Chapter 5

THE RIDE COMFORT VS. HANDLING DECISION

This chapter describes and analyses various methodologies that can be used to make the decision whether the suspension should be set to “ride comfort mode” or “handling mode”. This is referred to as the “**ride comfort vs. handling decision**”. It does not attempt to discuss or investigate possible control strategies for ride comfort and/or handling respectively. It rather assumes that these characteristics and control methods are known, *i.e*. that a set of “optimal” suspension characteristics and/or control laws exist for both ride comfort and handling. These two sets of conditions are in conflict as described in chapter 2. The importance of the ride comfort *vs.* handling decision cannot be overemphasized, as it is a safety critical decision. If the suspension system for example switches to the “ride comfort” mode during a severe handling or accident avoidance manoeuvre, the consequences might be severe and loss of control or rollover might result.

For the purposes of this study, the $4S_4$ will be switched to the soft spring and low damping characteristics when ride comfort is required and will switch to high damping and the stiff spring when handling is required. The effects of ride height on ride comfort and handling is excluded from the analysis. All the analyses will be made with the suspension set to the same ride height as the baseline suspension system. Effects caused by acceleration (*e.g.* squat) or braking (*e.g.* dive) are neglected at present. Figure 5.1 indicates where the ride comfort *vs*. handling decision fits into the study.

The aim of the present chapter is to find a strategy that uses parameters that can be measured directly, or otherwise easily calculated from direct measurements. This excludes the use of state estimators, integrators and artificial intelligence techniques such as neural networks.

No literature was found that is directly applicable to the ride comfort *vs.* handling decision as applied to off-road vehicles or controllable springs, although some of the concepts proposed by different authors are worth exploring and will be discussed now.

5.1 Literature

Stone and Cebon (2002) investigate semi-active roll control of a heavy vehicle. They make use of a system where an anti-rollbar is connected to the vehicle body with hydraulic cylinders providing switchable roll stiffness. The anti-rollbar can either be “free” (*i.e*. transmit little force) or “locked” (*i.e*. provides high roll stiffness). The low roll stiffness is intended for use when the lateral forces on the vehicle are small, thus providing good ride comfort. When large lateral forces are present, the system is switched to higher roll stiffness to improve handling. The vertical bounce stiffness of the suspension system is therefore unaffected. Although only a preliminary analysis is performed, the authors differentiate between a case where the lateral acceleration builds up slowly (general driving) and a case where rapid increases in lateral acceleration takes place (avoidance manoeuvres). For a rapid increase in lateral acceleration, using a **lateral acceleration threshold as control input** seems reasonable if control system delays are small. For the case of a slowly increasing lateral acceleration, a more sophisticated control strategy (*e.g.* one that uses steering inputs) might be beneficial.

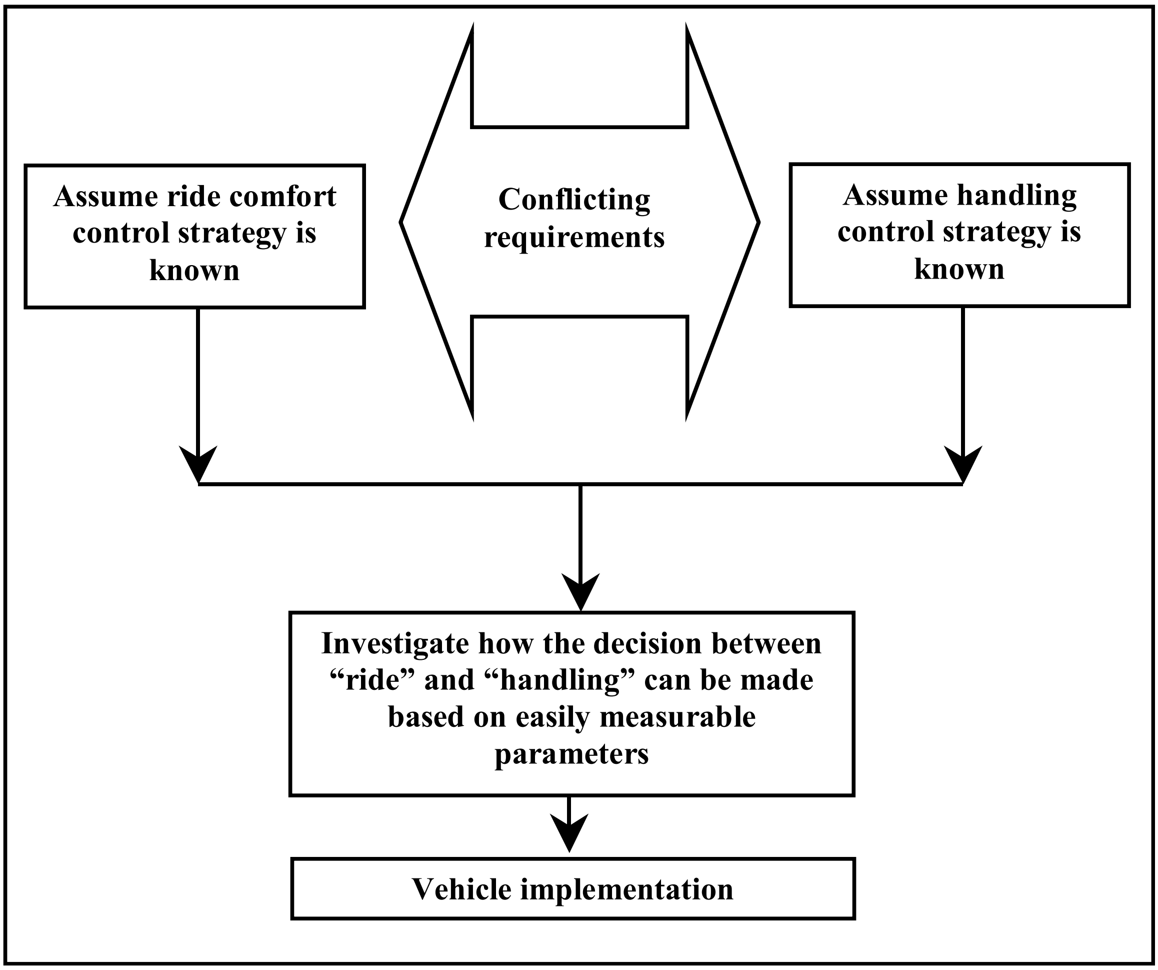


Figure 5.1 - The ride comfort *vs*. handling decision

Jost (2002a) describes the Continental Teve's four-corner air suspension with continuously variable semi-active damper control fitted to the Volkswagen Phaeton. The system adjusts damping force on each wheel within 10 to 15 milliseconds and automatically adjusts vehicle height. The system uses wheel acceleration sensors on the shock absorbers as well as body movement sensors (two at the front and one at the rear). Other inputs include data from the engine management, brake and electronic stability control systems. The system can recognise when the driver is steering into a curve. The driver can select between four fixed damper settings ranging from soft to sporty or firm. The control system will however temporarily override these settings when handling manoeuvres are encountered.

Nell (1993) and **Nell and Steyn (1998)** develop a general strategy for the control of two-state semi-active dampers in an off-road vehicle suspension system. Nell focussed on a full vehicle model taking all degrees of freedom into account instead of looking at each wheel separately. He defines suspension control as a “decision making” problem. The damper is switched to the high damping state whenever handling is required and controlled using a “minimum product” strategy whenever ride comfort takes preference. Roll movement over rough terrain is caused by suspension forces whereas lateral acceleration causes roll movement during handling manoeuvres on smooth roads. The ride comfort *vs.* handling strategy needs to differentiate between these two conditions. Nell measures and compares lateral and vertical acceleration on the centre of the (rigid) front axle. If this lateral acceleration is greater than the vertical acceleration, the handling mode (all dampers switched to high damping) is selected; otherwise the “minimum product” strategy is used to improve ride comfort.

Nell and Steyn (2003) apply the same basic idea to another off-road vehicle, in this case using measurements from two solid-state gyroscopes and two accelerometers as inputs to the control system. The control strategy is said to be a derivative of the method proposed by **Nell and Steyn (1998)**. It switches the two-state semi-active dampers to the conditions that will provide the highest accelerations opposing the motion of the sprung mass, or the lowest acceleration in the same direction. The relative damper velocities and absolute sprung mass velocity are no longer required. Handling is improved over the baseline vehicle by changing the “on” characteristic of the semi-active damper.

Darling and Hickson (1998) investigate the effect of an active anti-rollbar on the handling of a vehicle. They aim for a “flat” ride *e.g.* no body roll. They state that, although steering angle and vehicle speed can be used as control inputs, this relationship can vary significantly due to differences in tyre-road friction. They therefore make use of a lateral accelerometer mounted on the vehicle body in front of the centre of mass to measure a combination of lateral and yaw acceleration. A simple PID controller was implemented and the gains were optimised by a process of trial and error vehicle tests.

An electronic modulated air suspension system, as fitted to the 1986 Toyota Soarer is described by **Hirose *et. al.* (1988)**. The system is said to control spring rate, damping force and height with a response time of 70 milliseconds. There are three control steps namely: i) detection of vehicle travelling conditions, ii) classification of travelling conditions into one of several preset patterns, iii) adjust suspension parameters according to selected pattern. Sensors include three height sensors (left front, right front and left rear), steering angle sensor, throttle position, stop lamp switch and mode select switch. Vehicle height is detected in 16 steps between maximum and minimum height. Vehicle height is lowered if the vehicle speed exceeds 90 km/h and only increased again once vehicle speed drops to below 60 km/h resulting in a hysteresis of 30 km/h. On rough roads, height is increased above 40 km/h and only decreased again below 25 km/h to eliminate bump stop contact. Rough road conditions are detected by the left front wheel displacement using an observation duration of 0.5 seconds (half the sprung mass natural period). If the displacement measured during the observation duration exceeds a reference value four times in succession, ride height is increased. The detection period for changing ride height is 20 seconds to eliminate frequent ride height changes due to cornering for example. Spring and damper rates are changed simultaneously on all four wheels. Control of spring and damper settings are performed by either predictive control (see Table 5.1) or tracking control (see Table 5.2)

Table 5.1 – Predictive control as implemented by **Hirose *et. al.* (1988)**

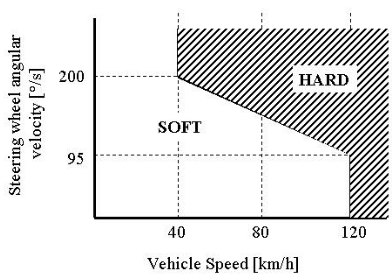
Situation	Sensor	Purpose
Anti-dive	Speed sensor Stop lamp switch	Suspension is changed to harder setting to restrict attitude change before the attitude change begins
Anti-roll	Speed sensor Steering sensor	
Anti-squat	Speed sensor Throttle position sensor	
Anti-bump	Speed sensor Height sensor	Irregularity of roads is detected by vertical movement of the front wheels and suspension is changed softer before the rear wheels pass through the detected irregularity to reduce shock

Table 5.2 - Tracking control as implemented by **Hirose *et. al.* (1988)**

Situation	Sensor	Purpose
Response to speed	Speed sensor	Suspension is set harder to improve travelling stability at high speed cruising. Since speed change is gradual, tracking control has satisfactory effect
Response to rough road	Speed sensor Height sensor	Suspension is set harder to restrict pitching and bouncing on rough road

A very similar system, fitted by Mitsubishi, is described by **Mizuguchi *et. al.* (1984)**. The suspension consists of air springs used in conjunction with coil springs and semi-active dampers. Sensors for vehicle speed, steering wheel angular speed, sprung mass acceleration (lateral, longitudinal and vertical), throttle speed and suspension stroke is used. Apart from ride height control, the suspension system can be switched from soft to hard quickly whenever any of the conditions in Table 5.3 are satisfied. The hard setting increases spring stiffness by approximately 50% and damping by 150%. The soft state is restored after 2 seconds in the hard state.

Table 5.3 – Strategy used by **Mizuguchi *et. al.* (1984)**

Case	Item	Sensor	Conditions
1	Vehicle speed	Vehicle speed	Soft to hard above 120 km/h Hard to soft below 110 km/h 10 km/h hysteresis
2	Steering speed	Steering wheel angular velocity	
3	Sprung mass acceleration	Acceleration sensor	Longitudinal acceleration: over 0.3g Lateral acceleration: over 0.5g Vertical acceleration: over 1g
4	Throttle speed	Throttle position sensor	Throttle wire moving speed: *over 0.25 m/s when accelerating *over 0.5 m/s when decelerating (with vehicle speed ≥ 3 km/h)
5	Front suspension displacement	Displacement sensor	Highest and lowest positions

Wallentowitz and Holdmann (1997) propose a frequency based control algorithm that generates high damping only when the vehicle is excited in the vicinity of the natural frequencies. They also propose a strategy where the vertical movement of each wheel is controlled individually, but with an overlying controller for roll and pitch movements. The spring must be switched to the stiff mode during braking and cornering to reduce roll and pitch angles. The soft spring is said to be only beneficial for frequencies lower than 5 Hz. No simulation or test results are given for the proposed controller.

Armstrong Patents Company Limited of York developed a practical intelligent damping system described by **Hine and Pearce (1988)**. The system consists of two or three state adaptive dampers combined with an auxiliary air spring to provide ride height control. Measurements indicate that reducing the damper setting below standard greatly improves vibration isolation at frequencies above 2 Hz, while higher than normal settings reduces the amount of motion around 1 Hz (roll *etc.*) The aim of the control strategy is to keep the damper in the lowest setting as long as possible and only switch to higher levels when required. Switching typically takes place in 12 milliseconds. The system has sensors for suspension movement, steering wheel angular velocity, vehicle speed, brake application and wheel or body acceleration. The control strategy is separated into a number of components namely:

i) Ride: Ride control is initiated by the relative suspension displacement and vehicle speed using displacement maps. If the displacement exceeds a pre-programmed limit for the specific vehicle speed, higher damping will be selected. For returning to the lower damper setting, the valves have been designed to delay until the pressure is below a preset limit to reduce hydraulic noise.
ii) Handling (including roll control): Handling is detected based on steering wheel speed and vehicle speed. Roll information, obtained from the suspension displacement sensors, is then used to switch back to the lower damper settings shortly after the vehicle returns to the level position.
iii) Acceleration: Information from the vehicle speed sensor is used to determine the acceleration. Damper rate is increased when acceleration exceeds a pre-defined level.
iv) Deceleration (Dive): Information from the brake and speed sensors is used to immediately switch the dampers to the hard setting. The system returns to the softer setting once the longitudinal acceleration drops to below a preset level.
v) Ride frequency control and vehicle levelling: Levelling compensates for changes in payload and aerodynamics.

The system was implemented on a 1986 model GM Corvette (5.7 litre) as well as a Ford Granada 2.8 Ghia. The improvements in ride comfort and handling is however not quantified.

An active suspension control approach that consists of an inner loop that rejects terrain disturbances, an outer loop that stabilises heave, pitch and roll response and an input decoupling transformation that blends the inner and outer control loops is proposed by **Ikenaga *et. al.* (2000)**. The ride control loop isolates the car body from uneven terrain while the attitude control loop maintains load levelling and load distribution during handling manoeuvres. Skyhook damping (the term used to describe the feedback of absolute sprung mass heave, pitch and roll velocities) improves heave, pitch and roll accelerations at all frequencies below the wheel frequency.

Truscott (1994) develops a composite controller for a high bandwidth (35 Hz) fully active suspension system. Only simulation results for a linear quarter car suspension system are presented. The proposed composite controller consists of two controllers operating together, but over different frequency ranges. The first controller cancels out low frequency dynamic loads experienced during cornering and braking, keeping the vehicle level. This controller operates at frequencies below 5 Hz. The second controller isolates the car from high frequency terrain induced vibration and operates at frequencies above 5 Hz. The vibration controller is fully adaptive and auto-tunes the system according to varying payload, tyre stiffness and varying road frequency spectrum. The 5 Hz frequency was chosen to be between the sprung mass and wheel-hop frequencies.

Trent and Greene (2002) propose a model-based genetic algorithm predictor to estimate the potential for rollover. The tyre deflection that will result in vehicle rollover approximately 50 time steps in future is calculated assuming all other operating conditions such as vehicle speed remain constant. Advanced rollover warning of 400 milliseconds may be possible, giving enough time for an intelligent suspension system or stability control (differential braking) system to react and decrease the rollover propensity.

Active roll control, as developed by TRW, is discussed by **Böcker and Neuking (2001)**. The system uses hydraulic cylinders fitted to the anti-rollbars. Figure 5.2 indicates the functioning of the control system schematically. Sensors include steering angle, lateral acceleration, hydraulic system pressure and vehicle speed.

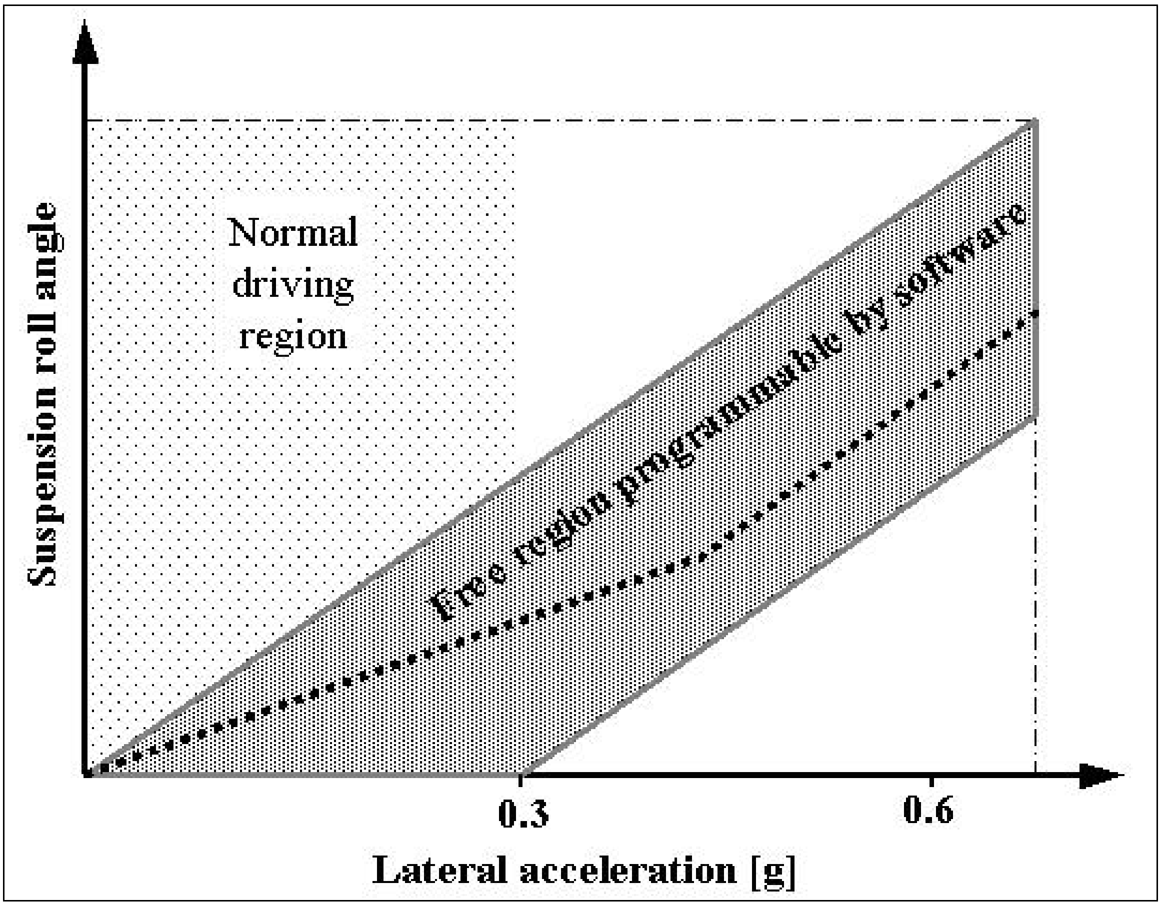


Figure 5.2 – TRW’s active roll control system according to **Böcker and Neuking (2001)**

Hamilton (1985) defines many general aspects for the theoretical operation of controllable suspension systems. No simulation or test results are given although prototype hardware was available. The proposed system consists of the following components:

i) *Ideal damping device*: must instantly provide the damping force required by the computer independent of suspension position or velocity.
ii) *Ideal energy storage device:* must be capable of changing it's energy storage capacity to a value demanded by the computer.
iii) *Ideal computer controller:* must have all the necessary inputs to calculate all required parameters.

The ideal theory of operation also consists of many aspects namely:

i) *Optimised ride comfort:* requires a very soft spring and virtually no damping force. Some force is required to control the kinetic energy of the wheel.
ii) *Cornering:* The centrifugal force on the vehicle's centre of gravity causes a torque on the sprung mass, about the roll centre, that can be counteracted by the damping device by applying equal forces in the opposite direction.
iii) *Ideal pitch control:* Attempt to maintain a level ride during acceleration or braking by looking at the change in pitch height.
iv) *Ideal level ride control:* This is required to compensate for substantial variations in the loading condition of the vehicle. Height control can also be used to decrease the frontal area of the vehicle during high speed driving or to increase ride height over rough terrain.
v) *Ideal roll control:* e.g. on a mountain road. Can be based on the difference in height between the sprung mass and the road surface on the left and right hand side of the car.
vi) *Ideal natural frequency control:* Observe the two primary natural frequencies (sprung and unsprung mass) using a Discrete Fourier Transform (DFT) and control each one of them separately.
vii) *Ideal high amplitude or high velocity control:* Road inputs that exceed the dynamic range of the suspension require forces to be applied to the sprung mass to move it up and over the obstacle. The magnitude of these forces should be such that the suspension movement limits are never exceeded.

Not all forces acting on the sprung mass can be totally eliminated. The system should aim to minimise them while optimally controlling vehicle movements within the suspension working space. There are many counteracting forces that are required simultaneously and these must be superimposed to control all the dynamics simultaneously.

The concept is to apply these forces when required and keep them as small as possible to ensure good ride comfort. The author states that only relative suspension movements need to be measured as all the other required parameters can easily be calculated from these.

The Mercedes-Benz Active Body Control (ABC) is described by **Birch (1999)**. The system was introduced on the CL Coupe and adapts both spring and damper characteristics to prevailing conditions. A hydraulic system (called a "plunger") acts on the coil spring to change the preload on the spring. The stiffness remains unchanged. The hydraulic system acts up to a frequency of 5 Hz thereby improving vehicle response to long wavelength road inputs as well as during braking and cornering. Anti-rollbars are not necessary and the system is also self-levelling. The driver can select sport and comfort settings.

A detailed description of the Citroën Hydractive I, II and III is beyond the scope of this text, but is described in substantial detail by **Nastasić and Jahn (2005)**. The control principles employed are however relevant to the ride *vs.* handling decision. The basic idea is to map different vehicle parameters against vehicle speed. Figure 5.3 indicates the steering wheel angle threshold as a function of vehicle speed. The suspension is switched to the handling mode whenever the measured steering wheel angle exceeds the threshold at a certain speed. The driver can select one of two different threshold levels by selecting a "normal" or "sport" mode with a switch. The steering wheel velocity threshold (Figure 5.4) exhibits a similar trend and operates on the same principle. At low vehicle speeds, large steering wheel angles and velocities are allowed *e.g.* during parking manoeuvres. As the vehicle speed increases the threshold levels become smaller, resulting in faster reaction times.

Body dive and squat (Figure 5.5) is determined by measuring the relative displacement of the front and rear suspension respectively. Threshold levels also decrease as vehicle speed increases. The accelerator pedal press and release rate is also used (Figures 5.6 and 5.7) to reduce squat and pitch during hard acceleration. Slow release of the accelerator pedal indicates that the driver desires to reduce speed gradually whilst a sudden release will often be followed by hard braking to quickly reduce speed. Dive and squat effects are further ignored for the present study.

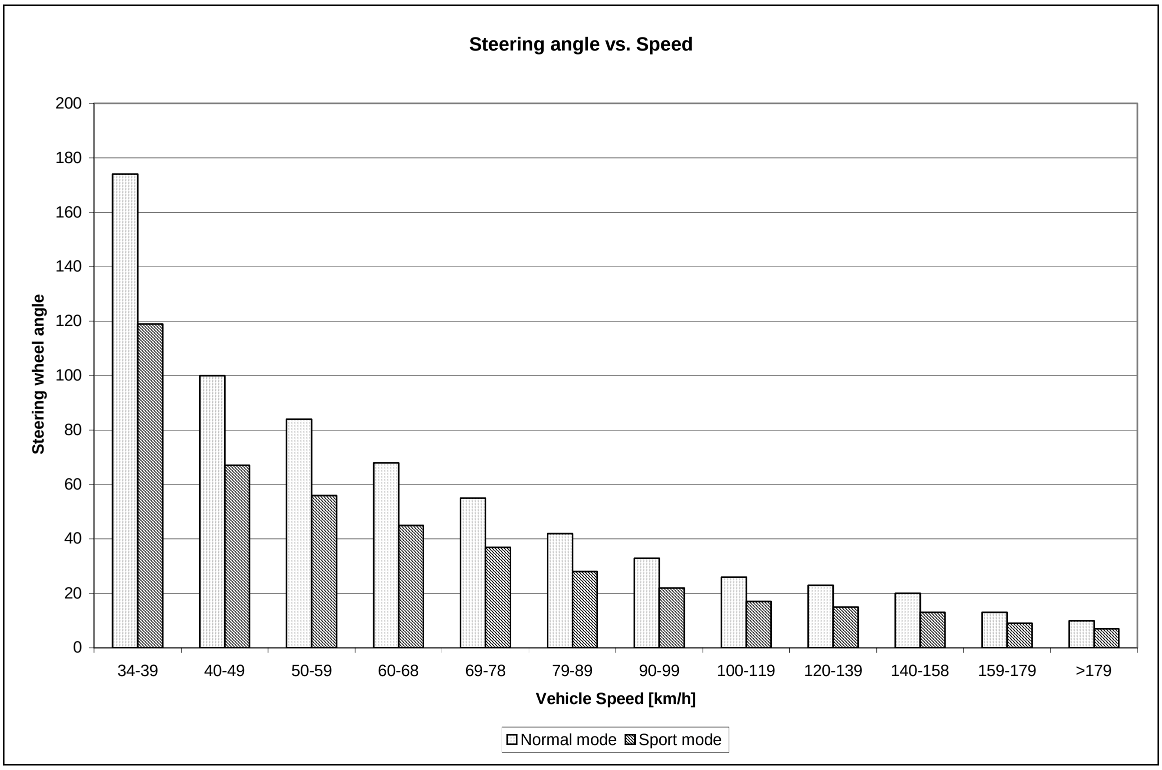


Figure 5.3 – Steering wheel angle *vs.* vehicle speed

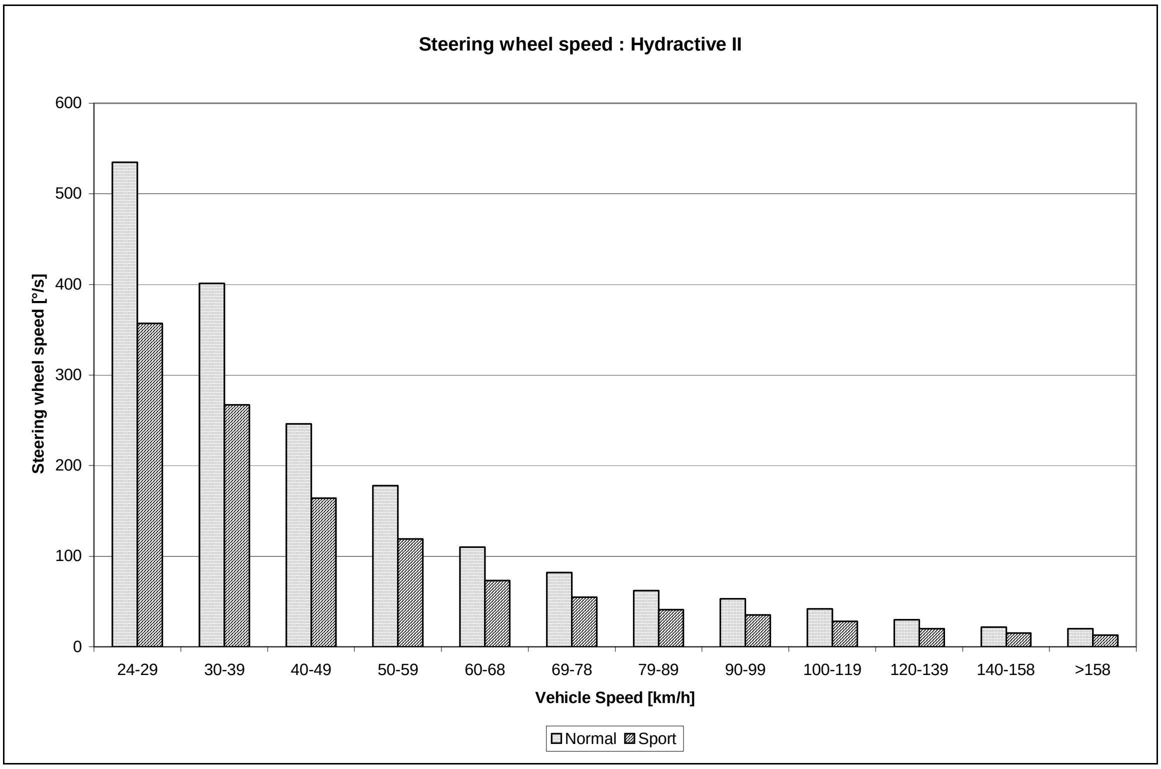


Figure 5.4 – Steering wheel rotation speed *vs.* vehicle speed

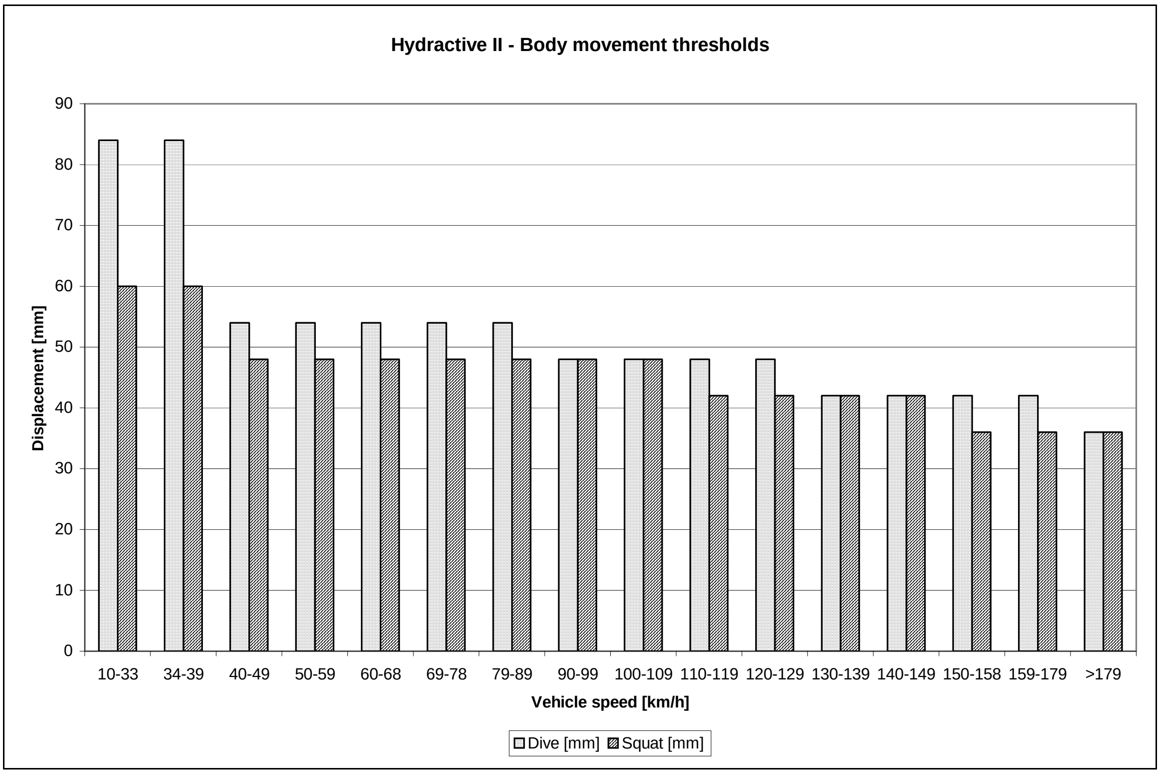


Figure 5.5 – Dive and squat *vs.* vehicle speed

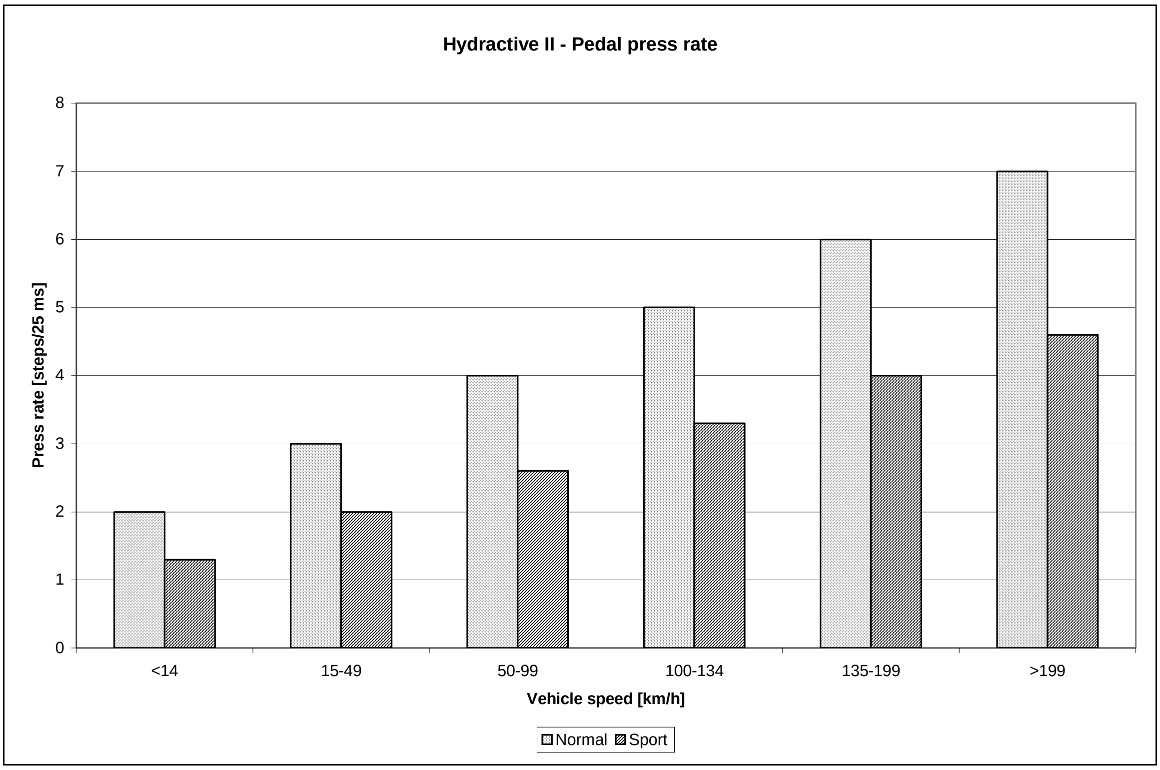


Figure 5.6 – Accelerator pedal press rate *vs.* vehicle speed

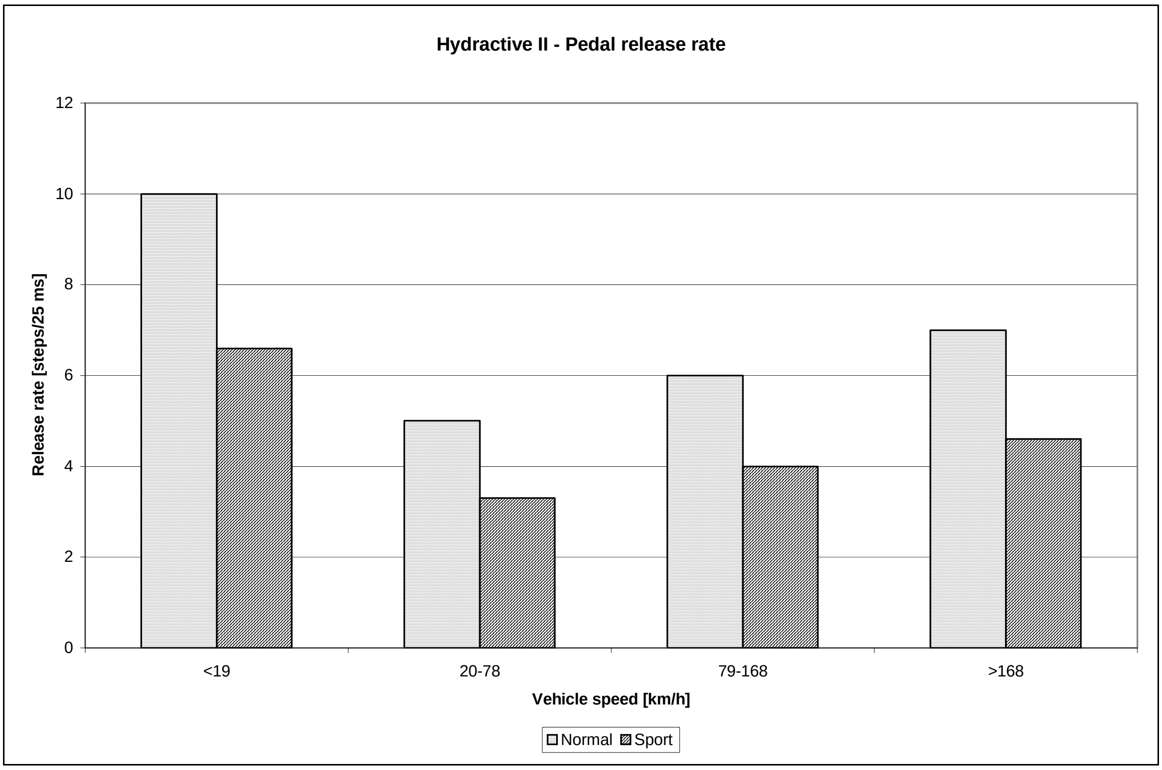


Figure 5.7 - Accelerator pedal release rate *vs.* vehicle speed

Interactive vehicle dynamics control on the 2000 Ford Focus is discussed by **Broge (1999)**. Although not controlling the suspension system of the car, the general concept may be applicable to the current study. Several parameters measured on the vehicle are compared to a dynamic handling map stored in the on-board computer. When any vehicle parameter deviates from the stored map, corrective action is taken by reducing engine power and braking appropriate wheels. Sensors include individual wheel speeds, steering wheel movement, yaw rate sensors and lateral accelerometers.

