BACHELOR'S THESIS

Development of an auto rickshaw vehicle suspension

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Preface

The work reported in this thesis was performed at Luleå University of Technology (LTU) as a part of the education to Bachelor of Science in Automotive Engineering. The work was conducted in cooperation with a project group attending the SIRIUS course (a course in product development at LTU). The foundation of this thesis came up as the project group's assignment was to design a new hybrid auto rickshaw and thus a new suspension had to be developed. In addition to LTU an Indian company by the name TVS Motors was also involved by providing an auto rickshaw to the SIRIUS project group. The TVS King was a key factor in this thesis and thus we want to thank TVS Motors, especially the vice president Jabez Dhinagar, who provided useful information.

We want to thank our supervisor Magnus Karlberg that helped us through this thesis. We also want to thank Peter Jeppsson for helping us through the use of the softwares as well as guidance through difficult steps and last but not least we want to thank the project group for giving us the opportunity to create this thesis work in collaboration with them. The project group consisted of Daniel Cook, Anders Gustafsson, Ulrika Grönlund, Martin Isaksson, Christoffer Sveder, Axel Wallgren and Hanna Winterquist.

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Abstract

An auto rickshaw is a three wheeled motor vehicle with one front steering wheel. Auto rickshaws are most commonly found in developing countries as they are a very cheap form of transportation due to low price, low maintenance cost, and low operation costs.
Auto rickshaws needs to be developed to take the step into the 21-century. Therefore a project was started by Luleå University of Technology where a hybrid auto rickshaw concept was developed in collaboration with TVS Motor Company Ltd.; which is an Indian manufacturer of auto rickshaws.

To develop a new product in the modern world one of the most important challenges is safety of the users. As for the automotive industries this challenge has a great importance since the outcome can be devastating. One important category from the safety point of view is the vehicle suspensions, as the suspensions control the movement of the wheels and thus keeping the vehicle on the road.
An important factor in developing the new hybrid auto rickshaw was to improve its safety. One way of improving the safety is to improve the suspension which is addressed in this thesis work. Hence a development of the suspension was carried out to analyse if the negative handling characteristics typical for a three wheeled vehicle could be improved in the hybrid auto rickshaw.

To save time and money, modern test cycles of the dynamic characteristics are often conducted in software simulations. A commonly used software in the automotive industries is Adams/Car from MSC Software, which is based on multi-body dynamics and motion analysis. Another well used program is Siemens NX which is a CAE software which was mostly used in this thesis. This thesis covers the basics of these two softwares where systems were modelled and simulated to evaluate the suspensions.

The goal of this thesis work was to develop a vehicle suspension intended for an auto rickshaw. A variety of different suspension types were investigated and evaluated until two suspension types were chosen; one type for the front and one type for the rear. These suspension types were then simulated and tested in Siemens NX 8.0 in different critical scenarios to gain useful information. The information was then evaluated to draw a conclusion if the developed suspension obtained good performance.
The final solution was simulated and partly verified and still work remains to get a full overview of the performance. This thesis covers the first steps in the design process.

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1. Introduction

In spring 2011 LTU, Luleå Technical University, entered a collaboration with the Indian company Trust, Value and Service (TVS). In 1978 TVS group opened a new subdivision called TVS Motor Company Ltd. located in Hosur which were concentrated in manufacturing mopeds [1]. Since then TVS Motor Company have grown into the largest company in TVS group and the third largest manufacturer in India of two wheeled vehicles, like scooters and motorcycles. TVS group was founded in 1911 and has today over 30 companies and about 40 000 employees with a turnover of four billion dollar [2]. It was not until 2008 that TVS Motor Company officially introduced the auto rickshaw formally known as TVS King. Today they manufacture around 5000 auto rickshaws a month of which 4000 are exported and 1000 sold in India.

In this collaboration 22 students with different majors joined a project with the task to develop a fully working auto rickshaw prototype (a three wheeled vehicle) in hybrid version. This vehicle should have a new design, good road characteristics and a fully working powertrain. The powertrain was developed to run on both electricity and fuel, i.e. a hybrid vehicle. Environmental problems and increased fuel prices leads to solutions with lower fuel consumption.

As a reference, a TVS King was sent from India to LTU and from this vehicle the guidelines were set e.g. size, current suspension, height etc.

The task for the whole project was to develop a completely new auto rickshaw. This included a better suspension to improve the handling of the vehicle which is the main objective for this thesis work. The configuration of one wheel in the front and two wheels in the back on this type of vehicle results in challenges in handling characteristics particularly when braking and cornering.

The suspensions main objective is to decay vibrations, both longitudinal and lateral, to increase the vehicles life and also to make the ride comfortable. Suspensions are generally a border between wheel and body and therefore located between the wheels and chassi. The border typically consists of control arms, dampers, springs connected to each other, which together is called suspension.
The main objectives for this thesis work were to improve the tipping risk and the overall handling performance.

This is a bachelor thesis which means it includes 15 ECTS-credits and should be finished in 10 weeks. Due to this time limit some operation will have to be disregarded, the areas this thesis will cover are detailed design of the suspension, creating CAD models of the auto rickshaw suspension which will be used in simulations, calculations of parameters which have critical impact in the simulations and simulations of the vehicle’s performance.

2. Method

To retrieve insight in vehicle dynamics e.g. to understand the function and behaviour of different suspensions literature surveys were conducted.

Before concept generation a thorough benchmarking were conducted within the auto rickshaw-, car- and motorcycle- section. It was decided that a weight matrix should be used to evaluate the different front and rear suspension concepts.

Some equations, e.g. centre of mass were calculated in MATLAB. Calibrations of the coils springs were also performed with calculations in MATLAB as the spring stiffness coefficient depended on the centre of gravity's position and weight.

After the concept generation and evaluation a decision of which suspension solutions to continue with were made. The final solution consisted of one front suspension type and one rear suspension type.
To improve the existing auto rickshaw's dynamic and handling characteristics, Siemens NX were used to simulate how the model would act on a road. Siemens NX was chosen due to possibilities to create 3D models as well as performing simulations with rigid-body motion analysis combined with flexible bodies.

Before the final tests could be simulated calibrations had to be performed on the dampers so that they would obtain good suspension characteristics. The final tests supplied the essential information needed for the final evaluations of the auto rickshaw's performance.

3. Vehicle dynamics

Vehicle dynamics can be used to describe the behaviour/characteristics of a vehicle in motion. There are two expressions that are commonly used when discussing vehicle dynamics; those are kinematics and elastokinematics. Kinematics describes the wheel travel, according to DIN often also called wheel (or steering/suspension) geometry, the movement of the wheels during vertical suspension travel and steering. Elastokinematics defines the changes on the positions of the wheel caused by forces and moments between the tyres and the road. The changes arise from the elasticity in the suspension parts. [3]

Deutsches Institut für Normung (DIN) is a German standard which is generally accepted. DIN is a non-profitable association that offers stakeholders a development of standards as a service for the industry. The primary task is to develop consensus standards that meet the requirements for the market. DIN is acknowledged to be a national standard body that represents Germans interest in European and international standard organizations. Out of the standards that DIN introduces 90 percent are made for international usage. [4]

Society of Automobile Engineers (SAE) is an American standards development organization which publishes 2.500 technical papers, 9.000 technical standards and 125 books annually. The organization writes more automotive standards than any other standard-writing organization in the world. [5]

There are five fundamental kinematic properties that the suspension links need to control. The roll steer and roll centre height are controlled by the roll axis and the anti-squat, anti-lift and the wheel path are controlled by the side view instant axis. [6]

3.1 Vehicle axis system

The vehicle moves around an axis system, with x-, y- and z-axles, with its origin in a line perpendicular to the road through the CG (centre of gravity) of the whole vehicle and at the intersection of the vehicle roll axis.
A body in space is determined by six degrees of freedom (DOF), one rotation and one translation about each axis. Since the vehicle chassis can be modelled as a rigid body its rotations can be described [6]. The SAE vehicle axis system is shown in Figure 3.1.

3.1.1 Roll angle

Roll angle is the rotation around the x-axis; the x-axis is pointed in the longitudinal direction of the vehicle, in other words the driving direction of the vehicle. When the vehicle rolls the body leans as when cornering. [6]
The vehicles positive longitudinal direction is defined along the x-axis according to ISO 8855.

3.1.2 Pitch angle

Pitch is the rotation around the y-axis; the y-axis is transverse to the vehicle. When the vehicle brakes or accelerates there is a pitch change. [6]

3.1.3 Yaw angle

Yaw is the rotation around the z-axis; the z-axis is perpendicular to the road pointing down through the vehicle in the SAE vehicle axis system. [6]

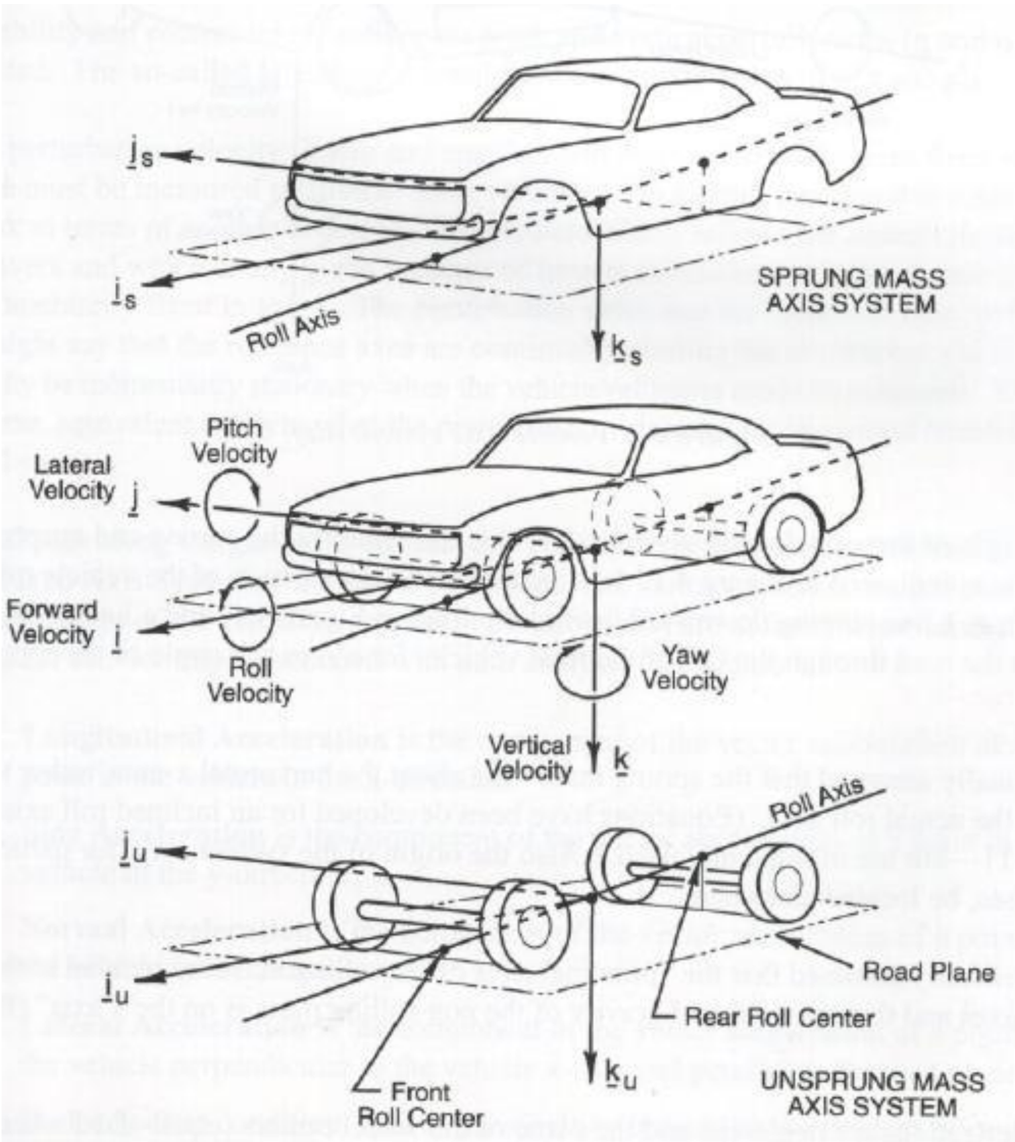

Figure 3.1: SAE vehicle axis system [7]

3.2 Lateral forces

Lateral forces are forces that act from the ground to the side of the vehicle, often at the contact patch.
Lateral loads are coupled through the roll axis.

3.3 Vertical forces

Vertical forces are forces that act, seen from behind, straight up on each wheel at the contact patch.
Vertical loads as acceleration and braking are coupled through the side view instant axis.

3.4 Toe angle δ_{vs}

According to standard DIN 70.000, seen from above, toe is the angle between the wheels centre plane and the centre line of the vehicle in longitudinal direction. Figure 3.2 shows the toe-in of a pair of wheels seen from above. Toe-in is when the front of the wheels are aimed inwards and toe-out (negative toe-in) is the opposite i.e. when the front aims outwards. Total toe-in is measured by the difference of front distance between right and left wheel and rear distance, often at the wheel centre height and described in millimetres. [3]
Toe-out on rear wheels with a front driven vehicle have the effect of making the vehicle less stable, this to make it easier to bend in corners and counter understeering. Disadvantage with toe in/out is the tire wear, which will be both larger and uneven, since the tires pointing direction are not the same as the vehicles. [8]

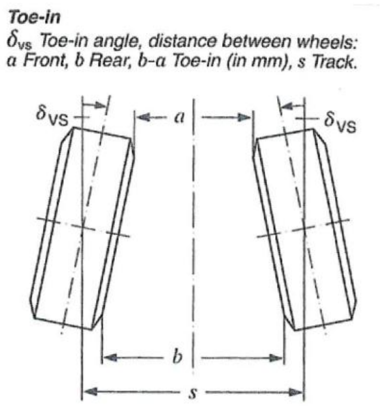

Figure 3.2: Toe-in seen from above [9]

3.5 Camber angle γ

Camber is the angle between the wheel centre plane and the vertical plane to the road, seen from the rear/front, according to DIN 70.000. Negative camber is when the top of the wheel leans inwards and positive when it leans outwards. Negative, zero and positive camber are shown in Figure 3.3. [3]
In the case of a loaded passenger vehicle a slight positive camber is preferred when entering a transverse-curved road surface, this to get even wear and a lower rolling resistance for the tires. Nowadays passenger vehicle uses negative camber to get better lateral tire grip and improved handling. [3]
Disadvantages of independent suspension are that when the body goes into a curve the outside wheel will gain positive camber and the lateral grip of this wheel will be reduced. To counteract this behaviour the suspensions are designed so that they will go into negative camber as they travel over a bump and positive when it rebound. [3]

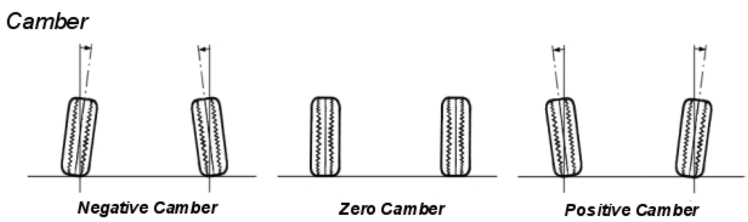

Figure 3.3: Camber angle seen from behind [10]

3.6 Camber force

Camber force is a lateral force in the direction of the tilt produced by the inclination (camber angle) of the wheel.
When the camber force occurs at zero slip angle, it is referred to as camber thrust. However there can also be lateral forces at other slip angles than zero [6]. The camber thrust affecting the wheel is shown in Figure 3.4.
The lateral force allows the vehicle to turn with a smaller radius than when there is no camber force [11].

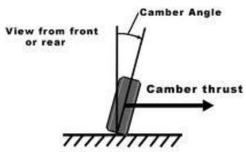

Figure 3.4: Camber thrust seen from behind [12]

3.7 Aligning torque

Aligning torque depends on camber angle and according to SAE aligning torque is the tires desire to steer about the vertical axis through the centre of the tires origin axis. Same effect as when the weather vane aligns to the wind direction. It is stabilizing in the linear direction and it tends to increase the slip angle. [6]

3.8 Caster angle τ

Caster angle, also known as steering axis angle or head angle on motorcycles. Seen from the side, it is the angle between the steering axis and the vertical plane; drawn through the centre of the wheel. The guidelines forming the caster angle is shown in Figure 3.5.
Positive caster angle is always preferred, and that is when the steering axis slopes in front of the wheel axle, this gives a self-centring effect of the wheel [3]. Negative caster angle will result in so called wheel wobble.
A steep steering axis, i.e. low angle, together with lot of rake or little trail will result in less effort to steer the wheel. A large caster angle together with high trail or little rake will give more resistance when steering. That is why a motorcycle with high head angle has a lot of rake to counteract this effect. [13]

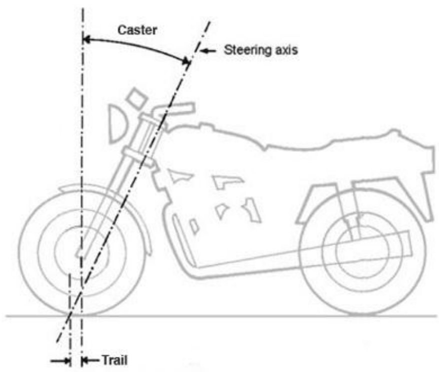

Figure 3.5: Caster angle on motorcycle [14]

3.9 Rake

Also known as offset, i.e. the distance between the wheels hub and the slope of the steering axis. The rake is shown in Figure 3.6: "Bicycle geometry". The offset together with the caster angle determines the size of the trail, the larger rake the more trail there will be. [13]
Rake angle on the other hand is the angle of the fork, i.e. the fork angle, which can differ from the steering axis angle, but seldom does because of the little positive or even negative trail that gives, which is undesirable as explained before. [13]

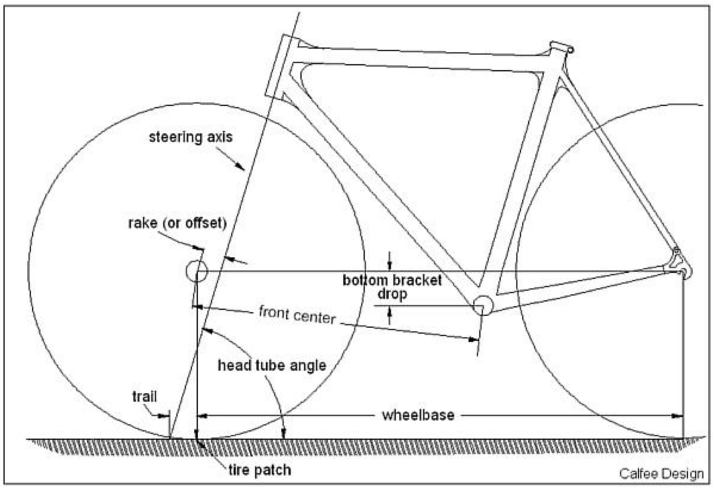

Figure 3.6: Bicycle geometry [13]

3.10 Trail

Ground trail is the distance between where the tire meets the ground and the point where the steering axis meets the ground, on the horizontal plane. The ground trail is shown in both Figure 3.6: "Bicycle geometry" and in Figure 3.7: "Motorcycle geometry". The primary function of trail is stability and also because the positive trail leads to a positive caster angle and therefore result in a self-centring effect. [15]
Real trail on the other hand is the distance between the steering axis measured at its slope angle and the wheels contact patch. The real trail is approximately 90 % of the ground trail when using normal rake angles. This trail decides the steering torque, more real trail results in less steering torque and less real trail will lead to higher steering torque. [15]

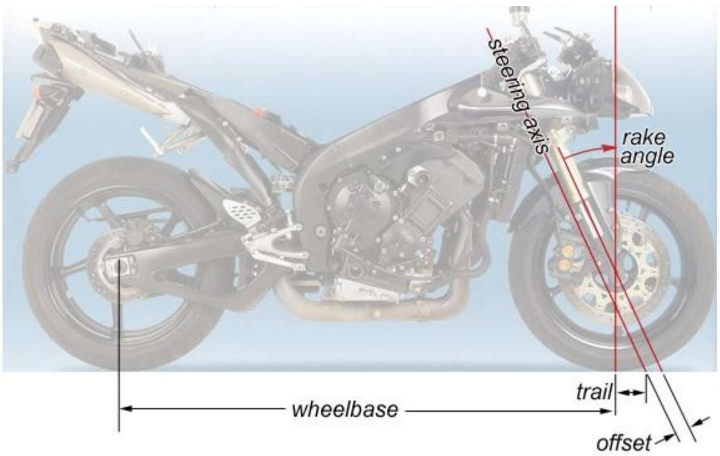

Figure 3.7: Motorcycle geometry [16]

3.11 Steer angle

At low speed, when no lateral forces are acting on the vehicle, it will corner precisely at the angle when the verticals are drawn starting at the middle of each wheel, perpendicular to the wheel, and meeting at one point i.e. the centre of the bend. [3]

3.12 Steering radius

Steering radius is an important aspect when it comes to low speed performance, especially when turning around the vehicle or during parking. The radius is determined by the wheelbase and the wheels turning angle. Shorter wheel base together with high steering angle gives a sharper turn and longer wheelbase together with low steering angle equals the opposite. The wheelbase for an auto rickshaw is the distance between the front wheel and the centre of the rear wheels. The turning angle is the same as for a motorcycle, due to the one steering wheel. [17]

The steering radius can be derived by drawing a perpendicular line from the rear wheels and the front wheel. Where the lines meet, is the middle point of the turning circle i.e. the distance will be the steering radius. The steering radius can be calculated theoretically with the same equation as for motorcycles. [17]

$r = \frac{w}{\delta \cos(\phi)}$ where *r* is the approximate radius, *w* is the wheelbase, δ is the steer angle, and ϕ is the caster angle of the steering axis [18].

3.13 Steering torque

Steering torque describes the load needed to steer the vehicle. The steering torque is affected by the caster angle, trail and rake. In the case of motorcycle the steering will be heavier when the caster angle is high together with either high trail or low rake. For easier steering a low caster angle together with high rake or low trail is needed.

3.14 Torque steer effect

An important criteria in handling characteristics is torque steer effect as it explains the force change in longitudinal direction while turning. The categories that can change the torque steer effect are longitudinal force changes, tire and ground adherence and of course the kinematic and elastokinematic chassis design. [3]

This effect often occurs during cornering when the vehicle has to slow down before turning, which result in centre of mass displacement and increases the load on the front axle and therefore reducing the rear axle load. If the longitudinal and lateral forces are sufficient and the increased transverse force due to the increase of normal forces from deceleration will result in a yawing moment that allows the vehicle to bend. If any of these forces is not sufficient the vehicle will either over- or under-steer. [3]

3.15 Slip angle

Also known as sideslip angle, tells how much the wheel has to turn before the tires contact patch twists. As long as the tire has traction the vehicle will move to the tires direction even if the wheel is not turned. [19]
The slip angle is the angle between the actual tire direction (the "footprint"/contact area of the tire) and the direction the wheel. The contact area (the grey lined area) and the slip angle (red field slice) are shown in Figure 3.8. The slip angle is caused by the tire distortion which leads to a lateral force and can thus make the vehicle corner. [20]

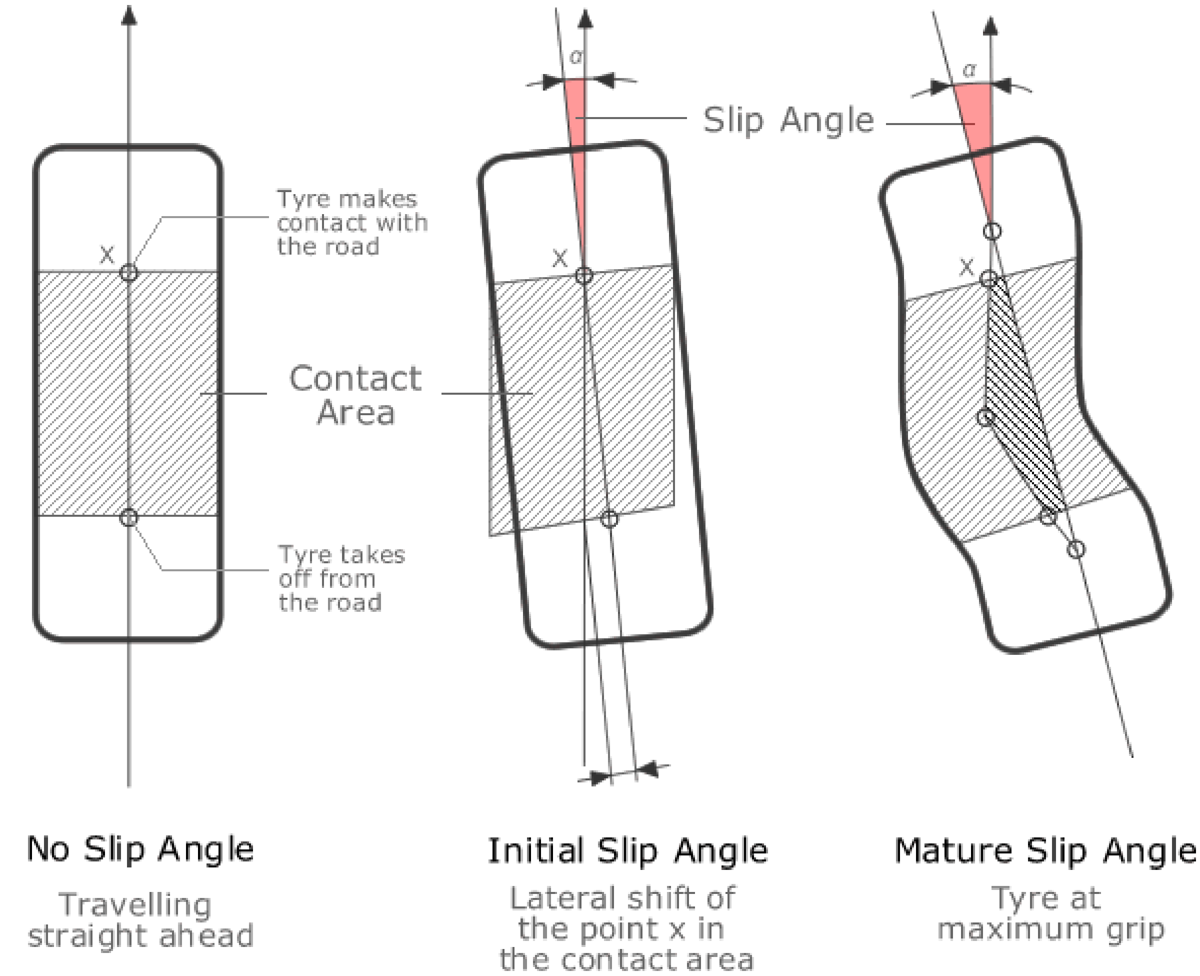

Figure 3.8: Slip angle on tyres seen from above [20]

