Detecting Defective Suspension Components

Master's thesis in Automotive Engineering

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Department of Mechanics and Maritime Sciences
Chalmers University of Technology
Gothenburg, Sweden 2020

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Abstract

In order to maintain the performance of the vehicle, detecting defective components and then informing the driver which parts should be changed is becoming more important. The target is to inform the driver through vehicle information system, when major components that are subject to wear and need changing. The area of focus in this thesis is Suspension components, e.g. dampers, springs and bushes. The plan is to detect defective suspension components by using the change in the transfer function from new to worn components and the target is to find a way to measure the change in transfer function and then inform the driver when it is time to change components.

Keywords: Defective components, Suspension components, Springs, Dampers, bushes, Macpherson, Transfer function, Wheel travel, MATLAB, Simulink, Vehicle dynamics, Vertical dynamics, Ride comfort

Acknowledgements

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Göteborg, September 2020-09-23

Anukaran M Patne | Hemanth Yadav Yekanthappa

Dedicated to

Our Parents for their constant love and support

Nomenclature

C_s	Supension damping in Ns/m
C_t	Tyre Damping in Ns/m
K_s	Spring Stiffness in N/mm
K_t	Tyre stiffness in N/mm
M_s	Sprung mass in Kg
M_{us}	Unsprung mass in Kg
V	Vehicle velocity in m/s
X	Distance in m
Z_u	Unsprung mass displacement in m
Z_s	Sprung mass displacement in m
A_u	Unsprung mass acceleration in m/s^2
A_s	Sprung mass acceleration in m/s^2
K_p	contact stiffness at upper bound
K_n	contact stiffness at lower bound
D_p	contact damping at upper bound
D_n	contact damping at lower bound
g_p	initial gap between the slider and upper bound
g_n	initial gap between the slider and lower bound

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Introduction

1.1 Background

Suspension is a system of subframes, linkages, control arms, springs, antirollbars, dampers and bushes that connects the wheels to the body structure and allows relative motion between the two. Important attributes for the Suspension system includes vehicle dynamics and ride comfort.

Suspension performance depends partly on the condition of spring and damper. Dampers and springs are in place to give the expected vertical control of the wheel and the body to suit the attribute targets for the vehicle. Dampers are used to dampen the movement of the wheels relative to body structure. The properties of the dampers is important because, they control the oscillation of the wheels. A damper that is tuned to suit the vehicle and the attribute targets controls the wheel and the body well, while a worn damper with less damping forces is not able to control the dynamics so well.

Suspension system plays an important role in vehicle handling and ride comfort of the vehicle. Hence, it is necessary to maintain it in a good working condition. In order to maintain the performance of the vehicle we need to be aware of the defective or worn components in the system, so that we can inform the driver about the same. Hence, it is required to develop a method to detect the change in the performance of the suspension, and by measuring the change we conclude if a change of component is necessary. By doing so the driver will always enjoy the ride and, the performance of the vehicle is maintained to optimal levels. [2]

1.2 Literature study

To start with the work in the master thesis project, it is important to understand the suspension system and the functionality of its various components. Suspension being one of the important systems in a vehicle, contributing mainly to handling performance and ride comfort, we must understand the suspension system and its components in detail. We had a basic understanding of suspension as it was a part of our course Vehicle dynamics in our master’s studies. We referred to lecture notes on suspension system by professor Ingemar Johansson.

And further knowledge about MacPherson suspension and damper characteristics was obtained from the textbook "Vehicle Dynamics" by Martin Meywerk [1]. To notice the change in damper characteristics we also needed to have in depth knowledge on damper properties and functions. The research paper "Analysis of damper data and design of portable measurement system" by Grant A. Malmedahl [2] helped us to understand damper and their importance in suspension system and the effect of malfunction of damper on vehicle. In order to get information regarding the defects in automotive components, research paper "Fault Detection in Automotive Semi-Active Suspension: Experimental Results" by Ruben Morales-Menendez, Jorge Lozoya-Santos, Diana Hernandez-Alcantara and John Jairo Martinez Molina was found to be very useful. Wherein a proposal of fault detection using Linear parameter Varying (LPV) systems is experimentally validated on embedded systems. These results open the application of LPV approaches to commercial vehicles since it is easy of implementation for several features such as low computation load, lumped parameter model and available for nonlinear dynamic systems [3]

1.3 Method

It starts with the literature study wherein the plan is to go through a number of research that has already been done on detecting defects in automotive components especially on suspension. It is also important to get insight about various components that go into suspension system and understand their function. Followed by the literature study the plan is to build two quarter car models (one basic and the other representing a vehicle) using MATLAB and Simulink. The quarter car models will be focusing on the front left corner of the vehicle. The suspension we are studying in this research is a "MacPherson suspension" and once the model is built, the next step is to obtain the nominal transfer function for the suspension system in good working condition on a given road profile. The next step involves replacing the damper with worn damper and obtaining the transfer function for the same road profile as earlier and observing the change in two transfer functions. After successful interpretation of this change in transfer function the plan is to come up with a method to measure the change and inform the driver through vehicle information system if change of component is necessary.

1.4 Deliverables

The objective of this master thesis project is to study how reduced performance in suspension components, e.g. dampers, can be measured on the vehicle by understanding the transfer function between the wheel and the body structure. A complete study of suspension system with a good damper as well as a worn damper should be done and identify the change in the transfer function. Based on the changed performance, a method should be implemented to measure that change. Finally, the driver should get the notification about the worn component.

- A model of a suspension that can be used to analyse the relation between wheel movement and body structure movement in different driving conditions.
- Present the difference, in the wheel movement and body structure movement, between new and worn components, e.g. dampers.

- Develop a transfer function that in a robust way can quantify the difference in movements between new and worn components.
- Develop a way to measure the proposed transfer function in the vehicle and propose a way to inform the driver.

1.5 Limitations

- MATLAB and Simulink will be used to simulate the transfer function between the wheel and the vehicle body. This will require that the model is simplified.
- The simulation will be run to develop a transfer function for the suspension system. The parameters in the simulations need to be selected to suit the suspension system and the vehicle.
- The initial model will only represent the front left corner of the vehicle and it cannot be applicable to the rear axle directly without any changes or modifications.
- No physical testing will be done.
- No integration of electrical and electronics will be done.
- The simulation will use a simplified model of the tire characteristics and a simplified model of top mount characteristics.

2

Theory

2.1 Suspension components

Before we begin with detecting defects in suspension components it is important to understand what goes into a suspension system. A suspension system consists of a number of components such as spring, damper, ball joints, bushes, Antiroll bar, knuckle etc. Each component has a unique design and functionality but when it comes to reducing the vertical disturbances that come from the road there are two main components which are primarily responsible and it's the spring and the damper system.

In the vehicle front corner model, we have only considered the spring, bumpstop and damper to represent the suspension system. Where in the spring show a linear behaviour and the damper and bumpstop has a nonlinear behaviour (this will be discussed in the later chapters).

To complete the quarter car model, we have a spring and a damper system representing a simple tyre model. Below can be seen a schematic representation of the quarter car model.

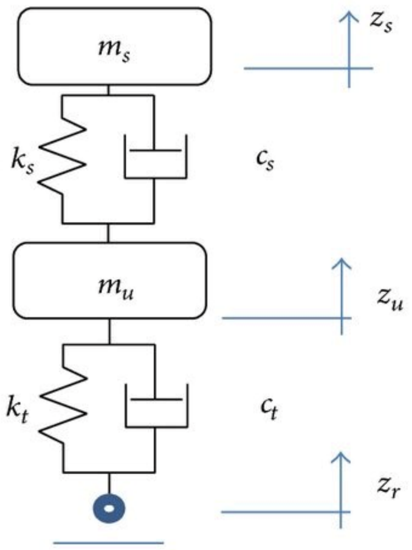

Figure 2.1: Quarter car model

2.2 Basic front corner

The basic front corner is a simplified linear model that consists of a spring (k1) and a damper (b1) representing the suspension system and a spring (k2) and a damper (b2) representing the tyre and, mass 1 and mass 2 representing sprung and unsprung mass respectively.

