semi-active suspension, the validity of the objective metrics must be considered. Any improvement in the ride and handling would be considered as an estimate since the vehicle model with a passive suspension could not be validated at that time. Nevertheless, the objective evaluation would provide estimates on the relative improvement of the suspension and could be used to remove unfeasible design iterations.

In the automotive industry, subjective evaluation is always required on a vehicle before it is release. Currently, there is not a published method for subjectively evaluating ride and handling of semi-active suspensions with a dynamic driving simulator. The next chapter presents the details of such a method that has been developed through the research of this project.

CHAPTER 8. Subjective Evaluation Method

The use of a dynamic driving simulator allows human drivers to subjectively evaluate vehicle ride and handling. This step brings the evaluation method one step closer to evaluating a physical semi-active suspension system. With the newly developed method, intermediate suspension design iterations can be accurately evaluated and tuned on a simulator while acquiring subjective feedback. This method reduces the time and costs associated with producing intermediate physical prototypes. Conventional subjective evaluation methods do not use a dynamic driving simulator. These methods rely on physical alterations to the vehicle's suspension, which require significantly more time and money compared to the virtual alterations that can be made on the simulator. The method described next provides this project's primary novelty as well as a path to avoiding the costs of the conventional evaluation methods.

The chapter begins with an overall description of the subjective evaluation method. Afterwards, the subjective metrics and the maneuvers on which they are evaluated are presented. Then, the rating scale used by the expert drivers for the evaluation is discussed. The overall questionnaire for the subjective evaluations will be presented, followed by the procedure used for implementing said method. A discussion of the drivers for the subjective evaluations and the validity of the subjective results is also included. The chapter closes with the description of an additional study for determining the correlation between objective metrics and subjective ratings for ride and handling of semi-active suspensions.

8.1 The Subjective Method

The method for subjectively evaluating semi-active suspensions involves drivers driving a vehicle model with each of the suspension modes on the DiM 250. The seven maneuvers are executed one at a time for each of the suspension modes, resulting in a minimum of twenty-one driving simulations. Drivers are given a questionnaire to be filled out upon completion of each driving simulation. This questionnaire records the drivers' ratings for the subjective metrics on ride and handling. At the end of the final subjective evaluations, the ratings are compiled and the relative improvement of the vehicle ride and handling over the passive suspension is presented. As a subjective evaluation method on semi-active suspensions for ride and handling has yet to be published, the development of this method was an iterative process. On several occasions, drivers would drive a vehicle model with intermediate versions of the semi-active suspension modes and provide subjective feedback on both the ride and handling performance as well as the questionnaire itself. Therefore, the development of the method was not the result of one researcher, but rather involved a collaboration with vehicle dynamics experts and expert drivers. The following subsections present details on all important aspects of the subjective evaluation method, beginning with the metrics rated by the drivers.

8.1.1 Subjective Metrics and Corresponding Maneuvers

The subjective metrics were chosen based on several factors. First, the subjective metrics from literature (See Table 2) were considered. Several of the handling subjective metrics were adopted since drivers felt a difference in the metric when evaluating different controller modes. From the ride metrics, the "Primary Ride Control" was adopted while changing the name to focus on the vertical direction of motion. A second factor for choosing a subjective metric was based on the objective metrics from Chapter 7 which could be directly perceived by the driver. The idea was to translate the most significant objective impacts on ride and handling into subjective aspects or qualities. For instance, the P2P Driver Seat Acceleration from Table 7 was translated into a subjective metric describing the severity of the impact felt by the driver when driving the vehicle over a cleat, for both the front and rear axles individually. A third factor for selecting subjective metrics regarded asking for the opinion of the drivers. They were asked which metrics they thought were most appropriate for a given maneuver as well as which ride and handling metrics they typically use for evaluation of ride or handling. This questioning was done to ensure the expert knowledge of industry professionals was captured by the questionnaire. After several driving sessions, the final list of subjective metrics was created. This list is summarized in Table 8. See Appendix F for the description of each metric.

Table 8: Subjective Metrics for Vehicle Semi-Active Suspension Ride and Handling

Maneuver	Subjective Metric	Ride or Handling?
Body Twist	(1) Vertical Ride Control (2) Lateral Head Toss	(1) Ride (2) Ride
Cleat	(1) Driver Disturbance – Front (2) Driver Disturbance – Rear (3) Disturbance Dissipation	(1) Ride (2) Ride (3) Ride

Frequency Response	(1) Roll Response (2) Roll Delay (3) Yaw Response (4) Yaw Delay	(1) Handling (2) Handling (3) Handling (4) Handling
Straight Braking	(1) Pitch Abruptness (2) Pitch Delay	(1) Ride (2) Handling
ISO DLC & Slalom	(1) Maneuverability (2) Delay (3) Steering Wheel Activity (4) Roll Response	(1) Handling (2) Handling (3) Handling (4) Ride
Max Performance Track Event	(1) Stability (2) Roll Response (3) Turn-In Response (4) Steering Wheel Activity	(1) Handling (2) Ride (3) Handling (4) Handling

The combination of ride and handling metrics in Table 8 are evaluated by drivers to capture the primary ride, secondary ride as well as the response of the vehicle to driver inputs. The combination of the maneuvers in Table 8 for the handling evaluation incorporates driver acceleration inputs, braking inputs, and steering inputs. Thus, the vehicle's vertical, lateral, pitch, yaw, and roll responses are captured through the implementation of the list of subjective metrics. Note that the metrics in Table 8 are the most significant aspects affected when changing between the default and semi-active suspension modes of the controller. See Appendix F for other metrics that were originally considered and then deemed insignificant upon testing. The exact naming of the subjective metrics were chosen to ensure consistency with the technical language of the drivers. The description of each metric was made clear and concise so that the drivers were not exposed to complex, unfamiliar descriptions. Finally, a descriptor was placed next to each metric to ensure the driver knew whether ride or handling was being evaluated. As a result, the published research on previous subjective evaluations, the identified objective impact that semi-active suspensions have on ride and handling, and the knowledge of expert drivers were combined to create a list of subjective metrics for evaluating semi-active suspensions.

8.1.2 Subjective Rating Scale

The rating scale implemented in the subjective evaluation method was a ten-point absolute scale based on an SAE standard. Specifically, the SAE J1060 rating scale was used since the expert drivers were already familiar with this scale [101]. The scale has integer rating values ranging from one to ten with an equivalent, qualitative text description for each rating. Furthermore, The SAE standard contains a list of additional text descriptions that state which ratings are noticed by certain skill levels of observers. For example, the integer rating ten corresponded to an "excellent" performance, which is deemed "not observed." In the research discussed in the literature review (See Section 3.2.1), the authors noted that the subjective metrics in their studies were rarely or never rated as their scale's maximum (best) value. The highest rating is regarded as a potential target to strive for when tuning a vehicle. Figure 44 is a condensed version of the rating scale used for this project. The remaining details of the scale have been removed to avoid conflicts of confidentiality.

Subjective Rating									
Intolerable	Severe	Very Poor	Poor	Marginal	Barely Acc.	Fair	Good	Very Good	Excellent
1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**

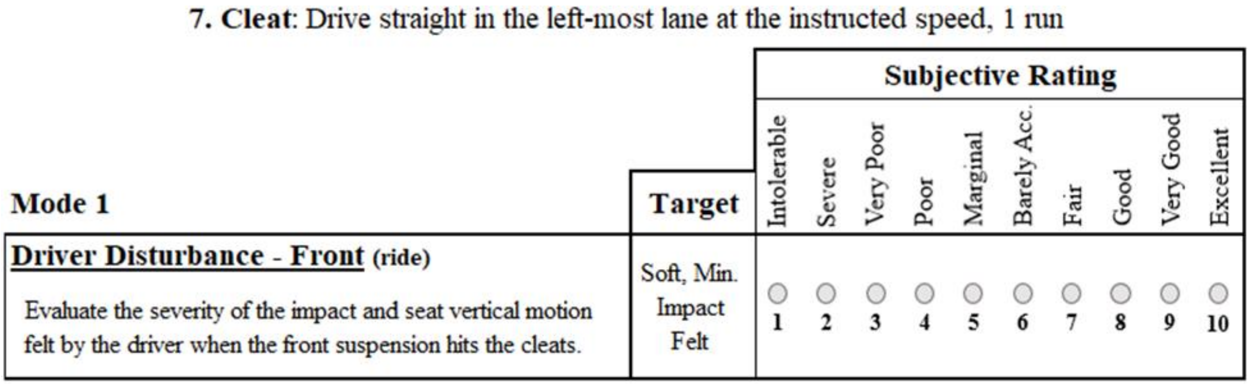
Figure 44: Subjective rating scale used for evaluation of vehicle semi-active suspension

An additional detail not shown for the rating scale in Figure 44 is a third text description pertaining to the acceptability of the performance perceived by the drivers. This additional description is meant to provide information to the evaluators on whether the subjective performance is acceptable or not. This aspect had a significant importance when tuning the semi-active suspension modes. For instance, when tuning the damper curves for the sport mode, a collection of hard damper settings ranging from factors of 1.5 to 3 times the default damper curve low-velocity region slopes were evaluated. At the same time, a collection of expert drivers were also asked to state whether the secondary ride tradeoff was "acceptable." This tuning methodology had each driver execute the body twist and cleat maneuvers and rate the ride metrics based on acceptability. This acceptability would determine whether the sport mode damper settings could be practically realized in a physical production vehicle. Thus, the subjective evaluation method not only evaluates the relative improvement of the semi-active suspensions ride and handling, but also the acceptability of the performance for consumers.

Upon completion of the subjective evaluations, the ratings are compiled in a MS Excel spreadsheet and averaged between the drivers. Both the absolute ratings of the drivers for each suspension mode and the change in ratings between the modes are recorded. This feature allows both the acceptability and the relative improvements of ride and handling over the default mode to be addressed. For a final note on the rating scale, the resolution can be increased to non-integer values as specified in SAE J1060 for evaluating less significant differences in ride hand handling [101]. Further discussion and use of a finer scale is discussed later in Section 8.3 for the correlation study conducted.

8.1.3 Subjective Questionnaire

The review of literature from Section 3.2.1 identified that a common requirement of subjective evaluations of ride and handling is to use some form of a questionnaire. The purpose of the questionnaire is to combine the list and description of subjective metrics, the rating scale, and additional information in a concise and clear format. This format allows the subjective evaluations to be executed in a timely manner while avoiding miscommunication between the drivers and the engineers conducting the testing. The drivers always had the questionnaire with them when driving. The questionnaire contains several pages. The first of which concisely displays the procedure of the evaluation method, the order and details of maneuvers being evaluated, and all details on the subjective rating scale. The remaining pages of the questionnaire provide a list of subjective metrics, their descriptions, a target improvement for each metric, and a condensed version of the ratings scale for each maneuver. The final page of the questionnaire is a feedback section requesting any additional comments the drivers could have on the suspension modes' ride and handling. Specifically, the vehicle's character (i.e., agile, sporty, boring, etc.) is requested. This feature provides additional information that could not be captured by the rating scale. Figure 45 presents the structure of a single page of the questionnaire related to one of the seven maneuvers. See Appendix F for the full structure of the questionnaire.



7. Cleat: Drive straight in the left-most lane at the instructed speed, 1 run

		Subjective Rating									
Mode 1	**Target**	Intolerable	Severe	Very Poor	Poor	Marginal	Barely Acc.	Fair	Good	Very Good	Excellent
Driver Disturbance - Front (ride) Evaluate the severity of the impact and seat vertical motion felt by the driver when the front suspension hits the cleats.	Soft, Min. Impact Felt	1	2	3	4	5	6	7	8	9	10

Figure 45: General structure of the subjective evaluation questionnaire

In Figure 45, only one of three sections for rating the suspension controller modes is displayed. In reality, there would be two more boxes similar to the one shown in Figure 45, but with the remaining subjective metrics. Note that the condensed subjective rating scale is only repeated at the top of each page of the questionnaire for clear accessibility for the drivers. The structure of the questionnaire was developed based on the aspects of other questionnaires discussed in the literature review (Section 3.2.1). However, this new questionnaire was created for subjectively evaluating the ride and handling of semi-active suspensions. Another new aspect of the questionnaire developed in this research was the addition of the metric "target" column. This column, found in Figure 45, provides text describing the target performance for the corresponding subjective metric. These targets provide the direction of improving rating values – a higher (better) integer corresponds to the performance matching the target column's description. This feature also clearly communicated the purpose of each subjective metric to the drivers, in case of any confusion. In the future, this target column could also provide a link between the knowledge of vehicle dynamics simulation engineers and professional drivers. This feature would help train future engineers working in the simulator lab to become more familiar with ride and handling subjective evaluations. Eventually, the simulation engineers could take on some of the responsibilities of performance engineers, alleviating some of the need to always book performance engineers for testing.

For each page of the questionnaire excluding the first and last page, the parameters for a given maneuver are displayed. These parameters include a target vehicle speed, steering input instructions, when and how to apply brakes (straight braking maneuver), and other information. This feature is another critical aspect of the subjective evaluation method since several maneuvers must be completed in a certain way. For instance, the body twist maneuver must be driven though at one specified speed to excite certain frequencies of the chassis motion. Finally, the number of driving simulations for each maneuver is specified. The number of runs for each maneuver was determined to allow ample time for rating the subjective metrics and keeping the total time of the subjective evaluations to a practical limit. However, if a driver requested an additional driving simulation, it was permitted. For details on the procedure for the complete subjective evaluation method, please refer to the next subsection.

8.1.4 Subjective Evaluation Procedure

The procedure for implementing the subjective evaluation method involves a sequential list of steps that are strongly recommended to be followed. Individuals conducting this procedure should also keep several important rules in mind when conducting the evaluations. The purpose of these rules is to ensure there is no bias from the drivers before driving the simulator. The rules are as follows:

I. Until a driver enters the simulator for the subjective evaluations, the driver should have no interaction with other drivers who have already completed an evaluation.
II. Drivers waiting to complete a subjective evaluation should not be able to see other drivers perform subjective evaluations, since viewing the simulator in motion can produce expectations (Simulator cabin motions were noticeably different when observing the evaluations of different suspension modes from the simulator control room)
III. The drivers should never be told which suspension mode they are driving with since it avoids the potential for drivers to set predetermined expectations before driving with the sport or ride modes

Note that the first suspension mode to be evaluated is always the default mode (baseline passive suspension). This decision was based on the fact that the drivers were already familiar with the default mode's damping settings. Also, the drivers were used to following a procedure where a reference vehicle is driven first. This approach was also highlighted in the literature review, where a reference vehicle is driven before an altered vehicle so that drivers can focus on ride or handling improvements from a validated baseline. Thus, the default mode (named "mode 1" in the procedure) was always driven first, followed by the ride and sport modes. This approach allowed the driver to always use the first evaluations with the default mode as a reference performance for subsequent modes. Recall that this subjective evaluation method is for determining if the improvements in ride and handling found during the objective evaluation are noticeable and can be captured on the dynamic simulator. The procedure was not designed to test the skill of the drivers to notice such improvements. A random order of suspension modes could otherwise confuse the drivers during the procedure and distract them from focusing on the ride and handling improvements. Considering these details, the steps for implementation of the subjective evaluation method are defined as follows:

1. **General Briefing**: Advise the drivers on the purpose of the procedure and briefly review the subjective rating scale, the subjective metrics, and review the questionnaire. Also state the rules during the testing.
2. **Subjective Evaluation** (approximately 45 mins/driver): Upon entering the simulator with the questionnaire and a pencil, each driver executes a maneuver with all suspension modes before moving onto the next maneuver as follows:
    - I. **Maneuver 1**: With Mode 1 set by the engineers in the control room, drive the simulator according to the maneuver instructions provided from the control room (these details are also on the top of each questionnaire page). Upon completion, the driver then fills out the current mode section of the page for the current maneuver. This section is where the drivers record their subjective ratings.
        - a) Engineers in the control room switch the suspension to Mode 2 (ride) and step I is repeated with this new mode
        - b) The suspension mode is switched to Mode 3 (sport) and step I is repeated again
        - c) The driver is asked to add any comments they have on the three modes and place them in the driver feedback section on the back of the questionnaire
    - II. **Maneuvers 2-7**: Repeat step I for the remaining maneuvers. Note that for the Max Performance Track Event, only certain portions of the track are driven on. The sections were chosen based on their ability to excite the dampers and promote chassis roll, pitch, and generally highlight certain handling aspects.

Thus, long and smooth straightaways and steady state cornering sections were skipped

III. **Final Comments**: Upon the driver's return to the control room, discuss the driver feedback section to discuss the results and determine if the drivers would accept Modes 2 and 3 as ride and sport modes, respectively.

3. **Post-Processing**: Once the subjective evaluations are completed for all drivers, compile the average ratings, and plot the absolute and difference in ratings between the suspension modes. The standard deviation between the drivers can also be plotted to observe the agreement in ratings between drivers.

This concludes the presentation of the subjective evaluation method. This approach utilizes a new list of subjective ride and handling metrics described for evaluating vehicle semi-active suspension. The rating scale provides a way to record the absolute acceptability and performance of each suspension mode, while also permitting the observation of relative performance between the modes. A questionnaire incorporating the subjective metrics, rating scale, targets for improvement, and the general procedure for the method provides a concise mode of implementing the subjective evaluation and recording the ratings from expert drivers. The results of implementing this method with the semi-active suspension controller developed in Chapter 6 can be found in the next chapter. The next section provides a discussion of the drivers used in this project.

8.2 Drivers for Subjective Evaluations

Similar to the discussion of validating the objective semi-active suspension performance, the validity of the results from the subjective evaluation is presented here. Details on the use of drivers for determining this validity and on the drivers used in this research are also presented.

8.2.1 Performance Engineers for Validating the Baseline Vehicle

The default mode of the semi-active suspension contains the damper curves replicating the passive suspension found in the physical vehicle. The validation of the vehicle model's subjective ride and handling performance has already been carried out with performance engineers. These performance engineer drove both the physical vehicle on proving grounds and the virtual model on the DiM 250. The parameters and K&C data curves of the CRT model were then tuned until the performance engineers found that the model replicated the vehicle with a high degree of fidelity and accuracy. For the subjective evaluation of the sport and ride modes, the evaluation method focuses on capturing the improvements in ride and handling relative to the default mode. Since the default mode performance is valid, the relative improvements can also be considered valid. The only difference between the three suspension modes is the magnitude and behaviour of the damper forces. It has already been proven that physical MR dampers can replicate the damper forces produced by the Hybrid strategy [75]. Noting also that the DiM 250 has been proven in other studies to replicate relative performance improvements when altering other suspension components [68], the subjective ratings found in this research can be considered valid.

8.2.2 Availability and Selection of Drivers

Due to the unfortunate global circumstances and resulting travel restrictions, it was not possible to use the performance engineers who validated the vehicle models studied in this project. However, as the evaluation methods in this research focus on recording the relative change in ride and handling improvements from a validated baseline vehicle, it is not necessary to use such performance engineers. Instead, individuals considered as both experienced drivers and vehicle dynamics experts from Stellantis kindly participated in the final subjective evaluations. Similar to the performance engineers, these drivers are familiar with the primary baseline vehicle model and subjectively evaluating ride and handling on a regular basis. Furthermore, these individuals work with performance engineers as part of their occupational responsibilities. A total of three drivers were available to evaluate the three semi-active suspension controller settings. Three drivers is one more than the typical number of performance engineers used for validating vehicle models. Therefore, for the purpose of developing and testing the subjective evaluation method, the drivers used for this research and their subjective evaluations are acceptable.

8.3 Objective Metric to Subjective Rating Correlation

The subjective evaluations carried out on the dynamic driving simulator present a significant opportunity to extend the research of this project into correlation studies. These studies can be used for discovering the correlation between objective metrics and subjective ratings when changing only a vehicle model's damper settings. Such a study utilizing a dynamic driving simulator has yet to be published and provides a new tool for implementing correlation studies in an accurate and timely manner. Through identifying the correlation between objective and subjective metrics, the prediction of future subjective ratings could be used based purely on objective results. This process could further reduce the development time for tuning suspension and vehicle models. A second opportunity presented is to study the sensitivity of the DiM 250 when evaluating ride and handling. The result of the study will also provide users of the simulator with information detailing how small of changes in ride and handling are noticeable on the DiM 250. The details and general procedure of the correlation study implemented are presented next.

8.3.1 Damper Correlation Study

To produce a larger set of data for the objective metrics and subjective ratings on ride and handling (see Tables 6 and 8) and to cover a wider range of changes to the suspension, the study involves evaluating a range of damper curve settings. Specifically, the damper curve settings from Section 5.1.1 (see Figure 25) were evaluated. These seven damper settings produce a wider, more comprehensive, and practical range of OEM damping settings to be evaluated compared to the three suspension modes of the semi-active suspension controller. Therefore, the results from this study provide correlations between objective and subjective metrics that cover the practical limits of damper curves for the vehicle models studied. Finally, the same expert drivers from the subjective evaluation of the semi-active suspension are used in this correlation study.

The procedure for the objective and subjective evaluations are the same as stated in Chapters 7 and 8, respectively. Here, three maneuvers chosen including the body twist, cleat, and ISO DLC maneuvers. The most significant changes in subjective ratings were found on these maneuvers and such maneuvers provide a wide range of ride and handling performance aspects. This method also permitted the study to be conducted in a timely manner where the total number of subjective evaluations and corresponding time required were reduced. Figure 46 contains the workflow of this correlation study.

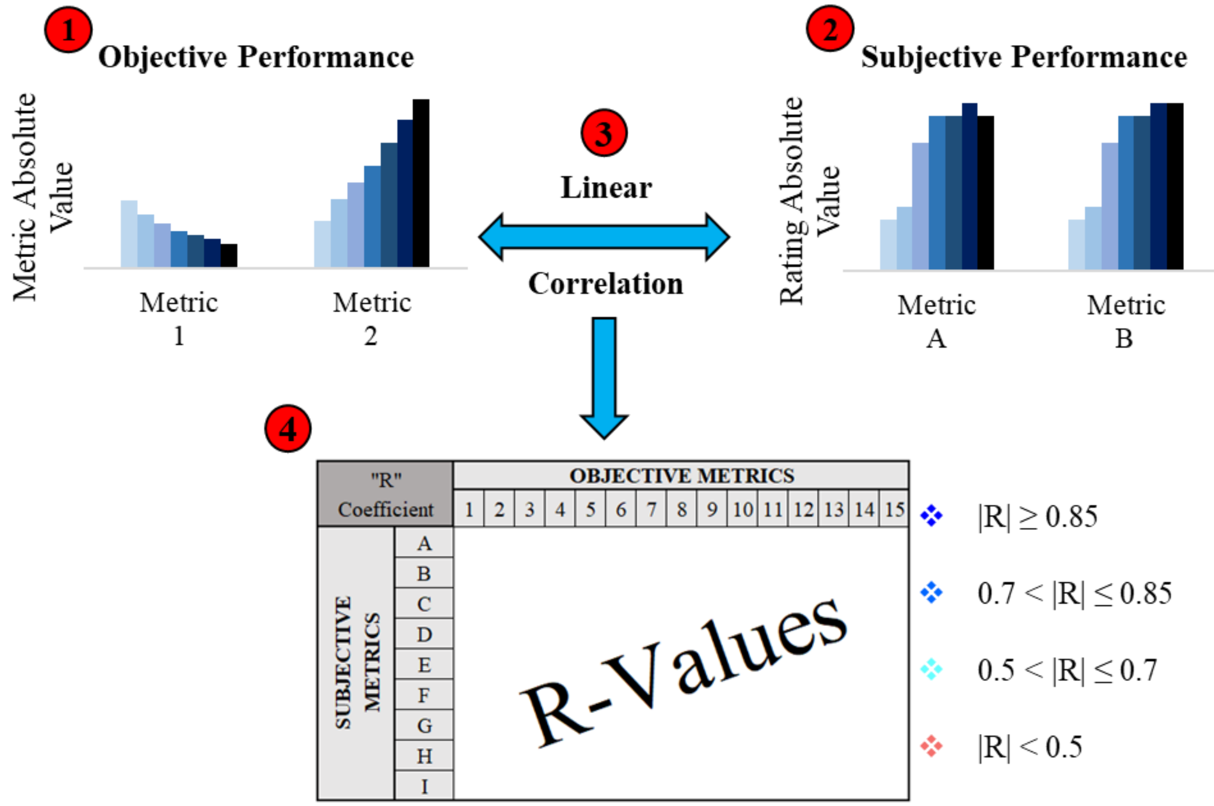


Figure 46: Correlation study workflow

Referring to Figure 46, the first step is to objectively evaluate the seven damper settings to obtain the values for the objective ride and handling metrics. Then, the same seven damper settings are subjectively evaluated with the dynamic driving simulator to obtain the values for the subjective ratings. The third step is to perform a linear correlation between the absolute values of the objective metrics and subjective ratings. This correlation is done by creating a grid with all possible pairs of objective and subjective ratings. From the three maneuvers, there are fifteen objective metrics and nine subjective ratings. Thus, there are 135 possible pairs. The four ranges of correlation coefficient "R" absolute values are adopted from linear correlation studies on steering and handling [57], [58]. In general, a coefficient absolute value above 0.7 resembles a good correlation between an objective and subjective metric. For absolute values above 0.85, a strong linear correlation exists. In the case a negative coefficient value is found, it means that as the objective metric is increased, the subjective metric will decrease, and vice versa.

A final important note to state on this correlation study is the alterations that were required in regard to the subjective evaluation questionnaire. Due to the reduced number of maneuvers, the questionnaire length was reduced. Moreover, the explanation on the purpose of the evaluation was changed. Here, the drivers were told to focus on identifying a change in ride or handling and determining how significant the change was. To capture less noticeable ride and handling improvements, the resolution of the rating scale was increased to 0.5 increments, rather than the whole integer values. According to the subjective rating scale, a 0.5 rating increment represents a "significant difference" that can be noticed by some customers. Moreover, the refinement was increased due the relatively small changes in objective metrics between damper settings found during the preliminary damper studies (See the results in Section 5.1.1). This alteration provided a larger number of ratings for drivers to choose from and it provided information on the simulator's sensitivity.

The final results of the damper correlation study and more are presented in the next chapter. The results of this study will provide OEMs with insight on which objective metrics to focus on when tuning their virtual suspension models (passive and semi-active). The objective results could be used to suggest how the resulting subjective ratings when evaluating such models will change and how noticeable these changes will be.

CHAPTER 9. Results & Discussion

The final results of all objective and subjective studies are presented in this chapter. First, the objective ride and handling of the vehicle models with the semi-active suspension controller are presented. The results with the main vehicle model, the Grand Cherokee, will be presented first, followed by the Renegade model results. Recall that the studies with the later vehicle model were completed to test the robustness of the semi-active suspension controller. After the presentation of the objective evaluation results, the results of the completed subjective ride and handling evaluations are presented, followed by the results of the correlation study.

The final settings for the semi-active suspension controller are summarized in Table 9. Most of the tuning process for developing these final settings was done in the offline environment since the alterations could be made and evaluated in the fastest time. Once a viable controller mode setting was found, the controller performance was subjectively evaluated on the DiM 250 to confirm the ride or handling improvement over the passive suspension (default mode).

Table 9: Final Semi-Active Suspension Controller Settings

Vehicle Model	**Controller Mode**	**Alpha Parameter** (Hybrid strategy only)	**Front Hybrid Gain** (Hybrid strategy only)	**Rear Hybrid Gain** (Hybrid strategy only)	**Front Damper Curves**	**Rear Damper Curves**
Jeep Grand Cherokee	Ride	0.95	2	1	Default	Default
	Sport	n/a	n/a	n/a	100% Hard	100% Hard
Jeep Renegade	Ride	0.8	1.3	1.3	Default	50% Hard
	Sport	n/a	n/a	n/a	100% Hard	100% Hard

For both sport modes, damper curves having 100% steeper slopes for the low-velocity damping region compared to the default mode were studied. Notice that the controller ride mode settings for the Grand Cherokee and Renegade are different. The difference in Hybrid strategy parameters likely resulted from the fact that the models represent vehicles from two different classes, having several dissimilar vehicle characteristics (See Section 4.3). Despite the difference in control settings, improvements in ride and handling were found with both vehicle models, as can be seen in the following chapter sections.

9.1 Final Objective Performance

The ride and handling results of the objective metrics from Table 6 are presented in this section. These results are from the offline objective evaluation method discussed in Chapter 7, where each vehicle model with the three semi-active suspension modes is simulated on the seven maneuvers. The plots in the following subsections highlight the percentage differences in objective ride and handling from the default mode.

9.1.1 Controller Objective Ride and Handling – Primary Vehicle Model

The objective results for the body twist and cleat maneuvers are presented in Figure 47. The first four metrics in Figure 47 originate from the body twist, whereas the remaining four originate from the cleat maneuver.