3.16 Roll centre

To find the roll centre a line is projected from the centre of the tire-ground contact patch and through the instant centre (IC) as shown in Figure 3.9. This is done for both sides of the vehicle. The intersection of those two lines is the roll centre of the sprung mass of the vehicle. The roll centre location depends on the instant centre distance from the tyre, whether the instant centre is outboard or inboard of the tyre contact patch and the instant centre height below or above ground. [3]
In other words the roll centre is the centre point, seen from the rear, which the body will roll around when a lateral force are applied on the body [3].
The roll centre is the force coupling point between the sprung and unsprung masses.
A low roll centre height is always preferred, this is because the lower the roll centre height is the larger the rolling moment becomes around the roll centre. The moment, that the tire will generate from the lateral force, around the instant centre, will push the wheel down and lift the sprung mass. This happens when the roll centre is above ground. If the roll centre is below ground the moment will push the sprung mass down. [6]

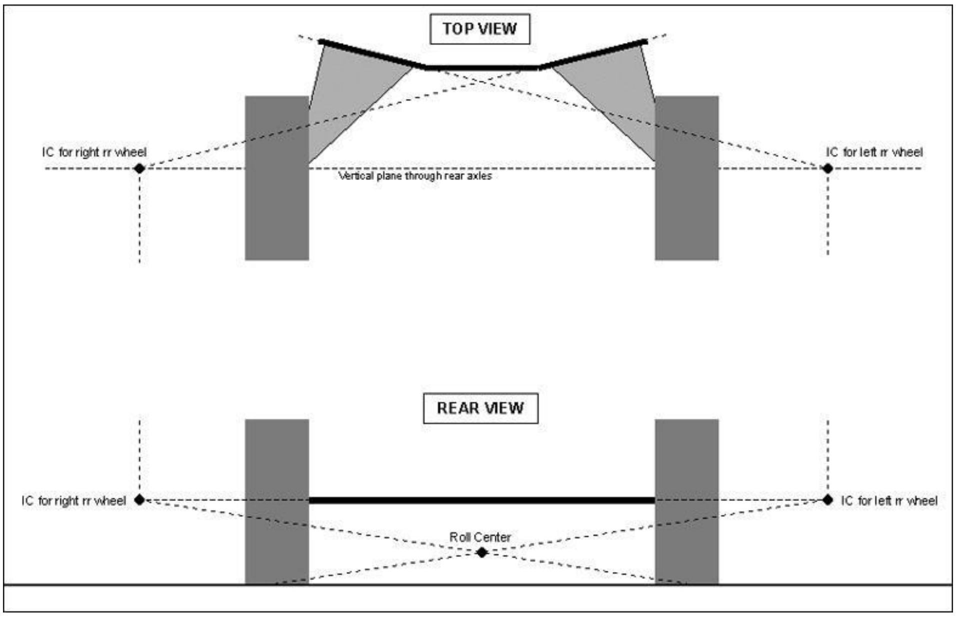

Figure 3.9: Roll centre construction [21]

3.17 Sprung/unsprung mass

Sprung mass is the mass of the vehicle that is supported by the suspension. Weights that are represented as the sprung mass are e.g. vehicle body, engine and passengers. [22]
The unsprung mass is the mass defined between the road and the suspension. E.g. wheels, brakes and parts of the suspension forms the unsprung mass. [22]
When driving over a bump the force acting on the unsprung mass will compress the suspension. In the rebound phase; when the suspension has to reach steady state, a low unsprung mass will allow the suspension to reach the steady state faster compared to a high unsprung mass that will require more time or higher force from the spring for the suspension to reach its steady state. Therefore a low unsprung mass is preferred as it provides better handling of the vehicle. [22]

4. Benchmarking

4.1 Auto rickshaws

Some common auto rickshaws on the market and their suspension solutions are discussed below.

4.1.1 TVS King

The first auto rickshaw developed by TVS Motor was named TVS King and was introduced in 2008, it was the first auto rickshaw model in India with a four stroke 200 cc engine.
The TVS King uses trailing link as front suspension and individual trailing arms as rear suspension and it has got constant rate coil springs with co-axial hydraulic dampers. [23]
The TVS King is shown Figure 4.1 and is the auto rickshaw that this thesis is based on.

Figure 4.1: TVS King [24]

4.1.2 Piaggio Ape

The first "auto rickshaw" was produced in 1948 by the Italian company Piaggio & C. SpA. The model was named Ape and they are produced in three different types; van, pick-up and rickshaw. Figure 4.2 shows the Piaggio Ape Callessino which is one of many versions. It has got a semi-monocoque frame and a steel body in the front that fits either one or two peoples, while the rear is built as a single load-bearing chassis in sheet metal. [25]
The rear suspension consists of trailing arms, spring coils and shock absorbers and in the front there is a leading link suspension [25].

Figure 4.2: Piaggio Ape Calessino [26]

4.1.3 Bajaj

Bajaj Auto is the fourth largest manufacturer of two- and three-wheelers in the world and the largest three wheel producer in India which makes them a rival to TVS. The Bajaj auto rickshaws are very common on the Indian market. One common version is the Bajaj RE which is shown in Figure 4.3. [27]
The first rickshaw built by Bajaj was licensed under Piaggio/Vespa and appeared on the market 1950. They used same mechanics as the Piaggio Ape but changed its appearance and even introduced a taxi version which Piaggio Ape did not have. [27]
This means that they still used the trailing arms together with shock absorbers and springs as rear suspension and leading link as front suspension [27].

Figure 4.3: Bajaj RE [28]

4.1.4 Mahindra & Mahindra

Mahindra & Mahindra is a company that doesn´t just aim for the commercialized vehicle but also to personal vehicle such as SUV:s, tractors, two-wheelers etc. The company was started in 1945 and is an underlying company of Mahindra group. [29]
The first three-wheeler to reach the market was the Alfa which was introduced in 2008. The Alfa is equipped with leading link, coil spring and hydraulic shock absorber as the front suspension. The rear suspension consists of independent trailing arms together with hydraulic shock absorbers and rubber springs. [30]

4.1.5 Tuk Tuk Forwarder Co., Ltd

This company is only aimed on rickshaws and was founded in 1993. They did not start their production until 1996 where they had a yearly capacity of 12.000 units. In 1994 the first two electrical Tuk Tuks were produced. [31]
Suspension wise this company goes in another direction than the rest as they use leaf springs and double shock absorbers together with a rigid axle for the rear suspension. In the front they use a telescopic fork with coil springs and dual shock absorbers. [31]

4.1.6 Monika Motors

Second largest company that specializes on auto rickshaws in Thailand, it's a part of the B.Grimm concern and was founded in 1883. First auto rickshaw came into production in 1960 and was named Monika L5. The L5 uses the same suspension combination as Tuk Tuk Forwarders auto rickshaws, i.e. a dual telescopic fork and a rigid rear axle with leaf springs. [32]

In other hand you can say that trailing arms are common as a rear suspension and trailing and leading link are common in the front. Those equipped with leading link has the advantage of anti-dive.

4.2 Front suspensions

The different front suspensions considered in this thesis are the common ones on motorcycles because of the one wheel in the front configuration on the auto rickshaw. Advantages and disadvantages, handling characteristics as well as the mechanics and how they work are discussed below.

4.2.1 Trailing link

Trailing link has got its pivot point in-front of the wheel axle which gives a diving effect when braking. Figure 4.4 shows a typical trailing link front suspension used on a Piaggio Vespa. Because the pivot is in-front of the wheel axle it will automatically have a large trail which makes the torque steering more resistant. The diving effect leads to a weight displacement. This is a drawback because it is preferred to have as small weight displacement to keep the handling characteristics close to constant. [33]
Since there is possibility to gain more trail on a trailing link a higher stability can be achieved when traveling with constant velocity, on a straight path compared to a leading link.
This type of suspension is the one used today on the TVS King.

Figure 4.4: Trailing link on a Piaggio Vespa [34]

4.2.2 Leading link

This kind of fork is often used on motorcycles with sidecars, which is because this suspension design requires less effort to steer the vehicle compared to telescopic forks and trailing link types. [35]
It suspends the wheel with a link behind the wheel axle; this is the main characteristic of this kind of fork. A side view of a leading link fork is shown in Figure 4.5. Advantages is the possibility to change the trail which adjust the torque steering resistance, this resistance is less than for the trailing link and thus appropriate for motorcycles with sidecars which can't lean in turns. Another advantage if fitted with floating callipers is that when breaking hard the force will be transferred via the rear vertical rod. I.e. the design includes anti-dive characteristics under heavy braking which gives only minor weight transfer and maintains normal suspension travel. [36]
One disadvantage is that when the suspension is operating some vertical forces is transferred to the steering head via the rear main fork tubes as the shock absorber can't absorb all the vertical forces.

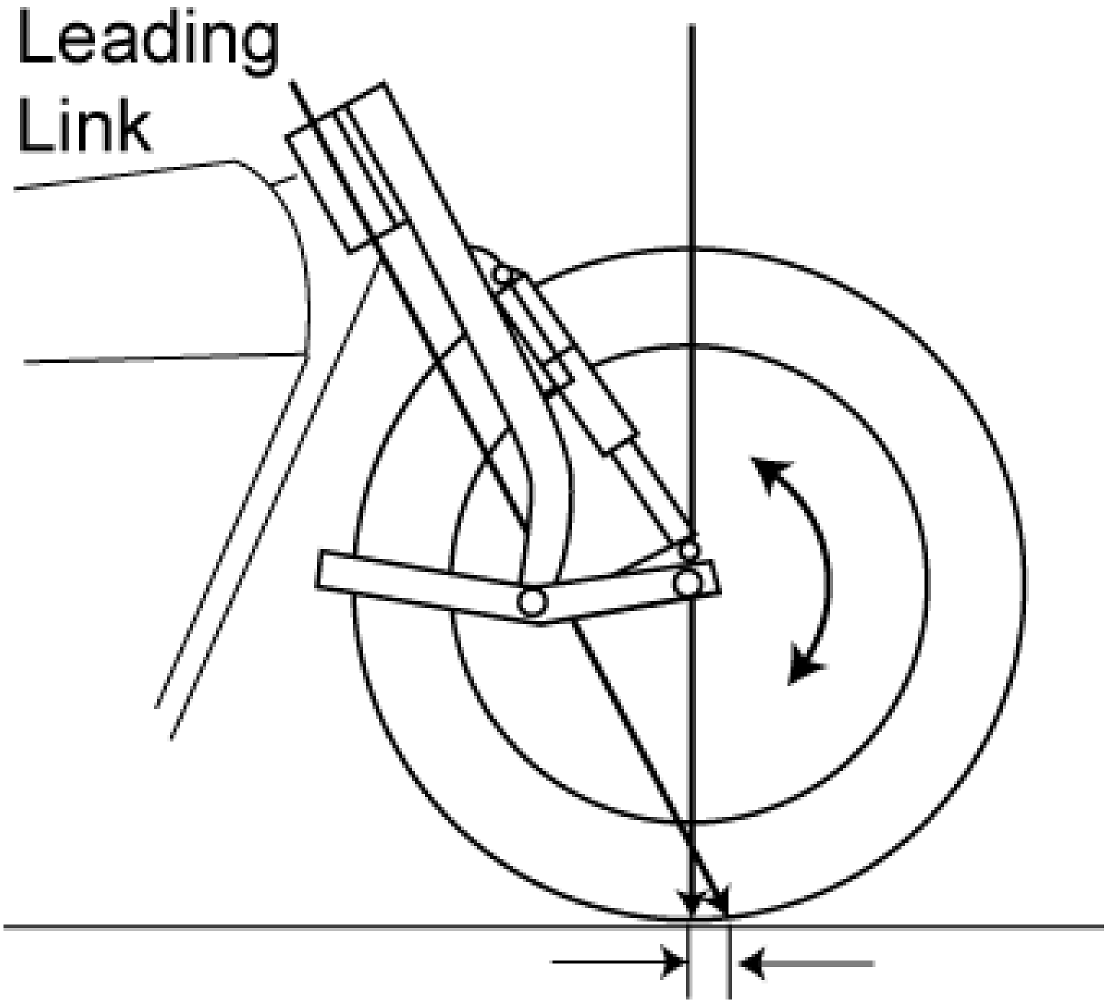

Figure 4.5: Leading link seen from the side [35]

4.2.3 Earles fork

The structure of this fork is triangular with its pivot point in the aft of the wheel axle which makes it a kind of leading link.
The patent drawing of the Earles fork describing the function and individual parts is shown in Figure 4.6. The fork splits up above the wheel where one rod (a) goes down behind the wheel and then connected with the horizontal rod (f) that goes from the wheel axle. From the split section a strut (g,i) is mounted and then connected to the horizontal rod (f) a bit aft the wheel axle, hence creating the pivot point. [37]
Advantages of this fork are the anti-dive characteristic and the ability to lift the wheel depending on angle of the strut. It also decreases the torque steering resistance compared to the trailing link. One disadvantage is like with leading link that some vertical forces are transferred to the steering head. [38]

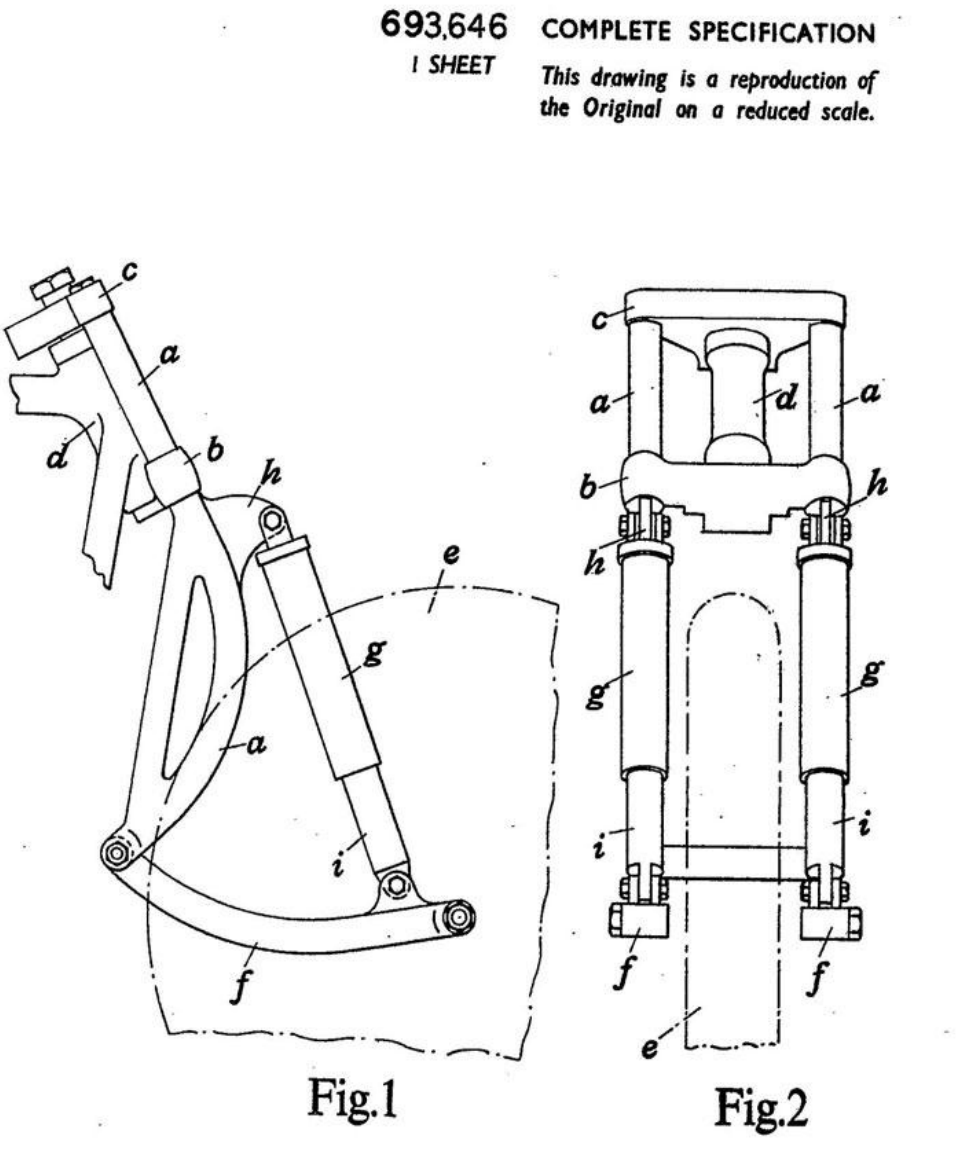

Figure 4.6: Earles fork patent drawing [37]

4.2.4 Telescopic fork

This kind of fork consists of a bar that is connected with the steering and the spindle. The damper and spring are built within the bar/hydraulic chamber. A picture describing the function of the telescopic fork is shown in Figure 4.7.
One problem with these forks is that if the bars are not parallel in the same vertical line the tire will start to wobble [38]. One other drawback with this design is the front ends tendency to dive when braking as the dampers compresses. This gives other undesirable handling effects such as head angle, trail and wheelbase change which can make the vehicle unstable. Telescopic forks are not suitable for higher rakes.
This is the most common used fork on motorcycles and mopeds today because of its simple design and low cost [39].

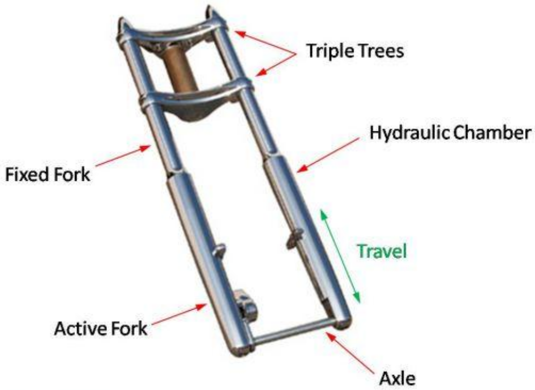

Figure 4.7: Telescopic fork [40]

4.2.5. Springer fork

The springer fork solution is an old design which consists of two bars on each side mounted parallel to each other; one bar is attached to a spring and the other to the steering [33]. Both bars; which are called the fixed fork respectively the active fork, are mounted to a kind of leading link at the bottom which is thereafter attached to the wheel axle. A springer fork and its primary parts are shown in Figure 4.8.
These sorts of forks are often without dampers and only depend on the main spring which makes the ride very bumpy [41].

Figure 4.8: Springer fork [40]

4.2.6 Girder fork

The girder fork consist of two beams, one on each side, then there is a transverse bottom triple clamp to hold them together, where the strut is mounted on and then connected to the triple tree which holds the steering rod. These are commonly mistaken for Springer forks because of the high mounted strut. They can be distinguished by the pivot being at its bottom triple clamp instead of behind the wheel axle as for the Springer forks. [33]
The Girder also has a shock absorber mounted together with the spring in contradiction to the Springer fork, which make the ride less bumpy [42]. Figure 4.9 shows the Girder fork and the function of its substantial parts.

Figure 4.9: Girder fork [40]

4.2.7 Hossack/Fior fork

The Hossack/Fior fork consist of two wishbones, a upper wishbone that connects the steering linkage to the body and a second one to keep the construction upright as shown in Figure 4.10. The pivot point of these two wishbones roughly has to be on the same vertical axis. The upright wishbone has the same task as a race car with a similar suspension set up. In the Hossack design the axle is rotated through 90 degrees and over hung. The steering link is designed in such a way that the handlebar pivot point carries none of the suspension loading and only has to handle the weight of the riders' upper half. [43]
The main advantages of this design are the handling capabilities due to the possibility of keeping the head angle, trail and wheel base constant [43].

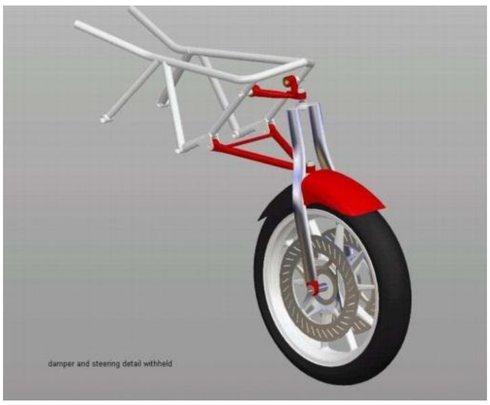

Figure 4.10: The Hossack design under normal running [44]

4.3 Rear suspensions

4.3.1 Independent rear suspensions

Independent suspensions are suspensions where the two wheel aren´t connected to each other.

Advantages of independent wheel suspension:

- Lower weight than dependent suspensions
- Smaller space requirement
- Kinematic/elastokinematic toe-in change which might lead to understeering possibilities
- Easier steerability
- No mutual wheel influence

4.3.1.1 Trailing arms

Trailing arms are often used as a rear suspension, and consists of a control arm connected between the wheel spindle/hub and chassis [45]. The arm lies longitudinally in the driving direction and has a triangular form with the smaller part by the wheel (to minimize the camber/toe change when cornering). Figure 4.11 shows a schematic view of the trailing arm from above.
The control arm has to withstand forces in all directions, and is therefore highly subjected to bending and torsional stress. If the arm withstands the forces no camber and toe change will be caused by vertical and lateral forces. [3]
Disadvantages with this suspension is that it´s designed for vertical movements only, when cornering it will roll together with the body and thus increasing understeering. The force on the control arm is also substantial as mentioned before and will then inevitable deform with unwanted positive camber as a result when cornering. There are also very small possibilities of a kinematic and elastokinematic effect on the position of the wheels. [3]
Advantages are the possibility of allowing for a flat floor and if the pivot axles lies parallel to the floor, the bump and rebound travel wheels undergo no track width, camber or toe change, and the wheel base only shortens a little. It is also cheap and it gives a smooth ride when travelling in a straight path. [3][6]

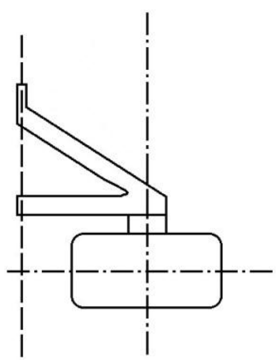

Figure 4.11: Trailing arm seen from above

4.3.1.2 Semi-trailing arms

Semi-trailing arm suspensions has similar construction as trailing arms, with the difference that the triangular arm has two skewed points to pivot around. When viewed from the top; as in Figure 4.12, the two pivot points forms a line that is somewhere between parallel and perpendicular to the vehicles centre line or point of travel [45]. That offset is often between 10 to 25 degrees and will decrease the vehicles tendency to understeer, semi-trailing arms have an elastokinematic tendency to oversteering. Camber and toe angle changes increase the higher the offset angle is. [3]

One disadvantage is the undesirable change of camber and toe angle when moving between bump and rebound because of the angled pivot points. Semi-trailing has a linear-lined camber change and a curved-lined toe change, the opposite of what you desire in a good suspension. [6]

The advantages are that it's compact, takes little space and can allow for a relatively flat floor.

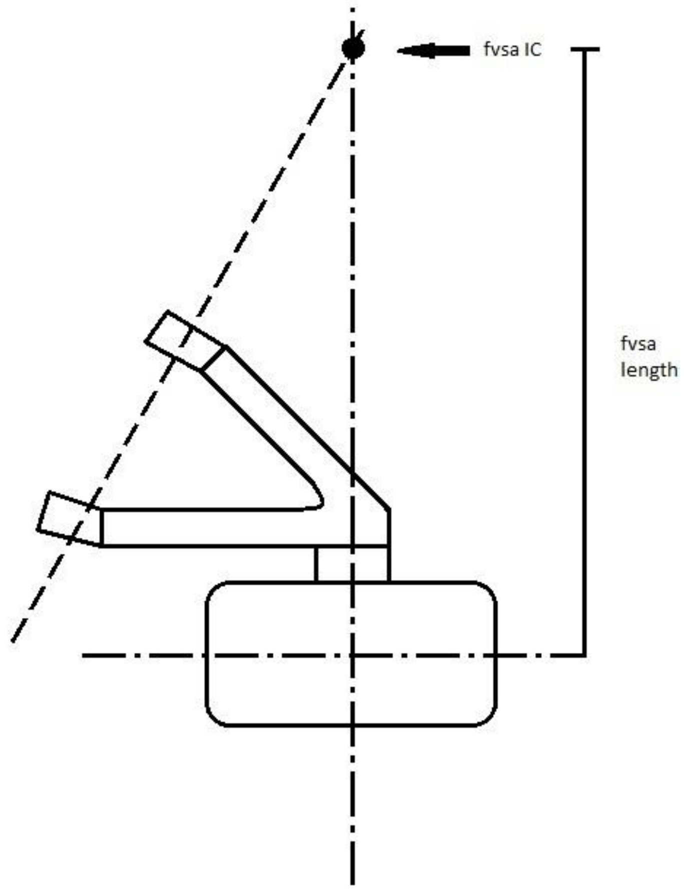

Figure 4.12: Semi-trailing arm seen from above

4.3.1.3 Double wishbone

Double wishbones are also referred to as double-A arm suspension and consist of two (occasionally parallel) control arms on each side of the vehicle, which are mounted by the two mounting points to the suspension subframe, frame or body. The wheel is mounted on the spindle, upright carrier or hub which then is mounted between the lower and upper A-shaped control arms. This structure allows the wheel to travel vertically up and down. [46]
The control arms often have the shape of an A, but can also be L-shaped or even a single bar linkage. L-shaped arms are often preferred on passenger vehicles as it usually gives a better compromise of handling and comfort capabilities. The shock absorber is mounted to the wishbones to control vertical movement and bushings or ball joints are mounted to the joints to resist fore-aft loads during acceleration and braking. The different bushings or ball joints can be at an angle from the horizontal axes or the vehicle centre line and thus anti-dive and anti-squat geometry can be built in. [3]
A double wishbone suspension with A-arms, hub and damper is shown in Figure 4.13.
Often the two control arms are of unequal length, a so called SLA (short and long arm) suspension, this to induce negative camber as the suspension jounces (rises).
These are very common today, especially on sport and racing cars, but as well on family cars particularly as a front suspension.
The main advantages are the kinematic possibilities. It is easy to carefully control the motion through bumps with good rejounce, and controlling parameters as camber, caster, toe angle, roll centre height, scrub radius and more. It is also easy to count the forces the different parts will be subjected to which allows more lightweight parts to be chosen. The design further has good load-handling capabilities. [3]
Disadvantages are the complex design, weight and that it demands longer service time than equivalent McPherson struts because of the increased number of components [3].

