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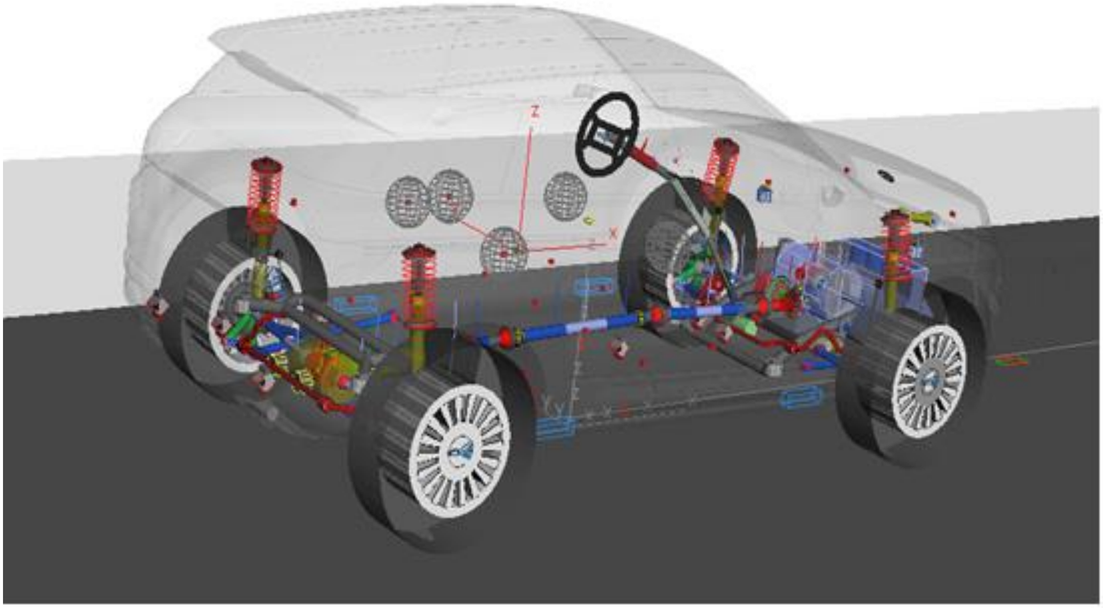

The use of wheel torque modulation to mitigate road induced longitudinal vibrations

Master's Thesis in Automotive Engineering

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Cover:
Land Rover Range Rover Evoque simulated in Simpack MBS software.

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ABSTRACT

Customers often require excellent ride comfort without compromising the handling characteristics. Extensive studies have been performed in the area of vertical motion in order to dissipate the forces from an impact whereas the study on longitudinal motion has not been so comprehensive.

The aim of this thesis is to present an analysis of how torque modulation can be used to mitigate the longitudinal vibrations felt by the passengers in the vehicle when it is subjected to impacts through speed lowering obstacles.

A study is performed to understand how road disturbances are inducing longitudinal vibrations and forces in the wheel assembly. The typical behaviour of the system is analysed in order to come up with a solution on the problem. The study has mainly been performed in a simulation environment using Simpack as MBS software and the vehicle model has been validated through proving ground tests.

In order to enable torque modulation when the vehicle is subjected to an impact, a hypothetical controller is created in Simulink and co-simulated with Simpack.

The results show improvement in lowering the high seat rail acceleration peaks of impact harshness although there is room for improvement regarding the controller to make it more stable and robust.

A very high torque is requested by the controller however further investigation shows that the torque can be limited to 250 Nm without significantly affecting the improvement of the vibrations caused by the impact. This gives a more realistic torque demand and flexibility in choosing actuators.

Key words: Suspension, Longitudinal vibrations, Torque modulation, Tyre, Bump, Simpack, Simulink, Multi Body System Analysis, CAE, Vehicle Dynamics

Contents

Preface

This master thesis work has been carried out at Jaguar Land Rover (JLR) in Whitley, Coventry, United Kingdom from September to December 2013. The project is performed at the Advanced Chassis R&D department as a final work completing a Master's degree in Automotive Engineering at Chalmers University of Technology.

We would like to express our gratitude to our supervisors at JLR – Riccardo Ficca and Phil Barber for supporting us through all stages of our work and making sure that we received necessary connections and resources within the organisation. Furthermore we would like to thank Saleem Zuberi for his support on proving ground testing, the practical stuff of getting started at the department and providing valuable input to the project.

We would also like to thank Libin Wang, Neel Shah and Lee Adcock for their help with our controllers and everyone not mentioned for a wonderful time at the Advanced Chassis R&D department.

Finally, we owe special thanks to our examiner, Gunnar Olsson and Professor Bengt Jacobson for guiding us through the thesis, providing feedback and much appreciated encouragement.

Göteborg, October 2014

Ibrahim Bakirci & Nino Katadzic

Notations

$e(t)$	Error over time
F_x	Longitudinal contact patch force
K_d	Derivative gain
K_i	Integral gain
K_p	Proportional gain
x	Body longitudinal position
$\dot{x}$	Body longitudinal velocity
$\dot{x}_{hub}$	Hub longitudinal velocity
$\ddot{x}_{hub}$	Hub longitudinal acceleration
$\ddot{x}_{seat}$	Seat rail longitudinal acceleration
$\ddot{z}_{seat}$	Seat rail vertical acceleration
T_{dr}	Wheel drive torque
T_{brk}	Wheel brake torque
ω	Wheel angular velocity

1. Introduction

The aim of this document is to give a detailed analysis of torque modulation to counteract longitudinal vibrations felt by the passengers in the vehicle when it is subjected to impacts in terms of road bumps.

1.1 Background

In recent years, the importance of ride comfort has steadily increased in vehicle design. The customers often require excellent ride comfort without compromising the handling characteristics.

Extensive studies have been performed in the area of vertical motion but the longitudinal aspect of vibrations has been disregarded even though it has a great impact on the human body. During the event of a vehicle driving over a bump, vibrations are transmitted thru the suspension in longitudinal and vertical directions. The vertical vibrations are handled with the spring and damper and to some extent the tyre whereas the longitudinal vibrations are only partially absorbed by the bushings and tyres. Conventional bushings are designed to handle many different situations i.e. braking, accelerating and cornering, hence they are not optimized for impact harshness only.

Impact harshness is an important metric for vehicle ride performance which is experienced by driver and passengers during an impact to the tyre from a road disturbance.

1.2 Objective

The aim of this master thesis is to simulate a vehicle driving over a bump and create a hypothetical controller using wheel torque to mitigate the longitudinal vibrations felt by the passengers.

1.3 Project scope

The scope of the project includes following activities:

- Understanding the problem - Investigate how various road disturbances are inducing longitudinal vibrations and forces in the wheel assembly through simulations in Simpack.
- Perform high speed camera footage to understand the problem
- Field testing – collect data on discrete events to correlate with simulations.
- Investigate the possibility of using torque modulation to mitigate the longitudinal vibrations and perform a study on which road disturbances the torque modulation can be used.
- Develop a control algorithm in Simulink that controls a hypothetical torque actuator and can be used to determine the feasibility of mitigating longitudinal vibrations felt by the vehicle occupants.

1.4 Delimitations

- The simulations will be performed in a multi body dynamics program called Simpack provided by Jaguar Land Rover.
- The controller will be assumed to have an ideal actuator that delivers the torque without any limitation on response time, maximum output and other typical characteristics of actuators.
- The model used for simulations is built by Jaguar Land Rover with some additions regarding torque delivery managed by the thesis writers.

1.5 Understanding the problem

In order to understand the problem when driving over a bump and what the passengers experience, certain parameters need to be determined that will provide information to characterise the problem. The parameters are used to determine the location of sensors to be used in simulations and proving ground tests.

Because the forces from the bump are transmitted through the wheel and suspension linkage into the body, one parameter that needs to be monitored is wheel hub acceleration and specifically in longitudinal direction. The measurement method used to monitor ride comfort is the seat rail acceleration which is also part of the ISO 2631 vibration standard. This standard claims as shown in figure 1, that for a longer period of time the human body is most sensitive for vibrations between 0-10 Hz. W_d, W_k and W_c are defined as frequency weightings with specific directions and primary context of use. (Rimell & Mansfield, 2007)

- W_d - fore-aft and lateral seat vibration
- W_k – vertical seat vibration
- W_c – fore-aft backrest vibration

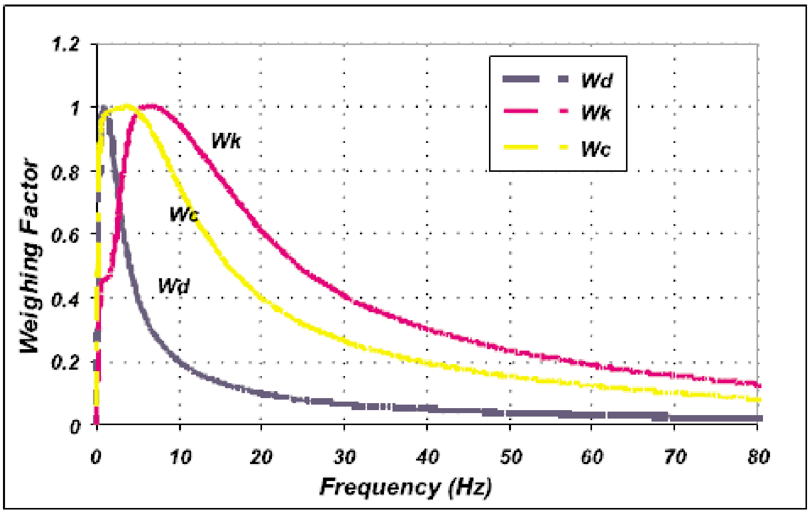


Figure 1. Frequency range and weighing factors - ISO 2631

To investigate the possibility of mitigating longitudinal vibrations a series of run cases are defined. The focus of interest lies in the type of disturbance that causes a clear impact during a short period of time on the wheels and suspension without damaging the vehicle.

The metal strip and the yellow-black rubber speed bump (RS bump) satisfy the requirements and are available at the Jaguar Land Rover (JLR) proving ground. The geometry (table 1) of the two obstacles will be used in simulations and tests.

The metal strip has a very short wave length and serves as a measure of how fast the control system has to be to control the vibrations during a short period of time. The RS bump has greater amplitude and longer wave length which causes a greater impact on the suspension and therefore serves as a measure of how much torque is needed to counter the vibrations.

Bump	**Size [mm]**
RS bump (figure 3)	H: 68 W: 400 (flat top of 180 mm)
Metal strip (figure 4)	H: 22 W: 100

Table 1. Run cases with different bumps

Figure 2. RS bump

Figure 3. Metal strip

1.5.1 Impact harshness and after-shake

When the vehicle drives over a bump, it is subjected to vibrations that can be categorised regarding their magnitude and duration. Figure 4 illustrates simulation data on seat rail longitudinal acceleration for the RS bump where it shows that the impact harshness is an event with a high magnitude, lasts for a short period of 70 ms and is the immediate reaction of the bump on the vehicle. The part after the impact harshness, where the oscillations starts at 5.2 seconds as shown in figure 4, lasts for a longer period and has lower magnitude is called after-shake. The frequency content of the impact is visible in figure 5 where the highest magnitude of frequencies are in the range of 5-25 Hz. The impact harshness part highlighted in figure 4 has a frequency of 15 Hz.

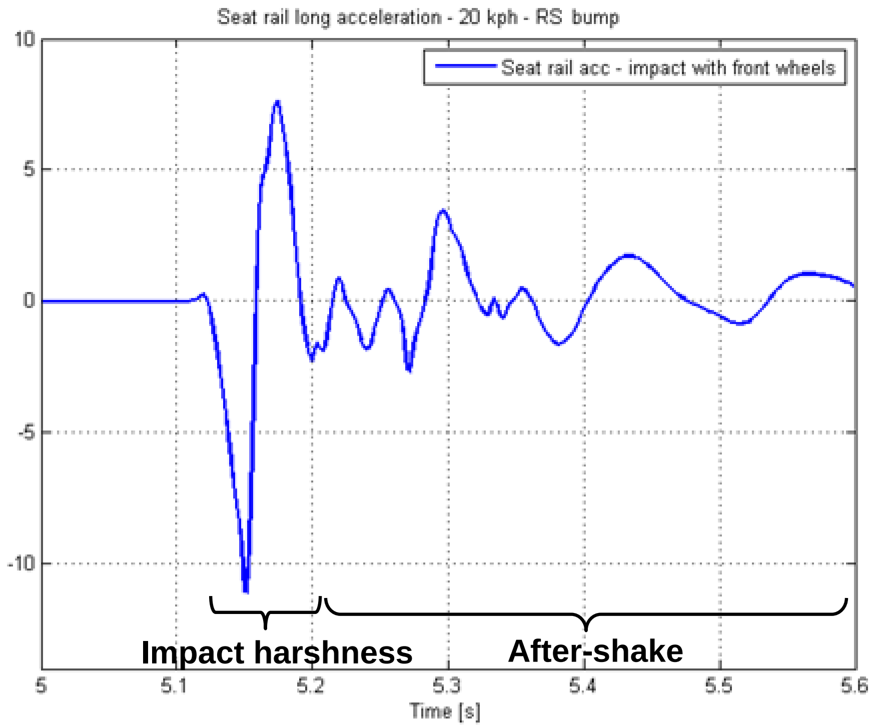


Figure 4. Definition of impact harshness and after-shake on longitudinal seat rail acceleration data