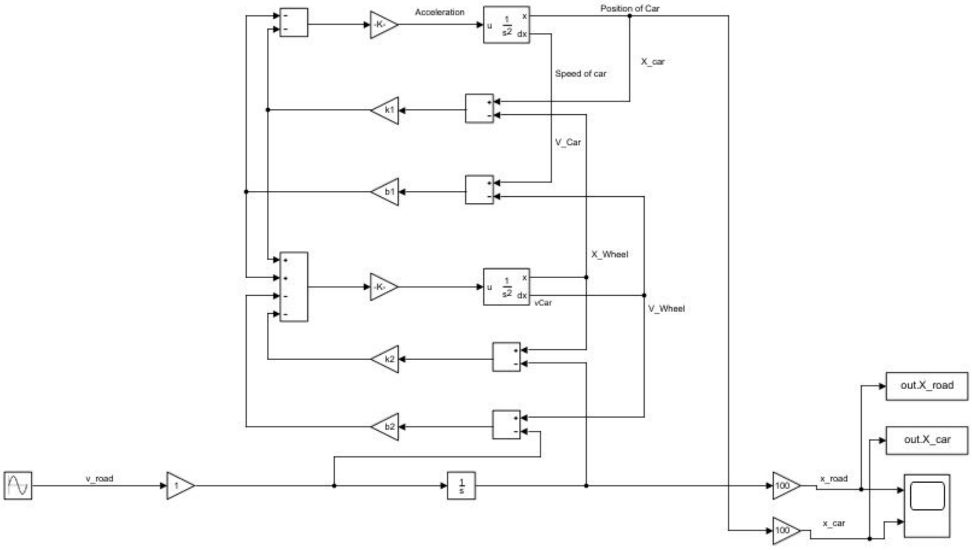

Figure 2.2: Basic quarter car model

The following was built inorder to understand the relation between wheel and body movement in a linear system consisting of linearly behaving components (spring and damper).
Below are the parameters and their respective value assigned in the system in order to test the simulation model:

Parameters	**Value**
M_s	$240\ Kg$
M_{us}	$36\ Kg$
K_s	$16\ N/mm$
K_t	$160\ N/mm$
C_s	$980\ Ns/m$
C_t	$100\ Ns/m$

Table 2.1: Parameters defined in the basic car model

The road input is a sine wave and when the simulation is run, we get an output that depicts the position of the sprung mass with respect to the position of the road.

The below figure is the output of the system when run for a time of 5 seconds.

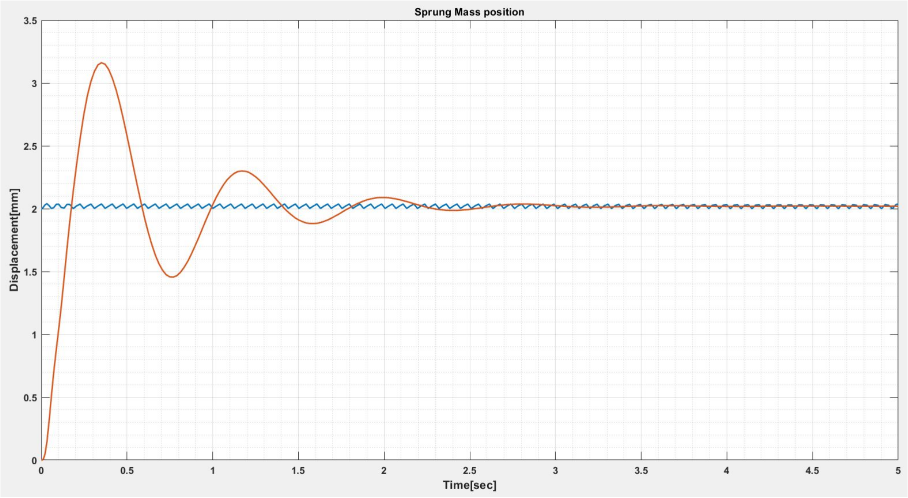

Figure 2.3: Position of the sprung mass with respect to road

As we can see the response of the system (blue line) is a sine function due to the input being a sine wave. The sprung mass oscillates as a sine function and it gradually dampens as it moves across the 5 second timeline.

This representation of the general front corner of the vehicle was a rather simple and a very linear representation which was used to get a basic understanding of how the model can be built in Simulink, and to understand the behaviour of the vehicle to different inputs such as sine wave or step input.

But, in order to study a behaviour that is closer to reality, we had to develop more sophisticated model that consisted of components exhibiting a nonlinear behaviour (Damper, Bumpstop). And it is these components we are interested in detecting the defects, starting with the Damper.

3

Methods

3.1 Vehicle front corner

The Vehicle front corner is built in a similar way as the general front corner with difference being the inclusion of a non-linear translational damper and a bumpstop. And all parameters of the vehicle such as the sprung mass, unsprung mass, spring rate etc, representing the actual values of a vehicle . The below figure shows the model setup.

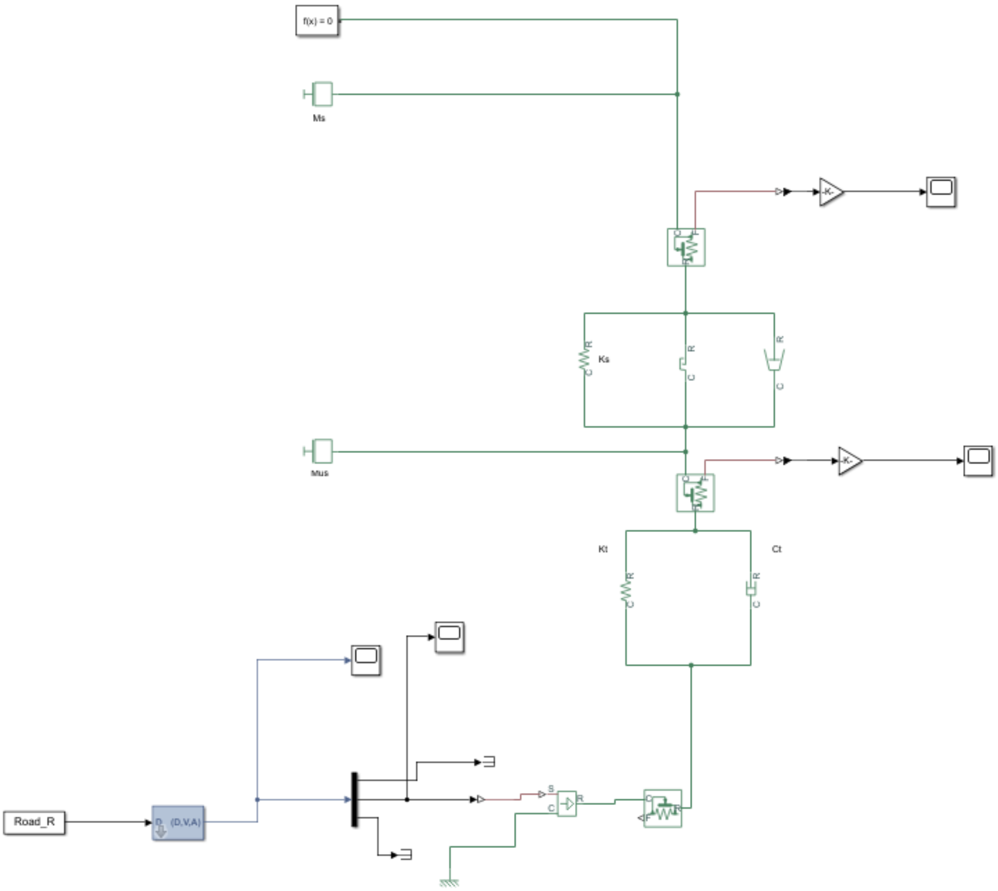

Figure 3.1: Vehicle Front Corner model

We use Simscape to model this front corner due to its flexibility when it comes to modelling and simulating the multi domain physical systems, it enabled us to

rapidly create models of physical systems allowing us to use a number of sensors such as “force sensor”, “motion sensor” etc and also its flexibility when it comes to allowing us to deploy our model into other simulation environment such as hardware in loop (HIL) and C-code generation[7].
The following are the parameters and their respective value assigned in the system:

Parameters	**Value**
M$_s$	566 Kg
M$_{us}$	54 Kg
K$_s$	28 N/mm
K$_t$	160 N/mm
C$_t$	100 Ns/m

Table 3.1: Parameters defined in the Vehicle quarter car model

3.1.1 Nonlinear damper

The damper is at the centre of our attention in this master thesis as it happens to be the first element in the suspension system we are interested in detecting a defect.