5.2 Suggested concepts for making the "ride comfort *vs.* handling decision"

From the literature discussed in paragraph 5.1, several concepts have been identified to assist in making the "ride comfort *vs*. handling decision". These concepts are listed in Table 5.4 and will be investigated further in paragraph 5.5. It is important to note that the majority of the applications discussed so far are related to road vehicles. Substantial differences might be required for off-road driving.

Table 5.4 – Suggested concepts for assisting with the "ride *vs.* handling" decision

Concept no.	Measurement parameters	Reference
1	Frequency analysis of acceleration	**Wallentowitz and Holdman (1997)** **Truscott (1994)**
2	Lateral acceleration *vs.* vertical acceleration	**Nell (1993)** **Nell and Steyn (1998)**
3	Steering angle *vs.* speed	**Hirose *et. al.* (1988)** **Hine and Pearce (1988)** **Nastasíc and Jahn (2005)** **Broge (1999)**
4	Pitch and roll velocity / acceleration	**Nell and Steyn (2003)**
5	Height, throttle position, brake application, mode select switch	**Hirose *et. al.* (1988)** **Hine and Pearce (1988)** **Nastasíc and Jahn (2005)** **Broge (1999)**
6	Lateral acceleration	**Stone and Cebon (2002)** **Daling and Hickson (1998)**

5.3 Easily measurable parameters

At the outset of this study, the decision was made to try and find a strategy that uses parameters that can be measured directly, or otherwise easily calculated from direct measurements. This therefore excludes the use of state estimators, integrators and artificial intelligence techniques such as neural networks. This decision was made based on several factors namely:

- This is the first study focussed on the "ride comfort *vs.* handling decision" for off-road vehicles
- No previous concepts or algorithms seem to exist
- Attempting to keep it as simple as possible and only as complicated as necessary
- The current focus is on a more fundamental understanding of the issues involved
- It is important to keep the cost of the sensors and control system within the project budget
- The "controller" to be used was a personal computer based system with an analog to digital converter card and a digital input-output card fitted. This excluded the use of digital signal processing (DSP) cards.

The parameters identified to be easily and directly measurable are listed in Table 5.5 while Table 5.6 lists parameters that can be easily calculated from the directly measured parameters. Displacement measurements can be differentiated with respect to time to give velocities. Although differentiation tends to add high-frequency noise, we are primarily interested in the low-frequency content of the velocity and good results can be achieved using simple mathematics and low-pass filters. Integration is also possible in theory, but creates many obstacles in practice due to the effect of drift. Small offsets in the zero reading of a sensor (*e.g.* accelerometer) can cause the integrated value to quickly drift to the limits. Because we are primarily interested in the low-frequency content, it is very difficult to control drift by for example high-pass filtering. The signal offsets are often influenced by effects such as change in temperature or attitude changes of the vehicle body due to varying load and road conditions. Absolute body movements can presently only be calculated by integrating acceleration signals twice. Absolute body movements are thus not easily measured directly, or calculated, and are therefore excluded at present.

Table 5.5 – Directly measurable parameters

No	Parameter	Position	Equipment
1	Vehicle speed	Roof	Global positioning system (GPS)
2	Relative displacement	Every suspension position	Rope displacement transducer
3	Angular velocity (roll, yaw, pitch)	Vehicle body	Solid state gyroscope
4	Relative displacement	Steering arm between axle and body	Rope displacement transducer
5	Acceleration	Vehicle body	Solid state accelerometer ±4g range
6	Kingpin steer angle	Kingpin	Potensiometer
7	Wheel speed	Any wheel	Optical speed sensor
8	Driveshaft speed	Gearbox output rear	Optical speed sensor

Table 5.6 – Parameters that can be easily calculated from measurements

No	Parameter
1	Relative suspension velocities
2	Relative angles between vehicle body and suspension components
3	Relative angular velocities
4	Angular accelerations

5.4 Experimental work on baseline vehicle

A test sequence, consisting of six different test routes and manoeuvres, was devised to be representative of the Land Rover Defender 110 vehicle's typical application profile. Tests were performed at representative speeds. For city and highway driving, tests were performed in and around the city of Pretoria. All other tests were performed at the Gerotek Vehicle Test Facility West of Pretoria. The legal speed limit was adhered to on all public roads. For off-road driving, the speed was determined by the driver's judgement of ride comfort while on the mountain pass the speed was limited either by vehicle performance on the steep uphill slopes or by handling around the corners. For the handling and rollover tests the speed constraint was the vehicle's handling combined with the driver's ability. The chosen test routes are summarised in Table 5.7 with the plan layouts of the test routes and tracks indicated in Figures 5.8 to 5.12.

It was also postulated that an objective vehicle parameter (*e.g.* lateral acceleration) might be correlated with an objective human physiologic parameter (*e.g.* heart rate or blood pressure). A series of tests were performed where heart rate and blood pressure was measured for both driver and passengers in attempt to obtain a correlation between vehicle parameters and change in heart rate. A total of 85 test subjects were used for the physiological measurements. Although very interesting trends were noticed, no correlation could be found between the measured physiological parameters and vehicle parameters.

Table 5.7 – Chosen tests and test routes

Test	Driving conditions	Test route	Driver	Duration [s]	Figure
1	City driving	Start: corner Dely & High (point 2) End: corner Rigel &Buffelsdrift (point 3)	Normal	704	5.8
2	Highway driving	Start: Fountains circle (point 4) End: corner Lynnwood & Kiepersol (point 5)	Normal	783	5.8
3	Off-road	Top 800m of Gerotek Rough Track	Normal	166	5.10
4	Mountain pass	Gerotek Ride & Handling Track - clockwise	Experienced	268	5.11
5	Handling	ISO 3888 Severe double lane change test	Experienced	13.4	5.12
6	Rollover	Fishhook rollover simulation test	Driving robot	7.1	5.9

5.5 Evaluation of concepts

The concepts identified in paragraph 5.2, and summarised in Table 5.4, will now be implemented on the test data measured on the baseline vehicle and evaluated. Although this approach is not strictly correct since the vehicle dynamics will change when the suspension settings change, this method is expected to illustrate trends and provide a first order evaluation of feasibility.

Only the first three concepts listed in Table 5.4 were investigated in more detail. Concept 4 is a “ride comfort” strategy while concept 5 focuses on the effect of longitudinal forces (*e.g.* due to acceleration or braking) on the vehicle and therefore not ride comfort or handling. Concept 6 is a “handling” strategy and ignores ride comfort.

At this point, the controllable suspension system is assumed to function as a two-state system that can be switched between a “ride comfort mode” and a “handling mode”.

Several requirements were set for evaluating the feasibility of a control strategy. These requirements are:

- Switching should not be too frequent, *e.g.* the strategy should not “hunt” between the “ride comfort” and “handling” settings
- The strategy should work for all six chosen tests without manual driver intervention.
- The strategy should rather err towards the “handling” mode.
- For both the handling test and the rollover test, the system should switch to the “handling mode” as quickly as possible, and remain in the “handling mode” for the duration of the manoeuvre. Ideally the “handling mode” should be selected before the start of the manoeuvre, but this is not possible without some kind of preview.
- During off-road driving, the system should remain in “ride comfort” mode for most of the time.

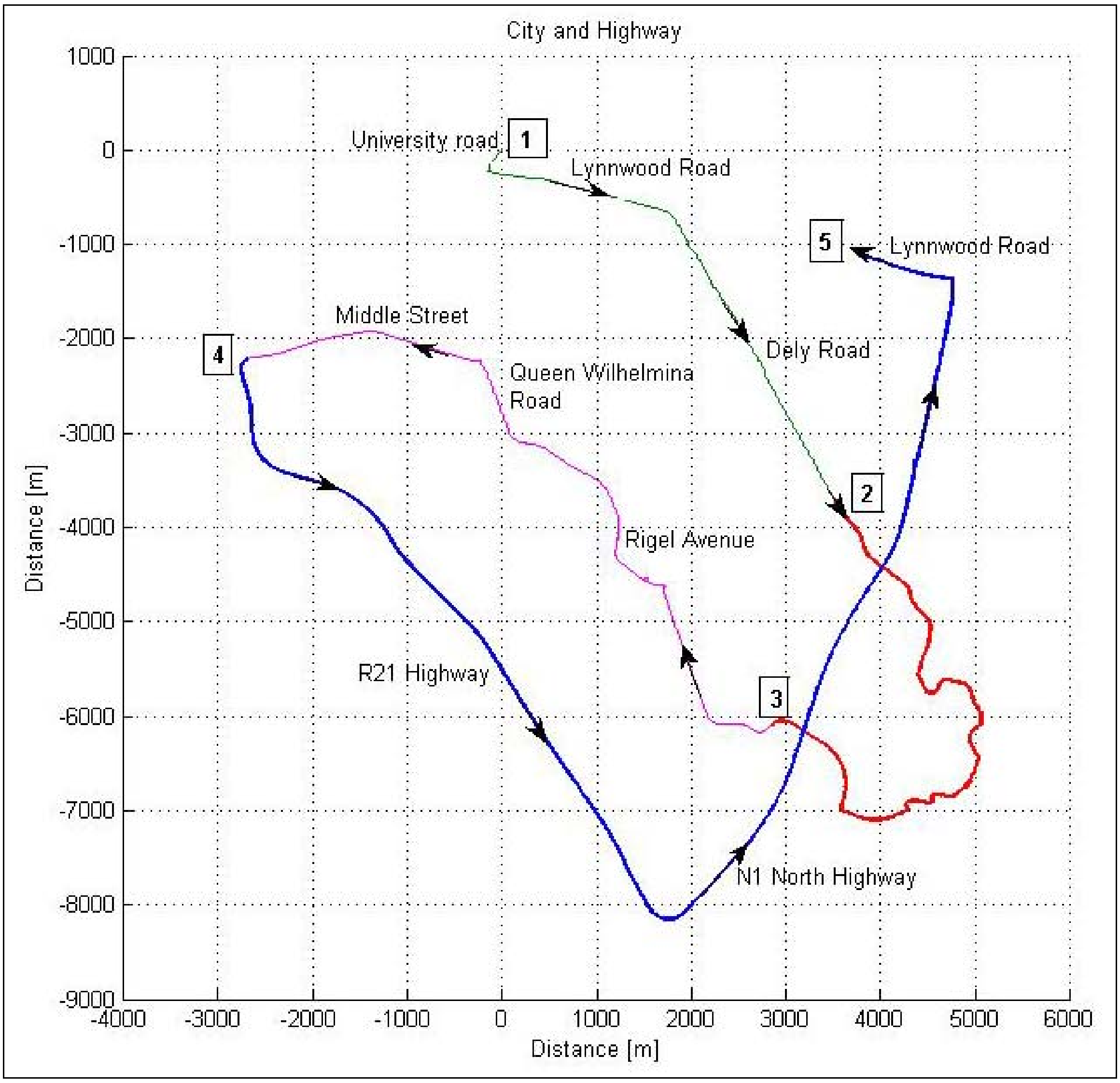


Figure 5.8 – City and highway driving route

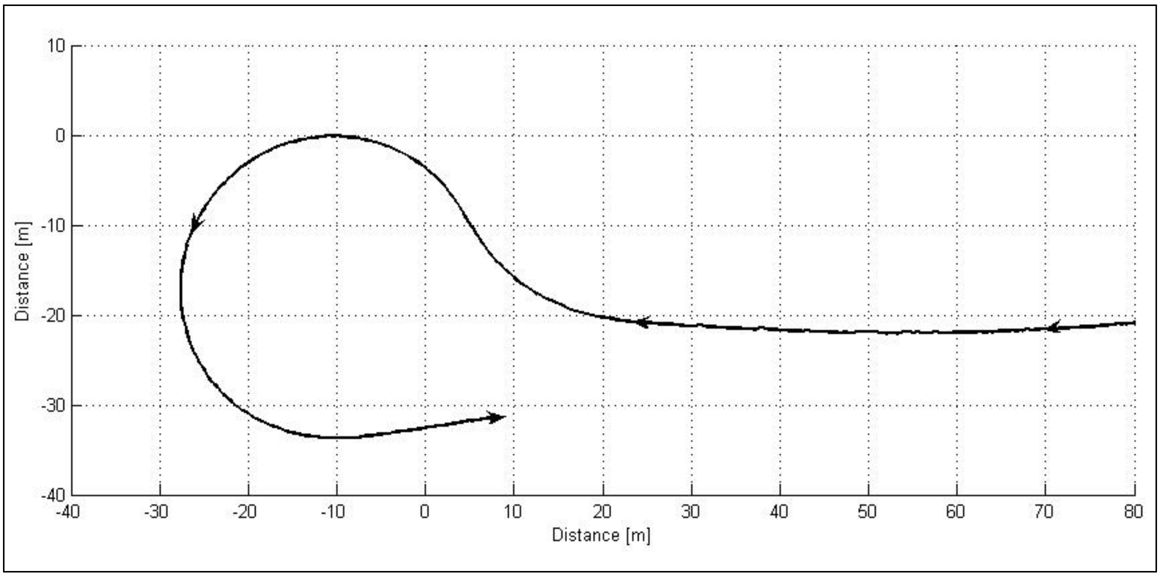


Figure 5.9 – Fishhook test

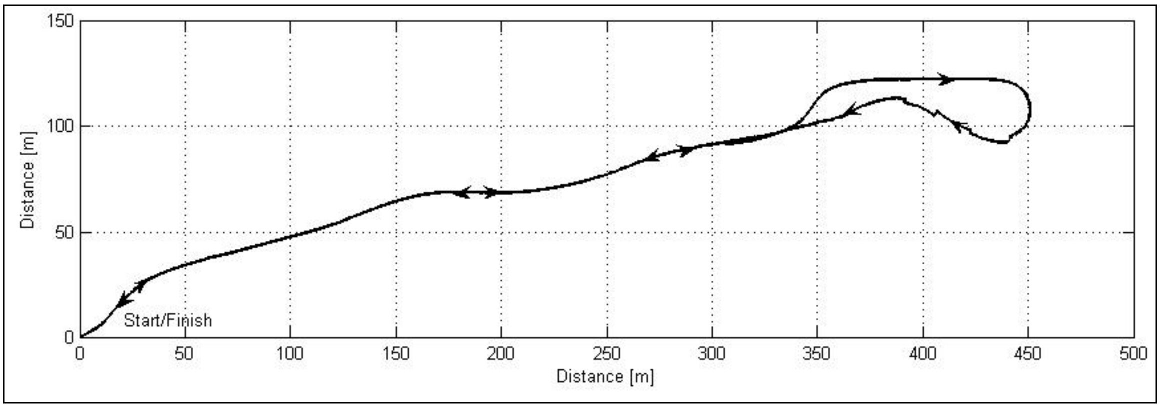


Figure 5.10 – Gerotek rough track top 800 m

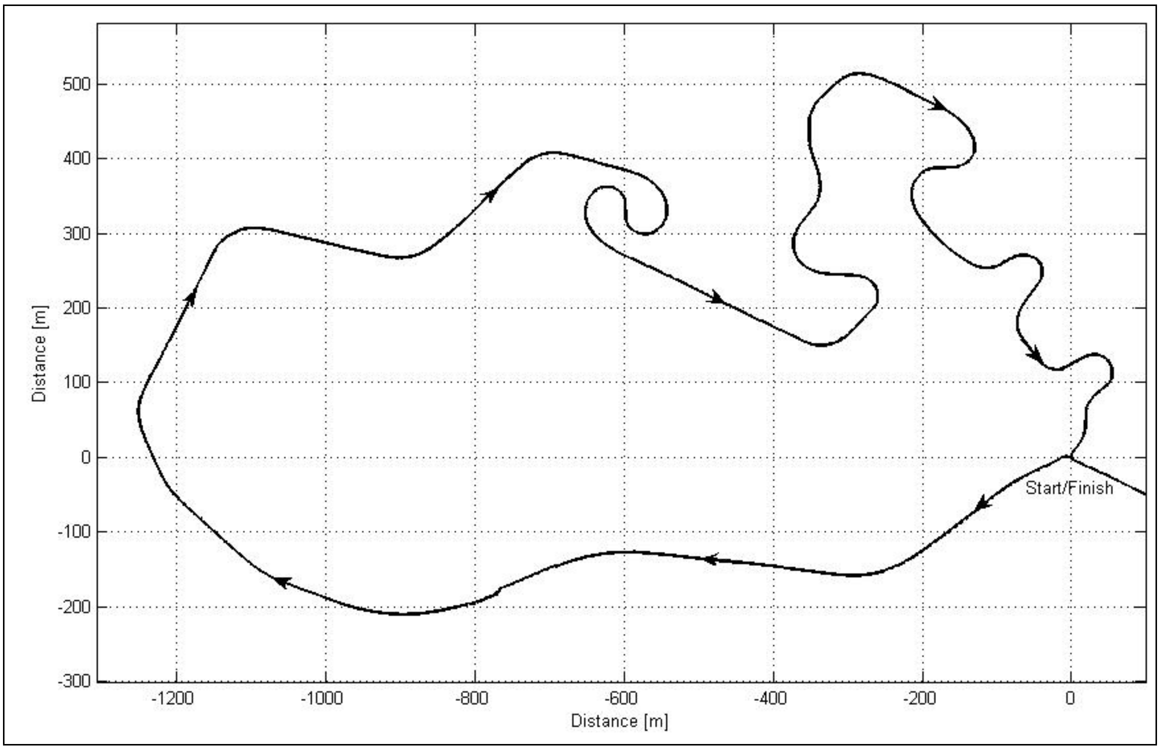


Figure 5.11 – Gerotek Ride and handling track

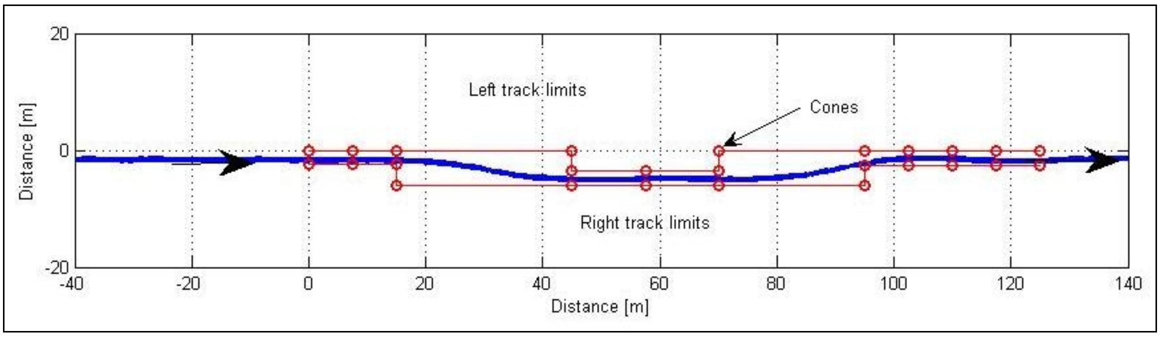


Figure 5.12 – Double lane change test

5.5.1 Frequency domain analysis

The first concept implemented is frequency domain analysis proposed by **Wallentowitz and Holdman (1997)** as well as **Truscott (1994)**. To determine whether this concept is feasible, FFT magnitudes were calculated for measurements over the six predefined tests. Each measurement was divided into bins of 1024 data points each with no overlapping. The FFT magnitudes were calculated for each bin of 1024 points, and then averaged for all the bins of the specific measurement at each frequency. Figure 5.13 indicates the average FFT magnitudes for the left rear and right rear vertical accelerations measured on the vehicle body. All six superimposed graphs indicate the same two peaks at 2 Hz (Body natural frequency) and 12 Hz (wheel hop frequency), although the magnitudes differ for the different terrains.

Figure 5.14 indicates that only very low frequencies can be detected by looking at the lateral acceleration. Trends are similar for all six tests. Roll, yaw and pitch velocities are indicated in Figure 5.15. The off-road track causes significant activity around 2 Hz that is absent in the other tests. Yaw velocity is restricted to low frequencies while the pitch natural frequency can be seen to be around 2 Hz for all six terrains. Relative suspension displacements (Figure 5.16) indicates the body natural frequency around 2 Hz. This is only really noticeable on the off-road test. The FFT magnitudes of the steering displacement and kingpin steering angle are indicated in Figure 5.17. All activity takes place at very low frequencies.

FFT magnitudes of relative suspension velocities are indicated in Figure 5.18. The relative velocity was obtained by differentiating the relative displacement in the time domain and then calculating the FFT. Again the frequency at 2 Hz is prominent with activity from about 1 Hz to 12 Hz. Trends do however look the same for all terrains.

The FFT magnitudes of the steering velocities, calculated by first differentiating the steering displacements in the time domain, are indicated in Figure 5.19. Figure 5.19(b) indicates the FFT magnitude of the steering velocity at the kingpin. Figure 5.19(a) indicates the steering velocity calculated from the measured displacement between the vehicle body and the steering link going to the wheels. The steering velocity clearly indicates activity around 8 to 10 Hz when driving off-road and through the mountain pass that is not present on the kingpin steering velocity. This is attributed to bump and roll steer as well as the kinematic effects resulting from the Panhard rod.

Although the frequency domain analysis provides valuable insight into the various excitation and natural frequencies, it is concluded that “ride *vs.* handling” cannot be detected from the frequency analysis. The same frequencies are excited regardless of terrain types, manoeuvres and speeds.

5.5.2 Lateral *vs.* vertical acceleration

The next strategy that was investigated is the one proposed by **Nell (1993)** and **Nell and Steyn (1998)**. They compare lateral and vertical acceleration as measured on the rigid front axle of a heavy off-road military vehicle. The semi-active dampers on their test vehicle is switched to “hard” when the lateral acceleration is higher than the vertical

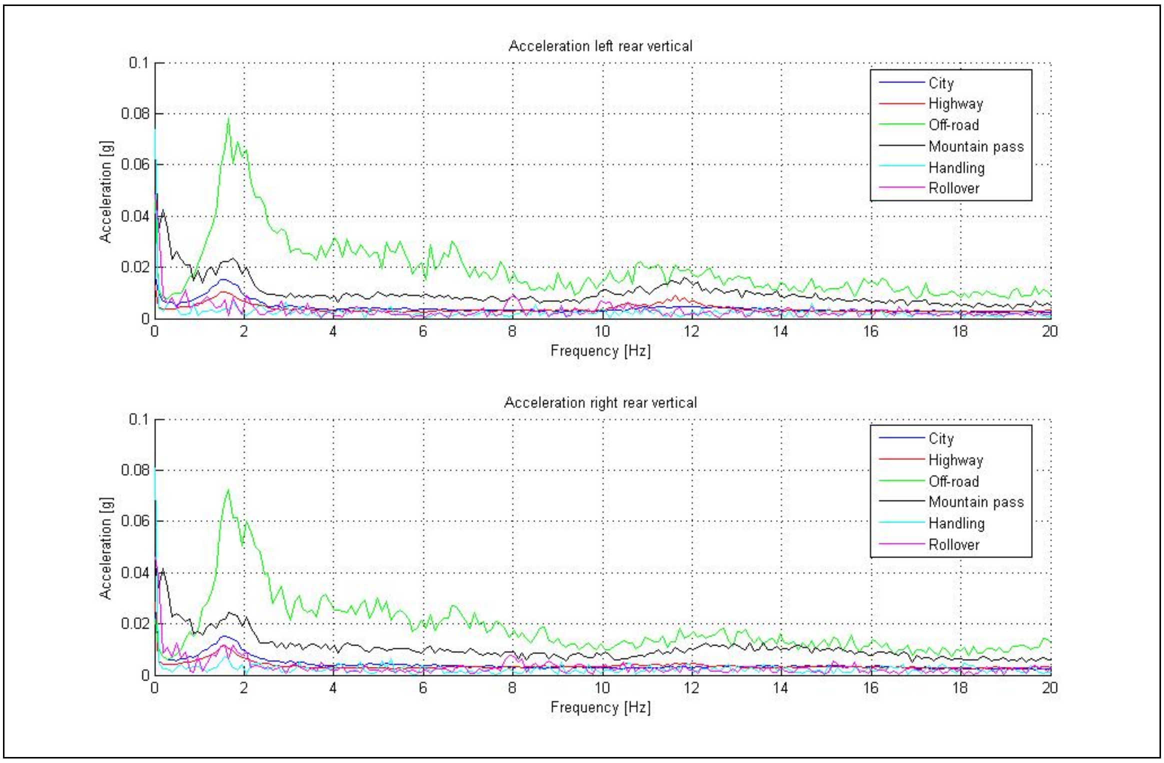


Figure 5.13 – FFT magnitude of vertical body acceleration (left and right rear)

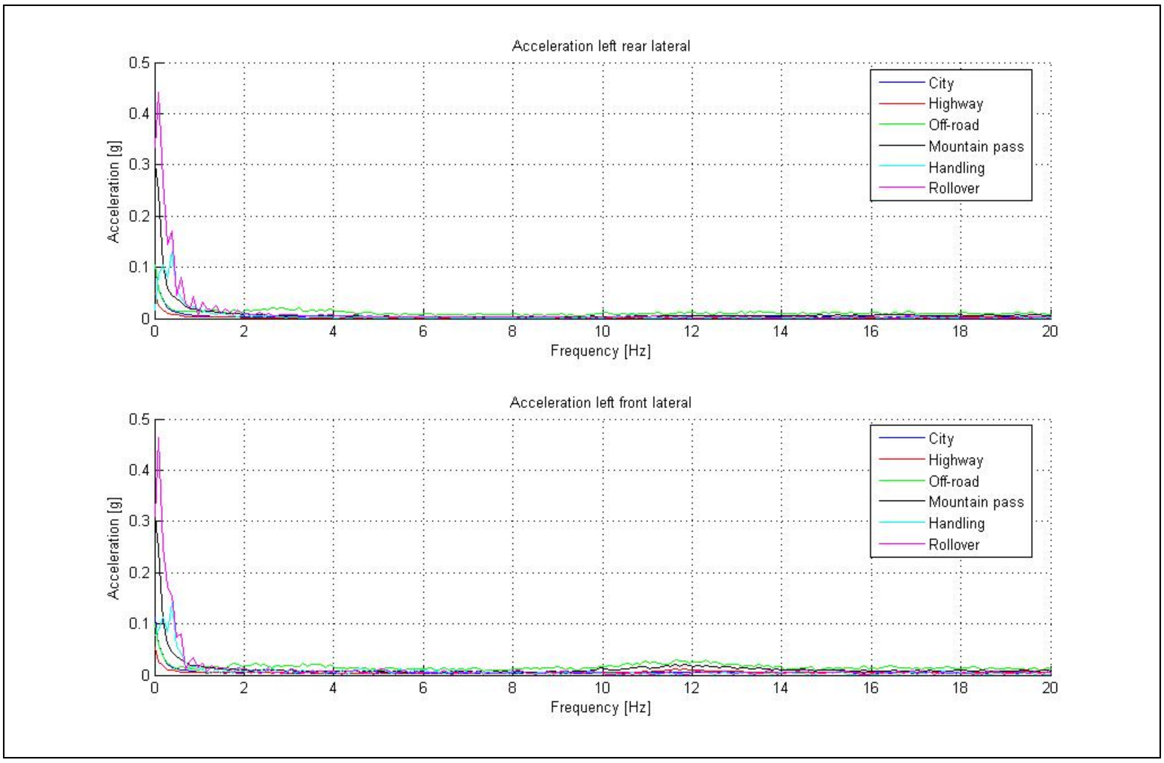


Figure 5.14 – FFT magnitude of body lateral acceleration (left front and left rear)