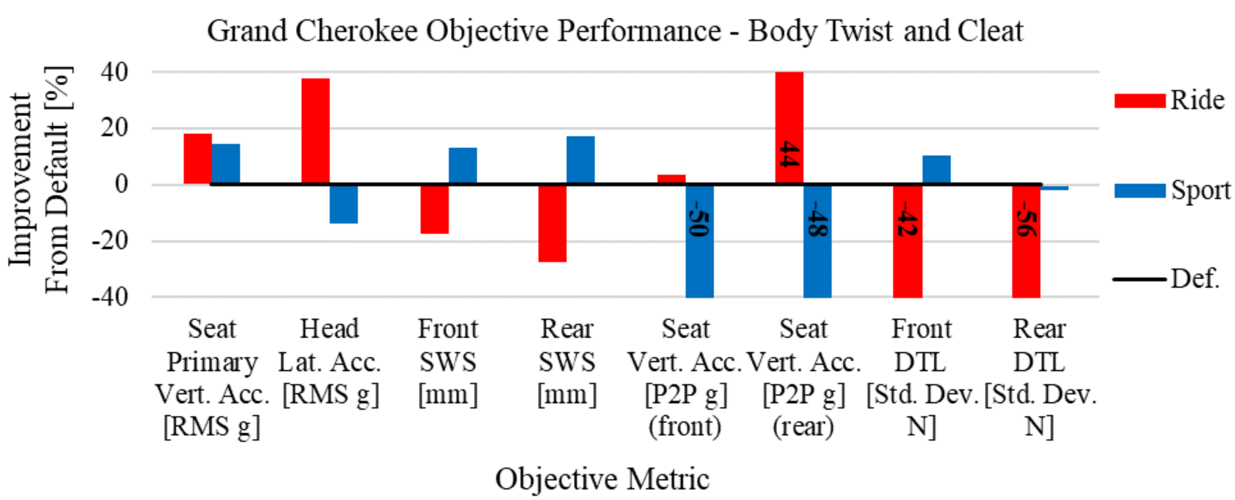


Figure 47: Grand Cherokee objective performance on ride maneuvers

In Figure 47, the objective results indicate that the ride mode improves both primary and secondary ride over the passive suspension. Most notably, the driver seat vertical accelerations and the lateral accelerations at the driver's head level are significantly improved (reduced). The two metrics presenting improvements in P2P vertical accelerations at the driver seat relate to secondary ride. Upon the front axle of the vehicle model impacting the cleat, the reduction of the driver seat vertical accelerations are not significantly improved with the ride mode. This aspect could be a result of the fact that the Grand Cherokee's front suspension is already soft, having relatively low spring and damping rates. Regarding the SWS and DTL, the ride mode worsens these objective metrics suggesting that the ride mode requires more suspension travel and has worse dissipation of disturbances and road holding compared to the other modes. Further results on these maneuvers are presented in Appendix G. Results for the frequency response maneuver can also be found in Appendix G. In general, the vehicle's roll and yaw responses are improved with the sport mode, having less body roll and higher yaw rates in response to driver steering inputs. However, the percentage improvements in the objective handling metrics on this maneuver are much less than body twist and cleat. Regarding the vehicle's pitch response to driver braking inputs, Figure 48 presents the objective results from the straight braking maneuver.

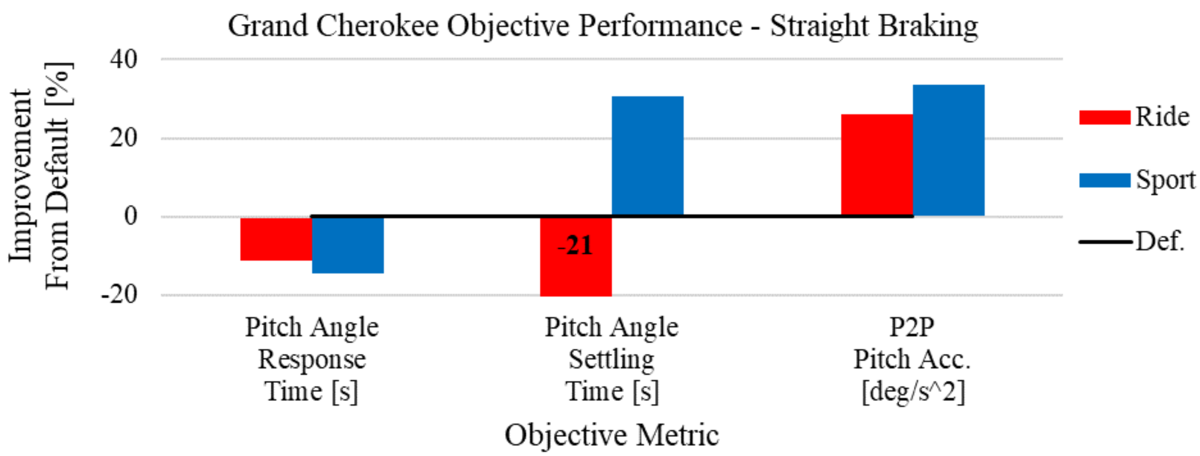


Figure 48: Grand Cherokee objective performance on straight braking maneuver

For the pitch responses of the semi-active suspension, the sport mode reduces the angle settling time and P2P acceleration the most. Note that the P2P acceleration metric is in the primary ride frequency range. The ride mode also reduces the pitch acceleration, but the pitch angle response and settling times are worsened resulting in a slower pitch response to the braking input. For both modes, the rear wheels did not lose contact with the road upon braking. In fact, this aspect was never an issue when tuning the controller for the Grand Cherokee model. Regarding the vehicle's response to aggressive steering inputs, Figure 49 presents the objective metric results for the ISO DLC maneuver. Note that the bolded zeros in Figure 49 depict a zero-percentage change in the objective metric for a given suspension mode.

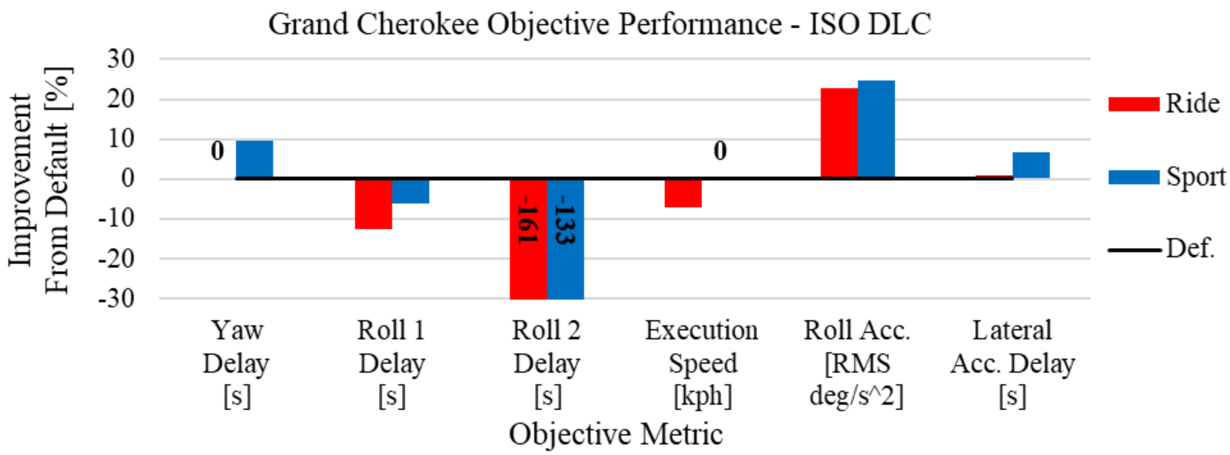


Figure 49: Grand Cherokee objective performance on ISO DLC

The most significant findings from the ISO DLC maneuver were the improvements in yaw response delay, lateral acceleration delay, and RMS roll acceleration for the sport mode. Note that the roll acceleration metric is in the primary ride frequency range. Similar to the frequency response maneuver, the roll delays (between roll angle and steering angle and roll angle and lateral acceleration) are considered insignificant (See Appendix G). In general, the findings on the ISO DLC maneuver depict a sport mode having a faster response to driver inputs as well as suppressed roll motion compared to the default mode. The ride mode also significantly suppressed the roll motion, but resulted in a slower vehicle execution speed and no improvement in the yaw response. The results for the slalom maneuver were similar to the ISO DLC objective results and have been placed in Appendix G. These results depict the objective performance for less aggressive steering inputs. Finally, the objective results from the performance track are depicted in Figure 50. Note that the vertical axis represents the percent difference in objective metrics rather than the percent improvement.

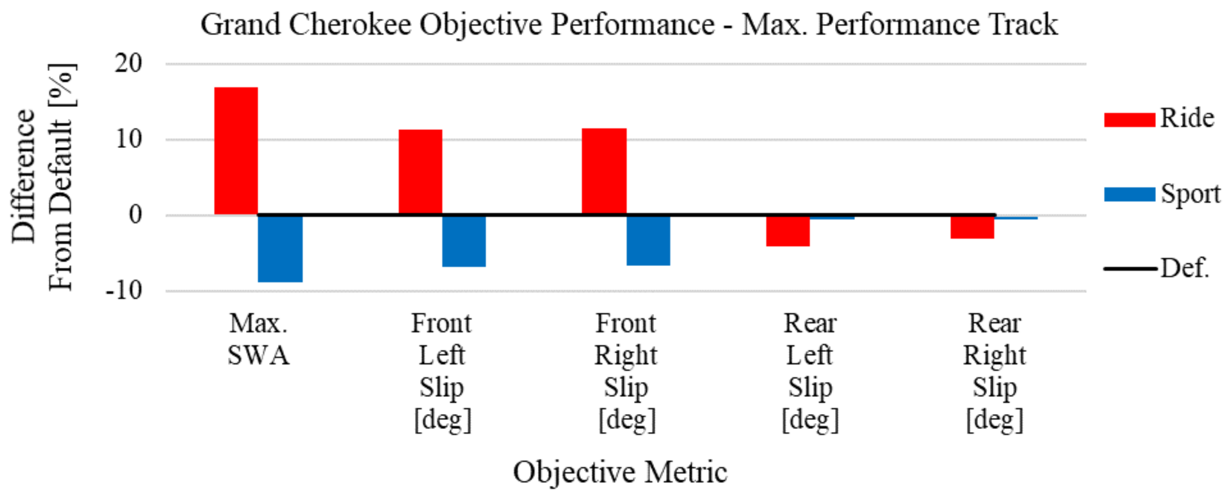


Figure 50: Grand Cherokee objective performance on performance track

The results in Figure 50 are from one of corners on the performance track on which the vehicle was simulated. This corner consisted of a decreasing radius and required more aggressive driver inputs compared to other corners on the track. However, similar results were found on other corners of the track. As a result, the vehicle dampers were significantly excited and their impact on performance could be realized. Here, the sport mode was found to have reduced front tire slip angles compared to the default mode. Where the default mode generally exhibits understeer, this reduction of front tire slip results in less understeer (or more neutral steer). Thus, the maximum SWA input is reduced for the sport mode. The opposite was found for the ride mode where the driver model had to apply a nearly 20% larger SWA compared to the default mode.

In general, the objective results indicate that the sport mode has the most significant overall improvement in handling. The vehicle roll and pitch motions are suppressed upon driver inputs, the SWS and DTL are reduced, the SWA demand for the driver on aggressive cornering is reduced, and delays for the lateral acceleration and yaw rates are reduced. These findings all result from a stiffer set of damper curves. For the ride mode, the primary and secondary ride are improved in regard to the accelerations measured at the driver seat location. The accelerations comprise of the vertical, lateral, pitch, and roll responses of the vehicle to road and driver inputs. Therefore, the ride and sport modes meet the general ride and handling targets presented in Table 7 (Section 7.1.3). The next subsection of this chapter presents the results of the same objective evaluation for the secondary vehicle model studied.

9.1.2 Controller Objective Ride and Handling – Secondary Vehicle Model

The Jeep Renegade vehicle model was studied to determine if the semi-active suspension controller could perform well within an additional vehicle class. The tuning of the controller for this vehicle model was found to be significantly more challenging than the controller settings for the primary vehicle. Please see Appendix G for objective findings for a collection of the best controller iterations for the Renegade. The objective results for the final controller settings on the body twist and cleat maneuvers are presented in Figure 51.

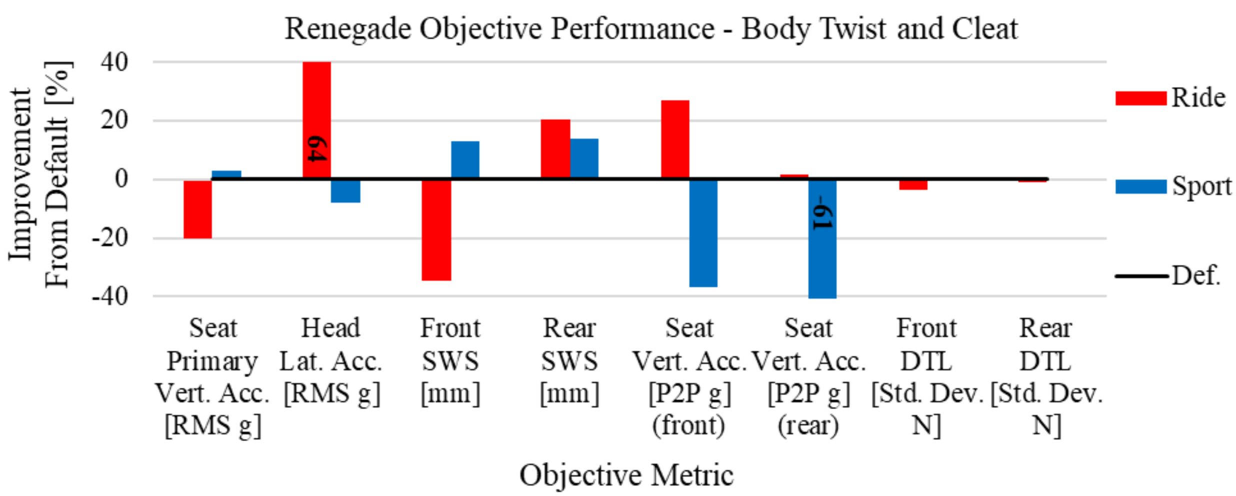


Figure 51: Renegade objective performance on body twist and cleat

In Figure 51, the objective results for the ride mode show significant improvements in the RMS driver head lateral acceleration on body twist and driver seat vertical P2P acceleration when the front axle hits the cleat. However, the driver seat RMS vertical acceleration is worsened with the ride mode. When tuning the Renegade, it was impossible to improve this metric with the Hybrid strategy while implementing different damper curves, alpha values, and hybrid gain values. Furthermore, trying to improve the driver seat accelerations lead to tradeoffs in other ride metrics on body twist and cleat maneuvers. Therefore, to retain improvements in other objective metrics, this worsening of the vertical primary ride on body twist was accepted. Even for the sport mode having the harder damper curves, the vertical primary ride could only be improved by 3%. Additional results on body twist can be found in Appendix G, where it was found that the secondary vertical ride performance on the body twist was improved by almost 20% with the ride mode. Furthermore, this can be attributed to the nearly nonexistent worsening of the DTL for the ride mode (See Figure 51), which otherwise affects the secondary ride acceleration frequencies.

Due to the increased damping rates of the rear dampers in the ride mode settings, the rear SWS on body twist is improved with the renegade and the driver seat vertical acceleration is barely improved when the rear axle hits the cleat. This metric could be improved, but it would require softer rear damping settings that were found to worsen the primary ride metrics. In general, the performance of the semi-active suspension controller on the body twist and cleat maneuvers is inferior to that with the Grand Cherokee when observing ride improvements. Regarding the frequency response maneuver, the objective results are less significant than other maneuvers. Thus, the results have been placed in Appendix G. Concerning the vehicle's pitch response, Figure 52 contains the objective results from the straight braking maneuver.

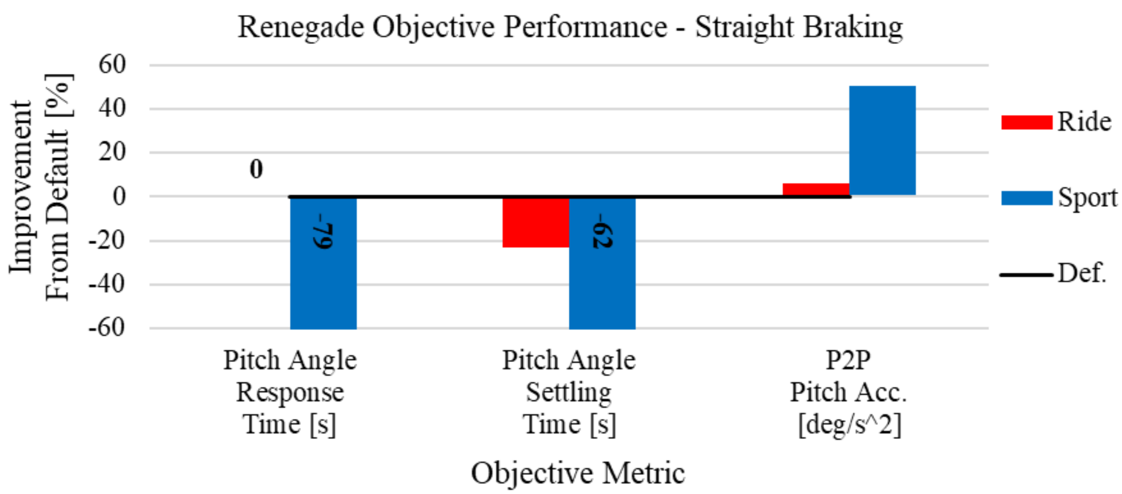


Figure 52: Renegade objective performance on straight braking

For the straight braking maneuver, the ride mode generally did not impact the pitch response of the vehicle, other than the length of the pitch angle settling time. On the other hand, the sport mode significantly suppressed the P2P pitch acceleration, while it increased the pitch angle response and settling times. In terms of handling, these increases in pitch response timing suggests a worsening of the handling. It was decided that these times should be addressed on the DiM 250 to determine if the changes are noticeable by drivers. The most significant finding was that the sport mode suppresses the pitch accelerations by 50%, providing a more rigid chassis response to driver braking inputs. Regarding aggressive steering inputs, Figure 53 presents the objective findings from the ISO DLC maneuver.

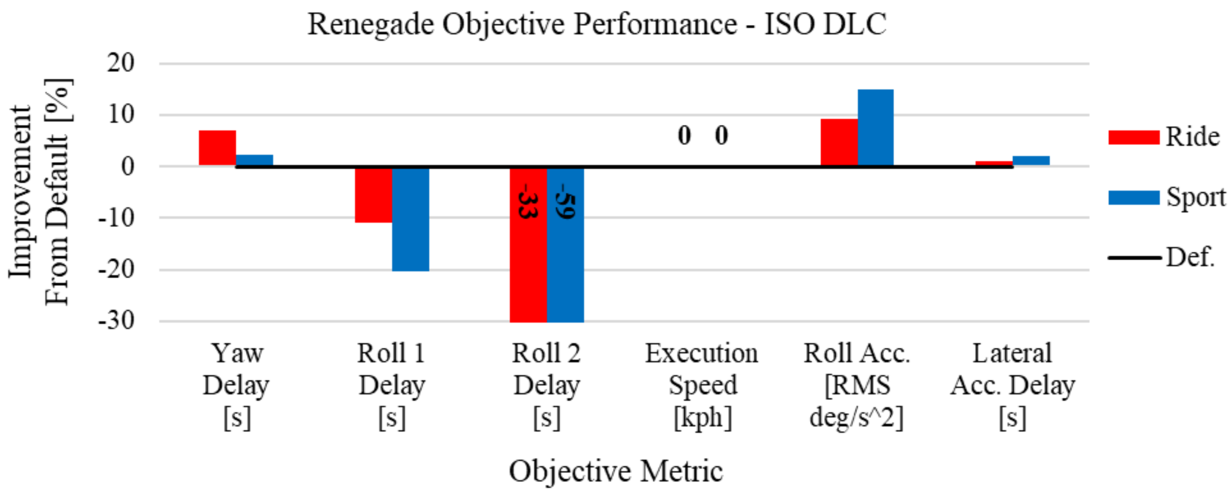


Figure 53: Renegade objective performance on ISO DLC

The objective trends on the ISO DLC for the primary and secondary vehicle models are similar. With the semi-active suspension controller performance with the secondary vehicle, the sport mode suppressed the chassis roll motion over the default mode the most. However, the ride mode provides the smallest delay in the vehicle yaw rate response. There was no significant change found in the maneuver execution speed or lateral acceleration delay the between the three suspension modes. Furthermore, the roll response delays were considered insignificant due to their extremely small magnitude, despite the results indicating that both modes significantly increase the delays. According to these findings, both of the controller modes improve some of the handling metrics, but the percent improvements over the default mode are less significant than the results with the primary vehicle model. The same objective handling metrics for the slalom maneuver are presented in Appendix G. Finally, the objective results from the performance track are presented in Figure 54. The vertical axis in this plot, the same as Figure 50, represents the percentage change in a metric, rather than the percentage improvement.

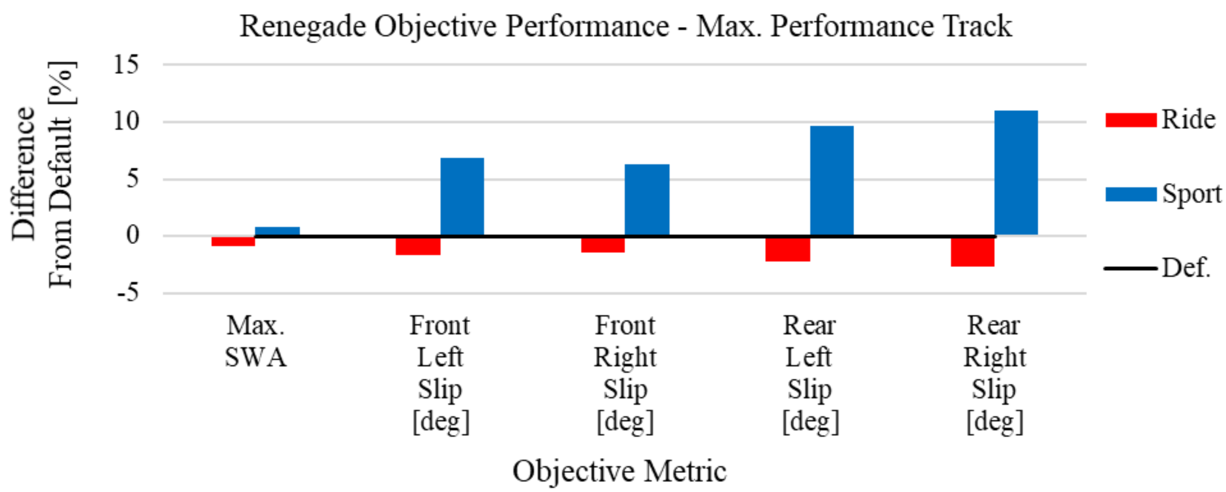


Figure 54: Renegade objective performance on performance track

The objective handling metrics depicted in Figure 54 address the vehicle steering behaviour on an aggressive corner of the performance track. Where the ride mode does not affect the vehicle steer, the sport mode increases all four tire slip angles by a range of five to ten percent. The default mode has a slight understeer behaviour where the front tires have higher slip angles than the rear. For the sport mode, the rear tire slip angles are increased slightly more than the rear, suggesting a reduction in the understeer behaviour. However, the change in SWA is insignificant for both suspension modes.

In general, the objective performance of the semi-active suspension controller for the secondary vehicle model indicate that robustness of the control strategy is satisfactory. The ride mode improves secondary vertical ride, but worsens the primary. The driver head level lateral acceleration is also improved, but the roll and pitch accelerations on the ISO DLC and straight braking maneuvers are not significantly affected. As for the sport mode, the SWS is reduced overall, the roll motion and pitch motions are suppressed the most on several maneuvers, but the steering behaviour on the performance track and several of the response delays are not improved. Therefore, the control strategies implemented in the semi-active suspension controller do result in some robustness and still meet several of the ride and handling objective targets from Section 7.1.3. The next chapter section presents the subjective findings from the completed testing on the DiM 250 to determine if the objective performance improvements lead to noticeable subjective improvements.

9.2 Final Subjective Performance

The ride and handling results of the subjective metrics from Table 8 are presented in this section. These results are from the subjective evaluation method discussed in Chapter 8, where each vehicle model with the three semi-active suspension modes is driven on the seven maneuvers using the DiM 250. The plots in the following subsections highlight the average absolute ratings for all suspension modes. This feature allows the acceptability and improvements in subjective performance to be observed simultaneously.

9.2.1 Controller Subjective Ride and Handling – Primary Vehicle Model

The first results presented correspond to the subjective performance of the semi-active suspension on the body twist and cleat maneuvers. Figure 55 contains five subjective metrics where the first two correspond to the body twist maneuver and the last three correspond to the cleat maneuver. For Figure 55 and similar, the italicized white numbers represent the standard deviation in the ratings.

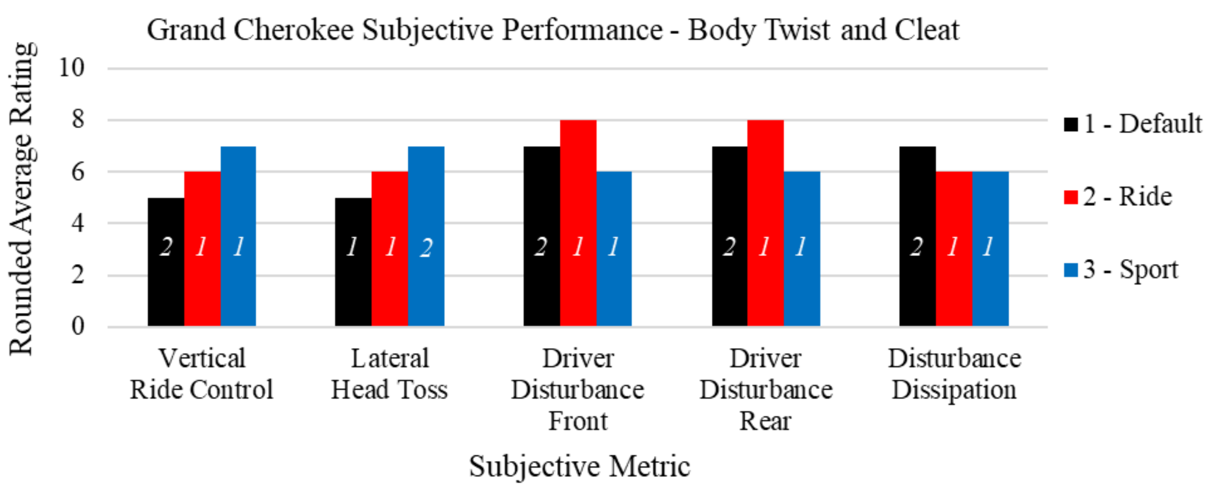


Figure 55: Grand Cherokee subjective performance on body twist and cleat maneuvers

In Figure 55 it is clear that the drivers found the ride mode to reduce the vertical accelerations (primary and secondary) compared to the default mode on both maneuvers. The head toss was also improved with the ride mode. These results match the objective evaluation findings. Each of these metrics were also rated above a value of five, meaning their performance was at least acceptable according to the subjective rating scale and drivers. Furthermore, both the ride and sport modes subjectively worsened the dissipation of road disturbances (related to DTL) on the cleat maneuver. As found in the objective evaluations, the ride mode worsened the front and rear DTL, but the sport mode did not. It is possible that the drivers rated the sport mode's disturbance dissipation as worse than the default if they focused more so on the magnitude of the disturbance instead of how long the disturbance lasted on the cleat maneuver. This aspect could be the cause of the unexpected rating since the sport mode's secondary ride performance was objectively and subjectively worse than the default mode. In the feedback section of the questionnaire, drivers noted that the ride mode had the worst dissipation, but it was not clear if it was a vibration coming from the seat or if the tire models were capturing noise from the virtual road profile.

An unexpected result was that the sport mode was rated as having the best head toss and vertical ride control. Objectively, the RMS head level lateral acceleration was the worst for the sport mode and the driver seat RMS vertical primary accelerations were inferior to the ride mode's performance, but still better than the default mode. Two of the drivers rated the sport mode's vertical ride control metric as only one point better than default, but the third rated it as four points better and resulted in a higher average rating than the ride mode. For the head toss subjective metric, the exact same instance occurred. Objectively, the sport mode is worse than the default for the head lateral acceleration metric. Further investigation into the objective data could not determine why the discrepancy in the head toss metric occurred. Other measured signals depicted the sport mode has producing larger RMS chassis roll accelerations and the same driver seat vertical and head level lateral displacements when compared to the default mode. However, one driver noted that with the highly damped sport more, he anticipated the stiffer ride and could have tried to compensate on the body twist maneuver. This compensation could affect the roll acceleration and head lateral acceleration. Regarding the subjective performance of the semi-active suspension in response to driver inputs, Figure 56 contains the driver ratings from the straight braking and frequency response maneuvers.

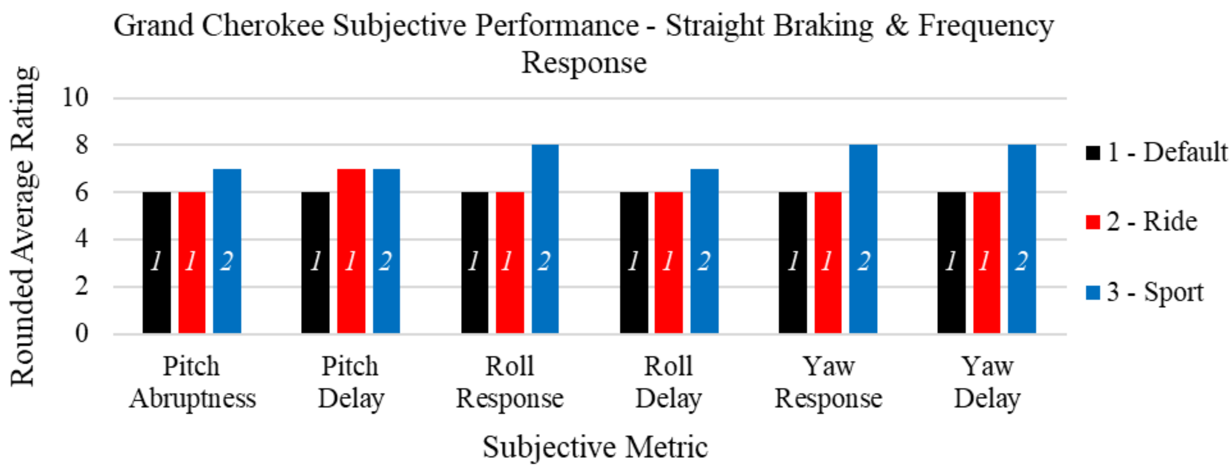


Figure 56: Grand Cherokee subjective performance on straight braking and frequency response

The first two metrics in Figure 56 correspond to the braking maneuver, whereas the last four correspond to the frequency response maneuver. The standard deviation of the ratings in Figure 56 all had a value of one point, suggesting the drivers were in good agreement. The drivers rated the sport mode has having the smallest pitch accelerations, smallest roll motion, and quickest yaw response. These subjective findings also match the objective findings from the same maneuvers. However, the pitch delay metric for the ride mode was rated as better than the default mode, while objectively this delay was longer. It is possible that the drivers were focusing more on the improvement of the pitch accelerations than how long it took for the vehicle to pitch upon braking since the vehicle could appear to stop moving sooner than the default mode. Additionally, the roll delay on the frequency response for the sport mode was rated better than the default, despite the objective worsening presented in Section 9.1. This result could also come from the fact that the drivers were focusing on the magnitude of the vehicle motion rather than the delay between the response and driver inputs, since the delays are generally extremely small. Not to mention, the objective metrics cover specific frequencies, whereas the drivers were focusing on the yaw and roll responses overall. Finally, the drivers noted that sport mode dampened the roll motion significantly near end of the frequency response maneuver where the steering frequencies are closer to 2Hz. The results of the next maneuver are presented in Figure 57, consisting of the subjective ratings for the ISO DLC.