The automotive industry most commonly uses single and twin-tube shock absorbers or dampers. The movement of piston rod causes the shock absorber piston to move up and down. During these movements the oil flows through either compression valve or rebound valve [1].

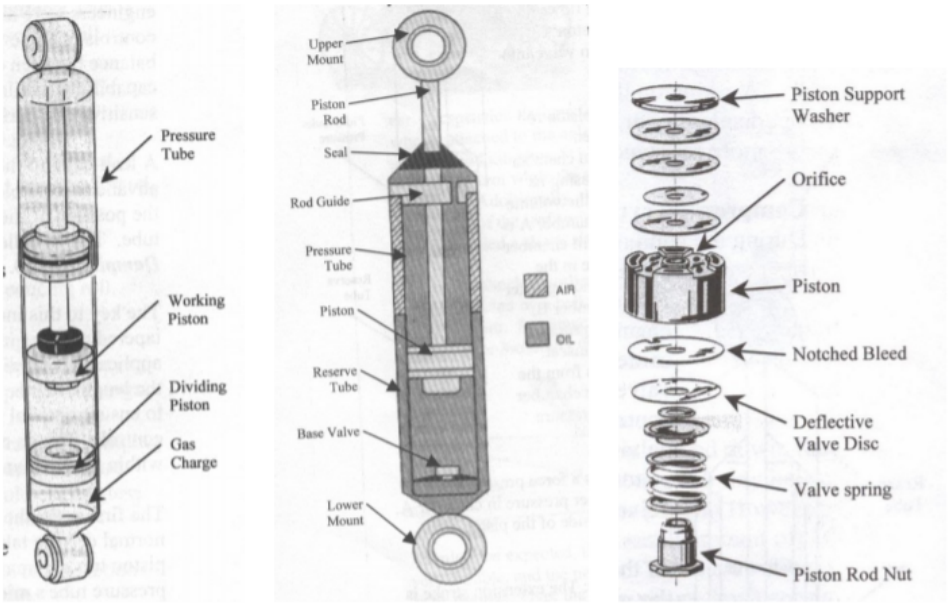

Figure 3.2: From left to right: Mono tube, twin tube and typical valve construction [2]

In order to capture a more realistic behaviour of damper in our system we had to consider its nonlinear behaviour in the modelling stage.
The way we capture this nonlinear behaviour of a damper is by plotting the force Vs velocity curve for a particular damper and it is through these curves we can extract different damping coefficients which then encapsulate the nonlinear behaviour of the damper in our system.
The below table gives the force for the respective velocity which was then plotted to get the damper curve for a Nominal damper.

Velocity (m/s)	0.05	0.1	0.3	0.6	1	1.2	1.5
Rebound	421	735	1303	1649	2127	2362	2744
Compression	284	343	461	627	862	1000	1196

Table 3.2: Force and velocity vector for a Nominal Damper in a Vehicle

The below figure is the curve for the Force Vs Velocity of a Nominal damper with blue line showing compression and the red line showing rebound. It can be seen that the rebound forces are almost twice the compression forces for any given velocity.

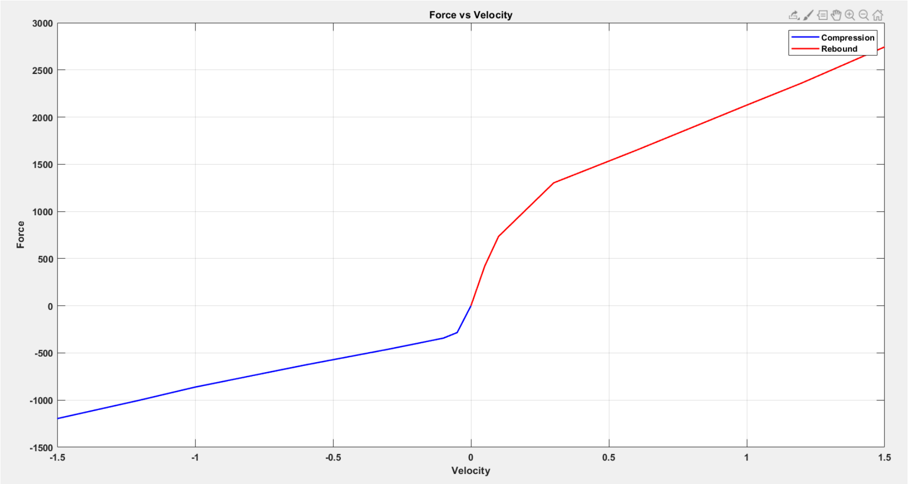

Figure 3.3: Force Vs Velocity curve for a Nominal Damper

3.2 Damper degradation

Damper degradations increases wear inside the damper i.e valves and piston, resulting in an increase in the clearance between piston and the cylinder walls causing leakage and, thereby reducing the damping force. There are a number of valves and orifices inside the damper and the oil is made to flow through, in order to achieve pressure and eventually the damping force. One of the reasons for damping force degradation is deformation of these valve springs, valve disc and orifices etc.

Cavitation can be considered as a temporary malfunction as it is mostly affected by erratic driving. During cavitation there is formation of air bubbles in the damping oil and its harmful because air acts differently from oil when passing through orifices. Cavitation is caused when fluid flows across a restriction, and the differential pressure across the restriction is great enough that the downstream pressure falls low enough to pull the dissolved air out and create a bubble. Then when the system equalizes again, the bubbles collapse. Cavitation reduces the damping force which as a consequence affects the ride comfort and safety [4].

Valve malfunction can occur due various reasons such as faulty installation, wear due to continual driving in harsh road conditions and this also leads to a reduced damping force eventually compromising on the comfort and safety.

There are different levels to which a damper can be considered degraded. In our work we have considered two dampers that are degraded by 20% and 30%, which means the damping force of these dampers is reduced by 20% and 30% respectively in comparison to the nominal damper. The table below shows the rebound and compression forces for the degraded dampers.

Velocity (m/s)	0.05	0.1	0.3	0.6	1	1.2	1.5
Rebound	336	588	1024	1319	1701	1889	2195
Compression	227	274	368	501	689	800	956

Table 3.3: Force and velocity vector for 20% degraded damper in a vehicle

Velocity (m/s)	0.05	0.1	0.3	0.6	1	1.2	1.5
Rebound	294	514	912	1154	1488	1653	1920
Compression	198	240	322	438	603	700	837

Table 3.4: Force and velocity vector for 30% degraded damper in a vehicle

The below figure shows the force vs velocity curve for the degraded dampers in comparison with the nominal damper. The red line represents the nominal damper, followed by green line representing 20% degraded damper and then blue representing the 30% degraded damper.