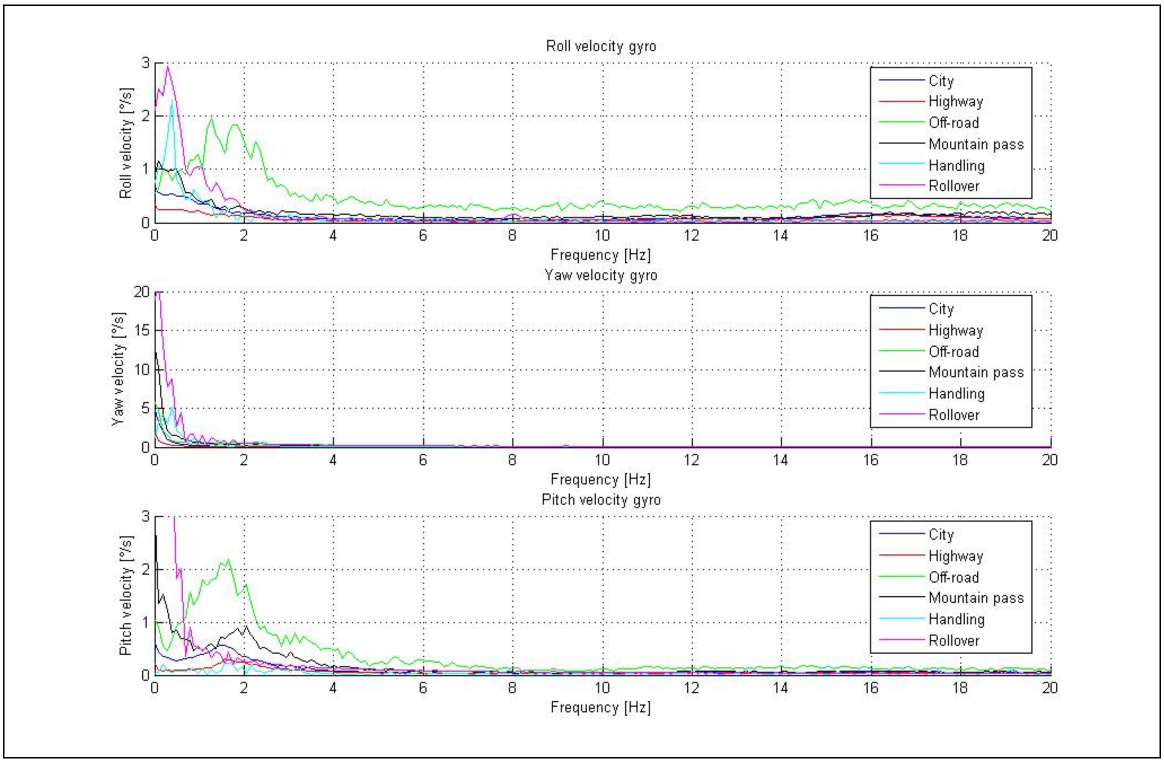


Figure 5.15 – FFT magnitudes of body roll, yaw and pitch velocity

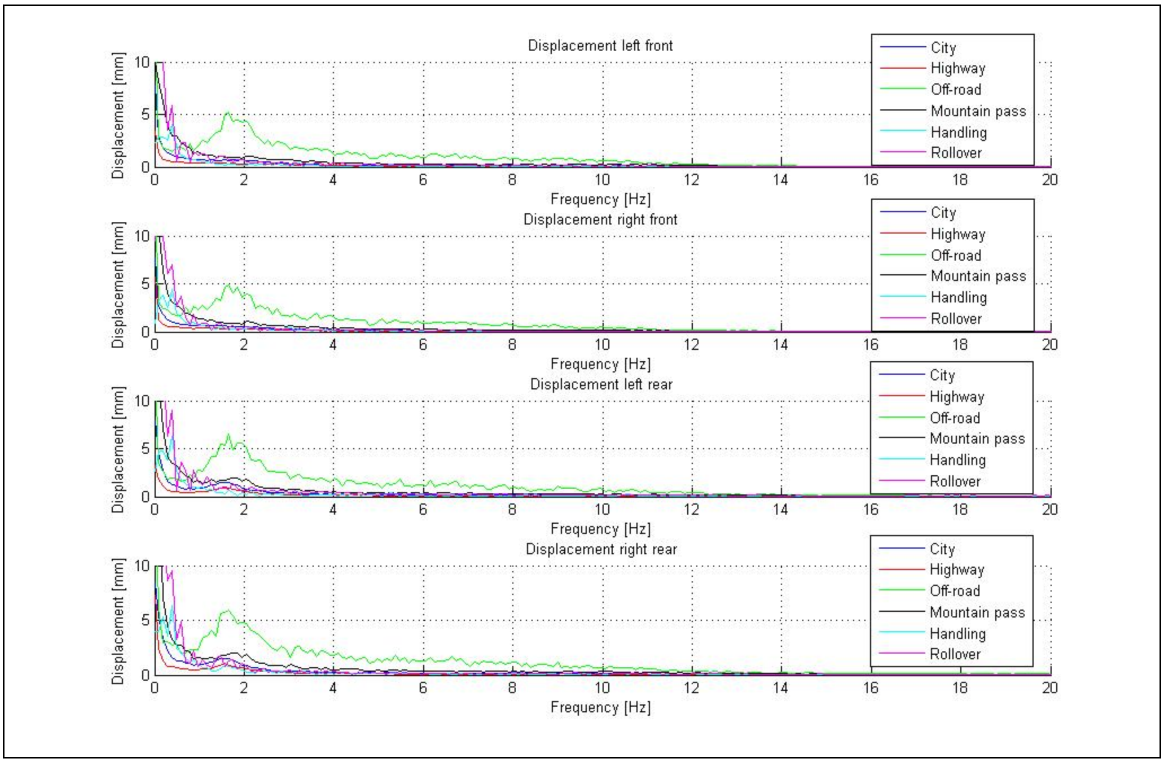


Figure 5.16 – FFT magnitude of relative suspension displacement (all four wheels)

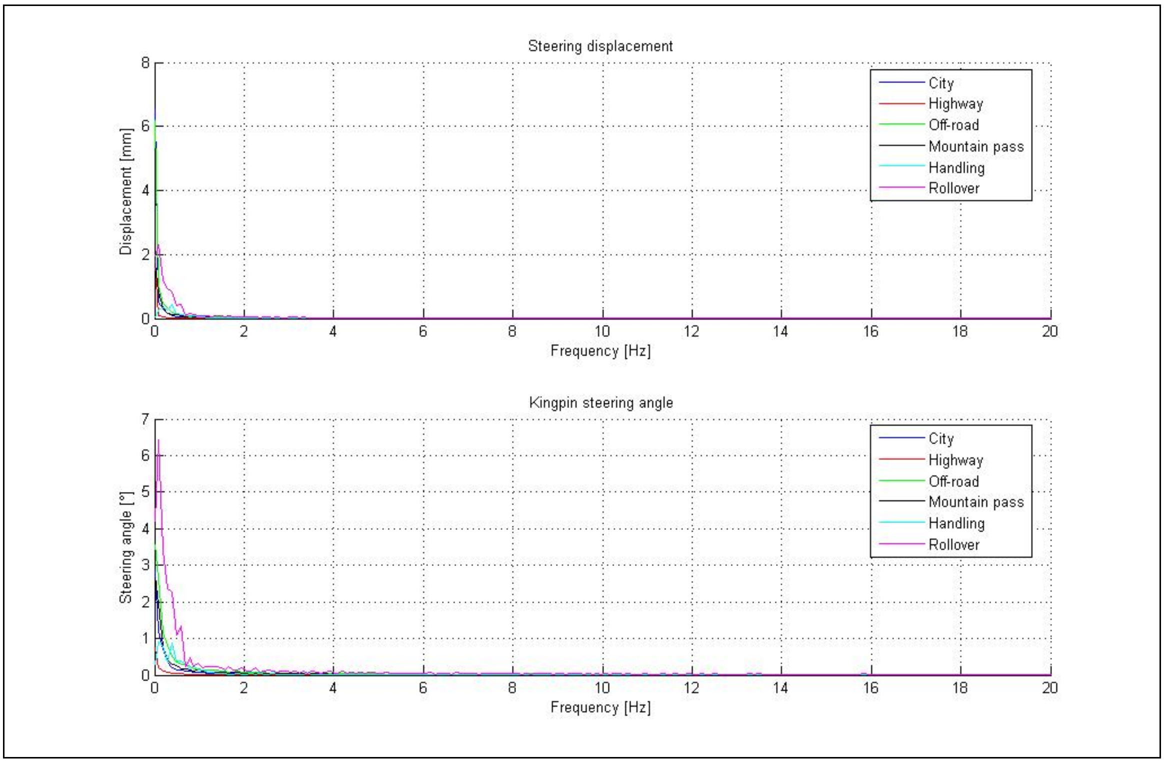


Figure 5.17 – FFT magnitude of steering displacement and kingpin steering angle

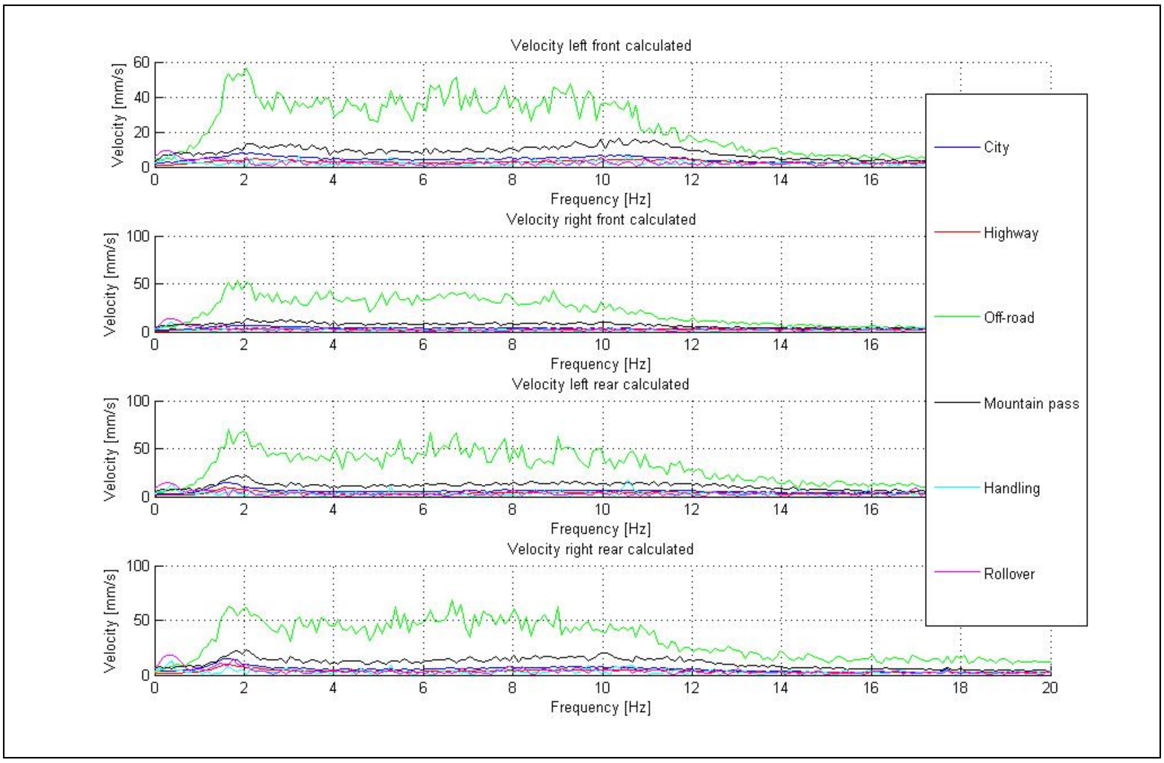


Figure 5.18 – FFT magnitude of relative suspension velocity (all four wheels)

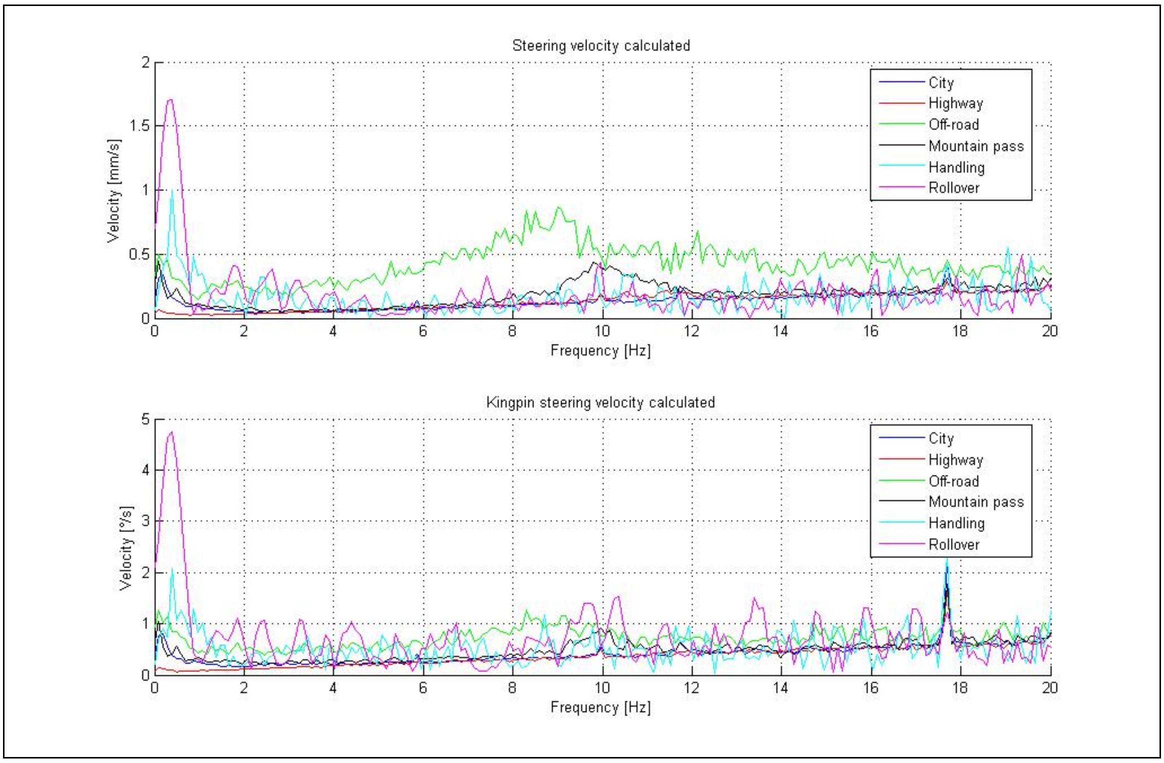


Figure 5.19 – FFT magnitude of steering velocity

acceleration and to the “ride comfort” mode when vertical acceleration is higher than lateral acceleration.

Figure 5.20 indicates the result of this analysis when applied to our measurements during city driving. Figure 5.20(a) indicates vertical and lateral accelerations measured on the left rear of the vehicle while Figure 5.20(b) indicates the suspension switching. A value of “0” in Figure 5.20(b) indicates “ride” mode while a value of “1” indicates “handling” mode. Switching seems spurious and random, *e.g.* between 530 and 600 seconds the vehicle is stationary, but the suspension switches all the time due to the background noise on the acceleration signals. The idling engine causes some of this noise.

Figure 5.21 indicates switching during the rollover test. It can be seen that the switching only works in one direction. The absolute values of the lateral and vertical acceleration should therefore be compared to enable correct switching. Two fundamental problems exist with the strategy as proposed by Nell namely:

- The absolute values of the accelerations should be compared
- Provision should be made for some type of dead band to prevent spurious switching when the measured values are close to zero

Nell intended the strategy to be used for accelerations on the front axle of the vehicle. The vertical acceleration on the axle is however significantly higher (can be 15 to 25g peak) than the lateral acceleration (about 1g peak). The strategy will therefore favour ride comfort, especially on rough roads. The strategy will also emphasize wheel hop frequency and not body motion due to the measuring position on the axle.

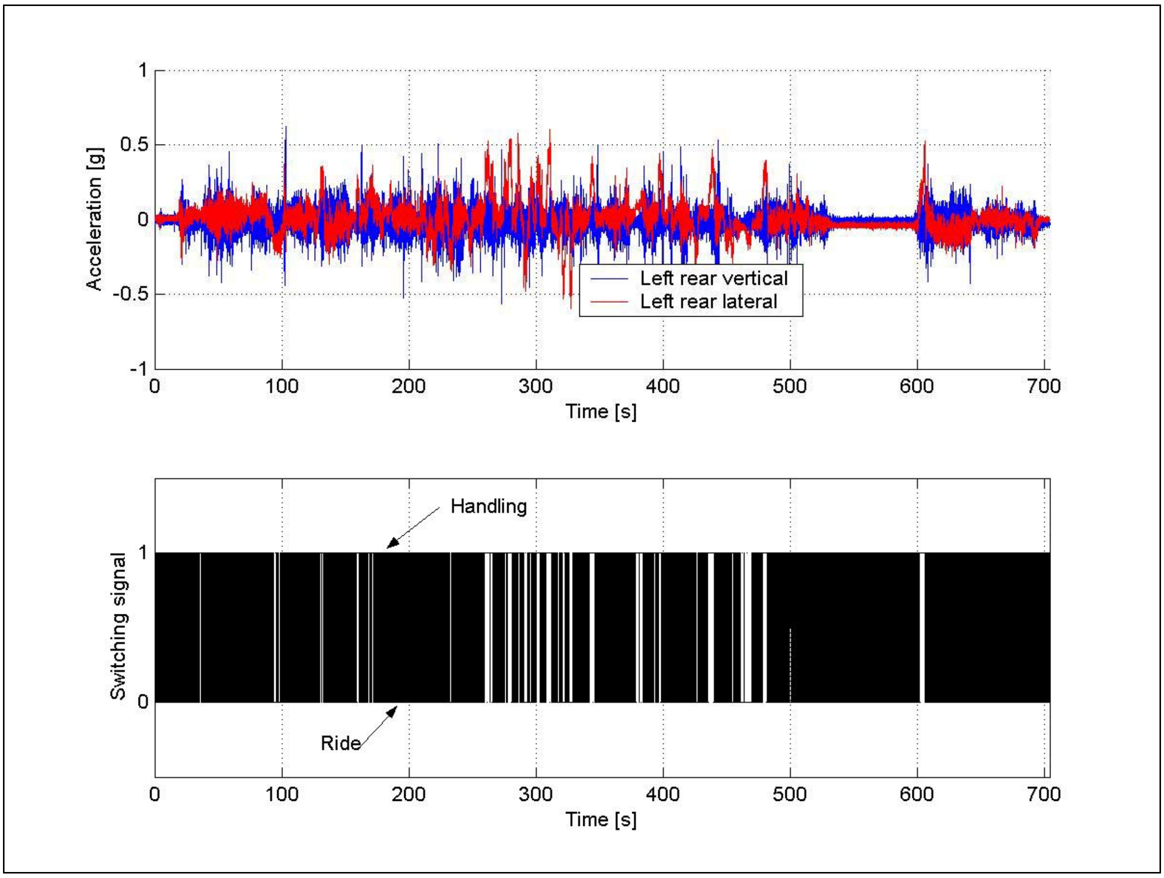


Figure 5.20 - Strategy proposed by **Nell (1993)** as applied to city driving

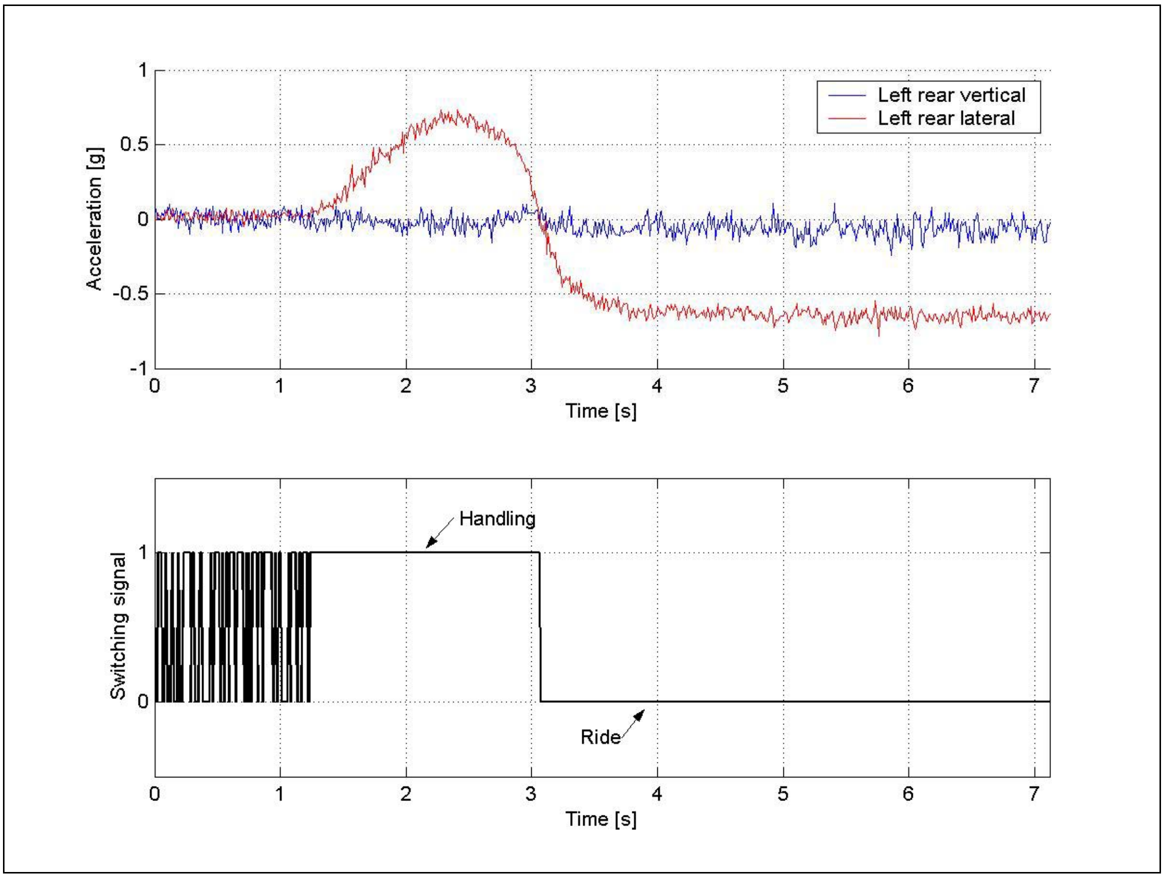


Figure 5.21 - Strategy proposed by **Nell (1993)** as applied to the rollover test

5.5.3 Lateral *vs.* vertical acceleration - modified

The strategy proposed in paragraph 5.5.2 is now modified in order to eliminate its drawbacks. The absolute values of the lateral and vertical accelerations on the vehicle body are used. A dead band is introduced to prevent spurious switching due to accelerometer noise and drift. The “ride” mode is always selected if the absolute lateral acceleration is less than 0.1 g. An upper limit is also included that forces switching to the “handling” mode when lateral acceleration exceeds 0.3g (see **Stone and Cebon, 2002** and **Darling and Hickson, 1998**). This however results in negligible improvement during highway driving (Figure 5.22) although the ride mode is at least selected during periods when the vehicle is stationary (*e.g.* between 650 and 700 seconds). A significant improvement is however noticed for the rollover test (Figure 5.23) where the handling mode is selected during most of the manoeuvre. The switching to the ride mode at 3.2 seconds is however problematic as this happens at a critical point in the test. The method is however an improvement on the previous case.

5.5.4 Steering angle vs. speed

The use of a speed dependant steering angle threshold has been applied frequently. Some examples are discussed by **Hirose *et. al.* (1988)**, **Hine and Pearce (1988)**, **Nastasíc and Jahn (2005)** as well as **Broge (1999)**. The steering angle *vs.* speed threshold used by Citroën was indicated in Figure 5.3.

The envelope for the Land Rover was determined by plotting steering angle against vehicle speed for all the tests. The results are indicated in Figure 5.24. The circles in Figure 5.24 indicate the measured data points obtained for city driving and the solid lines indicate the limiting values determined for different terrains. These curves represent the values of steering angle that is achieved during normal driving.

The strategy itself is very easy to implement and is an “input driven” strategy *i.e.* it will react on driver input and not vehicle reaction to driver input as is the case with lateral acceleration *etc.* It should therefore also give an early warning before the vehicle reaction can be detected.

The results of this strategy as implemented on test data is shown in Figures 5.25 to 5.30. The threshold value used for all the analyses indicated in Figures 5.25 to 5.30 is 50% of the steering angle limit for city driving (solid red line in Figure 5.24) at any given vehicle speed. During city driving (Figure 5.25), off-road driving (Figure 5.27) and on the mountain pass (Figure 5.28) the strategy works well due to the large steering angles involved. During the highway tests (Figure 5.26) switching seems to occur too often due to the fact that the steering threshold is very low and measurement noise and sensor drift has a significant effect on the results.

For the handling test (Figure 5.29) and rollover test (Figure 5.30) results are not entirely satisfactory, as the suspension will be switched to the “ride” mode whenever the steering goes through the zero position. This is dangerous as the effect takes place during critical parts of the manoeuvre.

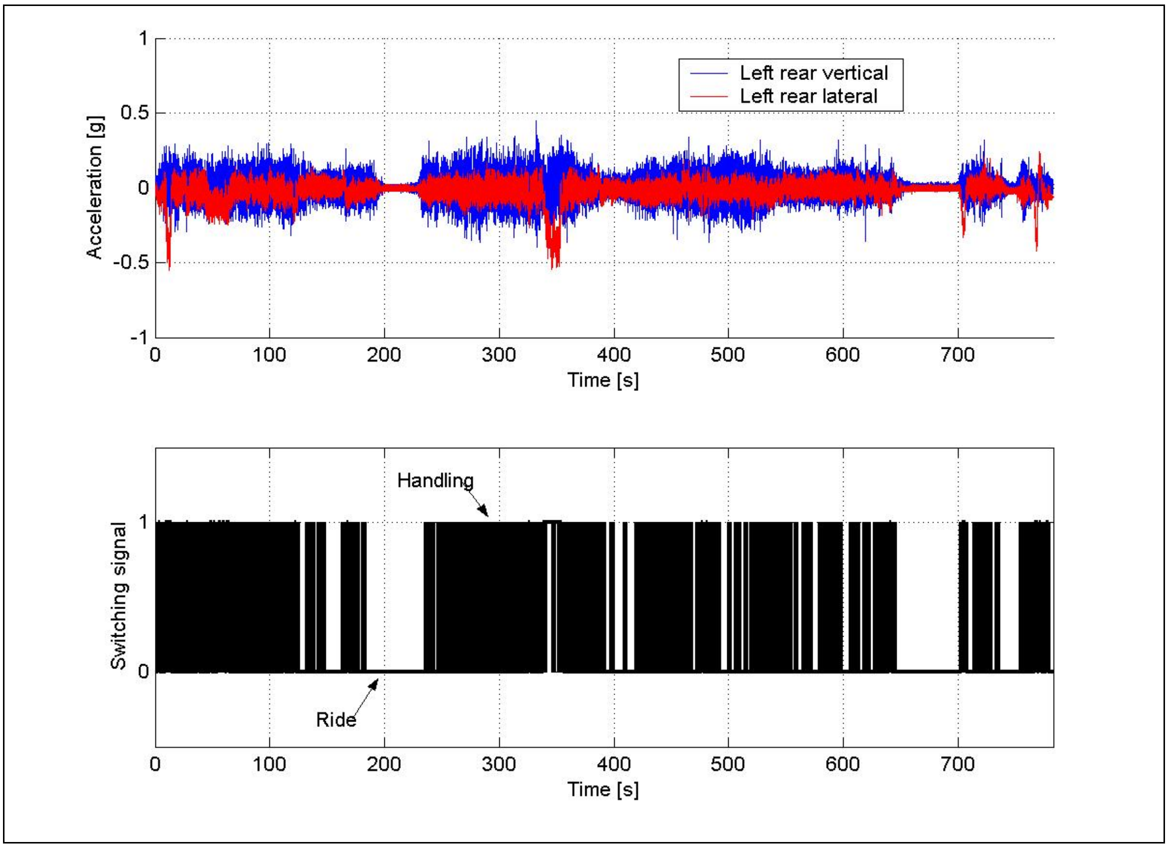


Figure 5.22 – Modified lateral *vs.* longitudinal acceleration for highway driving

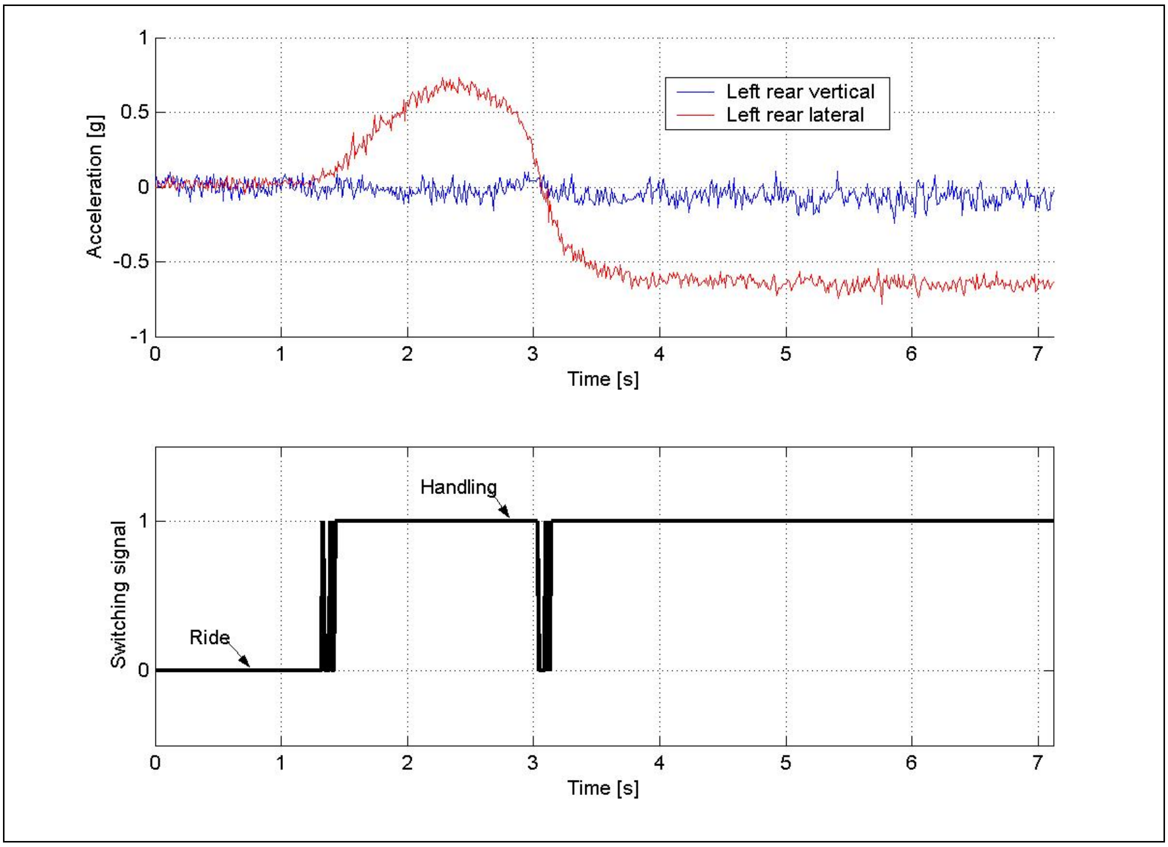


Figure 5.23 - Modified lateral *vs.* longitudinal acceleration for rollover test

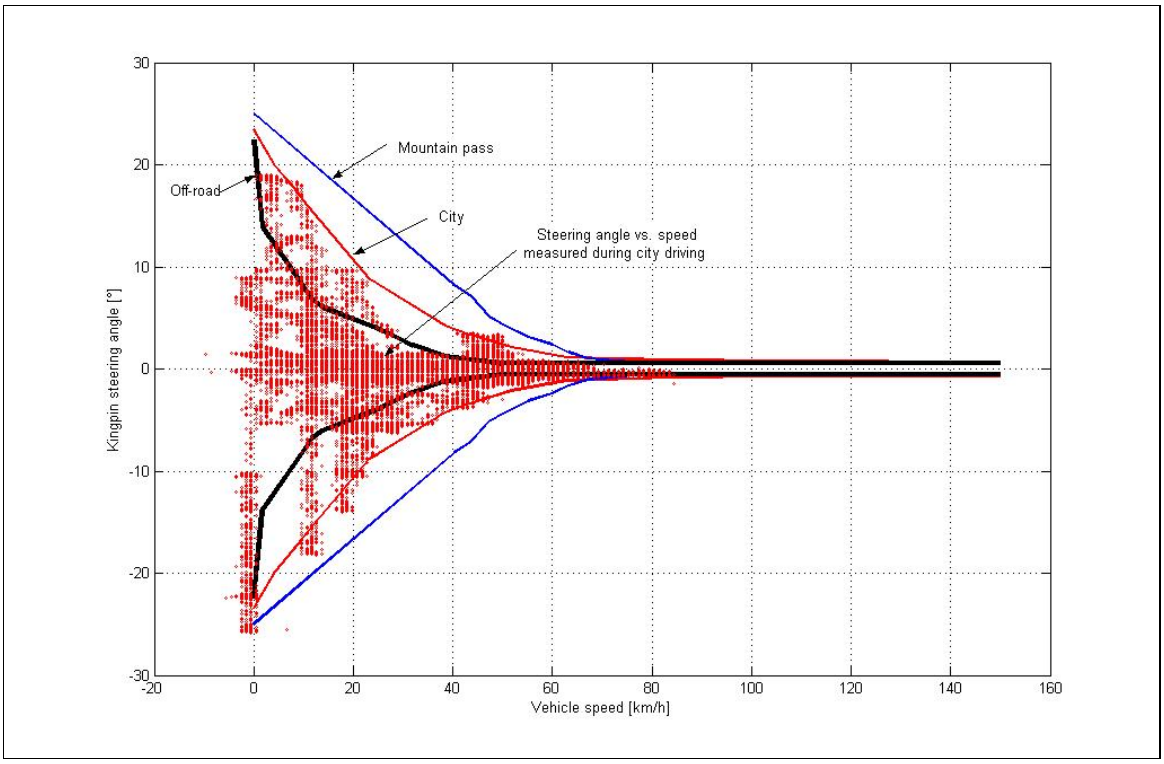


Figure 5.24 – Steering limits *vs.* vehicle speed measured during three tests

Spurious switching also sometimes occurs due to noise. This can be seen for example in the first 20 seconds of Figure 5.25 where the vehicle is stationary, but the steering wheel is turned. This problem can however be solved by using a dead band instead of a single limit.

The biggest difficulty when applying this strategy to the off-road vehicle is that the threshold values differ considerably depending on the terrain. If the terrain can be somehow "identified", and the threshold values adapted accordingly, then the performance of the strategy can be improved. Performance for the handling and rollover tests are however only expected to improve marginally because the system will still switch to "ride mode" when the steering angle goes through the zero position.

5.5.5 Disadvantages of proposed concepts

All the concepts investigated up to this point suffer from the same disadvantages namely:

i) Switching occurs too frequently
ii) All strategies don't work properly for all conditions
iii) Strategies that work well for on-road driving fail during off-road tests and *vice versa*

Unnecessary switching could be eliminated by applying a dead band, low pass filtering or delayed switching. It is imperative that absolute values be used.