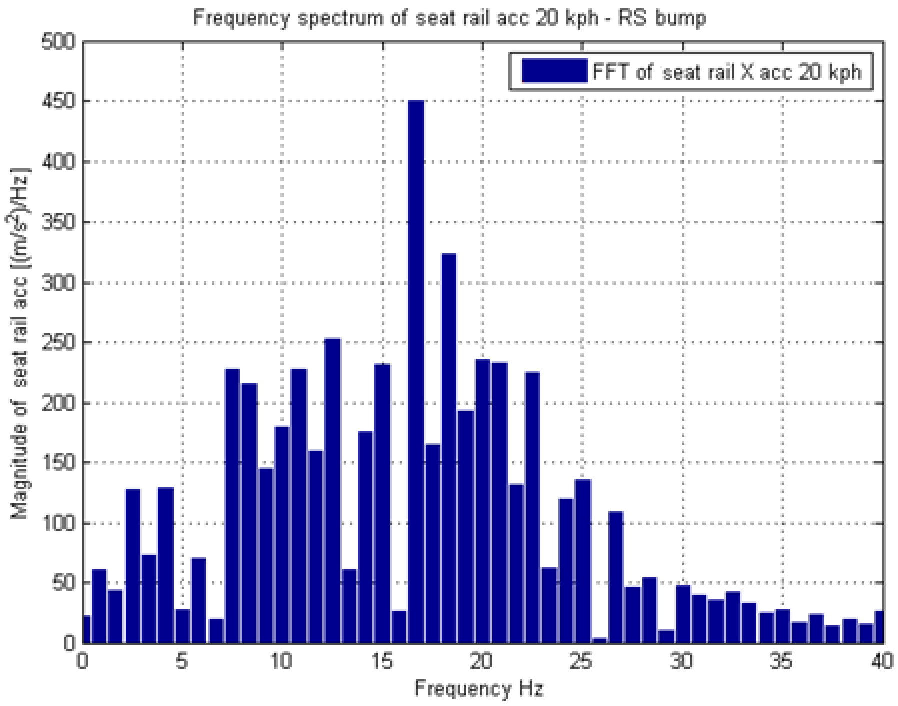


Figure 5. Magnitude of seat rail acceleration vs. frequency of the RS bump

1.5.2 High-speed camera test

Initial tests with high speed camera are performed to understand the motion of the wheel in vertical and longitudinal direction in relation to the body. In order to track the vehicle over the bump, reference stickers are attached to the wheel arch and the wheel hub.

The camera is positioned close enough to capture the wheel before and after the impact together with the RS bump.

The camera is manually triggered through the software Motion Studio X64 provided by the camera maker IDT with the settings listed in table 2.

Model	IDT Nx5
Sampling frequency	500 Hz
Exposure time	390 µs

Table 2. High-speed camera settings

The vehicle is driven with constant speed of 20 km/h over the obstacle and the footage is gathered for post processing. Tracking vectors are added on the reference stickers in Motion Studio X64 and the wheel motion is studied. Figure 6.1-6.3 shows that the wheel motion in vertical direction is significant whereas in longitudinal direction, the wheel has only a small displacement relative to the body. The displacement of the wheel in vertical direction is 58 mm and 12 mm in longitudinal direction.

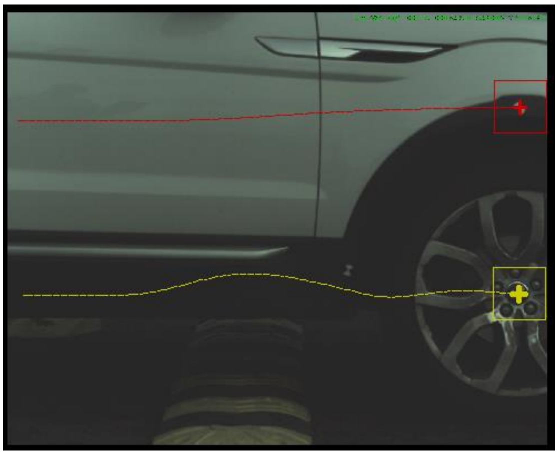
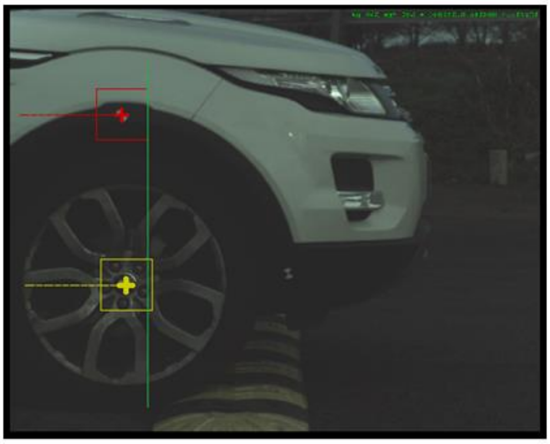


Figure 6.1 Motion tracking in Motion Studio X64



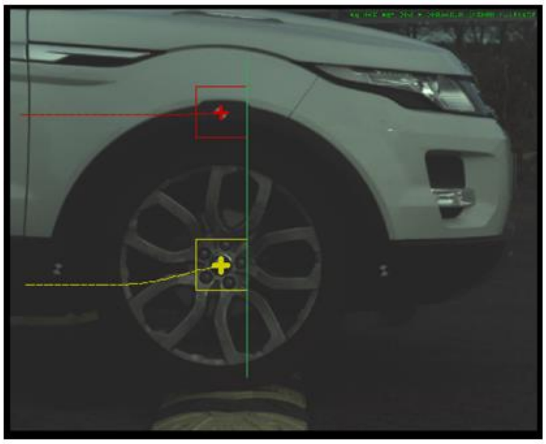


Figure 6.2 Relative position in x - before impact **Figure 6.3 Relative position in x - at impact**

2. Solving the problem

This project aims to investigate the possibility of mitigating the vibrations with torque modulation. The concept behind this idea is to apply a torque on front and rear wheel axle separately while the vehicle is driving over a bump.

Due to the impact of a bump, the wheel assembly is oscillating back and forth resulting in vibrations in the chassis and the seat rail. A longitudinal deceleration is acting on the hub when the wheel hits the bump and lasts until the wheel has reached the top of the bump. At this time the acceleration will turn sign into positive acceleration which is explained as the wheel has to catch up with the chassis.

When the wheel hits the bump, a longitudinal force is generated at the contact patch which results in a torque around the wheel centre with the loaded radius as a lever. The longitudinal force and torque is creating vibrations in the hub and the seat rail, which in theory means that if an opposite torque is applied on the wheel at the same time, the forces will cancel out and the resulting acceleration on the hub and the seat rail will be mitigated.

Figure 7 is representing simulation data of the front hub longitudinal acceleration from an impact and the hypothetical mirrored front hub acceleration. By applying torque on the wheel that results in the mirrored front hub acceleration seen as the dashed graph in figure 7, the resulting acceleration on the vehicle due to the hub acceleration would be zero. However the reaction torque from the powertrain has not been considered in this case.

The demand of additional torque can be done through different sources e.g. electrical powertrain, conventional powertrain and brakes.

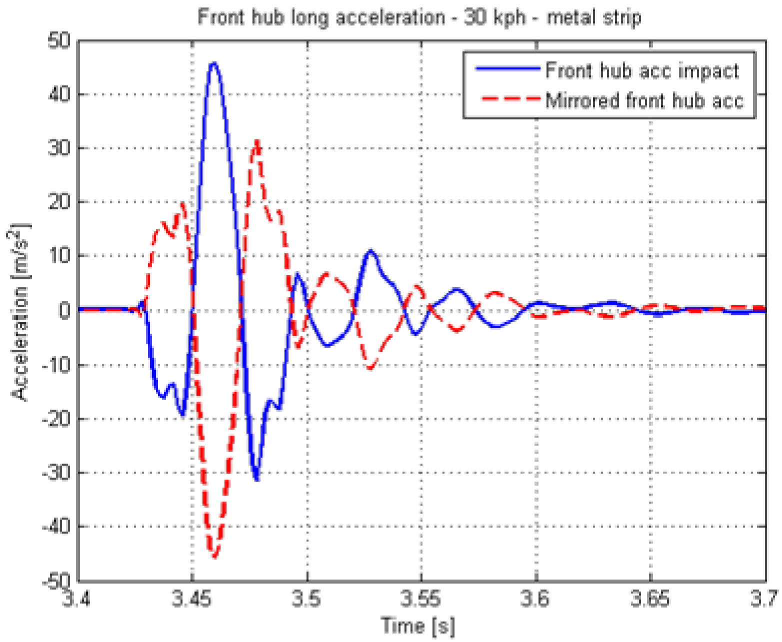


Figure 7. Hub acceleration from impact and mirrored hub acceleration

3. Multi-body simulation

To achieve high repetitive accuracy in simulations, a full vehicle multi-body simulation model (MBS) is used with flexible linkages, -tyres, -bushings together with all components in the chassis to investigate vehicle behaviour in a software called Simpack (ref: Simpack AG). A well-tested and developed vehicle simulation model for the Range Rover Evoque was available at Jaguar Land Rover and used as a reference in this project.

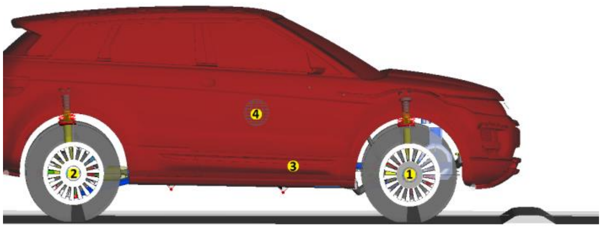


Figure 8. MBS vehicle

Vehicle specs	Unit	Value
Curb weight	kg	1800 kg
Wheelbase	m	2.66
Front suspension	-	MacPherson
Rear suspension	-	Tri link

Table 3. Vehicle specifications

A common speed in areas where road bumps are located is 30 km/h which will be the upper limit in the simulations. This is also decided to not promote illegal speeding over the obstacles. A second speed of 20 km/h is also used because that is a more reasonable speed over the specified obstacles. Sampling rate of 500 Hz is used in the simulations.

Simulation cases	Speed [km/h]
Metal strip	20 & 30
RS bump	20 & 30

Table 4. Run cases

3.1 Tyre model

In order to capture accurate results over the bump, a tyre model called FTire - Flexible Ring Tyre Model (ref: cosin scientific software) is used which is developed particularly for ride comfort simulations. The main characteristics of the FTire model are:

- Fully non-linear
- Valid in frequency domain up to 120 Hz
- High accuracy when passing single obstacles like potholes and bumps
- FTire is designed for vehicle comfort simulations and prediction of road loads on road irregularities even with extremely short wave-lengths.
- It can also be used as a structural dynamics based, highly nonlinear and dynamic tyre model for handling studies without limitations or modification to input parameters.

The FTire model consists of finite amount of flexible rings where each ring is carrying bending, radial, tangential and lateral stiffness which is visible in figure 9. FTire also animates the tyre during simulation which is visible in figure 10.1 with the distorted tyre over the bump.