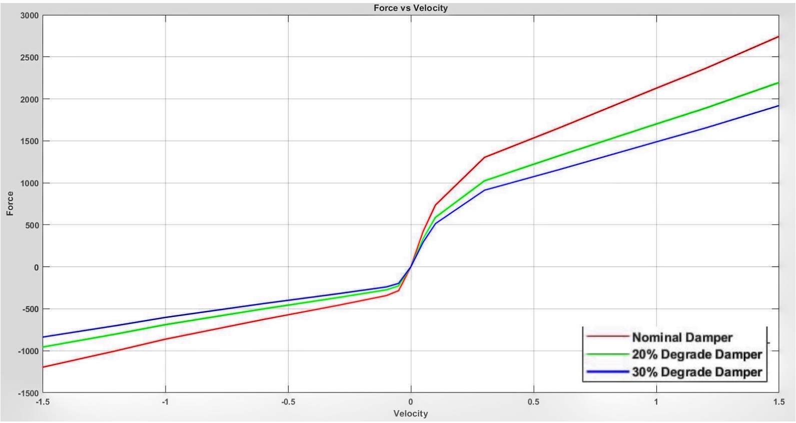

Figure 3.4: Force Vs Velocity curve for different dampers

3.3 Bumpstop

The bumpstop implemented in the model represents double sided mechanical translational hardstop, which can be found in the simscape library. It restricts the motion of the body between upper and lower bounds. The interaction between slider and the stop is elastic which means the stop is represented as a spring that comes into contact with the slider as gap is cleared and opposes slider penetration into the stop with force linearly proportional to the penetration. To account for energy dissipiation and non elastic effects, damping is introduced as the blocks parameter thus making it possible to account for energy loss. The schematic shows the idealization of the mechanical translational hard stop adopted in the block.

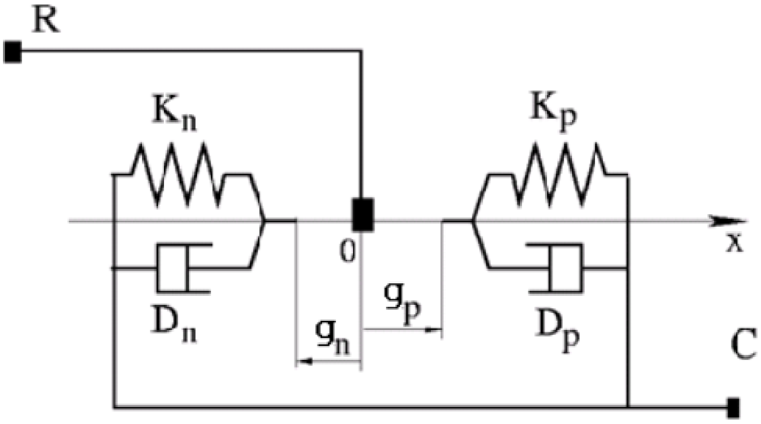

Figure 3.5: Bumpstop schematic representation

Where
K_p is contact stiffness at upper bound
K_n is contact stiffness at lower bound
D_p is contact damping at upper bound
D_n is contact damping at lower bound
X is the slider position
g_p is the initial gap between the slider and upper bound
g_n is the initial gap between the slider and lower bound
R is the slider

The bumpstop model operates such that stiffness and damping are applied smoothly through the transistion region, damped rebound. The slider travels smoothly through the transition region, the block smoothly ramps up the force from zero to full value. At the end of transition region, full stiffness and damping are applied during jounce/-compression. On the rebound, both stiffness and damping forces are decreased back to zero.

The below figure shows the nature of the suspension system with the spring and the bumpstop that is implemented in the quarter car vehicle.

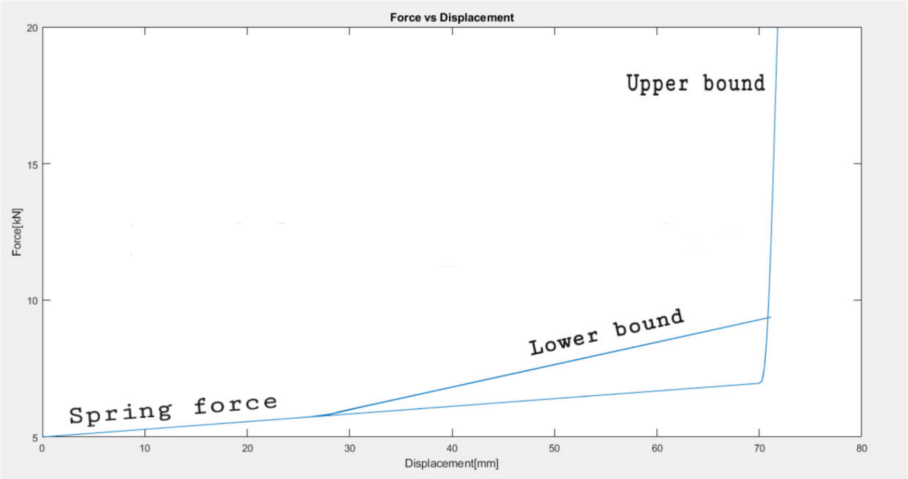

Figure 3.6: Spring + bumpstop curve

The bumpstop function in the vehicle is described in two regions, first the low stiffness region (lower bound) that is tuned to absorb energy and second, the high stiffness region (upper bound) which is present to control the wheel travel. The free travel is important for the ride comfort.

The spring force acts on the system until the spring is compressed to 27mm and after 27 mm the bumpstop comes into play. The bumpstop shows a linear behaviour upto 70 mm wheel travel which is the lower bound (soft region), and after 70 mm there is a sudden nonlinear increase in the stiffness which is the upper bound of the bumpstop. It is in this region the bumpstop has a higher stiffness to control the wheel travel.

3.4 Road Profile

The road profile for the simulations where defined by ISO 8608. This standard is categorised by roads with different levels of harshness from A to H, with A being the smoothest road with lowest elevations and H being the harshest road with the highest vertical elevations. Power spectral density is used as a basis for this classification. In his study Simulated road profiles according to ISO 8608 in Vibration Analysis, Peter Mucka concludes that A and B road profiles maybe recommended as a representative of a typical road profile that corresponds to road functional category such as motorway or first-class roads. Country or district roads can be classified as B or C class roads and Road classes from D to H often corresponds to unpaved roads [5].

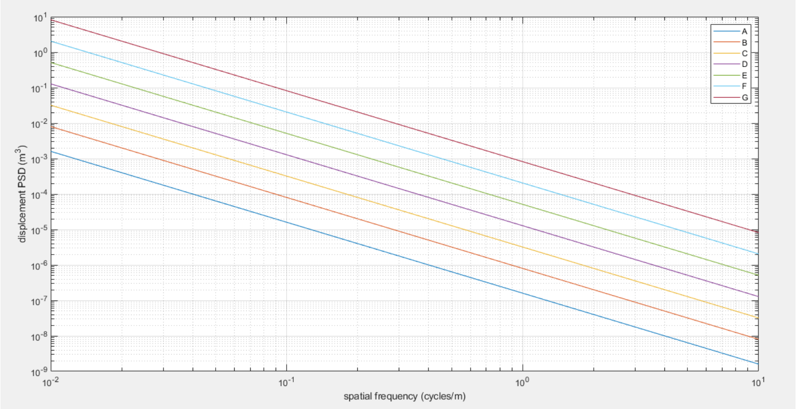

Figure 3.7: Classification of roads according to ISO 8608

The road profile is represented mathematically in work using a MATLAB script that takes the power spectral density as input and gives road elevations over the entire length of the road as output. These data points have been obtained from the ISO 8608 standard.

We have considered a range of road profiles to study the behaviour of the system, starting with road profile D and E which has elevations between $\pm 0.04\ m$ and $\pm 0.08m$ respectively and these elevations are spread over a 100 meters road, which is fed as an input to the front corner of the vehicle and the output is the disturbance in the system which is observed for various different speeds of the vehicle. Simulations were carried out on the harsher end of the road profile spectrum as well by choosing the road profile 'G'. Profile G has elevations between $\pm 0.3m$ and it was observed that the sprung mass accelerations where reaching 80-100 m/s^2 (close to 10 G's) due to the 30 cm elevations in the road. Such accelerations are practically impossible and never experienced, given the road conditions we see on an everyday basis. Smoother road surfaces such as AB were neglected because they have very negligible elevations and due to such small elevations, it would be difficult to capture the role played by the damper.