Figure 4.13: Double wishbone suspension [46]

4.3.1.4 McPherson strut

McPherson is a model where the damper and spring, also called strut, holds the spindle to the chassis. A 3D-model of a McPherson strut including the hub and control arm is shown in Figure 4.14. The strut's connection to the chassi is its pivot point which hence absorbs all the forces leading to bending stress on the strut. The long arm leads to a larger radius which makes it difficult for camber change. The design requires little space and is also easy to produce since it can be combined into one assembly. [3]
The McPherson strut is a common front suspension on modern cars, but not as commonly used as a rear suspension because of its space requirements upwards.

Other advantages are:

- Long spring travel.
- More space to fit engine compartment due to less space requirements.
- Easy to fit transverse engine.
- Simple and cheap to manufacture.

Disadvantages are:

- Force and vibrations that are transferred into the wheel arch which is a rather elastic area of the front end of a vehicle.
- More difficult to reduce road noise.
- Friction that is generated between piston rod and guide which decrease the springing effect.
- Greater height clearance requirements.

Figure 4.14: McPherson strut [46]

4.3.1.5 Chapman strut

The Chapman strut suspension is similar to McPherson, and is only used as a rear suspension. Figure 4.15 shows the patent picture of the Chapman strut. The upper strut (20) is still mounted to the chassis (19) but the lower one (13) is shrunk into a socket (12) on the hub-casing (9). The hub is then attached to a radius arm (10) which is horizontal and on the same plane as the half shaft. These can be changed to alter toe. The Chapman strut is designed to be angled so it can absorb more lateral force. [47]

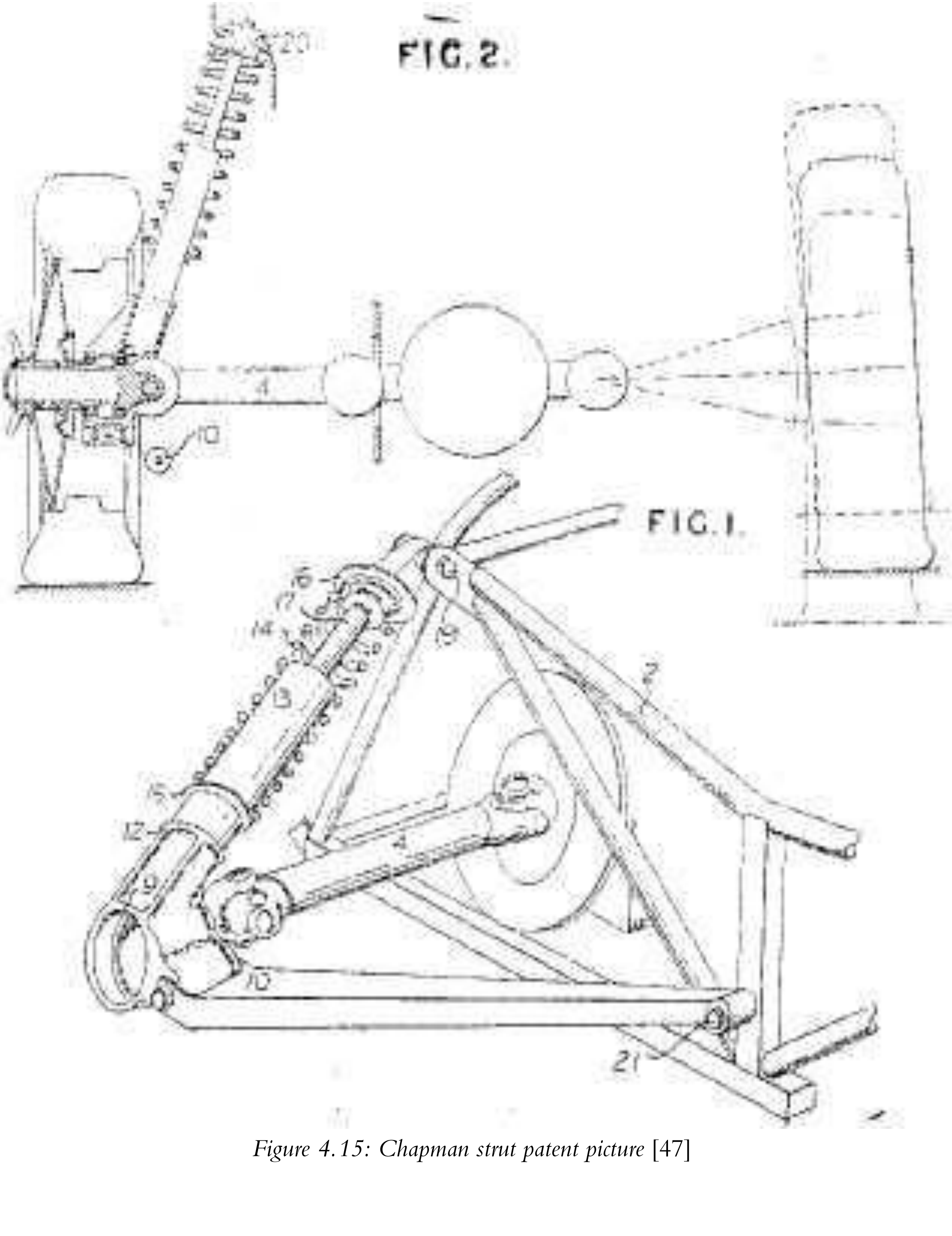

Figure 4.15: Chapman strut patent picture [47]

4.3.1.6 Multi-link

Multi-link is a kind of rear suspension that consist of trailing arms, one on each side, and two or more transverse control arms. One type of multi-link suspension including hub, spring and damper is shown in Figure 4.16.
Main objective of the trailing arms is to hold the wheel hub and when cornering allow the rear wheels to change toe to easier follow the curve. They also serve as borders for longitudinal forces/moment under braking and transferring them to the body via bushes. While the lateral forces that acts on the tires are transferred to the body via the subframe which is fastened in the middle via four bushes. The lateral forces are transferred to the subframe via the transverse control arms. The larger transverse axle carries both the spring and the joints for the anti-roll bar and it is on these control arms the majority of the longitudinal forces are transferred between axle and body. The control arms are positioned at an angle that together with the trailing arm bushes creates a desired elastokinematic characteristic. [3]

Advantages:

- Good kinematic and elastokinematic characteristics.
    - Elastokinematics characteristics:
        - Toe in under braking forces.
        - Lateral force compliance when understeering.
        - Prevent the effects of torque steer.
        - Stability when running straight and changing lanes.

Figure 4.16: Multi-link suspension [46]

4.3.1.7 Swing axle

Swing axle is an old and simple independent suspension. A single swing wishbone is mounted in the middle of the vehicle stretching out to where the wheels are mounted perpendicular to the wishbone. Swing axle suspensions traditionally use leaf springs and shock absorbers.
The swing axle has few advantages such as reduced unsprung weight and that it provides independent shock absorption compared to the live axle with enhanced ride comfort.
Among the disadvantages are severe toe-in and large camber change during bound and rebound and when body roll occurs. Especially body roll makes both wheels lean towards the corner resulting in heavy oversteering. Jacking, due to its high roll centre, which occurs on rebound causes positive camber change on both wheel which leads to low grip and vastly impaired handling. [48]

Figure 4.17 shows the movement of a swing axle configuration seen from behind; the upper case is when jacking occurs, the middle picture is under normal conditions, and the bottom picture shows body roll which occurs when cornering.
Very seldom used today mainly because of the poor handling [49].

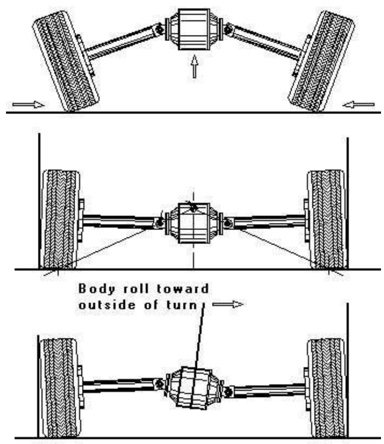

Figure 4.17: Swing axle seen from behind [49]

4.3.1.8 Torsion bar

A torsion bar suspension, also known as a torsion spring suspension, is a simple design consisting of a twistable bar mounted firmly to the vehicles chassis. Figure 4.18 shows two torsion bars.
At the end of the torsion bar a lever called the torsion key is mounted perpendicular to the bar. The torsion key is attached to a suspension arm, a spindle or the axle. When the wheel travels vertically the bar twists around its axis and is resisted by the bars torsion resistance. The effective spring rate of the bar is determined by its shape, length, material and cross section. The torsion bar or torsion key can easily be changed to adjust the ride height of the vehicle by either turning the adjuster bolts on the torsion key or by changing the torsion bar for different spring rate. [50]
Torsion bars are common on older cars and heavier vehicles as trucks and pickups today.
The main advantages is that it requires little space to mount and can be mounted both transversely and longitudinally, its durability and easy adjustability of ride height.
Some disadvantages are that torsion bars usually cannot provide progressive spring rate which leads to a stiff behaviour when unloaded and that it has poor handling. [50]

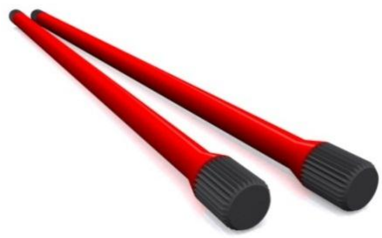

Figure 4.18: Torsion bars [46]

4.3.2 Non independent rear suspensions

Suspensions where the left side is influenced by the right side or the other way around. It is often a axle that connects the wheels together e.g. crosses the vehicle from one side to the other.

Disadvantages:

- Needs a lot of space.
- Lowers the ground clearance.
- Heavier then independent suspensions.
- The axle can be set in oscillation if one wheel hits a bump, this could amplify if a series of irregularities are hit.

Below is a list of different kinds of dependent axles.

4.3.2.1 Twist-beam

The twist-beam is also called a torsion beam suspension and shares some design with the trailing arm and the torsion bar. It combines features from trailing arms, torsion bar, and a sway bar. A patent drawing showing one type of twist beam including its different sections (number in the drawing) is shown in Figure 4.19. It has two trailing arms (2) which are connected transversely by a laterally mounted torsion beam (9). The two trailing arms (2) are welded to a twistable cross-member (9) and fixed to the body via trailing links (5). The twist beam absorbs lateral and vertical forces and acts as a sway bar that reduce body roll as the vehicle body leans in turns.
There are three basic types of torsion beams, with the main difference where the fore-aft location of the beam is.

The torsion beam could be described as the new rear axle design of the 1970's and is still today commonly used on small to medium front wheel driven cars because of its simple and space saving design [6].
Advantages are the price, easy to manufacture, it needs little space, the whole axle is easy to dismantle and assemble, the spring and damper are easy to fit, no need for external control arms and rods and thus only a few components to handle. There is also low unsprung weight, the beam can simultaneously work as an anti-roll bar and there is negligible camber, toe and track width change, low load dependent body roll understeering and low tail-lift during braking. [3]
Twist-beams have similar disadvantage problems as the trailing arm such as a tendency to oversteer due to control arm deformation and restricted possibilities for positioning the wheel and thus limiting the kinematic and elastokinematic qualities. Other disadvantages are shear and torsion stress in the cross-member and thus high stress in the welds which limits the rear axle load, the wheels are mutually affected, vibrations and noise are easily transferred to the body and the bodywork need to be the strong in the regions where the front bearings is connected to the body. [3]

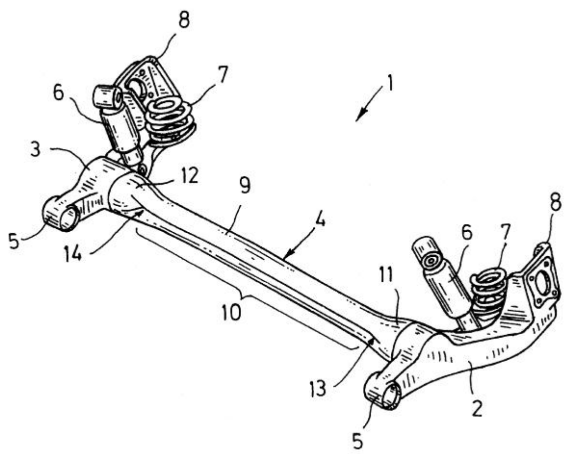

Figure 4.19: Twist beam axle patent picture [51]

4.3.2.2 Rigid axle

A rigid axle, sometimes called a solid, beam or live axle, is a transverse axle with mutual wheel influence. They are common on vintage cars and commercial vehicles as delivery vehicles and lorries. On heavier vehicles it is often attached to the chassis by longitudinal leaf springs which is both supporting and springing as damping and is good for absorbing forces in all directions but that solution takes a lot of space. A solid axle with leaf springs is shown in Figure 4.20. A commonly used rigid axle on front wheel driven vehicles is shown in Figure 4.21; which in this picture is a beam axle including a track bar to prevent side to side movement.
Some advantages are that it´s very simple and economical to manufacture, there is no wheel camber change when the body rolls during cornering and no changes to camber, toe and track width on full bump/rebound travel. This gives low tyre wear and optimal force transfer due to large spring track width. [3]
Disadvantages are the space requirement above the beam corresponding to the spring bump travel, mutual wheel influence and limited potential for kinematic and elastokinematic fine-tuning [3].
Because of those disadvantages today there are mostly heavier vehicles using this suspension designs where the disadvantages can be accepted [3].

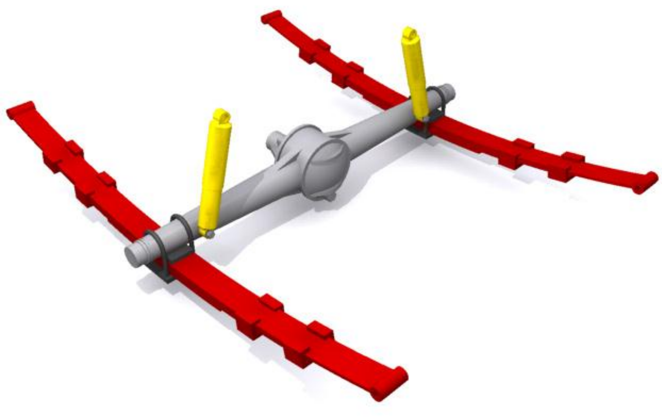

Figure 4.20: Solid axle with leaf springs [46]

Figure 4.21: Beam axle with track bar (also called panhard rod) to prevent side-to-side movement [46]

4.3.2.3 De Dion tube

The De Dion tube is a semi-independent rear suspension which is an improvement of the rigid axle and the swing axle with less unsprung weight and better ride quality. The design reminds of a rigid axle together with a fully independent trailing arm suspension. A type of De Dion tube is shown in Figure 4.22. A solid tubular beam holds the opposite wheels in parallel, but unlike a sway bar the tube is not directly attached to the chassis and without the torsional flexibility of a twist-beam suspension. The tube can move laterally and allows the wheel track to vary, this is necessary as the wheels are always kept parallel to each other, thus perpendicular to the ground during bound and rebound travel. In other words there are no camber change which is an advantage. There is a variety where the beam is non-telescopic, usually with trailing links and an A-bar for lateral location. [46]
There are disadvantages as it is a bit complex and heavy and therefore rarely used today. [46]

Figure 4.22: De Dion tube [46]

4.4 Anti-roll bar

Also called sway bar, which task is to counteract roll when a vehicle corners. As shown in Figure 4.23 the anti-roll bar is formed as a U with the endpoints mounted to each side of the suspension. The middle part is attached to the chassis with two bushings, often at its pivot point.
When cornering to the right the vehicle will roll to the left i.e. the left side wants to lift and this makes the anti-roll bar twist, to counteract it will transfer the same movement to the other side. Since it is impossible for the left side to lift the right side, the right side will instead push the left side back down. [46]

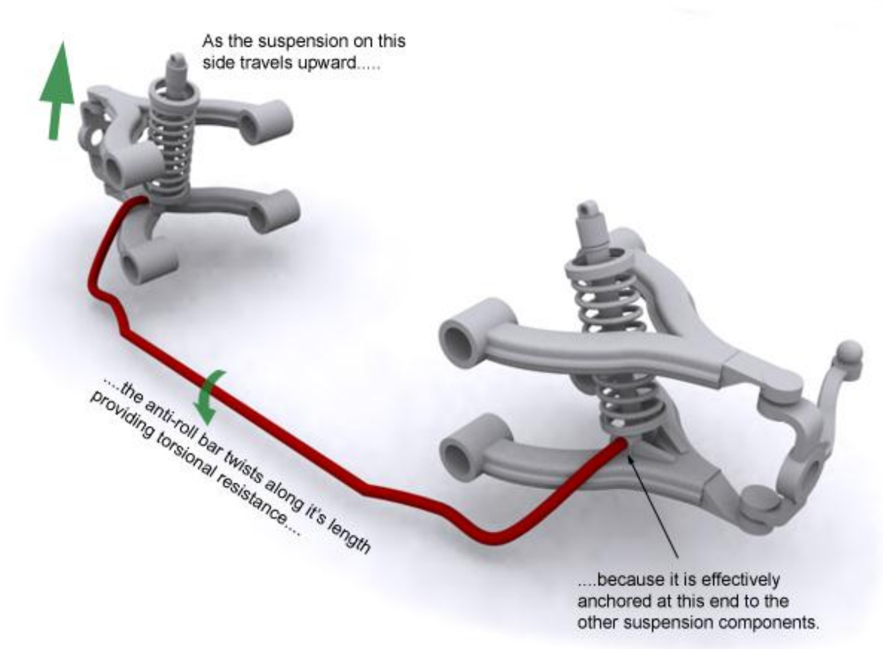

Figure 4.23: How an anti-roll bar works [46]

4.5 Software

This chapter describes the software programs that have been used in this thesis.

4.5.1 MSC Software

MSC Software is an American company founded in 1963 and its main field is to create software that simulates functionality of complex mechanical design. First developed simulation program was MSC/Nastran which was introduced in 1971 and aimed for the aerospace industry. Next program to be introduced by MSC was MSC/Patran; this was aimed for pre- and post-processing of engineering analysis. Since the company's beginning a couple of widely used programs have been introduced as Actran, MSC Fatigue, SimXpert, FEA, Adams etc. The solution areas for these programs are Integrated, Solver, Mid-sized business, Modelling and Simulation data and process management. [52]
These programs are targeted at optimizing product development and enable engineers to validate and optimize their designs using virtual prototypes. Nowadays even real physical prototypes that traditionally have been used in product design sometimes are replaced in favour of computer models and analyses made in MSC software.

4.5.1.1 Adams

Adams is the most popular multibody dynamics and motion analysis software in the world. With Adams there are possibilities to study the dynamics of moving parts and how forces and loads are distributed throughout the mechanical systems. Adams multibody dynamics software makes it easy to create and test virtual prototypes instead of building and test a physical one which is time consuming and expensive. [53]
Adams contains real physics and continuously solving equations for dynamics, kinematics, statics and quasi-statics, and it is also compatible with other programs which enable import/export of models from CAD programs. [53]
The program integrates mechanical components, hydraulics, pneumatics, electronics and control systems technologies so that it can be included in the virtual prototypes and thus be able to validate early system-level designs, and get an accurate result. [53]
Adams contains of different programs as for example Adams/View, Adams/Chassis, Adams/Motorcycle and Adams/Car.

4.5.1.1.1 Adams/Car

Adams/Car is a widely used Multibody Dynamics Simulation software made by the software company MSC Software. It is commonly used in the auto industry and on universities. For example it is common to use Adams/Car in Formula SAE; which is a competition among different universities on who makes the fastest Formula SAE car. There is even a pre-built Adams/Car model of a Formula SAE car that students can modify. [54]
Since Adams/Car is a widely used simulating program it is possible to import files from other computer programs e.g. NX etc., for example if creating complex models/parts are too time consuming in Adams/Car. [54]
By making simulations in Adams/Car issues as time, cost and risks in vehicle design development can be minimized, due to the absence of having to create a real vehicle before knowing the optimal parameters and then optimizing it through continuous drive cycles [54].
Adams/Car has different modules as Adams/3D road, Adams/Car Vehicle Dynamics, Adams/Car Suspension Design, Adams/Chassis and Adams/Driveline to mention a few [54].
In Adams/Car the option to model or modify existing templates is available. Virtual prototypes of complete vehicles and vehicle subsystems can be created with the parameters that are specific for just that kind of vehicle. [54]
There are standard testing procedures built in the program as for example straight-line driving, cornering, steering and quasi-static analyses [54].
Different designs can quickly and efficiently be evaluated early in the design cycle thanks to Adams/Car [54].
It is possible to run analysis of steering, suspension and full vehicle manoeuvres, and different properties for tyres can be modified or imported. Simulating the vehicle driving on a virtual road with various road conditions are possible, and during a simulation an action/load can be applied to see the reaction of the full vehicle/vehicle subsystems; with that information an optimization can be done. [54]
In addition to that, performing various analyses on the vehicle can be made to examine modifications done to components as damper, spring, bushing and anti-roll bar rates. Also changes to the design of the subsystems are possible to see as well as the influences on the

overall vehicle dynamics. [54]
The different parts in the vehicles suspension can be built and connected via joints and constraints, and the interaction between them and other parts of the vehicle as wheels etc. can be analysed [54]
There are possibilities for an overview on how the suspension transmits loads and to check standard suspension analyses of wheel travel, steering characteristics, vertical forces, static loads and body roll. Another area that is supported is the driveline and how that effects the vehicles dynamic behaviour under different operating conditions and study the driveline influence over chassis components as steering, suspension and brakes, etc. [54]

4.5.2 Siemens NX

NX™, also known as NX Unigraphics, is an advanced CAD/CAM/CAE software package product development solution from Siemens PLM Software.
Unigraphics has its roots in the early 70's when it was developed by United Computing from the Automated Drafting and Machining (ADAM) software code from MGS. Unigraphics and its owner UGS Corp. was then sold to Electronic Data Systems (EDS) in 1991. One other significant year was 2002 when the new "next generation" version of Unigraphics and I-DEAS was released. They were now gathered together into a single consolidated product with the functionality and capabilities from both products and now called NX. In 2007 UGS was sold to Siemens AG and since then it is marketed as Siemens PLM Software. Siemens PLM Software is a part of Siemens Industry Automation Division, and has sold over 6,7 million licences to 69.500 customers worldwide. [55]

The capabilities in NX are vast, from design to simulations and even process management. Like most software created for the industry, the goal is to minimize cost and time consumption. And with NX it is possible to improve the productivity throughout the product development line, reducing development time and get a more efficient manufacturing, thus minimizing the total cost. [56]

Some tasks that NX can be used for are:

- Design (Computer Aided Design (CAD))
    - Solid modelling
    - Freeform surface/shape modelling
    - Assembly modelling
    - Knowledge Based Engineering (KBE)
    - Sheet metal design
    - Product and Manufacturing Information (PMI)
    - Engineering drawing (drafting)
- Simulation (Computer Aided Engineering (CAE))
    - Motion simulation - kinematics and dynamic
    - Stress analysis - Finite Element Method (FEM)
    - Computational Fluid Dynamics (CFD)
    - Thermal
    - Electro-magnetical

- Manufacturing (Computer Aided Manufacturing (CAM))
    - Numerical control (NC) programming

Areas that NX can be used in are e.g.:

- Industrial design and styling
- Package design
- Mechanical design
- Electromechanical design
- Mechatronics concept design
- Mechanical simulation
- Electromechanical simulation
- Tooling and fixture design
- Machining
- Quality inspection

Computer Aided Design (CAD) is widely used in almost all product development today as the products get increasingly more and more complex as well as the cost has to be kept at a minimum. With NX the design and development process can accelerate thus deliver higher levels of quality at lower cost. [57]

It all starts with creations of 3D models in NX that rapidly can be built from freeform shaping, parameterised models or standard components. The models/parts can then be assembled and further be used for simulations; e.g. running Finite Element Method (FEM) analyses. FEM is a technique for finding solution of partial differential equations and integral equations, i.e. structural analysis solutions, and it uses Nastran® solver to solve the equations. [57]

The simulation application has capabilities in model preparation, solving and post processing within the mechanical and electromechanical area. By simulating the product/virtual prototype an estimation of the cost, and quality of designs can be made before physical prototyping and therefore it is possible to produce results faster, resolve product issues earlier, save significant time and effort, and minimize risks. That will minimize failure in real physical models by better quality and fewer design errors and thus reducing development time and costs. This is often referred as Computer Aided Engineering (CAE), which enables engineers to understand, predict, and improve the product performance digitally. [57]

NX is great for optimizing products as the design/product can be analysed, verified and validated. It is also well integrated with other systems including industrial design, simulation, tooling and machining applications and is fully associative with other CAD/CAM/CAE applications. This enables industries to capture and reuse products/design and process knowledge consequently improving efficiency and productivity. The design decisions can be made quickly with detailed knowledge of product performance and manufacturability issues resulting in faster time to market. [57]

NX allows for efficient design control and productivity beyond the design process which can reduce costs throughout all phases of product development. By using NX it is possible to consider the entire product lifecycle, from concept design, simulation and analysis through machine tooling design, manufacturing, assembly, service and support. [57]

Preparations for production of the parts using advanced tooling and machining technologies are possible as making drawings, render pictures and capture videos to mention a few things [57].