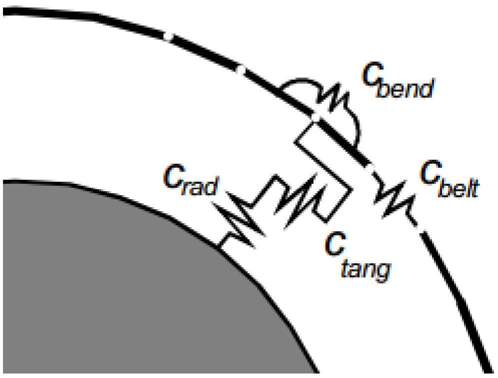


Figure 9. Belt representation in Ftire

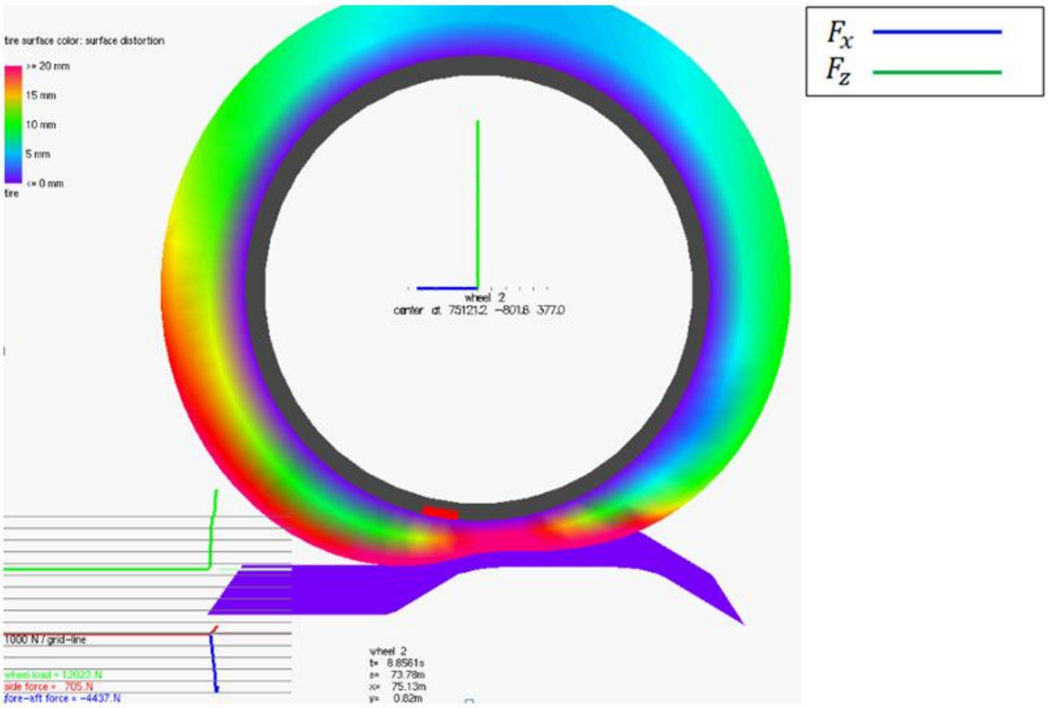


Figure 10.1 Ftire animation with distorted belt over the bump

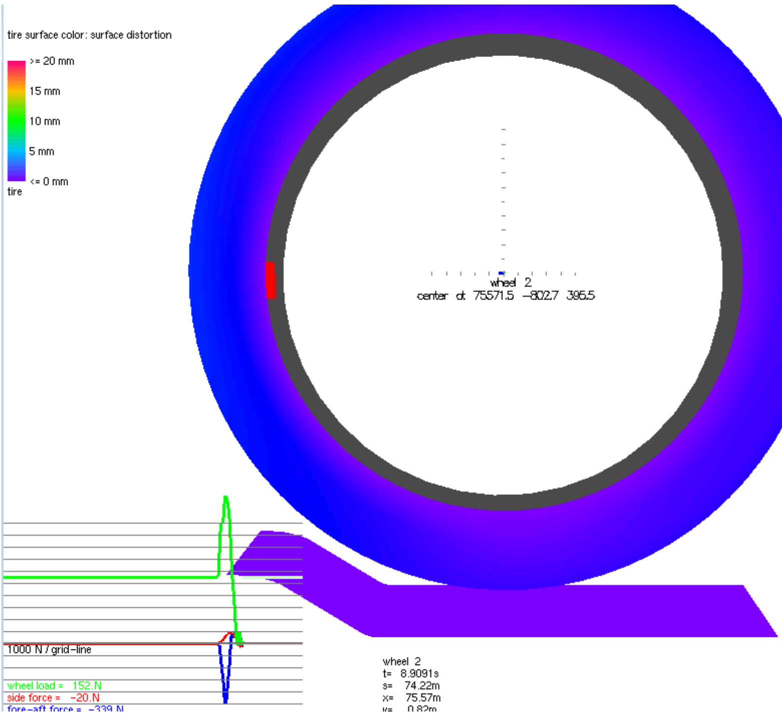


Figure 10.2 With 20 kph over the RS bump the wheel is in the air a short period after the bump

3.2 Co-simulation between Simpack and Simulink

In order to control the torque applied to the MBS vehicle in Simpack, a hypothetical controller is set up in Simulink.

The hypothetical controller will be assumed to have an ideal actuator that delivers the torque without any limitation on response time, maximum output and other typical characteristics of actuators. A simple representation of the Simulink layout is showed in figure 11 as a feedback loop where the plant is represented as a transfer function block that communicates externally with the MBS model and the bump in Simpack. This means that the controller in Simulink and the MBS model will work in co-simulation with each other.

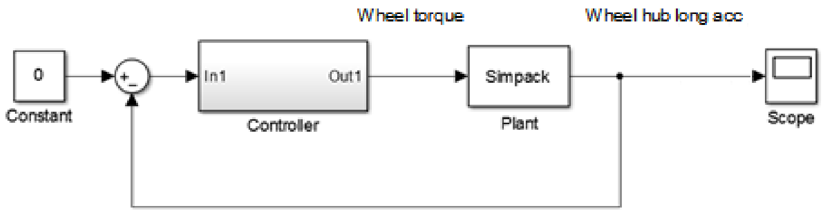


Figure 11. Simulink interface with Simpack co-simulation block

To control the vehicle over the bump, certain parameters would need to be sensed (table 5) and sent as measurements from Simpack into the hypothetical controller in Simulink. The controller calculates the torque needed to counteract the vibrations in the hub and applies individual wheel torques to the vehicle. Each wheel has torque inputs represented by application points in Simpack. The application points in Simpack are built up as Force Elements between two reference objects called markers. The application points for the drive torque is set between the outer and inner connection of the drive shaft which means that the drive torque should be transmitted through the drive shaft. The brake torque is applied between the knuckle and the wheel as can be seen in figure 12.

Output from Simpack to Simulink	Symbol	Unit
Body longitudinal position	x	m
Body longitudinal velocity	$\dot{x}$	m/s
Hub longitudinal velocity	$\dot{x}_{hub}$	m/s
Hub longitudinal acceleration	$\ddot{x}_{hub}$	m/s^2
Seat rail longitudinal acceleration	$\ddot{x}_{seat}$	m/s^2
Seat rail vertical acceleration	$\ddot{z}_{seat}$	m/s^2
Wheel angular velocity	ω	rad/s
Longitudinal contact patch force	F_x	N
Input to Simpack from Simulink		
4 x wheel torque (drive)	T_{dr}	Nm
4 x wheel torque (brake)	T_{brk}	Nm

Table 5. Inputs and outputs of the co-simulation

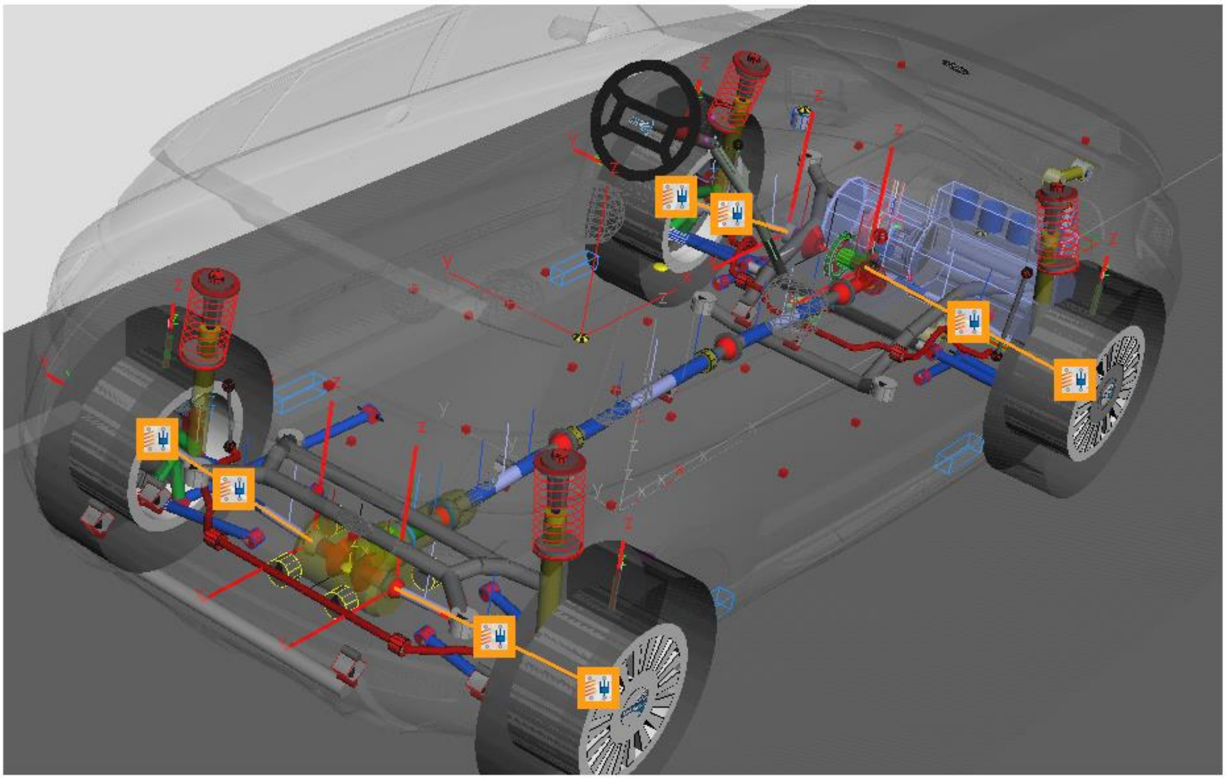


Figure 12. Application points of drive and brake wheel torque in Simpack

3.3 Verification of vehicle model

In order to study the vibrations in simulations and get accurate and robust results it is important that the model corresponds to the actual vehicle. The vehicle used in the MBS analysis is compared to the real vehicle with proving ground tests. The main objective of the comparison is to give an indication of how well the vibrations in Simpack correlate to real life vibrations. By doing this the torque modulation can be accurately tuned in a simulation environment and thereby save time, effort and financial resources.

The vehicle is driven over the metal strip with a constant speed of 30 km/h and data of the seat rail acceleration is gathered. Analysis of the data shows that the characteristics of the experiment and the simulation are similar to each other in impact harshness and after-shake although the rear wheels hit the obstacle earlier in the proving ground data. A validation of the velocities in both cases shows that the vehicle at the test was having a speed of 32 km/h which is the explanation of the second part of the acceleration plot coming earlier as shown in figure 13.

It is important to mention that it is hard to have an exact match between real life and simulation data due to large number of mass-spring systems with resonances that interfere with each other and the difficulty of recreating the simulation case.

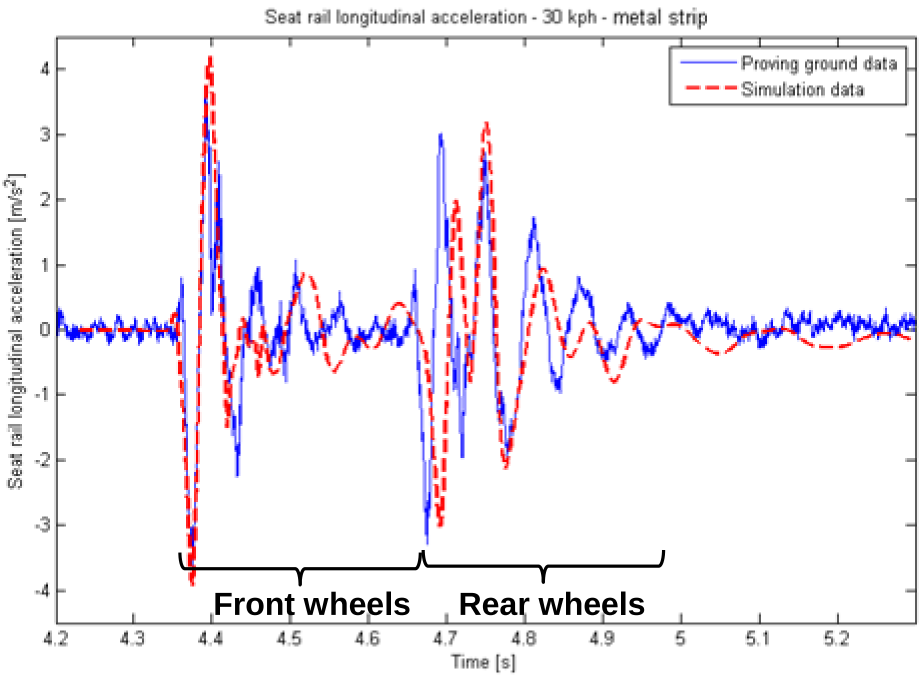


Figure 13. Proving ground data vs. simulation data at 30 km/h on metal strip

4. Controllers

In order to apply a negative and positive torque on the wheels with changing magnitudes and duration, different torque controllers are created.

4.1 Self-learning system – inverse TF

The idea with the self-learning system is that when a vehicle experiences a disturbance for the first time, the resulting hub accelerations are stored in a look-up table. Next time the vehicle runs over the same or a similar obstacle the system uses the accelerations to back calculate the required torque that should be used in order to mitigate the vibrations felt by the passengers. The system needs to recognise the obstacle and calculate the torque before the vehicle runs over the bump. The sensing of obstacles can be achieved by the use of different sensors like cameras and lasers. Pre-emptive sensing through stereo cameras allows 3D scanning of the road surface in real time up to a certain distance ahead of the vehicle.

The disadvantage of this system is the sensitivity of applying the torque at the wrong time.

To calculate the required counter-acting torque, an inverse transfer function of the system is needed. The first step is to calculate a transfer function. By knowing the response a torque input gives on the hub acceleration, it is possible to calculate the transfer function G(t) of the system.

$$Y(t) = G(t) \cdot U(t) \quad \rightarrow \quad G(t) = Y(t)/U(t)$$

In this case a torque input in the form of white noise is used to excite a random output. This is done in order to ensure that the transfer function reacts to a wide frequency spectrum. Before an inverse transfer function can be calculated the linearity of the original transfer function has to be validated.

To validate the transfer functions linearity it needs to be additive and homogenous. (Åström & Murray, 2012)

Additive: $Y(U_1 + U_2) = Y(U_1) + Y(U_2)$

Homogenous: $Y(aU_1) = aY(U_1)$

In this case the transfer function is proven to be linear between 0-600 Nm.

The hub acceleration response of the calculated transfer function is plotted against the actual response of the simulation model (figure 14). As can be seen, the transfer function gives a good representation of the model. However, the problem arises when the inverse of the transfer function is used to calculate the torque from a given hub acceleration. It is evident that the transfer function does not accurately calculate the counteracting torque in figure 15. Despite the linearity of the transfer function, the inverse does not work. At this moment it is not certain what the cause is to the problem. Looking at figure 16, it is evident that the transfer function seems relevant up to 50 Hz.

One idea in order to have a working transfer function is to use a white noise source of 0-50 Hz only.