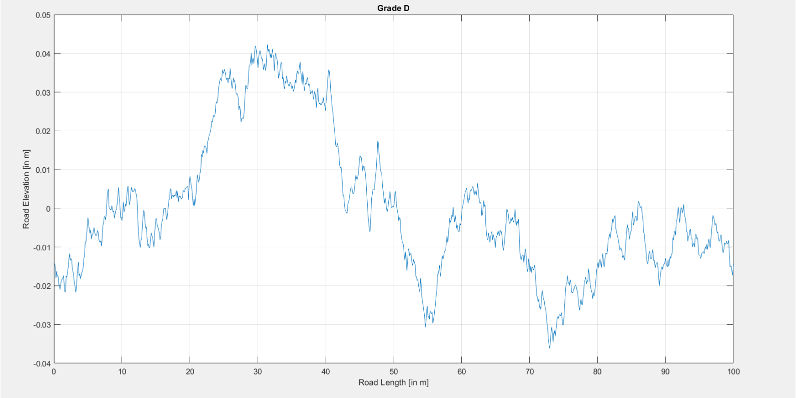

Figure 3.8: Road profile D

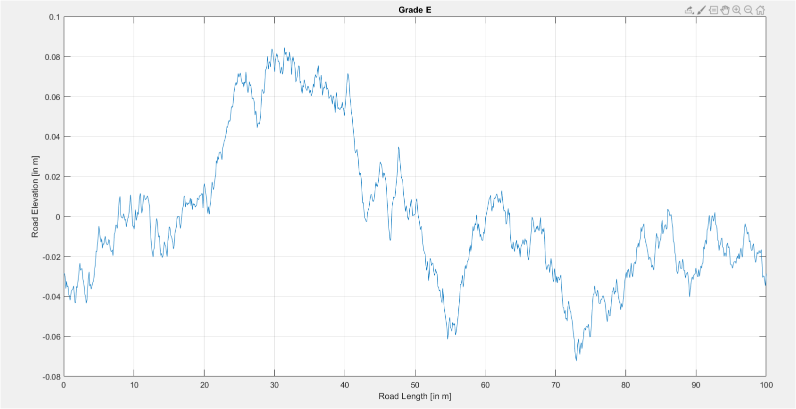

Figure 3.9: Road profile E

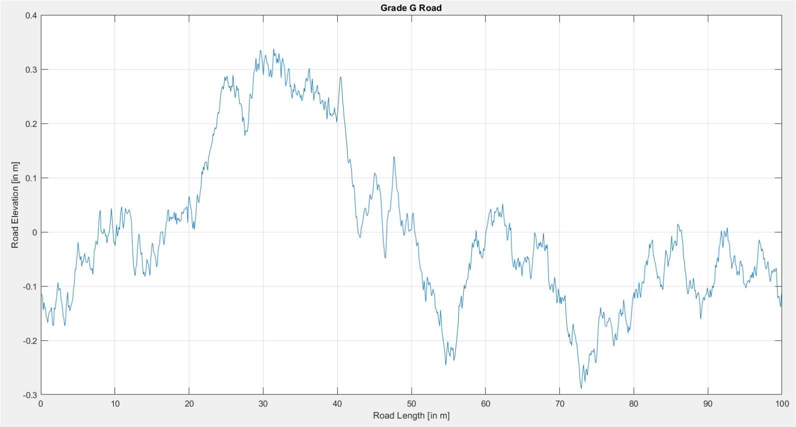

Figure 3.10: Road profile G

The results that were considered for observation are ”Sprung mass acceleration, Sprung mass position, Unsprung mass acceleration, Unsprung mass position, Suspension position, Suspension velocity, Sprung mass velocity and Unsprung mass velocity”

4

Results

4.1 Simulation results over time

4.1.1 Sprung mass position over time

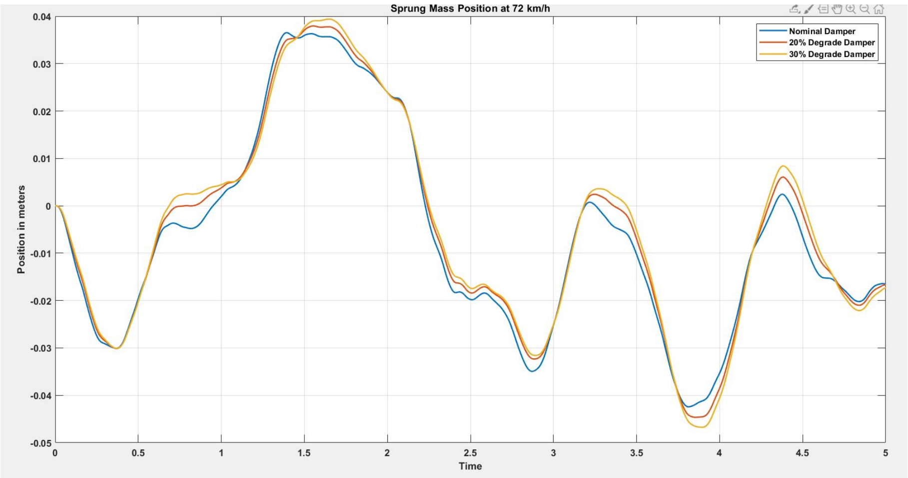

Figure 4.1: Sprung mass postion over time on a grade D road

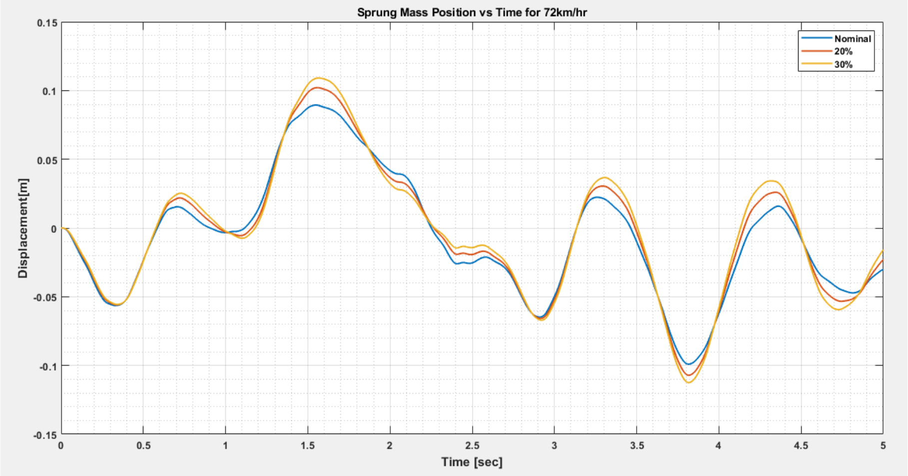

Figure 4.2: Sprung mass position over time on a grade E road

There is a difference in sprung mass displacement for the three dampers at the peak values in the order of few centimeters on both road profiles. However, the difference is not consistent and significant enough to be measured.