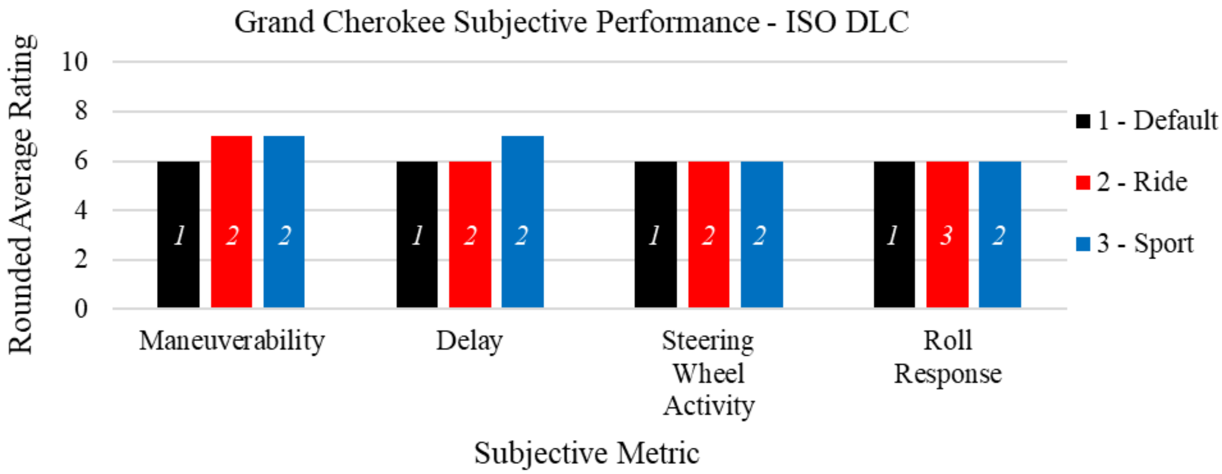


Figure 57: Grand Cherokee subjective performance on ISO DLC

In general, the subjective ratings from the ISO DLC in Figure 57 depict a sport mode having a batter capacity to maneuver through the cones and a faster responding vehicle. For the ride mode, it was perceived as having insignificant differences in handling compared to the default mode, except for the improvement in maneuverability. Objectively, both suspension modes suppressed the roll accelerations by more than 20% where the sport mode also improved the lateral acceleration and yaw angle delays. This aspect could be the reason why both modes were rated as having better maneuverability, but only the sport mode rated as having a shorter delay in the vehicle response. A difference between the implementation of the ISO DLC in the offline and simulator environments is that each driver can apply different acceleration, braking, and steering inputs on the simulator. In the offline co-simulations, the driver model applied essentially the same acceleration and SWA commands for all modes. Since the vehicle roll motion is dependent on the SWA input, this could have impacted the vehicle subjective ratings. For instance, if the driver applied a larger SWA input for the sport mode, than the chassis roll motion would be increased. Objectively, for the same SWA input the sport mode has a lower RMS acceleration. Thus, the sport mode's perceived roll motion could be close to the default's roll motion if the drivers unknowingly applied slightly larger SWAs when driving with the sport mode. Since the ISO DLC is a closed-loop maneuver, this phenomenon cannot always be avoided when using several human drivers.

In addition to the ratings presented in Figure 57, the driver found that the ISO DLC could be completed at the same speed for the default and sport setting, but when driving with the ride mode the speed was approximately 5kph slower. Furthermore, the standard deviation in ratings was found to be two points for most ratings, and three for ride mode's roll response rating. This result suggests the drivers were in less agreement on this maneuver than the first four. Better agreement was found on the slalom maneuver subjective findings where more standard deviations were at a value of one. See appendix G for the subjective findings on the slalom, having similar ratings as the ISO DLC. The final set of subjective results for the primary vehicle model are presented in Figure 58 for the maximum performance track.

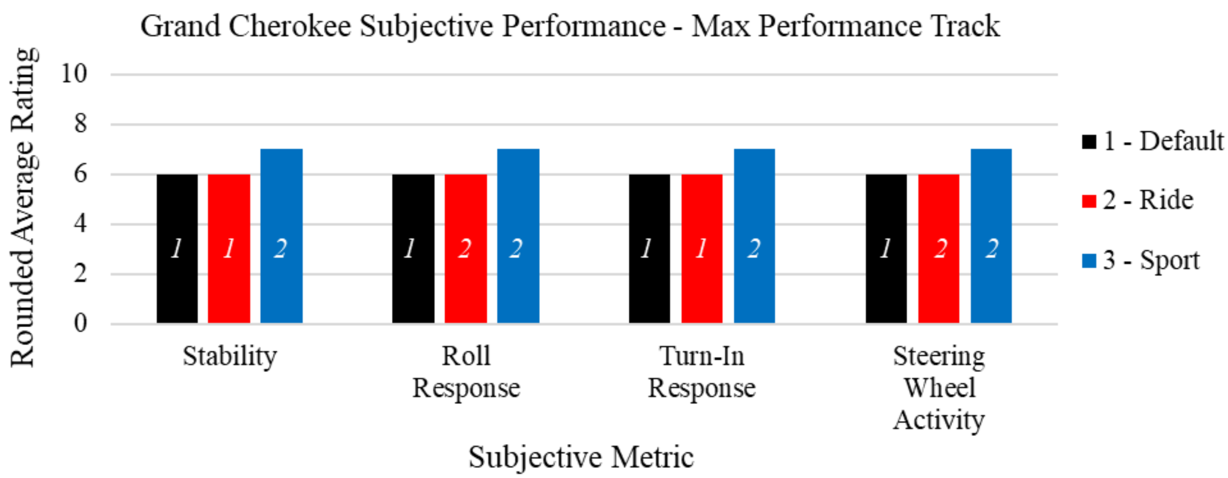


Figure 58: Grand Cherokee subjective performance on performance track

In Figure 58, the ride and default modes are rated as having the same handling performance. The ratings for the default mode all had a standard deviation of one, whereas the ride mode ratings had a standard deviation of two for the roll response and steering wheel activity metrics. This result suggests that the drivers were in more agreement for the default mode performance than the ride mode. For the sport mode, all metric averaged ratings had a standard deviation of two since one driver gave the same rating for the sport and default modes for all metrics. Otherwise, the other two drivers perceived the sport mode to have the best stability (rear wheel grip), roll response (vehicle roll rate and acceleration), turn-in response (response to initial steering input when turning), and steering wheel activity (total SWA when cornering). Objectively, the only significant difference between the sport and default modes was the reduction of understeer behaviour. This behaviour could account for the improved steering wheel activity rating for the sport mode. Additionally, two drivers noted that the sport mode felt sporty, promoted confidence in the handling, and felt predictable. This aspect was not captured objectively since the offline driver model cannot perceive such aspects as human drivers can. Nevertheless, the sport mode clearly is perceived as having better handling than the default, whereas the ride mode was perceived as having the same handling behaviour as the default.

In general, the recorded driver ratings always remained within the five-to-eight-point range. In many cases, a one-point difference between the default and another mode was noticed by the drivers. According to the SAE adapted subjective rating scale, a one-point difference between ratings is considered a major difference noticed by customers. The final results with the Grand Cherokee model suggested many one-point improvements in ride for the ride mode and handling for the sport mode. Thus, the subjective evaluation method was able to capture noticeable differences in ride and handling performance of the semi-active suspension modelled. Many of the subjective differences perceived by the drivers also followed similar trends found during the objective evaluation. The few discrepancies found between the objective and subjective performances could have resulted from disagreement between the drivers as suggested by higher standard deviations in the some of the averaged ratings. Moreover, the difference in driving behaviour between the drivers could also have impacted the differences in ratings between the suspension modes. Despite these differences, the subjective evaluation method developed and implemented was able to capture the improvements in ride and handling of a vehicle semi-active suspension. The next subsection presents the subjective results of implemented this method within another vehicle class.

9.2.2 Controller Subjective Ride and Handling – Secondary Vehicle Model

The first results presented correspond to the subjective performance of the semi-active suspension on the body twist and cleat maneuvers. Figure 59 contains five subjective metrics where the first two correspond to the body twist maneuver and the last three correspond to the cleat maneuver. As with the previous subjective result plots, the white italicized numbers inside each rating represents its standard deviation. Recall from the objective results for the Renegade controller where the vertical primary ride on body twist was not improved with the ride mode. Moreover, the sport mode had a significant tradeoff in secondary ride on the cleat maneuver. Finally, for all Renegade damping settings (semi-active suspension controller or from the correlation study) the vertical tire forces of the front and rear wheels drop to zero momentarily on the cleat, indicating a loss of road contact with this stiffer vehicle model.

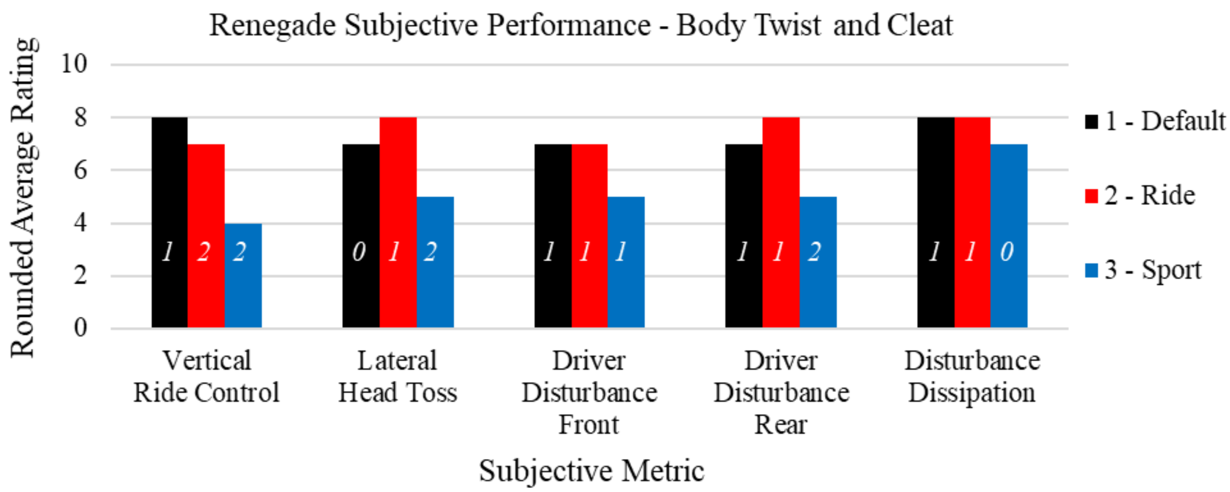


Figure 59: Renegade subjective performance on body twist and cleat maneuvers

For the first two metrics in Figure 59 on body twist, the ride mode has inferior vertical ride control and superior head toss compared to the default. These match the offline results found with the controller. For the remaining four metrics on cleat, the ride mode only improves the severity of the impact felt by the driver when the rear axle hits the cleat. As a result, the Renegade ride mode improves only some of the ride metrics compared to the model's passive suspension (default mode). Furthermore, these ride improvements are rated as one point higher than the default mode, indicating a major difference in performance. For the sport mode, there is an extremely significant tradeoff in secondary ride on cleat with the driver disturbance metrics being two points lower than the default mode. This result also matches the offline results predicting a tradeoff in secondary ride with the sport mode. Additionally, all drivers mentioned the ride was highly degraded and this is likely why the vertical ride control metric in Figure 59 was rated as unacceptable for the sport mode. As with the primary vehicle model, the subjective evaluation method has proven to capture a majority of the differences in ride performance on the body twist and cleat maneuvers. Next, Figure 60 presents the subjective handling results from the straight braking and frequency response maneuvers.

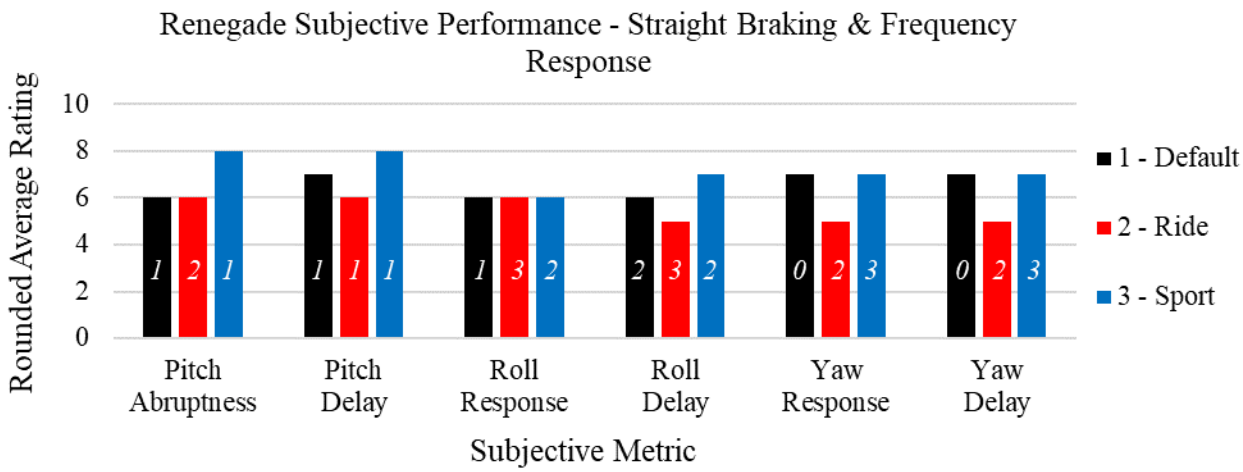


Figure 60: Renegade subjective performance on straight braking and frequency response

In Figure 60, the first two metrics from the braking maneuver indicate that the sport mode suppresses the pitch accelerations significantly over the default mode. This result was also objectively identified in the offline objective results from Section 9.1.2. For the sport mode's pitch response, the objective results indicated that both the pitch angle response and settling times were more than 60% longer than the default. However, the subjective pitch delay metric was rated as two points higher than default. One of the drivers commented that they preferred a slower pitch response upon braking, despite the description of the target column in the questionnaire. This preference could also have been the case with the primary vehicle model as the same discrepancy between the objective and subjective delay metric findings occurred.

For the remaining four metrics for the frequency response maneuver in Figure 60, the ride mode is generally rated as having poor handling compared to the default mode. Comments from the driver feedback section noted that the ride mode felt "sloppy" and "lazy" with worse handling, but better ride than the default mode. This feedback indicates the driver's acceptance for the ride mode as an acceptable ride setting, but the tradeoff in handling could be improved in future iterations. For the sport mode's handling performance, the drivers found it's handling similar to the default. Note that the standard deviations in the ratings are generally high for the frequency response maneuver, indicating some disagreement between the driver perceptions. The results for another handling maneuver are presented in Figure 61 for the ISO DLC.

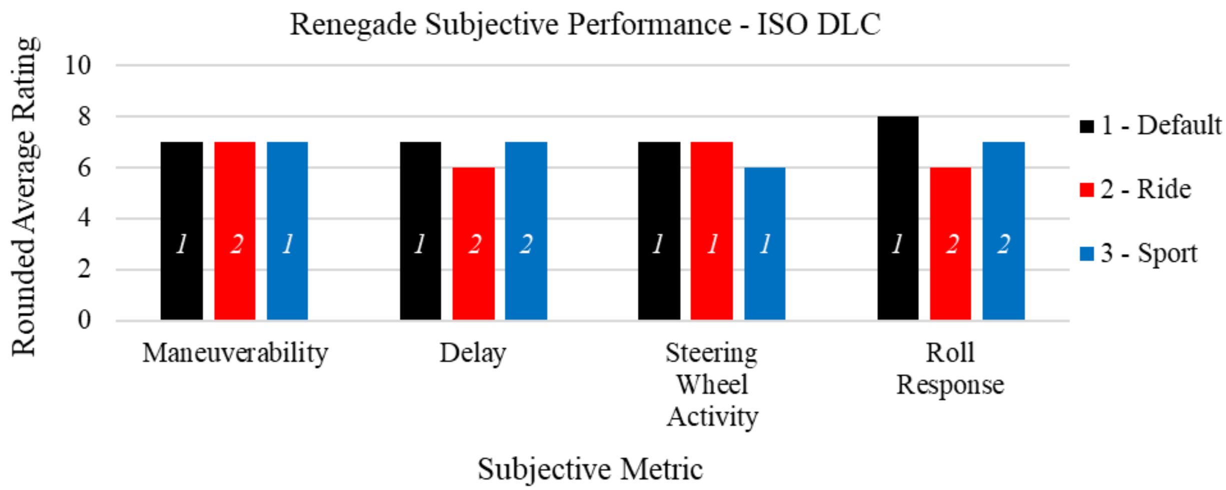


Figure 61: Renegade subjective performance on ISO DLC

The most significant finding from the ISO DLC maneuver was the degradation in roll response metric for the ride mode, despite the small objective improvement in the RMS roll acceleration metric. As mentioned before, the drivers could have applied different steering and accelerating inputs between the three modes compared to the offline simulations. This difference would certainly affect the handling performance. Moreover, the drivers commented that the vehicle model felt like the jounce bumpers were being hit with the ride mode, likely as a result of the increased SWS. Similar to the frequency response maneuver, the drivers felt as though the ride mode was "lazy", which could also be indicated by the objectively longer roll delay metrics which match the reduction in delay subjective metric in Figure 61. Regardless, the handling performance of the ride mode was still rated above five points, thus deemed acceptable for customers. Recall from Section 9.1.2 that the objective results from the Renegade controller did not have as significant differences in performance compared to the primary, larger SUV vehicle model. This result is likely why the difference in handling between the three Renegade suspension modes are smaller than that of the Grand Cherokee's controller. Similar findings can be found in Appendix H for the slalom maneuver. The subjective results for the secondary vehicle model so far have indicated satisfactory robustness of the modelled semi-active suspension controller with a smaller SUV model, compared to the primary vehicle model. The subjective handling results for the max performance track are presented next in Figure 62.

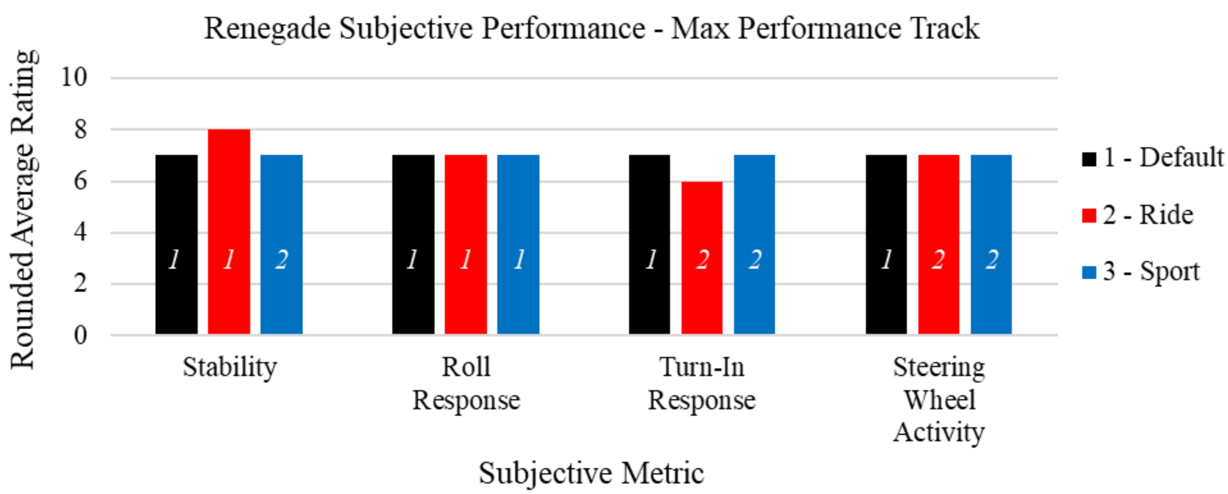


Figure 62: Renegade subjective performance on performance track

The sport mode's subjective handling performance on the 3D-scanned performance track was not improved as indicated by the ratings. Objectively, all four tire lateral slip angles were increased with the sport mode whereas the driver SWA demand was unchanged as predicted by the CRT max performance event algorithm. Note that the ratings are an average between the three drivers. In fact, two of the drivers rated two of the subjective metrics higher for the sport mode, but the third driver rated the sport mode two points lower. When rounding to the resolution of the rating scale, the sport mode's subjective handling improvement was diminished. This phenomenon can also be implied from the higher standard deviations in the sport and ride mode ratings in Figure 62. For the ride mode, the stability was rated highest. Objectively, the rear tire lateral slip angles were reduced with the ride mode, and this suggests the improvement in the stability subjective rating since it corresponds to the ability for the rear tires to maintain grip when cornering. However, the ride mode's turn-in response was rated lower which matches the lower delay ratings on the ISO DLC and the ratings for several handling metrics on the frequency response maneuver. Objectively, the ride mode induces longer delays in the yaw rate and roll angle at 1Hz steering inputs and more suspension travel in several maneuvers. This result can indicate the "lazy" and "sloppy" descriptors used by the drivers for this suspension mode. As with other handling maneuvers, the overall handling of the ride mode was still rated as acceptable, suggesting that the improvements in ride performance are worth the handling tradeoff.

The overall subjective performance of the semi-active suspension controller with the sporty, subcompact SUV (secondary vehicle model) indicate that the controller is not as applicable to this vehicle class. During the tuning of the controller for the subcompact vehicle model, there were more ride and handling tradeoffs discovered compared to the midsize SUV class (primary vehicle model). As a result, the semi-active suspension controller developed in Chapter 6 works better with larger SUVs. Despite the difference in overall subjective performance with the two SUV models, the implementation of the subjective evaluation method was able to identify several of the ride and handling tradeoffs, as indicated by the driver ratings. Thus, the subjective evaluation method developed could be applied to the two SUV vehicle classes. In the case of other vehicle classes, such as full-size SUVs or heavier, larger vehicles with significant ride heights and centers of gravity (such as pickup trucks which also utilize dampers), it is therefore possible that the subjective evaluation method would be applicable. For vehicle classes with less significant changes in ride and handling when implementing a similar semi-active suspension controller, the resolution of the scale could be increased to compensate. If a change in subjective ratings is discovered with the higher rating scale resolution, note that the magnitude of the difference noticed by customers will be smaller than those identified in this research according to the definition of the SAE adapted scale. The next subsection presents the results from the correlation study on altering damping rates and identifying the relationship between objective and subjective metrics.

9.3 Correlation Study Results

The results of the linear correlation study conducted when changing the vehicle damper curves from soft to hard settings are presented herein. To explicitly view the individual objective metric and subjective ratings results, see Appendix H. Table 10 contains the number and letter assignment for the objective and subjective metrics, respectively. These numbers and letters identify which objective-subjective metric pair correspond to each correlation coefficient in the final results. The naming of the metrics applies to both vehicle models.

Table 10: Metric Ordering Convention for Correlation Study

Maneuver	Objective Metric	Number	Subjective Metric	Letter
ISO DLC	Yaw Delay	1	Maneuverability	A
	Roll 1 Delay	2	Delay	B (1/2/3/4)
	Roll 2 Delay	3	SW Activity	C
	Lat. Acc. Delay	4	Roll Response	D (5)
	RMS Roll Acc.	5	-	-
Cleat	P2P Seat Acc. (front)	6	Driver Disturbance – Front	E (6)

Cleat (cont.)	P2P Seat Acc. (rear)	7	Driver Disturbance - Rear	F (7)
	Front Diss. Time	8	Disturbance Dissipation	G (8/9)
	Rear Diss. Time	9	-	-
	Front DTL	10	-	-
	Rear DTL	11	-	-
Body Twist	RMS Seat Vert. Acc.	12	Vertical Ride Control	H (12)
	RMS Head Lat. Acc.	13	Head Toss	I (13)
	Front SWS	14	-	-
	Rear SWS	15	-	-

The ordering of the metrics in Table 10 are arbitrary and the relative position between the objective and subjective metrics is not relevant. However, some of the letters in the rightmost column of Table 10 contain one or more numbers in parentheses. These numbers correspond to the objective metrics that address a similar vehicle objective aspect as the subjective metric with the letter before the parentheses. These metric connections were highlighted to determine if the subjective metrics that were developed in this research have any correlation with certain objective metrics that were chosen. The next subsection presents the results of the linear correlation computations.

9.3.1 Linear Correlation Results

Before reviewing the major correlation findings, a few notes should be addressed regarding the findings from the individual objective and subjective evaluations, whose results are in Appendix H. This discussion is focused on the results from the primary vehicle model correlation study. For all objective metrics on the body twist maneuver, the performance ranges from 20% worse to just less than 20% better than the default mode, depending on the objective metric and damper setting. Between damper settings, the objective metrics change by 5% to 10%, much less of a change compared to the differences in the sport and ride mode performances in the previous objective evaluations. However, the subjective ratings span a range from two to seven points. Thus, the DiM 250 was exceptional at capturing the small changes in damping settings. For the cleat maneuver, the P2P vertical acceleration objective metrics differed by 20% better to nearly 50% worse as the damping settings increased. Moreover, the DTL metrics changed insignificantly, and the front and rear tire loading dissipation times varied from more than 25% worse to nearly 50% better as the damping increased. Subjectively, the drivers did not perceive the changes in performance nearly as significantly as the body twist maneuver. Overall, the subjective ratings varied from five to six and a half points. An important point to make is that the road disturbances from the cleat maneuver are significantly smaller in amplitude and higher in frequency compared to the body twist maneuver, so it was expected that the ratings would differ less on this maneuver. Regardless, the effect of the altering the damping rates was noticeable on the DiM 250.

Finally, the objective handling findings from the ISO DLC were the smallest overall. None of the objective delay metrics changed by more than 10% from the default mode's performance. However, the RMS roll acceleration was improved by nearly 20% with the hardest setting. Subjectively, the relatively small objective handling effects from differing damping rates were noticeable on the DiM 250. As the damping increased, all subjective metrics increased from a rating of four to a rating of seven and a half points. As before, the DiM 250 was proven to be an exceptional tool for capturing the objectively small changes in handling to the primary vehicle model. Therefore, the sensitivity of the DiM 250 was high enough to notice at least 20% changes in the slopes of damping characteristic curves.

Now that once the general objective and subjective ride and handling performance effects are known, the significant of their correlation can be addressed. The results of the linear correlation study with the primary vehicle model are presented in Figure 63. The colours in the figure are representative of different ranges in the correlation coefficient as adopted from literature. The correlation is between the magnitudes of the objective metrics and the absolute values of the subjective ratings.

"R" Coefficient		OBJECTIVE METRICS														
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
SUBJECTIVE METRICS	A	-0.65	0.72	0.72	-0.73	-0.70	0.99	0.98	-0.93	-0.37	-0.96	-0.56	-0.94	0.89	-0.97	-0.96
	B	-0.69	0.74	0.75	-0.78	-0.75	0.99	0.99	-0.90	-0.32	-0.92	-0.47	-0.91	0.94	-0.95	-0.93
	C	-0.74	0.75	0.76	-0.80	-0.77	0.98	0.97	-0.95	-0.49	-0.97	-0.59	-0.96	0.89	-0.98	-0.98
	D	-0.80	0.81	0.83	-0.86	-0.84	0.96	0.96	-0.85	-0.36	-0.89	-0.42	-0.87	0.94	-0.92	-0.90
	E	0.48	-0.46	-0.49	0.56	0.54	-0.79	-0.80	0.60	-0.15	0.65	0.13	0.63	-0.80	0.72	0.66
	F	0.16	-0.21	-0.23	0.25	0.25	-0.49	-0.50	0.19	-0.57	0.24	-0.27	0.22	-0.59	0.32	0.25
	G	-0.86	0.86	0.87	-0.91	-0.88	0.87	0.87	-0.81	-0.53	-0.87	-0.55	-0.87	0.81	-0.89	-0.88
	H	-0.72	0.68	0.69	-0.75	-0.72	0.86	0.86	-0.95	-0.60	-0.95	-0.71	-0.96	0.72	-0.95	-0.96
	I	-0.73	0.70	0.71	-0.77	-0.73	0.90	0.90	-0.96	-0.53	-0.96	-0.68	-0.97	0.77	-0.97	-0.97

- $R \geq 0.85$
- $0.7 < R \leq 0.85$
- $0.5 < R \leq 0.7$
- $R < 0.5$

Figure 63: Linear correlation results with the Grand Cherokee

From the results in Figure 63, it is clear there are many correlations that exist between the objective and subjective metrics studied. All objective performance trends were linear except for the rear dissipation time metric on the cleat maneuver. Upon inspection of the column for the ninth objective metric in Figure 63 (corresponding to the rear dissipation time metric), a poor correlation coefficient was found with all subjective metrics. Similarly, for the front and rear driver disturbance subjective metrics on the cleat maneuver, generally lower correlation coefficient values were discovered. This result is likely due to the fact that the subjective ratings for these two subjective metrics were the least affected when changing the damping settings (See Appendix H). Furthermore, the eleventh objective metric in Figure 63 was found to have generally lower correlation coefficients with all subjective metrics. Recall that this objective metric (rear DTL) was not affected by increasing or decreasing the damping rates. As a result, nearly all of the weak correlation results are local to the cleat maneuver, having the least significant changes in ratings.