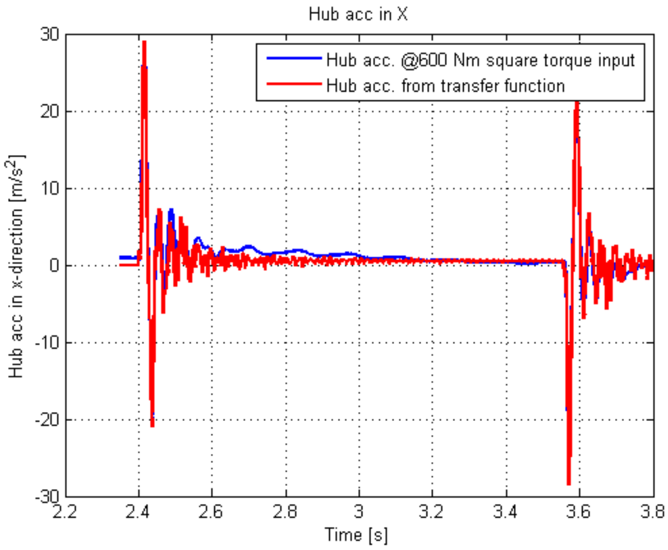


Figure 14. Calculated hub acceleration through transfer function vs. actual hub acceleration

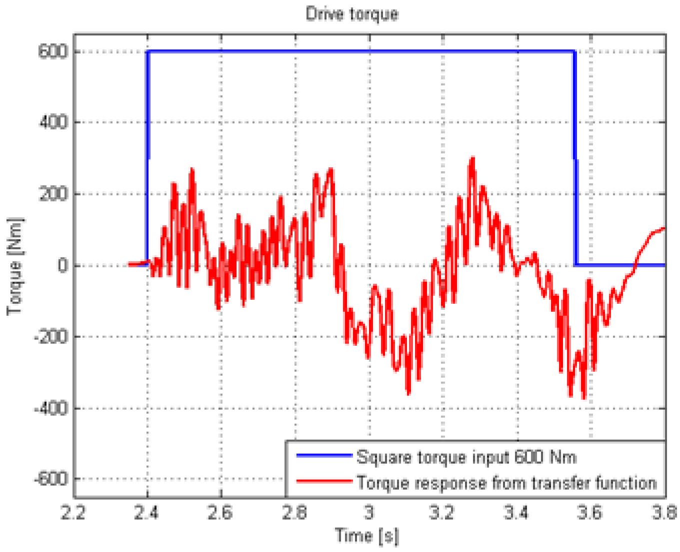


Figure 15. Mismatch between actual torque input and the torque calculated through the transfer function

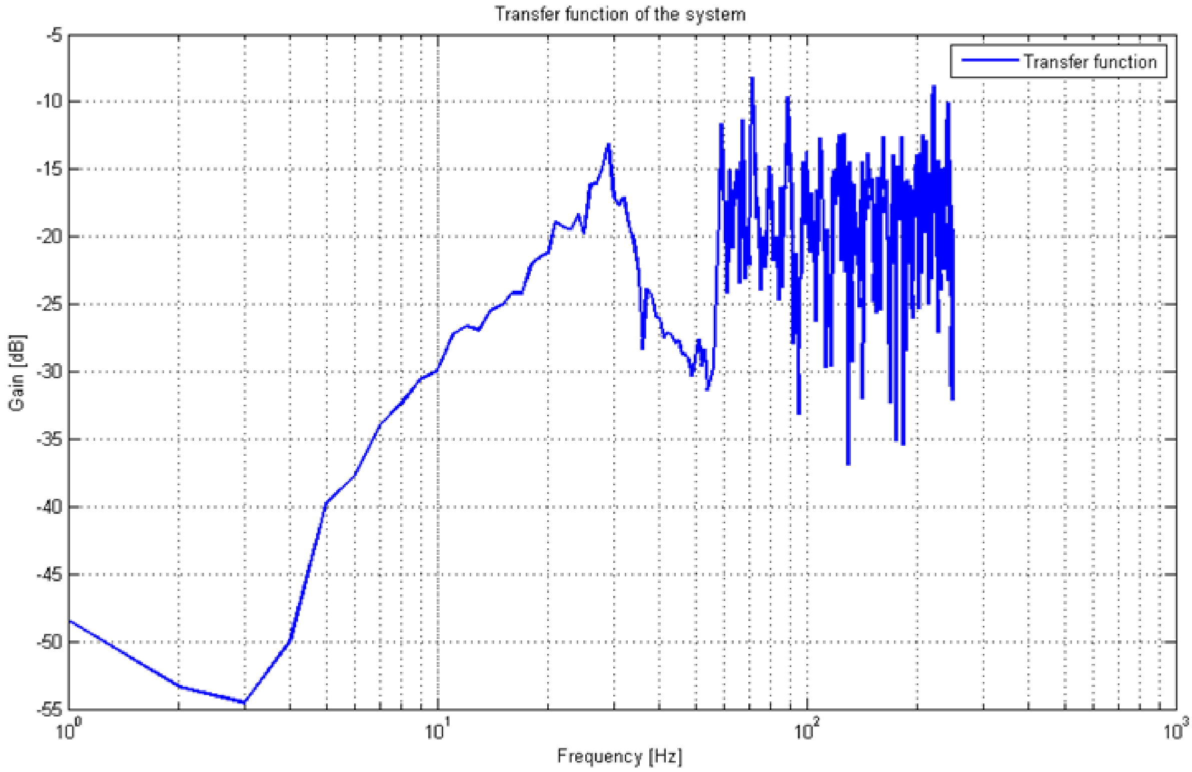


Figure 16. Relevancy of transfer function up to 50 Hz but with uncertainties at higher frequency.

4.2 Active system – feedback controller

In order for the controller to act independently without relying on other systems giving it required information about the bumps, an active system is created. This system is a real-time feedback controller that has a required value of zero on the hub acceleration, where it calculates an error and feeds it into a PI controller which calculates the torque that should be applied to the wheels. The derivative term of a PID controller is excluded due to the sensitivity to measurement noise in real life, which with the high rate of change on hub acceleration would cause excessive wear on the actuators.

The controller is enabling the front and rear wheels separately over the impact to not interfere each other.

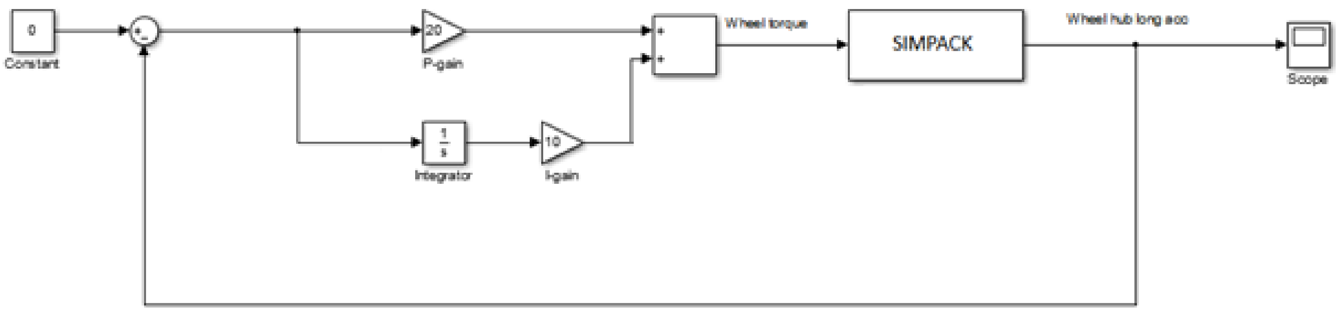


Figure 17. Simulink interface including the PI controller and the Simpack co-simulation block.

With the Ziegler-Nichols method (Thomas, 2008) initial gains are derived and used on the PI controller in simulation. The gains are manually tuned in order to get better results. In figure 18, the results show hub and seat rail longitudinal accelerations for the non-mitigated case compared to a proportional gain added in a feedback control. The impact harshness part of the curve is clearly mitigated with an increasing gain value. This is not the case for the after-shake where an increased gain generates larger oscillations. With a proportional gain of 30, the controller starts to become unstable which is the limit of what gain the controller should have. A proportional gain of 20 is chosen.

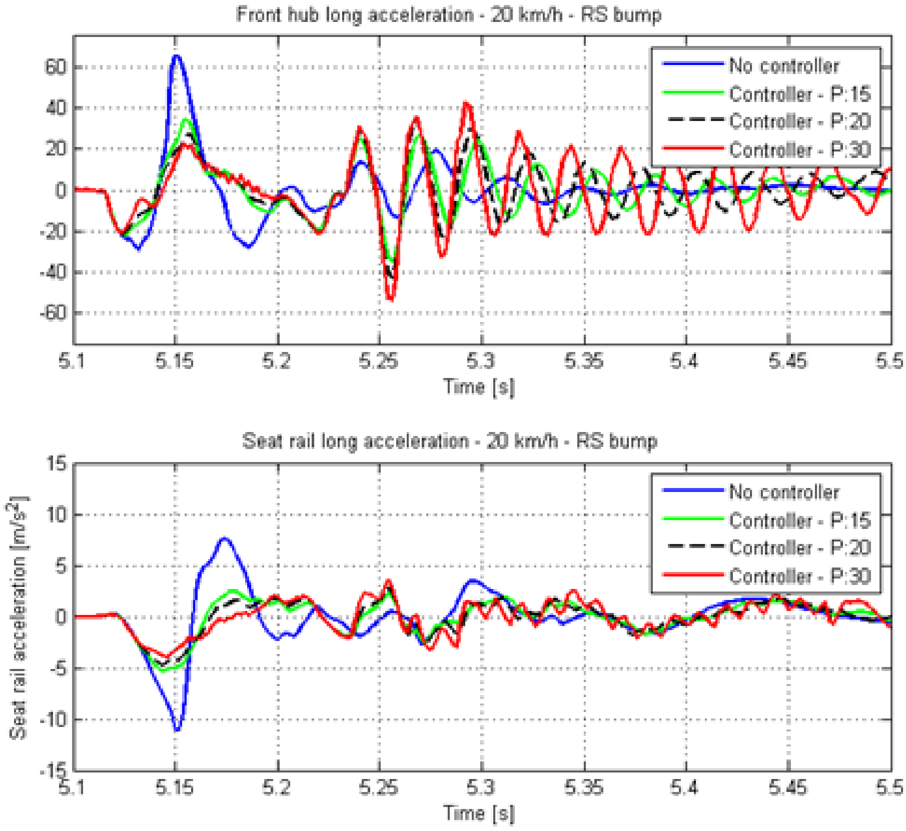


Figure 18. Proportional gain sweep and the results on hub and seat rail acceleration.

By including the integral gain in the controller, the system starts to become very unstable in the after-shake region. Figure 19 shows the time history of torque application over the bump where the torque is increasing with time and becomes constant. The reason for this is the function of PI controllers. Equation 1 shows that the integral part of the controller is adding the error over time to the output signal which gives a torque input steadily increasing instead of going back to zero between each pulse.

$$u(t) = K_p e(t) + K_i \int_0^t e(t) d\tau$$

Equation 1. PI controller

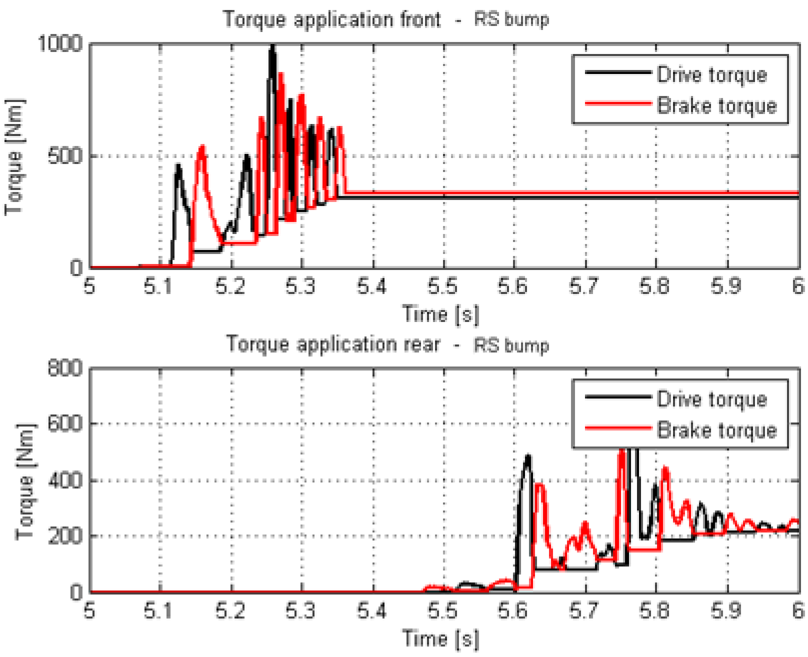


Figure 19. Increasing torque over time on front and rear wheel

By taking the output value from the integral part of the controller and feeding it back and subtracting it from the error in the next time instant (figure 20) the constant increase in torque was removed (figure 21).