4.1.2 Sprung mass acceleration over time

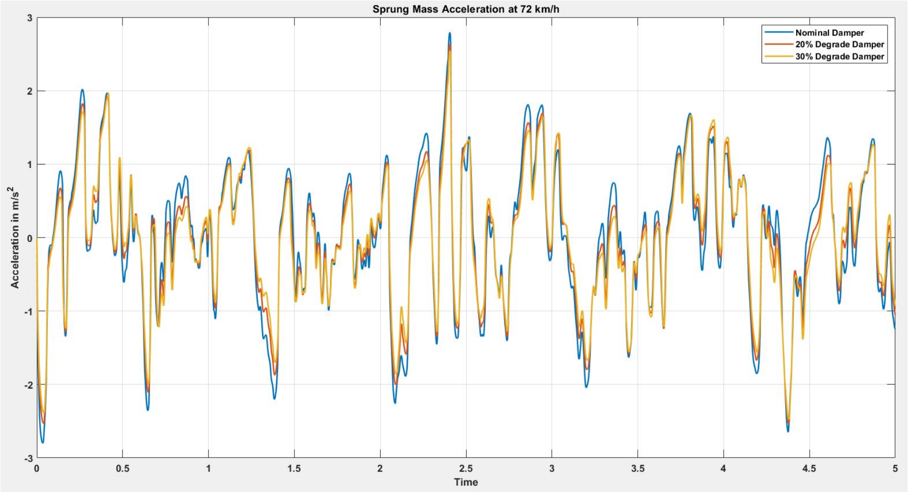

Figure 4.3: Sprung mass acceleration over time on a grade D road

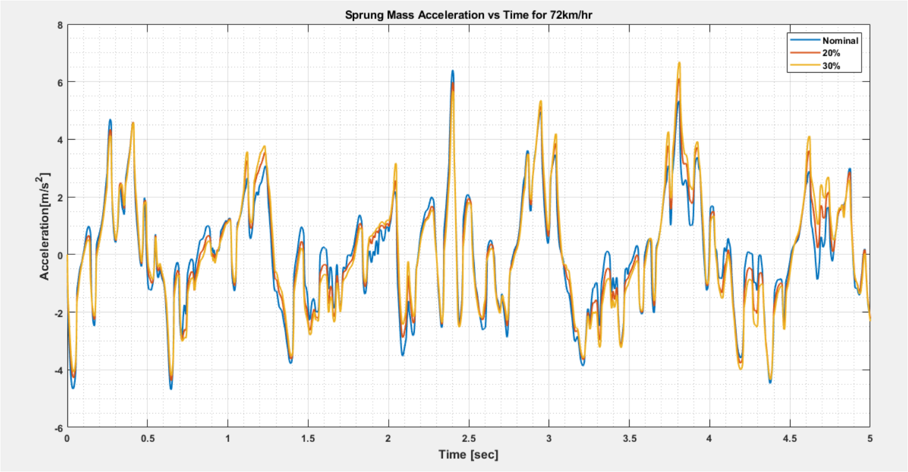

Figure 4.4: Sprung mass acceleration over time on grade E road

Sprung mass acceleration on the other hand show slightly different outcome. The sprung mass acceleration curves for these different dampers almost follow one another with deviations only at the peak values. Here the accelerations are high for a nominal damper while it is less for the degraded dampers which is something unexpected but, the difference in these accelerations is negligible. Studying the intensity of energy carried by these accelerations (Frequency response) might help us understand the reason behind these differences or the legitimacy of these results, and this will be done in the upcoming chapters.

4.1.3 Unsprung mass acceleration over time

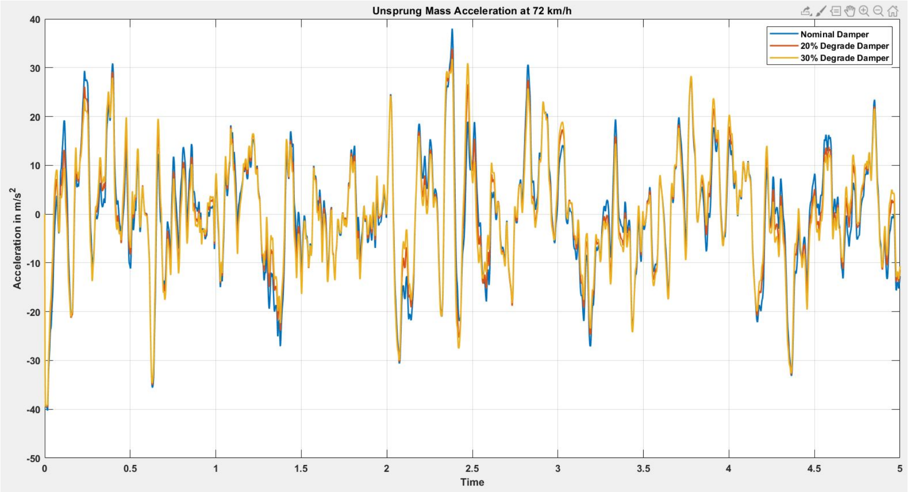

Figure 4.5: Unprung mass acceleration over time on grade D road

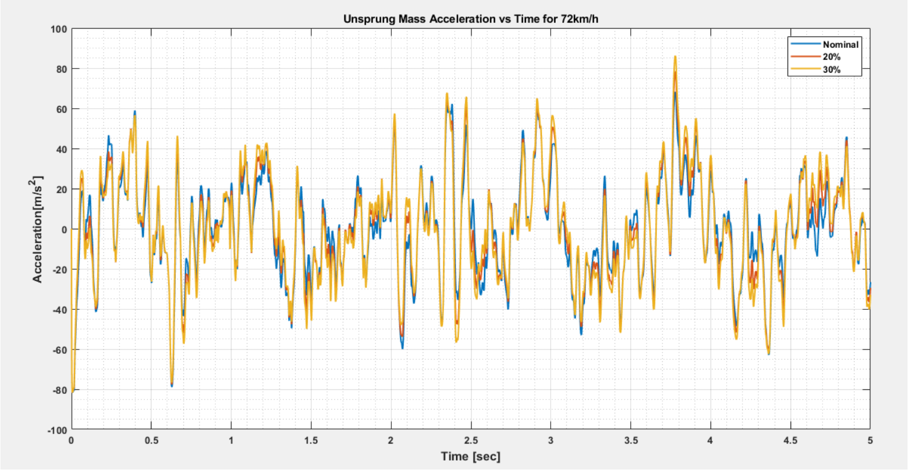

Figure 4.6: Unsprung mass acceleration over time on grade E road

The unsprung mass accelerations shows a similar behavior as the sprung mass accelerations with few peaks dominated by the nominal damper (in smaller road bumps where there are lower sprung mass accelerations, acc $< 20\ m/s^2$) while the rest dominated by the degraded dampers. There is no clear answer to this type of behavior but it can be speculated that this could be due to nominal damper transferring more force into the body while the degraded dampers loss of ability to generate more compression and rebound force as they are worn out. While the system performs slightly better with degraded dampers in small bumps it affects the ride greatly when there are larger bumps in the road (higher acceleration observed in sprung mass with 30% degraded damper in large road bumps).

4.2 Frequency response of the system

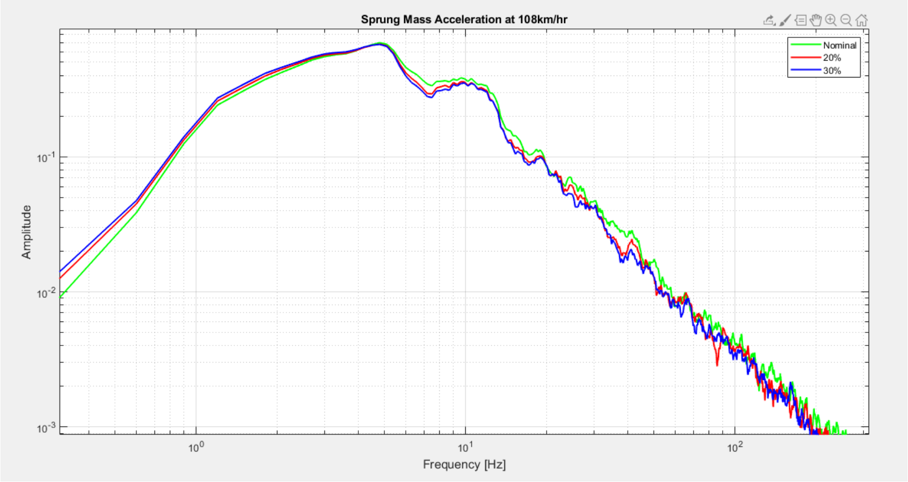

Figure 4.7: Sprung mass acceleration frequency response

To analyse frequency response of sprung mass acceleration, logarithmic scaled frequency is plotted as shown in fig.4.7 In sprung mass acceleration we can see a peak frequency at $5Hz$ for 108 km/hr and another peak at $10Hz$. This is because the system natural frequency is around $1Hz$ for sprung mass and 10Hz for unsprung mass. The amplitude of sprung acceleration is around 1(1/s). There is not much difference between nominal and worn damper but nominal damper amplitude is higher compared to worn damper which is justifies the sprung mass acceleration in time domain plots.