Besides the few weak correlation results, a large number of objective-subjective metric pairs were strongly correlated. Table 11 contains the strongest correlation results with the corresponding metric pairs. Note that the strong correlation results for the highlighted metric pairs from Table 10 are also included at the top in Table 11 (with an asterisk next to the R-value).

Table 11: Significant Correlation Results Found with the Primary Vehicle Model

Objective Metric(s)	Subjective Metric(s)	R-Value(s)	Increasing Damping Rates
12. RMS Seat Vert. Acc.	**H**. Vertical Ride Control	-0.96*	As 12 decreases, H improves
5. RMS Roll Acc.	**D**. Roll Response	-0.84*	As 5 decreases, D improves
8. Front Diss. Time	**G**. Disturbance Dissipation	-0.81*	As 8 decreases, G improves
6. P2P Seat Acc. (front)	**E**. Driver Disturbance – Front	-0.79*	As 6 increases, E worsens
4. Lateral Acc. Delay	**B**. Delay	-0.78*	As 4 decreases, B improves
12/14/15	**I**. Head Toss	-0.97	As 12/14/15 decreases, I improves
14/15	**H**. Vertical Ride Control	≤-0.95	As 14/15 decreases, H improves
14/15	**D**. Roll Response	-0.92/0.9	As 14/15 decreases, D improves
8/10/14/15	**A**. Maneuverability, **B**. Delay, and **C**. SW Activity	≤-0.91	As 8/10/14/15 decreases, A/B/C improves
5/14/15	**D**. Roll Response	≤-0.84	As 5/14/15 decreases, D improves
8/10/14/15	**G**. Disturbance Dissipation	≤-0.81	As 8/10/14/15 decreases, G improves
1/4	**C**. SW Activity	-0.74/-0.8	As 1/4 decreases, C improves
14. Front SWS	**E**. Driver Disturbance – Front	0.72	As 14 decreases, E worsens

5. RMS Roll Acc.	**I**. Head Toss	-0.73	As 5 decreases, I improves
5. RMS Roll Acc.	**H**. Vertical Ride Control	-0.72	As 5 decreases, H improves

All of the correlation results in Table 11 are for correlations with an absolute R-value above 0.7 since this value suggests a strong correlation between the two studied metrics. In the rightmost column of Table 11, the relationship between the metrics in a correlation pair is presented. OEMs can use these correlation trends when tuning the damping rates of their passive dampers and semi-active dampers which switch between discrete damping curves or damping rates (Switchable Damper Curves strategy). In other words, the correlation results are valid for similar, relative changes to the amount of force a damper produces for a given damper velocity. These correlation findings should not be strongly considered for other types of semi-active suspension since some of the correlations do not match the subjective results when the Hybrid strategy (ride mode) was evaluated. For instance, the ride mode of the semi-active suspension controller in this research increases the front and rear SWS (objective metrics 14 and 15) and improves the vertical primary ride (subjective metric H). This result is the opposite of what the correlation study findings in Figure 63 would predict. Thus, the results from this correlation study should not be extrapolated for semi-active suspension control strategies other than the Switchable Damper Curves strategy. A correlation study of tuning the Hybrid strategy would involve observing the impact of several interdependent control parameters (see Appendix G for some examples of tuning the Hybrid strategy with the Renegade). Such a study could be an entire project on its own. Therefore, the correlation study herein was focused on correlating objective and subjective metrics for one type of semi-active suspension strategy, as well as many passive damper settings.

In some cases of strong correlations found in Figure 63, it was not practical to consider such correlation since the objective-subjective pair are used to evaluate significantly different vehicle aspects. As one example, the pair of the RMS driver seat primary vertical acceleration objective metric on body twist and steering wheel activity subjective metric on ISO DLC had a strong correlation (absolute R-value at 0.96), despite the fact that both metrics are studied on significantly different maneuvers. The RMS driver seat acceleration metric is for evaluating primary ride on a rough road profile whereas the steering wheel activity evaluates the driver steering demand when executing an obstacle avoidance maneuver. Although both are improved with higher damping rates according to the objective and subjective evaluation results, OEMs typically would not tune the vertical acceleration objective metric in hope to reduce the steering demand on a handling maneuver. Instead, an OEM would hope to improve the subjective ride performance when tuning the objective ride offline. However, the individual trends of the two separate metrics identified could be used by OEMs as they indicate how the individual metrics are affected by the damping rates. Thus, impractical correlation results such as this example were not included in Table 11. The final subsection of Chapter 9 presents a brief discussion of how the results of the correlation study can be used by OEMs. See Appendix H for the results of the same correlation study with the secondary vehicle model.

9.3.2 Objective Metrics for Predicting Subjective Performance

At the end of Chapter 2, theory regarding linear regression was presented. Linear regression can be completed in the case a strong linear correlation is discovered between two variables. For this project, the strong linear correlation trends can be used as a predictive tool for determining subjective ride and handling on the DiM 250, when tuning the damper curves of a switchable semi-active control strategy. While implementing damper curves settings offline, the subjective performance changes can be predicted, further reducing the development time of suspension tuning. Figure 64 is an example of a linear regression with objective metric "12" and subjective metric "H" from Table 10. Both of these metrics are evaluated on the body twist maneuver.

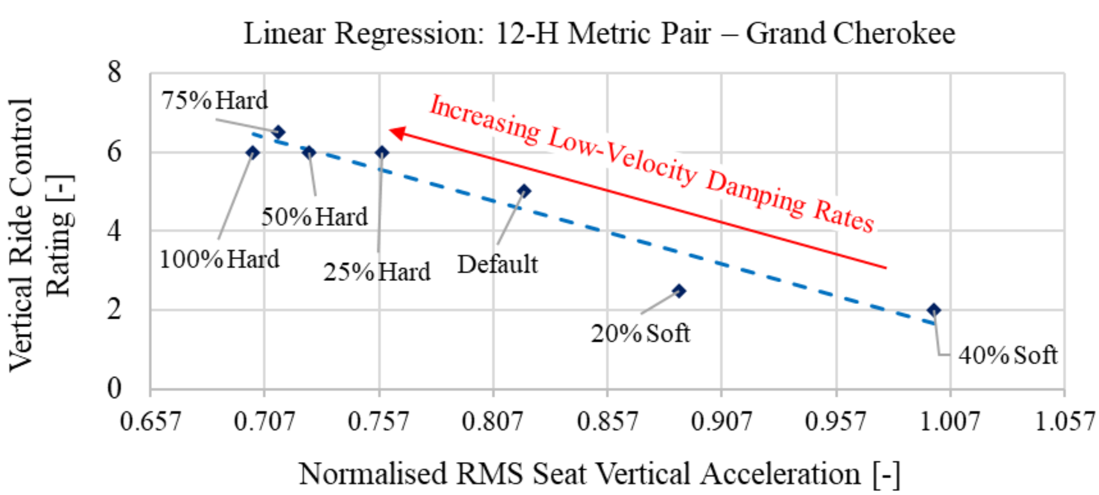


Figure 64: Linear regression for one of the correlation study findings

The results of the linear regression in Figure 64 present a powerful tool for tuning damper characteristics. The dashed line in the figure represents the regression line as found using equations (38) through (40). If an intermediate change in the low-velocity damping rate is made to the vehicle dampers, then the regression line can be used to map the recorded objective metric to a driver rating for the subjective metric. Finally, the mean-square-error and average absolute error in the ratings for the regression line in Figure 64 has an approximate value of 0.24 and 0.4, respectively. This result suggests acceptable accuracy for the subjective rating prediction. See Appendix H for similar results with the study on the secondary vehicle model.

This concludes the results and discussion chapter of the project. The next and final chapter concludes the research with comments on satisfying the project goals, effectiveness of the subjective evaluation method, and research limitations. Recommendations on future work are also presented.

CHAPTER 10. Conclusions & Recommendations

This final chapter returns to the novelty of the research and addresses the satisfaction of the objectives and sub-objectives stated in Chapter 1. The next chapter subsection highlights how the project goals were met and how the subjective evaluation method for the semi-active suspension was effective at capturing ride and handling improvements. The final chapter subsection expands on the findings of the research to set a path for future work related to this project.

10.1 Concluding Remarks

10.1.1 Meeting the Goals of the Project

Referring to the project goals and objectives listed in Section 1.5.2, the results of the evaluations in Chapter 9 have shown that these goals have all been met. Based on preliminary studies for determining the effect that vehicle dampers have on ride and handling performance, a comprehensive set of maneuvers was selected for evaluating semi-active suspension. A Simulink semi-active suspension controller was developed and tuned as presented in Sections 6.3 and 6.4. This controller implemented two well-known control strategies used by OEMs. The objective ride and handling performance of the controller was then evaluated and proven to improve many ride and handling metrics compared to a passive suspension (See Section 9.1.1), for a validated midsize SUV vehicle model. The robustness of the controller was objectively and subjectively tested with a secondary vehicle model in a different vehicle class. The results in Section 9.1.2 prove that the controller also showed acceptable performance in a subcompact SUV class.

The primary novelty of the research relates to the subjective evaluation method for evaluating a semi-active suspension. The developed method consisted of expert drivers from an OEM driving a state-of-the-art dynamic driving simulator over the set of maneuvers defined in Section 5.2. The drivers drove the validated midsize SUV models through each maneuver, comparing the subjective ride and handling performance of the virtual passive suspension with that of the semi-active suspension model. This DIL environment provided a timely manner to evaluate and tune multiple settings of the semi-active suspension controller. A questionnaire with an SAE subjective rating scale was used to record the driver ratings. The questionnaire contains a list of subjective metrics tailored for evaluating ride and handling of semi-active suspensions. The results of the subjective evaluation proved that the dynamic driving simulator could accurately replicate small and large percentage changes in ride and handling performance between the passive and semi-active suspension settings. For a majority of the subjective metrics, the drivers noticed one-to-two-point differences in the ratings, which suggest “major” performance improvements according to the SAE rating scale. Therefore, the subjective evaluation method provided a timely and cost-saving approach to evaluate the ride and handling of vehicle semi-active suspension.

The last stage of the research involved a linear correlation study aimed at providing two key advantages. The first was related to identifying which objective ride and handling metrics had strong correlations with subjective metrics, so that when tuning the dampers of a certain semi-active suspension control strategy (and passive dampers), the subjective performance on the dynamic simulator could be predicted with an acceptable level of accuracy. As a result, this correlation study provides a second opportunity to decrease the time required for subjectively evaluating intermediate damping settings. This advantage permits suspension designers and engineers to accurately evaluate and eliminate unfeasible damping settings before having to subjectively evaluate them. The second advantage of the correlation study identified the simulator sensitivity to relatively small percentage changes in damping rates of the vehicle suspension. The subjective results of the correlation study determined that percentage changes in low-velocity damping rates by at least 20% are significantly noticeable on the DiM 250. Moreover, these damping rate alterations only affected some objective metrics as much as 5-10% while the drivers still subjectively felt significant differences in correlated subjective metrics.

In conclusion, the overall goal and sub-objectives of the project were met according to their definition. This research provides a comprehensive objective and subjective study on virtual semi-active suspension, including the use of state-of the-art technology. Future research will increase in this area as dynamic driving simulators become more globally popular. Users of such simulators can benefit from this novel project, whether they adopt or adapt the methods to their research needs.

10.1.2 Effectiveness of the Subjective Evaluation Method

The findings of the subjective evaluations are credible as experts in the field of vehicle dynamics as well as ride and handling evaluations participated as drivers. As discovered in the preliminary studies of the research, dampers generally have a small impact on vehicle handling and have a more significant impact on vehicle ride. Despite this fact, the drivers noticed these impacts on ride and handling and the subjective results generally matched the objective trends found offline. In the literature review, there were more complex semi-active suspension strategies that have the potential to have more significant impacts on ride and handling compared to the control strategies evaluated herein. Due to the strong effectiveness of the developed subjective evaluation method, it is probable that the potential performance improvements of the more complex semi-active control strategies mentioned in Section 3.3 would be captured on the dynamic simulator. Finally, unprecedented vehicle behaviour on maneuvers not considered in the evaluations should not occur since a set of maneuvers concerning most aspects of the vehicle's directional responses was considered in this project.

The subjective evaluation method presented in Chapter 8 contains several features which made it effective at evaluating semi-active suspension. Since an absolute rating scale was implemented, the relative performance improvements and the absolute acceptability of the semi-active suspension ride and handling was captured. Secondly, a feedback section was incorporated to capture additional subjective perceptions of the ride and handling that could not be captured by a ratings scale, such as the characteristic feel (i.e., sporty, boring, stiff, confident, etc.). Thirdly, the structure of the questionnaire provides a simple layout that can be expanded for additional semi-active suspension settings. The concise display of information for each maneuver, the ratings scale, and the subjective metric targets provided a strong communication medium between the drivers and the intent of the subjective evaluation. Finally, aside from purely subjective evaluating ride and handling, the subjective method effectively provided a way to determine which objective ride and handling improvements were noticeable by drivers. An example of such was the insignificant impact that the relatively increased roll delays had on the driver perception of the sport mode handling.

10.1.3 Limitations of the Research

The subjective evaluations for the semi-active suspension and the correlation studies involved certain semi-active control strategies. The results of this study should not be extrapolated to other semi-active suspension control strategies, especially in the case the strategy relies on different vehicle dynamics signals and working principles. For such strategies, the objective ride and handling performances could be significantly different than the Switchable Damper Curves and Hybrid strategies. In such cases, the virtual controller should be completely objectively evaluated offline then debugged on a compact simulator before a subjective evaluation on a dynamic simulator. Furthermore, the correlations between the objective and subjective metrics can only be considered for the range of damping settings considered. To determine the correlation of the objective and subjective metrics when evaluating other semi-active suspension strategies, the same study should be repeated.

A final note on the research limitations pertains to the subjective ride and handling as influenced by the drivers and the simulator used in this research. Depending on the skill level of each driver used, it is possible that a difference in subjective ratings could arise if instead the performance engineers or other expert drivers were considered. The performance engineers have a different level of experience and slightly different skilled opinions on differences in ride and handling, where they could have the ability to notice even smaller changes in ride and handling compared to the expert drivers used in this study. This difference does not discredit the results, but it is important to consider that the results of the subjective evaluations depend on the drivers' relevant skills and experience. This aspect is why it is important to use expert drivers as opposed to random everyday consumers who don't know what ride and handling are or have the skill to complete complex maneuvers and evaluate at the same time. Otherwise, using driers with poor experience and skill drivers could lead to inaccurate evaluations. Finally, the DiM 250 dynamic driving simulator provided a state-of-the-art tool with innovative technology and motion cuing algorithms. For less advanced simulators with less DOF or a reduced sensitivity, some of the ride and handling subjective findings might not be reproduced on such simulators. Therefore, it is important that the capabilities and performance of a simulator are considered and compared to the DiM 250 before the findings of this research are adopted in studies with other simulators.

10.2 Recommendations for Future Work

The following recommendations are defined to provide future paths of improving and implementing the subjective evaluation method in additional areas of study.

10.2.1 Alternate Control Strategies

As highlighted in Section 3.3.3, the fuzzy logic controller with preview technology provides an opportunity for larger ride and handling improvements compared to the Hybrid and Switchable Damper Curves control strategies. Care should be taken when developing the virtual preview controller in regard to the resolution of the road data file, the preview time of the controller, and any time delays in the signal between a preview and semi active suspension controller. Moreover, practical implementation of such technology should also be strongly considered since the performance of the subjective ride and handling would rely on the ability of the preview technology to be practically realized. Otherwise, the objective and subjective results on the simulator would not be useful.

One of the advantages of the triple-mode-control structure (default, ride, and sport modes) that the virtual semi-active suspension in this research had was that switching between the three modes did not induce noise or undesired vibrations. It is possible that the sport and ride modes could be combined into an auto-selection mode which uses some form of a switching boundary, similar to that of the Karnopp strategy (See Section 2.3.3). Instead of using the product between the damper and sprung mass velocities, one of the virtual accelerometer signals could be used to switch between the three modes when certain levels of primary or secondary ride accelerations are experienced by the driver. A relatively simple controller could be constructed involving multiple discrete damper curves with switching boundaries between them. The subjective evaluation method could also be used to tune the switching boundaries and damper curves. Alternatively, artificial intelligence could be implemented to learn the suspension mode preference of the driver for an auto-selection mode. Finally, damper suppliers typically provide OEMs with “Black Box” Simulink models of their semi-active suspension controller [7]. These models could also be implemented on the DiM 250 to determine their ride and handling benefit with other vehicle models.

10.2.2 Extending the Subjective Evaluation Method to Other Vehicle Systems

The subjective evaluation method could be adapted to study the impact of other suspension systems and vehicle subsystems on ride and handling. The first and most obvious system to study would be a fully active suspension. In this case, the ride and handling performance improvements could be superior to the semi-active suspension. Since active systems input energy to the vehicle, it is possible that the active suspension could improve the ride and handling on steady-state maneuvers. In this case, stead-state cornering and step-steer maneuvers should be considered as additional maneuvers in the subjective evaluation.

In the CRT vehicle models, spring rates and anti-roll bar rates are represented by lookup tables just as the damper curves are. An uncommon suspension technology in passenger cars and heavy-duty vehicles are semi-active springs. The construction of a semi-active springs can have two or more discrete sections of different spring stiffnesses. Thus, lookup tables in Simulink could be constructed and accompanied by a switching boundary to determine when each section of the spring would be engaged during driving. The spring rates could be subjectively tuned and evaluated on the DiM 250 while using an adapted version of the questionnaire. Furthermore, the current questionnaire could be used to determine if such a technology provided any practical improvement in ride and handling.

The impact on ride and handling that other control systems have could also be evaluated using the current subjective evaluation method. For instance, active or semi-active roll control and ride height systems can be evaluated. Since the developed subjective evaluation method considers the vehicle's vertical, lateral, roll, pitch, and yaw responses through its comprehensive set of maneuvers, minimal adjustment would be required for the current method. Furthermore, newly emerging ADAS technologies can be implemented in the CRT vehicle models where their impact on ride and handling can be evaluated with the current subjective evaluation method. The combination of a semi-active suspension and an ADAS system such as collision avoidance could be evaluated. Such a study would evaluate how beneficial the handling improvement from the semi-active suspension is for improving the responsiveness of the collision avoidance system. The same could be done for autonomous driving systems.

10.2.3 Automated Evaluation Tools and More

To improve the data post-processing for the objective and subjective results, an automated transfer of data between Adams Post Processor, Matlab, Simulink, and CRT could be created. A Matlab script could be created which reads the results file from an offline co-simulation, organizes the data, executes post-processing computations, and creates a table of data that can be easily opened in MS Excel or other software to plot the final results. A small extent of this idea was used for post processing the frequency response, and some of the ISO DLC and Slalom maneuvers. However, the idea could be further expanded to implement an automated tuning method which reads the objective ride and handling performance after a co-simulation between the Simulink controller and the CRT vehicle model. A Matlab or Python script could determine if the objective performance was acceptable, implement a change to the Simulink suspension controller, and re-run an offline co-simulation to determine a new objective performance. In this case, an optimization algorithm could be implemented to tune the virtual semi-active suspension controller.

In the simulator environment, a virtual questionnaire implemented on a smart device would permit real time data uploading to a cloud-based system. In this case, the subjective ratings could be displayed in the simulator control room. Personnel in the control room could then use the subjective feedback to suggest semi-active suspension controller alterations to the driver. This virtual tool would be beneficial in scenarios where the drivers are unaware of the control strategy behind the controller. The engineer who developed the controller, using their knowledge of the controller working principles, would change the control parameters according to the subjective feedback on ride and handling. In general, this approach provides a more concise and effective way to communicate and implement alterations when tuning a new semi-active suspension model.

In the future, the recommendations stated herein should be implemented to fully exploit the new subjective evaluation method and its potential for improvement and use in other areas of vehicle dynamics. The combination of the current subjective method and the recommendations would also permit further reduction in vehicle development times, especially when several active control systems are implemented in a given vehicle. OEMs and suppliers can benefit from the use of dynamic driving simulators as they provide the key advantage when implementing subjective evaluation methods. Full vehicle and subsystem models can be rapidly tuned and tested without having to physically fabricate and evaluate undesirable intermediate design iterations. The use of dynamic driving simulator will likely become the standard in vehicle development and subjective evaluation if OEMs and suppliers continue to consider these aspects.

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APPENDICES

Appendix A

Bicycle Vehicle Model Assumptions and Simplifications (from [11])

- The velocity of the vehicles centre of gravity is constant in the longitudinal direction
- Vertical, roll, and pitch responses are neglected
- Entire mass of the vehicle is concentrated at the vehicle's centre of gravity
- Front and rear tire pairs are represented by a single tire in the front and rear, respectively. The tire forces act at the centre of tires A and B from Figure 10
- Pneumatic trail and resulting aligning torque on each tire is neglected
- The load distribution between the front and rear axles is assumed to be constant
- Since the longitudinal velocity of the model is assumed constant, the longitudinal tire forces are neglected
- The steering angle is small and the radius of curvature for turning is large relative to the vehicle size
- Linear relationship between the tire lateral forces and tire slip angles (constant cornering stiffness, which is generally true for small side slip angles)

Half-Car Vehicle Model Equations of Motion (from [35])

$$[m]\ddot{z} + [c]\dot{z} + [k]z = F \tag{A1.1}$$

$$z = [z_s \quad \theta \quad z_{u.1} \quad z_{u,2}]' \tag{A1.2}$$

$$[m] = \begin{bmatrix} M_s & 0 & 0 & 0 \\ 0 & I_x & 0 & 0 \\ 0 & 0 & m_1 & 0 \\ 0 & 0 & 0 & m_2 \end{bmatrix} \tag{A1.3}$$

$$[c] = \begin{bmatrix} 2C_u & b_1 C_u - b_2 C_u & -C_u & -C_u \\ b_1 C_u - b_2 C_u & C_u b_1^2 + C_u b_2^2 & -b_1 C_u & b_2 C_u \\ -C_u & -b_1 C_f & C_u & 0 \\ -C_u & b_2 C_u & 0 & C_u \end{bmatrix} \tag{A1.4}$$

$$[k] = \begin{bmatrix} 2K_u & b_1 K_u - b_2 K_u & -K_u & -K_u \\ b_1 K_u - b_2 K_u & K_u b_1^2 + K_u b_2^2 + K_r & -b_1 K_u & b_2 K_u \\ -K_u & b_1 K_u & K_u + K_t & 0 \\ -K_u & b_2 K_u & 0 & K_u + K_t \end{bmatrix} \tag{A1.5}$$

$$F = [0 \quad 0 \quad z_{r,1} K_t \quad z_{r,2} K_t]' \tag{A1.6}$$

Twin Track Model (without kinematic wheel suspension) Additional Information

Assumptions and Simplifications

- Vehicle suspension is represented by a spring and damper, no bump stops, mounts, bushings, or other linkages are considered
- Camber, caster, and toe angles are neglected and assumed to be zero
- The rear wheels are not steerable
- The applied forces (from the spring and dampers) connect the unsprung masses to the sprung mass. These forces act at discrete points on the four corners of the sprung mass
- Air resistance and aerodynamic forces are considered and modelled as forces acting at discrete points on the sprung mass or vehicle chassis
- Anti-roll bars are represented as applied torques on the chassis near the front or rear axles
- Inertia of the wheel carriers and wheels are lumped together
- The suspension spring and damping characteristics are considered linear
- Tire lateral, longitudinal, and vertical stiffnesses are constant
- The model does not consider chassis or other body compliance

Appendix B

Grand Cherokee Objective Validation Frequency Response (0.3g)

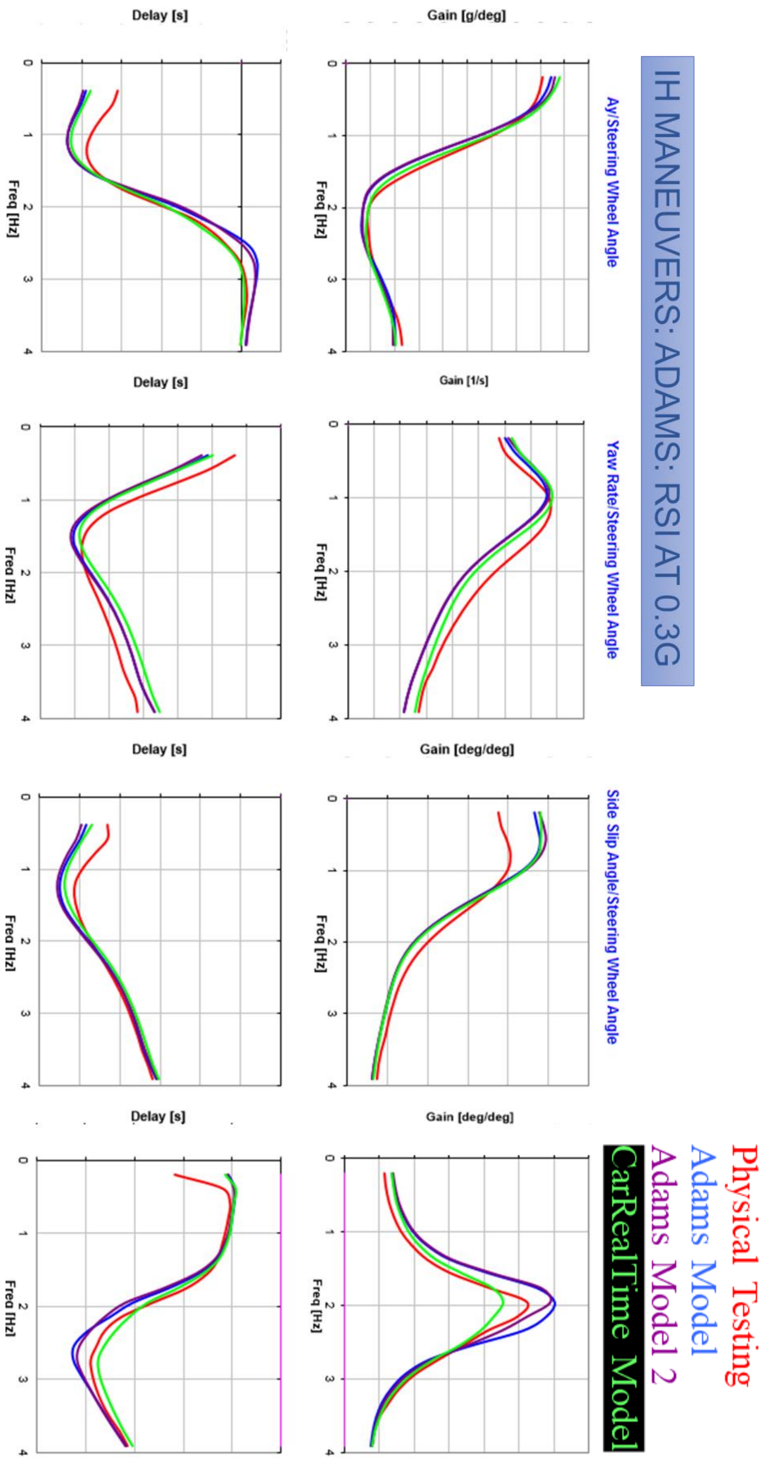

Figure 65: Objective validation of vehicle model on frequency response

Grand Cherokee Objective Validation Step Steer

The black contours in the following plots represents the data from the Grand Cherokee virtual model, whereas the plots with colour represent the data recorded during physical testing. There are several sets of data in each plot since multiple step steer maneuvers were executed. The importance lies in the similarity between the model and the physical vehicle.