4.5.2.1 NX 8.0

The NX 8.0 version was released in October 2011. NX 8.0 has High Definition (HD3D) support for better design decisions as well as many other improved features like solving simulations faster, new tools and enhanced feature modelling. [58]

4.5.3 MATLAB

MATLAB® is a high level technical computing software product from MathWorks Inc. which can be used for numerical computation, algorithm development, data analysis and visualization [59].
With MATLAB it is possible to solve technical computing problems faster than with traditional programming languages as Fortran, C, and C++ [59].
MATLAB can be used for various applications as test and measurement, signal and image processing, control design and financial modelling and analysis. Data can be analysed and visualized in 2D and 3D graphics and the program includes mathematical functions for numerical integration, statistics, filtering, optimization, linear algebra and Fourier analysis. [59]
MATLAB can also be integrated with other software as e.g. Simulink and can be used with external applications as Microsoft Excel, Java, Fortran, C, and C++ [59].

5. Specification of requirements

There are some specifications that need to be fulfilled and these are given by law regulations, permits and parameters as e.g. space dimensions from groups within the project. Some requirements are listed below. Parameters as top speed, grade ability, ground clearance, turning radius, wheel base, wheel track, overall length, overall width and overall height are taken from the specifications for the TVS King [60].

The parameters from the TVS King specifications follow beneath:

- Top speed: ~50 km/h
- Grade ability: >7,0 degrees or 16 %
- Ground clearance: >165 mm
- Turning radius: <2860 mm
- Wheel base: ~1985 mm
- Wheel track: ~1150 mm
- Overall length: ~2645 mm
- Overall width: ~1330 mm
- Overall height: ~1740 mm

Parameters listed below are specifications provided from law regulations and the other groups in the project:

- Acceleration: >6,3 m/s^2
- Brake distance: <13 m @ 30 km/h
- Load capacity: 1 driver + 4 passengers
- 120/70x12" wheel in front
- 4.00-8, 6 PR original TVS wheel in rear
- 40x40x65cm rear wheel house space
- 30 cm distance from rear of the rear wheel to rearmost of the vehicle
- No transverse axle

Some of the listed parameters above can be changed while some specifications have to be kept as closely as possible to fulfil today's permits. The reason for that is if the new auto rickshaw shall be able to be sold to the intended buyers who are low income takers, the auto rickshaw must keep the limits of the permits for small lightweight three wheeled vehicles in India. If the vehicle does not fulfil those parameters a new permit has to be issued which is a too large expense for the contemplated buyers.
This gives a smaller control field but the main goal is still to create a safer auto rickshaw with better handling and vehicle dynamic behaviour. Parameters that can be changed are steering angle, caster angle, camber angle, toe angle and rake. Some of these parameters results in the change of other parameters, e.g. steering angle together with wheel base changes the turning radius which must be less or equal to the recent value of 2860 millimetres. Another

specification that can be changed is the trail which the caster angle together with the rake changes and that affects the steering torque.
Other requirements are the wheel base which needs to be approximately 1985 millimetres and maintain the wheel track around 1150 millimetres. The ground clearance on the current TVS King is 165 millimetres which needs to be remained, or even higher because of the poor road conditions in India. However a higher ground clearance results in higher centre of mass which needs to be kept in mind. The wheels used are a 12 inch front wheel and original TVS wheels at the rear with the dimension; 4.00-8, 6 PR. The rear wheel arch including the suspension and the wheel travel are constrained to a space of 40x40x65 centimetres, and the front wheel arch size should be around the front wheel diameter plus five centimetres. The rear wheel had to be mounted ~30 cm from the rear of the wheel to rearmost of the vehicle, i.e. the wheel centre needed to be mounted ~50 cm from the rearmost of the auto rickshaw (to keep the wheel protected in an event of a small impact, and thus keeping the vehicle drivable although it might have some cosmetic damages.)
A transverse axle cannot be used since the auto rickshaw is intended to have a flat floor in the middle while keeping the ground clearance high enough, without effecting the body space of the vehicle, i.e. there is no space for a transverse axle.
There are some parameters to consider when running the simulations, as the top speed around 50 km/h, the acceleration which is going to be at least 6,3 m/s^2 at some point, and the brake distance at 30 km/h which needs to be 13 meter or less according to law regulations. The auto rickshaw should also be able to manage grade ability of 7,0 degrees or 16 % inclination. These numbers are inputs for the simulations.
The vehicle was designed for one driver and four passengers. The people situated in the auto rickshaw should be included in the weight distribution which also is an input for the simulations.

6. Concept evaluation

The suspension solution must fit within the space specified by the platform group as well as fulfil the necessary tolerances with respect to geometry and stability. The solution must be cheap, ease of use, meet desired behaviour with regard to the passive safety of passengers and other road users and to keep wheel forces into the body as low as possible [3].

6.1 Initial suspension evaluation and selection

To separate the category influence on the evaluation criterias a weight matrix was used, since all categories don´t have the same impact. Rear and front suspensions were graded in separate matrices which can be found in Appendix A: I,II. The suspensions were graded from 1-5 in each category, one being the lowest grade and five being the highest. A category importance matrix was used and evaluated by comparing the categories with each other, 1 having equal importance and 2/0 having greater/less importance. After grading each category a percentage value was acquired, this value was then multiplied with each suspensions grade in that category. Appendix A: III shows the category weight values. Total score for rear and front suspension with the total graded importance matrix are shown in Appendix A: IV.
Without the category importance matrix, suspensions that gets high grades in low importance categories can thus get a high total score and advance to the next phase of evaluation. With the category importance matrix there is a possibility to get a view over the total score including each categories impact.

Starting with eleven different rear suspension types and seven different front suspension types, a weight matrix was used to reduce the different solutions to three front suspensions and three rear suspensions.
The eleven rear suspensions and the seven front suspensions are described in the previous benchmarking chapter.
Each category in the weight matrix for the front and rear suspensions and its evaluation process is discussed below.

6.1.1 Simplicity

In this category the simplicity of the suspensions were evaluated by looking at how many movable parts that were included, how each part were connected and also how complex the shape of each part were. A solution with few movable parts and simple design was searched for, because the market requests such.

6.1.2 Space demand

Space limit for the rear suspension were discussed together with the vehicle platform project group and the result was 65 cm front to back, 40 cm depth and 40 cm height. Those solutions that demanded more space were graded with a lower score and those that better fulfilled the space demands got a higher score.
The suspensions with transverse axle were discarded because of the low positioning of the platform floor, that together with a transverse axle would cause a to low ground clearance due to the criteria set up. The criteria for the ground clearance was set to minimum 16,5 cm which

is referred to the ground clearance of the 2011 TVS King.
The space limit for the front wheel were different partly because it was placed in the middle, and given that the front body will form around it and create a spherical wheel arch with a measure of the wheels' diameter plus 5 cm. Increasing the ground clearance to get more space for the front wheel would mean a higher centre of mass which is not wanted, and positioning it further away longitudinally was not allowed because of the permits restricting the wheelbase. That is one reason why a front suspension that did not require much height would get a higher score.

6.1.3 Handling

With the book "*The Automotive Chassis: Engineering Principles - Second edition*" [3]; desired handling aspects for the rear suspensions were acquired. Such aspects are for instance kinematic and elastokinematic properties, cornering characteristics and roll centre.
The handling aspects for the front suspensions were collected from different sources, mainly from the references of each type with a more thorough research.

6.1.4 Dynamics

Desired dynamics for the vehicle are high transient response, suspensions ability to absorb both vertical and lateral forces and sufficient bump travel.
The dynamic characteristics for the different suspensions were partially estimated with logical reasoning and partially with facts from the book "*The Automotive Chassis: Engineering Principles - Second edition*" [3].

6.1.5 Economic aspect (manufacturing)

The economic aspects were estimated, due to difficulties of finding facts that give a direct answer. To give a correct cost for each solution would require extensive research and would be far too time consuming for this thesis. Therefore the result was estimated with reasoning and logical discussion with a reasonable manufacturing cost in mind. The manufacturing cost was evaluated depending on number of parts, complicated manufacturing processes depending on each parts structure and the entire structural design.

6.1.6 Economic aspect (maintenance, durability)

This category was evaluated according to the buyers' perspective and estimated with logical decision. Poor durability and expensive for the customer to maintain and service were given a low grade. Durability, length between each service, and future service cost were estimated according to how many low sustainable parts included in the suspension.

Criterias	Front suspension				
	Trailing link	Leading link	Earles fork (type of leading link)	Telescopic fork	Springer fork
Simplicity (Low grade = low implicity)	0,33	0,33	0,25	0,33	0,17
Space demand (Low grade = takes a lot of space)	0,33	0,33	0,25	0,25	0,17
Handling (Low grade = poor handling)	0,67	0,89	1,11	0,67	0,22
Dynamics (Low grade = poor dynamics)	0,67	0,67	0,67	0,89	0,44
Economic aspect (manufacturing cost) (Low grade = high manufacturing costs)	0,78	0,78	0,78	0,97	0,78
Economic aspect (maintenance, durability) (Low grade = high costs for the customer)	0,78	0,78	0,97	0,78	0,58
Grade:	**3,56**	**3,78**	**4,03**	**3,89**	**2,36**
Total result:	**Discarded**	**Chosen**	**Chosen**	**Chosen**	**Discarded**
The three types with the highest grade will be chosen for further evaluation.					

Table 6.1: Front suspension weight matrix – Total score

Criterias	Rear suspension						
	Double wishbone	McPherson strut	Chapman strut	Semi-trailing arm	Trailing arm	Multi-link	Torsion bar
Simplicity (Low grade = low simplicity)	0,17	0,25	0,25	0,33	0,33	0,08	0,42
Space demand (independent 65x40x40cm) (Low grade = takes a lot of space)	0,33	0,08	0,08	0,42	0,42	0,08	0,42
Handling (Low grade = poor handling)	1,11	0,67	0,89	0,44	0,44	1,11	0,22
Dynamics (Low grade = poor dynamics)	0,89	0,67	0,67	0,44	0,44	1,11	0,22
Economic aspect (manufacturing cost) (Low grade = high manufacturing costs)	0,39	0,58	0,58	0,78	0,97	0,19	0,97
Economic aspect (maintenance, durability) (Low grade = high costs for the customer)	0,39	0,58	0,58	0,97	0,78	0,19	0,97
Grade:	**3,28**	**2,83**	**3,06**	**3,39**	**3,39**	**2,78**	**3,22**
Total result:	**Chosen**	**Discarded**	**Discarded**	**Chosen**	**Chosen**	**Discarded**	**Discarded**
The three types with the highest grade will be chosen for further evaluation.							

Table 6.2: Rear suspension weight matrix – Total score

Above the weight matrices with the total score for the rear and front suspension types are shown. Table 6.1 refer to the front suspension types and Table 6.2 refers to the rear suspension types.

From those matrices the suspension types to further evaluate were chosen.
The three front suspensions chosen were; leading link, Earles fork and telescopic fork.
The three rear suspensions chosen were; double wishbone, semi-trailing arm and trailing arm.

6.2 Final evaluation and selection

Further investigation of the different suspension solutions generated by the weight matrix follows.
There were nine different suspension combinations to choose from, three different front suspensions and three different rear suspensions. The next step was to choose which one of them to use for optimization. A thorough theoretical investigation of the remaining suspension types and their qualities, advantages, disadvantages etc. was needed to be able to bring down the remaining concept to only one.

6.2.1 Front suspensions

Beginning with the remaining front suspensions which are telescopic fork, Earles fork and leading link, the telescopic fork could be discarded because of the odd configuration needed if fitted as a single sided type which is a requirement in this case.
If fitted to only one side the telescopic fork needs to be configured as dual tubes fitted longitudinally at one side, otherwise the wheel will swivel around its steering axis as there is nothing to stop it from pivoting around its axle as the telescopic tube is round.

Another solution could be to use an elliptical or square tube to prevent the wheel from revolving around its steering axis, but all of those types of tube configuration gives large tensions in the bearing in the head tube as well as on the fork itself. A different configuration that is possible to use is a very big single tube fitted on top of the fork in the middle of the frame, since it needs to be steady enough to handle all the forces that will be transferred into that single point.
None of those types of configuration have been used on a larger scale, only on concept motorcycles and some odd bicycles. The performance and reliability is thus only covered in a small scale and it is difficult to know all the disadvantages/advantages that this peculiar shape of telescopic fork can give.
Other disadvantages associated with telescopic forks mentioned before are the wobble tendencies and the dive effect that occurs during braking when the dampers compresses. The main advantages as low cost and simple design, does not apply in this case as the design gets atypical wherefore the advantages does not outweigh the disadvantages.
The telescopic fork is thus a possible, but not reasonable choice.

Remaining front suspension solutions were Earles fork and leading link. Those two types have a similar design with the main difference where the aft pivot point is located. On the Earles fork this point is behind the wheel as described before, and on the leading link it is just aft the wheel axle. Earles fork can be considered as a type of leading link. Otherwise they share their advantages as anti-dive characteristics, light steering, simple design and low cost as well as disadvantages like large forces transferred into the steering head when the suspension is operating.
Because of the very limited space in the front of the auto rickshaw the choice was made to use the leading link design.

6.2.2 Rear suspensions

The three different rear suspensions remaining from the matrix evaluation were trailing arms, semi-trailing arms and double wish-bones. To sort out which one to use a thorough enquiry follows down under for the three different types of rear suspension.
The first one to more deeply analyse is the double wishbone. The double wishbone is the one that gives the best performance, as mentioned before it has superb kinematic characteristics, good elastokinematic behaviour, good load-handling capabilities, a lot of adjustable parameters etc.
The question is then if optimal characteristics can be achieved within the small space available for the suspension? A double wishbone design can be implemented for the auto rickshaw, but to keep the ground clearance high enough and avoiding the wishbones to take up more space without affecting the coupe would be difficult. Also the wishbones would have to be quite short which is not preferred seen from a kinematic perspective since longer A-arms gives less scrub [46]. The distance between the links should be as long as possible to minimize the forces in the control arms and their mountings, i.e. component deformation gets smaller and wheel control more precise [3]. Due to needs for relatively small distances between the mounting points, the mounting points need to be firmly fitted, which transfer more forces into the body and decreases the ride comfort [9].
As the double wishbone is such a complicated design it can be difficult to assign all the many

available parameters correctly keeping in mind the overall motion movement the suspension will carry out.
Another big question to consider is how expensive the construction is and how recognised it is by the intended buyers? The answer is that the double wishbone suspension consists of a lot of parts which means that it will have a very complicated structure that perhaps won´t be recognised by the average layman in India. The complicated structure with its large amount of suspension links compared to the trailing, and semi-trailing arm design also makes the construction more expensive to manufacture and repair in event of failure in comparison to the two other types discussed. Those two arguments would unfortunately weigh more than the performance the double wishbone would provide.

With the double wishbone out of the equation only two suspension types remained, trailing arms and semi-trailing arms.
Semi-trailing arms gives better kinematic characteristics compared to trailing arms. On the contrary it will instead take more space in the horizontal plane than the trailing arms, and the elastokinematic properties are poor due to the high forces affecting the suspension links.
Vehicles fitted with semi-trailing arms have a tendency to oversteer due to the camber and toe change when the wheel bumps and rebound-travel. The camber change is linear-lined and the toe in/out change is curved lined, the opposite of what is desired in a good suspension.
As shown in Figure 6.1 the roll centre of semi-trailing arms is just above the ground. A low roll centre is preferred as described earlier in this thesis.

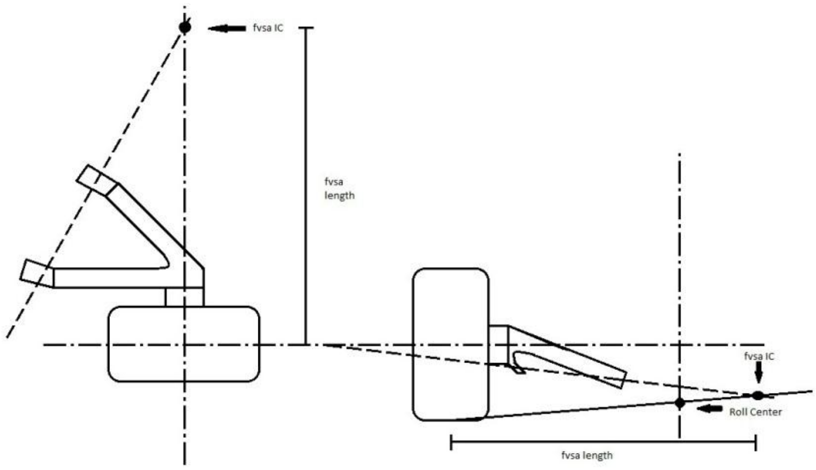

Figure 6.1: Schematic view of semi-trailing arms and its roll centre

Although the semi-trailing arm has a simple basic design, it can be difficult to fully see the effects various positions of the radius arm axis can result in. The radius arm axis is the axis as the triangular arm pivots around as described earlier in this thesis under benchmarking of semi-trailing arms. The radius arm axis is often angled between 10 to 25 degrees from the line perpendicular to the vehicles centre line. Depending on that offset angle, the camber and toe in/out changes, the wheelbase alternate, and the track width, roll centre height changes.
The greater the offset angle is from the line perpendicular to the vehicles centre line, the more the camber, toe in/out changes during wheel movements [3].
Semi-trailing arm designs are cheap to manufacture, but not as cheap as trailing arms due to the more material that will be needed.

Trailing arms has the advantages of low manufacturing cost, small space requirements (especially inwards to the middle), simple design used today by several auto rickshaw manufacturers and thus will the average user recognize the construction and be able to repair it with simple means.
Trailing arms have some disadvantages as the design causes the movements only to be vertical, there are small possibilities for kinematic changes and weak elastokinematic properties, due to bad change in camber when cornering due to deflection of the control arms and movements at the attachment to the body. When cornering the lateral forces will induce a positive camber which results in understeering. The attachments will move during lateral forces due to high torsional and bending stress in the position of torque fulcrum.
The trailing arm design also result in large change in caster trail and some change in wheelbase [9].
Although the trailing arms have some disadvantages, they were overcame by the significant advantages as the cheap and simple design that is space saving. Especially cheapness and space saving properties are important as the final product must be kept inexpensive enough for the average low income taker in India to be able to buy it. The suspension must be simple enough to make it easy and cheap to maintain and repair. Space need to be kept at a minimum due to the small total vehicle size, and that it is intended to accommodate a flat floor in the middle of the vehicle while keeping the ground clearance high enough. The trailing arms fulfil those criteria as the trailing arms take little space transversely.
The roll centre is on the ground for trailing arms which is shown in Figure 6.2. The roll centre is thus lower than for semi-trailing arms. As said before a lower roll centre is preferred which makes trailing arms better regarding this feature.
The trailing arm design also makes the ride comfortable when travelling in a straight path.

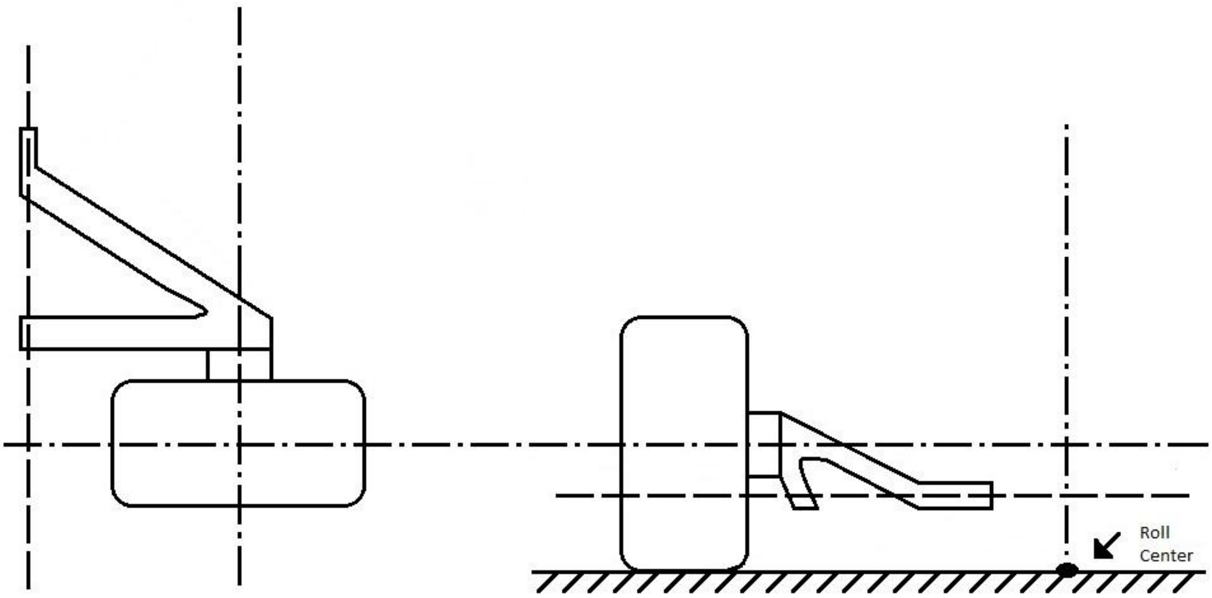

Figure 6.2: Schematic view of trailing arms and its roll centre

The chosen solution for the concept was the trailing arms as the rear suspension, but with an additional sway bar to reduce body roll.

7. Detailed Design

This chapter contains a guidance of each process during the detailed design of the chosen suspension solution.

7.1 Leading link

The front suspension caster angle was chosen to 19 degrees as which is commonly used today. The lower link should keep a length within the radius of the wheel to fulfil the strict space criteria in the front of the vehicle. The leading link thus got the approximate length of 200 mm. The leading link is preferred to be level to the ground to perform well dynamically, e.g. when steering (no change of height of the wheel). To keep the tight space requirements while having the longest spring and damper possible, the second link (the upper one connected to the head tube) were given an angle of 80 degrees from the leading link.
Other measures as the thickness of the tube, hub, etc. were approximations of the measurements on the 2011 TVS King.

7.2 Trailing arms

As mentioned before the kinematic design possibilities are few for the trailing arm suspension as it has a very simple design. Trailing arms are designed only for vertical movements and must be strong regarding bending in all directions to resist camber torque, braking torque and steer direction torques from the lateral and vertical forces that are exerted on the tire, as there is only one link used per side [6]. The trailing arms thus have weak elastokinematic properties.
The trailing arms have a triangular form with the top "corner" attached to the wheel hub, this to have a space saving design while keeping the strength high. Because of the high torsional and bending stresses in the position of the attachments to the body and to minimize unwanted positive camber change due to deflection of the trailing arms when cornering (which results in understeering) the trailing arms should be as wide as possible.
The trailing arms must further fit inside the wheel houses that were constrained to a space of 40x40x65 centimetres because of the flat floor that was intended to be accommodated in the middle of the vehicle. The rear wheel centre was desiderated to be mounted ~50 cm from the rear end of the auto rickshaw, which would give a distance of ~30 cm from the rear of the wheel to the rearmost of the vehicle and thus having a safety margin to the wheel in an event of impact.
The axle that the trailing arms pivot around should lie parallel to the ground/floor and consequently there will be no camber, toe or track width changes during wheel movement through bump and rebound travel. The trailing arms should be as short as possible to increase force that pulls the tail end down during braking. Although shorter arms induce larger wheel base change during wheel movements, it does not affect the handling properties. [3]
Due to this information, the trailing arms were given the approximate width of: 270 mm, and length of: 370 mm.
As with the caster angle for the front suspension the toe and camber angle for the rear suspension will be 0 degrees. No toe in/out will give an even and low tire wear and a low roll resistance. By having no camber likewise results in less tire wear and rolling resistance,

although the lateral tire grip will be less when cornering as negative camber is preferred to counteracts this. Less grip in the rear tires is not that crucial since that is not the most substantial parameter for keeping the auto rickshaw stable; i.e. to prevent the vehicle from tipping over. The prime variables for tipping over regards the height of the centre of mass, roll centre and sidesway. The trailing arms will thus be added a sway bar to reduce the sidesway. The trailing arms will be angled around 10 degrees downwards from the vehicles horizontal line to get the desired ground clearance.

The structural design of the rear suspension derived from the existing rear suspension of the 2011 TVS King. Since the trailing arm design is similar to the one already used on the TVS King, a production line for it already exist that only needs some minor modifications, and will thus minimize the cost for the company, which is a benefit.

7.3 Adams/Car simulations

This chapter will guide through the usage of Adams/Car starting with the modelling of the chosen suspension.

7.3.1 First time using Adams/Car

A detailed list over the fundamental expressions used in Adams/Car and explanation of the terms follows:

- **Template:** The file format that Adams/Car uses when creating/building a model.
- **Subassembly:** A file format used in Adams/Car. Templates must be converted into subassemblies before being assembled.
- **Assembly:** The file format that is used when performing simulations.
- **Grid:** A plane in space to use as a reference, to know the X-, Y- and Z-directions and also where origo is.
- **Construction frame:** Used when a part needs a different orientation in space than the one given by the grid and it is visualized as an orange coordinate system.
- **Hardpoints:** By assigning an X-, Y- and Z-coordinate, a green point in space will appear.
- **General part:** A weightless part that is visualized as a yellow coordinate system, placed either between two hardpoints or on a hardpoint.
- **Geometry:** Give the weightless part created in "General part" a weight and visualization.
- **Wizard part:** Wizard part is used instead of making both the "General part" and "Geometry" steps, i.e. it creates both a part and gives it a weight in one step, but this is only possible for two different parts; link and arm.
- **Joints:** Connects two parts together and locks DOF between the two parts, i.e. locking one parts movement from the second one.
- **Bushing:** Bushing creates a force resistance in desired directions, i.e. has the same function as a real bushing.

- **Forces:** This category includes different functions as for example dampers and springs, i.e. parts that will cancel/absorb forces.
- **Communicators:** Connects and communicates between templates, different communicators can be assigned depending on the purpose. Communicators are divided into two categories; input and output. The input communicators are visualized as a yellow coordinate system and receive information, while output lies in the background and transmits the information.
    - **Mount:** Most important communicator, it sends/receives information from one template to another.
    - **Location:** Informs about a templates whereabouts.