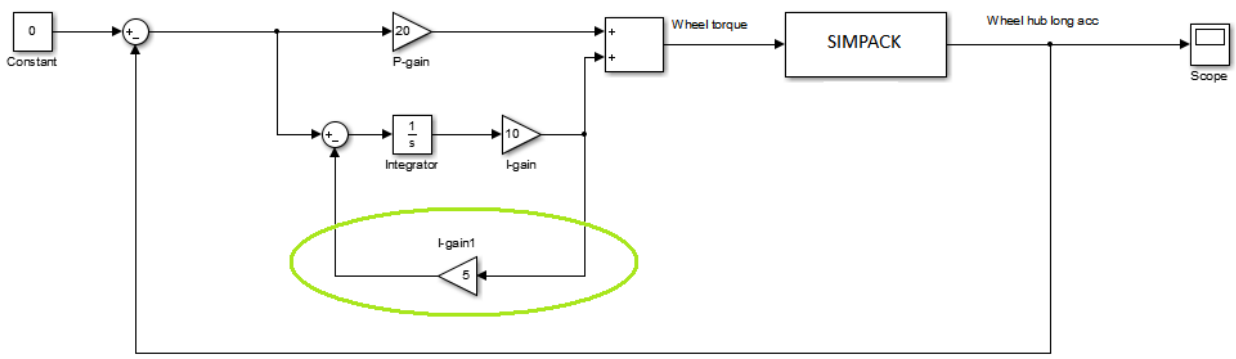


Figure 20. Integral feedback added to the controller

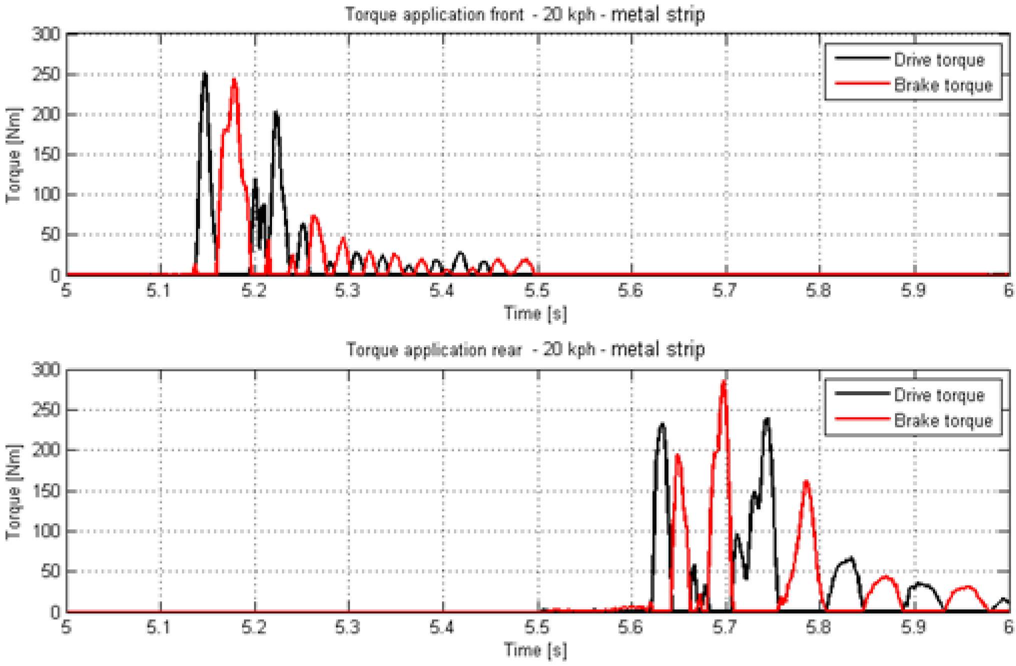


Figure 21. Torque application on front and rear wheel vs. time with feedback integral

5. Simulation results

The PI controller is used on the metal strip and the RS bump with a vehicle speed of 20 km/h. Each case has been analysed in the time domain and frequency domain with a comparison between an enabled and disabled controller.

5.1 PI controller on metal strip and RS bump

Since the controller uses the hub acceleration as input it is of great interest to see if there are any improvements regarding vibrations and acceleration amplitudes on this area. In figure 22, the first peak of the impact harshness shows a minimal improvement while the second and third peaks are improved by a great amount.

The after-shake in the case of metal strip shows no significant improvements although the amplitude is much lower than the impact harshness. In figure 23 with the RS bump, the after-shake is worse. In the case of RS bump the wheels are in the air for a moment and a reason for the oscillations can be a non-robust controller that over-reacts when the wheel hits the ground after the bump.

Although the oscillations are significant on the hub it has very little effect on the seat rail longitudinal acceleration. However, this is still something that needs to be resolved since this kind of behaviour is not good for the actuators and the components on the vehicle.

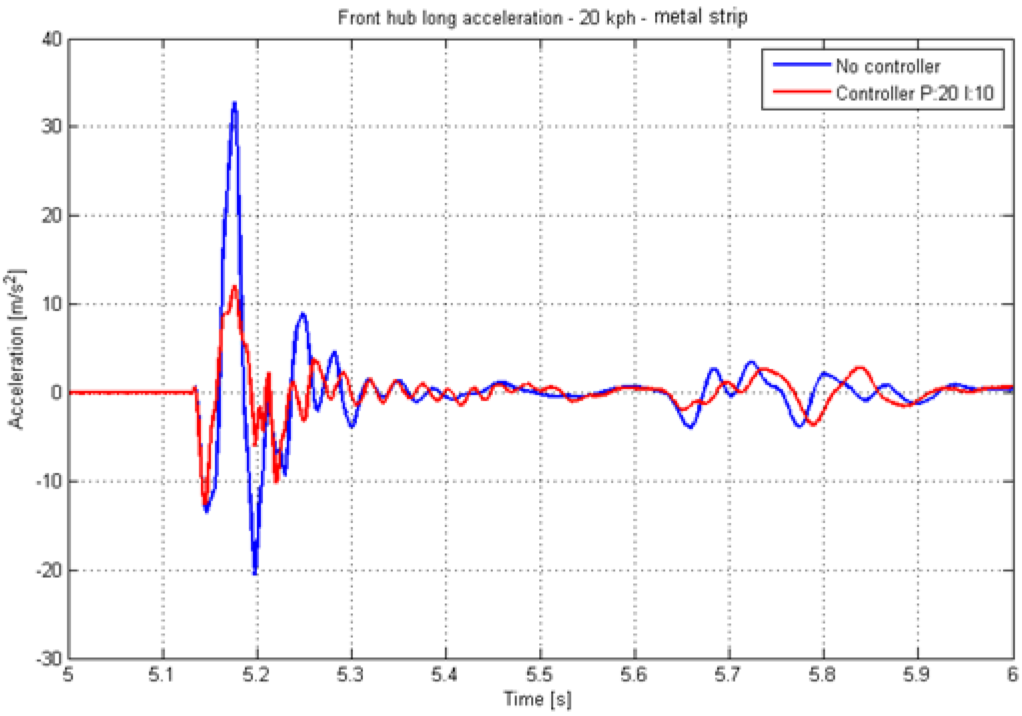


Figure 22. Front hub long acceleration over metal strip with and without a controller enabled

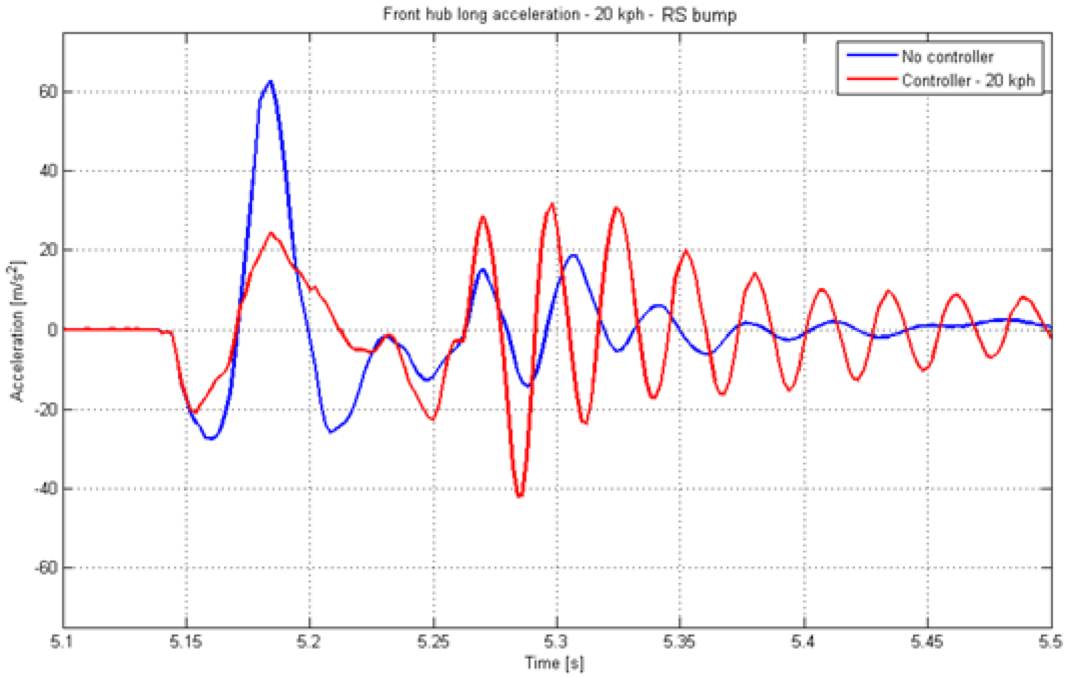


Figure 23. Front hub long acceleration over RS bump with and without a controller enabled

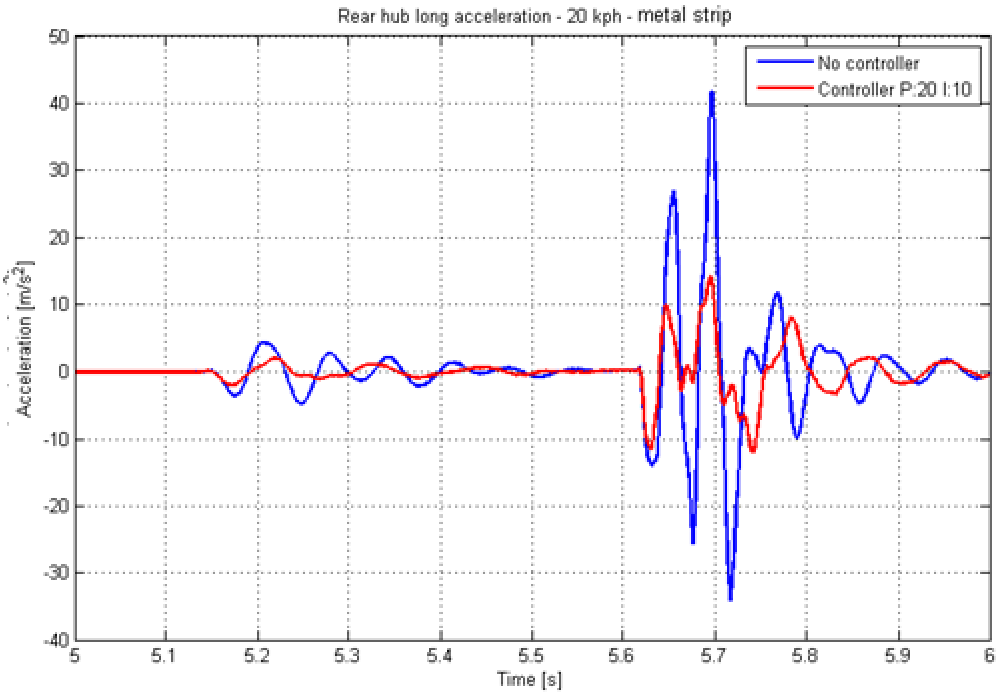


Figure 24. Rear hub long acceleration over metal strip with and without a controller enabled

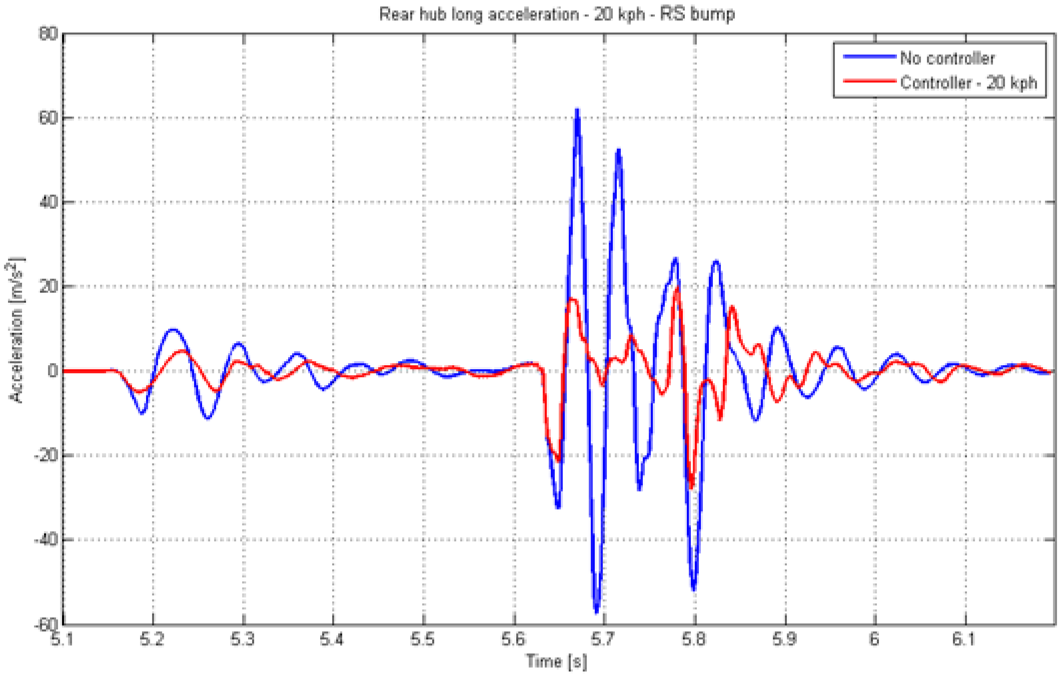


Figure 25. Rear hub long acceleration over RS bump with and without a controller enabled

5.1.1 Seat rail longitudinal acceleration

Looking at the seat rail acceleration, which is what the occupants of the vehicle are subjected to and which ultimately needs to be mitigated, it is evident that the peaks of impact harshness are significantly improved. The peaks of the impact harshness are lowered by 50-80 % in figure 26.