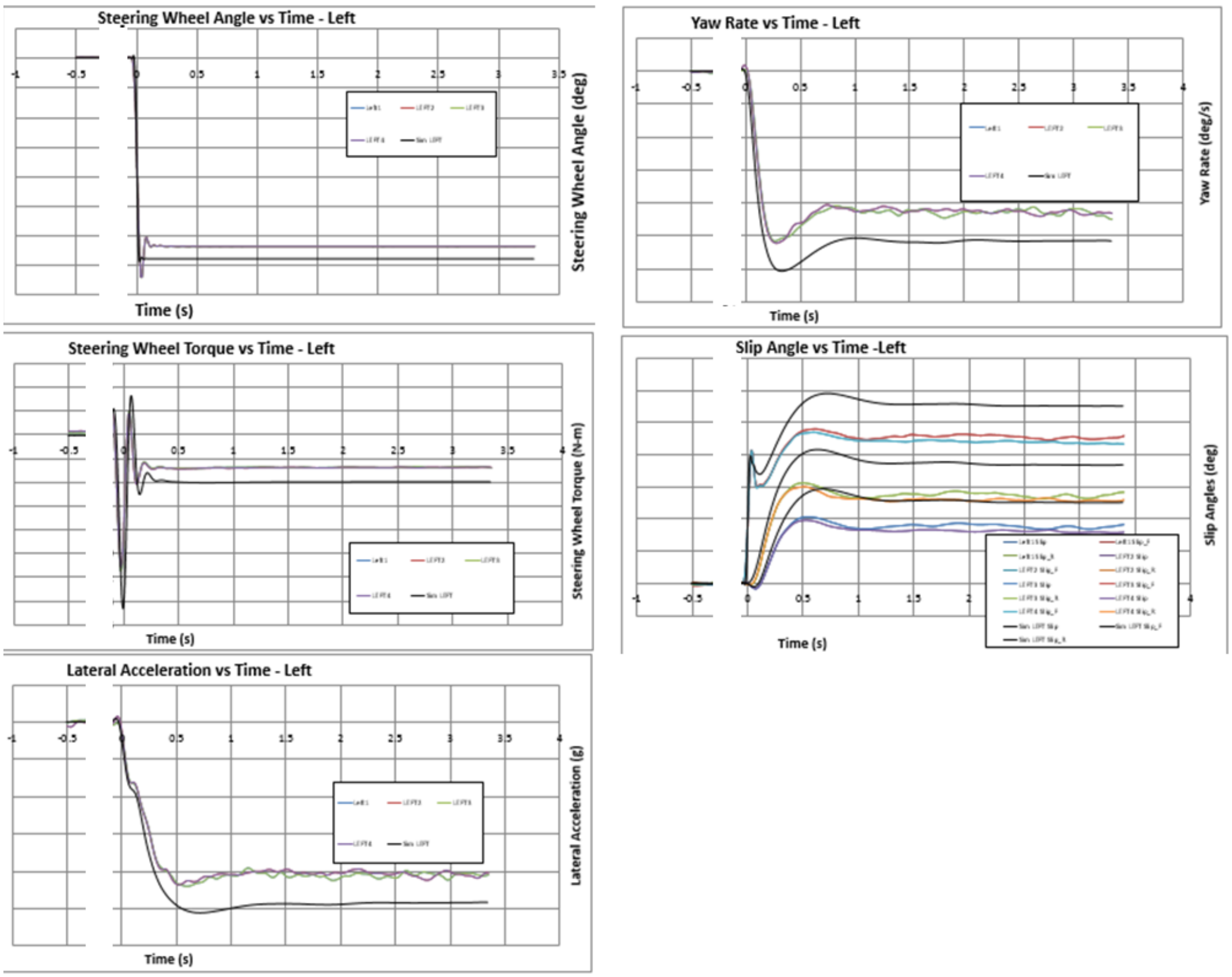

Figure 66: Objective validation of vehicle model on step steer

Appendix C

Preliminary Damper Curve Study Results Front-to-Rear Damper Alterations

Below are the damper curves with the harder and softer front settings with the Grand Cherokee.

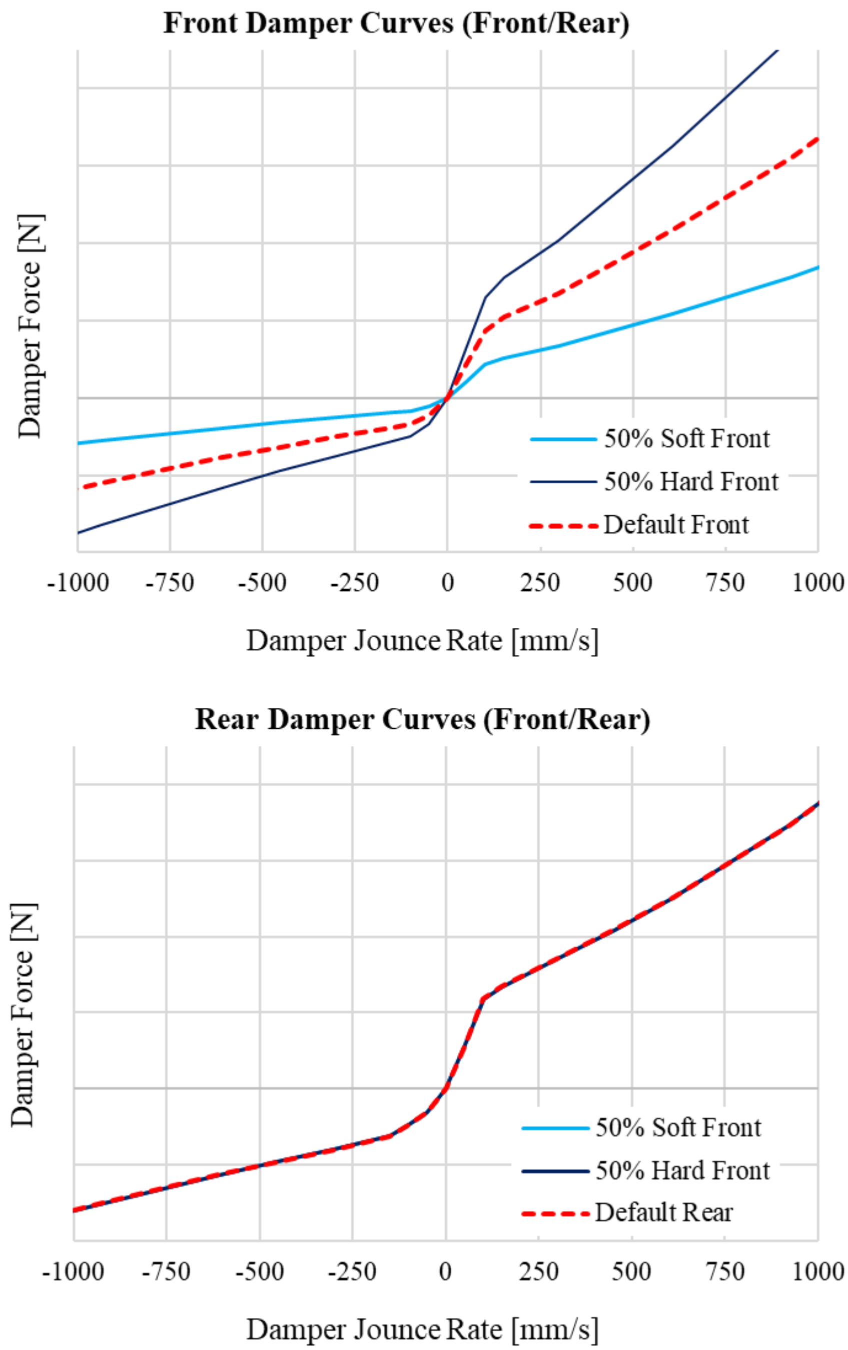

Figure 67: Front-to-rear damper curve alterations

Preliminary Damper Curve Study Results Steady State Cornering Results

The results below indicate that the dampers are not excited significantly during steady state cornering. The steering behaviour is unchanged; all configurations exhibit the same understeering behaviour. Similarly, the yaw gain is unaffected. The roll acceleration is always less than 2 deg/s^2, which is significantly low and would not be felt by the driver.

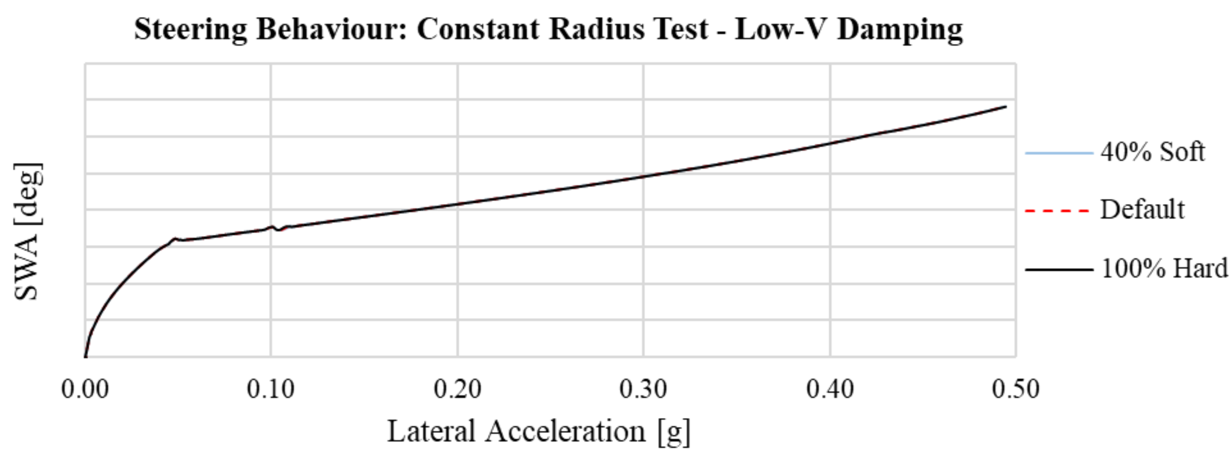

Figure 68: Steering behaviour for low-velocity damping on steady state cornering

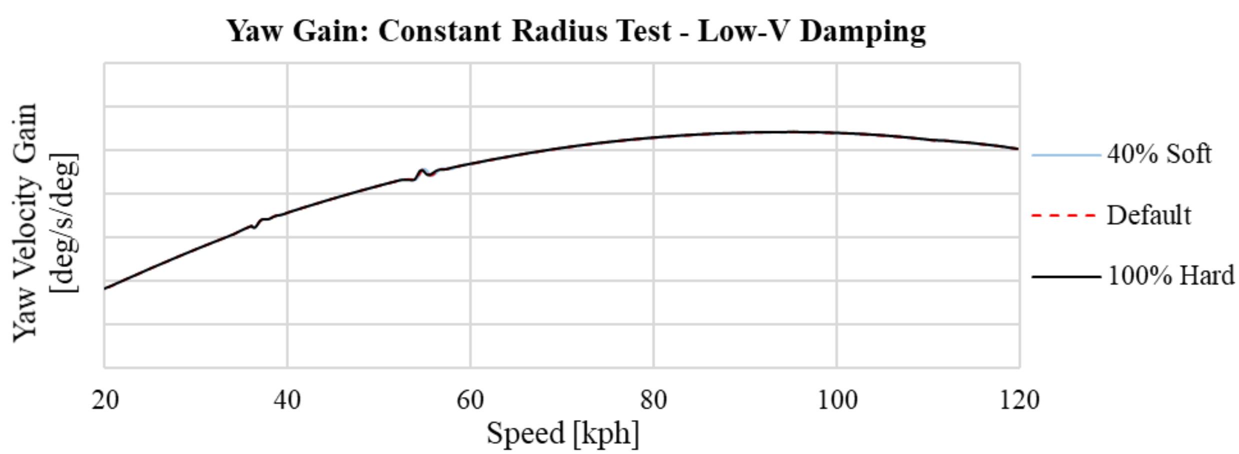

Figure 69: Yaw response for low-velocity damping on steady state cornering

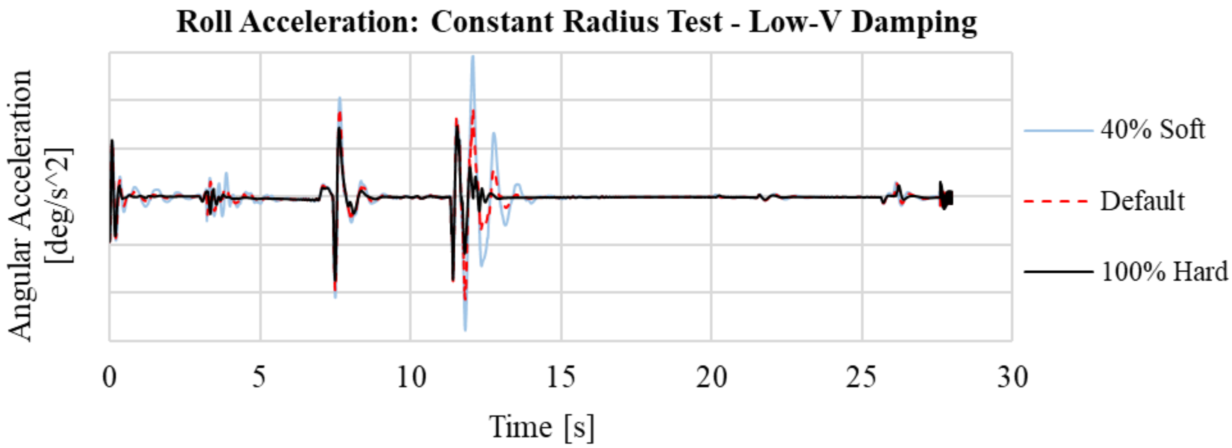

Figure 70: Roll response for low-velocity damping on steady state cornering

Preliminary Damper Curve Study Results Low-Velocity Damping on Cleat Maneuver

The vertical tire loading on one of the rear tires for the cleat maneuver is plotted below, for the low-velocity damper curve alterations. As the damping is increased, the transient response of the system has reduced overshoot and a faster settling or dissipation time. The horizontal dashed lines represent the 5% error margin around the steady state value of the vertical tire loading. Note that for the hardest setting, the response seems overdamped as the tire loading takes more time to reach steady state than the red, default damping response. The overshoot of this setting is minimal, but the dissipation time is increased as a result of the significant increase in damping. This result was found only for the rear tires. It suggests the damping is reaching a practical maximum, where a further increase in the damping would worsen the dissipation time further. At the same time, the secondary ride performance on the cleat maneuver would worsen further.

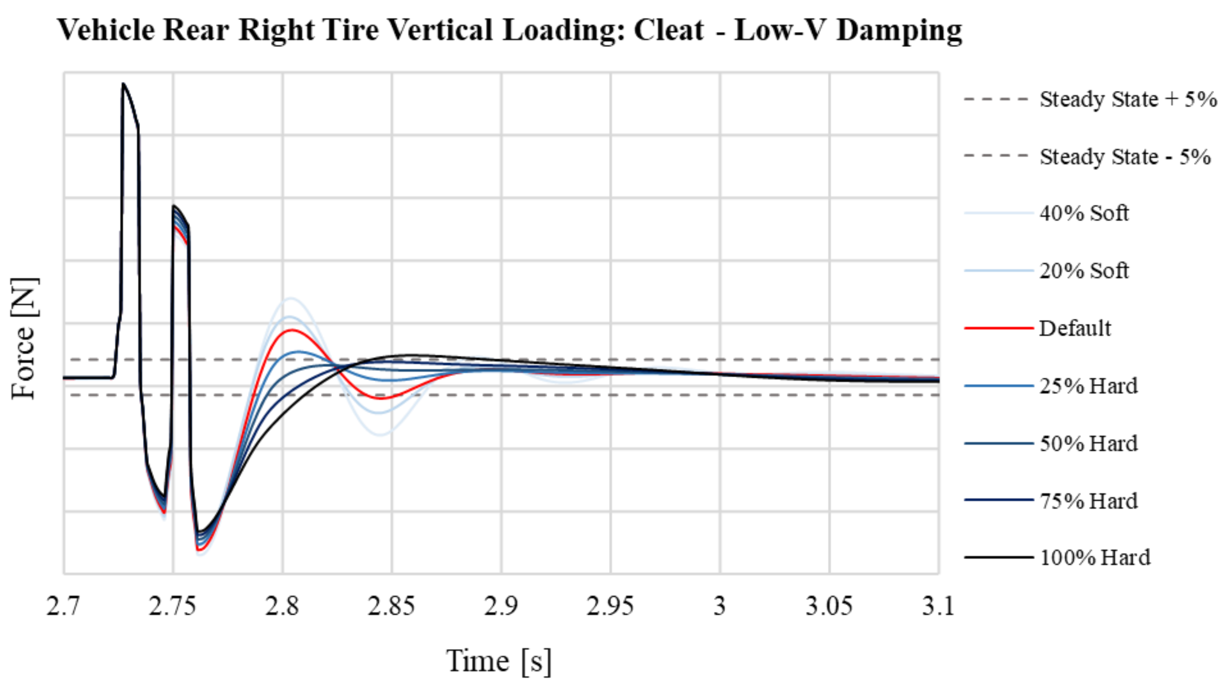

Figure 71: Rear tire vertical loading on cleat for low-velocity damping

Preliminary Damper Curve Study Results High-Velocity Damping Objective Ride

Below are the objective ride results for the high-velocity damping curve alterations. The results are less significant than the findings from the low-velocity damping results. In the second plot, the rear dissipation time for the hardest setting was found to be an outlier since the vertical tire loading for that setting barely remained within the 5% margin from the steady state value. The other settings passed outside the steady state settling time error margin directly after the first overshoot, hence the hardest setting appears to have a significantly better dissipation time. Regardless, this percent improvement is much less than the findings from the low-velocity study.

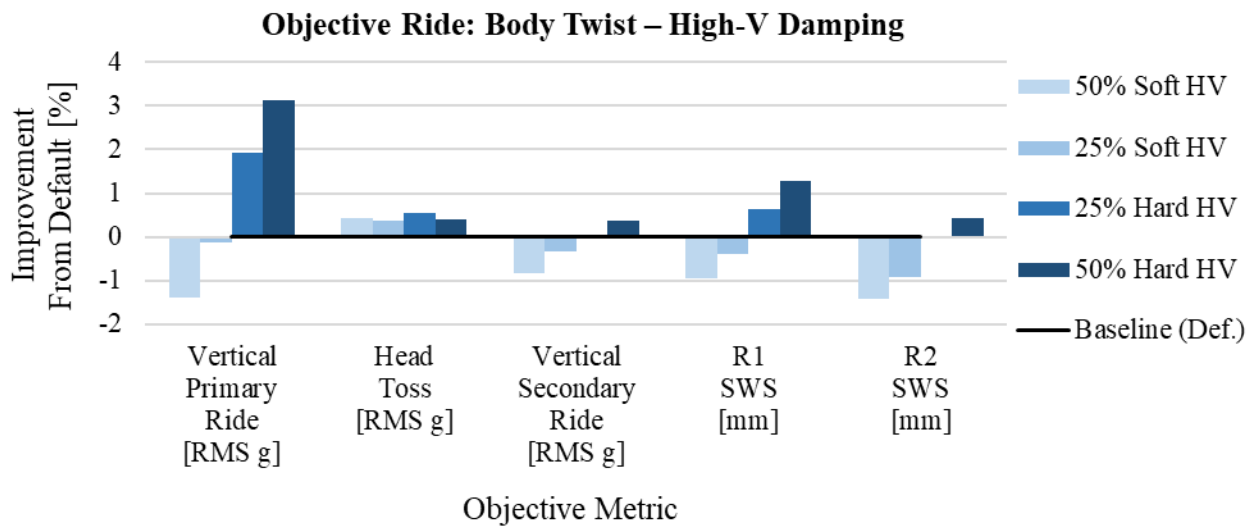

Figure 72: High-velocity damping on body twist

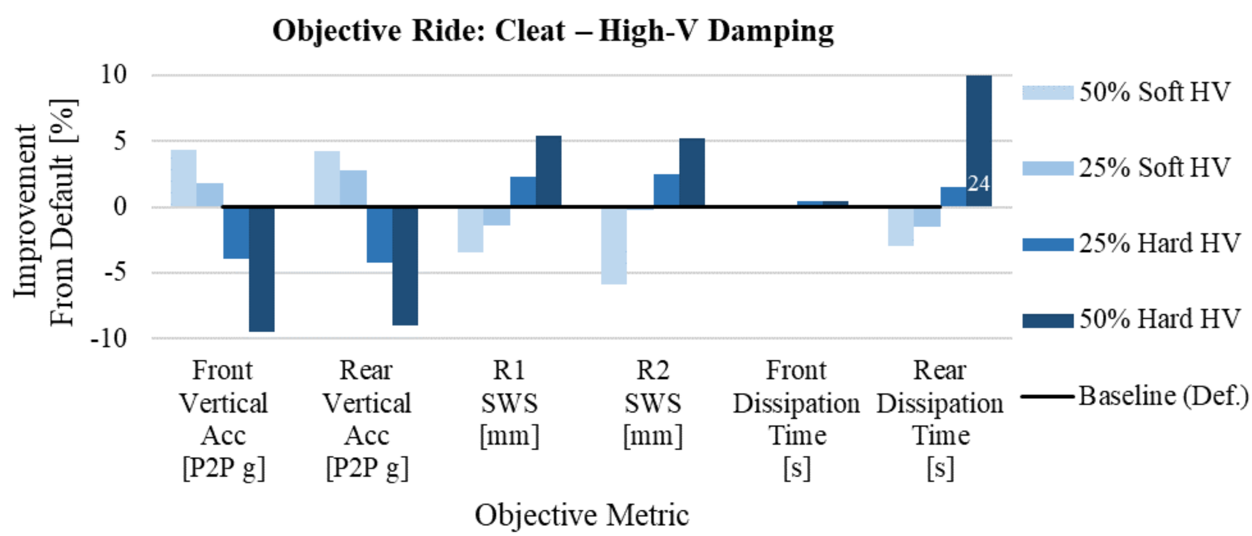

Figure 73: High-velocity damping on cleat

Preliminary Damper Curve Study Results Rebound-Compression on Step Steer Maneuver

The results below present the change in the roll acceleration response of the vehicle to a step steering input while changing the rebound and compression damping of the Grand Cherokee. As the damping is increased, the P2P roll acceleration is reduced. The highest damping setting, 50% Hard Rebound, results in an 8% reduction of the P2P roll acceleration. These findings are less significant than the low-velocity damping curve alterations. Overall, it is better to keep the same rebound-to-compression ratio as the default setting, and tune only the low-velocity regions of the damping curves when improving ride and handling.

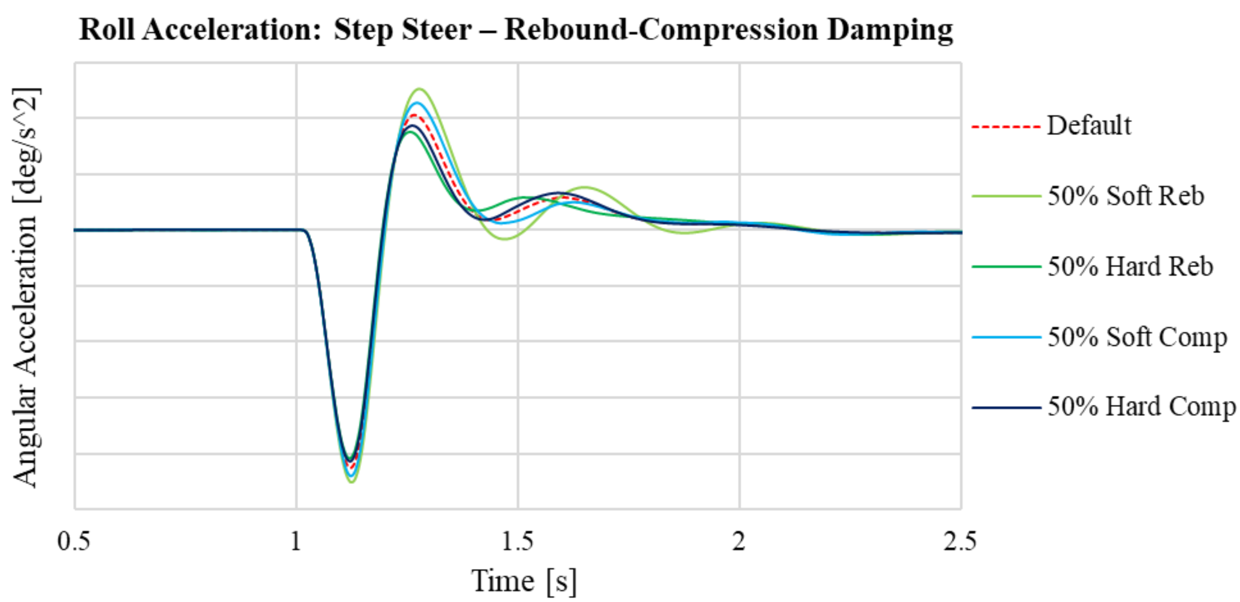

Figure 74: Rebound and compression damping on step steer

Preliminary Damper Curve Study Results Rebound-Compression Damping Objective Ride

Below are the most significant objective ride results found for the rebound and compression damping curve alterations. With the higher rebound or compression damping settings, the vertical primary ride and SWS are improved on the body twist maneuver, but the vertical secondary ride (P2P vertical acc.) on the cleat maneuver are worsened. This trend is common to all damper curve alterations that involve the increase of the low-velocity region damping. The increase of any part of the low-velocity damping results in a tradeoff between primary and secondary ride.

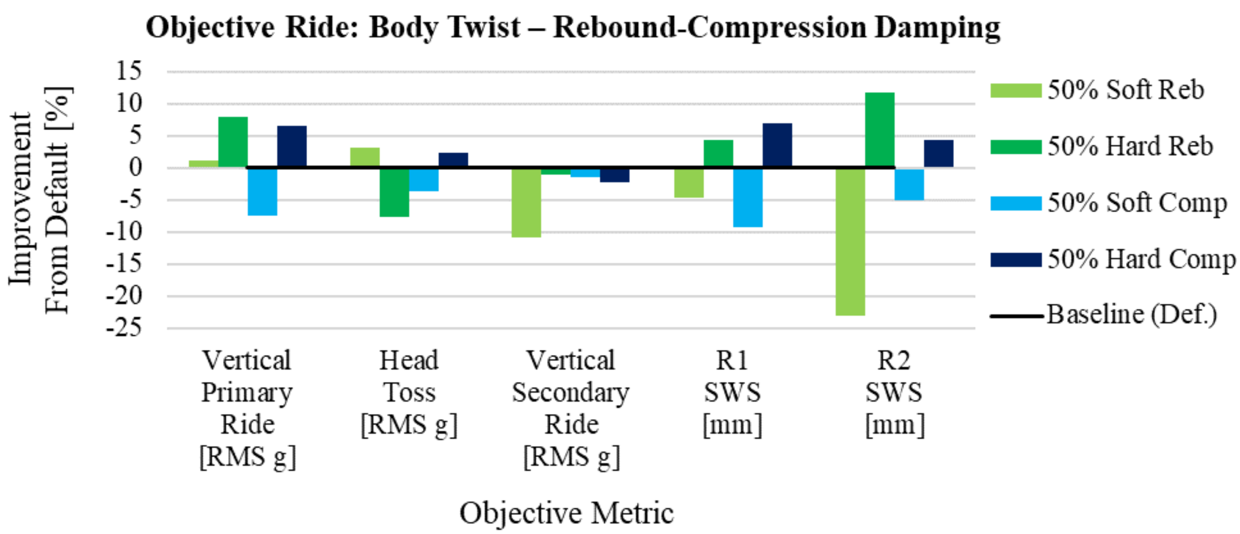

Figure 75: Rebound and compression damping on body twist

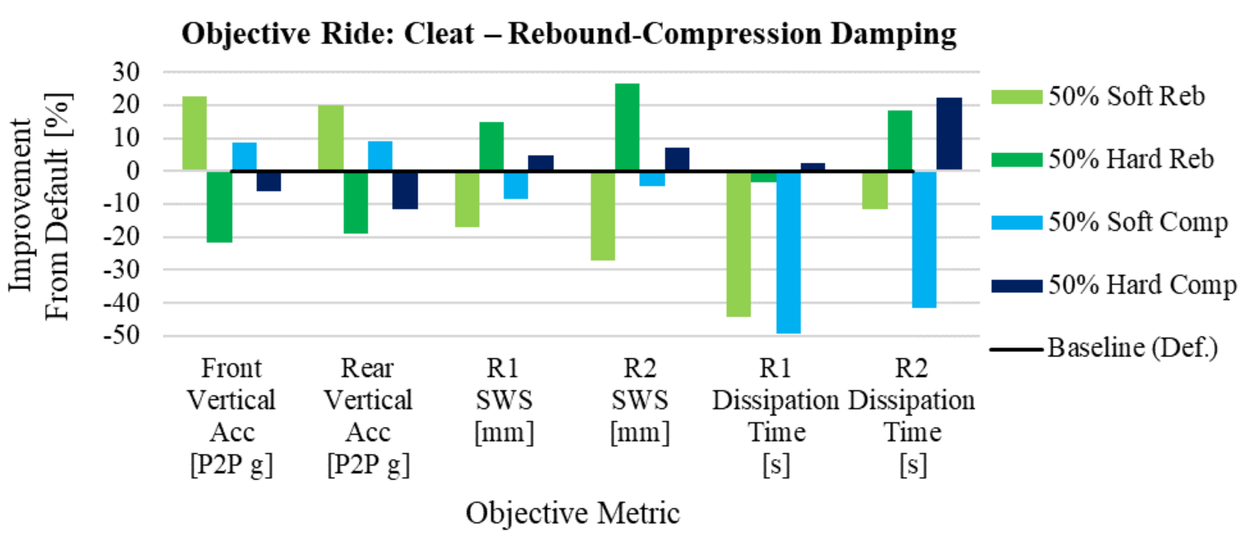

Figure 76: Rebound and compression damping on cleat

Preliminary Damper Curve Study Results Excited Damper Curve Velocities on Maneuvers

The table below summarizes the maximum and minimum damper velocities found in the vehicle response to all of the maneuvers considered in the preliminary study and more. In some maneuvers, each damper is experiencing a different input from the road, so not all dampers have the same rebound or compression velocities during a given maneuver. These damper velocities are based on simulations with the default dampers. If the damping is increased, the absolute values of the damper velocities decrease. The opposite occurs if the damping is reduced. The table is meant to give the reader an idea on how much the dampers are excited for a given maneuver. Clearly, the steady state cornering maneuver excites the damper the least, hence the insignificant results found in the preliminary damper curve study.

Table 12: Damper Velocities for Various Maneuvers

Maneuver	Absolute Damper Velocity Range [mm/s]
Step Steer	≤ 65 for 0.5g lateral acceleration
Cleat	≤ 350
ISO Double Lane Change	≤ 160
Braking	≤ 70
Body Twist	≤ 350
Swept Steer	≤ 125 for 0.5g lateral acceleration
Calabogie	≤ 400 (600 at one instance)
Slalom	≤ 120
Steady State Cornering	≤ 35

Appendix D

Controller Strategy Performance Comparison Body Twist Extended Results

Below are two figures containing data recorded during the body twist maneuver simulations for the Skyhook, Groundhook, and Hybrid control strategies with the Grand Cherokee.