First task was to understand how to build models in Adams/Car which was done by a couple of steps as follows:

1. Designating hardpoints: With the default mode the length is viewed in millimetres and by knowing this the distance of each hardpoint could be placed to match the real measures. When appointing hardpoints a choice of applying either dual or single points is available. The dual choice is mirrored on the X-Z plane while the single choice will not be reflected.
2. Creating parts: This can be done in two different options; one option is to first create a non-geometrical part that is placed on one or between two hardpoints and then adding geometry to that part. The second option is to create a wizard part which has to be constrained between two hardpoints, wizard parts are either an arm or a link that already have geometry.
3. Creating springs, dampers and bushings: When creating springs and dampers both parts and coordinates are needed. For example the upper/lower connection point of the damper/spring must have a part to join to and a coordinate where it is supposed to be attached. While bushings only needs one connection coordinate and two parts that are supposed to be joined together. In this category the stiffness coefficients and damping ratios of the springs, dampers and bushings can be altered as desired.
4. Creating joints: Joints are placed at the connection point between every part by knowing the desired movements of the model. The desired movements are decided by how many degrees of freedom (DOF) that are locked, and by knowing the DOF:s redundant joints can be avoided. If there are any redundant joints they will be shown after the simulation by an appearance in the error box.
5. Creating communicators: When creating communicators three criteria must be kept in mind, name, minor role and type of communicator. If these three criteria are not fulfilled the output and input communicators will not trade information and will therefore not work. If an input communicator has no matching output communicator an error sign will appear. This does not apply for output communicators since them only lies in the background.
    - Mount communicators are placed within a template at the point on the parts where information are meant to receive from or transmit to another template. Mount communicators only receives/transmit information and does not report its location.

- Location communicators are communicators that inform a templates position to another. For example the wheel is centred to the wheel hub with the help of a location communicator.

After all these steps a model of the desired suspension should be completed.

7.3.2 Rear suspension

The first suspension to be modelled was the rear suspension of the trailing arm type which was modelled from scratch by going through step 1-4 in "First time using Adams/Car". Although modelling from scratch is not recommended, especially not when the user lacks experience in the program.
The trailing arm suspension was modelled with the geometry and measurements found in the design chapter, but the trailing arm suspension model could not be completed since one vital step was missing in "First time using Adams/Car"; mount- and location communicators. Therefore it led to the search of existing templates and a couple templates were found but unfortunately trailing arm suspension was not among them. The suspension found most similar to trailing arms were semi-trailing arms and instead of modifying the semi-trailing arms to trailing arms it was used as a reference template, partly because a trailing arm suspension template already been initiated.
The details on each communicator's position could then be retrieved from the semi-trailing arms template, both input and output. Communicators found in the existing semi-trailing template are communicators for e.g. driveline as differential, drive shafts, etc. However the only communicators needed were mount communicators, location communicator and brake communicator.
Now a rear suspension had been modelled and was ready for simulations.
A picture of that as a screenshot taken in Adams/Car is shown in Figure 7.1.

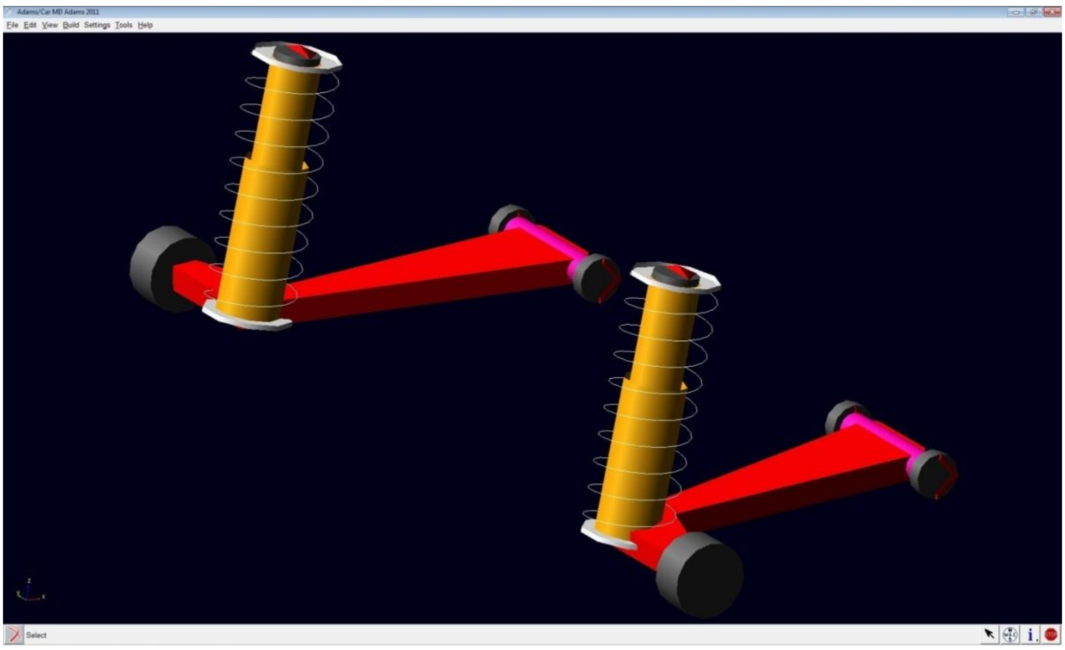

Figure 7.1: Trailing arms modelled in Adams/Car

7.3.3 Front suspension

A single leading link suspension was modelled from scratch with the measurements from the design chapter since the front suspension was going to be created as a single sided motorcycle fork. After going through step 1-4 and reaching step 5 in "First time using Adams/Car" an obstacle appeared, an output communicator called suspension parameter array needed symmetrical suspensions i.e. dual sided suspensions to be able to be applied. This communicator informs the suspension strut which parts it is supposed to travel between. Since Adams/Car was built for four wheeled vehicles, the test rig for suspension travels were only compatible for either separate two wheel front/rear suspension travel; called parallel wheel travel, or suspension simulation for all four wheels. The parallel wheel travel test rig consists of two platforms placed horizontally and parallel to each other. Because of this the simulations for the single front wheel became an issue since the test rig was built for two wheels and an option for a single suspension parameter array was not available as mentioned earlier. This issue was postponed and a dual sided front suspension of the same type was modelled, by going through step 1-5 from the "First time using Adams/Car" section. Simulations were conducted to get information about what kind of joints and communicators the suspension model would need to make proper suspension travels. With that information the joints and communicators were added to the single sided front suspension. Since no solution on how to create a suspension simulation on a single sided suspension was found an alternative solution was implemented. The solution was to model a dual sided suspension with minimal distance between the wheels which would resemble a one wheel configuration, both visually and functionally.
This setup is the one referred to as dual suspension later on in suspension/steering simulation section.
In Figure 7.2 a screenshot taken in Adams/Car shows the modelled dual sided leading link.

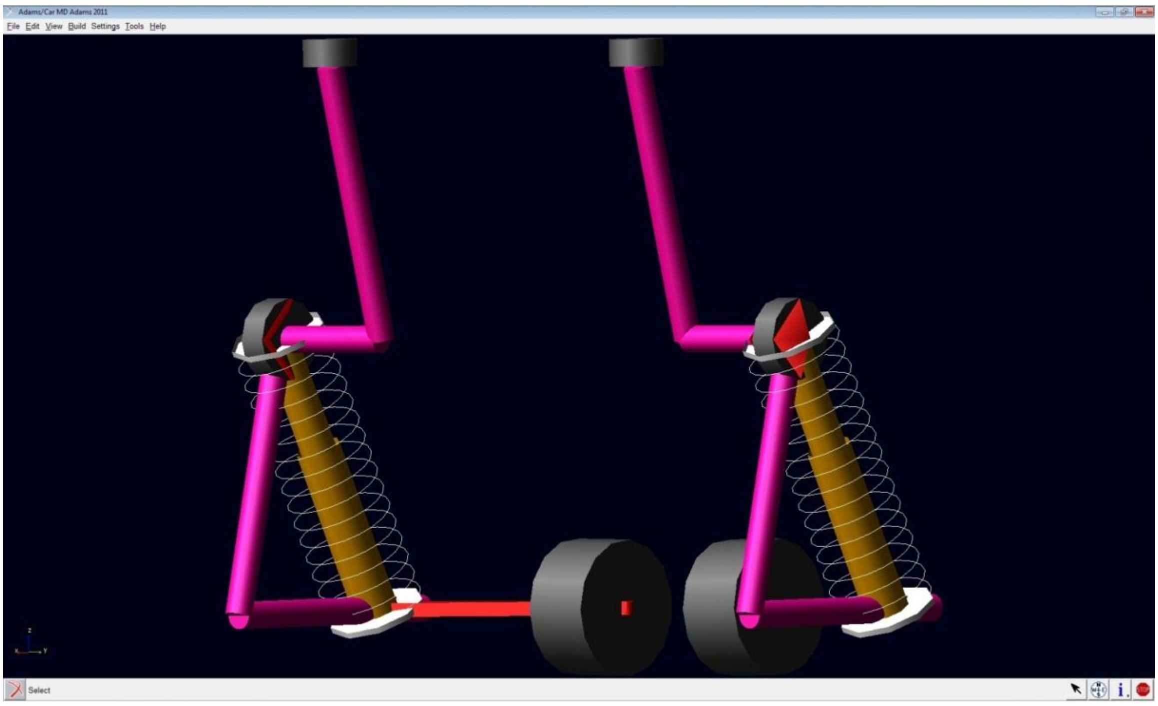

Figure 7.2: Dual sided leading link modelled in Adams/Car

7.3.4 Body

The auto rickshaw body frame was modelled in NX by the vehicle platform group and then exported from NX by converting the UGS NX .prt file into a Parasolid .x_t file. The Parasolid .x_t file was then imported into Adams/Car.
Having the structure in Adams, suspension templates could be adjusted to fit within the frame structure by adjusting the hardpoints defining the position of the different parts in the template. Matching communicator to the other templates was also implemented so that later on when a full vehicle were to be simulated it would send information to the other subsystems.

7.3.5 Suspension and steering simulation

To create a suspension simulation, the suspension templates had to be converted into subassembly file types before being assembled together. To make the suspensions transmit information to the standard test rig the communicators had to match the three criteria mentioned in step 5 at "First time using Adams/Car" section.
In this application a series of different tests could be conducted e.g. steering, parallel wheel travel, etc. First parallel wheel travel simulation was performed on the trailing arms, to test that all joints and communicators acted as desired. The suspension simulations performed on the front suspensions were executed on the dual front suspension and not on the single front suspension, this due to the issues about the test rig explained earlier in the "Front suspension" section.
Steering simulations would also be performed on the dual front suspension and not on the single sided front suspension since this test uses the same test rig as in the parallel wheel travel simulation. With some search within Adams/Car an existing steering template used for car suspensions was found and since the front suspension was going to be a fork kind, the existing steering template would require some extensive modification to suit the dual front suspension. The purpose of these simulations were to make sure that the joints used on the dual suspension during suspension simulations still worked properly during the steering simulations and to know if the modifications of the steering would work properly. In this simulation three templates had to be converted into subassemblies; steering, suspension and test rig, and for these to interact the communicators had to be matched. With the information retrieved from Adams/Cars existing templates, right communicators could be placed on the templates used in the simulations. The joints used in parallel wheel travel simulation were found to not perform correctly in the steering simulation.
With the results from these simulations further work on suspension/steering simulation was postponed because of the difficulties mentioned about the single front wheel, joints in parallel wheel travel simulations and steering simulations.

7.3.6 Full vehicle simulation

In this application a full vehicle could be assembled for different kinds of simulations; Course event (3D road, ISO-lane change), Cornering events (brake in turn, constant radius cornering, cornering w/Steer release etc.), etc. To run a full vehicle simulation several templates were needed as; front and rear suspension, steering, brakes, powertrain, wheels and body/frame. With the suspension, steering and frame in order there were only powertrain, brakes and wheels missing. With the knowledge of what templates that were missing to create a full

vehicle simulation, a search was conducted. Existing templates of those were found and altered to match the auto rickshaw's setup.
To create a full vehicle assembly the templates mentioned above had to first be converted into subassemblies before being united into the full vehicle assembly. With all the subassemblies merged into the assembly, it was possible to move each subassembly into its specific location. An attempt to create the full vehicle simulation rose up to more errors, some of those reflected back to the earlier errors that occurred. With the time spent an idea of how much more time the solutions to the errors might consume, indications showed that it would probably be too time consuming. Hence a decision was taken to switch over to Siemens NX 8.0 and the vision of a fully functional Adams/Car simulation was abandoned.

7.4 MATLAB calculations

The MATLAB version used in this thesis was R2011a. MATLAB was used to calculate the position of the centre of gravity with different loads and to calculate the stiffness coefficient for the front and rear suspension springs.

The centre of mass position for the vehicle, including the driver and passengers, was calculated from the centre of gravity of each part, which included the auto rickshaw structure inclusive body sheet, fuel engine, batteries, dash board, rear seats, front seat, passengers and driver. These parts together represent the sprung mass. The mass of those parts were acquired from the responsible team of that area and the positions were acquired from the NX model/assembly of the concept auto rickshaw.
Below is a list of the different masses:

- Auto rickshaw body structure inclusive body sheet: 180 kg
- Fuel engine (Honda GX270): 22 kg
- Main batteries (LiFePO4 48 V, 40 Ah): 25,6 kg
- Small battery (LiFePO4 12 V): 3,4 kg
- Dash board: 20 kg
- Rear seats: 10 kg on each side
- Front seat: 5 kg
- Passenger/driver: 49 kg (Acquired from the 50:th percentile of male Indian agricultural workers [61])

The centre of mass was calculated for two different cases; minimum load (only driver), and full load (four passengers plus driver).
The total mass including vehicle for those cases follows underneath:

- Minimum load – Only the driver: 325 kg
- Full load – Driver plus four passengers: 521 kg

The mathematical formulas used to calculate the centre of gravity position was:

Position: $$p_a = (X_a, Y_a, Z_a) \tag{1}$$

Centre of mass: $$CM = \frac{(m_a * p_a)*(m_b * p_b)}{m_a * m_b} \tag{2}$$

The MATLAB script for the calculation could easily be modified if the positions or masses of the "parts" would be changed, the script can be found in Appendix B.
The position of origo was set to the left rear side on the ground, to only have positive position coordinates for the different centre of masses.

With two rear wheels and one front wheel; called a delta shape, a stable triangular zone formed from the wheel positions could be drawn. The position of the centre of gravity should be within the triangular zone to maintain stable characteristics. In Figure 7.3 the two weight scenarios are shown inserted in the triangular zone. The picture indicates that the vehicle would be reasonable stable.

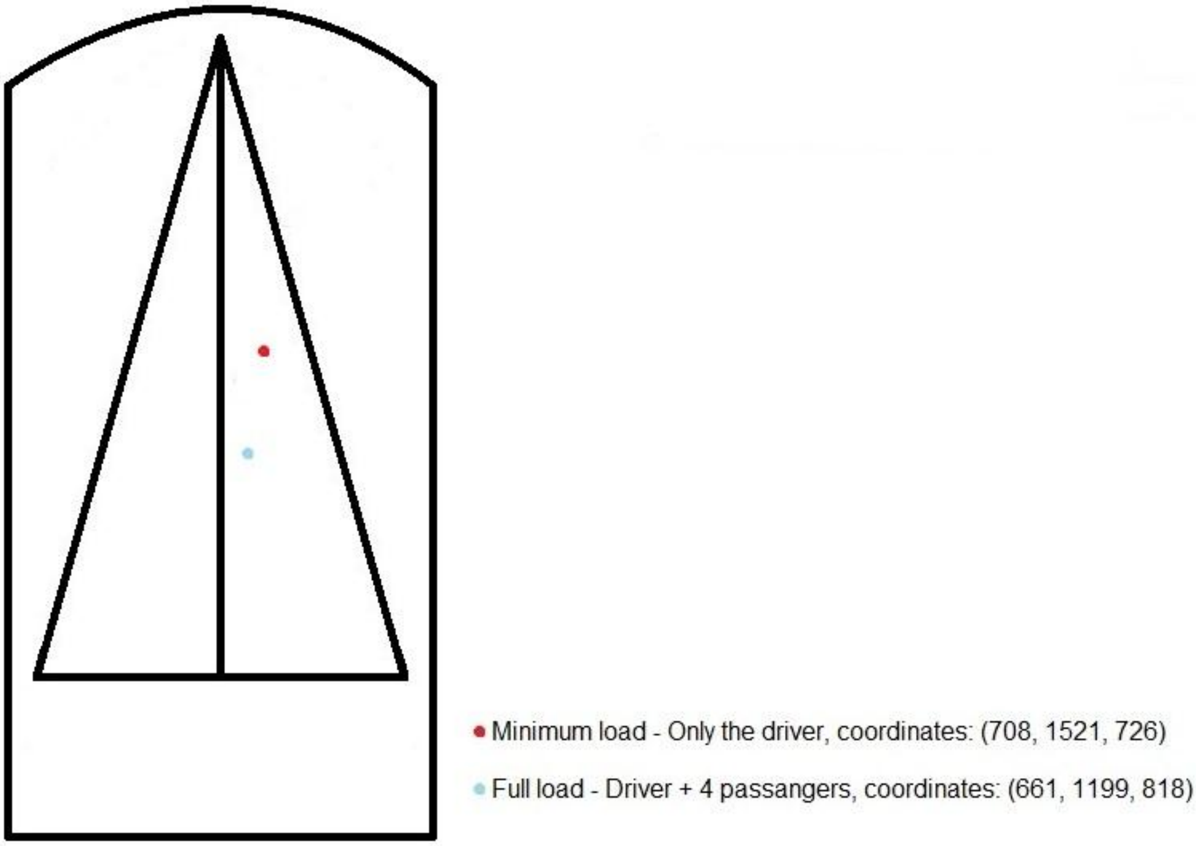

Figure 7.3: Centre of gravity position seen from above

To calculate the spring constant for the front and rear springs a different MATLAB script was created.
Within the MATLAB script, the weight distribution on each wheel was calculated, since the centre of gravity is not located in the centre of the vehicle. An auto rickshaw got three wheels instead of four which implies that the front wheel might have to carry more weight than the rear wheels. As the position of the centre of gravity for full load is located more to the rear than the front, more weight will be carried by the rear springs. With only one front wheel, the weight that is distributed to the front will be carried by that spring alone. For the rear part, the weight distribution on each side needed to be calculated (since there are two rear wheels). The spring stiffness coefficient was calculated and retrieved from the side that carried the most weight, but the same value were used on both sides since the rear springs are placed symmetrical and should have the same spring stiffness coefficient.
When the weight distribution is known the resulting force on each wheel could be calculated, which is the same as the force the springs should withstand for a given length.
The spring stiffness coefficient:

$$k = \frac{F}{x} \tag{3}$$

where F is the force acting on the spring and x is the springs' displacement from its free length to its initial length.

The spring stiffness coefficient was tested at static state with full load (521 kg) resulting in the stiffness coefficients:

- Front: 60 N/mm
- Rear: 60 N/mm (each)

The MATLAB-script for the spring calculations can be found in Appendix C.
After the position for the centre of gravity and spring stiffness coefficient were derived, they were used in NX 8.0 to simulate performance in different tests.

7.5 Siemens NX 8.0 simulations

With the transition over to the latest NX version available (8.0), a different modelling approach was used. Even though NX was not intended to carry out advanced vehicle dynamic simulations it can be conformed for that purpose by using suitable features and assigning right parameters. These modelling and simulation approaches are described below.

7.5.1 Building models

With Siemens NX 8.0 all necessary parts was modelled as solids. These parts were leading link fork, front/rear dampers, front/rear wheels, wheel hubs, trailing arms, sway bar (anti-roll bar) and end links to connect the sway bar to the suspension. Additional parts needed to be able to create a simulation and test the vehicle's performance were an auto rickshaw body, road, ramp and bumps. An auto rickshaw frame was modelled by another group within the project. The auto rickshaw frame was constructed as a simple structural model of the prototype vehicle with

the measurements from the specification of requirements chapter.
A small cube was modelled and constrained to the centre of gravity position, this to gain the possibility of changing the position. The centre of gravity position located inside the auto rickshaw frame could then be changed depending on the sprung weight.
The few parameters mentioned in the design chapter for the leading link and trailing arms together with studies on the existing TVS King provided the measurements needed to construct a 3D model of the leading link and trailing arms. The prototype were given an additional design possibility as the positions of the dampers/springs and links could be altered which gave the potential of trying out different setups. The parts that were not discussed in the design chapter i.e. wheels, dampers, end link and sway bar were modelled with measurements derived from the existing TVS King as well as from literature studies. The road gained the measurements of a 120x120 m square field, this to fit a skid pad; a circular course with a radius of 40 m.

To carry out an accurate simulation the most significant weight and dimension values are listed below.

Wheels:

- 12" hub motor front wheel
  - Weight: 15,0 kg (Including generator, rim and tire)
  - Outermost diameter: 530 mm
  - Width: 140 mm
- 8" rear wheel (Original TVS King wheel)
  - Weight: 10,3 kg (Including rim and tire) [62]
  - Outermost diameter: 441 mm
  - Width: 117 mm

Wheel hub (also represents the electric motor in the simulation):

- Weight: 5,0 kg

Leading link:

- Weight: 4,7 kg (Total weight)
- Lower leading link length: 200 mm
- Lower leading link width: 69,5 mm
- Upper leading link length: 285 mm
- Upper leading link vertical height: 272 mm
- Upper leading link tube outer diameter: 50 mm, 4 mm thickness
- Upper leading link width: 120 mm

Trailing arm:

- Weight: 11,7 kg (Measured in NX with material properties for steel)
- Length of the trailing arm: 370 mm
- Width at the attachment points: 270 mm
- Trailing arm box size: 70x40 mm, 3 mm thickness

Front damper:

- Weight: 3,3 kg (Including spring and damper)
- Spring outer diameter: 55 mm
- Upper part:
  - "Piston" outer diameter: 30 mm
  - "Piston" length: 75 mm
  - Total length: 124 mm
- Lower part
  - "Piston" outer diameter: 35 mm
  - "Piston" length: 69 mm
  - Total length: 105 mm

Rear damper:

- Weight: 5,5 kg (Including spring and damper)
- Spring outer diameter: 72 mm
- Upper part:
  - "Piston" outer diameter: 35 mm
  - "Piston" length: 105 mm
  - Total length: 138 mm
- Lower part
  - "Piston" outer diameter: 40 mm
  - "Piston" length: 107 mm
  - Total length: 142 mm

End link:

- Weight: 0,7 kg (Measured in NX with material properties for steel)

Sway bar:

- Weight: 1,6 kg (Measured in NX with material properties for high carbon steel type 1095)
- Width from side to side: 400 mm
- Length from front (where it is connected to the frame) to the attachment of the end links: 320 mm

- The sway bar thickness was set to 15 mm and solid. This measurement derived from literature studies on what a reasonable sway bar thickness might be depending on the vehicles weight.

Weight for the different parts derived from groups within the project, literature studies, and from different reports, these were compared with the weights of the TVS King to verify that the estimations were reasonable.
All parts were fully parameterised to lighten the workload in case a measurement had to be modified, e.g. the caster angle was parameterised with the connection plate which the horizontal part of the leading link were connected to as shown in Figure 7.4.
The trail was possible to adjust by the different holes on the connection plate. However by adjusting the trail, the length of the spring changes since it is mounted between the horizontal and vertical leading link parts. Using the middle hole and a caster angle of 19 degrees resulted in a trail of 24,2 mm.

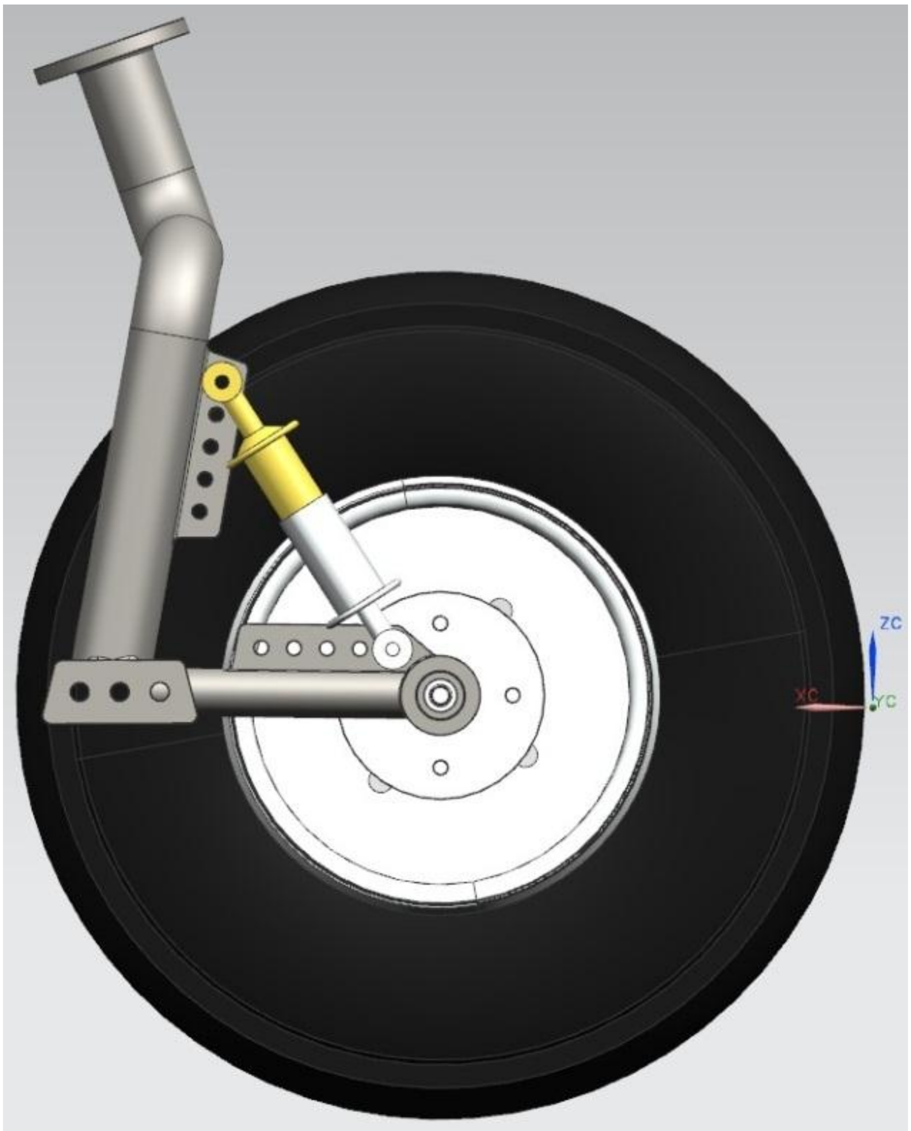

Figure 7.4: NX model of front suspension – Right side view

The modelling stage was now finished.