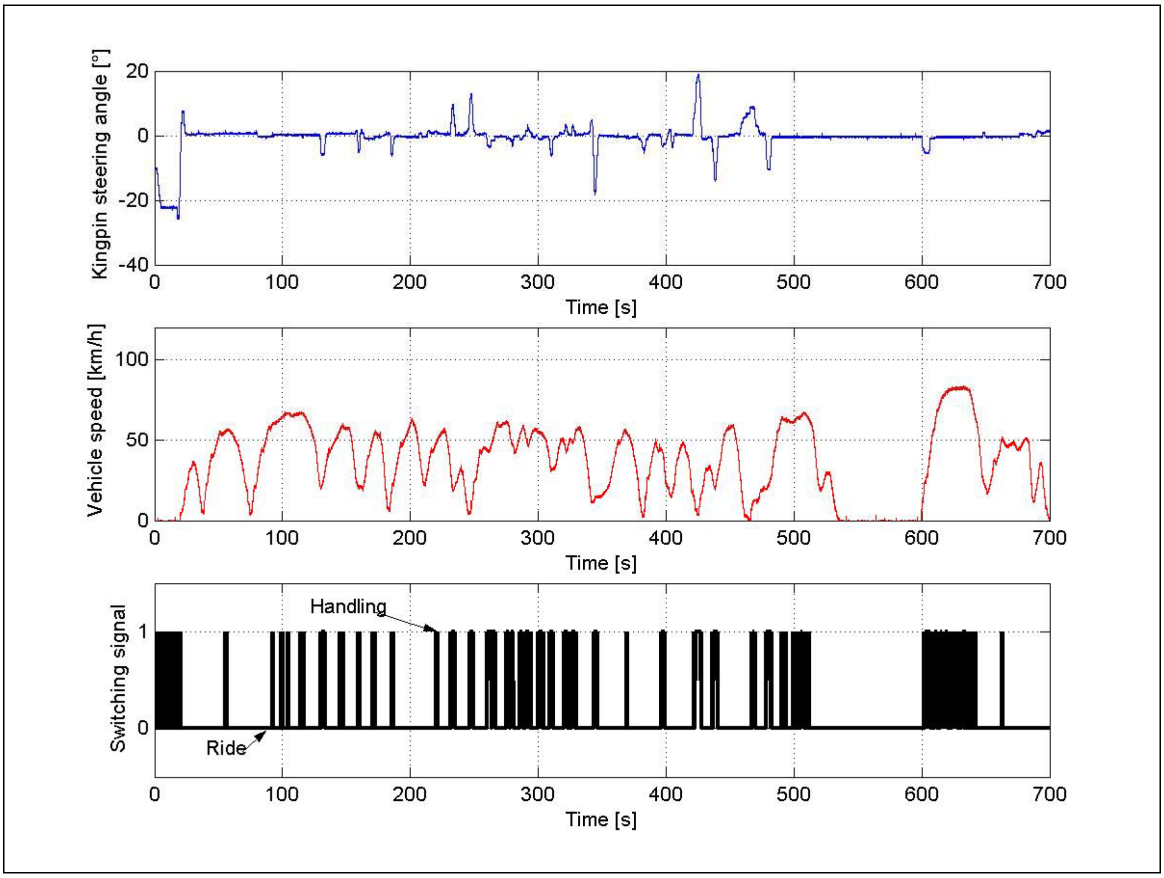


Figure 5.25 – Steer angle *vs.* speed implemented for city driving

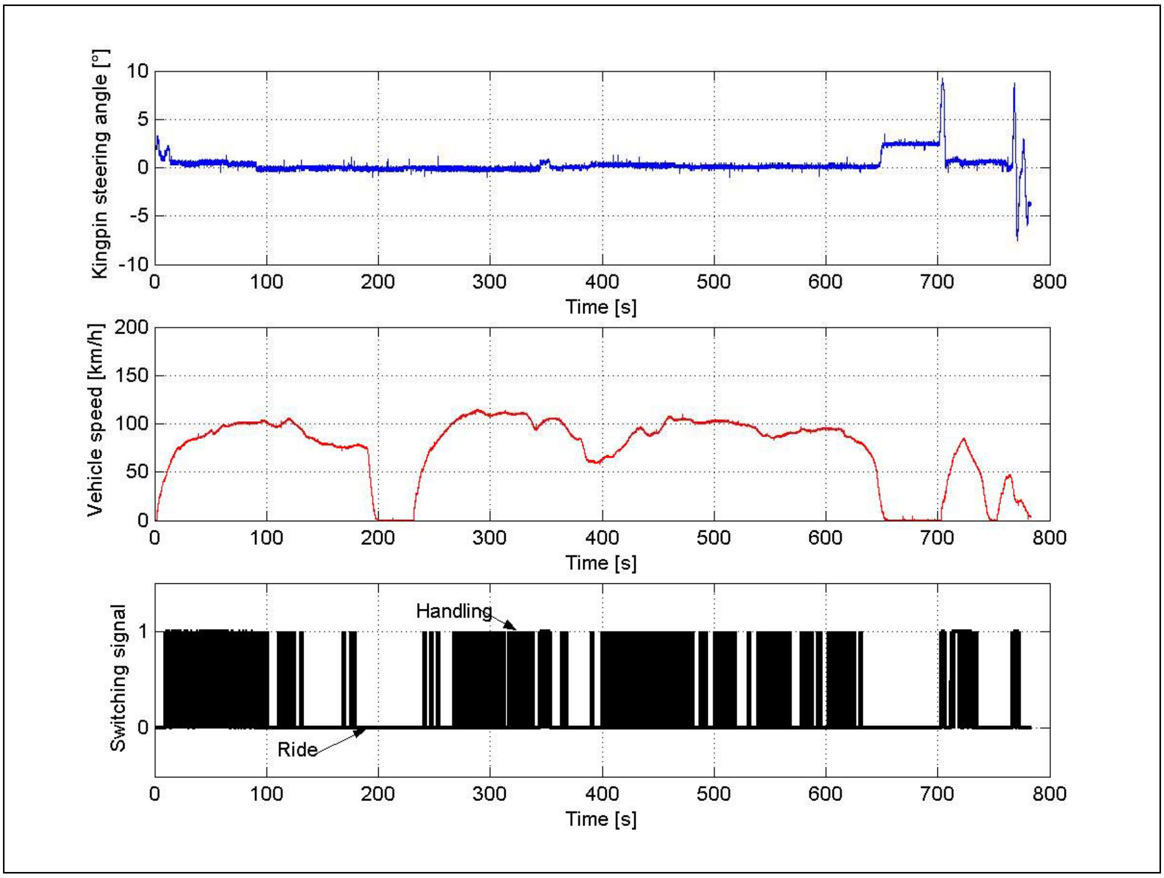


Figure 5.26 – Steer angle *vs.* speed implemented for highway driving

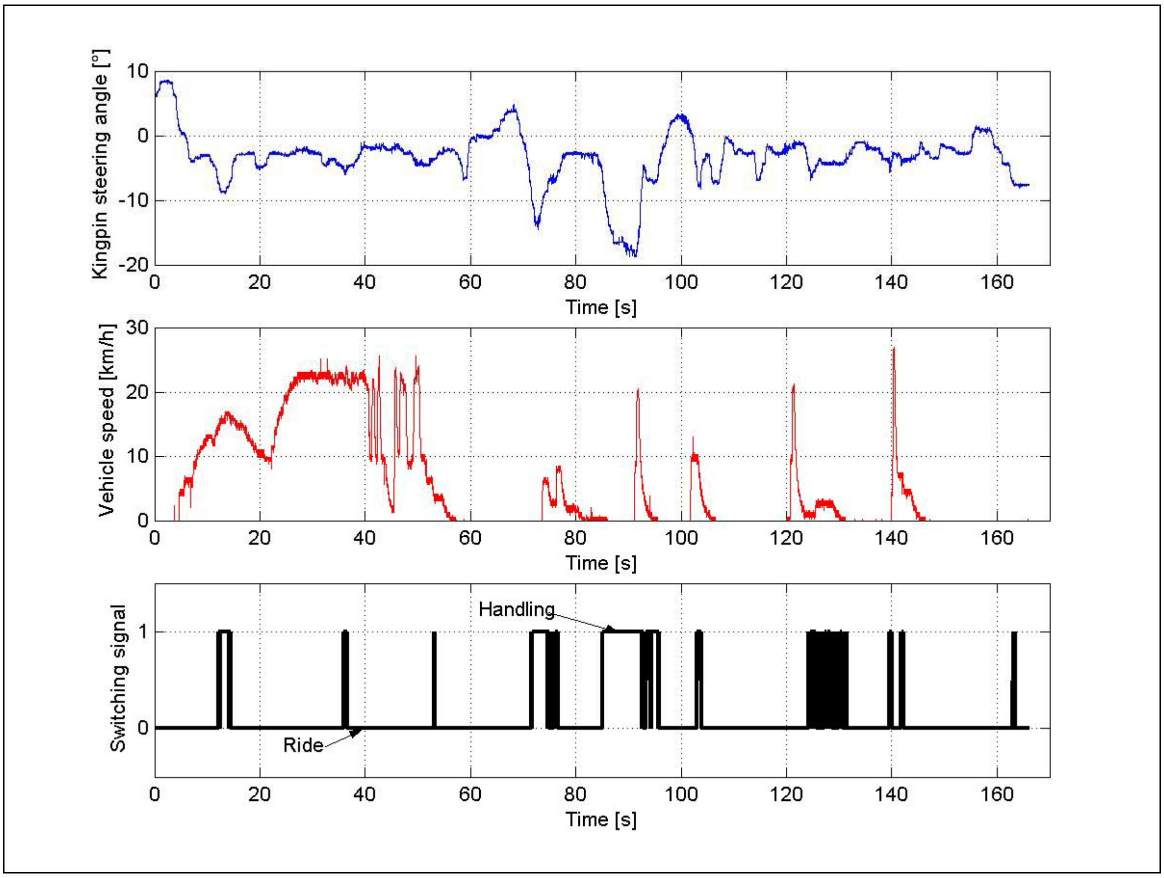


Figure 5.27 – Steer angle *vs.* speed implemented for off-road driving

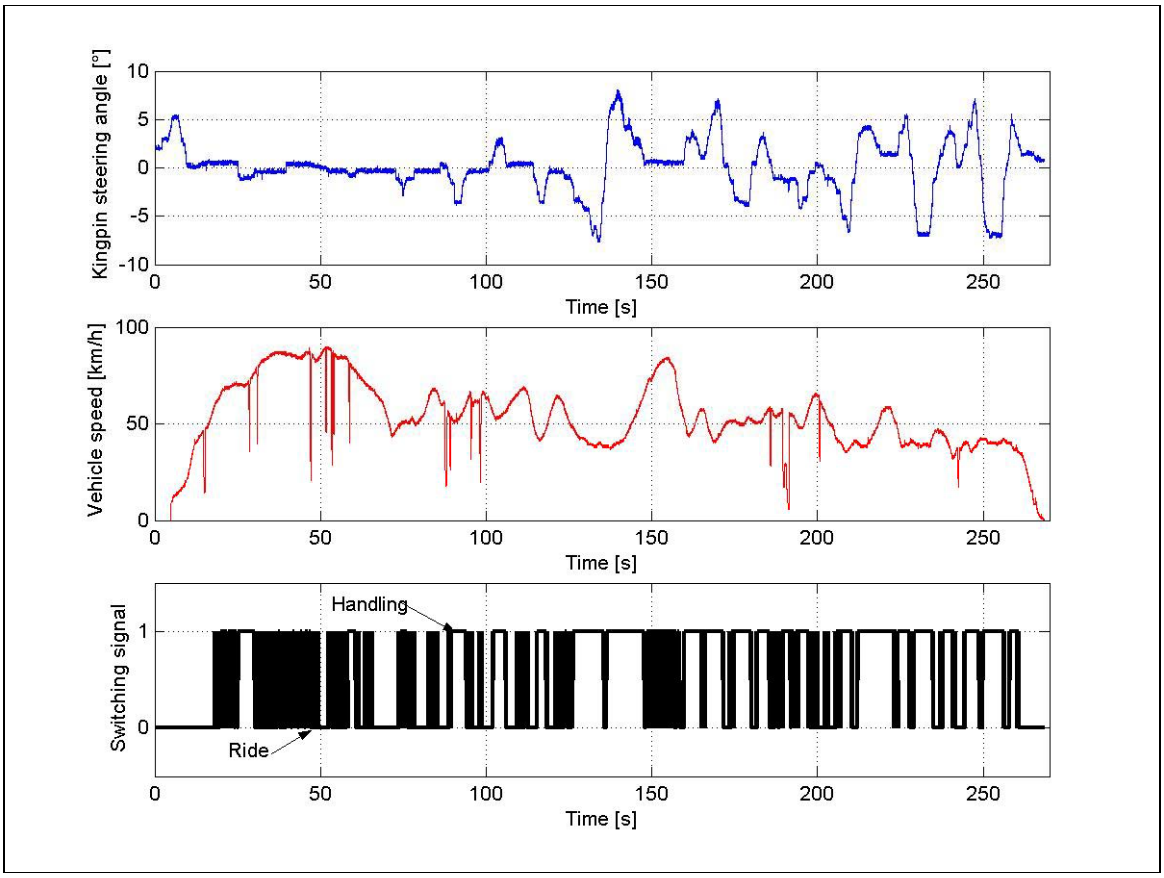


Figure 5.28 – Steer angle *vs.* speed implemented for mountain pass driving

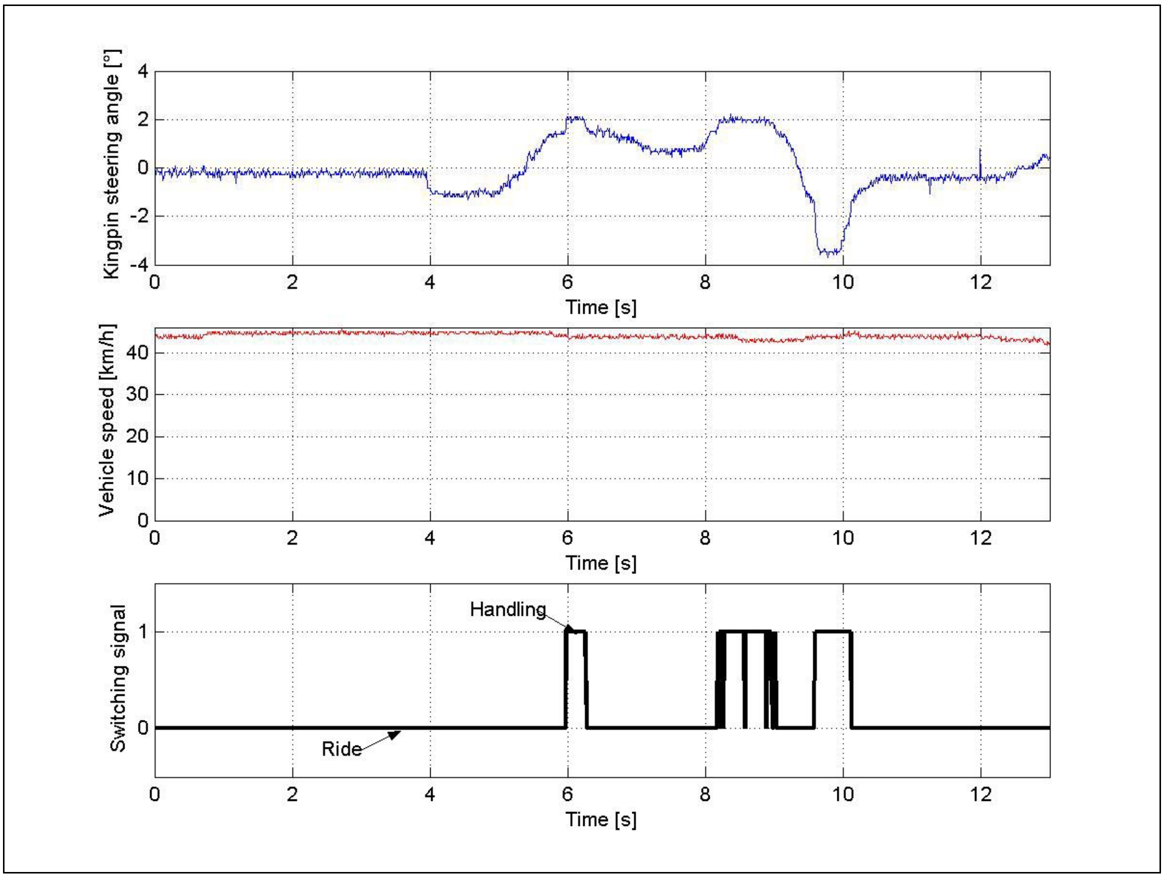


Figure 5.29 – Steer angle *vs.* speed implemented for handling test

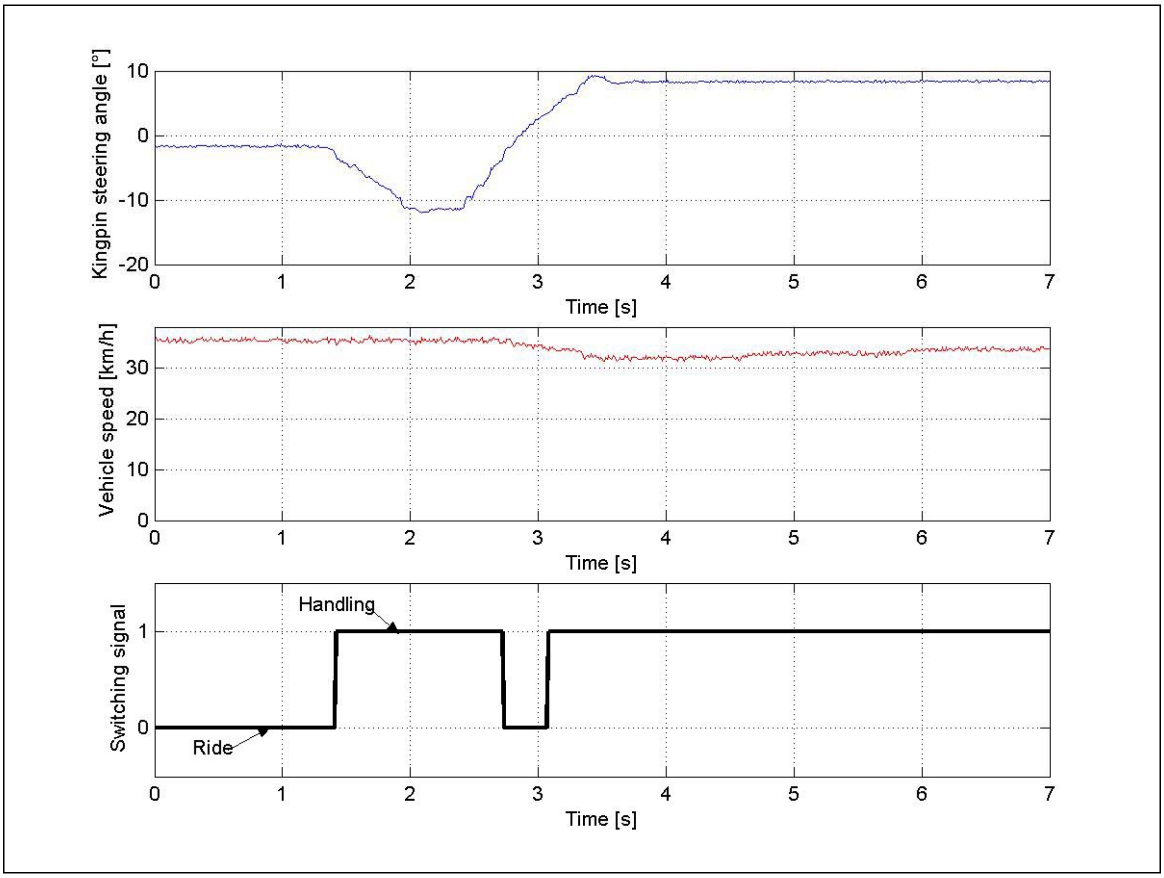


Figure 5.30 – Steer angle *vs.* speed implemented for rollover

5.6 Novel strategies proposed

To overcome the problems mentioned in paragraph 5.5.5, two additional strategies are proposed namely relative roll angle and running RMS (RRMS).

These proposed strategies will be discussed in paragraphs 5.6.1 and 5.6.2.

5.6.1 "Relative roll angle" calculated from suspension deflection

Roll angle was identified as a good measure of handling in paragraph 2.1.2.5. Absolute body roll angle is however very difficult to measure directly. The first proposal is to use the relative body roll angle between the vehicle body and axle, calculated using the relative suspension deflection of the left and right suspension systems. Because the body roll angle is small (< 5°), the roll angle is proportional to the difference between left and right relative displacements, divided by the distance between the left hand and right hand displacement measuring points. The difference between left and right displacements is therefore directly compared to a threshold value, without calculating the actual body roll angle. If this difference exceeds the threshold, the suspension system is switched to the handling mode. Results of this concept, applied with a threshold value of 20 mm, are indicated in Figures 5.31 to 5.36. The left front and right front relative suspension displacements were used in these calculations, but the same concept could be applied to the displacements measured for the rear axle.

The strategy works well for city driving (Figure 5.31), highway driving (Figure 5.32) and mountain pass driving (Figure 5.34). It switches to "handling" mode too frequently during off-road driving (Figure 5.33). During the handling test (Figure 5.35), the "ride" mode is selected for most of the manoeuvre. "Handling" mode is selected only at the most critical part of the test where the vehicle returns to the initial lane (between 70 and 100 meters in Figure 5.12, corresponding to between 8 and 11 seconds in Figure 5.35). This switch to "handling" mode at this critical point of the test might have disastrous effects. Although behaviour during the handling test can be improved by reducing the switching threshold of 20 mm, "ride" mode will still be selected whenever the relative roll angle crosses through the zero position. A reduction of the threshold will also result in more unwanted switching during off-road driving. During the fishhook rollover test (Figure 5.36), dangerous switching to the "ride comfort" setting occurs where the relative roll angle crosses the zero position. This is however also the place where the roll velocity (and therefore kinetic energy due to body roll) is maximum. Switching to "ride" mode under these conditions is highly undesirable.

5.6.2 Running RMS vertical acceleration *vs.* lateral acceleration

The second proposal is to use the running RMS (RRMS) value of lateral acceleration compared to the running RMS of vertical acceleration. This concept will result in an average absolute value of the required parameters and should therefore reduce spurious switching and noise.

The running RMS (RRMS) is calculated determining the RMS value of the last N number of points. The strategy includes hysteresis and will always select the "ride comfort" mode if the RRMS lateral acceleration is less than 0.05g. It also always selects "handling" mode when the RRMS lateral acceleration is greater than 0.3g. Between these two limits, handling mode is selected only when the RRMS lateral acceleration exceeds the RRMS vertical acceleration. A running RMS of 1 second (or 100 previous data points) has been used for this analysis and seems to successfully remove noise without affecting response time detrimentally.