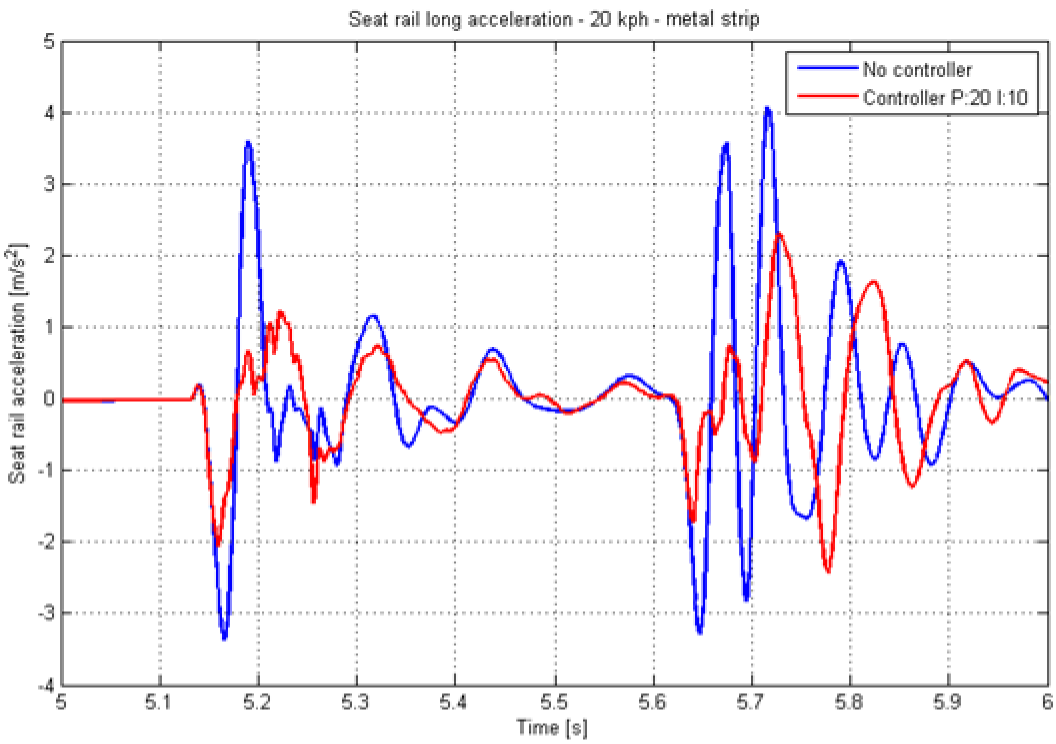


Figure 26. Seat rail long acceleration over the metal strip with and without a controller enabled

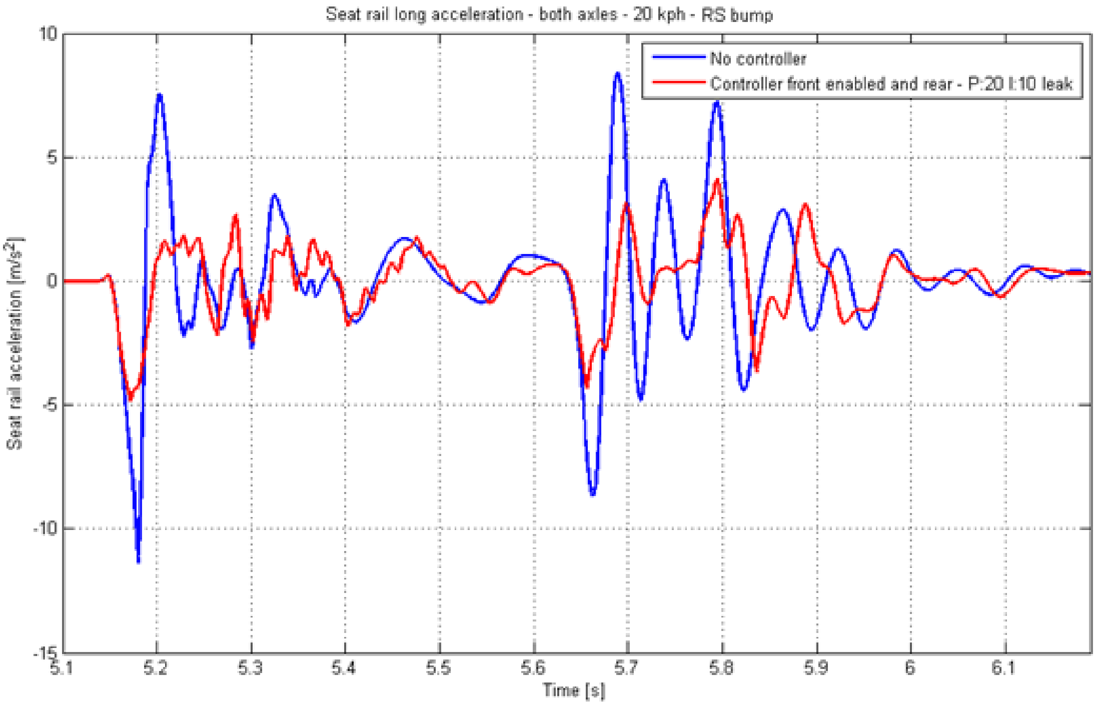


Figure 27. Seat rail long acceleration over the RS bump with and without a controller enabled

Looking at the power spectral density of the longitudinal seat rail acceleration, the magnitude of the power can be analysed for certain frequencies. In the range between 0-5 Hz where internal organs has a resonance frequency, no improvements are made, however the exposure of this specific frequencies is in the order of milliseconds and therefore does not have huge effect on the human body. Major improvement is done in the area between 10-25 Hz on the RS bump in figure 28 and between 15-25 Hz on the metal strip in figure 29. The effect of this improvement is that since the natural frequency of components in the suspension is in the range of the improved spectra, it should lead to a lower NVH and a perception of more comfortable and robust vehicle.

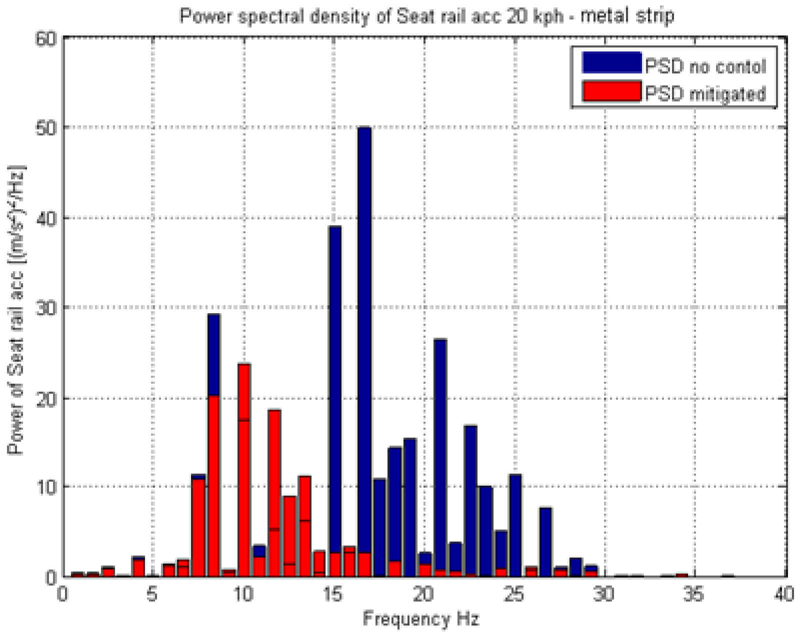


Figure 28. Power spectral density comparison between mitigated and non-mitigated on the metal strip.

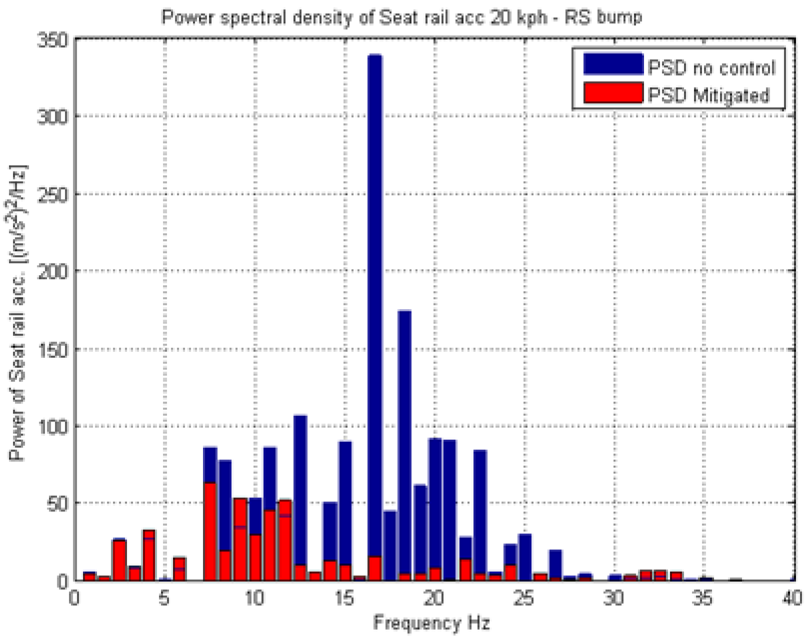


Figure 29. Power spectral density comparison between mitigated and non-mitigated on the RS bump.

5.1.2 Torque profile and torque limitation

In order to determine the specifications of an actuator, the first thing to look at is the amount of torque that needs to be supplied to mitigate the vibrations. The torque required to mitigate the vibrations in the case of the RS bump, reaches as high as 800 Nm per wheel as can be seen in figure 30. It would be very difficult to find actuators that could provide this amount of torque in a short period of time. For this reason, an investigation is done on how much the torque can be limited compared to the maximum torque and still give a significant improvement. In figure 33, the torque is limited to 250 Nm and only a small worsening is visible.

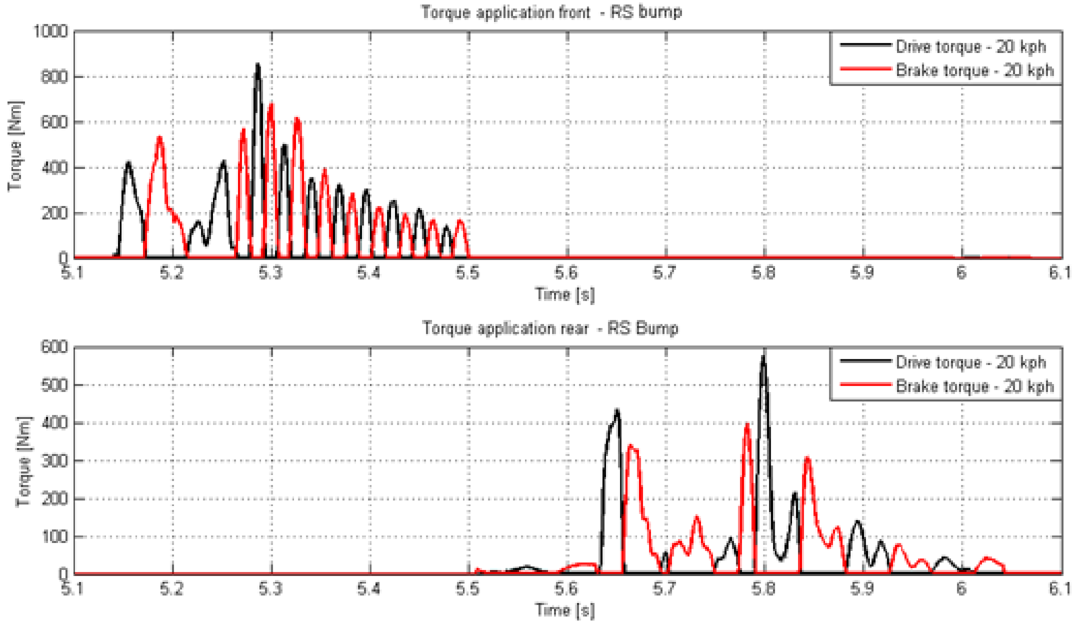


Figure 30. Drive and brake torque applied on the front and rear axle respectively

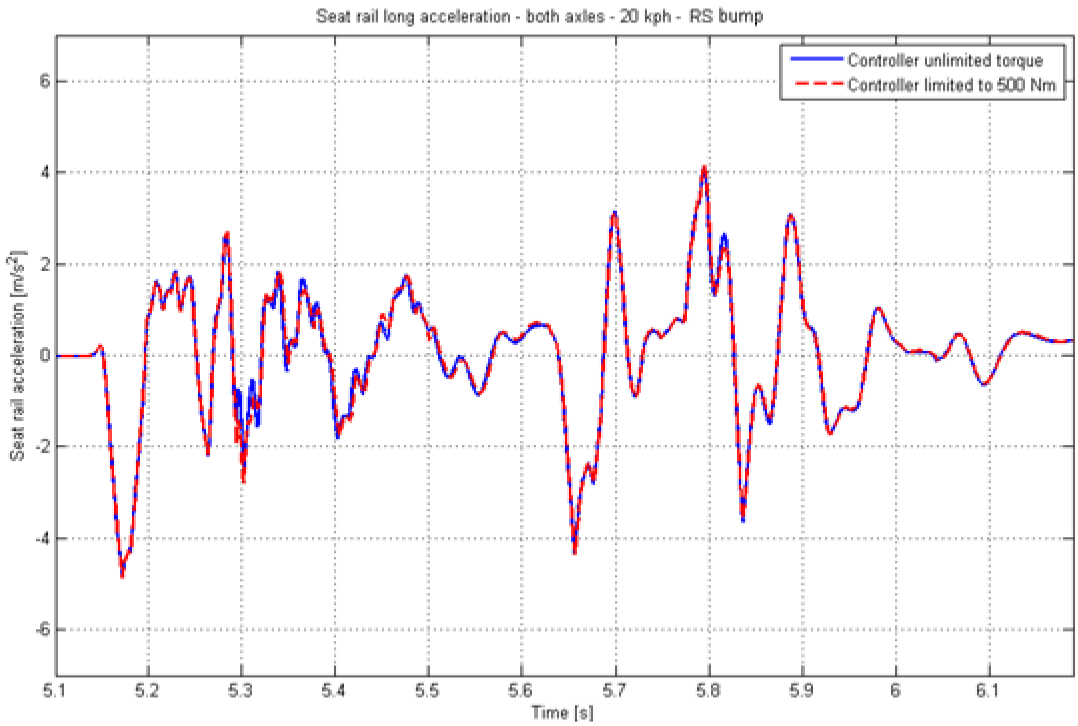


Figure 31. Sensitivity analysis of limited torque to 500 Nm compared to unlimited