7.5.2 Assembly

With all the parts in order, next step was to assemble all parts to a full vehicle. By opening the assembly option within the modelling application, this was made possible. Each part was then attached to each other with the help of constraints as touch, align, infer center/axis, etc. For example the centre of the wheel had to be constrained to the centre of the wheel hub which was done with the infer center/axis constraint. In some cases more than one constraint was needed to lock a part's desired DOF:s.
The design group had a finalized structure with complicated details and all parts assembled to it. However a simplified version of this auto rickshaw model was used to keep the simulation time as low as possible. All the parts were then assembled to the simplified auto rickshaw frame with the same positioning as in the finalized structure. Therefore a simplified version could be used without affecting the result.
In addition to the measurements mentioned earlier, the positions and measurements acquired from the finalized structure and trailing arm/leading link sections (Detailed design chapter) provided some additional measures.

- Wheel base: 1979 mm
- Wheel track: 1119 mm
- No toe or camber
- Sway bar level to the ground at static conditions
- Lower leading link was angled 0°; i.e. level to the ground
- Trailing arm angled: 12 degrees
- Rear damper angle: 11 degrees
- Rear damper height vertically: 227,5 m

The data given above were set to the vehicle in steady state at full load, since that scenario is the most common one.

In Figure 7.5 a screenshot taken in NX shows the assembled rear suspension including the trailing arms, dampers, sway bar, end links and wheels.

Figure7.5: Rear suspension assembly

Next phase was to create a motion simulation of the fully assembled vehicle.

7.5.3 Motion Simulation

With the fully constrained auto rickshaw transferred over from the modelling application to Motion Simulation in NX all the constraints had to be switched out to joints. Some constraints were automatically changed to joints when switched over to the Motion Simulation application. In some connections no joints were added by the application and thus it had to be done manually.
Before starting a simulation all joints had to be looked through to ensure that the automatically converted joints were changed to the correct kind of joint for the desired movements. When redundant joints are detected in a simulation an error sign will appear, these joint functions will then be removed from the animation in contrast to Adams/Car were the animation would not even work. With the joints in place, connector features as springs and damper had to be assigned. The spring and damper connector were mere a function then a visual feature. The connectors used were attached within the damper model; created in an earlier stage, and then assigned with a linear spring stiffness coefficient and damper constant. With the correct joints distributed among the connections and springs/dampers attached to the suspensions next step was to attach connectors called 3D contact.
The 3D contact enabled the tires to be restrained to the road and contained a couple parameters that had a significant impact on the tires movement towards the road. The parameters included friction, material damping, etc. To acquire all the information about the parameters needed for the 3D contact to work properly research on Internet, NX help application and reports were carried out.
A list of the parameters, its meaning and the value assigned are shown below.

3D contact parameters:

Basic:

• Stiffness:	160 - 240 N/mm
• Stiffness Exponent:	1,1
• Material Damping:	16 - 24 N*sec/mm
• Penetration Depth:	0,1 mm
• Rebound Damping Factor:	0,25

Friction Parameters:

• Static Coefficient:	0,55 - 0,9
• Stiction Velocity:	0,1 mm/sec
• Dynamic Coefficient:	0,45 - 0,75
• Friction Velocity:	10 mm/sec

Starting by sorting out the basic parameters the Stiffness is the first one to peruse.
The software uses the Stiffness value to calculate the normal force for each unit of penetration, i.e. the harder the materials, the higher the stiffness. In the rear wheel case the parameter was taken from a report where the dynamics of the TVS King were analysed and information about the tire stiffness was found and therefore set to 160 N/mm [62]. As for the front wheel

the stiffness of a standard reference tire is about 240 N/mm (National Center for Asphalt Technology (NCAT) at Auburn University) [63].

Some of the parameters could not be set by using empirical data and therefore they were added by using the sample values provided in "Guidelines for contact materials" found in the NX Help application.
The Stiffness Exponent was one of those parameters and it compensates for the nonlinear behaviour in the contact forces. The software uses the Stiffness Exponent value when calculating the contribution of the material stiffness to the contact normal force. The value must be equal or greater than one and increasing the value increases the motion response. In this case the Stiffness Exponent was 1,1 according to the "Guidelines for contact materials".
The Material Damping sets the coefficient of maximum damping. When the penetration depth is zero, which occurs before the moment of impact, the damping is zero. The damping is increasing gradually to the Material Damping coefficient as the penetration increases to the value defined for the Penetration Depth. According to the "3D contact dialog box" found in the NX Help application the Material Damping was recommended setting in the range of 0,01 to 0,1 percent of the Stiffness value, i.e. 0,16 and 0,24 in this case. If there was too much bounce in the contact the guidelines suggested to incrementally increasing the Material Damping by a factor of 10.
Penetration Depth value controls when the solver applies the maximum damping coefficient. As mentioned earlier the solver uses the Material Damping coefficient when the penetration is equal to or greater than the threshold value for Penetration Depth. From the guidelines the Penetration Depth was set to 0,1 mm.
Rebound Damping Factor controls the damping force that occurs when the bodies interact. After contact occurs the factor reduces the damping forces when the bodies separate, but not when the bodies are being pressed together. The default value 0,25 was used for this parameter.

Next parameters to sort out were the Friction Parameters.
Different values of the friction coefficients were set depending on what kind of surfaces the simulations were going to be tested on. Friction coefficients for different surfaces including both static and dynamic values can be found in Appendix D.
Static Coefficient is the friction coefficient used by the software when the Stiction Velocity is smaller than the Friction Velocity.
The Stiction Velocity is the velocity where the static friction peaks and at velocities higher than the threshold value for Stiction Velocity the solver gradually transcends from static to dynamic friction value. According to the "Guidelines for contact materials" the Stiction Velocity was set to 0,1 mm/s.
Dynamic Coefficient describes the coefficient of friction at the contact point when the Stiction Velocity is larger than the Friction Velocity.
The specified Friction Velocity is equal to the Stiction velocity as the effective friction coefficient is set to dynamic friction coefficient. The Friction Velocity was added the velocity of 10 mm/s pursuantly to the "Guidelines for contact materials".

To make the auto rickshaw travel forward a driver parameter was needed which assigned to the revolute joint that constrained the front wheel to the wheel hub. To get the auto rickshaw to move at the desirable speed a specified constant velocity was also needed, and this was assigned in the unit degree/s. To get a more suitable unit of the velocity the following equation was used:

$$v\ (m/s) = \frac{\text{Constant Rotation Driver (degrees/sec)}}{360 * \text{circumference of the wheel (m)}} \tag{4}$$

This transformed the velocity unit from degrees/s to m/s.
The purpose of a skid pad test is to gain information on which velocity a vehicle can manage before losing control when driving in a circular track.
For the skid pad test a steer function on the front wheel was required, this was assigned to a revolute joint that connects the leading link to the frame and angled in the same direction as the caster angle. A driver was added to the revolute joint, but instead of using the constant velocity motion driver an initial displacement in degrees was used, this gave the steering angle. A rewritten equation of the law of sine was used to calculate the steering angle i.e.

$$\beta = sin^{-1}(\frac{\sin\propto}{x} * y) \tag{5}$$

where β is the steering angle, α = 90 °, x = wheelbase, and y = turning radius.

With the skid pad radius and the wheelbase of the vehicle known a steering angle could be calculated.

When joints and connectors had been added to the assembly, simulations without the sway bar could be executed. Preparations for simulations including the sway bar are described below.

7.5.4 Finite Element Simulation

As all parts modelled in NX are managed as rigid bodies a Finite Element (FE) simulation had to be carried out on the sway bar to make it behave as a real one. The FE simulation application allowed a model to be turned into a flexible body discretised by finite elements. One of sway bars objective is to allow the vehicle to corner in a higher velocity, this is made possible with a torsional twist in the fore end of the sway bar and thus transferring forces from one side to the other. The sway bar was therefore modelled as a flexible FE-model.
Nodes were placed at the connection points on each side of the sway bar where the end links were connected. The connection points were described with a function named 1D connection which provided a list of both rigid and flexible connection. The nodes were chosen to be rigid with the type Rigid Body Element 2 (RBE2). RBE2 was used to define a single grid point where the rigid body had independent degrees-of-freedom and connect it with an arbitrary number of grid points with dependent degrees-of-freedom.
Next step was to create a mesh of the sway bar where tetrahedral 3D elements of medium size 4 mm were used. The element size defines the resolution of the deformation/displacement calculations. Smaller element size gives a higher resolution and slower solve performance. On the other hand too large element size can give inaccurate results. With this information an element size of 4 mm was found reasonable. Before being able to solve this mesh two more

operations had to be accomplished; assigning material property and constraining the sway bar. Material property of the sway bar was assigned to spring steel of type AISI 1095 (high carbon steel) [64] since it is a common material used in anti-roll bars [65]. The point of making a FE analysis is to retrieve information about plausible deformations, therefore it had to be constrained in all DOF:s at the nodes.
With all pre-processing finished, a simulation was conducted where all the results were gathered in a .rfi file. In addition to the .rfi file an animation on plausible deformation scenarios with a graded colour chart was provided. As stated previously, a FE analysis informs about plausible deformations and the .rfi file provides that information which would be used in the final simulations to describe the alteration of the sway bar when cornering.

7.5.5 Motion Simulation combined with Advanced Simulation

Simulations were performed in a skid pad test to observe the behaviour of the auto rickshaw with the pre-tuned parameters and the sway bar, including the .rfi file.
First the sway bar had to be added to the assembly in the modelling application and instead of changing the constraints over to joints bushings were used since a replication of a real scenario as possible was desirable. The bushings were found in the same tag as springs, dampers and 3D contact i.e. Connector. Bushings were added to the sway bar - end link connections and at the front end of the sway bar to constrain it to the auto rickshaw frame. The stiffness for these bushings was retrieved from the "*Automotive suspension engineering Forum*" [66] and the damping coefficient from the report analysing the vehicle dynamics of the TVS King. The stiffness of modern bushings used for connecting sway bars were found to have a value of 5000 N/mm which was used both for radial and longitudinal directions. The bushings described in the report were used for the trailing arms on the TVS King and had a damping coefficient of 98 N*sec/mm which was used in both directions.
With the sway bar constrained the .rfi file could be added. To be certain that the .rfi file was added a mesh pattern embracing the sway bar was shown.

7.5.6 Calibration

The calibration tests were performed with a bump response test on a standardized proving ground which is the same as Hankook Tire Co. in Korea uses. Sung et. al. [67] presented bump response tests at a velocity of 30 km/h on a proving ground with bumps with the height of 70 mm and width of 800 mm.
The calibration tests were performed with the centre of gravity for full load; i.e. one driver and four passengers. No sway bar was used since the time to solve the simulation would increase significantly while the results should not be affected drastically.
Friction values used in this test was a combination of earth road (dry) and dry gravel, which is defined in static and dynamic coefficients (see Appendix D). Those road types were chosen since auto rickshaws are commonly driven on them.
Friction coefficients used in this case were:

- Static coefficient: 0,63
- Dynamic coefficient: 0,6

7.5.6.1 Calibration of 3D contact parameters

Front "tire properties" (3D contact parameters):

- Stiffness: 240 N/mm
- Stiffness Exponent: 1,1
- Material Damping: 24 N*sec/mm
- Penetration Depth: 0,1 mm
- Rebound Damping Factor: 0,25
- Stiction Velocity: 0,1 mm/sec
- Friction Velocity: 10 mm/sec

Rear "tire properties" (3D contact parameters):

- Stiffness: 160 N/mm
- Stiffness Exponent: 1,1
- Material Damping: 16 N*sec/mm
- Penetration Depth: 0,1 mm
- Rebound Damping Factor: 0,25
- Stiction Velocity: 0,1 mm/sec
- Friction Velocity: 10 mm/sec

As shown in the list Material Damping gained the values of 24 N*sec/mm respective 16 N*sec/mm. These values were discovered after calibration test as the initial values gave too much bounce in the contact. Through the "Guidelines for contact materials" the Material Damping was multiplied twice with a factor of 10.
The parameters listed above were used in all tests.

7.5.6.2 Calibration of springs

The spring stiffness coefficients were calculated with MATLAB from knowing the maximum weight the vehicle is intended to carry together with the distance between spring free length and spring initial length. The spring initial length is the spring length in steady state for full load. The rear spring free length was set to 210 mm and the rear spring initial length was set to 180 mm i.e. a difference of 30 mm. The same procedure was carried out for the front spring where the free length of 144 mm and a spring initial length of 114 mm gave a distance of 30 mm. Calculations for the spring stiffness coefficient done in MATLAB can be found in Appendix C.
The spring parameters were hence set to:

- Spring stiffness coefficient:
  - Front: 60 N/mm
  - Rear: 60 N/mm
- Spring free length:
  - Front: 144 mm
  - Rear: 210 mm

In Figure 7.6 a static wireframe picture shows a side view of the front suspension in steady state.

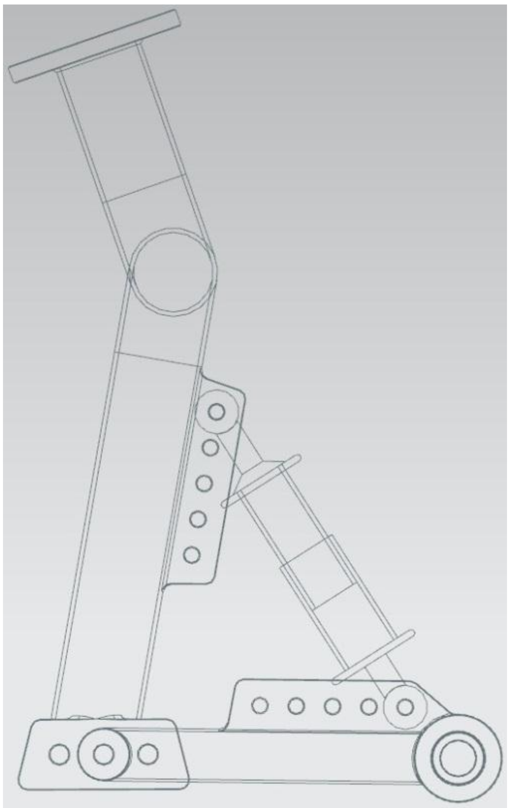

Figure 7.6 – Side view of front suspension in steady state

Total spring travel is the distance the spring can travel from its free length to fully compressed state. With the known distance of the maximum possible spring travel, the minimal length of spring in compressed state should be the same as the longest piston and if possible it is preferred to add an additional length to prevent the upper spring bracket from colliding with the lower damper piston. The piston lengths deciding the front respective the rear were 75 mm and 107 mm.

For the front the total spring travel was from 144 mm - 75 mm which gave a travel of 69 mm and for the rear it was 210 mm - 107 mm; i.e. 103 mm travel.

To calculate the spring coil turns and coil diameter which defines the fully compressed damper an equation was needed. The diameter of the spring:

$$D = \sqrt[3]{\frac{Gd^4}{8kn}} \qquad (6)$$

where d is the coil diameter, n is the number of coil turns, k is the spring stiffness coefficient and G is the shear modulus [68].

The already known essential parameters were the reasonable spring outer diameter, the spring stiffness coefficient and the information on the defining piston length. The shear modulus for AISI 9260 is 190-210 GPa [69]. AISI 9260 is a type of high carbon steel commonly used in automotive coil springs [70].

With the known essential parameter mentioned above the number of coil turns and coil thickness could be computed. According to the equation two parameters were missing; number of coil turns and diameter of the coil. To acquire both parameters a trial and error run had to be conducted. The minimum compressed spring length is given by multiplying the number of coil turns with the coil diameter. A value closest possible to the chosen spring outer diameter could be retrieved with different combinations of coil turns and coil diameters.
The parameters used to calculate the number of coil turns and spring coil thickness were:

- Minimum compressed length:
  - Front: 75 mm
  - Rear: 107 mm
- Spring outer diameter:
  - Front: 55 mm
  - Rear: 72 mm
- Spring coefficient:
  - Front: 60 N/mm
  - Rear: 60 N/mm
- Shear modulus:
  - Front: 200 GPa
  - Rear: 200 GPa

This together with equation (6) gives:

Front: $$D = \sqrt[3]{\frac{(2*10^{11}Pa)*0{,}007^4 mm}{8*60\frac{N}{mm}*11}} = 44{,}97\ mm \qquad (7)$$

Spring outer diameter = 44,97 mm + 7 mm = 51,97 mm ≈ **52 mm** < 55 mm

Rear: $$D = \sqrt[3]{\frac{(2*10^{11}Pa)*0{,}009^4 mm}{8*60\frac{N}{mm}*12}} = 61{,}07\ mm \qquad (8)$$

Spring outer diameter = 61,07 mm + 9 mm = 70,07 mm ≈ **70 mm** < 72 mm

The trial and error runs resulted in the following number of coils turns and coil diameter and thus also the compressed length:

- Coil turns
  - Front: 11
  - Rear: 12
- Coil diameter:
  - Front: 7 mm
  - Rear: 9 mm
- Compressed length:
  - Front: 77 mm > 75 mm
  - Rear: 108 mm > 107 mm

The compressed spring length values were kept within the stated values and gave the front a margin of 2 mm and the rear a margin of 1 mm.
To verify that the springs were assigned an appropriate spring stiffness coefficient value a test with static conditions was carried out. Static condition is when the vehicle is standing still and only gravity is affecting the system. In this test the gravity were applied on the vehicles centre of gravity and forced the springs to its steady state in the full load case. When starting the simulation the gravity will immediately be applied and thus pull the vehicle towards the ground. The springs will then counteract the downward movement and push the vehicle upwards which results in the spring oscillating until the system reaches steady state. From the test the displacement of the springs could be plotted and analysed in a graph to ensure that the calculations were correct.
Figure 7.7 shows the displacement of the front spring while Figure 7.8 shows both rear spring displacements.

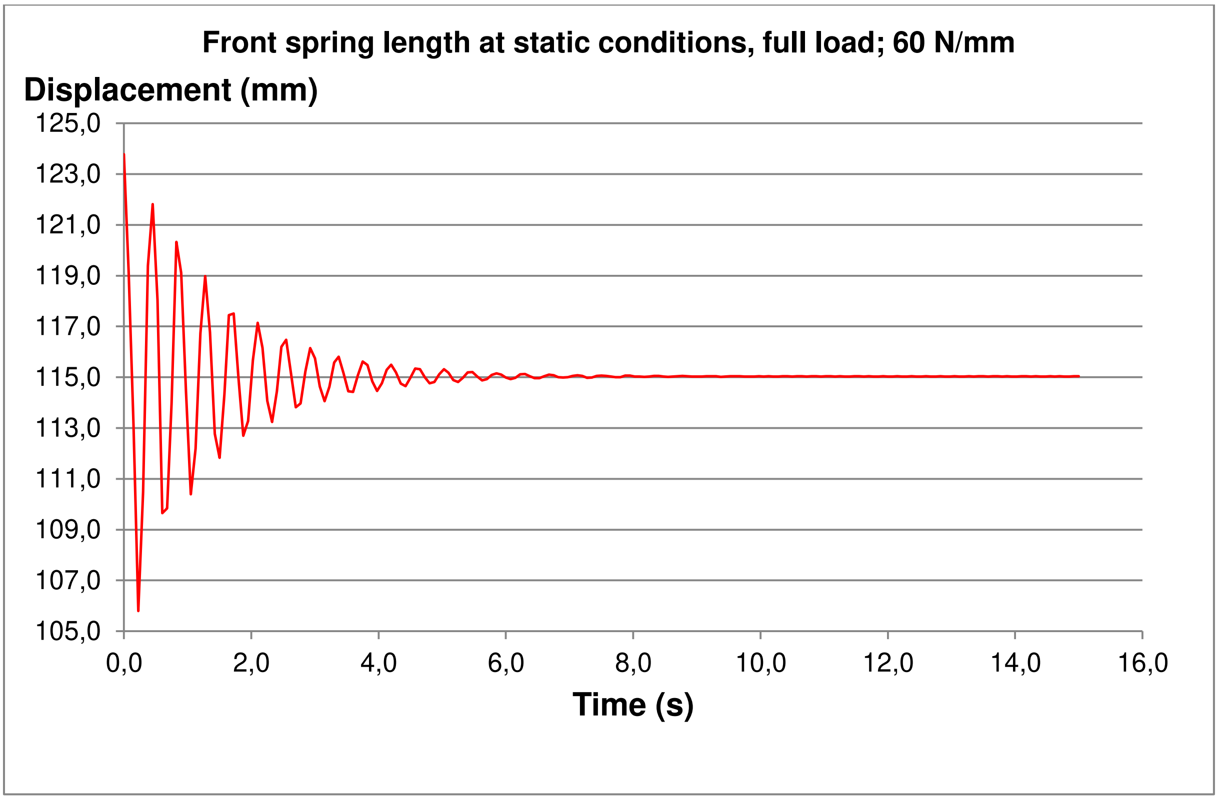

Figure 7.7: Front spring length without damper at static conditions with full load

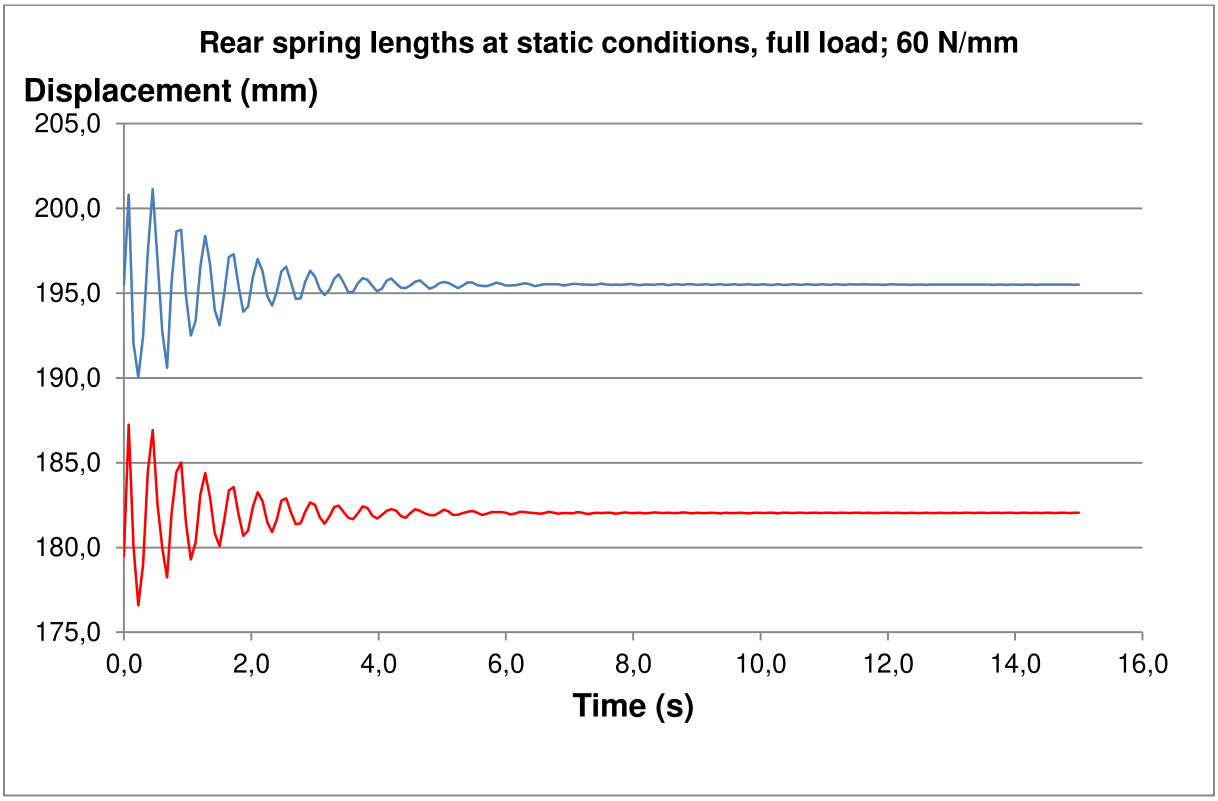

Figure 7.8: Rear spring lengths without dampers at static conditions with full load

Deduced from Figure 7.7 and 7.8 the front had a displacement of 115 mm, the rear right spring (blue line) had a displacement of 195,5 mm and the rear left spring (red line) had a displacement of 182 mm at steady state.

The values from the test differed a little from the calculations and the reason for that the springs were seen as single parts carrying a weight and not as three interacting springs. Although there is more weight distributed to the right side the rear right spring has larger displacement than the left one. That is because both the rear right and front spring (which is mounted to the right side of the wheel) are interacting and partially carrying the weight distributed to the right side of the vehicle.

7.5.6.3 Calibration of dampers

In this test the dampers are calibrated with linear properties. Linear damping was used since the solving time would increase considerably otherwise.
The test was carried out by driving on the proving ground as mentioned above, i.e. 70 mm high and 800 mm wide bumps at a velocity of 30 km/h. Graphs were plotted on the movement of the spring travel to see how fast front and rear reaches steady state; i.e. its initial length after driven over a bump.

Parameters essential for this simulation to calibrate the dampers are:

- Spring free length:
  - Front: 144 mm
  - Rear: 210 mm
- Spring stiffness coefficient:
  - Front: 60 N/mm
  - Rear: 60 N/mm
- Damper constant:
  - Front: 19 Ns/mm
  - Rear: 2 Ns/mm

A too high damper constant will counteract the springs' task to reach steady state slower than critical damped and a too low damper constant will allow the spring to oscillate with gradually decreasing amplitude.
Figure 7.9 shows a graph where the front spring had difficulties quickly reaching its steady state position after running over the bumps. The front damper counteracted the springs desire to reach its steady state fast which indicated that the system was overdamped. The rear damper showed underdamped characteristics as it oscillated which is shown in Figure 7.10 where the blue line is the right and red line is the left suspension.