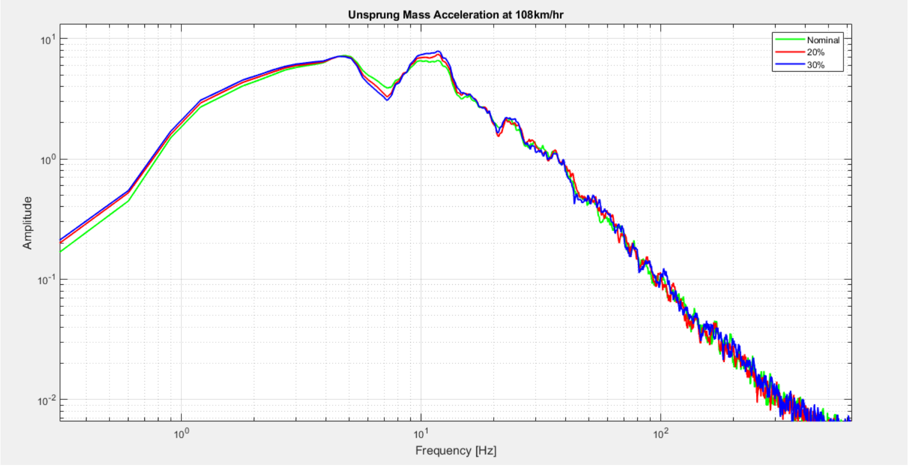

Figure 4.8: Unsprung mass acceleration frequency response

Frequency response of unsprung mass acceleration, logarithmic scaled frequency is plotted as shown in fig.4.8. In unsprung mass acceleration we can see a peak frequency at $5Hz$ and at $10Hz$. Natural frequency of unsprung mass is $10Hz$ hence we see higher peak at $10Hz$. The amplitude of unsprung acceleration is around 10(1/s).

4.3 Transfer function

To validate results from the simulation and to come up with a robust transfer function that would allow us to detect the change that is consistent throughout the run, following calculations were made and analysed in time domain and later on in frequency domain.

4.3.1 Relative displacement between sprung and unsprung mass (Zu -Zs)

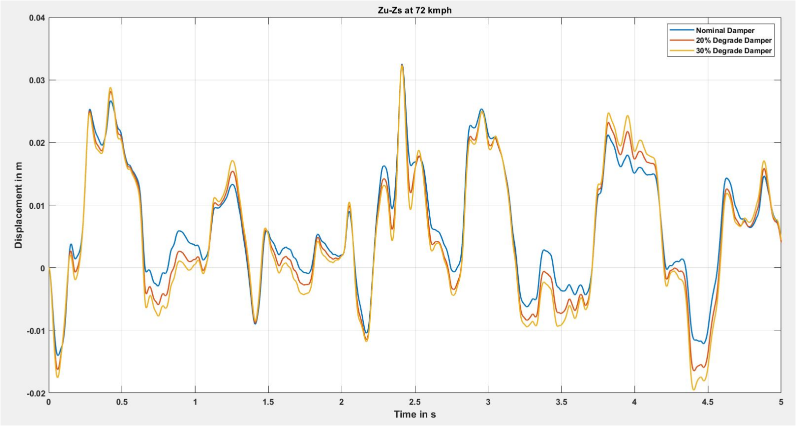

Figure 4.9: Relative displacement between sprung and unsprung mass on grade D road

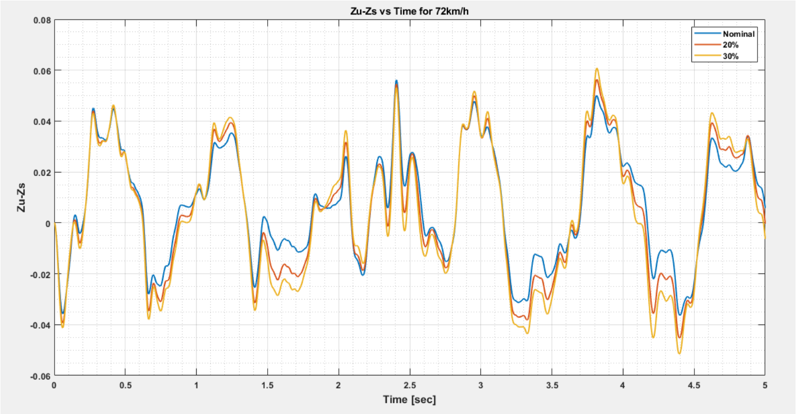

Figure 4.10: Relative displacement between sprung and unsprung mass on grade E road

The displacement between sprung and unsprung mass position shows a good display of the difference in the nominal and degraded damper but this transfer function fails due to its inconsistency as the difference can be seen only at the peak values or in instances where there are larger bumps. In positions where there are small or no bumps in the road both the nominal and degraded dampers display the same nature or response. Further more the change in response is clearly distinguishable in the negative region of the displacements at time 1s, 3.5s, 4s on road grade D and 1.5s, 3.5s, 4.5s on grade E road which hints that change in compression forces has significant difference on the system output than chanage in rebound forces.

4.3.2 Unsprung mass acceleration by sprung mass acceleration (Au/As)

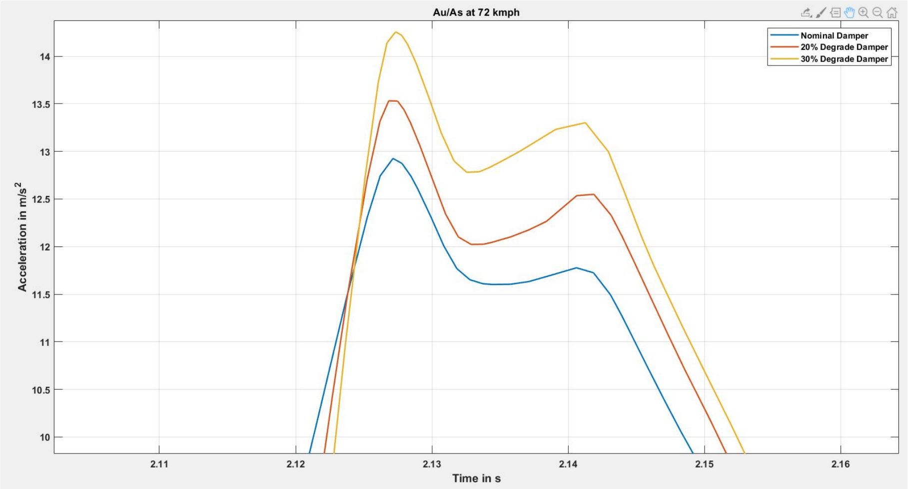

Figure 4.11: Au/As over time - grade D road

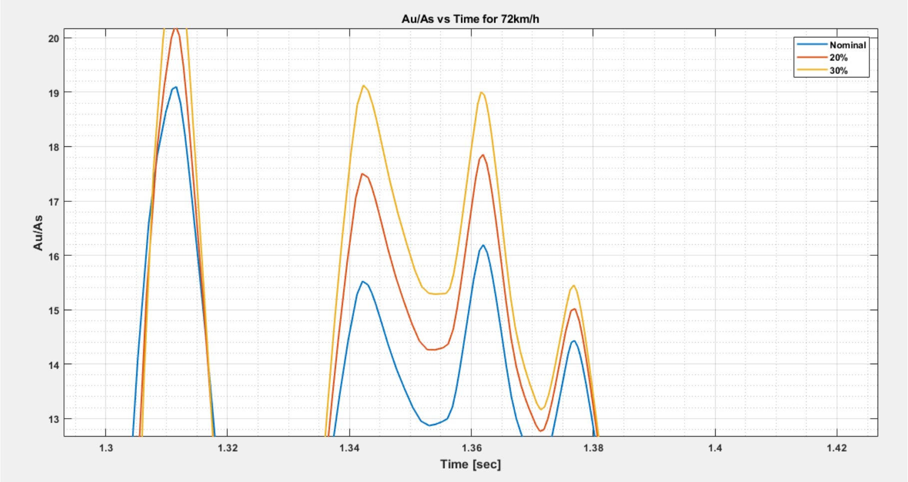

Figure 4.12: Au/As over time - grade E road

In this transfer function we were able to see big difference in the three dampers but it is only at very small time instances and not throughout the run, which makes it a very unreliable transfer function.