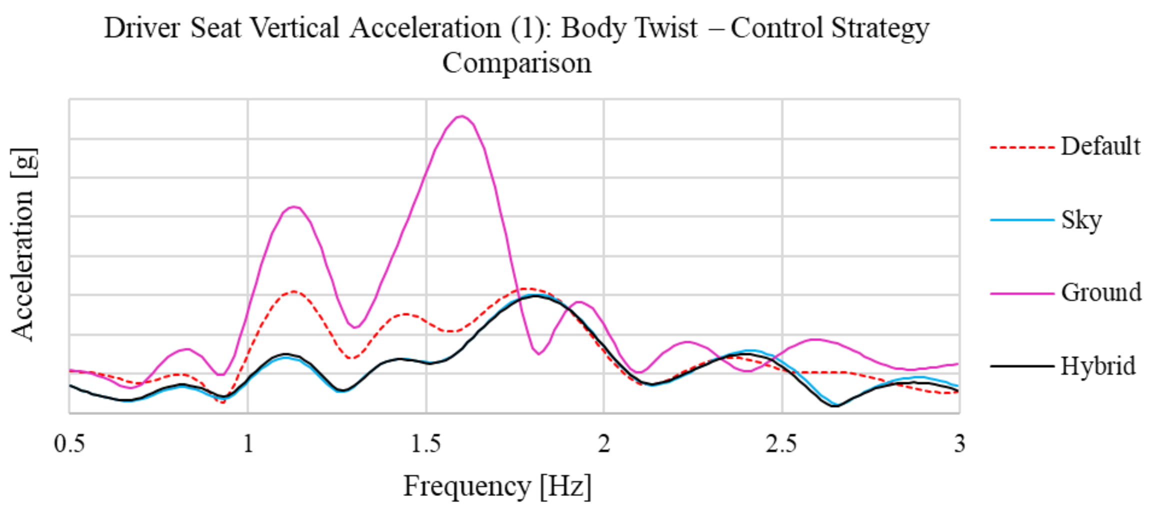

Figure 77: Control strategy objective ride performance on body twist

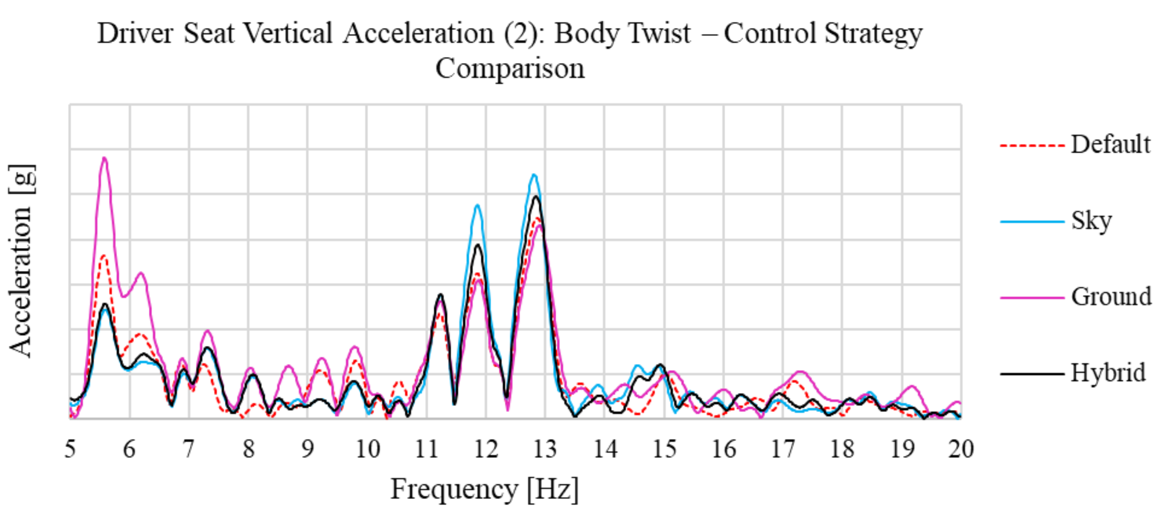

Figure 78: Control strategy objective ride performance on body twist continued

Notice that the Groundhook strategy only improves the driver seat accelerations over the default damper setting at the unsprung mass natural frequency in the second figure. In both figures, the Skyhook and Hybrid strategies have similar performance, except at the unsprung mass natural frequency. In this 11-13Hz range, the Hybrid strategy alleviates the loss of unsprung mass control found with the Skyhook strategy, as captured by the reduction in acceleration plot.

Controller Strategy Performance Comparison Cleat and ISO DLC Extended Results

Below are two figures containing data recorded during the cleat and ISO DLC maneuver simulations for the Skyhook, Groundhook, and Hybrid control strategies with the Grand Cherokee.

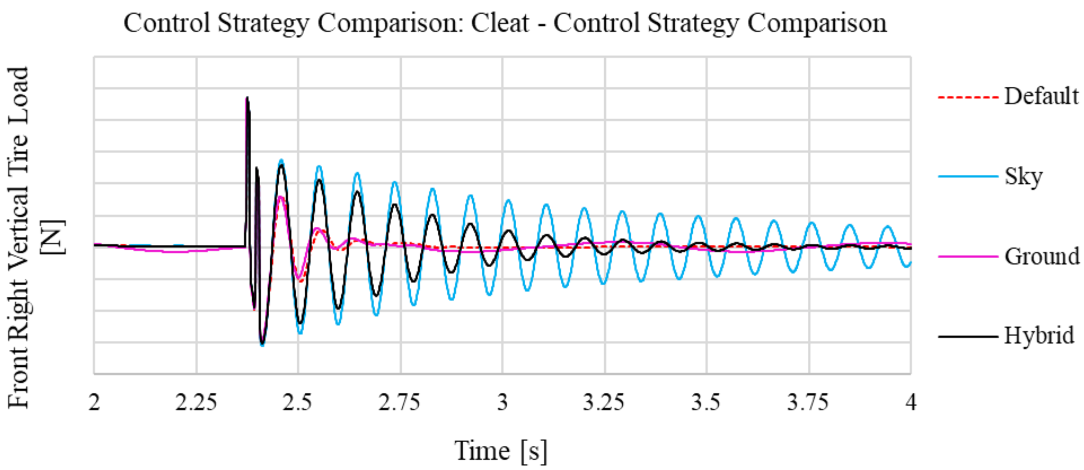

Figure 79: Control strategy objective DTL on cleat

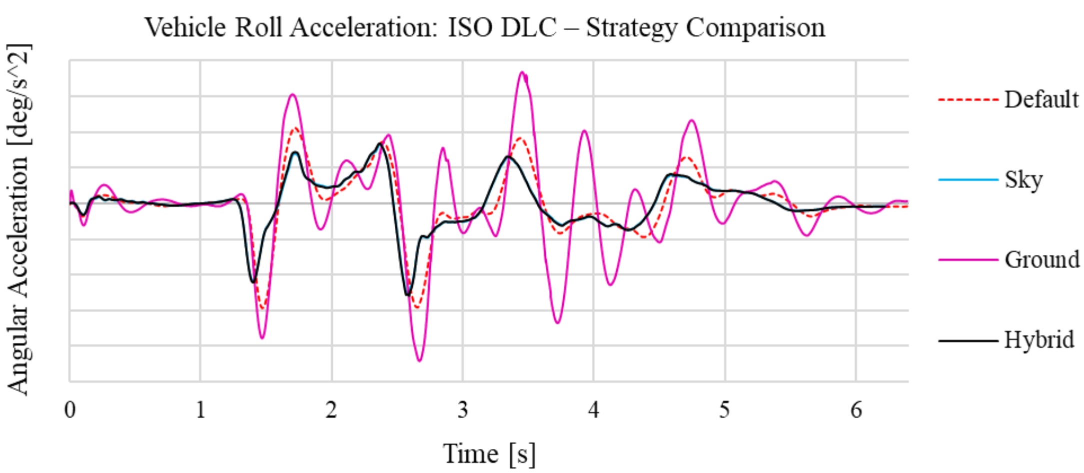

Figure 80: Control strategy objective RMS roll acceleration on ISO DLC

In the Figure 75, the Skyhook strategy has the worst fluctuation in vertical tire loading as suggested in Figure 39 from the DTL objective metrics. The Groundhook strategy appears to have the same performance as the default damper setting with no noticeable improvement. In Figure 76, further evidence that the Groundhook strategy results in poor control of the sprung mass is shown by the large fluctuations in the chassis roll accelerations on the ISO DLC maneuver. Throughout the maneuver, the vehicle chassis roll excessively and would result in poor primary ride performance and poor handling when executing the DLC maneuver.

Appendix E

Insignificant Objective Metrics Insignificant Metrics for Each Maneuver

The table below summarizes the objective metrics for each maneuver which did not result in significant ride or handling performance changes between the semi-active suspension modes.

Table 13: Insignificant Objective Metrics

Maneuver	Insignificant Objective Metrics
Body Twist	Driver Seat and Head RMS Longitudinal Accelerations
Cleat	Driver Seat P2P Longitudinal Accelerations
Frequency Response	N/A
Straight Braking	Driver Seat and Head P2P Longitudinal Accelerations
ISO DLC & Slalom	Maneuver Execution Speed, Lateral Acceleration Delay, Tire Slip Angles
Max Performance Track Event	Steering Wheel Torque, Maximum Yaw Rate, RMS Roll Acceleration

Appendix F

Subjective Metrics Description of Subjective Metrics

The table below consists of the descriptions used for each subjective metric.

Table 14: Description of Subjective Metrics

Maneuver	Subjective Metric	Metric Description
Body Twist	Vertical Ride Control	Evaluate the magnitude of the heave motion (vertical accelerations) felt at the driver seat.
	Lateral Head Toss	Evaluate the extent to which the driver's head is 'tossed' in the lateral direction while driving over the road.
Cleat	Driver Disturbance – Front	Evaluate the severity of the impact and seat vertical motion felt by the driver when the front suspension hits the cleats.
	Driver Disturbance – Rear	Evaluate the severity of the impact and seat vertical motion felt by the driver when the rear suspension hits the cleats.
	Disturbance Dissipation	Evaluate how quickly the vehicle's vertical motion dissipates after hitting the cleat.
Frequency Response	Roll Response	Consider how much/fast the vehicle rolls. Evaluate the magnitude of vehicle's roll response to the steering input.
	Roll Delay	Evaluate the roll motion in terms of the delay between the lateral dynamics (and steering) and the vehicle roll.
	Yaw Response	Evaluate the vehicle's yaw rate response. Focus on the vehicle's directness and crispness of its response.
	Yaw Delay	Evaluate the delay between the steering input and the vehicle's yaw rate.
Straight Braking	Pitch Abruptness	Evaluate the magnitude of the pitch motion in response to the braking input (focus on pitch acceleration).
	Pitch Delay	Evaluate the delay between the braking input and the pitch angle to reach steady state.
ISO DLC & Slalom	Maneuverability	How well can the vehicle turn in and out of the cones? Focus on the vehicle's yaw rate and tire slip.
	Delay	Evaluate the time delays between the steering input and the vehicle's roll angle and yaw rate motion.
	Steering Wheel Activity	Evaluate the required amount of steering the driver has to apply to maneuver through the cones (focus on angle only).
	Roll Response	Evaluate the roll response to steering inputs. Consider how much/fast the vehicle rolls while weaving the cones.
Max Performance Track Event	Stability	Do the rear wheels lose grip easily? Evaluate the tire slip in the rear during transient steering/cornering.
	Roll Response	Evaluate the roll response. Consider how fast the vehicle rolls (rate and acc.) while executing corners and chicanes.
	Turn-In Response	Evaluate the vehicle's response to initial steering input when entering corners on the track.
	Steering Wheel Activity	Evaluate the required amount of steering the driver has to apply to maneuver through corners (focus on angle only).

Subjective Metrics Insignificant Subjective Metrics Removed from Evaluation

The table below contains a list of the subjective metrics that were removed from the subjective evaluation due to their insignificance when evaluation semi-active suspension. Note that the step steer maneuver was removed during the early stages for development for the subjective evaluation method. Objectively, the results indicated the least differences in handling between the suspension modes. Subjectively, there were minimal differences between vehicle's responses noticed by the drivers and thus were deemed insignificant for the evaluation. This decision also allowed for a timelier evaluation process.

Table 15: Insignificant Subjective Metrics

Maneuver	Insignificant Subjective Metrics
Body Twist	Longitudinal Head Toss
Cleat	N/A
Frequency Response	N/A
Straight Braking	Wheel Control (rear wheel lift)
ISO DLC & Slalom	N/A
Max Performance Track Event	N/A

Questionnaire Format General Format of the Questionnaire Pages

The following images present the structure of the questionnaire. The first image pertains to the front page, the second pertains to one of the seven maneuvers, and the third image pertains to the feedback section on the back of the questionnaire. Certain information specific to Stellantis' internal standards have been hidden for confidentiality.

Subjective Evaluation of Vehicle Semi-Active Suspension for Improved Ride and Handling

Driver Name: _______________ Date: _________

Instructions: Please complete each maneuver with the selected controller mode and fill in the corresponding subjective ratings. The subjecting rating scheme can be found below. Each page of this questionnaire corresponds to a different maneuver. Additional instructions will be provided as you work through each maneuver. We will select the damper controller modes for you.

Maneuvers		
1	Straight Braking	3 Lanes
2	Frequency Response	3 Lanes
3	Slalom	Proving Grounds
4	Double Lane Change	Proving Grounds
5	Max Performance	Grattan Track
6	Body Twist	CPG South Tort.
7	1" Cleat	9 Lanes

General Steps (for each maneuver)
1 - Complete the maneuver/obstacle
2 - Rate the current mode (start with 1)
3 - Select controller mode "2"
4 - Complete the maneuver again
5 - Rate mode 2
6 - Repeat steps 3 to 5 with mode "3"
7 - Move onto next maneuver/obstacle

Subjective Rating Scale

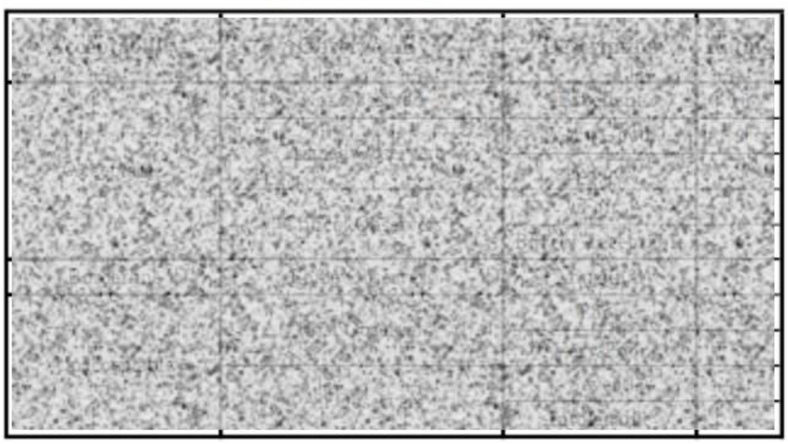

Figure 81: Questionnaire first page format

This first page of the questionnaire was designed to repeat the purpose and procedure of the study to the drivers. The rating scale contains information on the acceptability, text descriptions for each integer rating and observability of certain drivers, and the corresponding integer values. Additionally, refinement of the rating values and a description for higher levels of refinement are placed below the rating scale. This information was removed due to copyright from SAE.

3. Slalom: 7 Cones x 30.5m, 3 trials

Mode 1: Record fastest speed here: ______	Target	Subjective Rating									
		Intolerable	Severe	Very Poor	Poor	Marginal	Barely Acc.	Fair	Good	Very Good	Excellent
Maneuverability (handling) How well can the vehicle turn in and out of the cones? Focus on the vehicle's yaw rate and tire slip.	Crisp Yawing, Min. Slip	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Delay (handling) Evaluate the time delays between the steering input and the vehicle's roll angle and yaw rate motion.	Minimal to No Delay	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Steering Wheel Activity (handling) Evaluate the required amount of steering the driver has to apply to maneuver through the cones (**focus on angle only**).	Smooth and Small Angles	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Roll Response (ride) Evaluate the roll response to steering inputs. Consider how much/fast the vehicle rolls while weaving the cones.	Less is Better	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10

Mode 2: Record fastest speed here: ______	Target	Subjective Rating									
Maneuverability (handling) How well can the vehicle turn in and out of the cones? Focus on the vehicle's yaw rate and tire slip.	Crisp Yawing, Min. Slip	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Delay (handling) Evaluate the time delays between the steering input and the vehicle's roll angle and yaw rate motion.	Minimal to No Delay	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Steering Wheel Activity (handling) Evaluate the required amount of steering the driver has to apply to maneuver through the cones (**focus on angle only**).	Smooth and Small Angles	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Roll Response (ride) Evaluate the roll response to steering inputs. Consider how much/fast the vehicle rolls while weaving the cones.	Less is Better	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10

Mode 3: Record fastest speed here: ______	Target	Subjective Rating									
Maneuverability (handling) How well can the vehicle turn in and out of the cones? Focus on the vehicle's yaw rate and tire slip.	Crisp Yawing, Min. Slip	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Delay (handling) Evaluate the time delays between the steering input and the vehicle's roll angle and yaw rate motion.	Minimal to No Delay	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Steering Wheel Activity (handling) Evaluate the required amount of steering the driver has to apply to maneuver through the cones (**focus on angle only**).	Smooth and Small Angles	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10
Roll Response (ride) Evaluate the roll response to steering inputs. Consider how much/fast the vehicle rolls while weaving the cones.	Less is Better	○ 1	○ 2	○ 3	○ 4	○ 5	○ 6	○ 7	○ 8	○ 9	○ 10

Figure 82: Questionnaire maneuver page format

Driver Additional Feedback

Mode 1	Please use the space below for any additional comments you have on the ride and handling of the vehicle. If possible, comment on the vehicle's character (i.e. agile, sporty, boring, nervous, stiff, etc.).

Mode 2	Please use the space below for any additional comments you have on the ride and handling of the vehicle. If possible, comment on the vehicle's character (i.e. agile, sporty, boring, nervous, stiff, etc.).

Mode 3	Please use the space below for any additional comments you have on the ride and handling of the vehicle. If possible, comment on the vehicle's character (i.e. agile, sporty, boring, nervous, stiff, etc.).

Figure 83: Questionnaire feedback page format

Appendix G

Objective Ride and Handling Body Twist and Cleat Extended Results for Primary Vehicle Model

The figure below contains additional objective metrics for secondary ride and SWS on the body twist and cleat maneuvers, respectively.

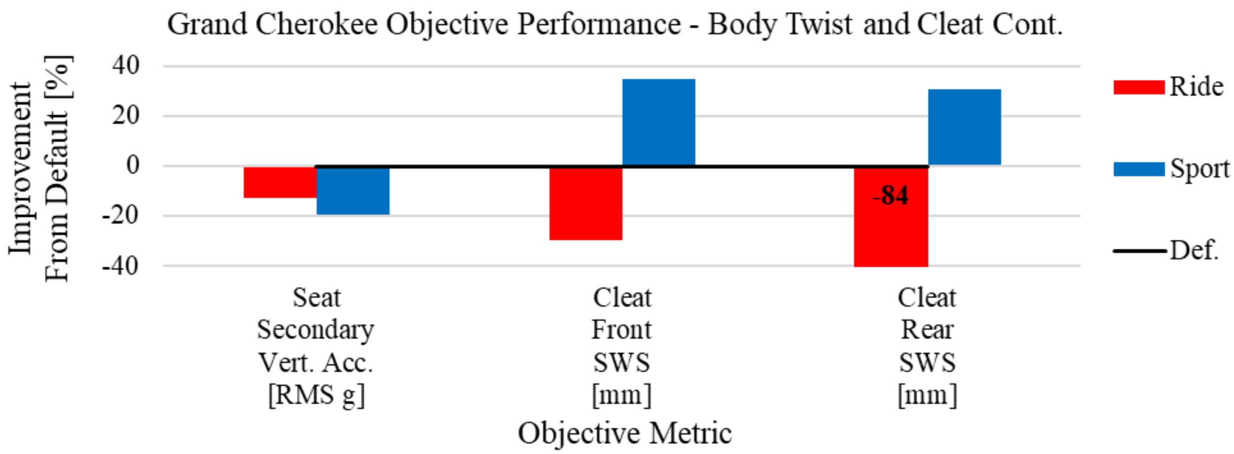

Figure 84: Grand Cherokee objective ride additional results

For the driver seat secondary vertical acceleration metric on body twist, the ride and sport modes worsen this objective wide compared to the default mode. Due to the ride mode's worsening of DTL and SWS metrics from Figure 47 (Section 9.1.1), the dissipation of disturbances is also worsened where the suspension could take too long to dissipate one disturbance before being introduced to a subsequent road disturbance. Recall that the secondary ride is in the frequency range of 5-20Hz, typically where the unsprung mass's (wheel, tire, and certain suspension components) natural frequency resides. Since the DTL objective metric relates to the fluctuation in tires' vertical loading, a deterioration of DTL can directly impact the secondary ride felt at the driver seat location. Thus, the DTL and secondary ride metrics are linked on the body twist maneuver.

Objective Ride and Handling Frequency Response Results for Primary Vehicle Model

The figures below contains objective handling metrics from the frequency response maneuver.

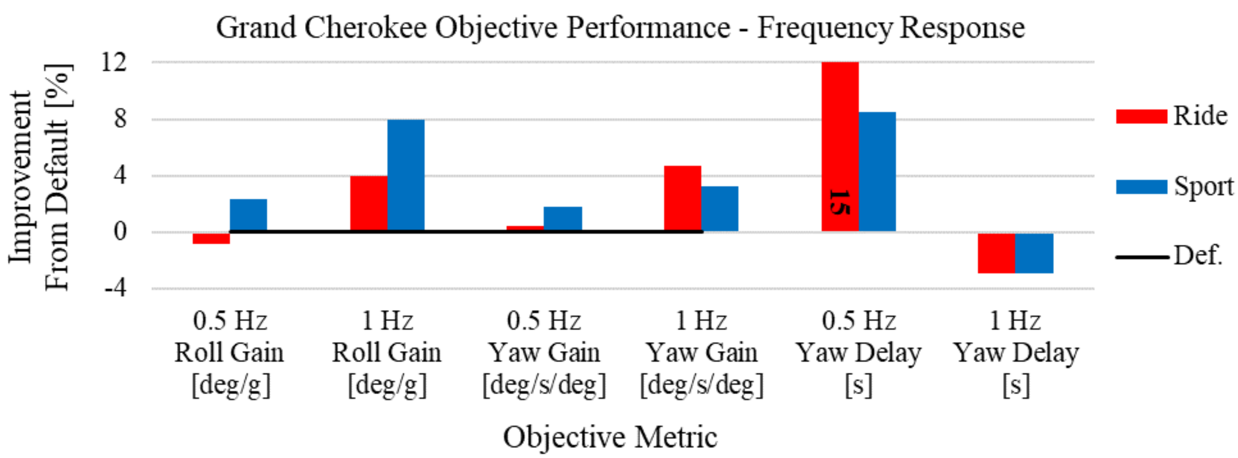

Figure 85: Grand Cherokee objective handling additional results on frequency response

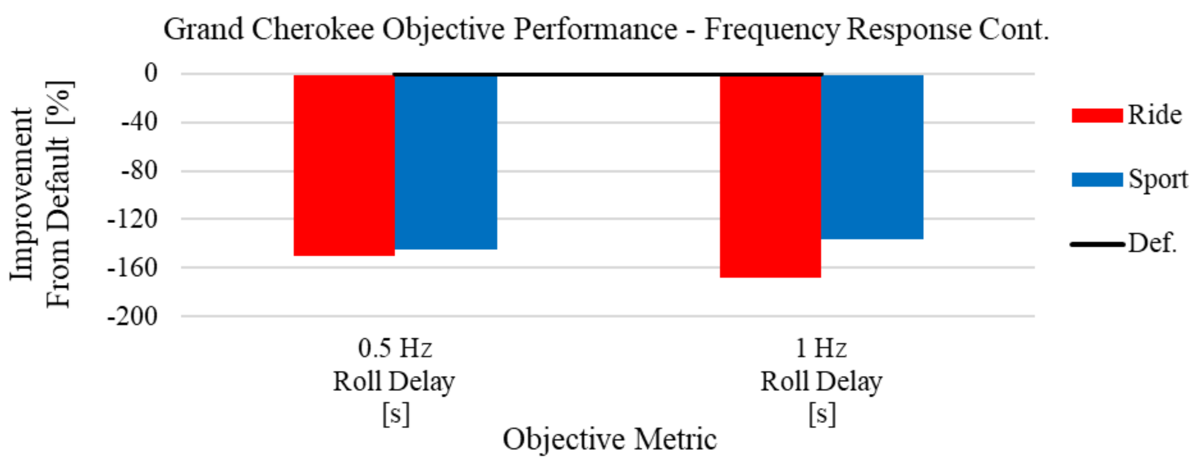

Figure 86: Grand Cherokee objective handling additional results on frequency response (cont.)

In general, the sport mode reduces the chassis roll and increases the yaw rate the most at the steering frequencies depicted in the figures. At the lower frequency, the ride mode slightly worsens body roll. The delay between the roll angle and yaw rate with respect to the input SWA are also shown. The objective roll angle delay is worsened by both suspension modes at both frequencies depicted in the second figure. However, the magnitude of the metrics is extremely small, where any change in said metric would result in large percentage differences from the reference default mode. Thus, the findings for the roll delay were considered insignificant.

Objective Ride and Handling Slalom Results for Primary Vehicle Model

The figure below contains objective ride and handling metrics from the slalom maneuver.

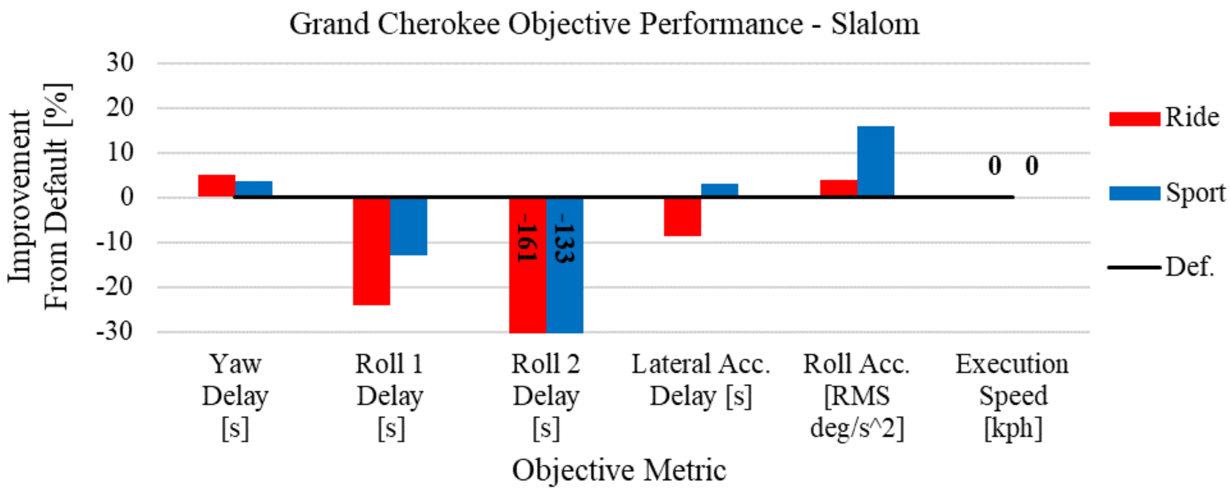

Figure 87: Grand Cherokee objective handling additional results on slalom

The objective results for the slalom maneuver suggest similar changes in ride and handling as found on the ISO DLC maneuver. The sport mode reduces the RMS roll acceleration the most and slightly improves the lateral acceleration and yaw rate delays. The roll delays were considered insignificant here as well as the ISO DLC maneuver. For the ride mode, the yaw delay is improved more than the sport mode. Upon inspecting the yaw delay metrics for the frequency response maneuver having similar steering frequency inputs, it appears the ride mode is able to improve the yaw response delay at low steering input frequencies. This result is supporting the evidence from literature that the hybrid strategy can produce improvements in both ride and handling at the same time, at least objectively.

Objective Ride and Handling Tuning the Renegade Controller

The following figures contain objective ride results found when tuning the ride mode of the semi-active suspension controller. These figures contain the results of the best intermediate controller settings.

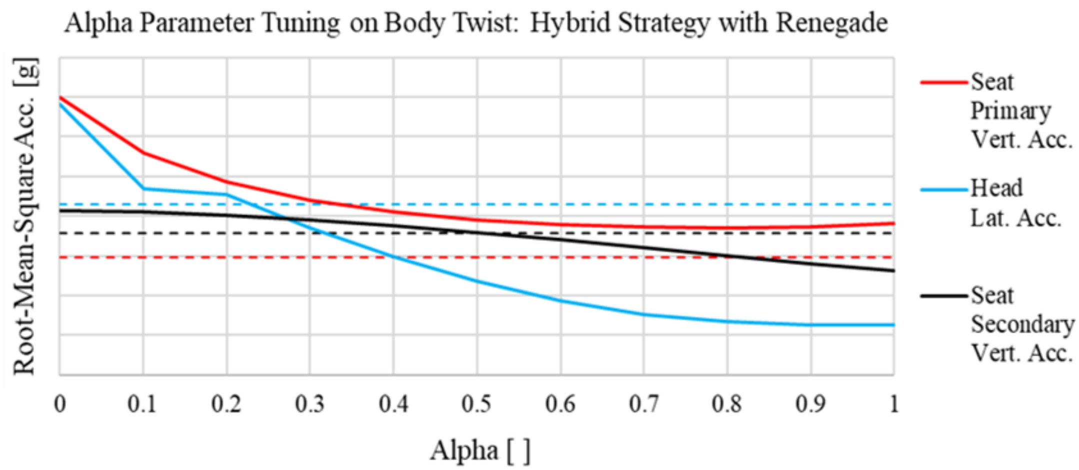

Figure 88: Tuning the alpha parameter with the Renegade controller on body twist

The first image depicts the vertical acceleration metrics for RMS primary and secondary ride frequencies measured at the driver seat, as well as the RMS lateral acceleration at the driver's head level, in the primary frequency range. The horizontal dashed lines depict the default mode's performance on these three objective ride metrics. Along the x-axis is the alpha parameter for the Hybrid control strategy. Note that for all values of alpha, keeping the front and rear Hybrid gains at a maximum of 2.2, the driver seat primary vertical acceleration could not be improved over the default. At values of alpha above 0.5, both the head lateral acceleration and driver seat secondary vertical acceleration could be improved over the default mode. A value of 0.8 results in the best driver seat primary vertical acceleration.