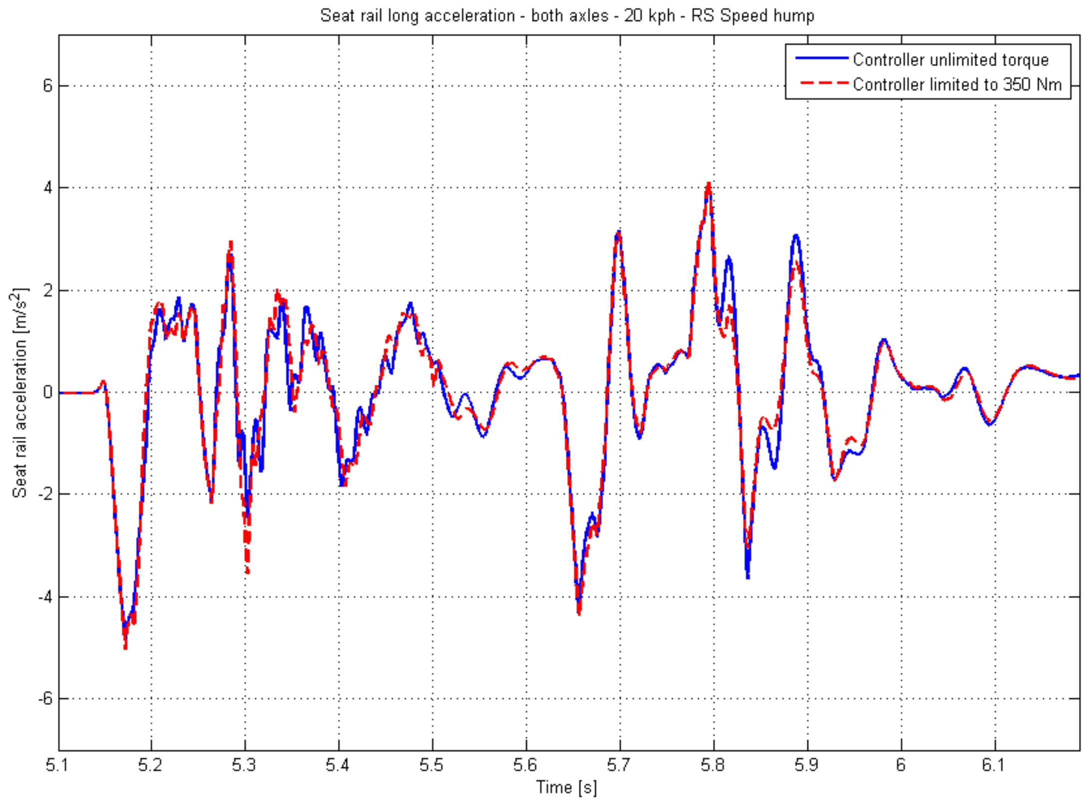


Figure 32. Sensitivity analysis of limited torque to 350 Nm compared to unlimited

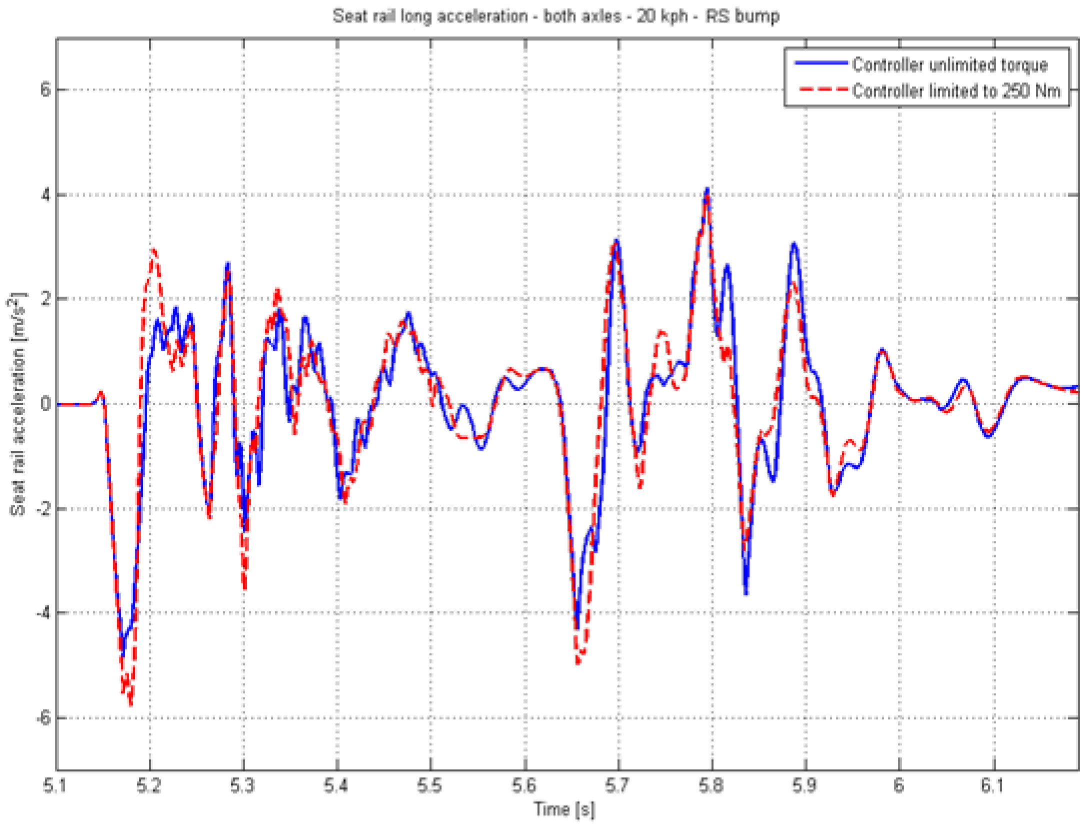


Figure 33. Sensitivity analysis of limited torque to 250 Nm compared to unlimited

5.1.3 Response times of torque demand

Another important factor to investigate is the shortest response time needed from an actuator. In table 6, the duration of torque applied is between 18-44 milliseconds which mean that the actuator needs to be able to vary the torque within this time range.

	Front axle		Rear axle	
	Impact	After-shake	Impact	After-shake
Drive	34 ms	20 ms	26 ms	24 ms
Brake	44 ms	20 ms	40 ms	18 ms

Table 6. Torque times for RS bump at 20 km/h.

An example of an ABS system is presented in table 7 where a total time of around 60 milliseconds is needed to communicate with the internal systems in the vehicle, apply pressure and decrease pressure.

	Event	Time [ms]
1	Communication and processing	5
2	Pressure model decrease	5
3	Actual pressure decrease response	8
4	Pressure decrease completed	30
5	Pressure model increase begins	5
6	Actual pressure increase response	8
	Total time	**61**

Table 7. ABS system response times - Bosch

5.1.4 Control system delay

In a real vehicle there will always be a delay from the time the system senses a disturbance to the moment it reacts to it. Compared to a simulation environment which reacts according to the sample rate, the real system will have some delay in communication between the control units and the actuators. To test the sensitivity of the system, a delay is introduced. Looking at figure 34, a delay of 20 milliseconds does not affect the impact harshness by very much but the after-shake is worse. In figure 35, with a delay of 40 milliseconds the result is much worse in both impact harshness and after-shake. It is also noted that the after-shake, in the case of 40 milliseconds delay is much worse for the front axle then in the rear. This might have something to do with the instability of the controller as previously seen in figure 23 which also shows greater after-shake amplitude for the front axle.

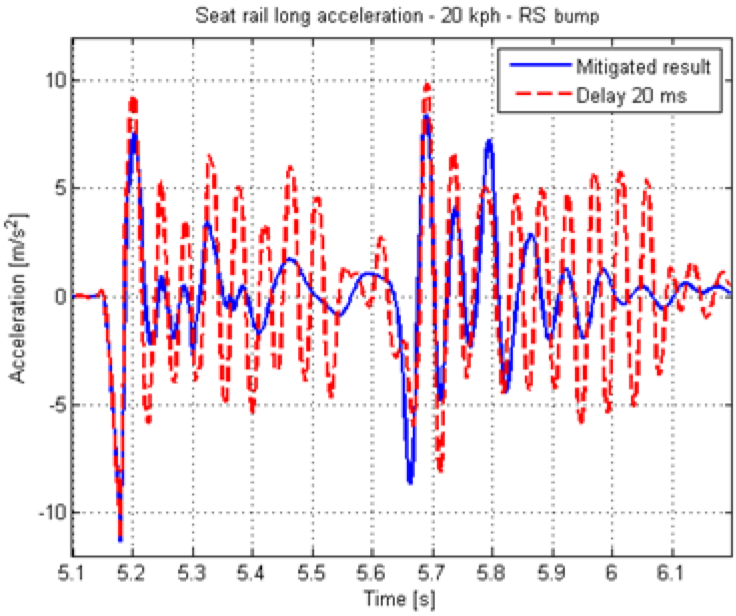


Figure 34. Mitigated seat rail long acceleration compared to a delayed controller of 20 ms

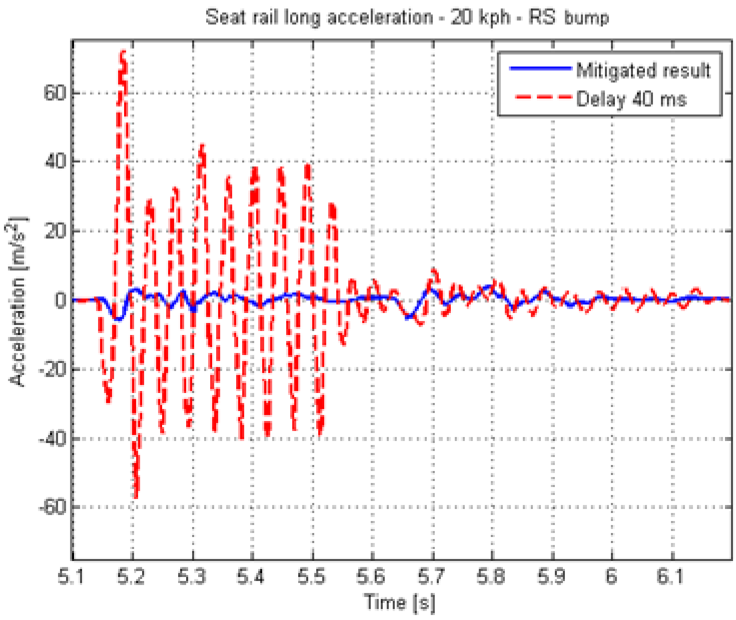


Figure 35. Mitigated seat rail long acceleration compared to a delayed controller of 40 ms

6. Discussion

The results shows that in theory it is possible to mitigate the road induced longitudinal accelerations however in reality there are some limitations regarding actuators, sensors and signal processing.

An actuator for an ideal system needs to work in the area between 20 Hz to 55 Hz. In a real system an actuator needs to be significantly faster because of delays in the entire system (i.e. sensors, signal processing, decision making) which makes it challenging to find actuators that satisfy the demands.

The self-learning system uses the information provided by the camera in order to apply correct torque with a specific timing on the obstacle. Therefore the accuracy of the information provided by the camera has to be high which can be hard to reach with the technology currently available.

In the case of the active system, the cameras are only used to activate the controller for a certain distance enabling the system on discrete events. In this case there is no need for high accuracy since the controllers can be enabled for a longer time than the duration of the bump. The system is thereby only active over discrete speed lowering obstacles and not activated over stochastic road irregularities.

A significant signal processing time delay is present which ultimately affects the performance in the active system. However in a self-learning system there is a possibility of compensating the delays by sending the signals before the disturbance.

7. Conclusion

The investigation done during this project show that it is definitely possible to significantly mitigate the longitudinal vibrations felt in both wheel hub and seat rail. The results show improvement in lowering the high accelerations peaks of impact harshness. With the controller activated, the amplitude of the oscillation after the impact peaks is amplified in the case of front hub acceleration. However the rear axle does not show the same effect. The reason for this can be found by investigating the complex model used in the simulations or developing a more stable and robust control system. Designing the controller with a variable gain and thereby being active initially and decreasing with time could be investigated to reduce the remaining oscillations.

Initial results showed a very high torque demand during the event. However further investigations have showed lower torque demands can be used without significantly affecting the improvement. This allows for more realistic torque demand and flexibility in choosing actuators.

Testing the sensitivity of the sensors by applying a delay in the control system shows that the system can still mitigate the impact harshness.

It is difficult to draw any conclusion on the unstable after-shake whether it is insufficient stability of the controllers or the time delay affecting the vibrations. Therefore, it is recommended to further investigate the choice of controller and how the torque is applied in the model. Additionally, a recommendation is to investigate that the powertrain internal components are modelled correctly.