4.3.3 Unsprung mass position by sprung mass position (Zu/Zs)

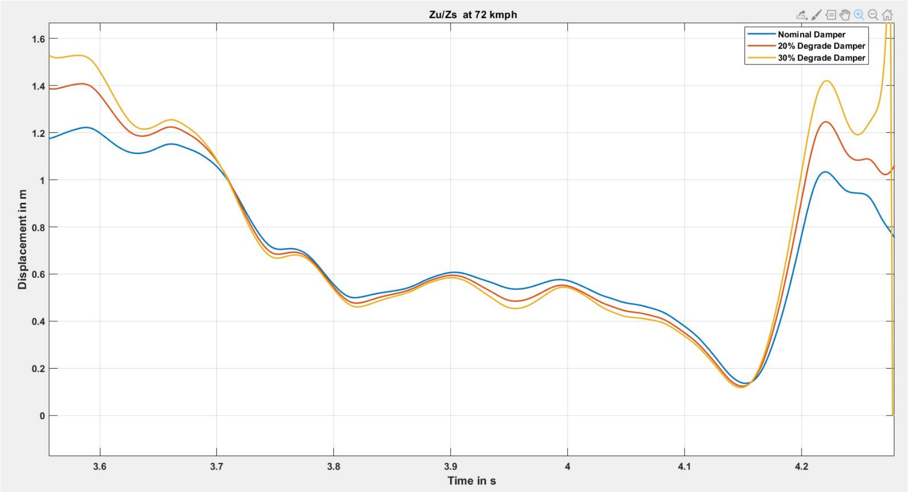

Figure 4.13: Zu/Zs over time - grade D road

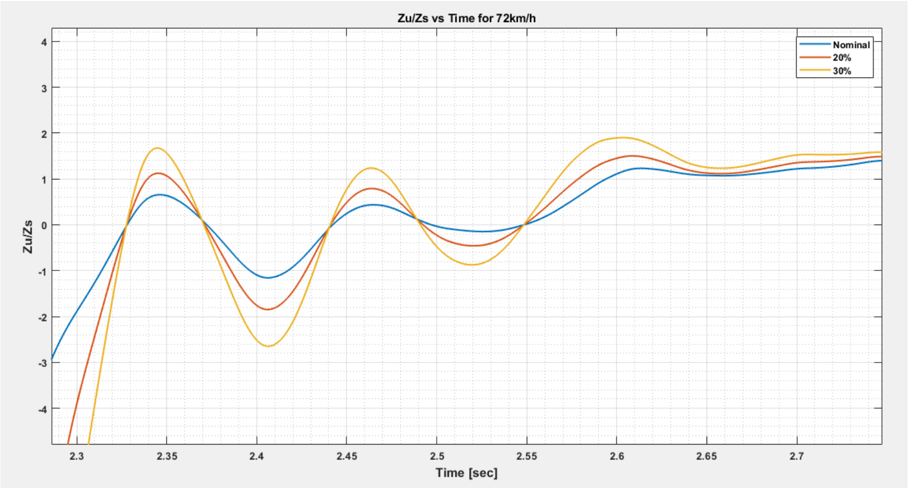

Figure 4.14: Zu/Zs over time - grade E road

In this transfer function, the division of unsprung and the sprung mass shows nice display of difference in the system output especially in grade E road (fig.4.14), which has higher amplitude of road disturbances. After 2.65s this response changes and the difference in the output of the three different damper fades away. In grade D a significant difference is seen only at the beginning and the end of the run whilst showing an unreliable response in the major portion of the run.

4.4 Frequency response of the relative speed between the sprung and the unsprung mass

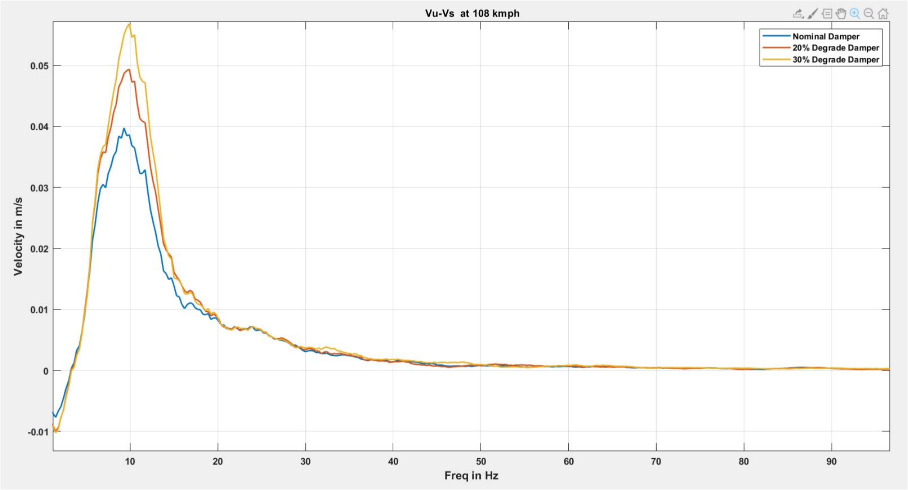

Figure 4.15: Relative speed between sprung and unsprung mass – grade D

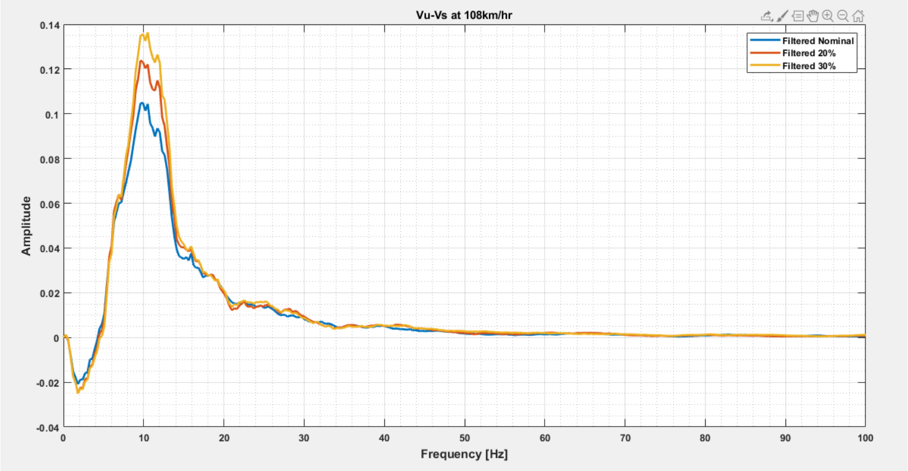

Figure 4.16: Relative velocity between sprung and the unsprung mass- E

The above figures shows the frequency response of the relative speed between the sprung and the unsprung mass for the two different road grades (D,E). At 10 hz frequency we have a peak and we can see a clear distinction between the three dampers, the difference is in the order of 0.01- 0.02 m or 1-2 cm.

5

Conclusion

The following conclusions can be made from the work carried out in this project.

- A complete study of suspension system with a good damper as well as a worn damper was done.
- A quarter car model was developed incorporating non linear dampers and bumpstop, in order to analyse the relation between wheel movement and body structure movement.
- The relation between wheel and the body movement was studied on three different road profiles of 100 meters recognized by ISO standards.
- Different transfer functions were analysed (Zu-Zs, Zs-Zu, Zu/Zs, Zs/Zu, Vu-Vs,Vs-Vu, Vu/Vs, Vs/Vu, Au-As, As-Au, Au/As, As/Au) in time and frequency domain, in order to detect the defect in the dampers.
- Although a change in the behavior of the system was noticed when damper was changed to degraded dampers, this change was found to be inconsistent and difficult to measure. As the result, a robust transfer function was not identified.

6

Future Work

- The model developed in this master thesis was a simple quarter car model, to get more realistic results the model can be built in more detail by including more suspension components. Perhaps, a half car model or a full car model can be a good start.

- The current model uses a spring and a damper system to represent a tire, an improved tire model which takes actual tire characterstics into consideration is one way to go.

- An actual physical testing can help verify the simulation results or can help us in approaching the problem with a different perspective.

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`html`.

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