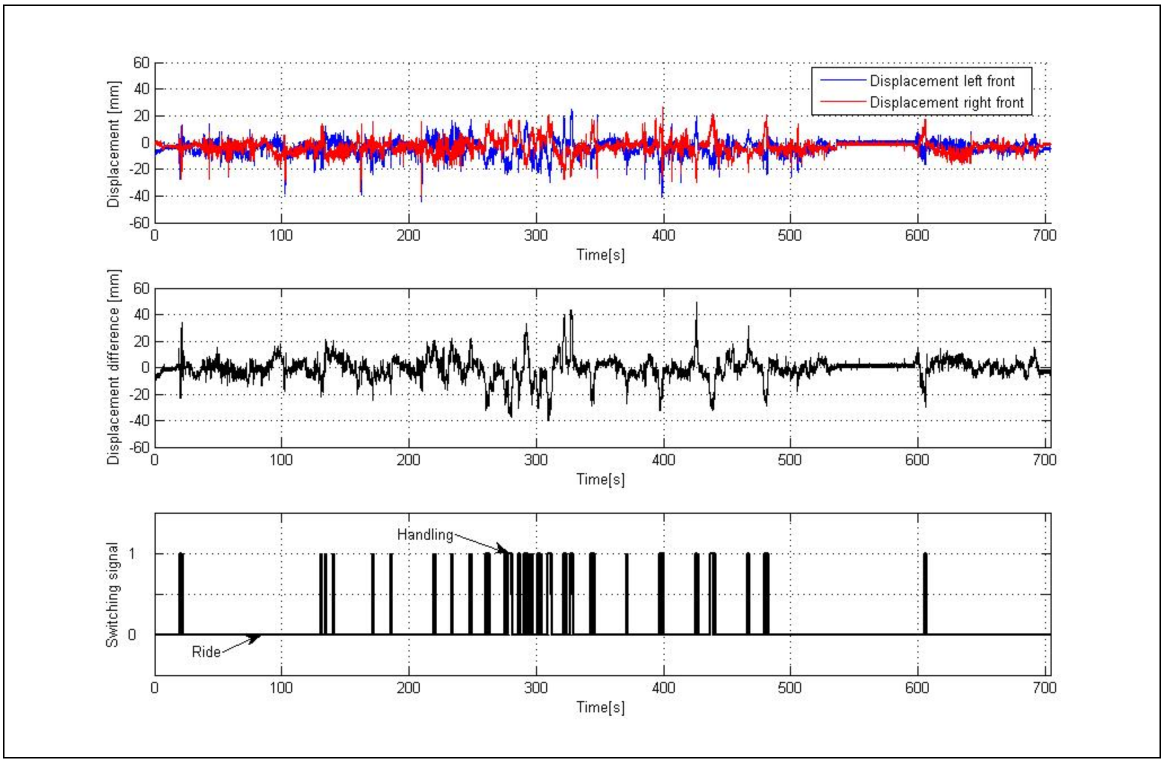


Figure 5.31 – Relative roll angle strategy for city driving

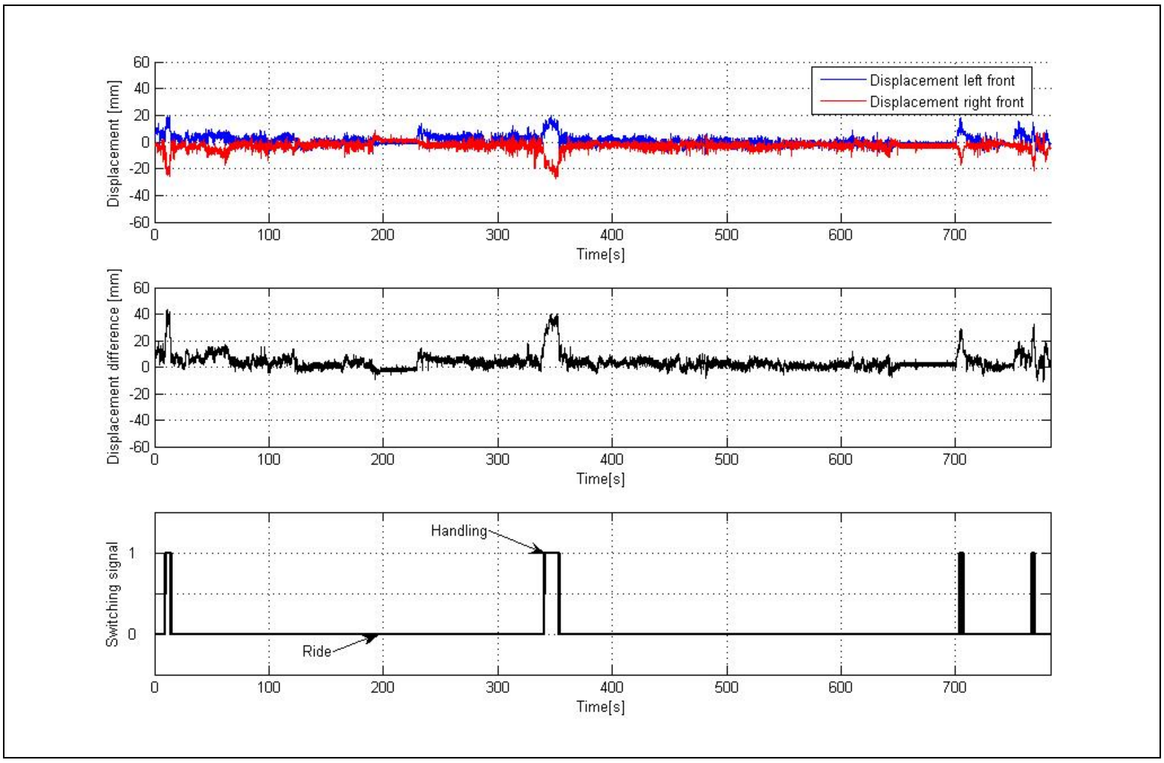


Figure 5.32 – Relative roll angle strategy for highway driving

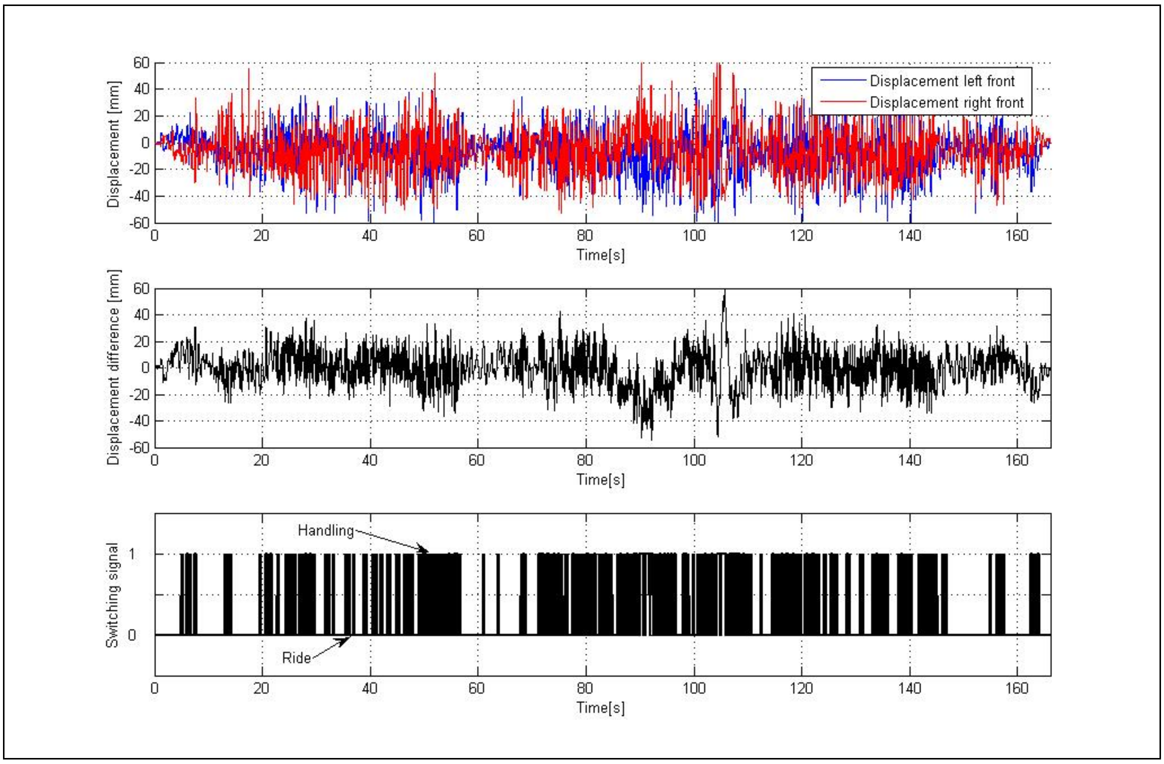


Figure 5.33 – Relative roll angle strategy for off-road driving

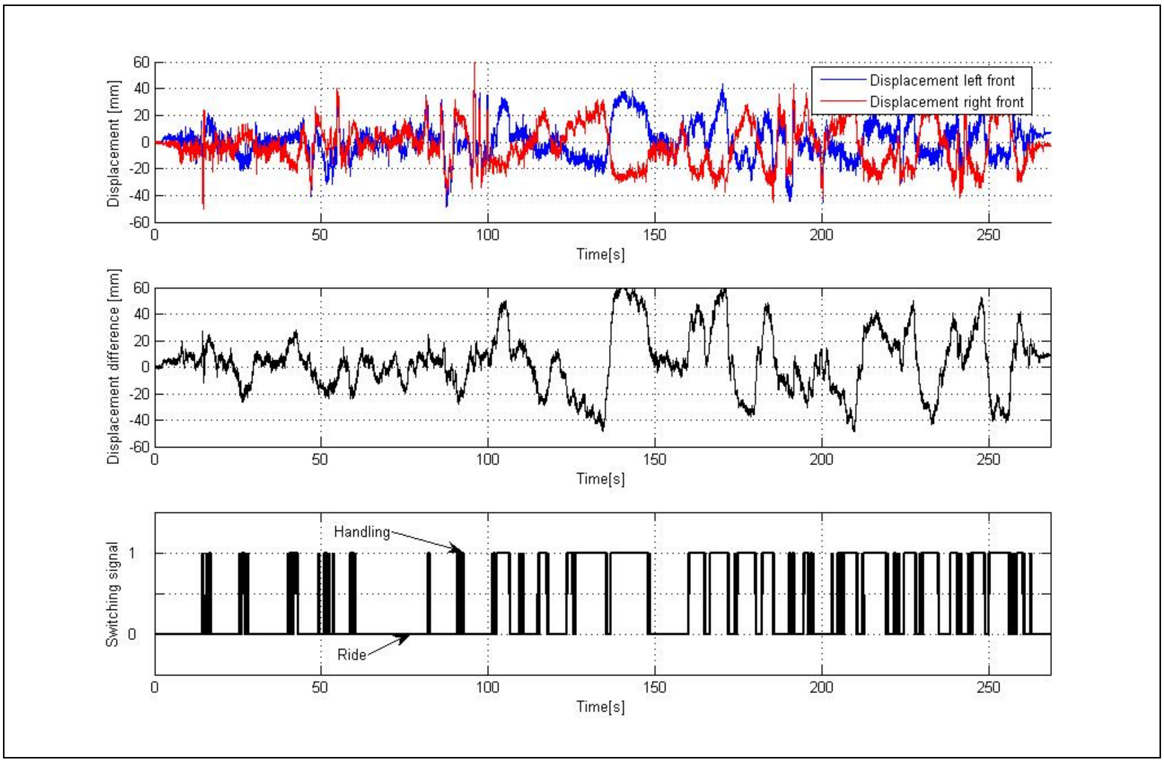


Figure 5.34 – Relative roll angle strategy for mountain pass driving

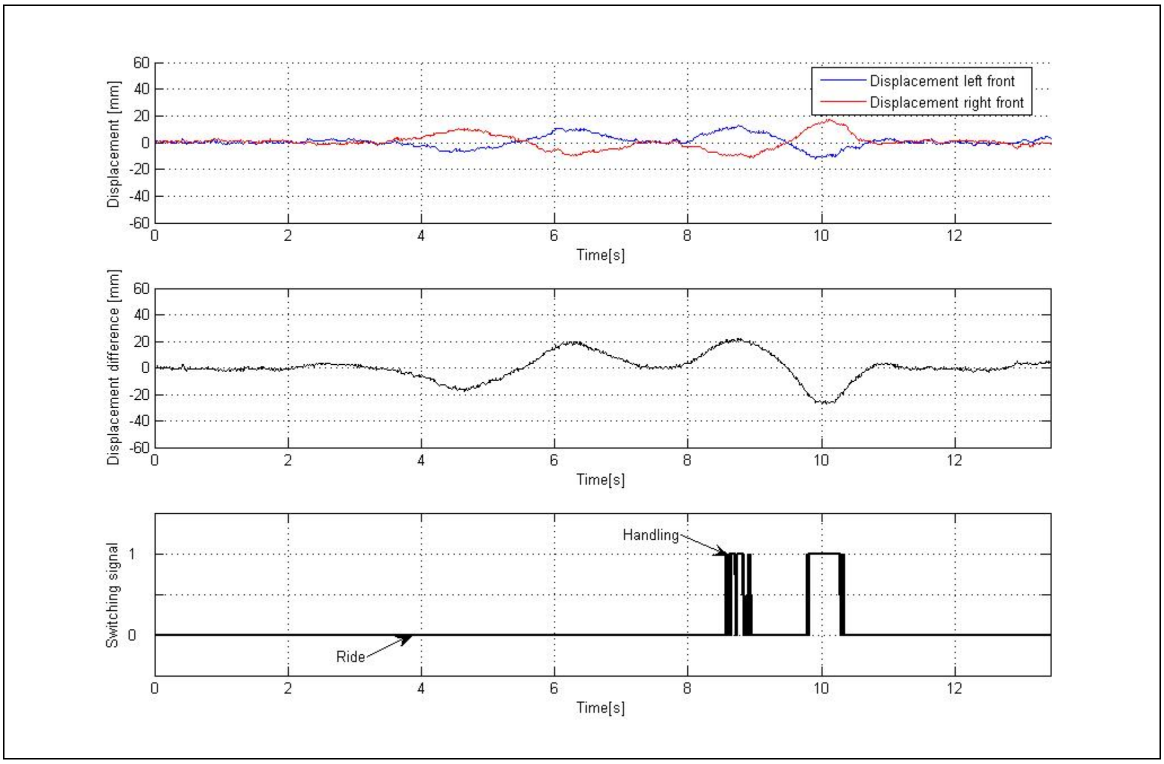


Figure 5.35 – Relative roll angle strategy for handling

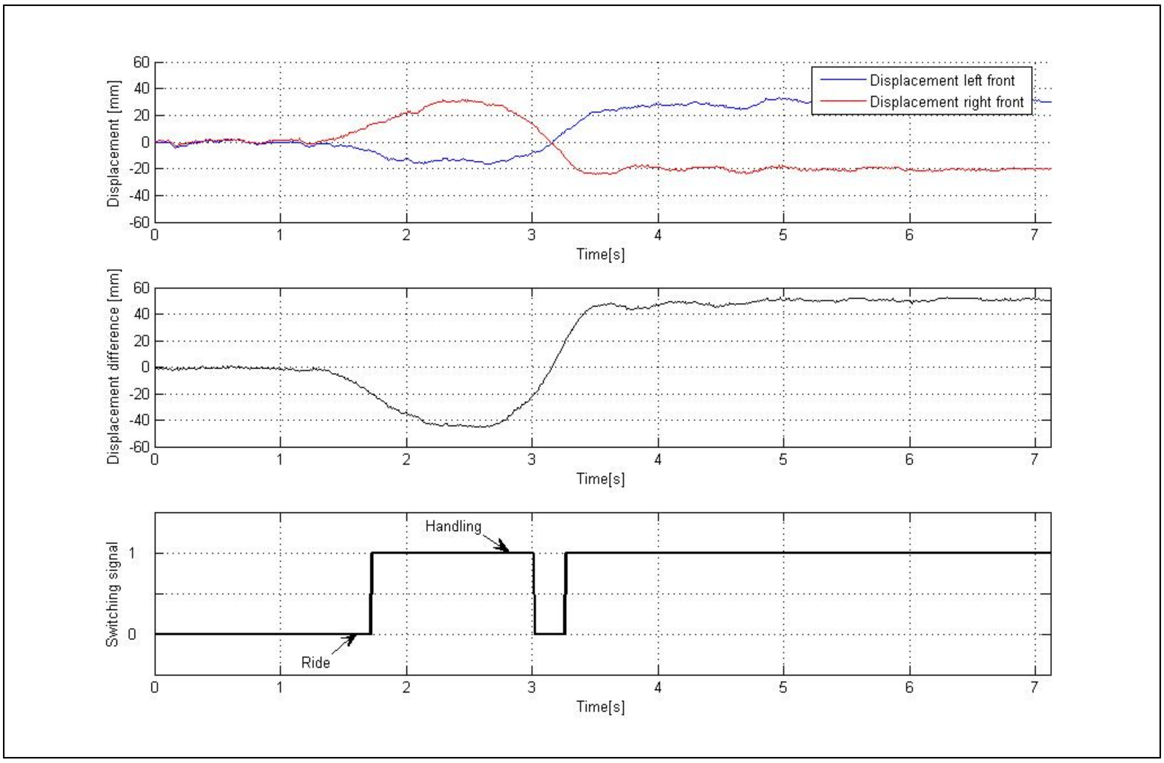


Figure 5.36 – Relative roll angle strategy for rollover

RRMS strategy results are indicated in Figures 5.37 to 5.42. This strategy works well for all conditions except for the double lane change where the "ride comfort" mode is selected about halfway through the test (see Figure 5.41). This is however the point where the vehicle is in the second lane before it starts turning back into the first lane. This should not result in serious problems, as long as the switching back to "handling" mode happens quickly enough.

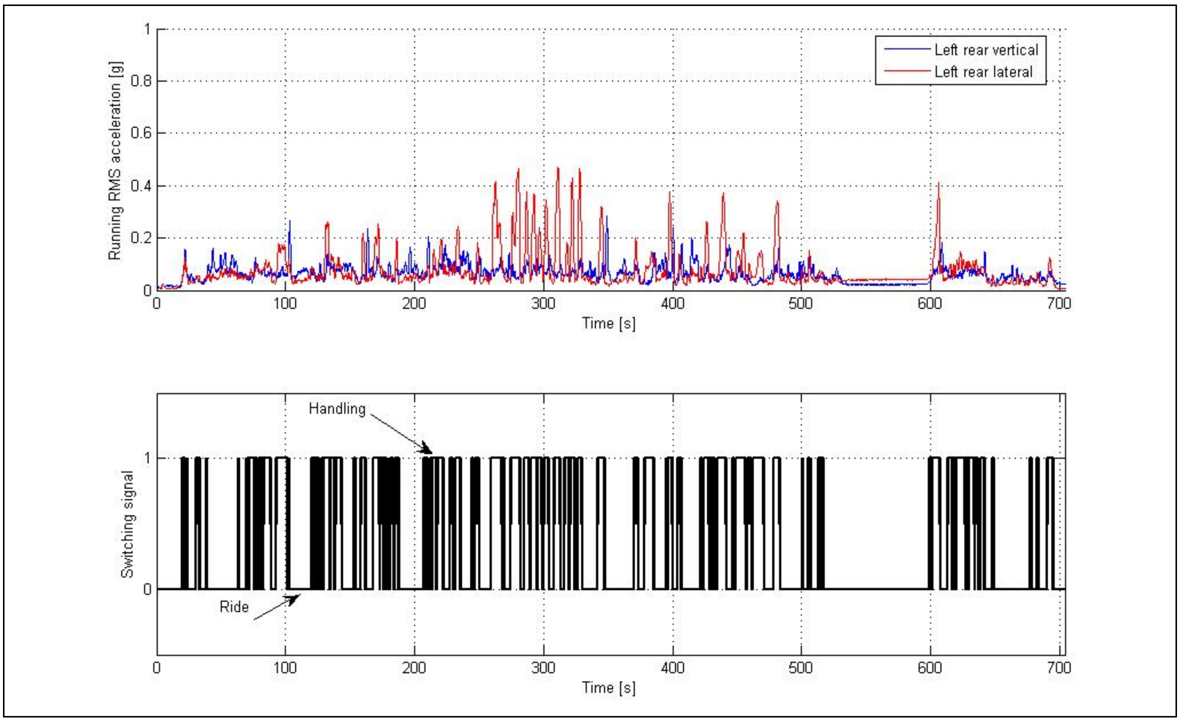


Figure 5.37 – RRMS strategy for city driving

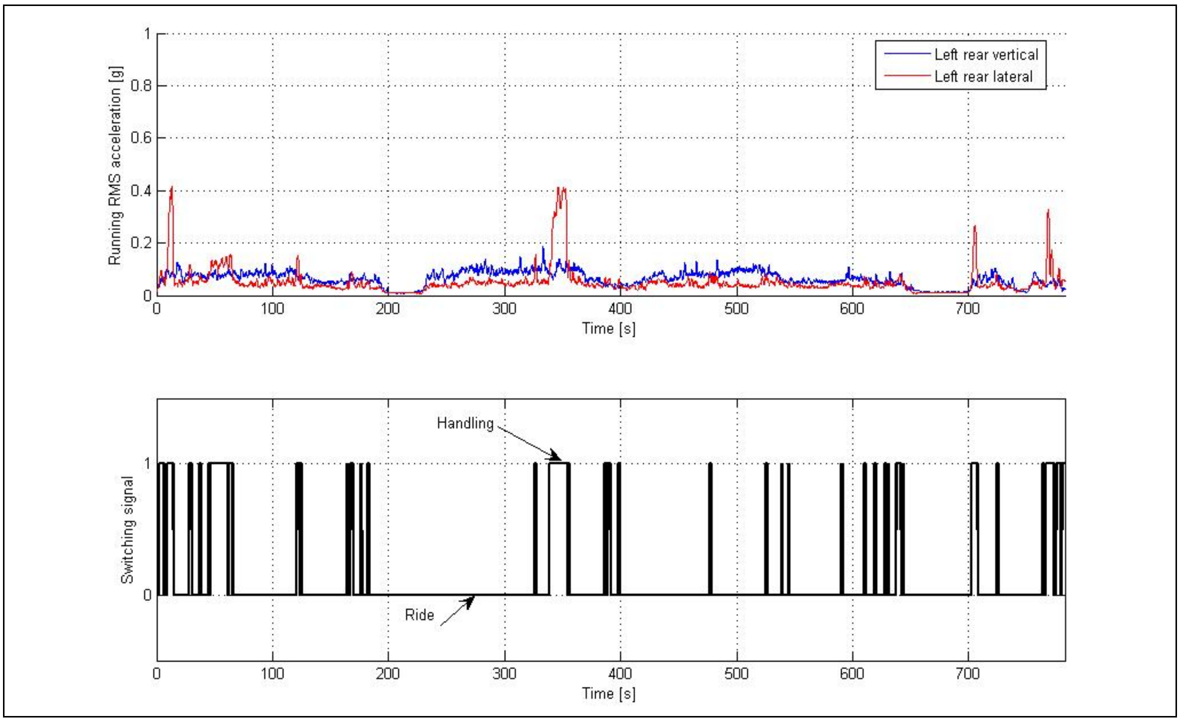


Figure 5.38– RRMS strategy for highway driving

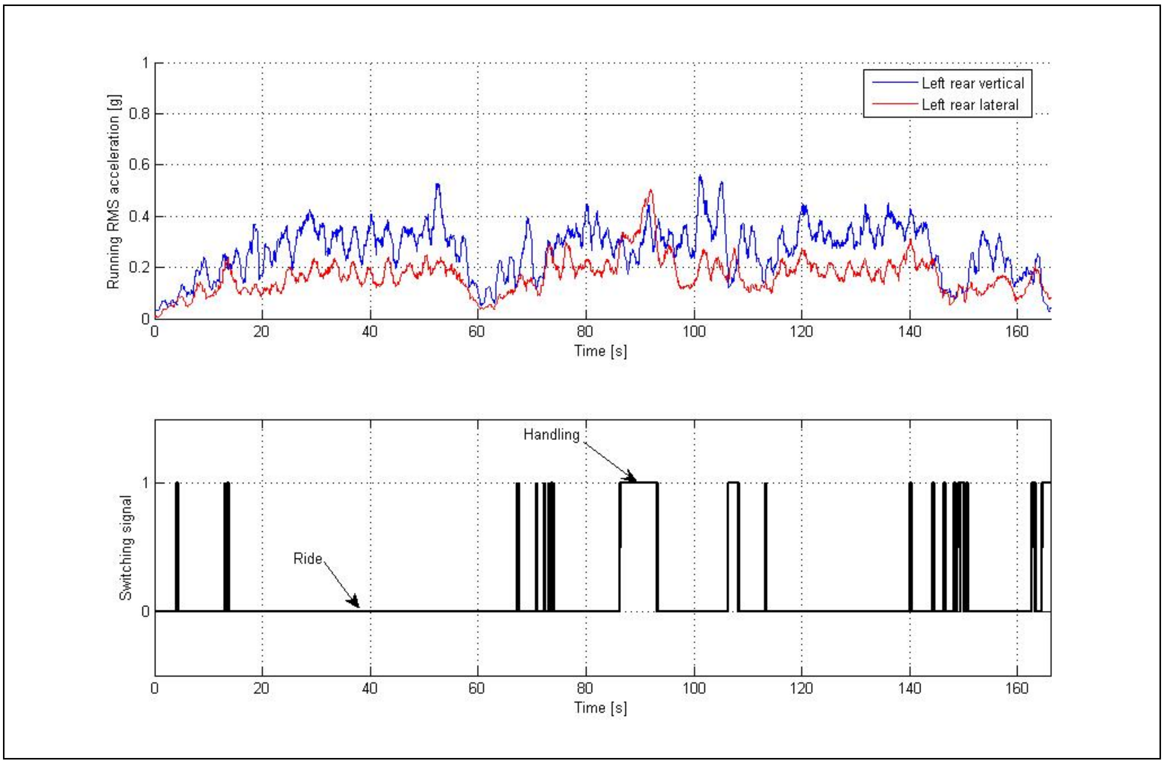


Figure 5.39– RRMS strategy for off-road driving

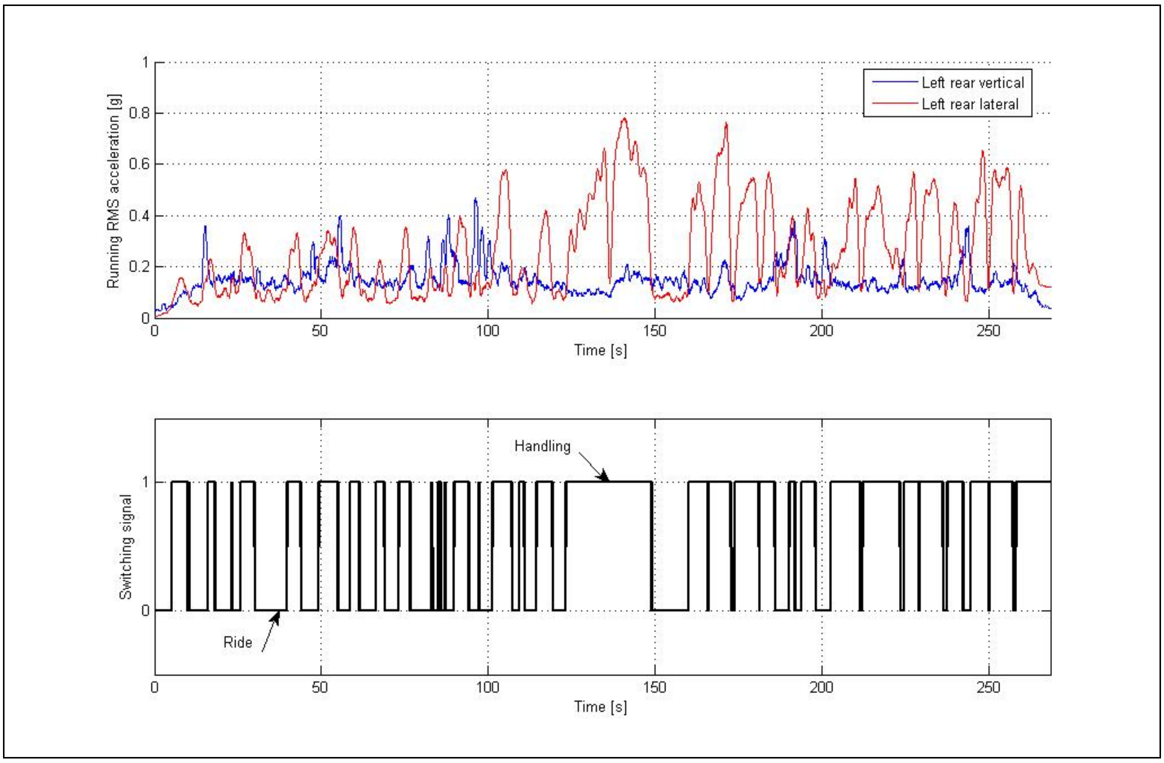


Figure 5.40– RRMS strategy for mountain pass

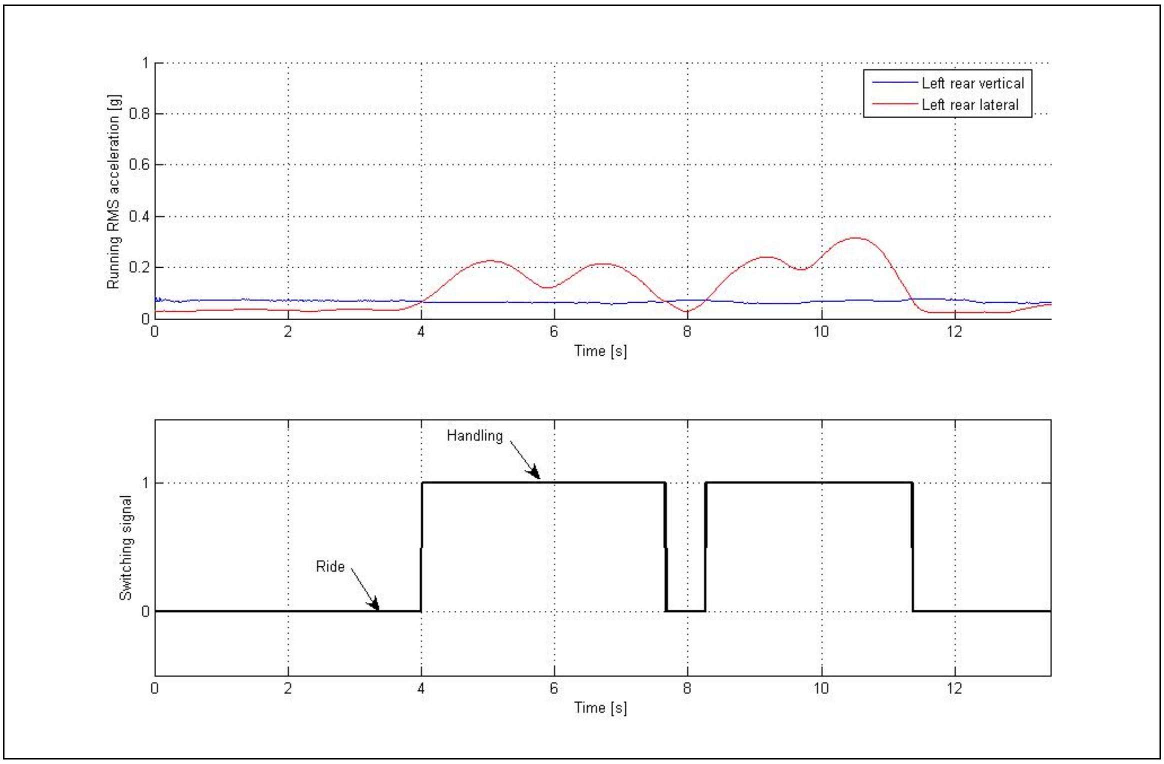


Figure 5.41– RRMS strategy for handling test

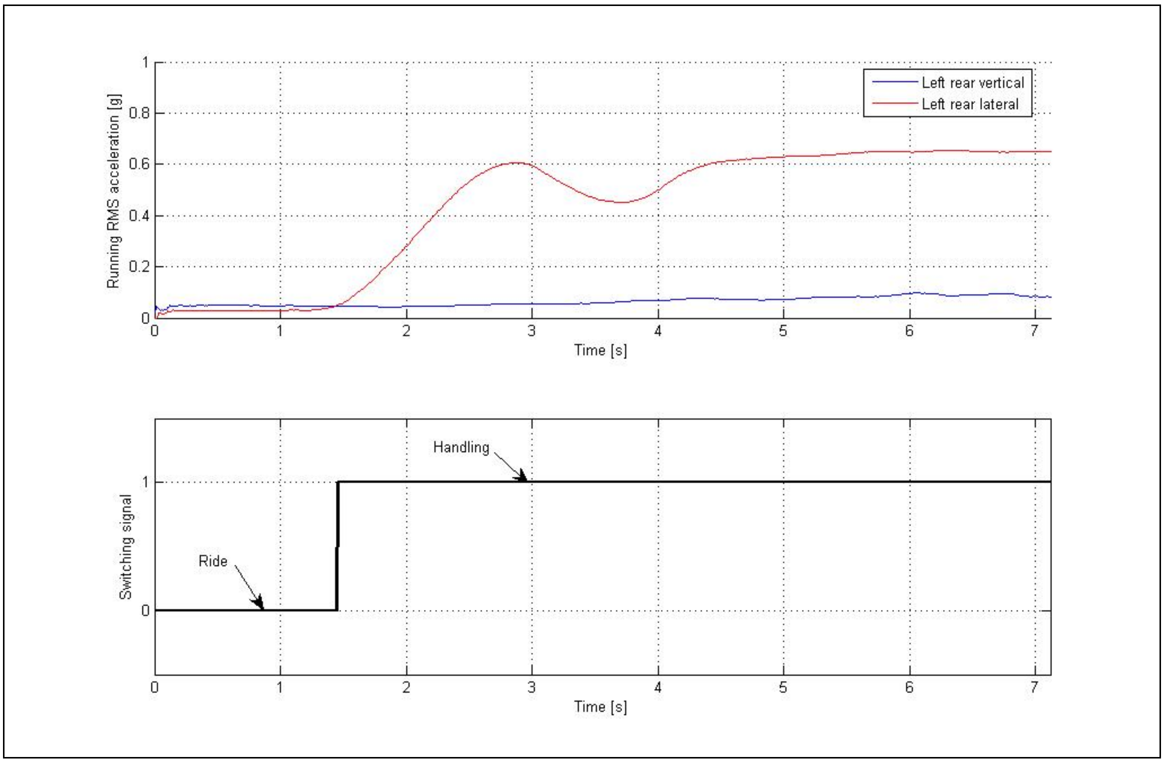


Figure 5.42– RRMS strategy for rollover test

For the analyses discussed above, a 100-point or 1 second RRMS was used. The number of points in the RRMS is expected to influence the response time, threshold levels and rejection of noise for short duration events. Figure 5.43 indicates the effect of the number of points in the RRMS on both the RRMS value and the resultant switching of the system for the handling test. The ideal behaviour would be if the system switches to "handling" mode immediately upon starting the test (*i.e.* at 3.7 seconds), and then remains in "handling" mode for the duration of the test. Figure 5.43(a) indicates the RRMS of the lateral acceleration for number of points from one to 500. The one point RRMS corresponds to the absolute value of the measured acceleration, while the 500 point RRMS corresponds to a five second RRMS. An increase in the number of points results in more "smoothing". The RRMS magnitude also decreases with an increase in the number of points. This means that the threshold levels should be decreased as the number of points is increased.

The corresponding switching according to the RRMS strategy is indicated in Figure 5.43(b). The y-axis has no units but just indicates the switching pattern for the eight different analyses. For the one point RRMS, switching occurs quickly after the start of the test (at 3.7 seconds). The switching delay as a function of the RRMS duration is indicated in Figure 5.44. As the RRMS duration increases, the switching delay increases accordingly. A one point RRMS does however result in many switchovers between "ride" and "handling" mode. As the RRMS duration is increased, the number of switchovers decreases. RRMS durations of 2 seconds and higher result in the system staying in "handling" mode for the duration of the test. The percentage time spent in the "handling" mode is indicated in Figure 5.45 as a function of the RRMS duration. As the RRMS duration increases above 2 seconds, the initial delay results in a reduction of time spent in the "handling" mode. The choice of RRMS duration is therefore a trade-off between response time and switching behaviour. Values between one and two seconds seem to be a reasonable starting point.

5.7 Conclusion

It is concluded that, of all the proposed strategies, only the running RMS (RRMS) appears to work for all the test conditions. Vehicle tests must be performed to validate the strategy.

A combination of strategies may also result in improvements, *e.g.* the steering angle can be used to determine the switching point from "ride comfort" to "handling", but switching back to "ride comfort" may then be based on the running RMS, or simply delayed by a fixed time to eliminate spurious switching.

If the terrain or driving conditions could be successfully identified, using for example artificial intelligence techniques (self organising maps, neural networks *etc*.), other concepts (*e.g.* steering angle *vs.* vehicle speed) may be successfully implemented by adapting thresholds according to operating conditions.

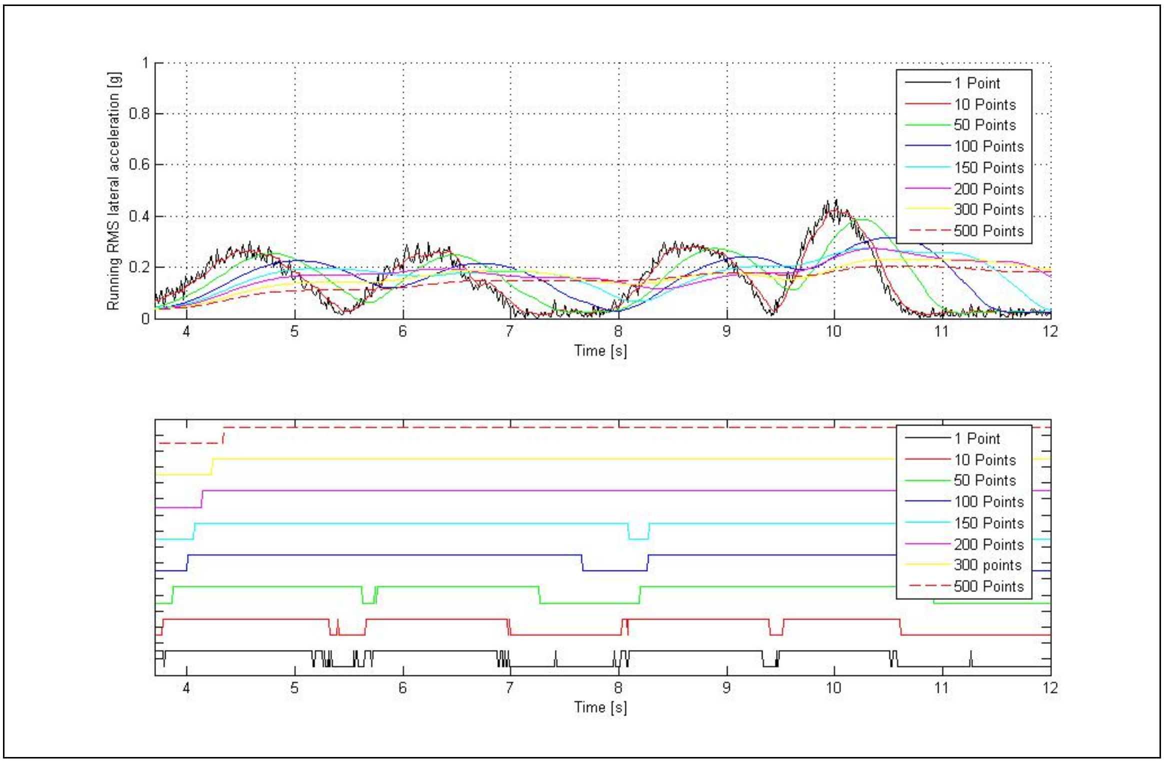


Figure 5.43 – Effect of number of points in the RRMS on switching for handling test

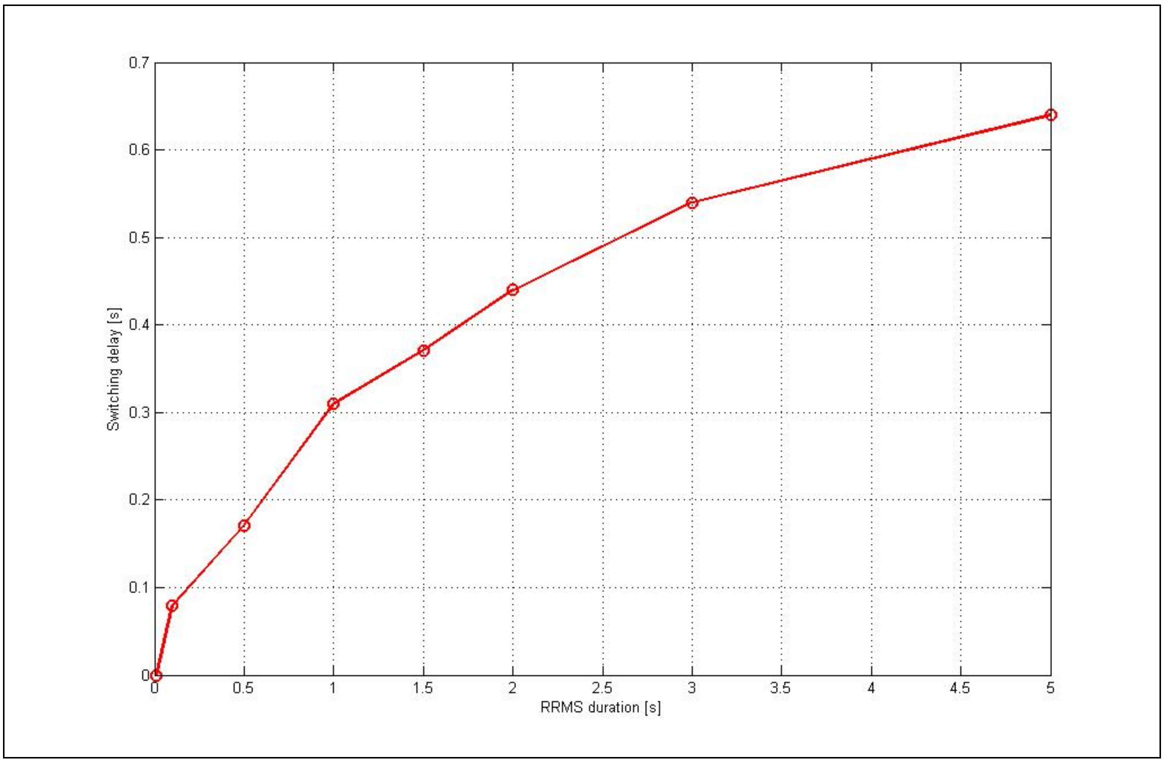


Figure 5.44 – Effect of number of points in the RRMS on the switching delay for handling test

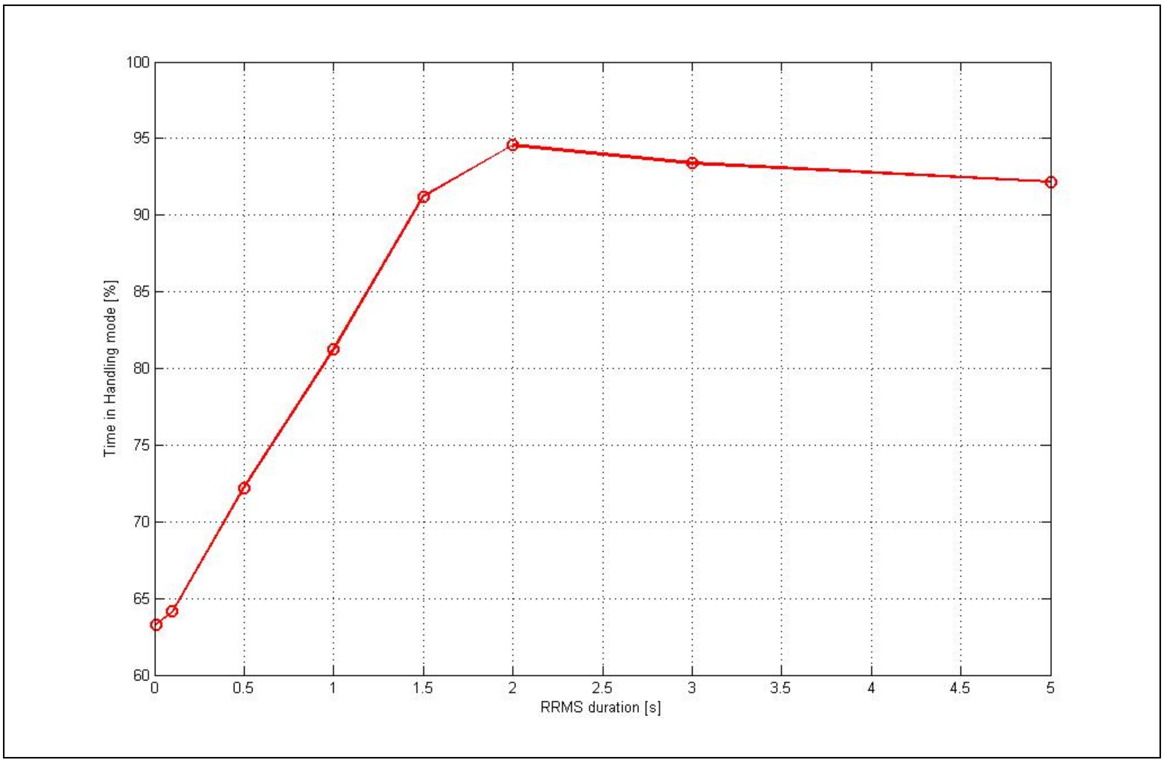


Figure 5.45 – Effect of number of points in the RRMS on time spent in “handling” mode for handling test

Chapter 6

VEHICLE IMPLEMENTATION

The integration of the $4S_4$ suspension hardware, associated hydraulics and electronics on the test vehicle is discussed in this chapter. Ride comfort and handling test results, performed on the vehicle with the $4S_4$ system fitted, are quantified, discussed and compared to baseline values obtained from testing of the baseline vehicle. Results are interpreted to determine whether the system works as intended and if the proposed "ride comfort *vs.* handling" decision strategy performs as predicted.

6.1 Installation of $4S_4$ hardware on test vehicle

Mounting of the new suspension system to the test vehicle required relatively minor modifications to the chassis and axle mounting points. Mudguards on the inside had to be cut to make provision for the units. The struts are mounted on the same centerline as the baseline suspension system. One notable change is the absence of any rubber elements in the mounting arrangement compared to the baseline suspension system, where the dampers were mounted to the chassis and axles with rubber bushes. The original rubber bump stops and axle-locating links were not modified. This results in exactly the same suspension travel and suspension kinematics as the baseline suspension system.