8. Alternative methods of mitigation

This project aims to investigate the possibility of mitigating the vibrations with torque modulation but it is worth mentioning that the problem can be mitigated through other methods.

8.1 Longitudinal dampers and springs

Similar to a damper and a spring in vertical direction to take up the movement of the wheel, the same can be done in longitudinal direction as can be seen in figure 36 (Zhu, 2011).

Advantages: Higher dissipation of forces

Disadvantages: Heavier & packaging

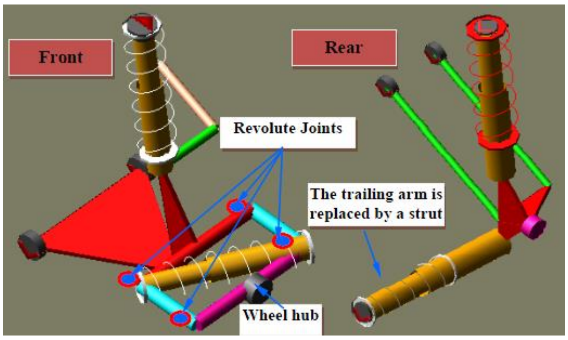


Figure 36. Longitudinal dampers and springs. (Zhu, 2011)

8.2 Semi-active dampers

A damper is a device that dissipates kinetic energy and reduces the amplitude of oscillations. The amount of reaction force generated from the damper against the oscillations is related to the velocity of the vibrations (i.e. road input). The damper is usually designed with a nonlinear force-velocity characteristic and is the most commonly used damper.

Suspension systems can be categorised as active, semi-active and passive ones. The traditional suspension is passive due to the unchangeable characteristics of force-velocity. For active systems energy is partly added to the vehicle motion whereas for semi-active and passive systems energy can only be dissipated. Active suspension is being controlled through external actuators, applying an external force on each individual wheel suspension to control the wheel motion. Semi-active systems changes the damping coefficient, resulting in variable damping in rebound and compression. The system works stepwise or in real time using different technologies such as a solenoid activated system.

Various technologies can be found; valve based systems as well as systems based on magneto- or electro rheological fluids. Magneto rheological damper is a semi-active system in the sense of changing the viscous coefficient in the fluid inside the damper. The fluid contains iron particles and the damper piston contains an electromagnetic coil that can generate magnetic flux across the fluid passages.

By controlling the viscous coefficient or having controllable valves one can have damper characteristics suitable for comfort situations and later changing the properties to suit handling manoeuvres where a stiffer damper is needed. (Villarreal, Wilson & Abdullah, 2004) Other types of semi-active damping techniques are used in suspension systems that can be applied here.

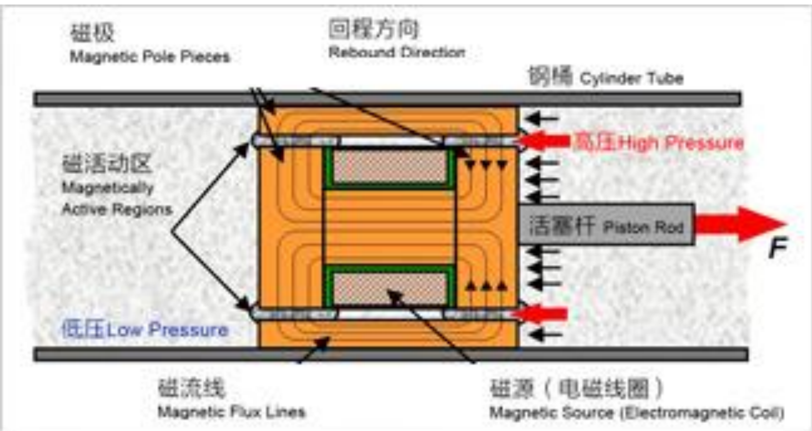


Figure 37. Description of a MR damper (BWI Group, 2014)

9. Future work and recommendations

Recommendations for future work would be to investigate how the hypothetical controller can be implemented in an actual vehicle. Since there is a difference between simulation and test data, the system would need to be implemented on a real car to validate the performance and tune the system to mitigate the vibrations caused by the impact.

- **Improvement of PI controller**

  Deeper study of the instability of the system during the after-shake needs to be conducted in order to improve the controller.

- **Self-learning system - Inverse transfer function**

  An investigation of why the inverse transfer function does not model the system correctly has to be performed. If an inverse transfer function that gives correct torque profile is achieved, an alternative way of mitigating the seat rail acceleration can be used. A recommendation is to start comparing the relevancy of the calculated transfer function (figure 16) to the simulation results gathered from Simpack.

- **Sensitivity analysis on torque**

  Perform simulations to investigate the sensitivity of the system by applying wheel torque with different durations, phase-offset and amplitude.

- **Sensitivity analysis on time delays**

  In a realistic application understand the sensitivity of timing issues and how to mitigate these.

10. References

- Zhu, J. J. (2011). *Study of Vehicle Dynamics with Planar Suspension Systems (PSS).* Waterloo, Ontario, Canada: University of Waterloo
- Rimell, A. N. & Mansfield, N. J. (2007). *Design of Digital Filters for Frequency Weightings Required for Risk Assessments of Workers Exposed to Vibration.* Loughborough University, Loughborough, Leicestershire, UK
- Villarreal, K., Wilson, C. & Abdullah, M., (2004). *Effects of MR damper placement on structure vibration parameters.* Tallahassee, Florida, USA: FAMU-FSU College of Engineering
- Åström, K. J., & Murray, R. M. (2012). *Feedback Systems, An introduction for Scientists and Engineers.* Princeton University Press, Princeton, USA
- Thomas, B. (2008). *Modern Reglerteknik.* Bertil Thomas och Liber AB, Stockholm, Sweden
- BWI Group. (2014) *MagneRide* Available at: http://www.bwigroup.com/en/images/oe1_img2_1.jpg (Accessed: 2014-01-24)

Appendix A – Simulink Model

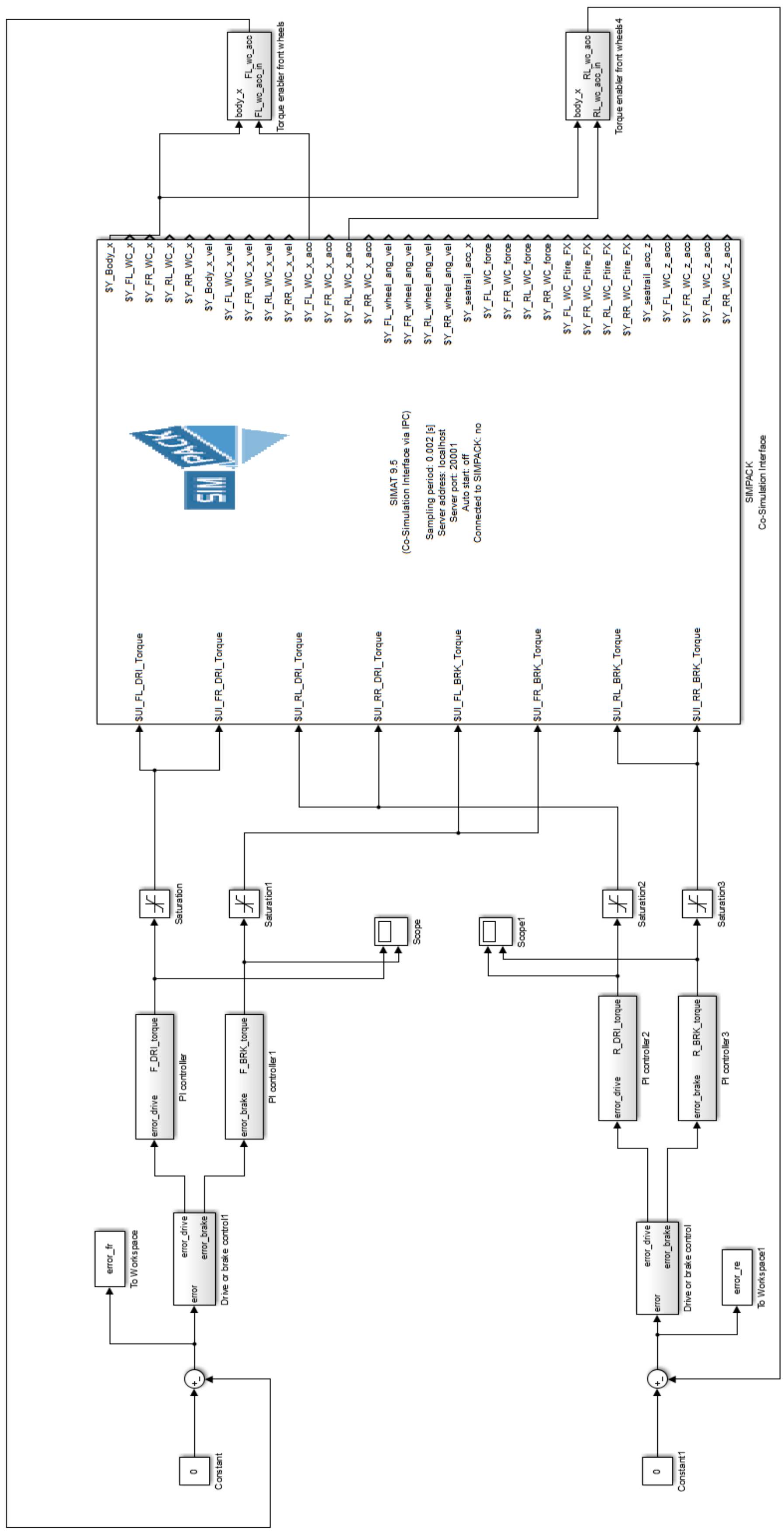


Figure 38. Overall layout of Simulink model

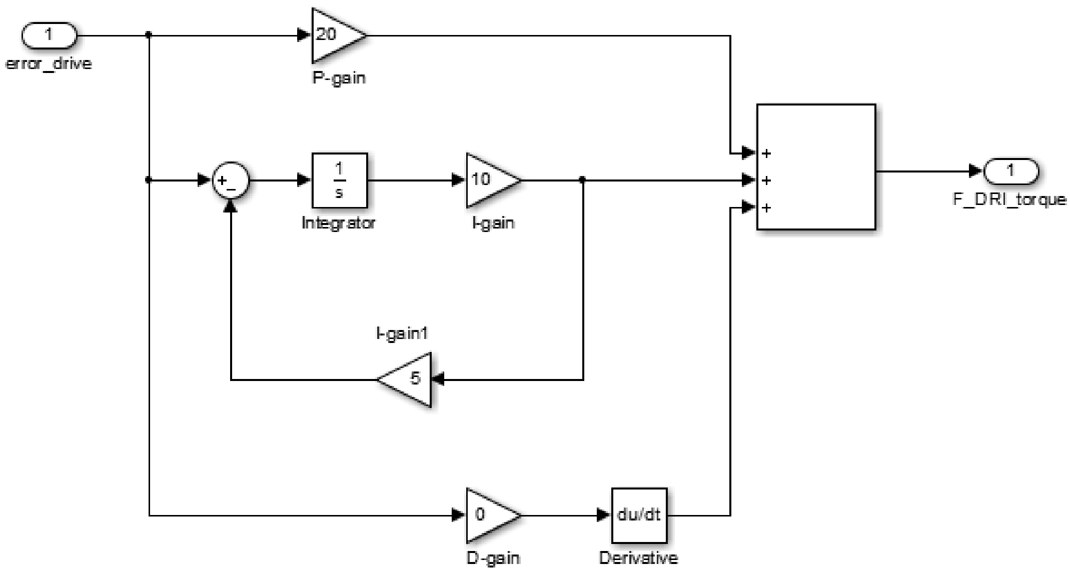


Figure 39. Controller in the feedback loop. One for both front/rear and drive/brake torque.

Appendix B – Matlab code for Simpack simulation

TORQUE ENABLER WHEELS

```
function FL_wc_acc = fcn(body_x,FL_wc_acc_in)

if body_x > 27.5 && body_x < 30.5

   FL_wc_acc=FL_wc_acc_in;

else

   FL_wc_acc=0;
end
```

DRIVE AND BRAKE CONTROL

```
function [error_drive, error_brake]= fcn(error)

    if error > 0 % If the error is positive; the hub acceleration is negative, apply a drive torque.

   error_drive = error;
   error_brake = 0;

    elseif error < 0 % If the error is negative; the hub acceleration is postive, apply a brake torque.

   error_drive = 0;
   error_brake = -error; % In the SIMPACK the brake torque has to be positive input.

   else
   error_drive = 0;
   error_brake = 0;
   end
```

Appendix C – Matlab code for test data

```
clc
clear all
close all

filename1 = 'gaydon_impact_strip_data.xlsx';
A = xlsread(filename1);
A=A(1:10136,:);
sr=2000;
filename2 = 'simpack_impact_strip.xlsx';
B = xlsread(filename2);
c = zeros(3450,1); d = zeros(3885,1);
A_x = [c; B(:,2); d];
% Plotting Seat X-acceleration
figure(1);
plot(A(:,1),(0.35+A(:,2)).*9.81); % Raw data
hold on
plot(A(:,1),A_x,'r','LineWidth',2)
xlabel('Time [s]')
ylabel('X-Acceleration [m/s^2]')
title('Seat rail longitudinal acceleration')
legend('Raw impact data','Simpack simulation')
% % Filtered
% [b,a] = butter(3,50/(sr/2),'low');          % IIR filter design
% y = filtfilt(b,a,A(:,2));                   % zero-phase filtering
% plot(A(:,1),(0.35+y).*9.81,'m','LineWidth',1.5);


grid ON
```

Appendix D – Simulation case in Simpack