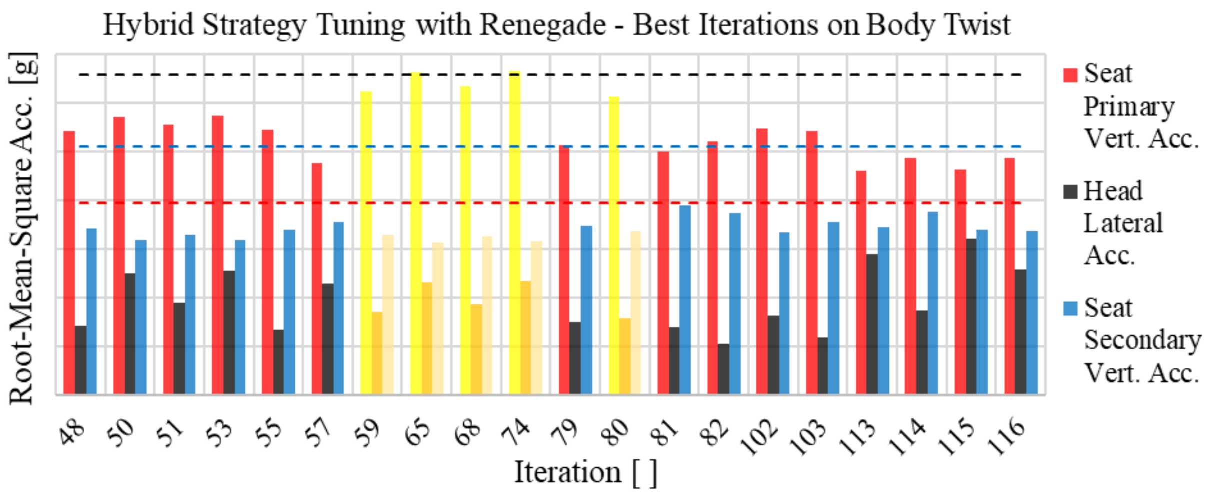

Figure 89: Objective ride for Renegade controller best iterations

The second image presents results of the same objective ride metrics as the previous figure, except for several of the best-found controller iterations are presented. Again, the horizontal dashed lines represent the default mode's performance. The results of the iteration highlighted in yellow depict unfeasible iterations, since the driver seat RMS primary vertical accelerations were worsened the most of all iterations. See the table below for the settings of the controller for each iteration. In general, tuning the controller involved observing the impact of altering the damper curves and the hybrid gains, whereas the alpha parameter was best held at a value of 0.8.

Table 16: Controller Settings for Renegade's Best Iterations

Iteration ID	Controller Settings				
	Front Damper Curves	Rear Damper Curves	Front Hybrid Gain	Rear Hybrid Gain	Alpha
48	50% Hard	50% Hard	0.5	0.75	0.8
50	50% Hard	Default	0.5	0.75	0.8
51	50% Hard	Default	0.5	1	0.8
53	50% Hard	25% Soft	0.5	1	0.8
55	50% Hard	75% Hard	0.5	0.75	0.8
57	Default	50% Hard	1.5	0.75	0.8
59	Default	50% Hard	0.5	0.75	0.8
65	Default	Default	0.5	0.75	0.8
68	Default	Default	0.5	1	0.8
74	Default	25% Soft	0.5	1	0.8

79	Default	75% Hard	1	0.75	0.8
80	Default	75% Hard	0.5	0.75	0.8
81	Default	75% Hard	1.5	1	0.8
82	Default	75% Hard	1	1	0.8
102	25% Soft	50% Hard	1	0.75	0.8
103	25% Soft	50% Hard	1	1	0.8
113	Default	50% Hard	1.7	0.5	0.8
114	25% Soft	50% Hard	2	1	0.8
115	25% Soft	50% Hard	2.2	0.5	0.8
116	25% Soft	50% Hard	1.3	0.6	0.8

Once the performance of an iteration was recorded from the body twist maneuver, the ride performance on the cleat maneuver was recorded to ensure there were no significant tradeoffs in ride and handling. In general, the SWS and the DTL metrics changed in the same way when altering the controller settings, but the SWS is easier and quicker to record, straight from Adams Post Processor. Therefore, the secondary ride and the SWS metrics were recorded to observe the performance trends. Figures 86 and 87 present the results of these metric for best controller iterations. As before, the dashed lines represent the default mode's performance.

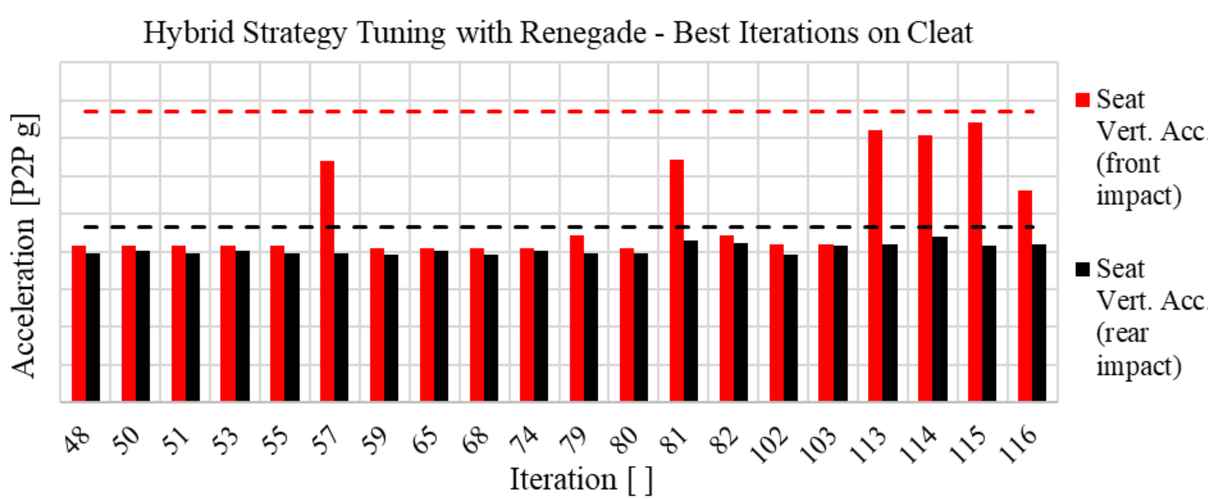

Figure 90: Renegade controller best iterations' objective secondary ride

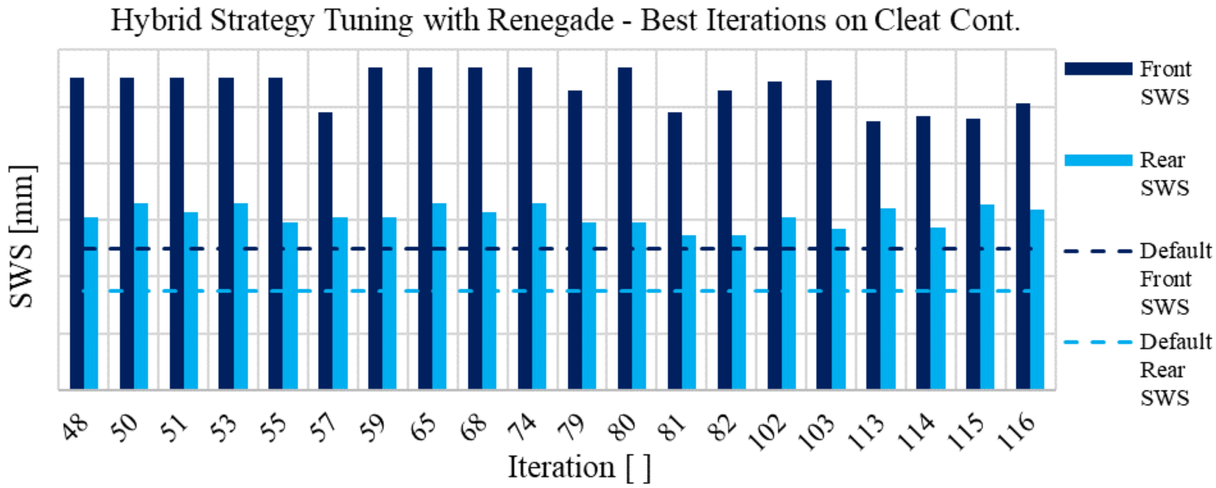

Figure 91: Renegade controller best iterations' objective SWS ride on cleat

For all of the best iterations, driver seat P2P vertical accelerations in the secondary ride frequency range are improved, thus there were not unfeasibly controller settings in this case. Finally, one last check was made on the straight braking maneuver to ensure that rear wheel lift did not occur. For the Renegade, this issue was more abundant than with the Grand Cherokee. The following figure presents the rear wheel vertical load (equal for both rear wheels) in response to the braking input on the straight braking maneuver.

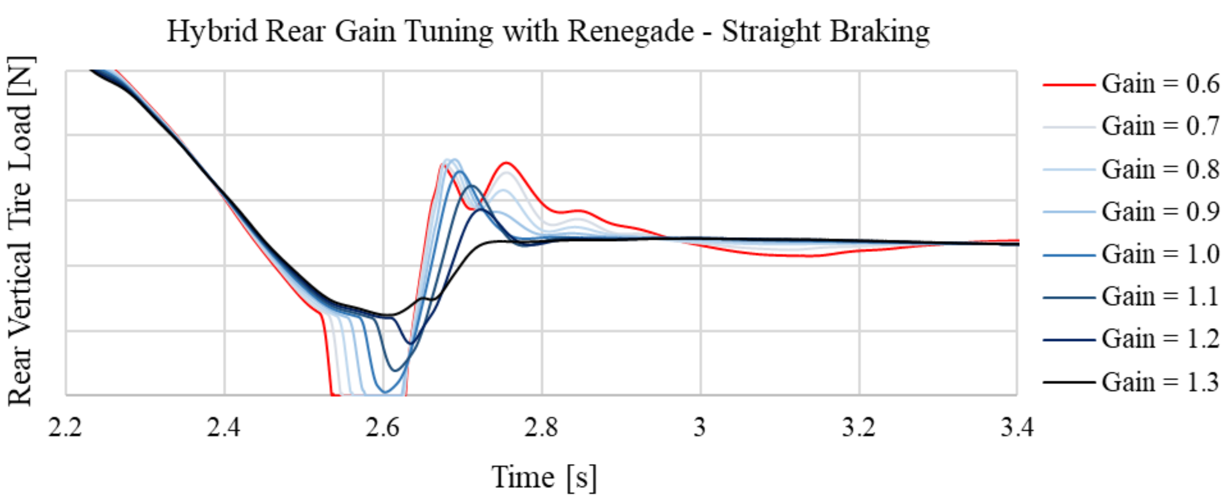

Figure 92: Renegade controller best iterations' wheel lift on straight braking

Recall from the previous table that iteration 116 is nearly the same as the final iteration of the controller (See Table 9). In the figure above, the red contour depicts the rear wheel load for Iteration 116 on the straight braking. Clearly, wheel lift occurred as the vertical tire loading dropped to zero upon braking. The rear hybrid gain was then increased until the real wheel lift was removed and a significant drop in the vertical tire loading was alleviated. The result was a rear hybrid gain of 1.3. The results of the controller with these final settings are the ones presented in Chapter 9.

Objective Ride and Handling Body Twist and Cleat Extended Results for Secondary Vehicle Model

The figure below contains additional objective metrics for secondary ride and SWS on the body twist and cleat maneuvers, respectively. These results are for the Renegade vehicle model.

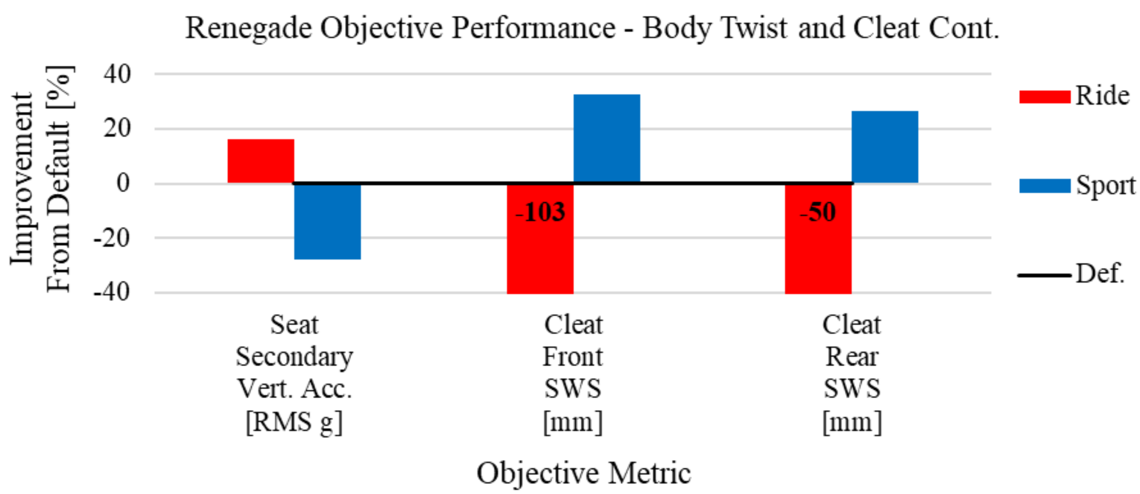

Figure 93: Renegade objective ride additional results

On the cleat maneuver, the rear SWS is not worsened as significantly as it was on the Grand Cherokee ride mode. Since the DTL (See Figure 51) was not worsened and the dissipation times of the vertical tire loading were not affected for the sport and ride modes of the Renegade, it was found that the driver seat secondary vertical RMS acceleration metric in the figure above was improved for the ride mode. Furthermore, for all settings of the semi-active suspension controller of the Renegade, the vertical tire forces dropped to zero on upon impact with the cleat, for the front and rear tires. On the body twist maneuver, the rear damper jounce of the ride mode appeared to exhibit a slight jacking down effect. Here, the dampers remained more compressed for some of the maneuver, compared to the default mode. This effect results in a lower total SWS, thus the rear SWS was shown to be improved on the body twist maneuver.

Objective Ride and Handling Frequency Response Results for Secondary Vehicle Model

The figures below contains objective handling metrics from the frequency response maneuver for the Renegade controller.

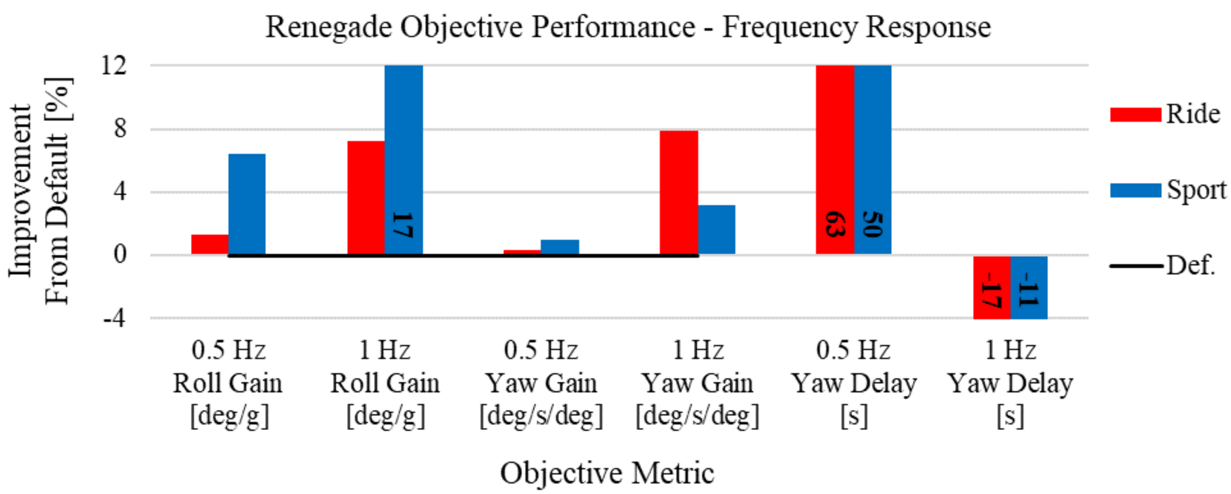

Figure 94: Renegade objective handling additional results on frequency response

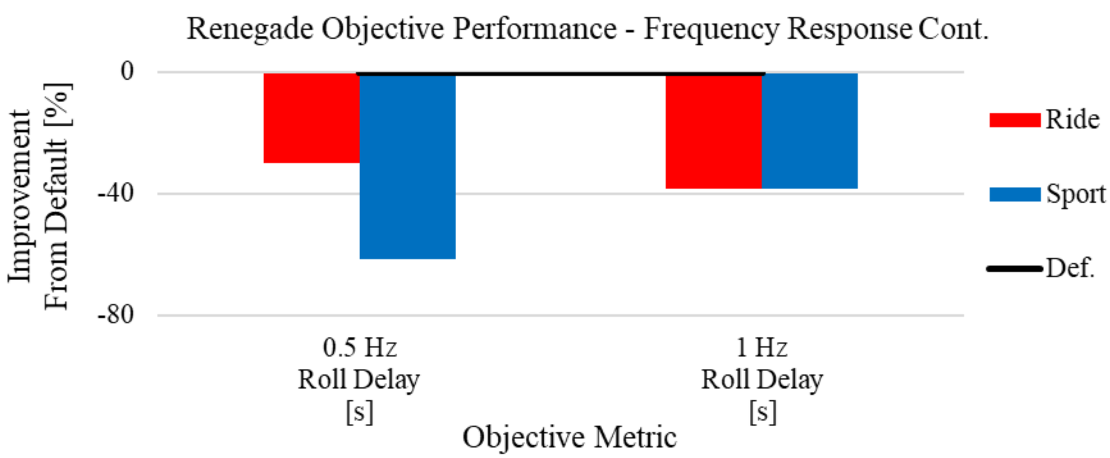

Figure 95: Renegade objective handling additional results on frequency response (cont.)

The sport mode reduces the chassis roll motion the most among the three suspension modes. The ride mode increases the yaw rate in response to driver inputs at the higher frequency specified in the first figure. As with the primary vehicle model, the hybrid strategy has proven to improve certain ride and handling metrics over the default passive suspension. However, the magnitudes of the handling metric improvements on the frequency response maneuver are generally small. Finally, the roll angle delay metrics result in significant increases for both the sport and ride modes, suggesting a more delayed roll response. However, the magnitude of the metrics are extremely low, even lower than the primary vehicle model's delay values. Moreover, the yaw delay magnitudes are smaller and likely less noticeable than the primary vehicle model.

Objective Ride and Handling Slalom Results for Secondary Vehicle Model

The figure below contains objective ride and handling metrics from the slalom maneuver while considering he performance of the semi-active suspension controller with the Renegade vehicle model.

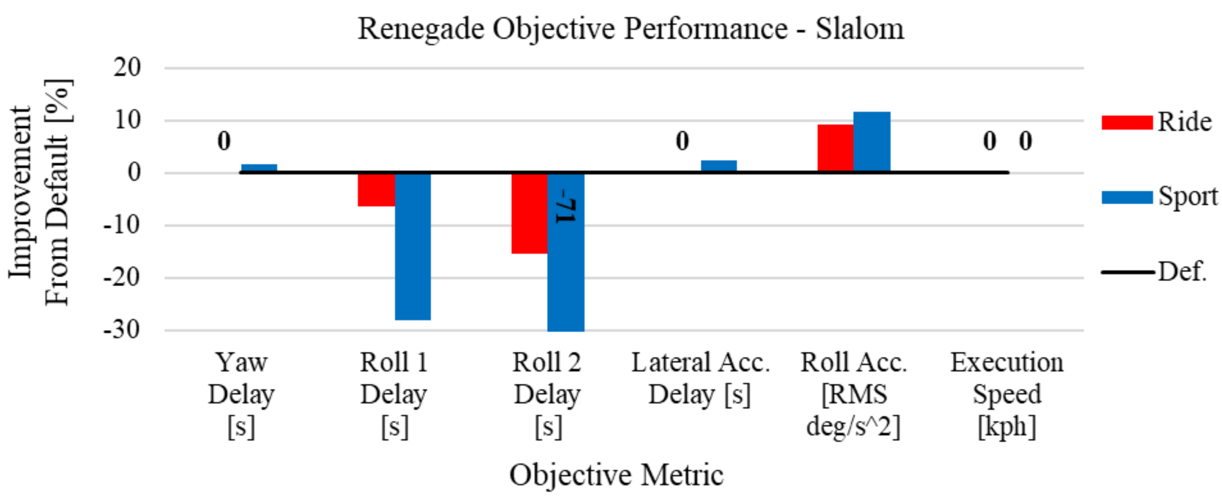

Figure 96: Renegade objective handling results on slalom

Nearly identical trends in the slalom objective performance between the two vehicle models were found, when implementing the two versions of the semi-active suspension controller. These trends are also similar to the objective performances found on the ISO DLC maneuver. However, the findings with the Renegade model generally have lower percentage improvements than the Grand Cherokee. The only metric to be significantly improved was the RMS roll acceleration for the sport mode (10% reduction), which was improved by 15% for the Grand Cherokee sport mode.

Subjective Ride and Handling Slalom Results for Primary Vehicle Model

The figure below contains subjective ride and handling metrics from the slalom maneuver while considering the Grand Cherokee vehicle model.

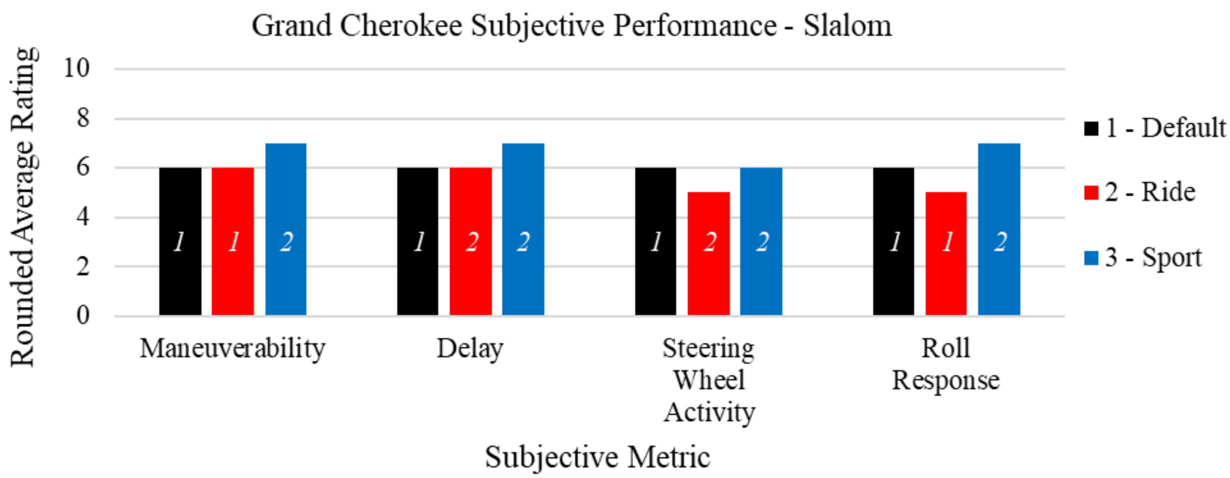

Figure 97: Grand Cherokee subjective handling results on slalom

The results of the subjective evaluation for the slalom maneuver depict a sport mode as perceived to have better maneuverability, a faster responding vehicle while having suppressed roll motion as compared to the default mode. In general, the drivers rated the ride mode similar to the default mode while having a slightly worse roll response and requiring more SWA input from the driver to execute the maneuver. However, the performance of all modes was accepted by the drivers as all ratings had a value above or equal to five. Objectively, the sport mode had shortened lateral acceleration and yaw delays. Thus, the improved delay subjective metric rating for the sport mode matches this result. Moreover, the sport mode reduced the RMS roll acceleration objective metric. This result could have led to the improved subjective rating for the roll response metric, which is related to how fast the vehicle rolls when maneuvering through the cones. Objectively, there was no difference in the SWA for all three suspension modes, thus it was unexpected that the ride mode had a lower subjective rating on the slalom. However, difference in driving behaviour between the three drivers cold lead to the difference in driver inputs, which would also impact the roll response of the vehicle. Finally, the drivers all completed the slalom at the same speed. This result matches the objective result.

Note that the sport mode ratings had a standard deviation of two for all metrics, whereas the default mode ratings always had a value of one suggesting that the drivers were in better agreement for the default mode's performance than the sport mode. The ratings for the ride mode had a standard deviation of one for the maneuverability and roll response metrics, and a deviation of two for the other metrics.

Subjective Ride and Handling Slalom Results for Secondary Vehicle Model

The figure below contains subjective ride and handling metrics from the slalom maneuver while considering the Renegade vehicle model.

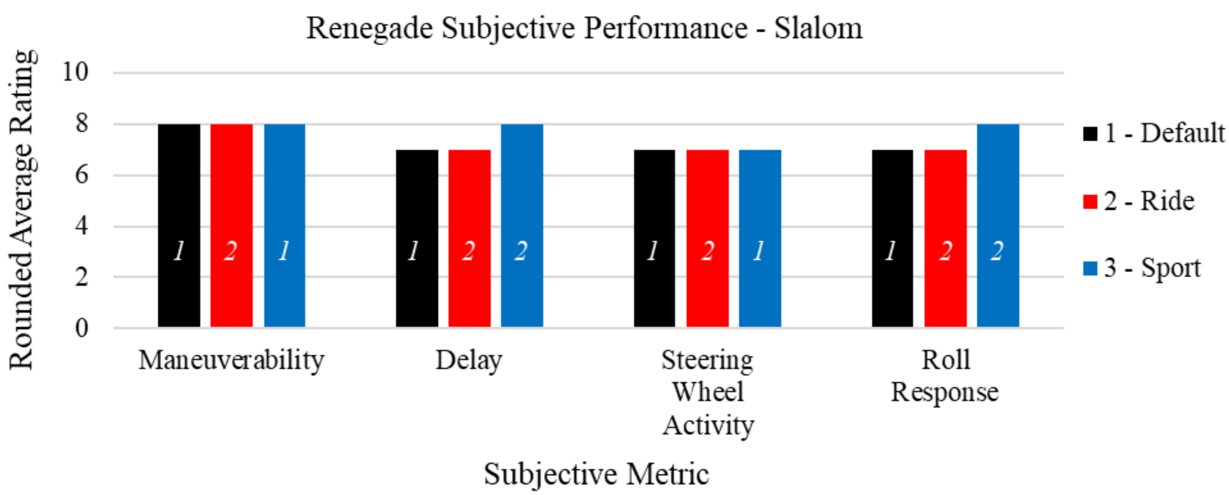

Figure 98: Renegade subjective handling results on slalom

The subjective evaluation of the Renegade semi-active suspension controller indicate the ride mode has having the same handling performance as the default. As the objective differences in handling for this maneuver were discovered to be relatively small, the result was expected. Furthermore, drivers commented that the ride mode had acceptable handling for quick and short maneuvers, especially for the slalom having less aggressive maneuvering compared to the ISO DLC. For the sport mode, the delay and roll response metrics were rated higher than the default. Driver comments also stated the sport mode as feeling "sporty" and having slightly better handling than the default mode. Although the objective difference in handling were limited, the yaw delay, lateral acceleration delay, and RMS roll acceleration were all improved with the sport mode.

Appendix H

Correlation Study Results Primary Vehicle Model Objective and Subjective Data

The following figures contain the objective and subjective ride and handling results from the correlation study on the primary vehicle model.