The prototype $4S_4$ units, as fitted to the right hand side of the test vehicle, are illustrated in Figures 6.1 to 6.5. Purpose-made top and bottom mounting brackets can be seen in Figure 6.1. The required wiring to the solenoid valves, as well as the hydraulic pipe for height adjustment is visible in Figure 6.5.

The pressure transducers, used to measure strut pressure, can be seen on top of the aluminium valve blocks in the figures.

Ride height adjustment capability was also incorporated on the test vehicle. The requirement for the ride height adjustability is that the system should be able to raise or lower the vehicle body up to the maximum or minimum elevation in 30 seconds. The minimum required oil flow for all four struts was calculated to be 1.57 l/min. The pump used has a volumetric displacement of 1.0 cm³/rev and delivers 3.0 litres per minute at a motor speed of 3000 r.p.m. The required oil reservoir should hold sufficient oil to guarantee functionality during lowering or raising of the vehicle. In order to have a sufficient reserve, a reservoir with a usable capacity of 5.9 litres was selected. The hydraulic power pack consists of a 12 Volt direct current (DC) electric motor, hydraulic gear pump and oil reservoir supplied by SPX Stone (**Anon, 2005c**). The assembled DC power pack is shown in Figure 6.6. The power pack is driven from a supplementary 12 Volt battery that is connected in parallel to the vehicle's 12 Volt battery.

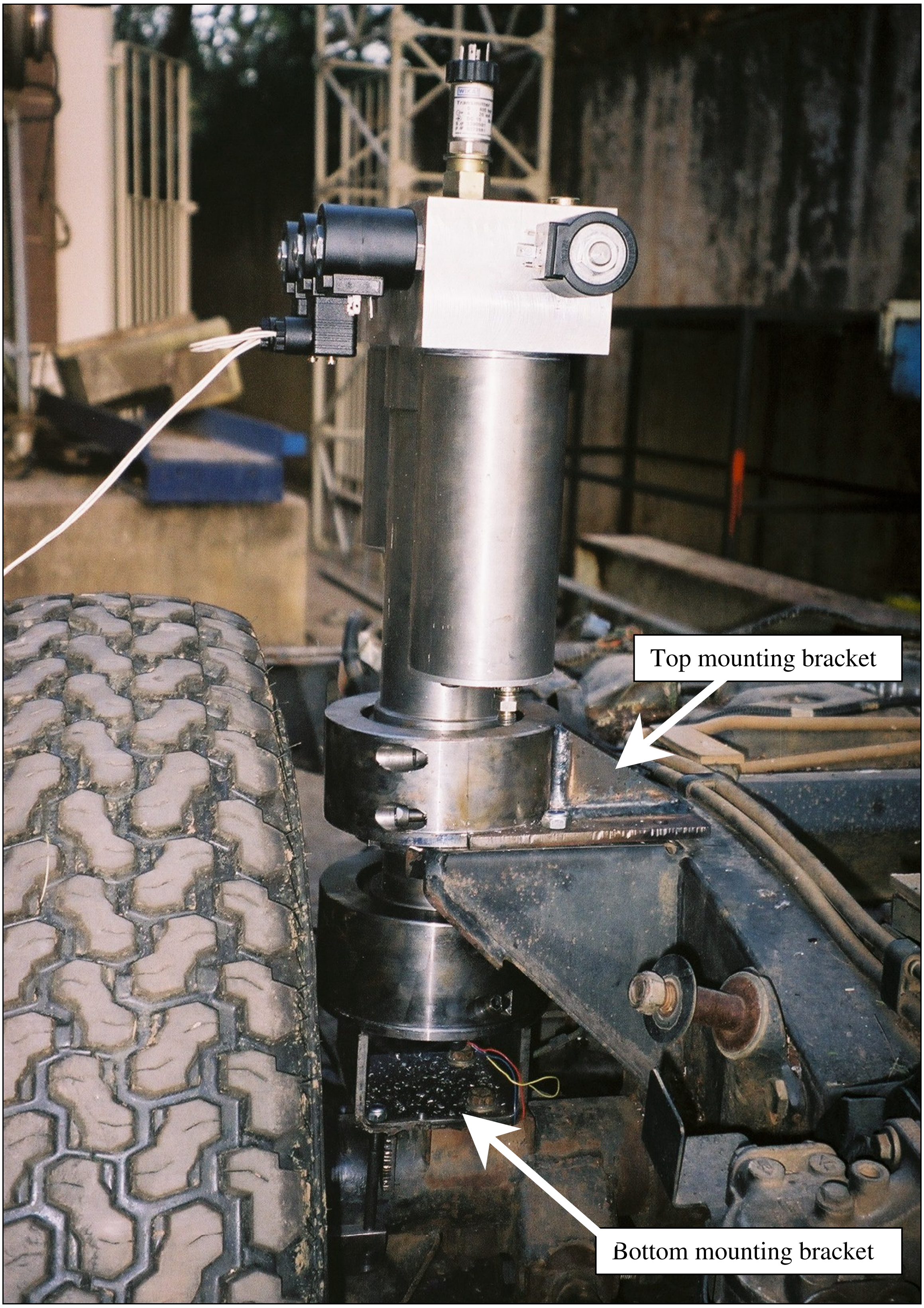


Figure 6.1 - Right rear suspension fitted to chassis – front view

Figure 6.2 - Right rear suspension fitted to chassis – inside view

Figure 6.3 - Right front and right rear suspension fitted to chassis

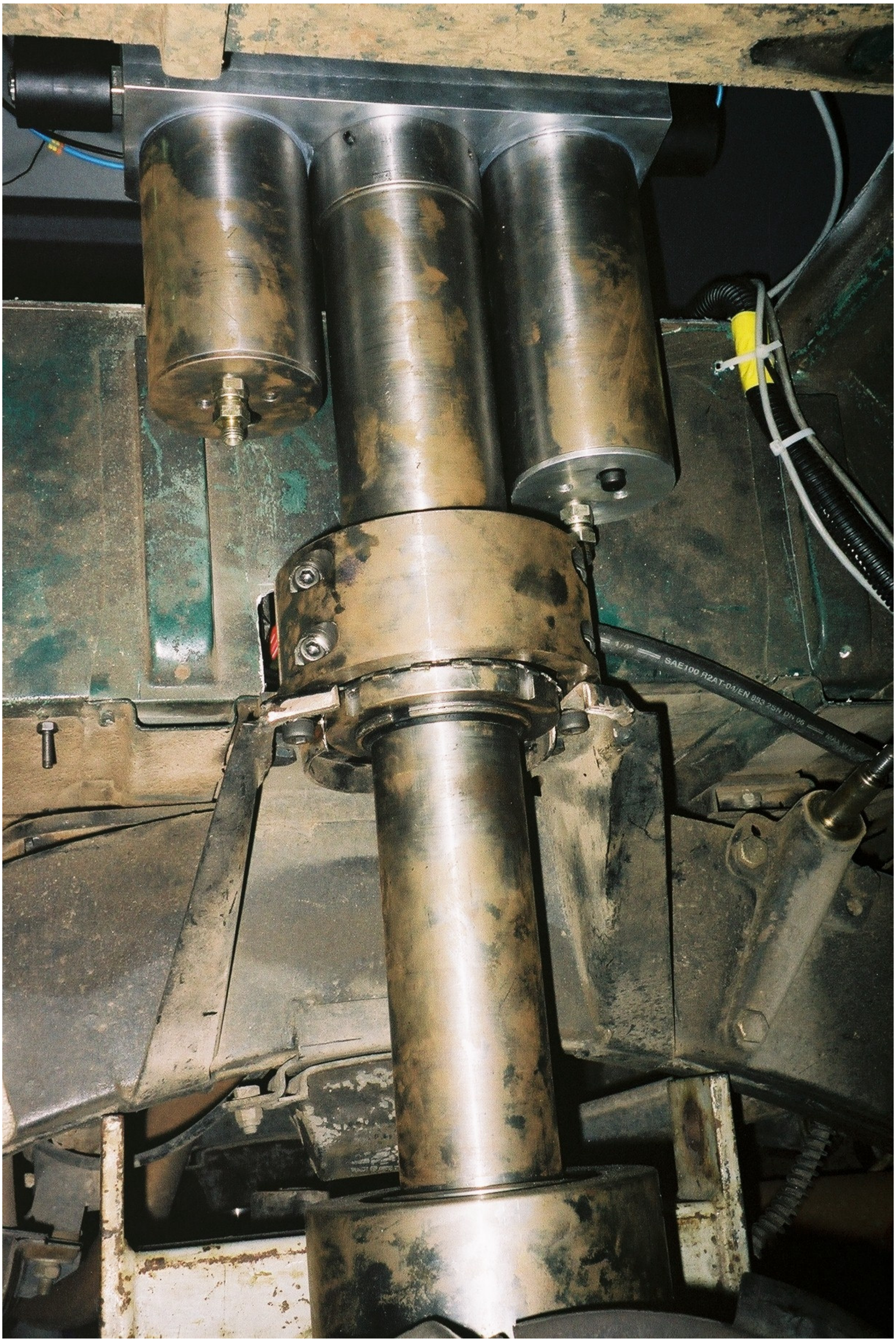


Figure 6.4 - Right rear suspension fitted to test vehicle – side view

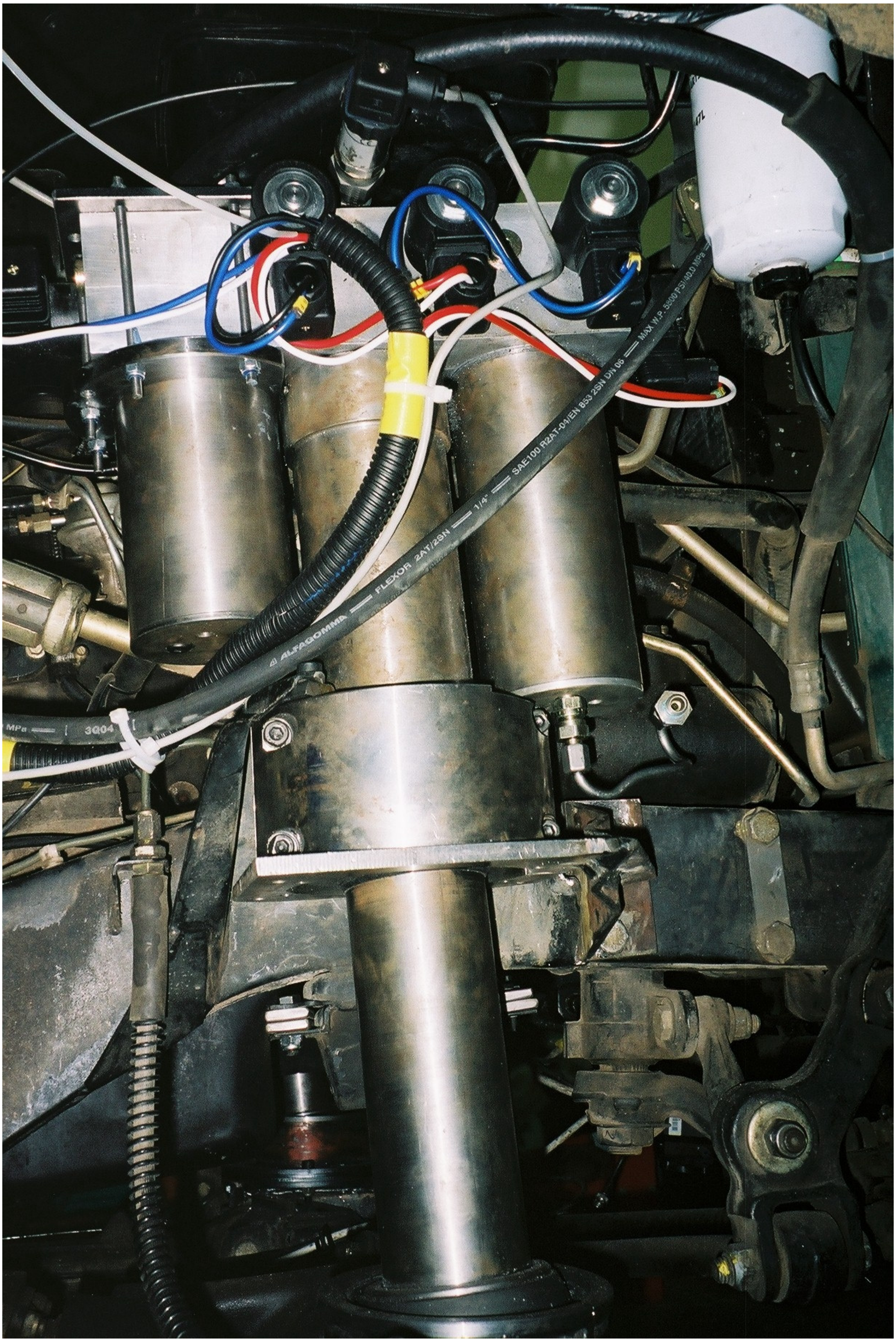


Figure 6.5 - Right front suspension fitted to test vehicle – side view

Figure 6.6 - Assembled hydraulic power pack

A control manifold (Figure 6.7) is used to regulate the oil flow from the power pack to the individual struts, or to let the oil flow back to the oil reservoir.

Figure 6.8 indicates the hydraulic pump and associated reservoir and valves used for height adjustment, mounted in the load area of the vehicle. The solid-state relays used to switch the solenoid valves are also shown.

6.2 Control electronics

The $4S_4$ control system controls ride height as well as the different spring and damper settings by means of solenoid valves. For this purpose it is necessary for the controller to process analog signals, from sensors measuring the vehicle's current operating conditions, to switch the solenoid valves and hydraulic power pack.

The control unit is based on a Coremodule 420 computer (PC-104 form factor) from AMPRO. Analog inputs are measured with a Diamond Systems MM-16-AT 16-bit analog to digital convertor card. The digital outputs, controlling the solid-state relays, are provided by a Diamond Systems Onyx-MM-DIO card. A schematic diagram of the control unit is provided in Figure 6.9.

Figure 6.7 - Control manifold for ride height adjustment

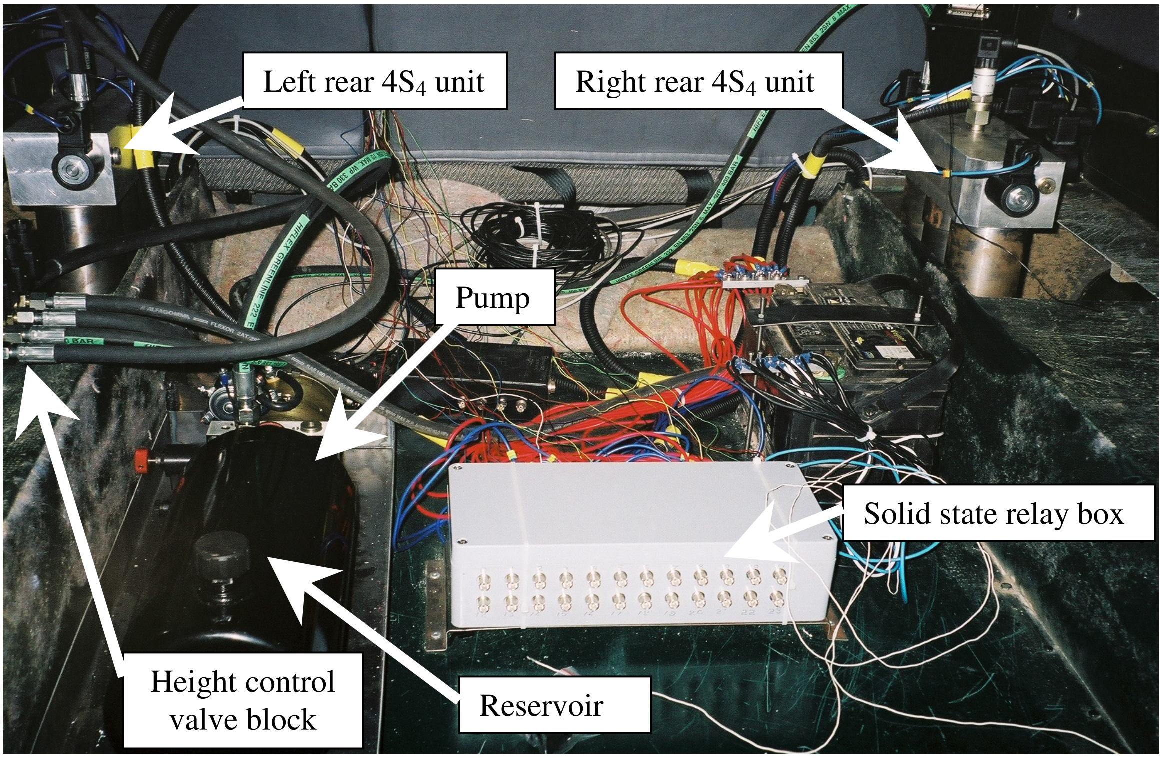


Figure 6.8 – Piping, wiring and electronics

The four relative strut displacements (one for each $4S_4$ strut) as well as lateral and vertical accelerations are digitised by the analog to digital converter card. The ride height adjustment algorithms use the relative strut displacements while the “ride *vs.* handling” decision uses only the vertical and lateral body accelerations. After computing the required settings for all the valves, the valves are switched via the digital output card and solid state relays.

The control algorithm used for the “ride *vs.* handling decision” is the running RMS (RRMS) strategy proposed in chapter 5. The control loop runs at 100 Hz and employs a 100-point (or 1 second) RRMS. Both lateral and vertical accelerations are measured using a single Crossbow CXL04LP3 tri-axial accelerometer with built-in signal conditioning.

There are several issues that require special attention including zero positions, signal drift and noise. Initially the aim was to mount the accelerometers on the test vehicle in the vicinity of the center of mass. This mounting position resulted in high noise content from presumably the engine or drivetrain vibration. Although the mean signal was zero, the RMS resulted in an unacceptably high value. The accelerometer was subsequently moved to a position under the rear seat, where the engine vibration levels were significantly reduced. As an additional precaution, these accelerations were filtered with a 6^{th} order analog low-pass Butterworth filter, with a 50 Hz cut-off frequency, to prevent aliasing and to filter out engine related vibration. The software also recorded measurements before each test in order to obtain the zero values on all sensors.

Relative strut displacements are measured using ICS-100 In-Cylinder Sensors from Penny & Giles. The linear potentiometer positioning sensors are mounted inside the struts, surrounded by the hydraulic oil. They offer low hysteresis, low electrical noise, stable output under temperature extremes and good dither vibration performance. No signal conditioning is necessary and the sensors only require a stable supply voltage to operate reliably.

All the valves are normally closed *i.e.* in the event of power failure (*e.g.* due to a flat battery, cable breaking or control computer that reboots), the $4S_4$ system will revert to the “handling” mode (*i.e.* stiff spring and high damping with no height adjustment). This adds a failsafe capability to the system. Due to the required reverse logic, the switching signals indicated in the rest of this chapter have the opposite meaning to those in Chapter 5, i.e. a logic “1” now means “ride” mode (all valves open) and a logic “0” indicates handling mode (all valves closed).

6.3 Steady state handling

The steady state handling characteristics of the vehicle were tested using a constant radius test. In this test, the vehicle was driven around a circle of 25-meter radius starting at crawl speed and gradually increasing speed until the vehicle reached it’s handling limit (based on either sliding out or impending rollover). Data is represented as a graph of steering link displacement against lateral acceleration. A zero slope on this graph indicates neutral steer. A positive slope (steering angle increases as lateral acceleration increases) indicates understeer while a negative slope (steering angle decreases as lateral acceleration increases) indicates oversteer.

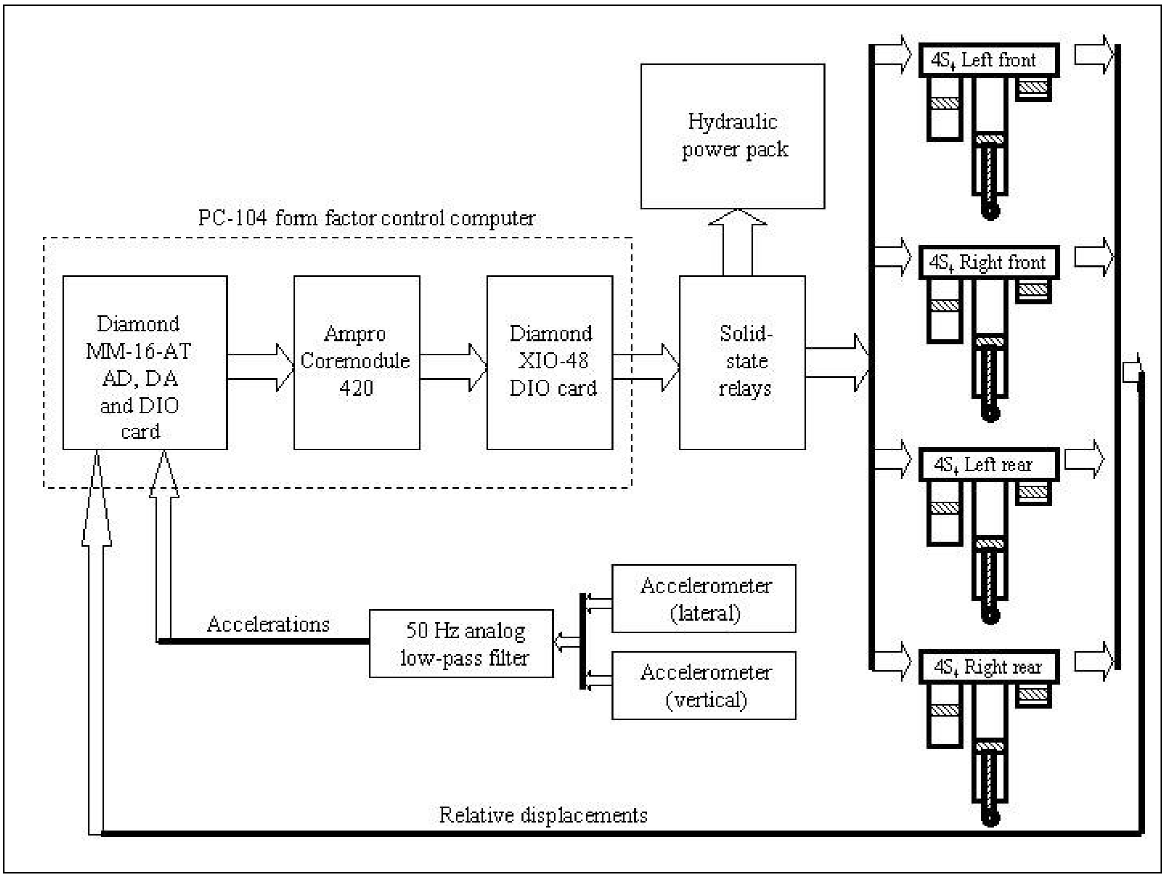


Figure 6.9 - Control computer schematic

The effect of front:rear roll-stiffness balance was determined experimentally by performing preliminary tests without any control applied, but just switching the valves manually.

Measured characteristics are indicated in Figure 6.10 for the "handling" (all springs hard) mode, front suspension hard (rear soft) and rear suspension hard (front soft). All three settings steer neutrally up to 0.3 g after which oversteer develops for "all springs hard" and "rear springs hard". In the case where the front suspension is hard, the vehicle steers neutrally up to 0.4 g and thereafter understeers. This indicates that switching the front suspension to the hard setting can induce understeer. The opposite scenario is probably also valid (*e.g.* switching the rear to hard will result in oversteer) although this is not as evident from the data in Figure 6.10. A possible handling strategy then is to switch the front suspension to hard when oversteer is detected and *vice versa* to counter understeer.

Figures 6.11 to 6.14 indicate the relative roll angle between the body and axle against lateral acceleration for different combinations of spring stiffness and ride height. It is clear that stiffening the suspension, as well as lowering the ride height, considerably reduces the body roll angle.

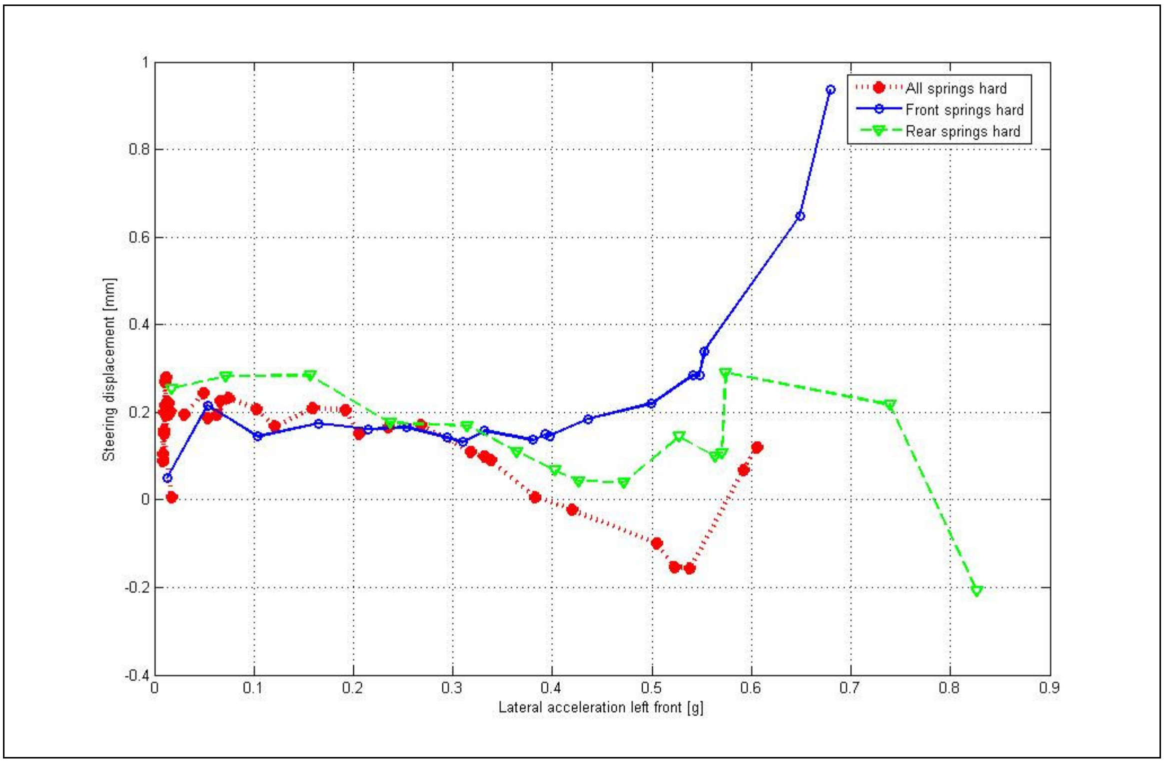


Figure 6.10 – Constant radius test results

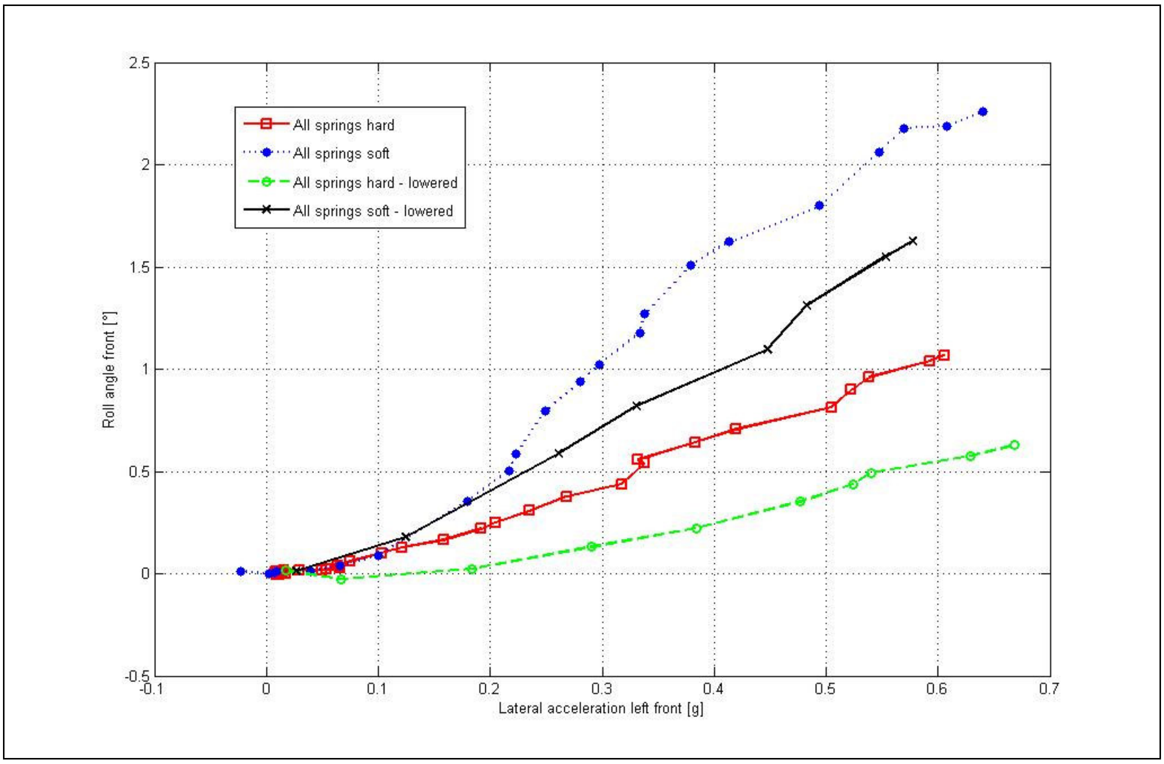


Figure 6.11 – Relative roll angle front – effect of ride height

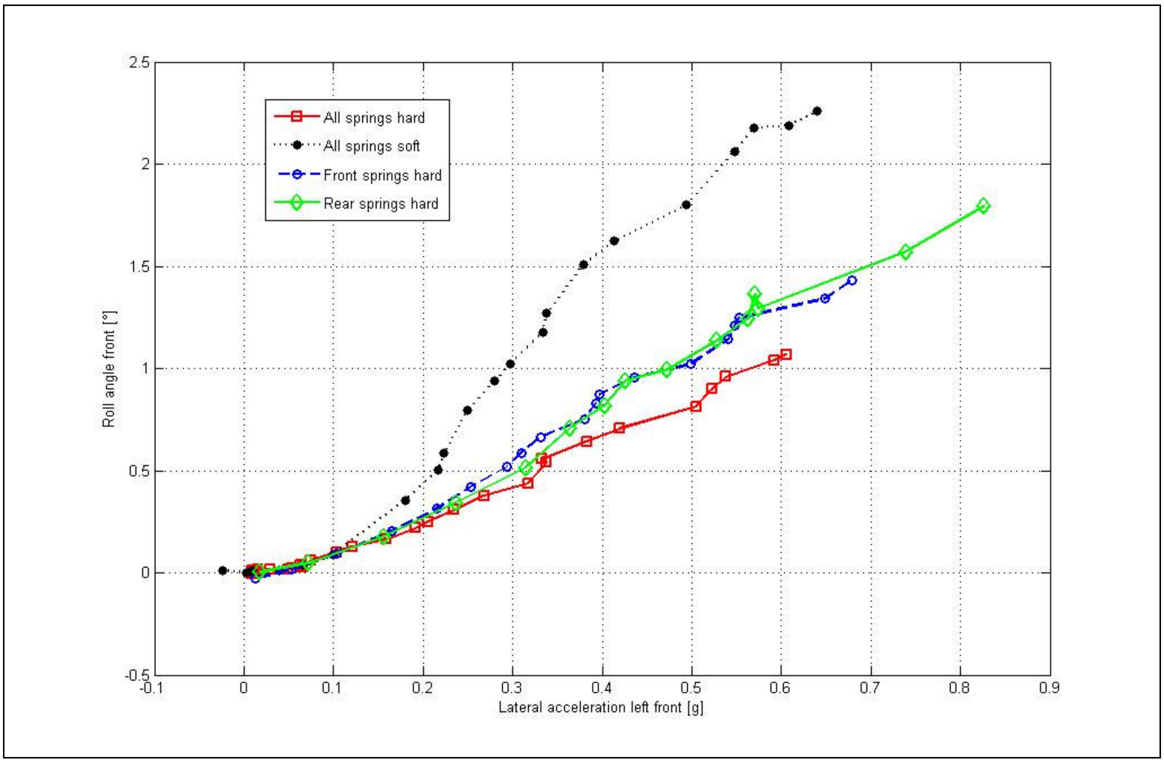


Figure 6.12 – Relative roll angle front – effect of stiffness

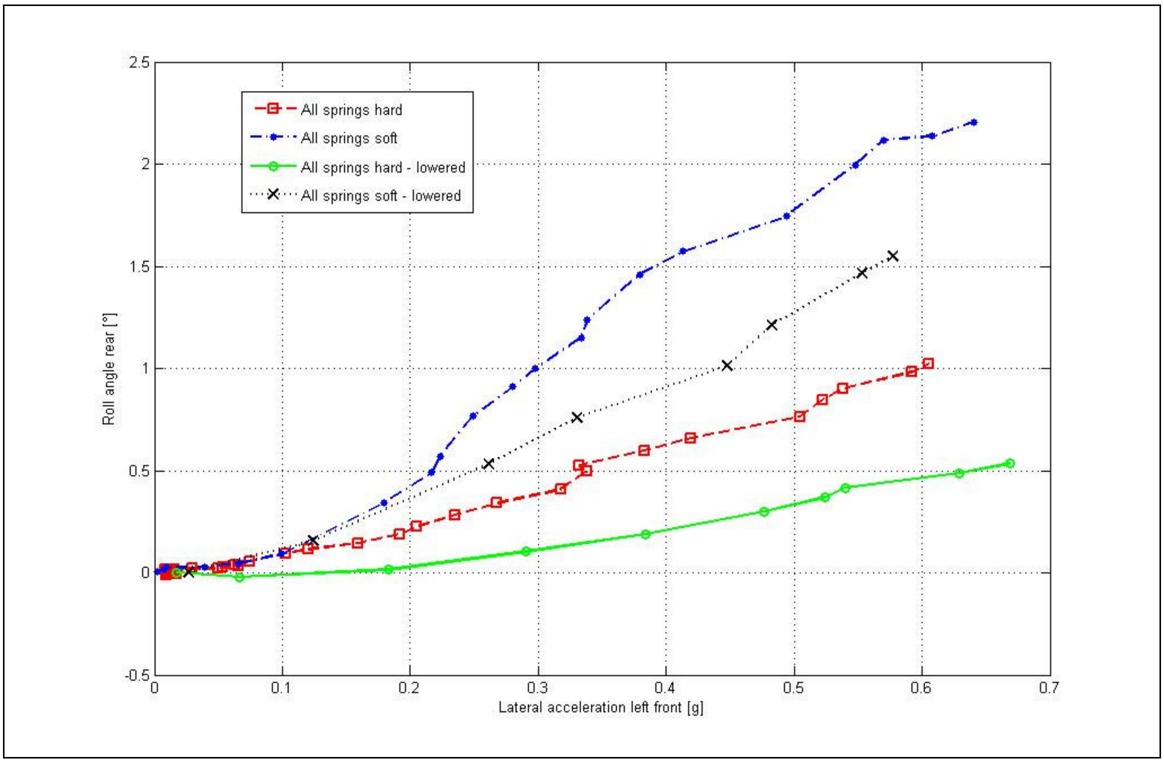


Figure 6.13 – Relative roll angle rear – effect of ride height

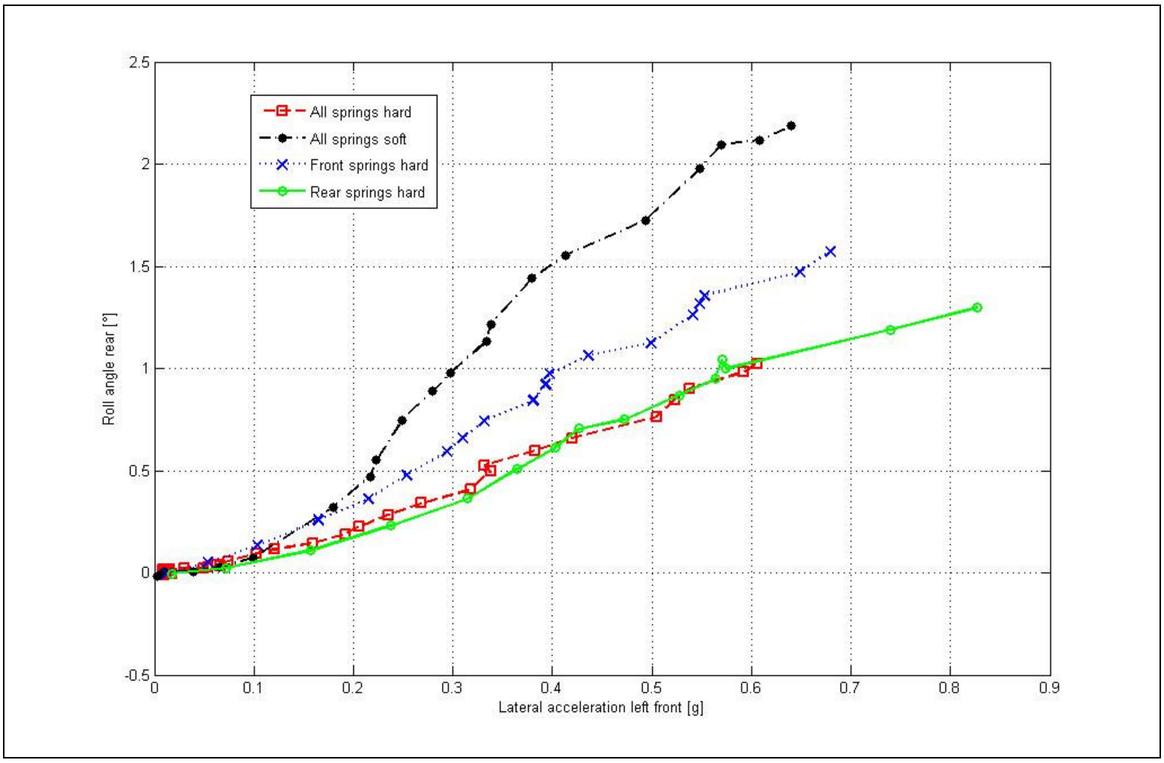


Figure 6.14 – Relative roll angle rear – effect of stiffness

It is concluded that the hard suspension setting results in a considerable decrease in body roll. Further improvements might be obtained by switching the roll stiffness balance between front and rear to counter over- or understeer.