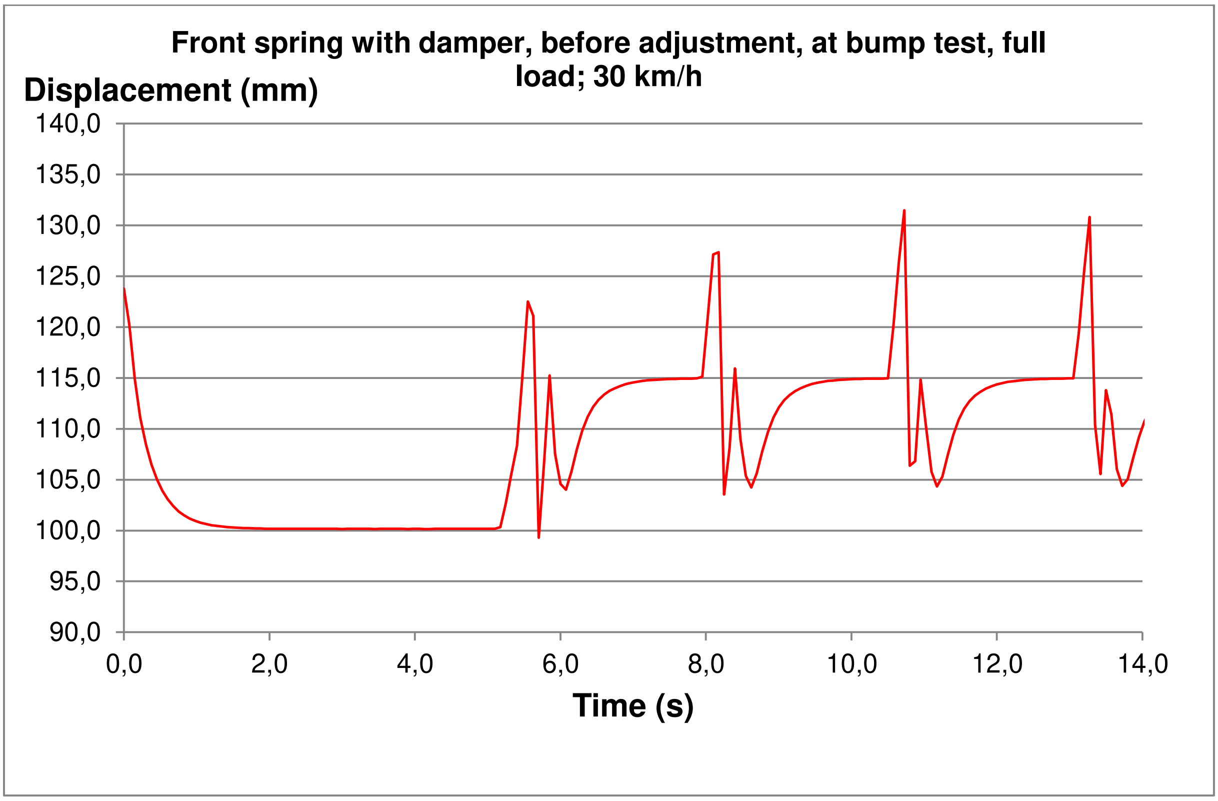

Figure 7.9: Overdamped front suspension with full load at 30 km/h

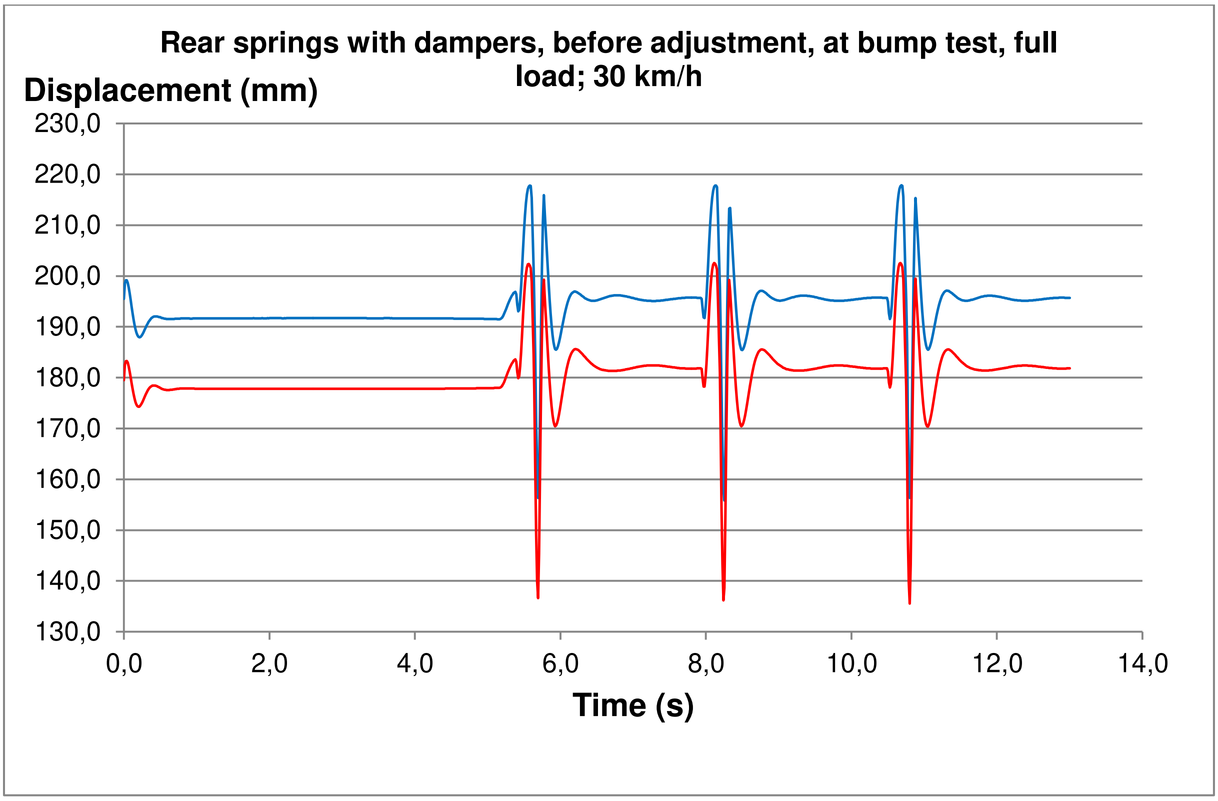

Figure 7.10: Underdamped rear suspension with full load at 30 km/h

After some trial and errors the critical damped values were derived to 15 Ns/mm for the front and 15 Ns/mm for the rear dampers.
The spring travel with critically damped values are shown in Figure 7.11 and Figure 7.12, where Figure 7.11 shows the front spring travel and Figure 7.12 shows both the right and left rear spring travel.

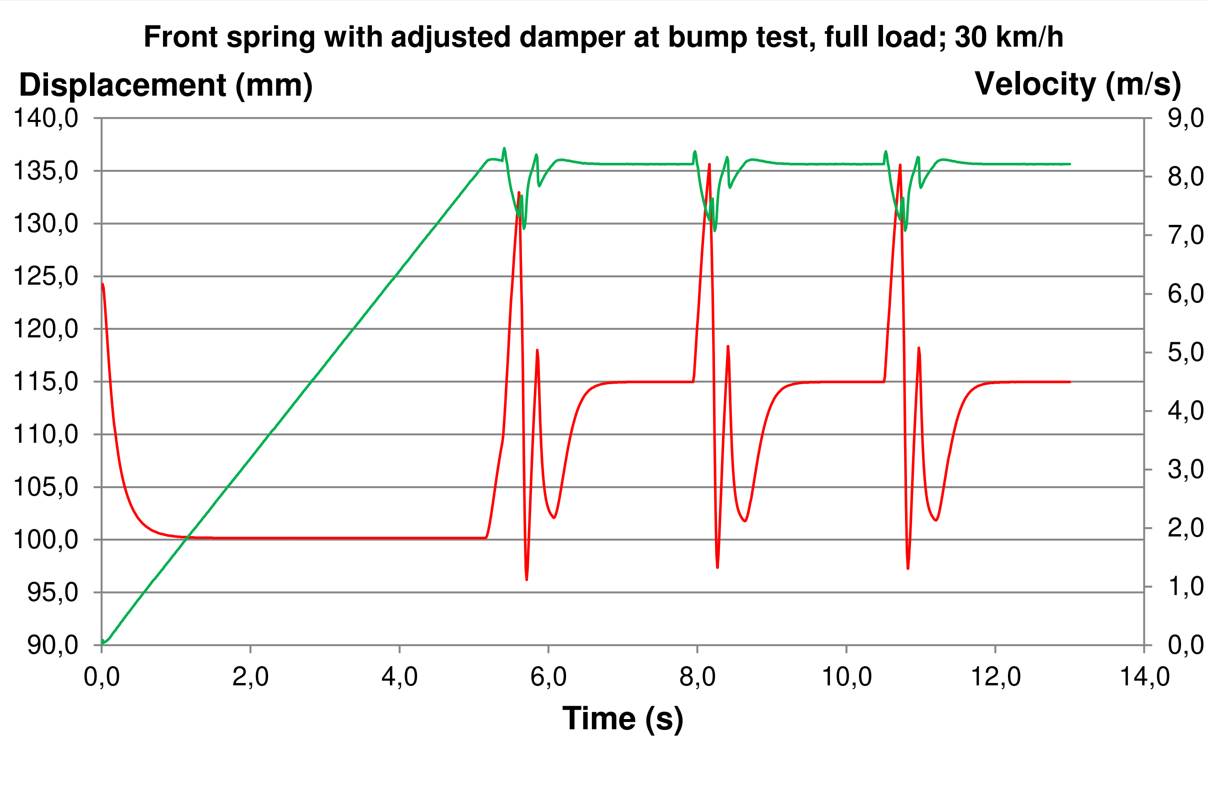

Figure 7.11: Critical damped front suspension with full load at 30 km/h

In Figure 7.11, the velocity is shown as a green line, and the suspension travel for front is shown as a red line.

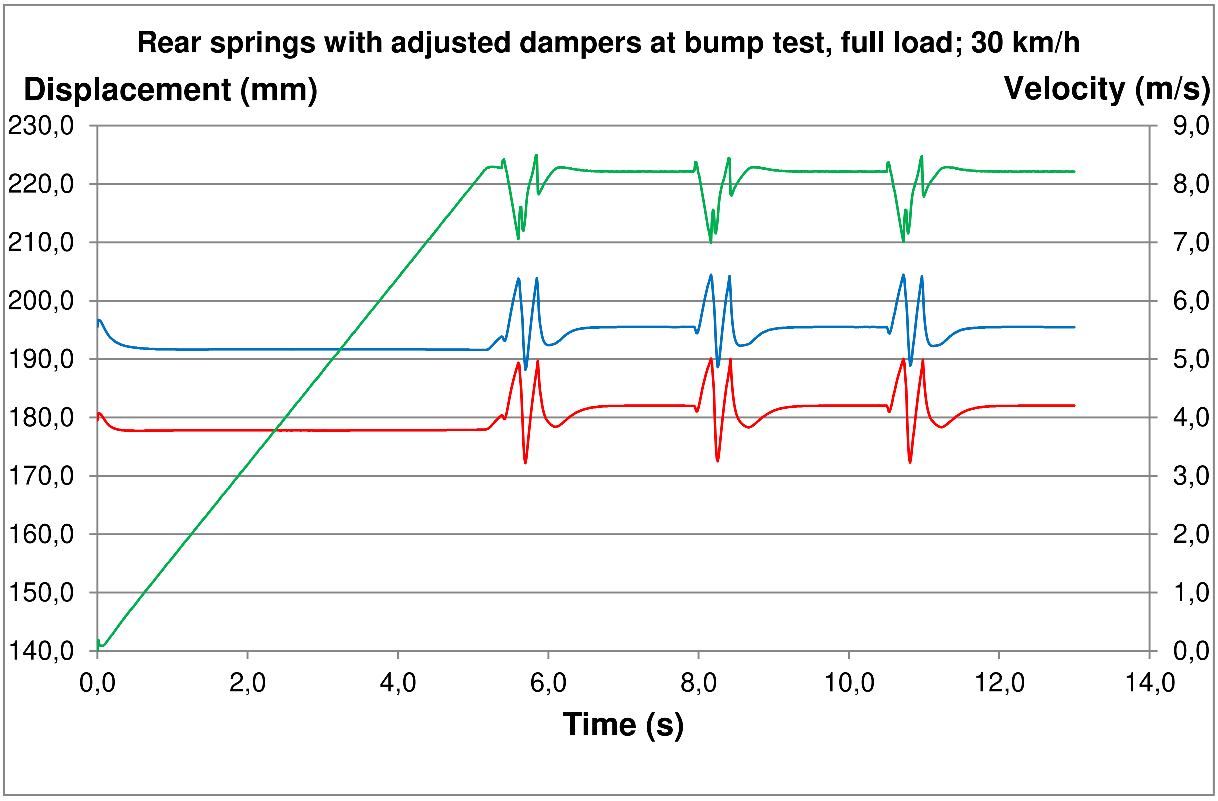

Figure 7.12: Critical damped rear suspension with full load at 30 km/h

Figure 7.12 shows the travel for the rear where the green line is the velocity, red is left suspension travel and blue is the right suspension travel.

As shown in Figures 7.11-7.12 the springs reach their steady state right before the first bump. One reason was the inflict of gravity which are shown in the beginning of both figures as the lines declines very fast, within a couple of milliseconds. Another reason was that during the acceleration of the vehicle to its input velocity of 30 km/h the springs were kept more compressed. When the vehicle reached the velocity of 30 km/h the springs rose to their steady state.

7.5.7 Verification

With both the springs and damper calibrations finished different tests were performed to verify that these calibrations coincide with other real actions within reasonable measures. These verification tests were conducted under different conditions as for example another load case and a different road type.
Friction values used in this test were the same as in the calibration tests.

Verification test 1 - Suspension travel with minimal load

This verification test was performed on the proving ground with the 70 mm high and 800 mm wide bumps to ensure that the springs and damper worked reasonably and that damping stayed around critically damped even if the sprung mass changed. The test included only the driver seated which meant a lower sprung mass and with hence another position for the centre of gravity.
The centre of gravity calculation was made with the existing MATLAB script used for minimal load (see Appendix B: V).

Figure 7.13 and 7.14 indicated that the suspension kept critical damping since the springs reached steady state relatively fast. Figure 7.13 shows that the front suspension got a higher steady state position than in the calibration tests since the sprung mass was decreased.

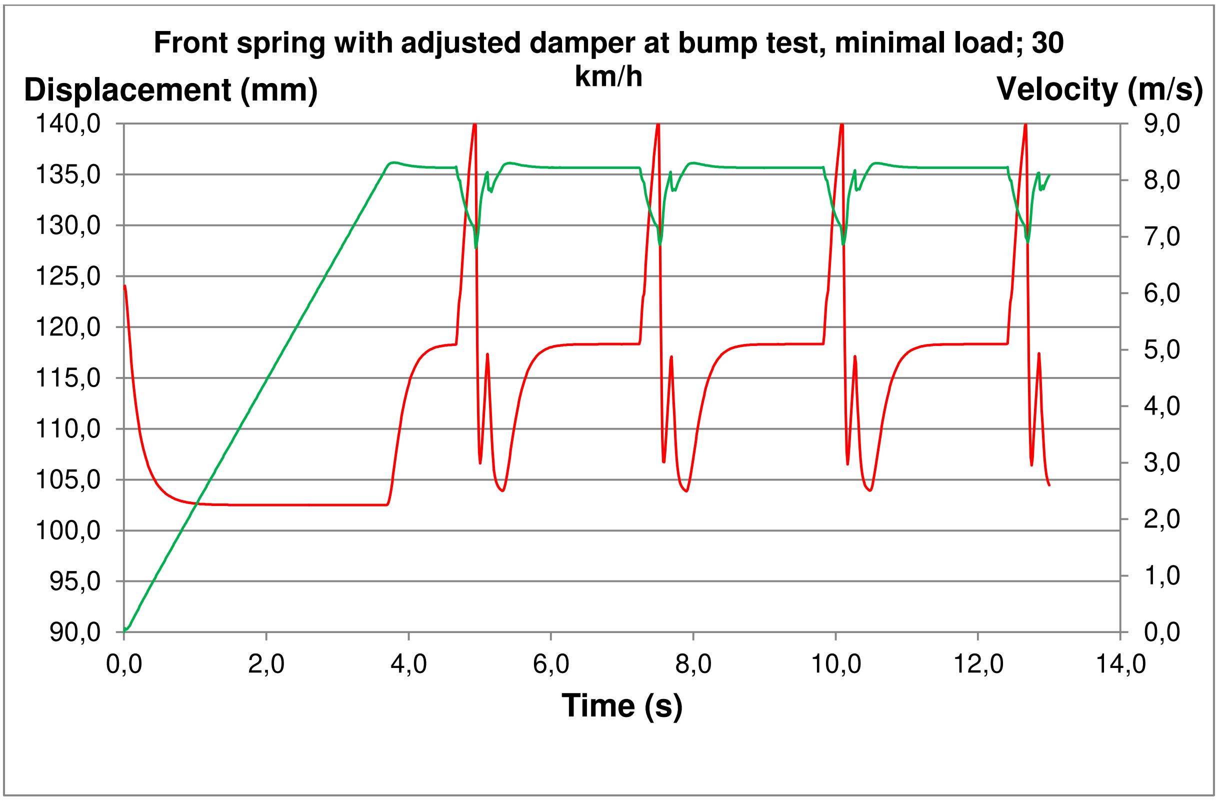

Figure 7.13: Critical damped front suspension with minimal load at 30 km/h

The green line represents the velocity and the red line represents the front suspension travel.

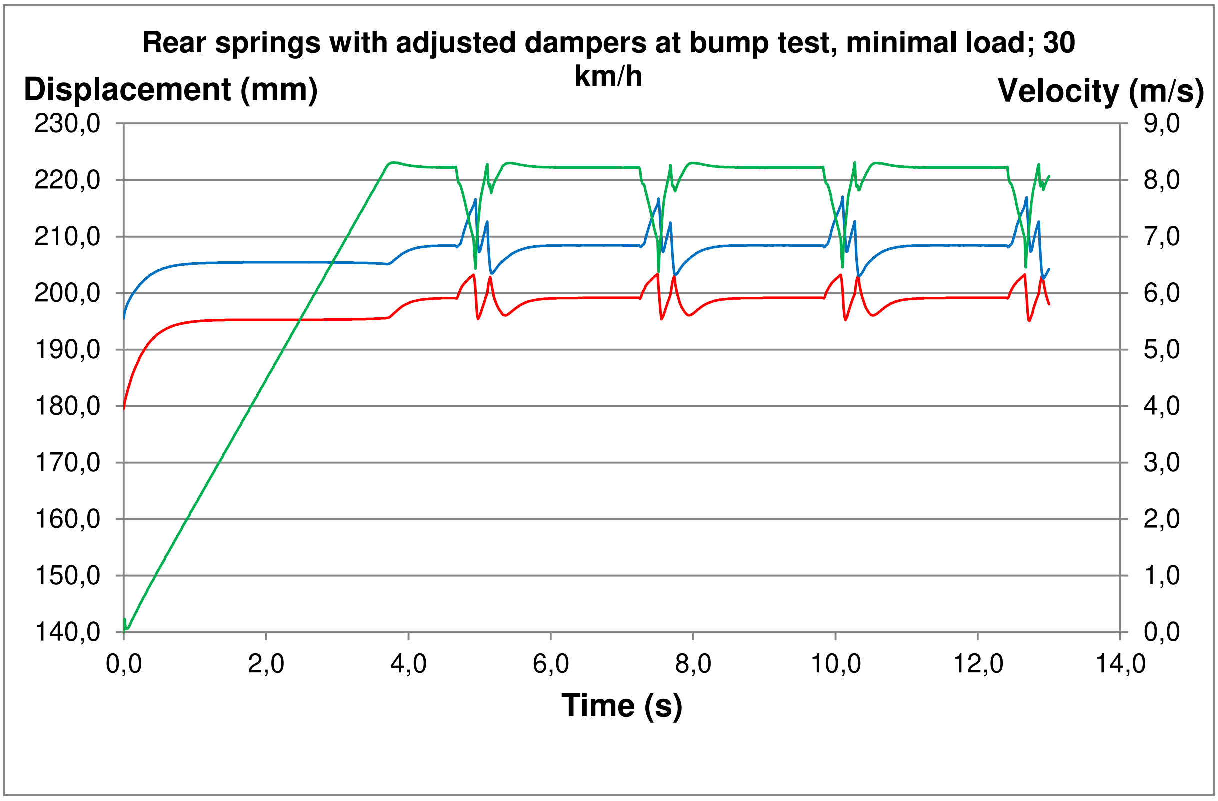

Figure 7.14: Critical damped rear suspension with minimal load at 30 km/h

In Figure 7.14 the green line represents the velocity, blue line represents the rear right suspension travel and the red represents the left suspension travel.

The reason for the quick increase in displacement at the beginning is the decreased sprung mass than for the load case that the system was designed for. Because of the low sprung mass the vehicle managed to reach the increased steady state position earlier than in the case of full load.

Verification test 2 - Suspension travel on rough road

The second verification test was made to ensure that the suspension performed well on a different test road. This test was carried out on a road with smaller bumps; 40 mm high and 300 mm wide, and with a larger amount of bumps with smaller distance between them than on the proving ground. The larger amount and smaller bumps were chosen since the aim was to build a road that would resemble a rough road. These verification tests were performed with the same velocity as in the previous tests; 30 km/h, and with the centre of gravity for both minimal and full load.

Starting with the full load case Figure 7.15 indicates that the front suspension still performs well as the transient quickly die out after the last small bumps. After the small bumps at the end of the test, the auto rickshaw hits the larger 70 mm bumps again. Figure 7.16 shows the same kind of result for the rear suspension.

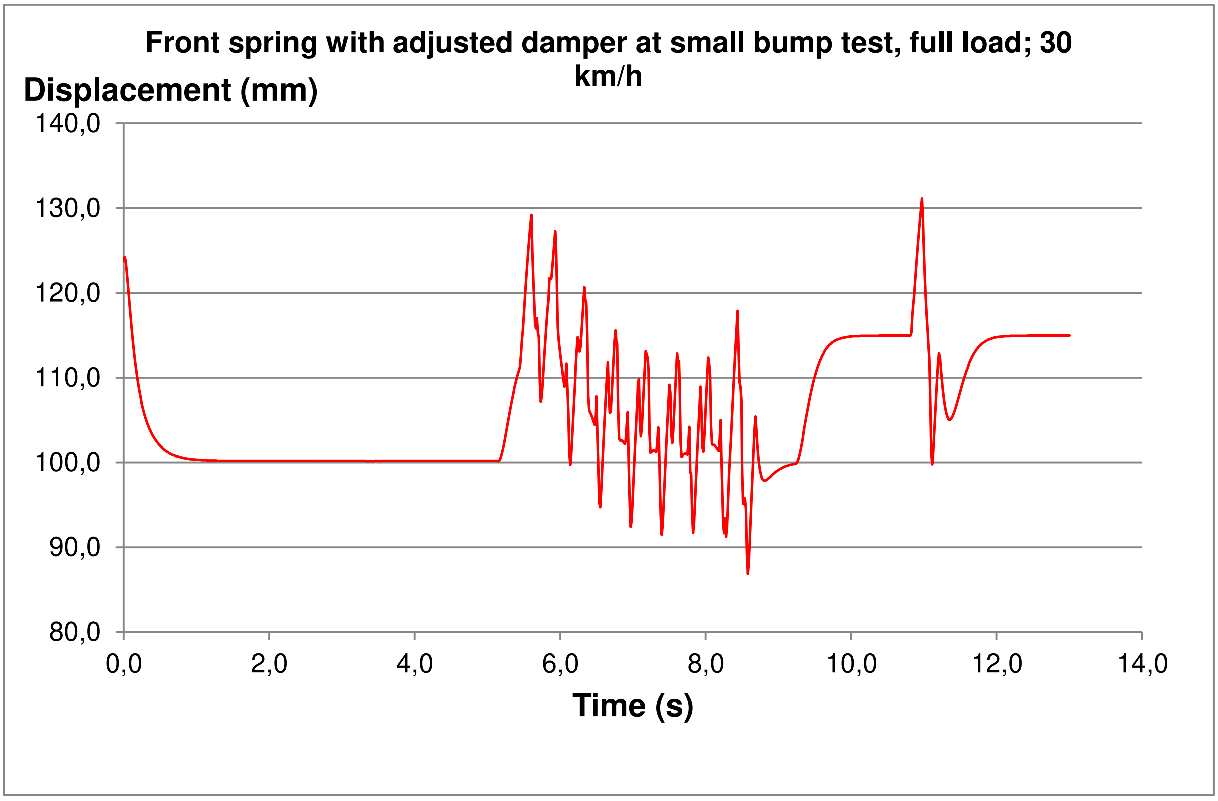

Figure 7.15: Front suspension travel on rough road with full load at 30 km/h

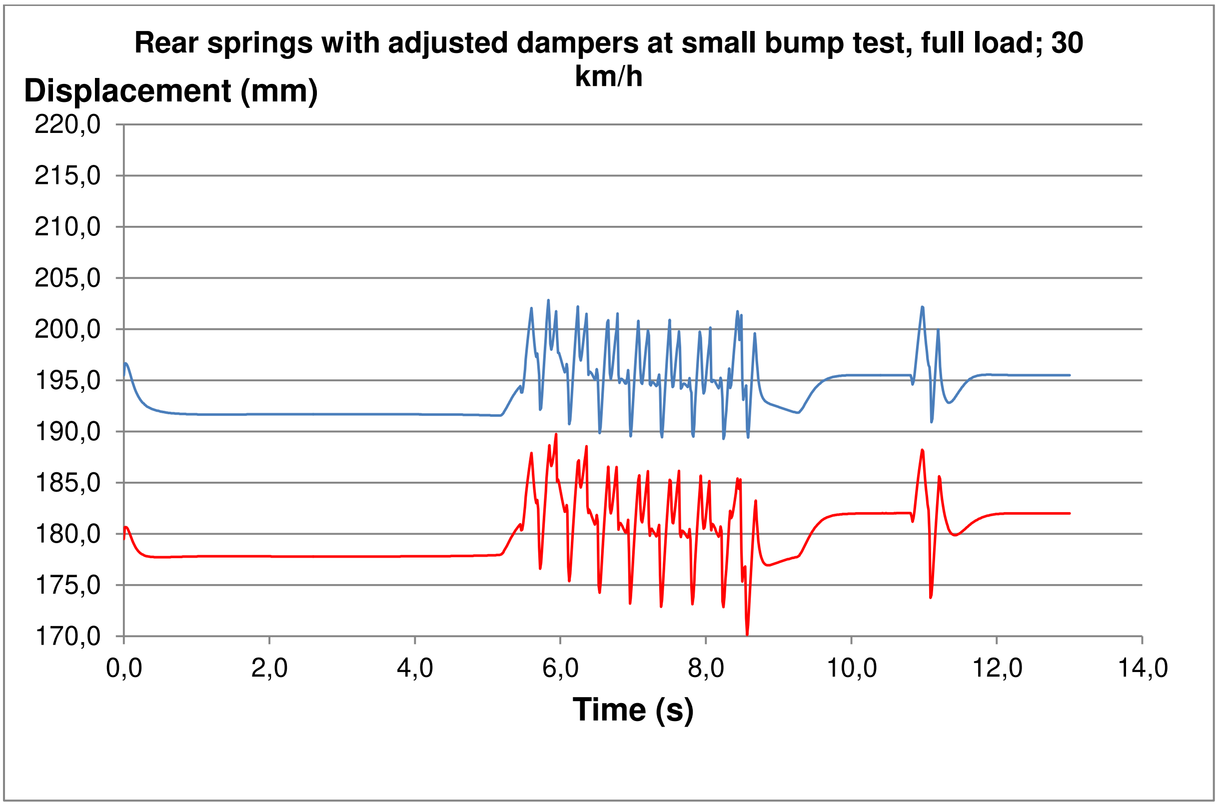

Figure 7.16: Rear suspension travel on rough road with full load at 30 km/h

Blue line represents the rear right suspension travel and red represents rear left suspension travel.