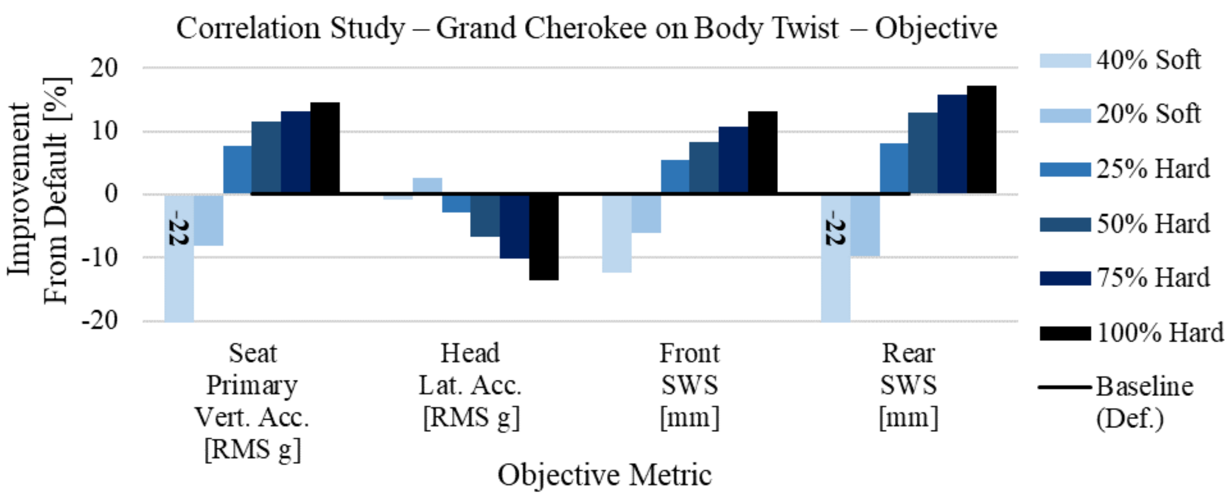

Figure 99: Grand Cherokee objective correlation results on body twist

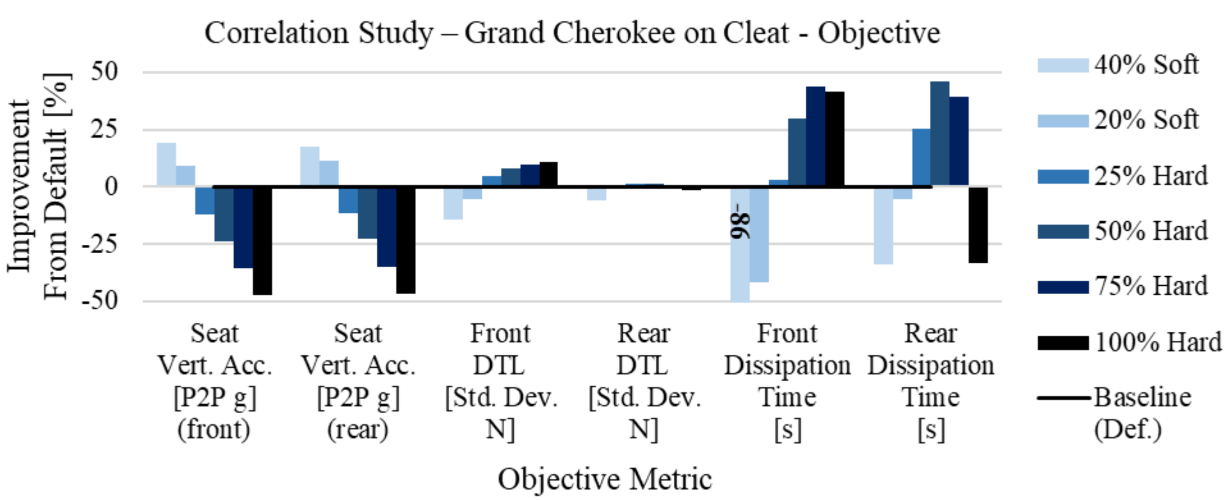

Figure 100: Grand Cherokee objective correlation results on cleat

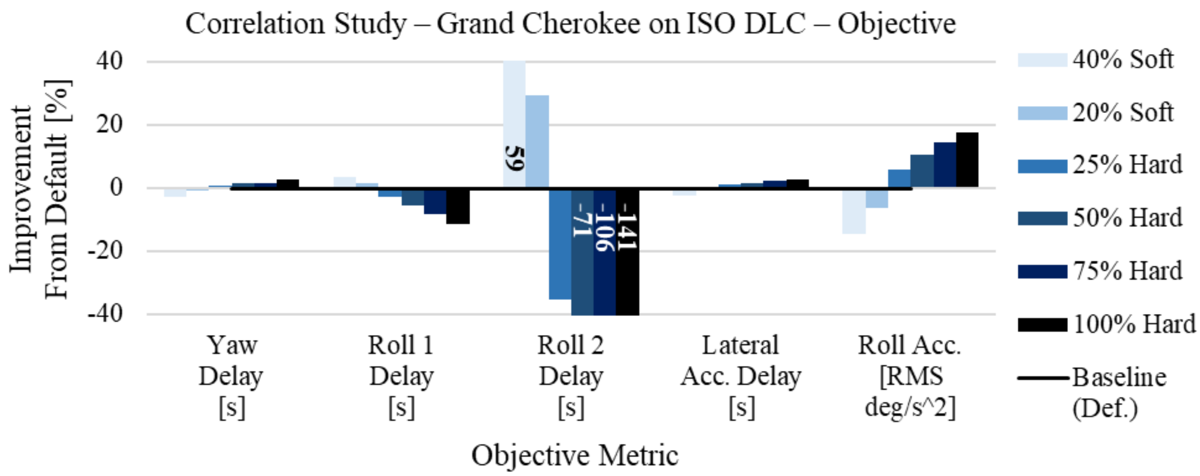

Figure 101: Grand Cherokee objective correlation results on ISO DLC

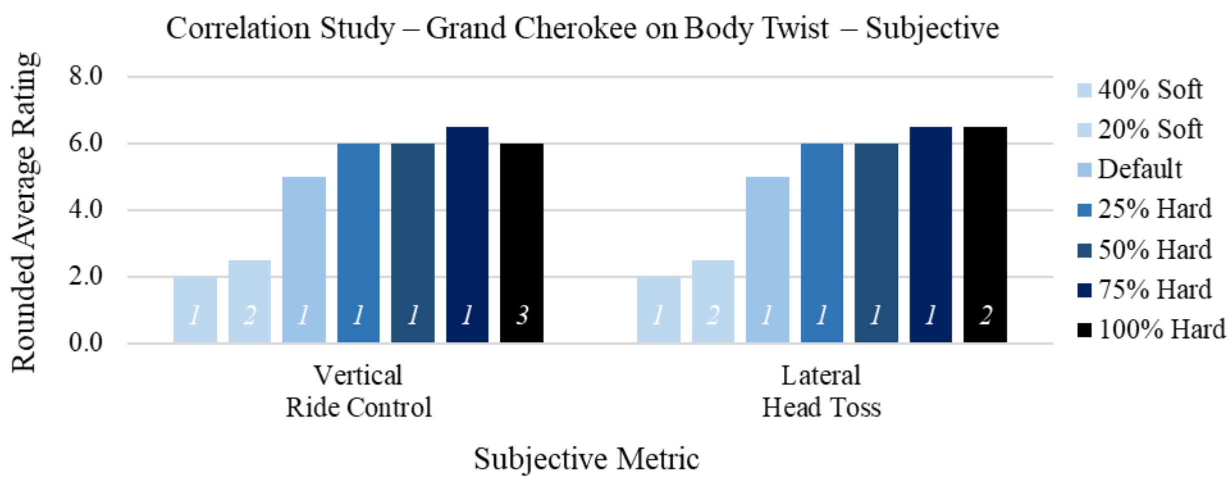

Figure 102: Grand Cherokee subjective correlation results on body twist

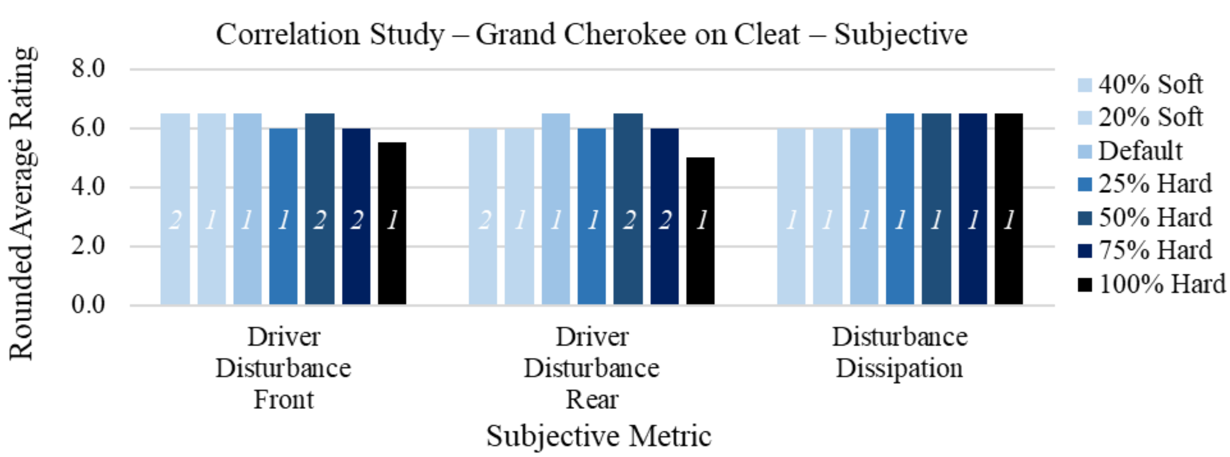

Figure 103: Grand Cherokee subjective correlation results on cleat

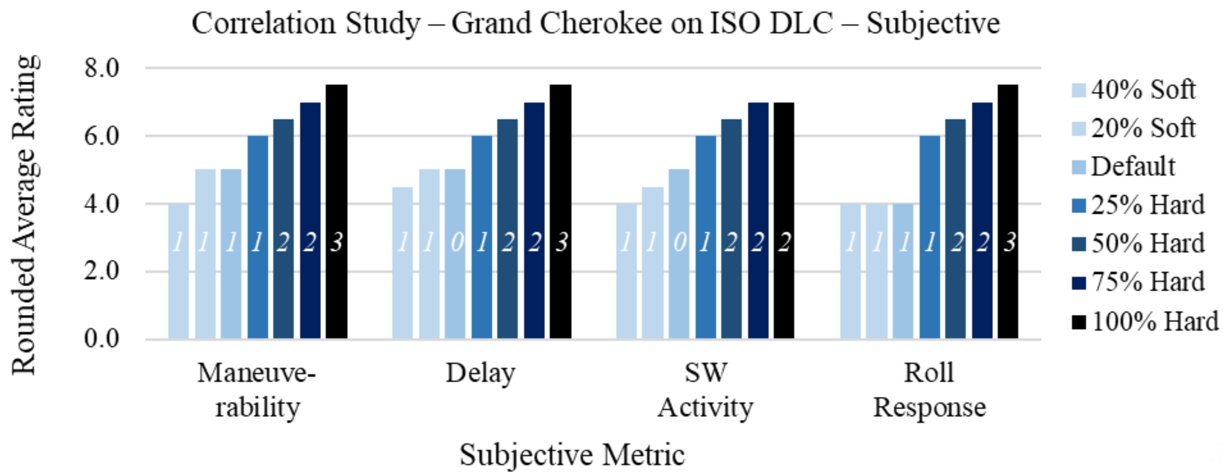

Figure 104: Grand Cherokee subjective correlation results on ISO DLC

Correlation Study Results Secondary Vehicle Model Objective and Subjective Data

The following figures contain the objective and subjective ride and handling results from the correlation study on the secondary vehicle model.

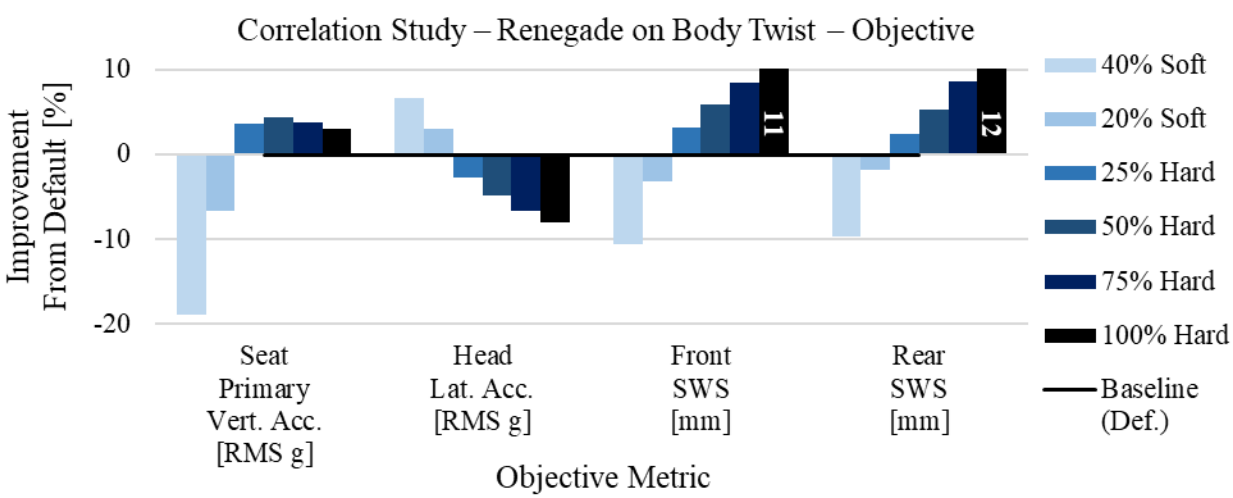

Figure 105: Renegade objective correlation results on body twist

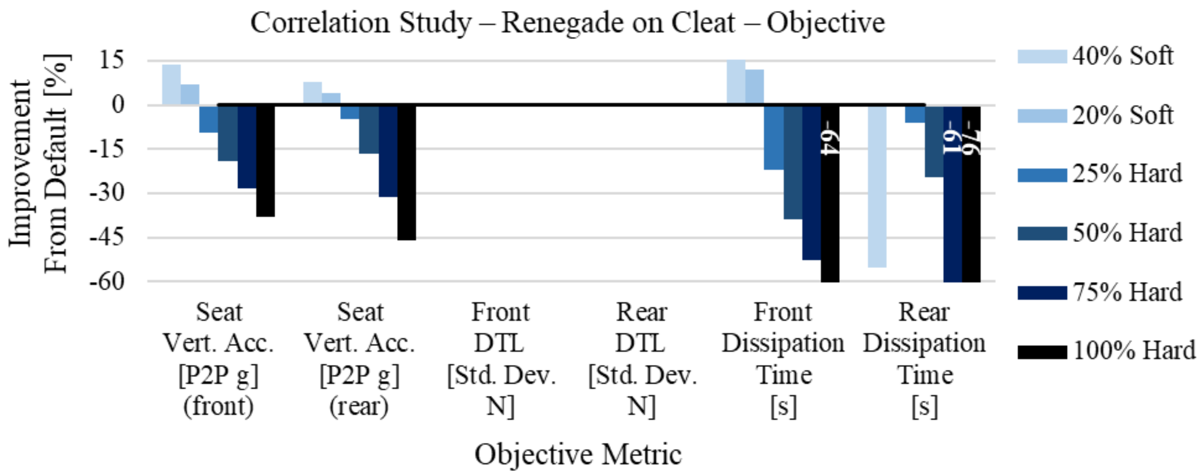

Figure 106: Renegade objective correlation results on cleat

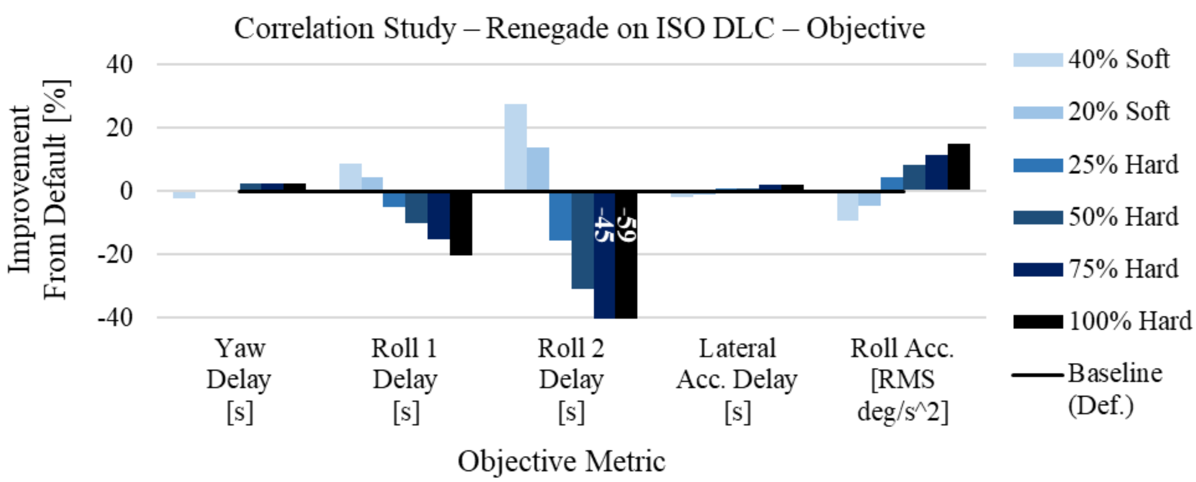

Figure 107: Renegade objective correlation results on ISO DLC

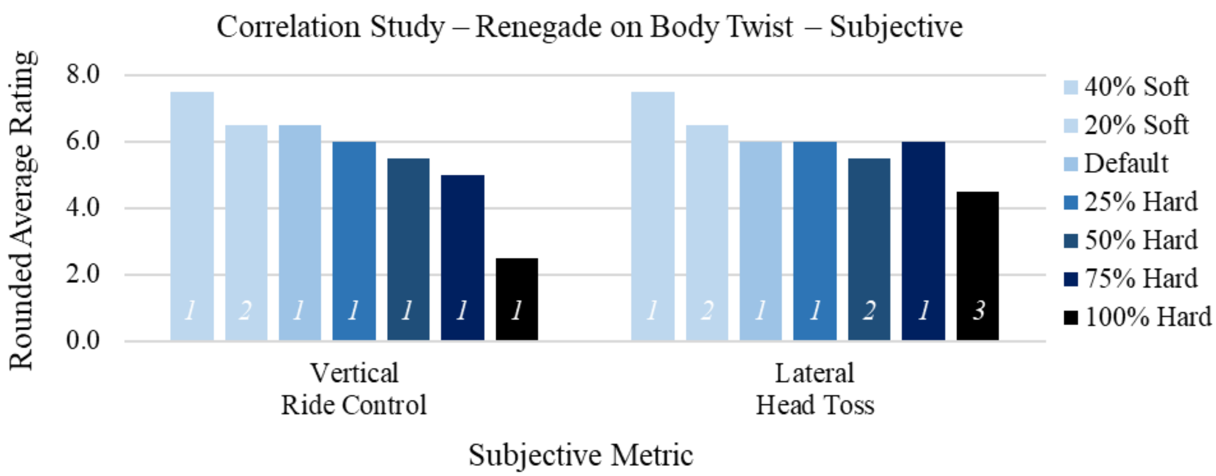

Figure 108: Renegade subjective correlation results on body twist

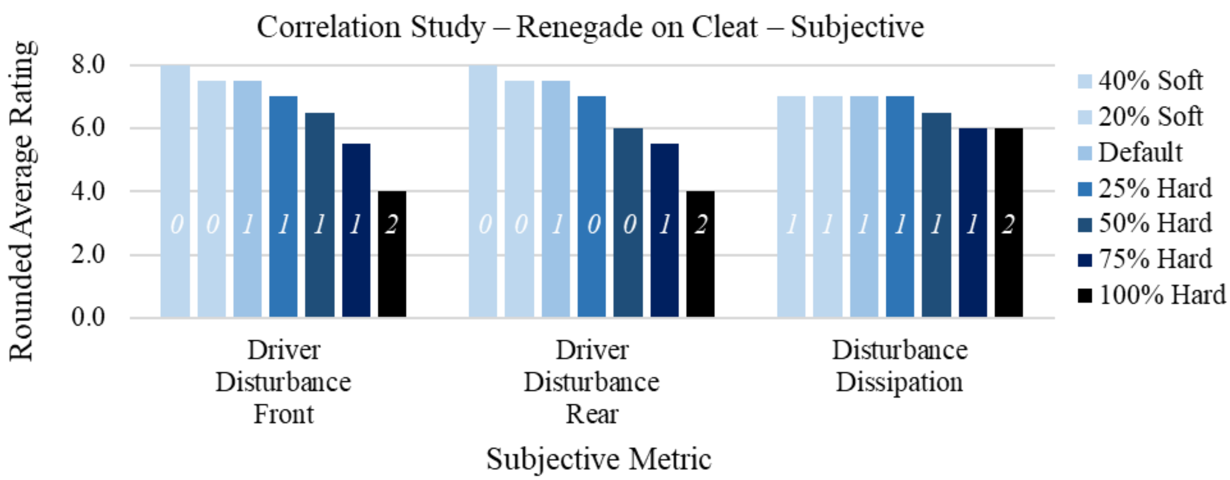

Figure 109: Renegade subjective correlation results on cleat

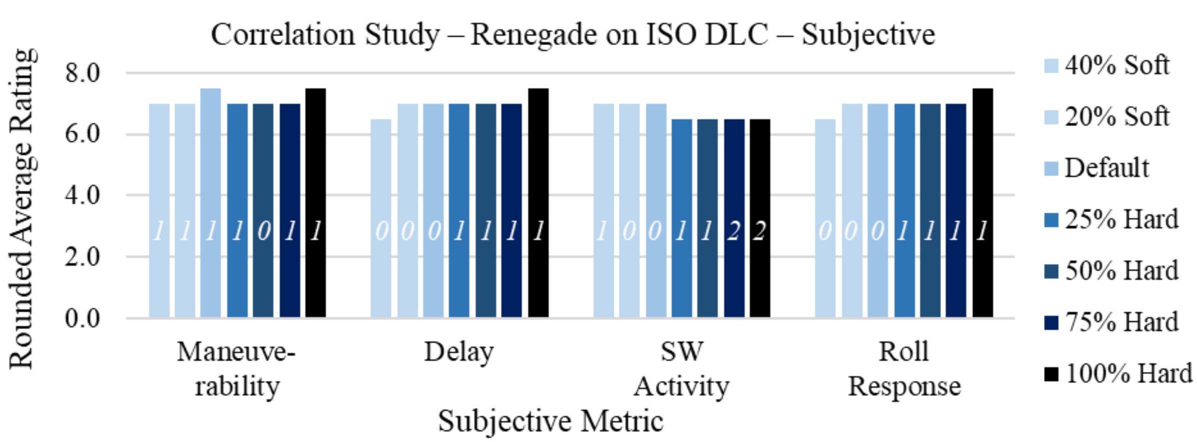

Figure 110: Renegade subjective correlation results on ISO DLC

Correlation Study Results Secondary Vehicle Model Linear Correlation Matrix

The following figure and tables present the correlation coefficients for all of the objective and subjective metric pairs and the significant correlation findings.

"R" Coefficient		OBJECTIVE METRICS														
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
SUBJECTIVE METRICS	A	-0.19	0.31	0.30	-0.26	-0.30	0.30	0.36	0.23	0.13	-0.26	-0.18	-0.24	0.29	-0.32	-0.34
	B	-0.76	0.80	0.79	-0.76	-0.80	0.79	0.78	0.72	0.19	-0.79	-0.69	-0.74	0.80	-0.84	-0.87
	C	0.77	-0.87	-0.87	0.88	0.88	-0.86	-0.77	-0.90	-0.39	0.80	0.80	0.76	-0.88	0.84	0.82
	D	-0.76	0.80	0.79	-0.76	-0.80	0.79	0.78	0.72	0.19	-0.79	-0.69	-0.74	0.80	-0.84	-0.87
	E	0.81	-0.95	-0.94	0.85	0.91	-0.95	-0.99	-0.93	-0.68	0.71	0.60	0.61	-0.88	0.88	0.91
	F	0.86	-0.96	-0.95	0.86	0.93	-0.97	-0.99	-0.95	-0.65	0.74	0.64	0.64	-0.90	0.90	0.93
	G	0.83	-0.90	-0.89	0.79	0.87	-0.91	-0.96	-0.91	-0.77	0.64	0.53	0.53	-0.83	0.82	0.84
	H	0.80	-0.92	-0.91	0.82	0.89	-0.93	-0.96	-0.89	-0.58	0.73	0.61	0.63	-0.87	0.88	0.91
	I	0.85	-0.88	-0.88	0.85	0.89	-0.87	-0.83	-0.83	-0.25	0.86	0.79	0.82	-0.89	0.91	0.92

❖ R ≥ 0.85

❖ 0.7 < R ≤ 0.85

❖ 0.5 < R ≤ 0.7

❖ R < 0.5

Figure 111: Linear correlation results with the Renegade

Table 17 presents the most significant correlation findings with the secondary vehicle model. Recall that the R-values in the third column with an asterisk pertain to the highlighted metric pairs from Table 10. These pairs correspond to subjective metrics that were created to address the same ride or handling aspect as one of the objective metrics.

Table 17: Significant Correlation Results Found with the Secondary Vehicle Model

Objective Metric(s)	Subjective Metric(s)	R-Value(s)	Increasing Damping Rates
7. P2P Seat Acc. (rear)	**F**. Driver Disturbance – Rear	-0.99*	As 7 increases, F worsens
6. P2P Seat Acc. (front)	**E**. Driver Disturbance – Front	-0.95*	As 6 increases, E worsens
13. RMS Head Lat. Acc.	**I**. Head Toss	-0.89*	As 13 increases, I worsens
5. RMS Roll Acc.	**D**. Roll Response	-0.80*	As 5 decreases, D improves
8/9 Dissipation Times	**G**. Disturbance Dissipation	-0.77/-0.91*	As 8/9 increases, G worsens
1/4	**B**. Delay	-0.76*	As 1/4 decreases, B improves
15. Rear SWS	**F**. Driver Disturbance – Rear	0.93	As 15 decreases, F worsens
14. Front SWS	**E**. Driver Disturbance – Front	0.88	As 14 decreases, E worsens
14/15	**B**. Delay	≤-0.84	As 14/15 decreases, B improves
5/12/14/15	**I**. Head Toss	≥0.82	As 5/12/14/15 decreases, I worsens
5/14/15	**D**. Roll Response	≤-0.8	As 5/14/15 decreases, D improves
2/3	**C**. Steering Wheel Activity	-0.87	As 2/3 increase, C worsens

One major difference in the correlation study results with the secondary vehicle model is the front and rear tires' response on the cleat maneuver. For all damping settings, the vertical tire forces abruptly increase upon impact with the cleat and immediately drop to zero momentarily before quickly returning to the steady state values. Thus, as the damping was increased, the vertical tire force response appeared to become overdamped with the harder damping settings. Upon observing the objective metric trends on the cleat maneuver in Figure 106, neither of the DTL objective metrics are affected whereas the front and rear dissipation times worsen with the higher damping. This was a result of the over-damping of the vehicle. Furthermore, subjective metrics in Figure 109 for the driver disturbance on the cleat degrade to unacceptable ratings with the highest damping. This result could attribute to the worsening of the vertical ride control "H" (combined primary and secondary vertical ride) subjective metric on body twist, which was the opposite case found with the primary vehicle model. Here, the worsening of the secondary ride could dominate the slight objective improvement in the primary ride displayed for the secondary vehicle model in Figure 105. As a result, the correlation coefficient values for pairs including objective metrics "8" and "9" as well as subjective metric "G" have the opposite signs as the results from the study on the primary vehicle model. For subjective metric "C" pertaining to the steering wheel demand for the driver on the ISO DLC, it only changed by half of a rating between the softest and hardest setting. Thus, correlations with this metric were deemed insignificant.

Despite these difference in the correlation results due to the significant tradeoff between primary and secondary ride, many practical correlation results were discovered as identified in Table 17. Each of these correlation coefficients from the pairs in Table 17 have an absolute value above 0.7, indicating that a strong correlation exists. These correlations provide clear indication on which objective metrics should be considered when tuning vehicle dampers offline. Furthermore, linear regression is applicable in the same way as presented in Section 9.3.2. Here, linear regression is applied to the metric pair "13-I" corresponding to the objective head level lateral acceleration and head toss metrics. Figure 112 presents the regression line which could be used to predict future subjective ratings with similar damping settings on the secondary vehicle model.

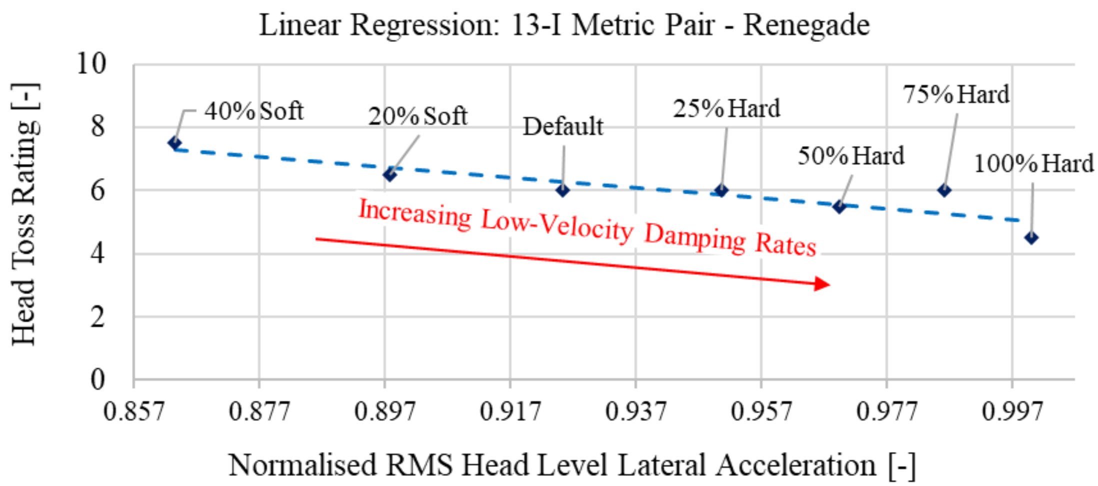

Figure 112: Linear regression for one of the correlation study findings with Renegade

The linear regression line in Figure 112 has a mean square error and average absolute error in the ratings from the regression line of approximately 0.14 and 0.31, respectively. This result indicates an accurate prediction method for observing this particular vehicle model and altering low-velocity damping rates. The same approach could be repeated for all other highly correlated metric pairs to predict other subjective ratings when tuning the secondary vehicle model's dampers if future alterations are necessary. For similar subcompact SUV models, the fact that a strong correlation exists suggests that consideration on improving the head level lateral acceleration will lead to an improvement in the head toss rating during the subjective evaluations.

VITA AUCTORIS

NAME:	Zachary Sinasac
PLACE OF BIRTH:	Windsor, ON
YEAR OF BIRTH:	1996
EDUCATION:	General Amherst High School, Amherstburg, ON, 2014
	University of Windsor, B.A.Sc., Windsor, ON, 2019
	University of Windsor, M.A.Sc., Windsor, ON, 2021