6.4 Dynamic handling

In order to evaluate the dynamic handling characteristics of the vehicle, the ISO 3888 double lane change test was performed. The vehicle body roll angle is used as a measure of handling.

Handling test results through the ISO 3888 double lane change at a vehicle speed of 58 km/h is indicated in Figure 6.15. At first valve selection was performed manually without any control applied. The vehicle was driven in a specific gear against the diesel engine's governor in an attempt to keep the vehicle speed as constant as possible, and to ensure the same speed for different test runs. Test speeds did however vary slightly *e.g.* between 57 and 61 km/h, 70 and 75 km/h and 82 to 84 km/h respectively for the three gear ratios used for testing. The "ride" setting (soft spring and low damping), "handling" setting (stiff spring and high damping) and baseline vehicle is compared to each other at the same vehicle speed. It is observed that the "handling" setting results in significant improvements in roll angle (between 61 and 78 %) compared to the baseline vehicle. The "ride" setting is, however, very soft and results in unsatisfactory handling as expected. Roll angle was determined in two ways. The top graph indicates the body roll angle obtained by integrating the roll velocity measurement and correcting for drift. The bottom graph indicates the relative roll angle between the vehicle body and the axle, calculated from the measured relative displacement on the left and right hand struts. The values for all four peaks, based on the relative roll angle between the body and the axle, are summarised in table 6.1. The “handling mode” results in significant improvements in roll angle, compared to the baseline vehicle.

Table 6.1 – Comparison between baseline and $4S_4$ relative roll angles through double lane change at 57 to 61 km/h

Peak	Baseline roll angle [°]	“Handling mode” roll angle [°]	“Ride mode” roll angle [°]	Improvement of “Handling mode” over baseline [%]
1	1.6	0.6	3.0	63
2	-2.1	-0.8	-4.5	62
3	-2.3	-0.9	-3.7	61
4	1.8	0.4	4.0	78

Figure 6.16 illustrates the effect of a 50 mm reduction in ride height on the body roll angle at 58 km/h. There is a slight improvement in roll angle for the “handling mode”. The major advantage is, however, seen in the “ride comfort mode” where the body roll angle is reduced substantially to the same levels as for the baseline suspension system. Note that the vehicle speed for the soft suspension with lowered ride height is marginally lower than for the other three test runs.

With these large differences between the “handling mode” and the “ride comfort” mode, it is imperative to investigate whether the RRMS control strategy will switch the $4S_4$ system to “handling mode” for the duration of the manoeuvre. Figure 6.17 indicates results for RRMS control at 61 km/h. After a delay of 0.8 seconds, the system switches to “handling mode”. It does however switch back to “ride mode” between 3.1 and 4.2 seconds. Figures 6.18 and 6.19 indicate that at speeds in the region of 75 km/h, the system switches back to “ride mode” in some of the tests (Figure 6.19) but remains in the “handling mode” for others (Figure 6.18). At higher speeds (above 80 km/h) the system stays in “handling mode” as indicated in Figures 6.20 and 6.21. The switching between settings at the lower speeds is not regarded as a problem as the vehicle is still far from the handling limits. When approaching the handling limits at higher speeds, the RRMS control functions correctly by switching to “handling mode” and remaining in “handling mode” until the manoeuvre is completed. The initial switching delay is also reduced from 0.8 seconds at 61 km/h to 0.5 seconds at 75 and 0.4 seconds at 84 km/h. The vehicle will therefore travel 13.5 m at 61 km/h and 9.2 m at 83 km/h before the $4S_4$ system switches from the “ride mode” to the “handling mode”. In actual fact the valve response time of between 0.04 and 0.09 seconds (see paragraph 4.7.6 in Chapter 4) should be added to this initial switching delay of the control strategy.

The comparison between roll angle for the “handling mode” and RRMS control is indicated in Figures 6.22 and 6.23. Both figures indicate that the RRMS control does not perform as well as the “handling mode” with a definite offset noticeable in the data. This is attributed to the delay from the start of the test until the RRMS strategy selects the handling mode. This switching delay results in an initial roll angle on the soft suspension. Once switching takes place, the large accumulator, and the oil in it, is isolated from the rest of the system. The portion of oil removed, results in a differential change in ride height between left and right and therefore an initial body roll angle. After switching takes place, the resulting roll angle corresponds to the “handling mode”, with an offset equal to the initial roll on the soft suspension. This offset in body roll angle is eliminated when the system switches back to “ride comfort mode”. The roll angles at 70 and 82 km/h however still compare favourably with the baseline roll angle at 57 km/h.

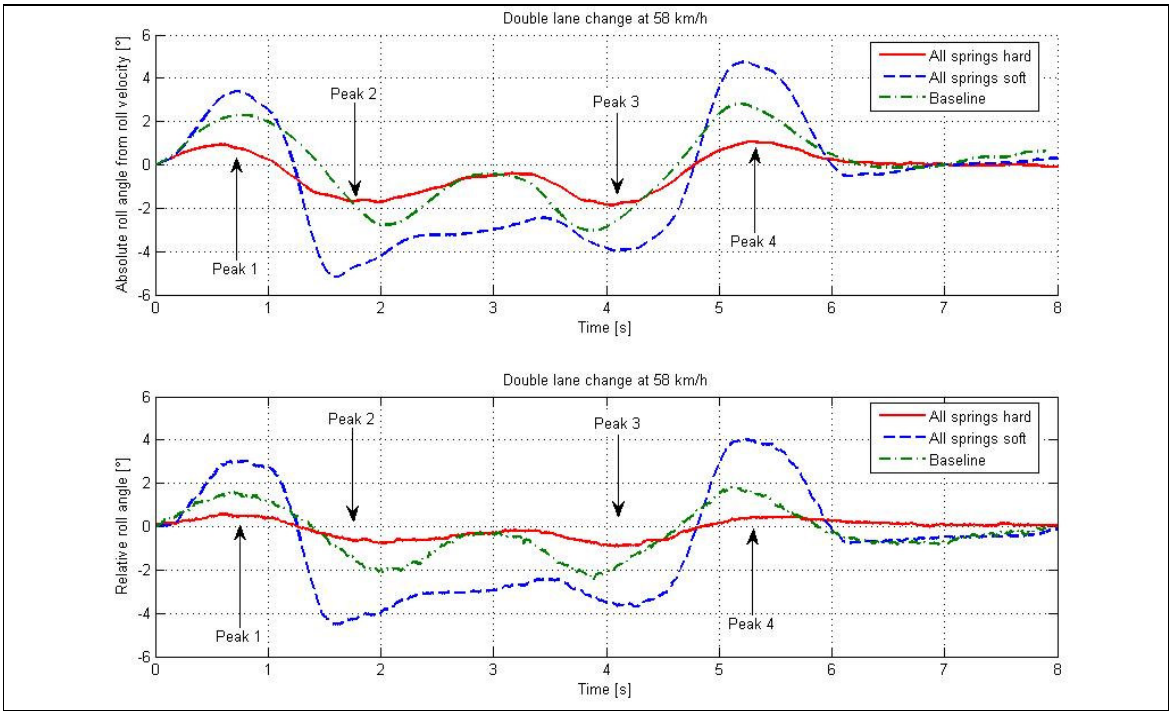


Figure 6.15 – Body roll with $4S_4$ settings compared to baseline at 58 km/h

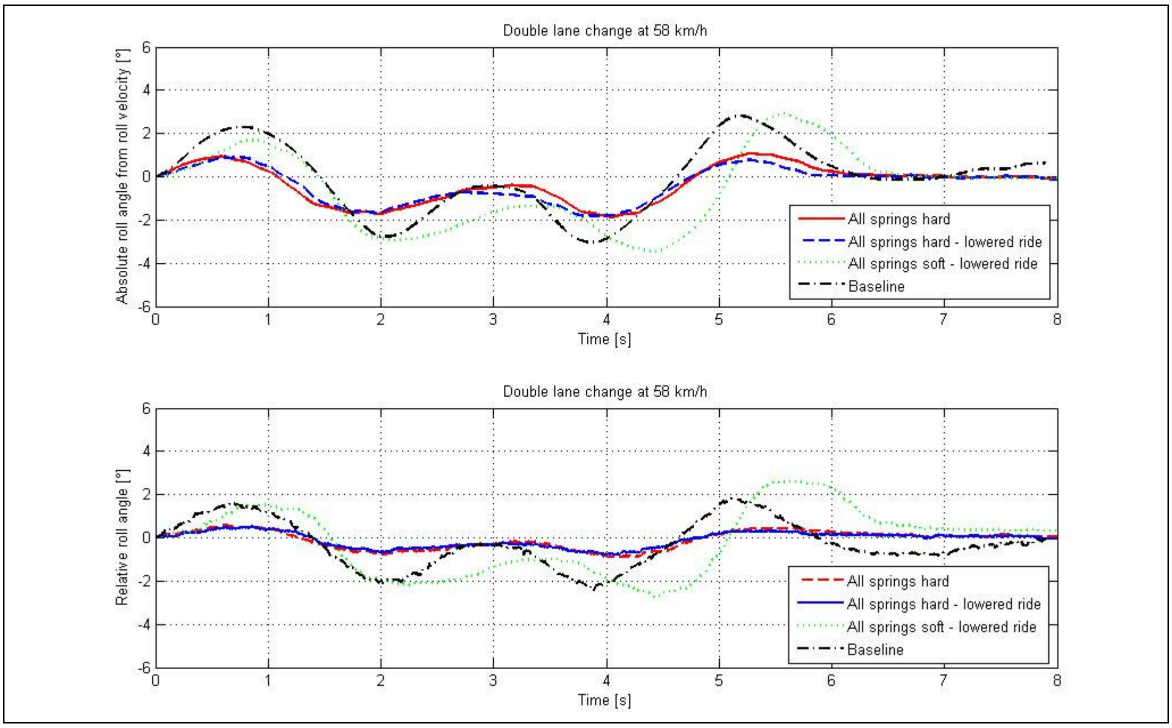


Figure 6.16 - effect of ride height on body roll at 58 km/h

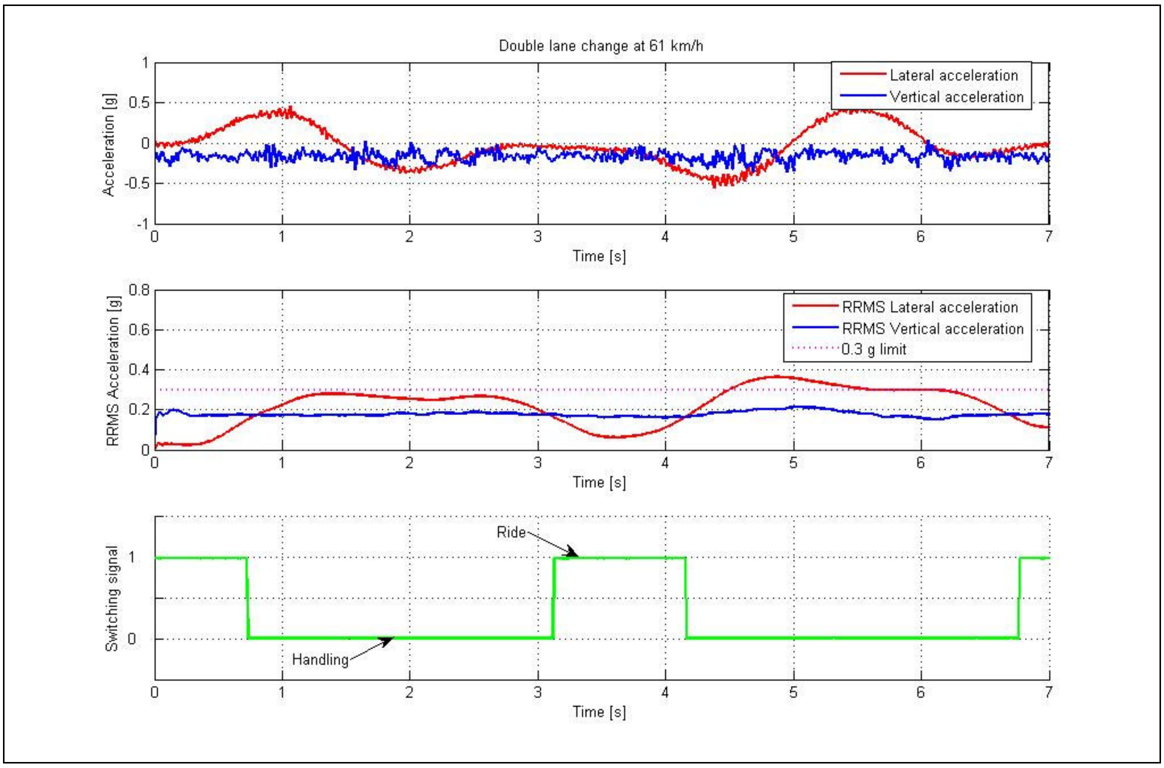


Figure 6.17 - RRMS control at 61 km/h

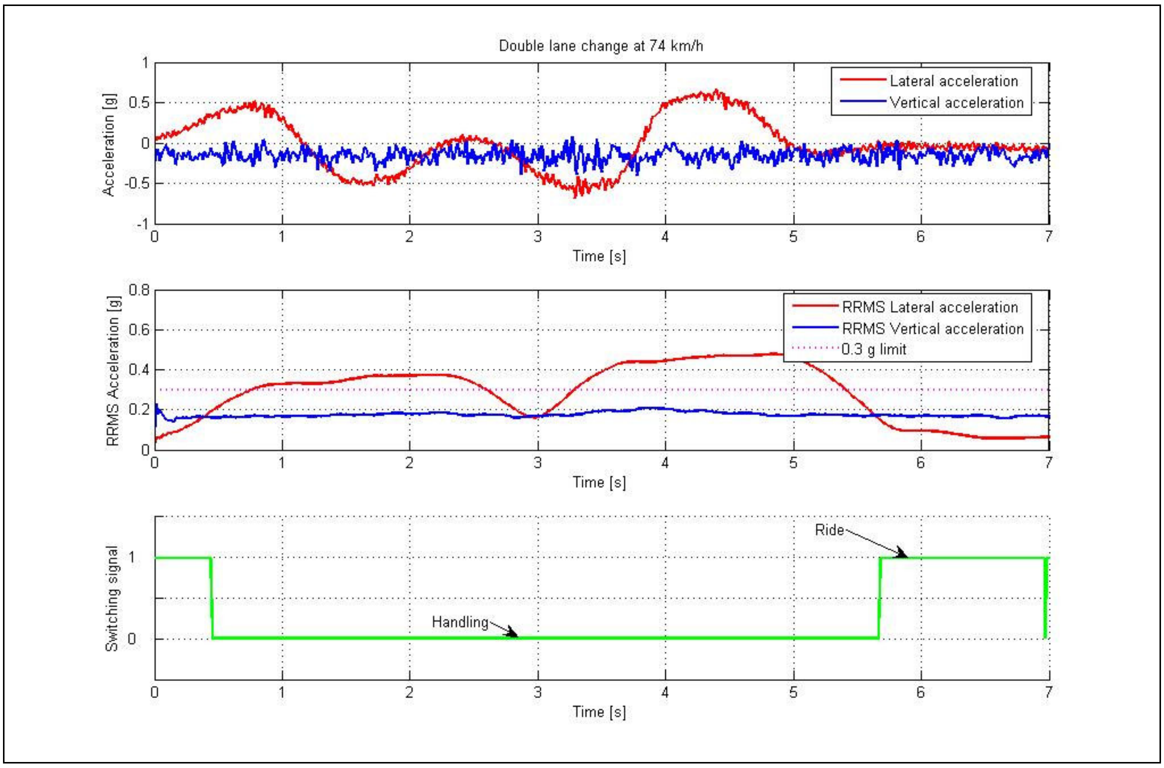


Figure 6.18 - RRMS control at 74 km/h

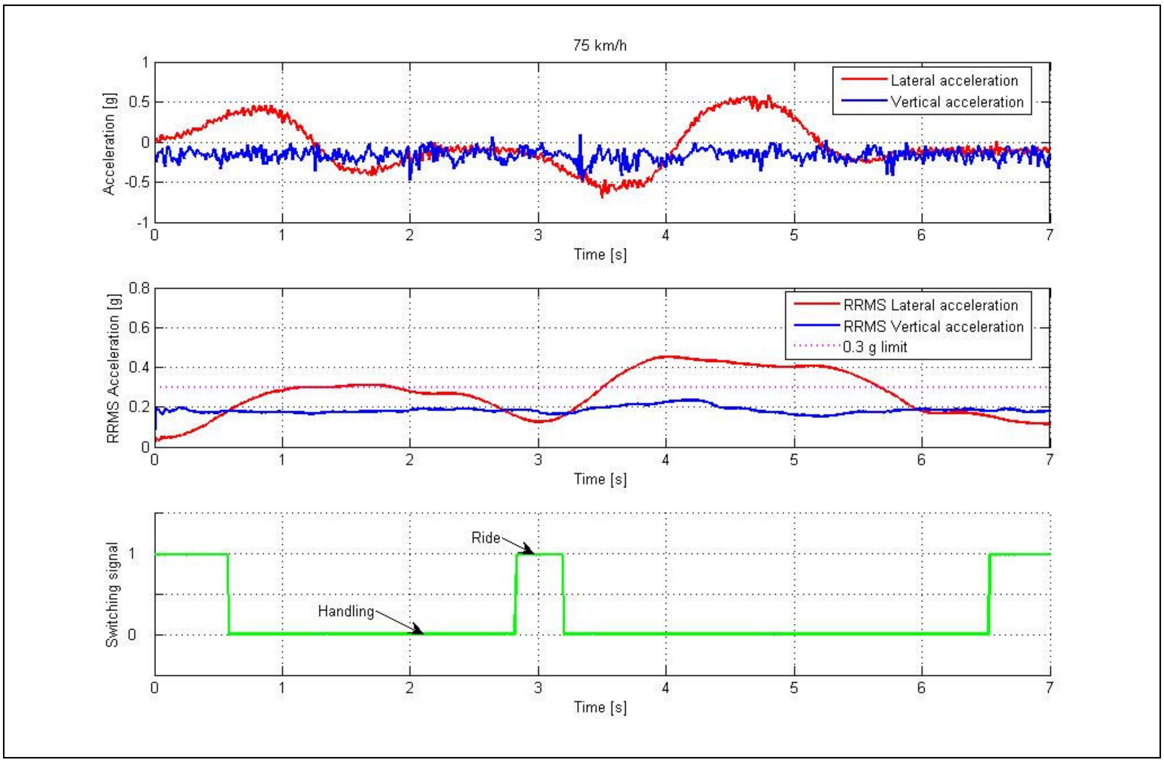


Figure 6.19 - RRMS control at 75 km/h

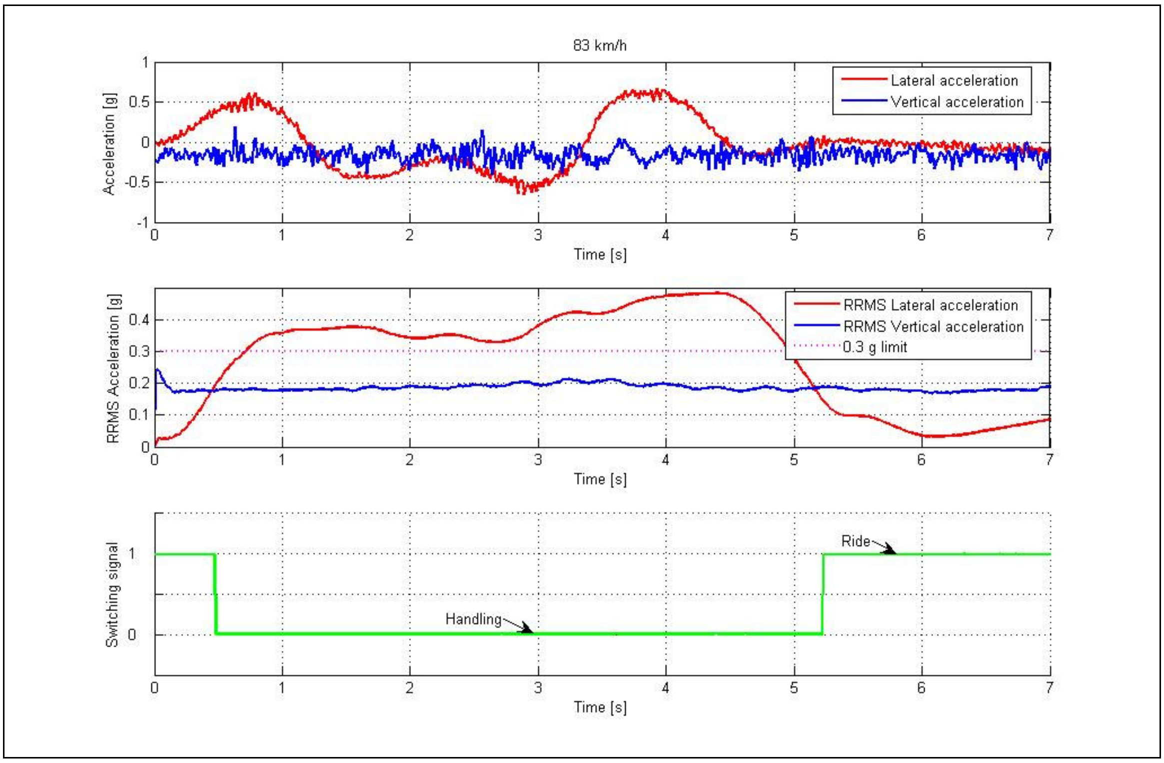


Figure 6.20 - RRMS control at 83 km/h

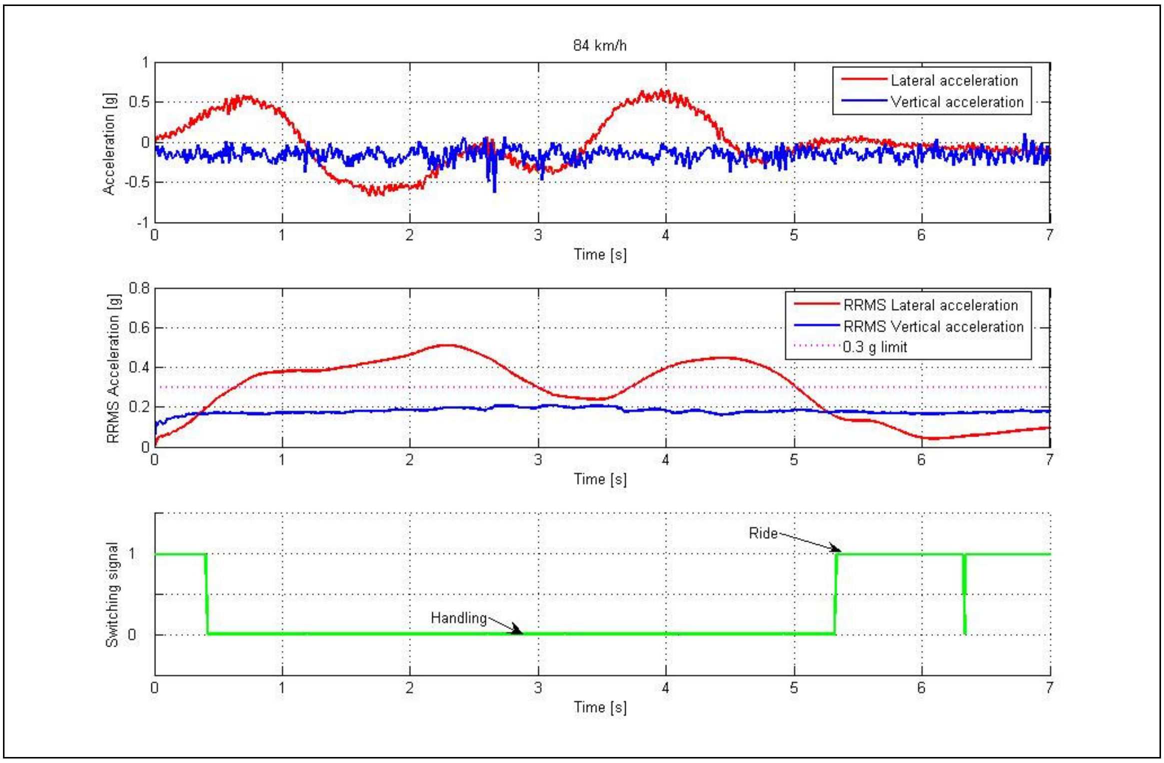


Figure 6.21 - RRMS control at 84 km/h

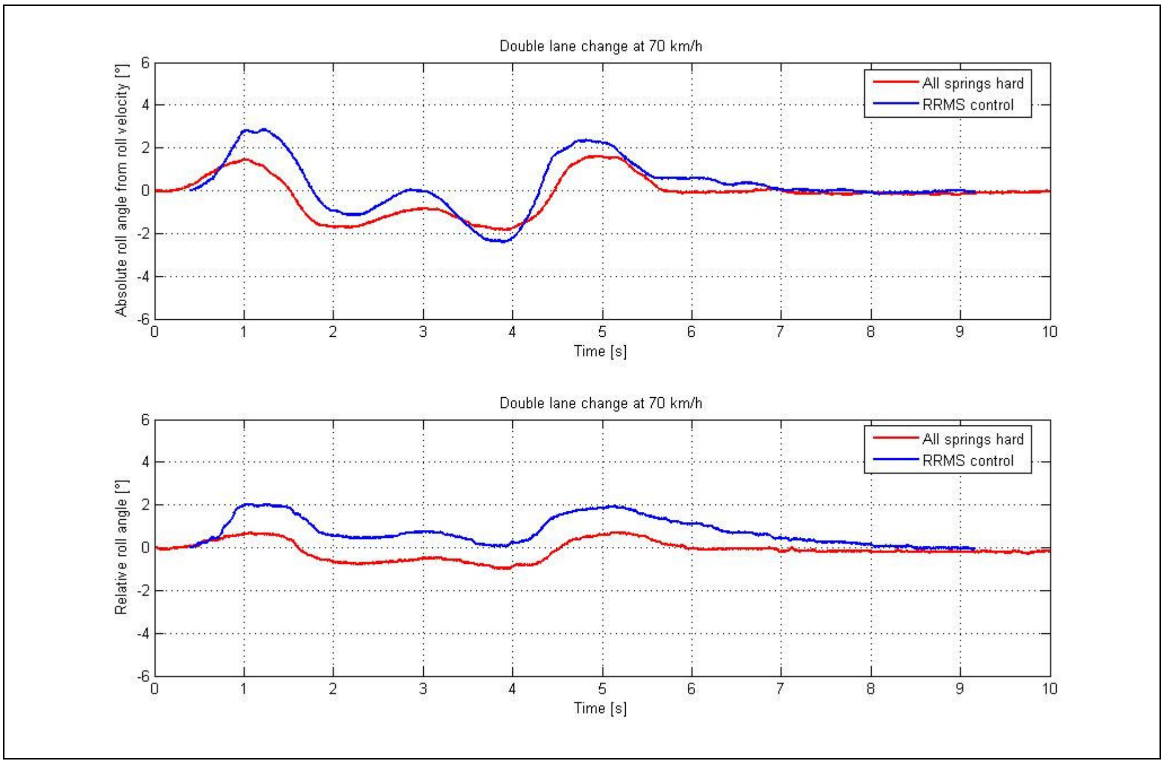


Figure 6.22 - RRMS control compared to “handling mode” at 70 km/h

As a final comparison, Figures 6.24 and 6.25 indicate the roll angle for the handling mode at three different speeds (Figure 6.24) and the corresponding roll angle for the RRMS control mode (Figure 6.25). The peak-to-peak roll angles of the RRMS strategy at 73 and 83 km/h are significantly lower than at 60 km/h, primarily due to the fact that the strategy does not switch between "handling" and "ride" modes during the manoeuvre, as it tends to do at 60 km/h. This is favourable as it will improve ride comfort at lower speeds but, at the onset of a handling manoeuvre, switch to handling as the vehicle speed increases, improving high-speed vehicle stability.

6.5 Ride comfort

For the evaluation of ride comfort, the vehicle is driven over the Belgian paving (see Figure 2.21 in Chapter 2) at five speeds. The vertical accelerations, measured at three positions on the vehicle body and weighed according to the BS6841 standard, is used as a measure of ride comfort.

In order to test if the RRMS control strategy performs correctly, the vehicle was driven over the Belgian paving track at different speeds in both the "ride comfort" mode (all soft) and the RRMS control mode. Figure 6.26 indicates that the strategy indeed switches to the soft setting on the Belgian paving. At 4.8 seconds the driver changes direction to avoid the very rough test track following the Belgian paving. During this manoeuvre the RRMS control strategy switches the suspension to "handling" mode. Figure 6.27 confirms that there is no significant difference in the ride comfort, at the three measuring positions and five speeds, when the "ride mode" is compared to the RRMS control. The data points for "handling mode" are only indicated for the lowest speed of 17 km/h.

The RRMS strategy performs correctly for driving in a straight line over a rough road. The "ride comfort mode" results in an improvement in ride comfort, of between 50 and 80%, compared to the "handling mode". A significant improvement in ride comfort with respect to the baseline values is however not experienced due to the following reasons:

i) The current $4S_4$ hardware has the same damper setting front and rear while on the baseline vehicle, front damping is considerably lower than rear damping.

ii) The low damping characteristic on the current $4S_4$ hardware has more or less the same characteristics as the rear dampers on the baseline vehicle due to pressure drops in the bypass valves and valve block channels. Significant improvements in ride comfort are only expected for damper characteristics less than 50% of the baseline values.

iii) The baseline vehicle's rear dampers are installed at an angle while the $4S_4$ dampers are vertical, thus exerting greater damper force even though the force-velocity characteristics are similar.

Refinement of the $4S_4$ damper settings for the low damping characteristic is necessary before ride comfort improvements will be noticed. This will mean enlarging the diameter of the existing ports and channels, and fitting valves with a lower pressure drop or higher capacity.

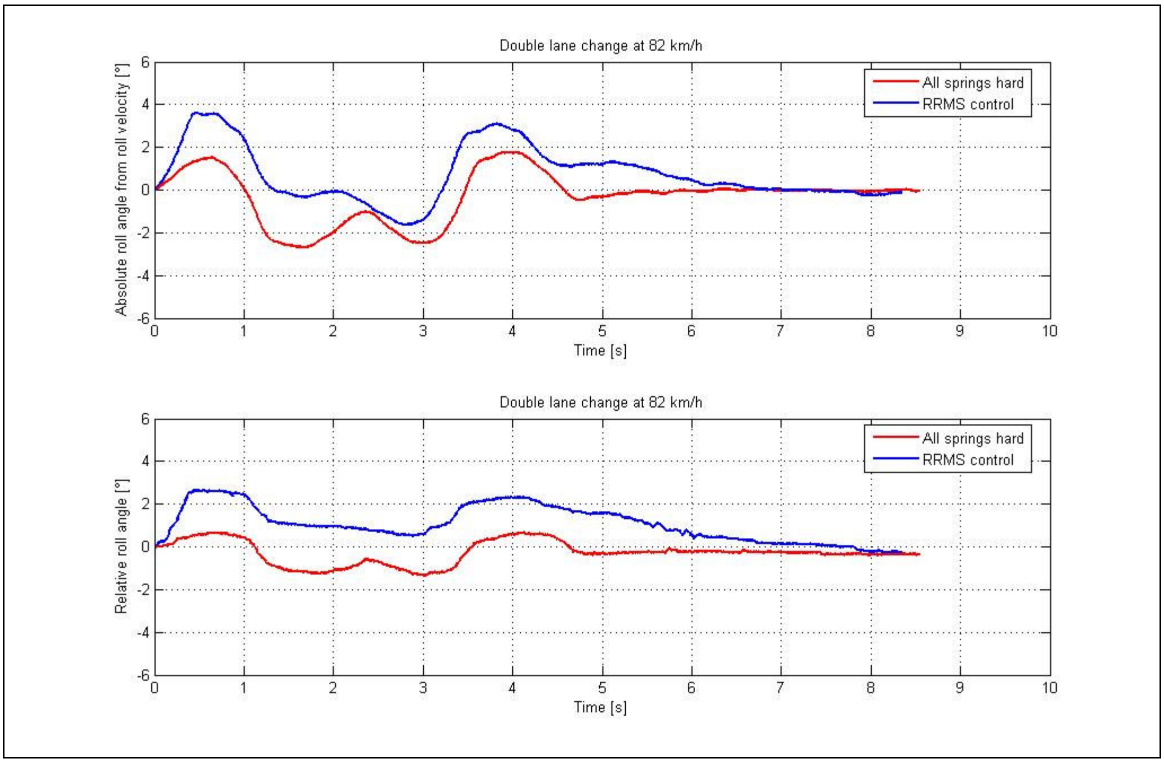


Figure 6.23 - RRMS control compared to "handling mode" at 82 km/h

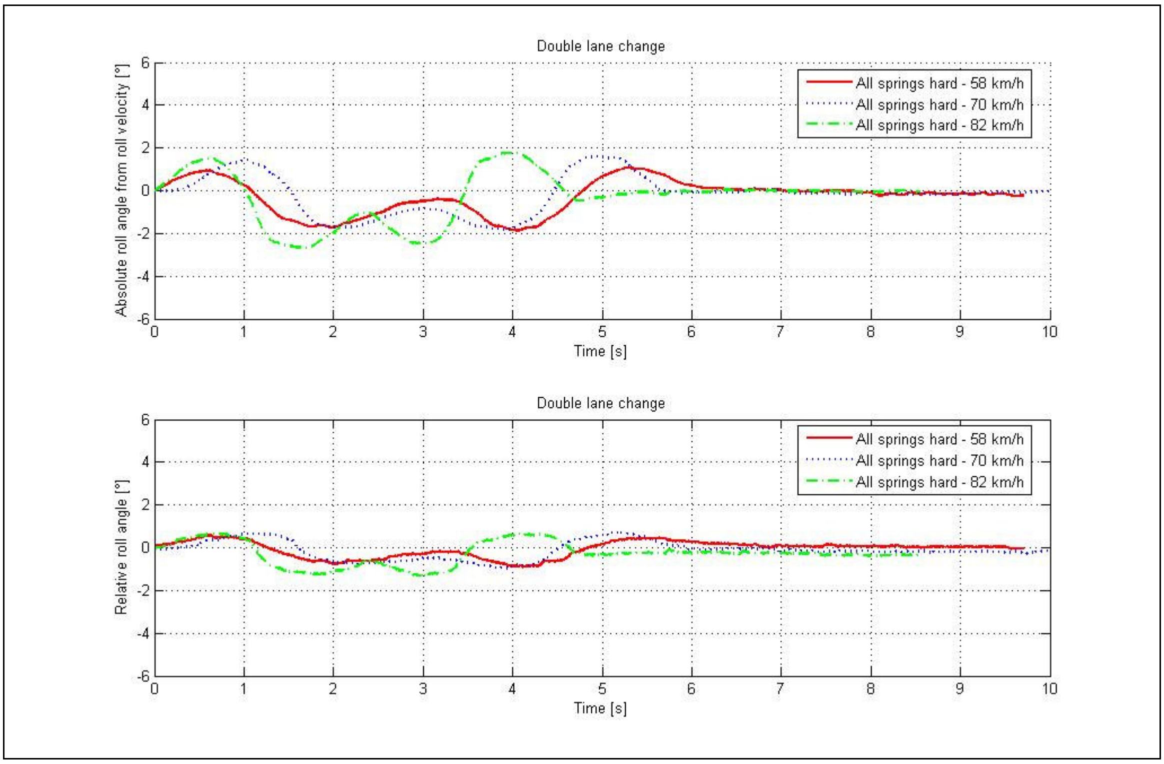


Figure 6.24 - Body roll for "handling mode" at different speeds

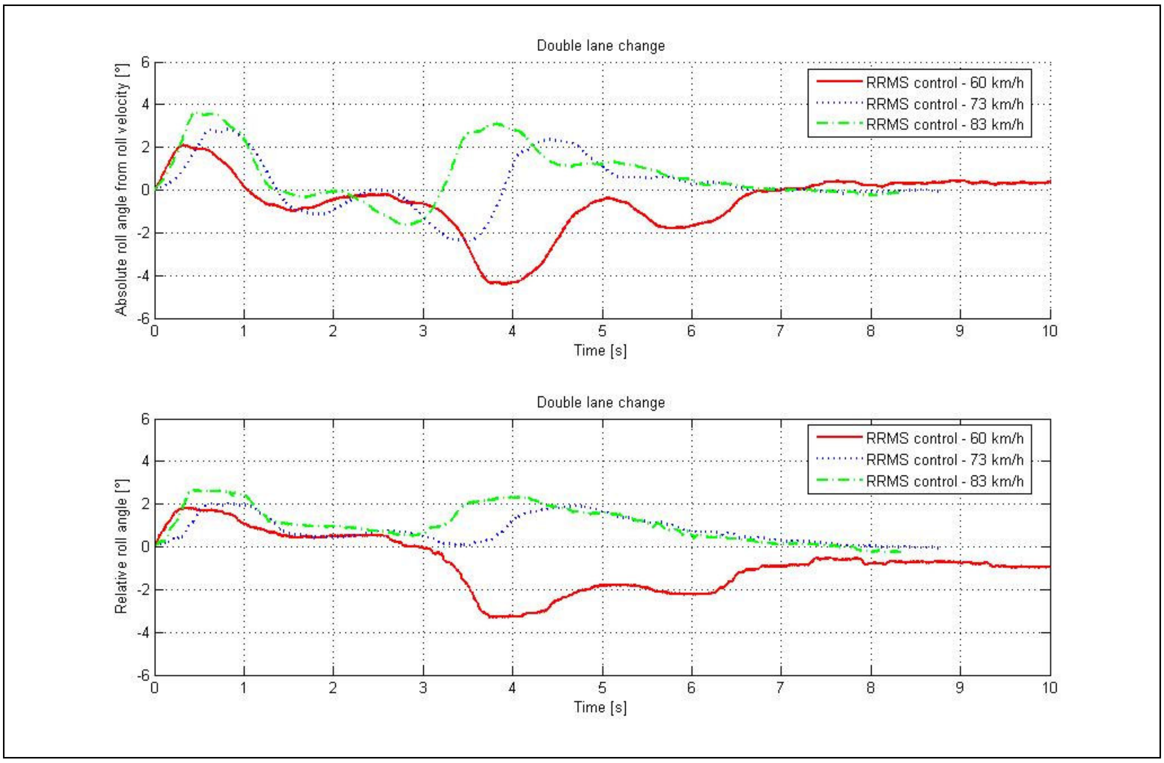


Figure 6.25 - Body roll for RRMS control at different speeds

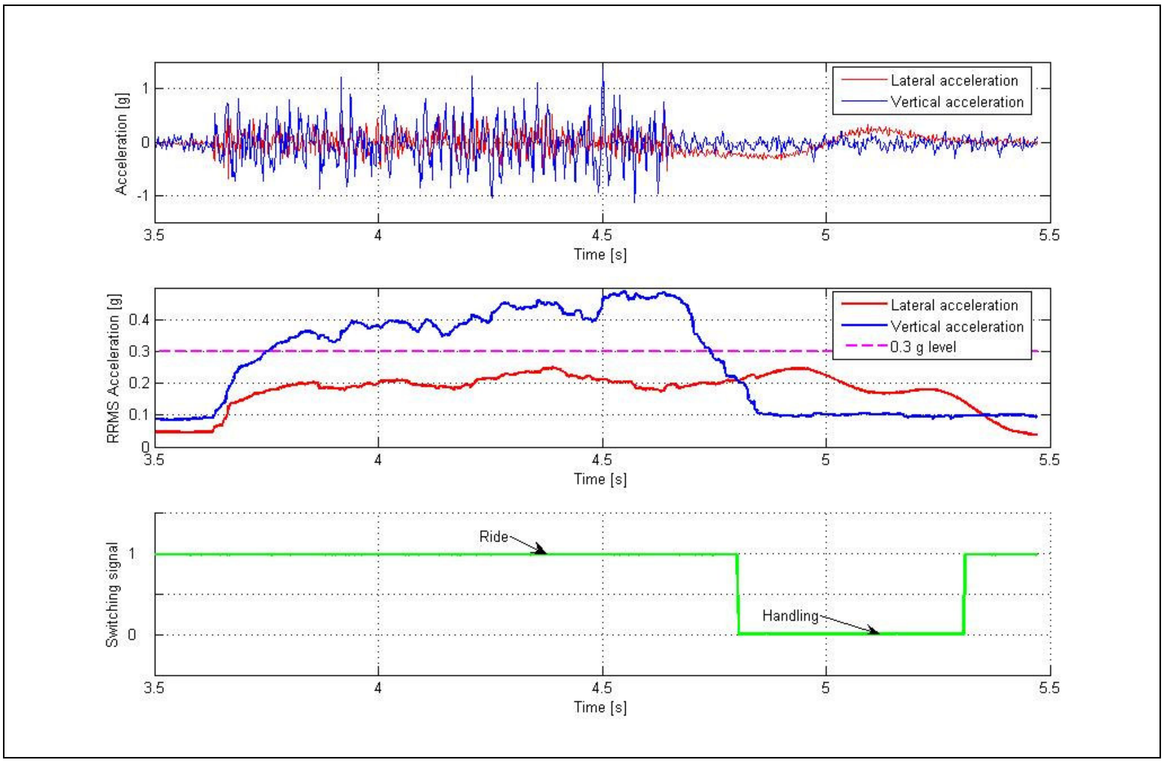


Figure 6.26 - RRMS control over Belgian paving at 74 km/h

6.6 Mountain pass driving

Performance of the RRMS strategy during mountain pass driving is shown in Figure 6.28. The RRMS control switches to "handling mode" whenever the RRMS lateral acceleration exceeds the vertical acceleration. Subjectively the vehicle feels very stable. The subjective improvement in ride comfort is considerable compared to "handling" mode.

6.7 City and highway driving

Results for city driving and highway driving are indicated in Figures 6.29 and 6.30 respectively. Switching to "handling mode" occurs rarely and only when cornering or changing lanes. Again subjectively the system performs as expected with a very noticeable improvement in ride comfort compared to "handling" mode, but also inspiring confidence when performing handling manoeuvres.

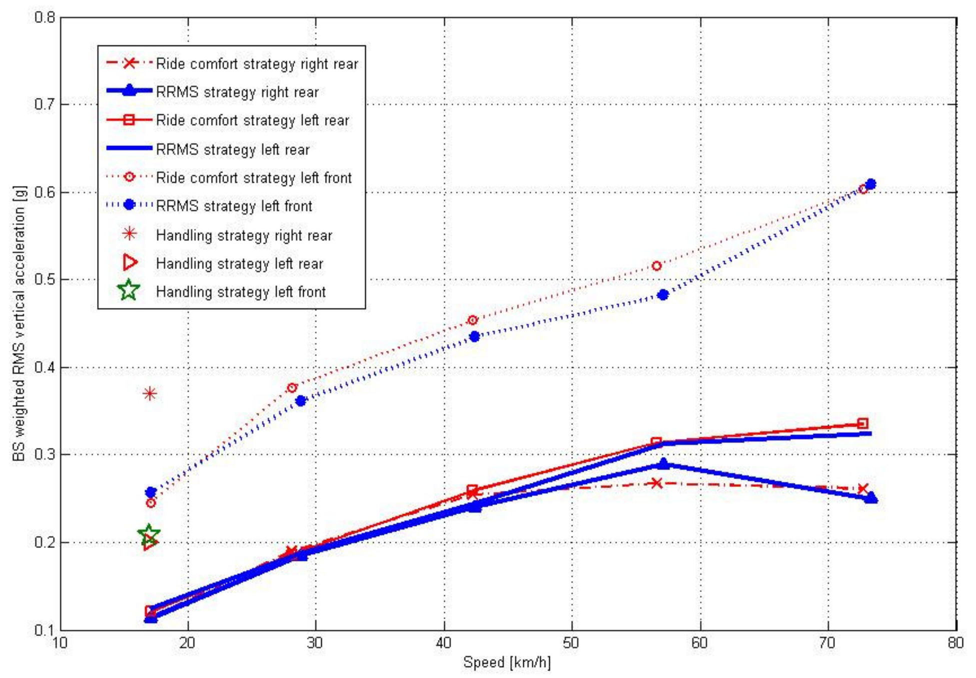


Figure 6.27 - Ride comfort of RRMS control compared to "ride mode"

6.8 Conclusions

The $4S_4$ suspension system performs according to expectations. Ride comfort in the "ride" setting, is a 50 to 80 % improvement over the "handling" setting. Body roll angle in the "handling" setting, is a 61 to 78 % improvement over the baseline vehicle and a 47 to 90 % improvement over the "ride comfort" setting.

The RRMS control strategy performs well under most circumstances, the only drawback being the time taken to switch to "handling" mode during the double lane change manoeuvre. Switching between "ride comfort mode" and "handling mode" occurs seamlessly, without the driver noticing the switching. The low damper characteristic is not sufficiently low enough to improve the ride comfort compared to the baseline vehicle.

The differences between "ride comfort mode" and "handling mode" are significant, illustrating that the principle works according to expectation.

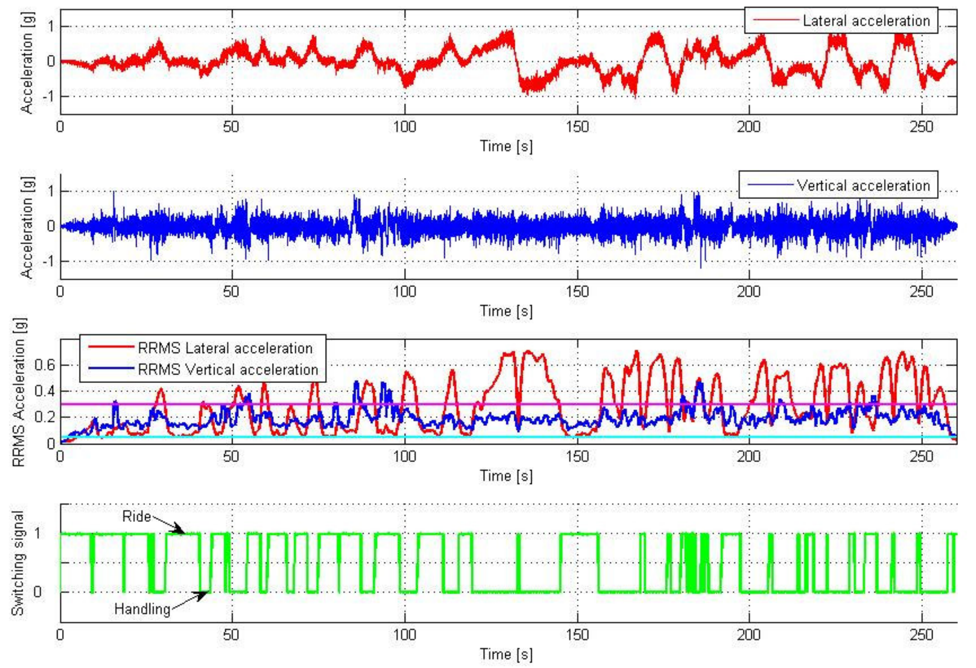


Figure 6.28 - RRMS control during mountain pass driving

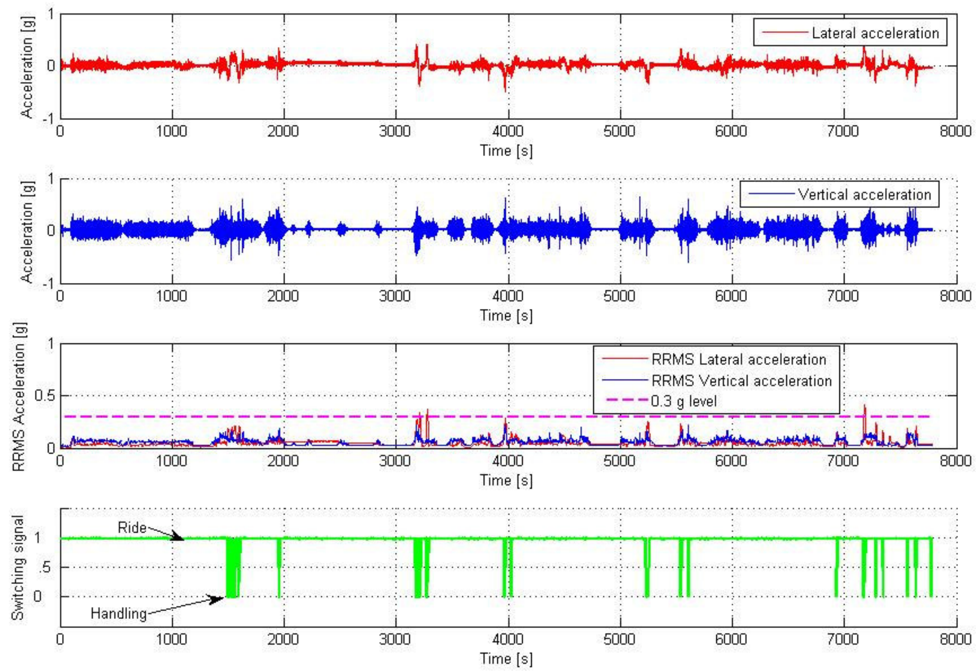


Figure 6.29 – City driving

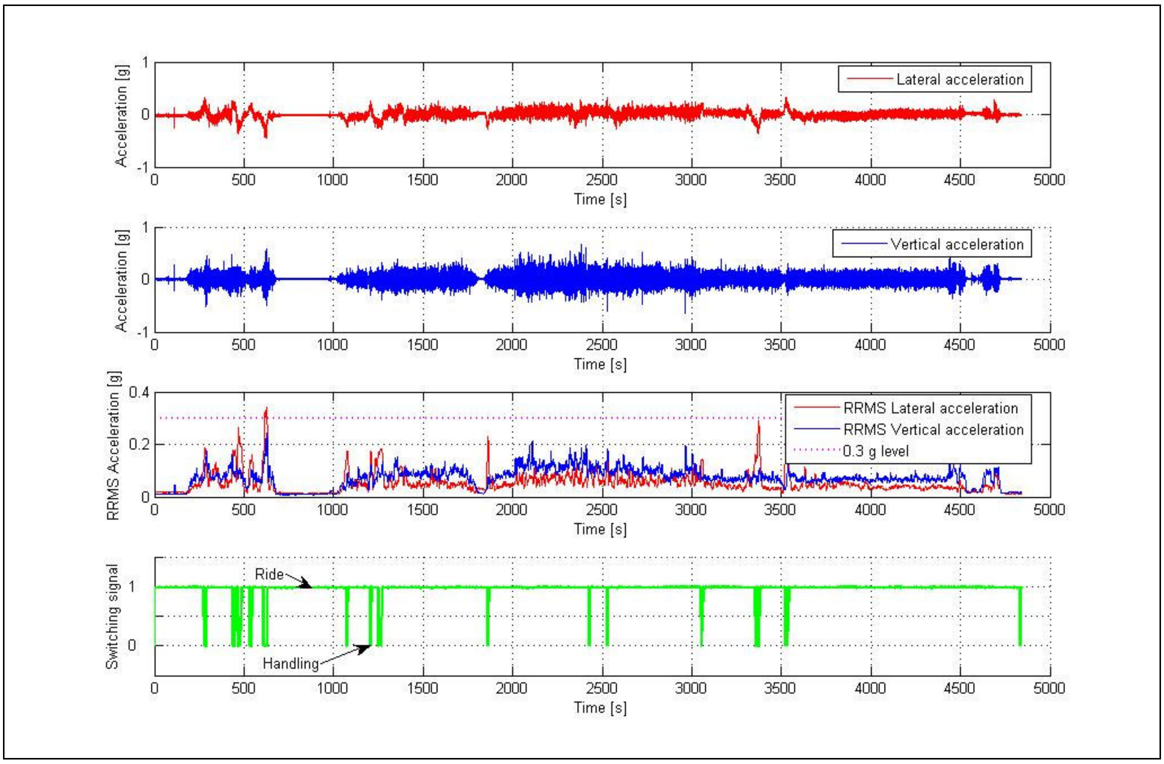


Figure 6.30 – Highway driving

Chapter 7

CONCLUSIONS AND RECOMMENDATIONS

7.1. Conclusions

Controllable suspension systems have been implemented successfully in top-end passenger cars and are regarded by industry specialists as the development trend of the future. Basic systems employ a "mode switch" where the driver manually selects a suspension setting *e.g.* "comfort" or "sport". More advanced systems react quicker and use some form of control to determine suspension settings.

Application of controllable suspension systems to vehicles that require good off-road capability (high ground clearance, large suspension travel and soft springs), but also good handling and stability on smooth roads at high speeds (low centre of gravity and stiff springs) are rare. Military wheeled vehicles, Sports utility vehicles (SUV's) and Crossover utility vehicles (CUV's) all fall within this category. This thesis attempts to fill this gap.

For off-road vehicles, a "mode switch" where the driver manually selects a suspension setting *e.g.* "off-road" or "on-road" can be used, but if the design in any case offers "ride comfort" and "handling" settings, automatic switching may just as well be employed to get the best possible benefit from the system. This also relieves the driver from making this decision. Furthermore, good handling is often required during off-road driving and good ride comfort is desirable when driving on bad roads. A successful "ride comfort" *vs.* "handling" decision can automatically select the required suspension settings according to the prevailing driving conditions. An important point worth noting is that current production systems still employ compromised characteristics, *i.e.* the "low" and "high" characteristics are often not optimised for ride comfort and handling respectively. The "low" setting is merely biased towards ride comfort but still results in acceptable handling. The "high" setting is biased towards handling, but still gives tolerable ride comfort. The suspension settings used in the present study are at the limits of the design space, i.e. the "low" setting gives the best possible ride comfort, but with unacceptable handling. The opposite holds for the "high" setting, *i.e.* best possible handling with intolerable ride comfort. This configuration results in large improvements in both ride comfort and handling respectively, but its successful application in vehicles rely on the "ride comfort *vs.* handling decision"

7.1.1 The ride comfort *vs*. handling compromise

Although no clear-cut answer is available for a metric that quantifies vehicle handling, the body roll angle was used in this research as an indication of handling.

The following hypotheses were made:

i) Ride comfort and handling have opposing requirements in terms of spring and damper characteristics.
ii) Suspension requirements for off-road use differ substantially from requirements for high-speed on-road use.
iii) A set of passive spring and damper characteristics, called the “ride comfort characteristic” can be obtained that results in excellent ride comfort over prescribed off-road terrains at prescribed speeds. Additional improvements may be possible by using “control”, but is not considered for the purposes of this research.
iv) A set of passive spring and damper characteristics, called the “handling characteristic”, can be obtained that results in excellent handling for prescribed high-speed maneuvers on good roads. Additional improvements may be possible by the use of “control” but is not considered for the purposes of this research.
v) Advanced suspension system hardware that can switch between the passive “ride comfort” and “handling” spring and damper characteristics, can be feasibly implemented. Response time must be rapid enough to enable control of the sprung mass natural frequencies.
vi) A robust decision can be made whether “ride comfort” or “handling” is required for the prevailing conditions.

A validated, non-linear full vehicle model was used to investigate the “optimal” characteristics for both ride comfort and handling. The conflicts between these requirements were investigated and analysed using simulation. The following conclusions are made based on the evidence presented:

i) A passive suspension system is a compromise between ride comfort and handling, as the respective requirements for ride comfort and handling are at opposite ends of the design space.

ii) To eliminate the “ride comfort *vs.* handling” compromise the following is required:
   a. At least two discrete spring characteristics are required namely:
      - A stiff spring for excellent handling.
      - A soft spring for excellent ride comfort.
   b. At least two discrete damper characteristics are required namely:
      - High damping for excellent handling.
      - Low damping for excellent ride comfort.
   c. The capability to rapidly switch between the two spring and two damper characteristics.
   d. A control strategy that can switch between “ride comfort” mode and “handling” mode in a safe and predictable way.

7.1.2 Possible solutions to the ride comfort *vs.* handling compromise

The solution proposed to solve the “ride comfort *vs.* handling” compromise, is to use a twin accumulator hydropneumatic spring (two-state) combined with a two-state (on-off) semi-active hydraulic damper. Although more than two spring and/or damper characteristics can be incorporated, two is considered sufficient based on the simulation results presented. The pre-requisite is however that a successful ride comfort *vs*. handling decision-making strategy can be developed that will switch automatically between the “ride comfort” and “handling” modes. This switching must be safe and quick enough to prevent accidents, but not disturbing to the driver.

Preliminary investigation indicates that further improvements in ride comfort using control techniques are unlikely, especially when the spring and damper characteristics have been determined by optimising for ride comfort.

The proposed solution to the “ride comfort *vs.* handling” compromise is the 4 State Semi-active Suspension System or $4S_4$.

7.1.3 The four-state semi-active suspension system ($4S_4$)

A possible solution was formulated and investigated in greater detail in Chapter 4 where the design, manufacturing, testing and mathematical modelling of the proposed prototype four-state semi-active hydropneumatic spring-damper system ($4S_4$) system was described.

The design meets all the initial design specifications and can be fitted to the proposed test vehicle with minor modifications to the test vehicle. The manufactured prototypes have been extensively tested and characterised. Although several problems were identified on the first prototype, these have been addressed and eliminated on the second prototype. Prototype 2 meets all the dynamic requirements, except that the low damping characteristic is too high to achieve the maximum ride comfort benefit.

A mathematical model of the suspension unit was developed and implemented in SIMULINK. Agreement between the model predictions and the measurements was generally good. Some aspects where the model or the quantification of its parameters needs improvement were identified. In particular, the tests to date clearly identified the need for a better method of quantifying the mass of gas loaded into the accumulators.

7.1.4 The ride comfort *vs*. handling decision

The crucial “ride comfort” *vs.* “handling” decision was investigated in chapter 5. Numerous tests were performed for different driving conditions and the data thoroughly analysed. Based on this analysis, different decision-making ideas were investigated. It is concluded that of all the proposed strategies, only the running RMS (RRMS) strategy appeared to work for all the test conditions.

A combination of strategies may also result in improvements, *e.g.* the steering angle can be used to determine the switching point from “ride comfort” to “handling”, but switching back to “ride comfort” might then be based on the running RMS, or simply delayed by a fixed time to eliminate spurious switching.

7.1.5 Vehicle implementation

The implementation of the proposed hardware and decision-making strategy in the vehicle, as well as final test results is discussed in Chapter 6.

The $4S_4$ suspension system performs according to expectations. Switching between "ride comfort mode" and "handling mode" occurs seamlessly without the driver being aware of the switching. Ride comfort with the "ride" setting is 50 to 80 % better than with the "handling" setting. The "ride comfort mode" does not present an improvement in ride comfort compared to the baseline vehicle, because the low damping characteristic on the $4S_4$ prototypes is too high. Body roll angle on the "handling" setting is improved by 61 to 78 % compared to the baseline vehicle and 47 to 90 % compared to the "ride comfort" setting.

The RRMS control strategy performs well under most circumstances, with the only drawback being the time taken to switch to "handling" mode during the double lane change manoeuvre.

7.1.6 Final comments

The proposed solution successfully eliminates the "ride comfort *vs*. handling" compromise when designing vehicles for both on- and off-road use. The $4S_4$ suspension system can be successfully implemented in hardware form, as this research has proven. The "ride comfort *vs*. handling" decision can be made using easily measurable parameters from freely available sensors.

7.2 Recommendations

Several recommendations to improve the system, and aspects that warrant further investigation have been identified.

7.2.1 The ride comfort *vs*. handling compromise

The handling study, presented in chapter 2, should be expanded to include more vehicles (especially off-road vehicles) and more drivers. This should enable better limits to be obtained.

For the present study, suspension characteristics for optimal ride comfort were obtained by simulating the vehicle driving over the Belgian paving at a speed of 60 km/h. Optimal characteristics for handling were obtained by performing a double lane change at 60 km/h on a smooth level road. The issue of combined ride comfort and handling was briefly investigated by performing the double lane change over the Belgian paving.

Before a final verdict can be reached with respect to the optimal suspension characteristics for ride comfort and handling respectively, it is necessary to investigate the effects of the following aspects in greater detail:

i. Different terrain roughnesses
ii. Different vehicle speeds
iii. Different handling manoeuvres

iv. Combined ride comfort and handling over a rough terrain *e.g.* performing the double lane change manoeuvre over the Belgian paving
v. More design variables such as the low speed and high speed damping characteristics, different compression and rebound characteristics, as well as the transition point between the low- and high speed characteristic.
vi. Effect of ride height
vii. Different vehicle loading conditions
viii. Improving vehicle handling compared to passive "handling" setting by applying control.

7.2.2 Possible solutions to the ride comfort *vs.* handling compromise

The effect of ride height, on the ride comfort and handling of the vehicle, should be investigated in more detail. Limited test results discussed in chapter 6 indicate that handling, with the soft suspension, can be considerably improved by lowering the ride height. A control strategy to change ride height, while the vehicle is moving, should be investigated.

7.2.3 The four-state semi-active suspension system ($4S_4$)

The current $4S_4$ system can be improved in several ways namely:
i. The "off " characteristic for the damper is currently too high and compares to the baseline damper value. This characteristic should be lowered significantly to between 20% and 50% of the baseline value before substantial improvements in ride comfort will be realized. This should be achievable by enlarging the ports and channels in the valve block or replacing the valve with a valve of larger flow capacity.
ii. The low-speed "on" characteristic for the damper needs to be increased.
iii. The gas charging procedure needs to be improved to ensure that the correct mass of gas is initially charged into the unit.
iv. Weight and cost should be reduced before the system can be commercially viable.
v. The $4S_4$ simulation model should be further verified to determine if the transient response, during valve opening and closing, is correctly simulated.

7.2.4 The ride comfort *vs*. handling decision

For further improvement of the "ride comfort *vs.* handling" decision, the use of artificial intelligence techniques (self organising maps, neural networks, fuzzy logic *etc*.) to identify the terrain and operating conditions is suggested. If the terrain or driving conditions can be successfully identified, then other concepts such as the steering angle *vs.* vehicle speed limit values can be implemented and thresholds adapted according to operating conditions. Possible reduction of the delay time caused by the length of the RRMS calculation, using additional information, should be investigated.

The SIMULINK model, comprising four of these units, should be incorporated into the ADAMS vehicle dynamics model of the sport utility vehicle in question. This will enable investigation of control strategies using simulation instead of vehicle testing.

The possibility of controlling vehicle over- and understeer by altering the front:rear roll stiffness balance should be investigated.

The capability of the $4S_4$ system to reduce rollover propensity has not been investigated. Suspension characteristics required to prevent rollover, and the effect of ride height and control, must be determined. Early rollover warning systems might be beneficial in this application because ride height and suspension characteristics can be adapted to operating conditions. It might for example be possible to reduce ride height rapidly by dumping oil in the reservoir and prevent rollover. Reduction in centre of gravity height of up to 150 mm may be achieved in this manner.

7.2.5 Vehicle implementation

Concerning implementation of the $4S_4$ system on a vehicle, the measuring position for the two accelerometers needs to be investigated. If the lateral accelerometer is mounted at the front, it might react earlier during a handling manoeuvre. The installation of a steering angle sensor should also be investigated.

7.2.6 Additional possibilities

Additional improvements may be possible using integrated chassis control, where the ABS braking system and automatic stability control is linked to the $4S_4$ suspension control. Not only can sensors be shared, but additional information can be used *e.g.* the system pressure in the $4S_4$ gives vertical wheel load (not true when bump or rebound stops are in contact). This could be used as input to the brake or stability control system to determine which wheels should be braked. This early warning could improve the performance of the other systems.

Installation of a higher capacity hydraulic pump could facilitate slow-active control, such as active body control or active anti-rollbars, without need for additional suspension hardware. The suspension system can then be used as an actuator or force generator instead of the current application as an adaptive element.

Reduced rollover propensity might require a third set of spring and damper characteristics or a different combination e.g. soft springs with high damping. The effect of front:rear stiffness balance has been indicated, but not used in the control yet. Switching spring and damper characteristics individually for each wheel might also have possible benefits in other driving scenarios that were not investigated.

Many other driving scenarios (other than the six investigated) should be investigated to ensure that the switching strategy works under all conditions, or otherwise adapt the strategy accordingly.