To be sure that the suspension performed as desired another test with minimal load was carried out. The results from those tests are shown in Figure 7.17 for the front and Figure 7.18 for the rear suspension.

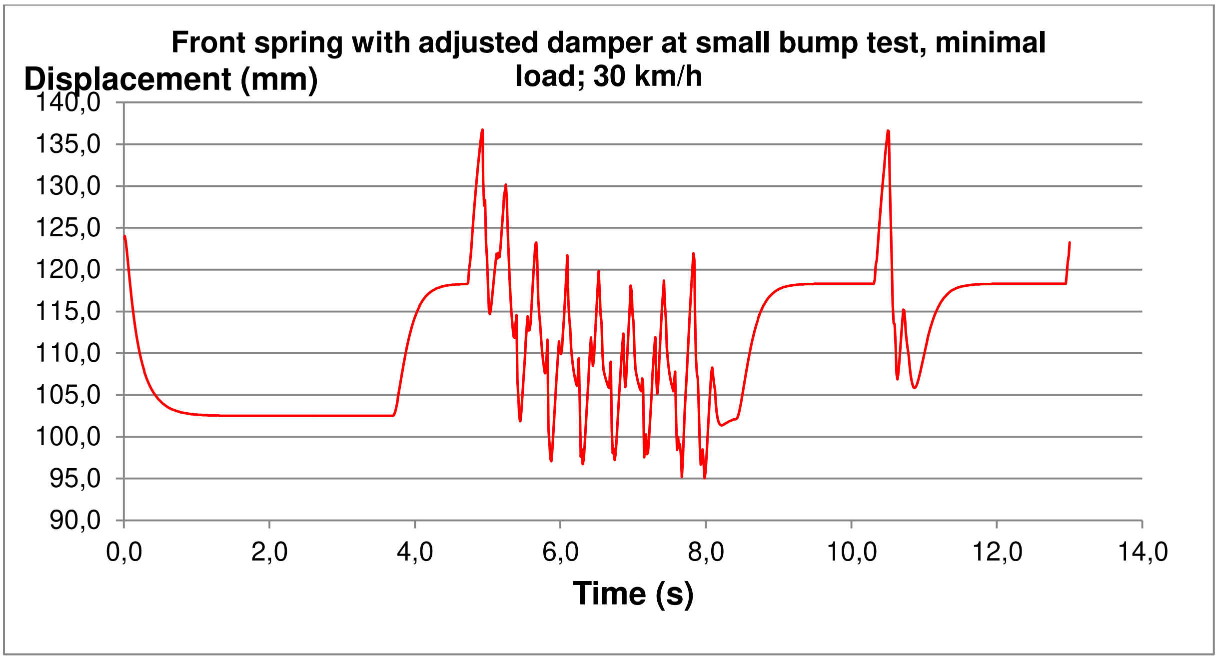

Figure 7.17: Front suspension travel on rough road with minimal load at 30 km/h

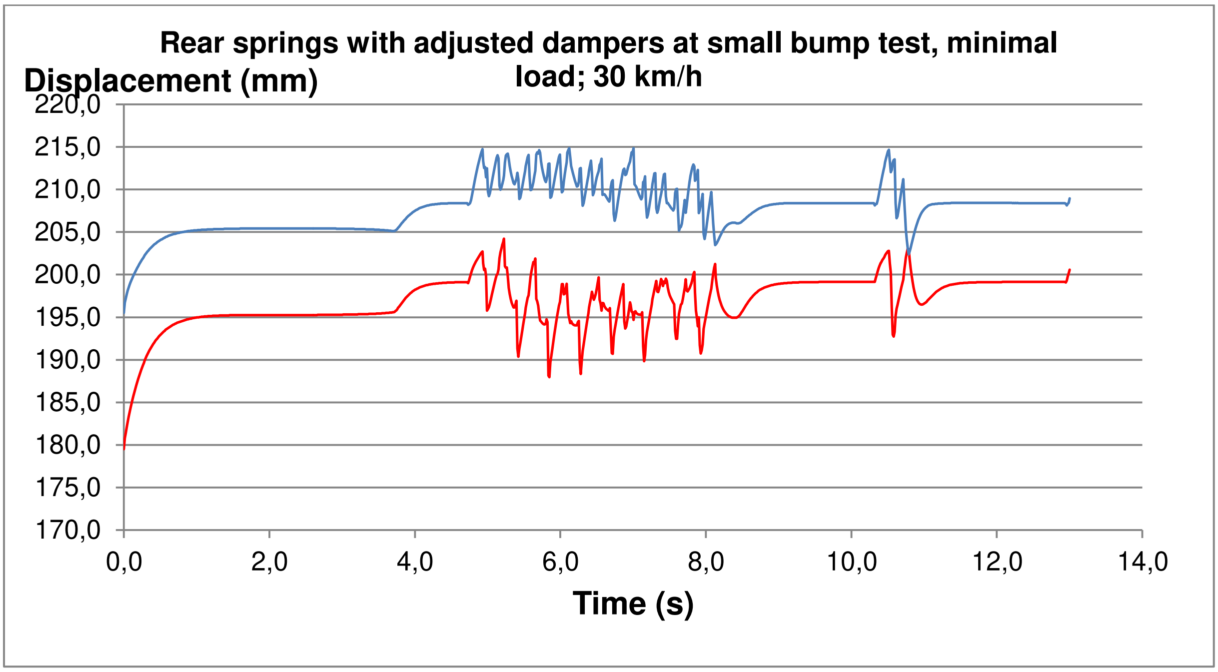

Figure 7.18: Rear suspension travel on rough road with minimal load at 30 km/h

The blue line represents the rear right suspension travel and red represents rear left suspension travel.

Both front and rear suspension showed good damper characteristics.

Verification test 3 - Suspension test with sway bar and full load

A final verification test was performed with the sway bar included in the model. The test was performed on the same proving ground as the one used in "Verification test 1" and with the same velocity of 30 km/h. The reason for this test was to ensure that the sway bar did not have any negative effect on the suspension when driving in a straight path over the bumps. Since the test is performed on a road that will execute the same amount of force on both the rear wheels when driving straight, the sway bar should not influence the result.

Figure 7.19 and 7.20 shows the result at fewer steps (100 compared to 1000 steps) than used for the damper calibration test. Comparing the results with and without the sway bar (Figure 7.11 and 7.12), no differences were found. Fewer steps were used because of the time issue for solving a simulation mentioned earlier.

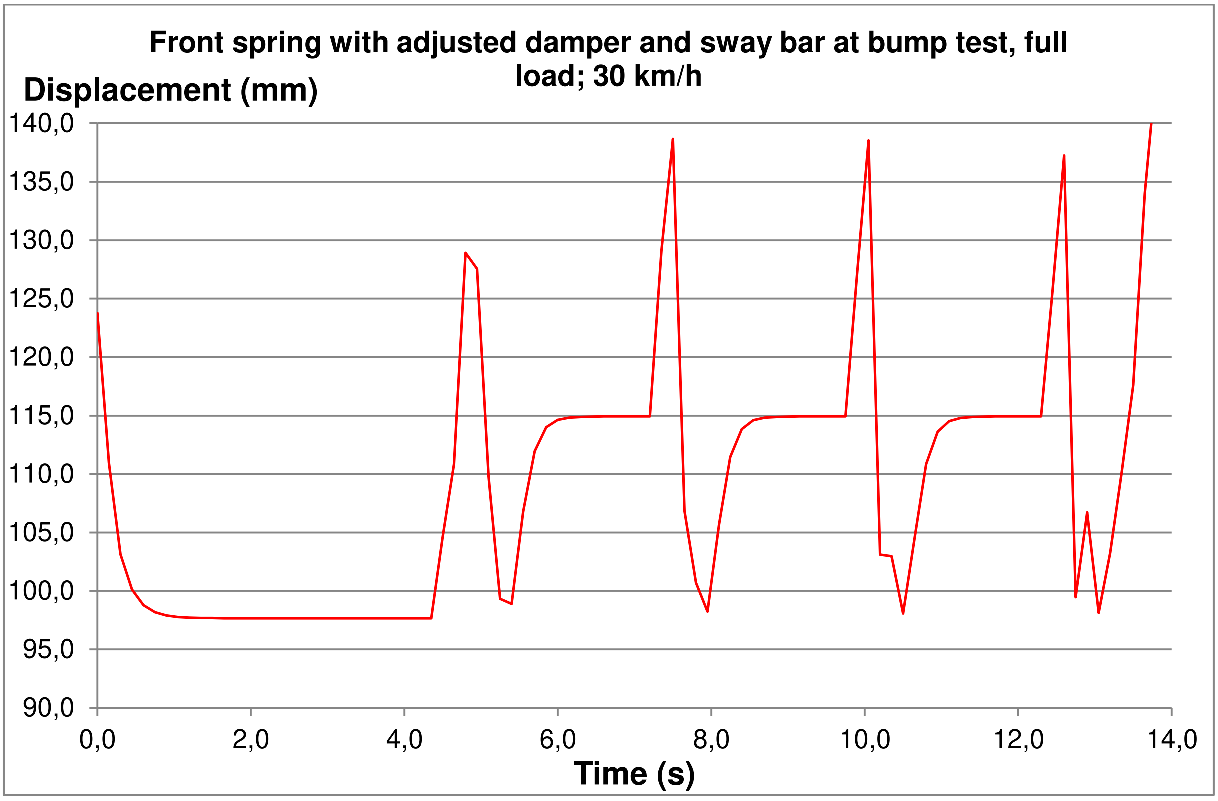

Figure 7.19: Front suspension travel including sway bar with full load at 30 km/h

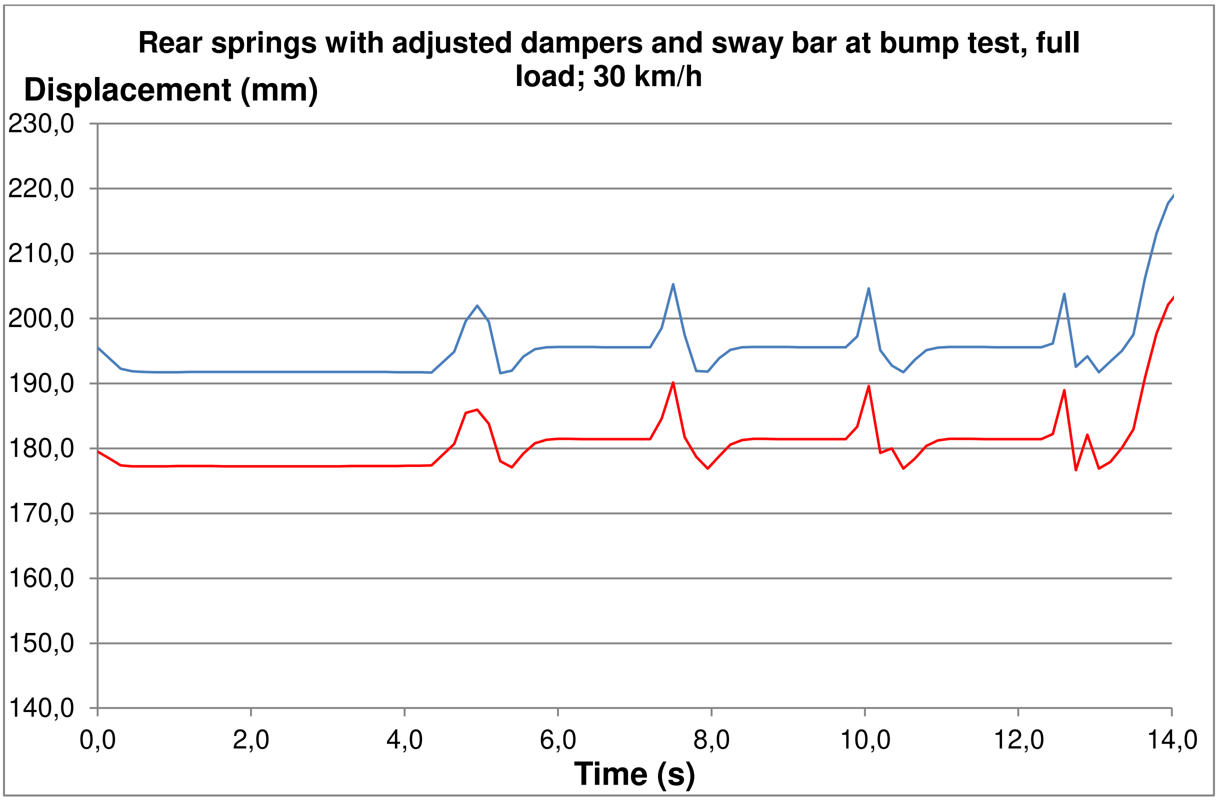

Figure 7.20: Rear suspension travel including sway bar with full load at 30 km/h

Blue line represents the rear right suspension travel and red represents rear left suspension travel.

All verifications indicated that the suspension would perform well under most common conditions.

7.5.8 Auto rickshaw performance tests

In this section simulations of different conditions were carried out to study and review the performance of the auto rickshaw. The first simulations conducted were the ramp tests described below.

7.5.8.1 Ramp tests

From the specification requirements list the vehicle must manage to start from standstill and drive up a slope with seven degrees inclination. These tests started at the bottom of the ramp with the front wheel close to the ramp. The front wheel was assigned an initial velocity of five km/h which represent the same as starting on the ramp at a low velocity. All the ramp tests were conducted without the sway bar due to the slow solving process issue mentioned earlier. Although no sway bar was used it should not affect the results since there were no differences in the normal forces acting on the rear tires under these tests.
Figure 7.21 shows a screenshot of the auto rickshaw as it is going to start up the ramp.

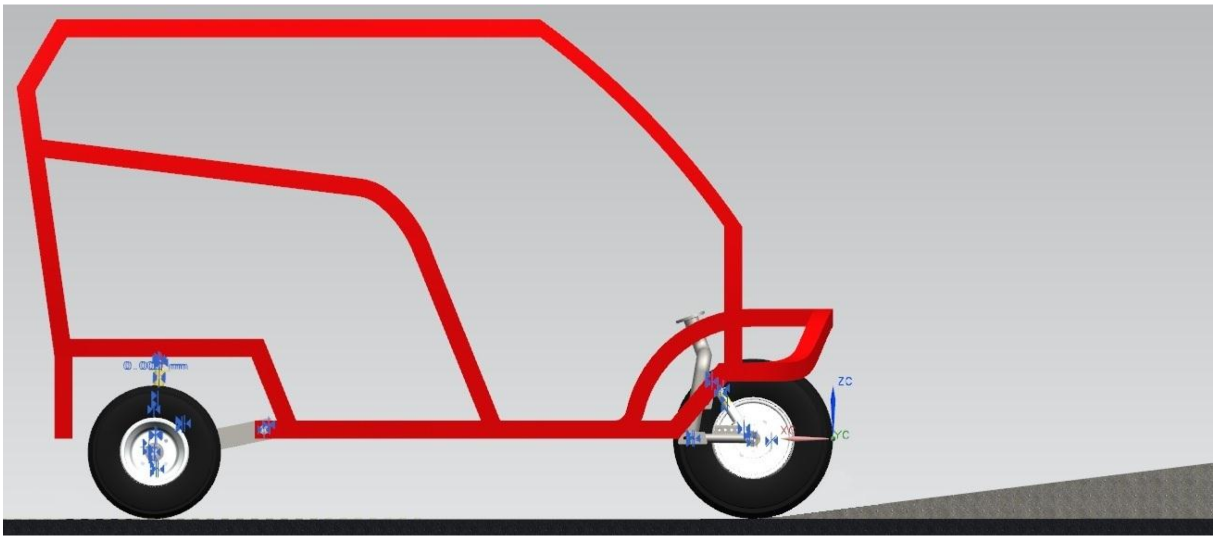

Figure 7.21: Side view of auto rickshaw before starting to drive up the ramp

Test 1 - 7˚ inclination with full load on wet earth road

The first ramp test was conducted on a wet earth road surface with full load. Wet earth road is the surface with the lowest friction coefficient that an auto rickshaw commonly will be used on. Frictions coefficients for wet earth road can be found in Appendix D. The crucial parameters used in this test were:

- Full load
- Ramp angle: 7 degrees
- Velocity: ~5 km/h
- Friction coefficients:
    - Static: 0,55 (wet earth road)
    - Dynamic: 0,4 (wet earth road)

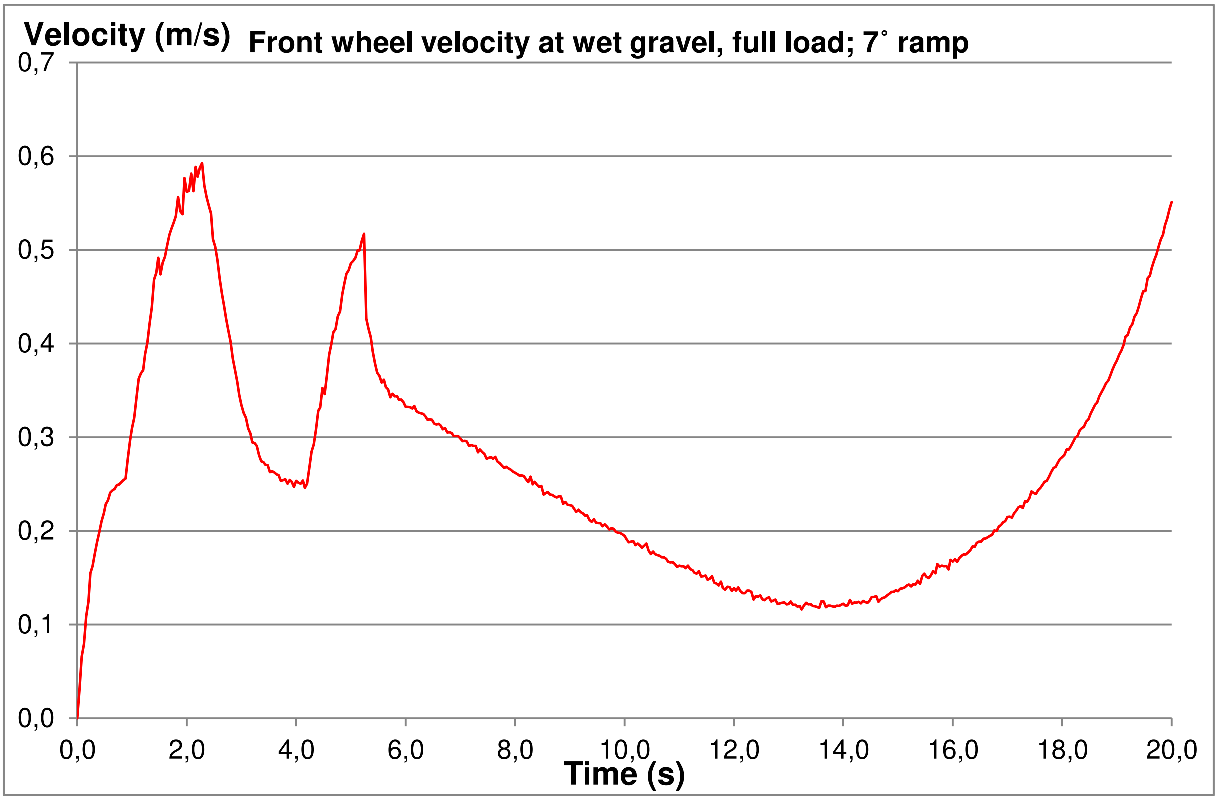

Figure 7.22: Front wheel velocity on 7˚ ramp with full load at wet gravel

As shown in Figure 7.22 the vehicle did not manage to drive up the seven degree slope and instead side-slipped. Figure 7.22 shows the front wheels velocity forward in the wheel's direction. Given from the graph a steady increase of the velocity occurs until both rear wheels roll onto the ramp which is represented by the first peak. After the first peak in the graph a steady decrease occurs until the front wheel gains a small increase in velocity. At 5,2 seconds the second peak turns up which represented the front wheel losing its forward momentum that the friction generated, and starts to side-slip. This event ends when the front wheel gains enough momentum to increase in speed again which is when the auto rickshaw points to the side of the ramp. The increase of forward momentum occurs at that point because the front wheel does not have to pull the auto rickshaw up the ramp.

Test 2 - 6° inclination with full load on wet earth road

This test was based on a follow up of the previous test, since the auto rickshaw did not manage to drive up a seven degree slope. A ramp with an inclination of six degrees was used instead while all other parameters were remained.

- Full load
- Ramp angle: 6 degrees
- Velocity: ~5 km/h
- Friction coefficients:
  - Static: 0,55 (wet earth road)
  - Dynamic: 0,4 (wet earth road)

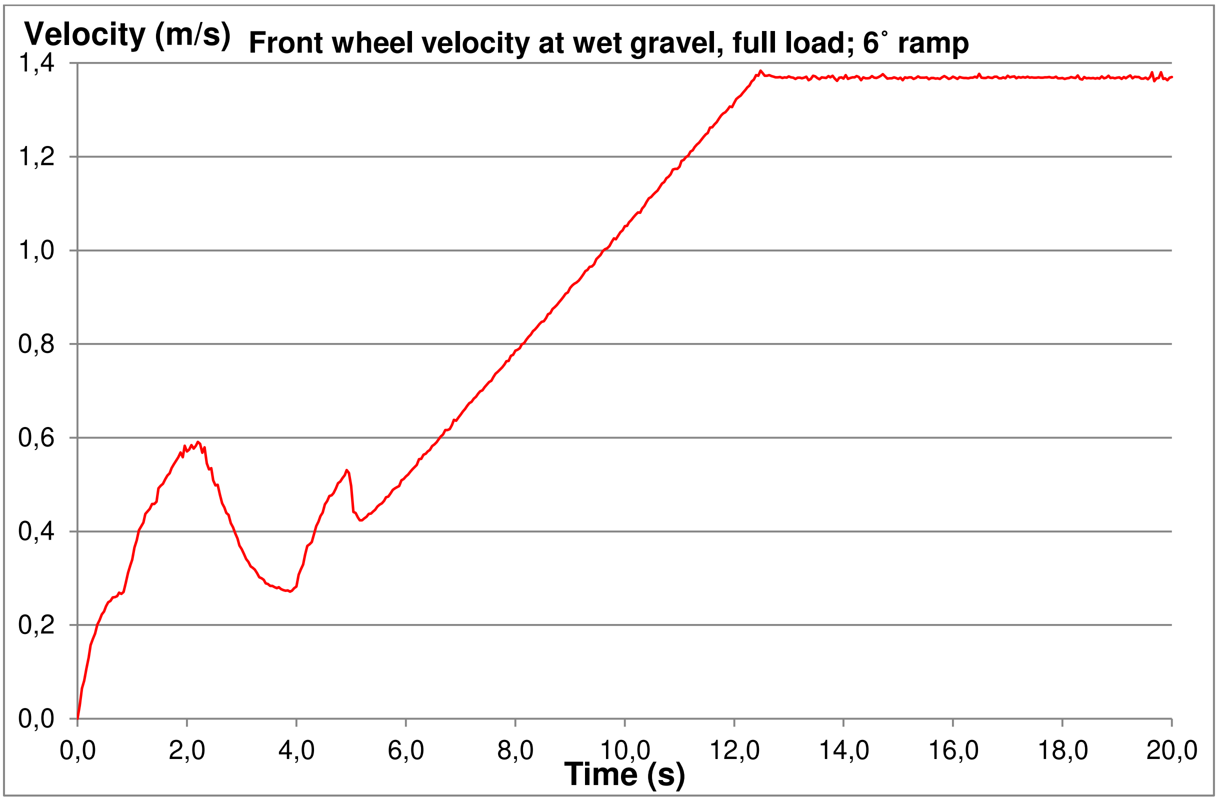

Figure 7.23: Front wheel velocity on 6° ramp with full load at wet gravel

Figure 7.23 shows that the auto rickshaw managed to drive up with some difficulties in the beginning. The auto rickshaws velocity increased slowly until the rear tires entered the ramp and a quick decrease in velocity occurred since it gained more weight to pull upwards which is the same behaviour as in the previous test. Thereafter the vehicle climbed up the slope in a steady pace until it after 12,5 seconds reached the predetermined velocity of 5 km/h.

Test 3 - 7˚ inclination with full load on dry gravel

Another test was regarded whether the auto rickshaw managed a 7 degree slope with the friction coefficient of dry gravel which was the surface used in the earlier test as it is a common surface that the auto rickshaws likely will be used on.

- Full load
- Ramp angle: 7 degrees
- Velocity: ~5 km/h
- Friction coefficients:
  - Static: 0,63 (dry gravel)
  - Dynamic: 0,6 (dry gravel)

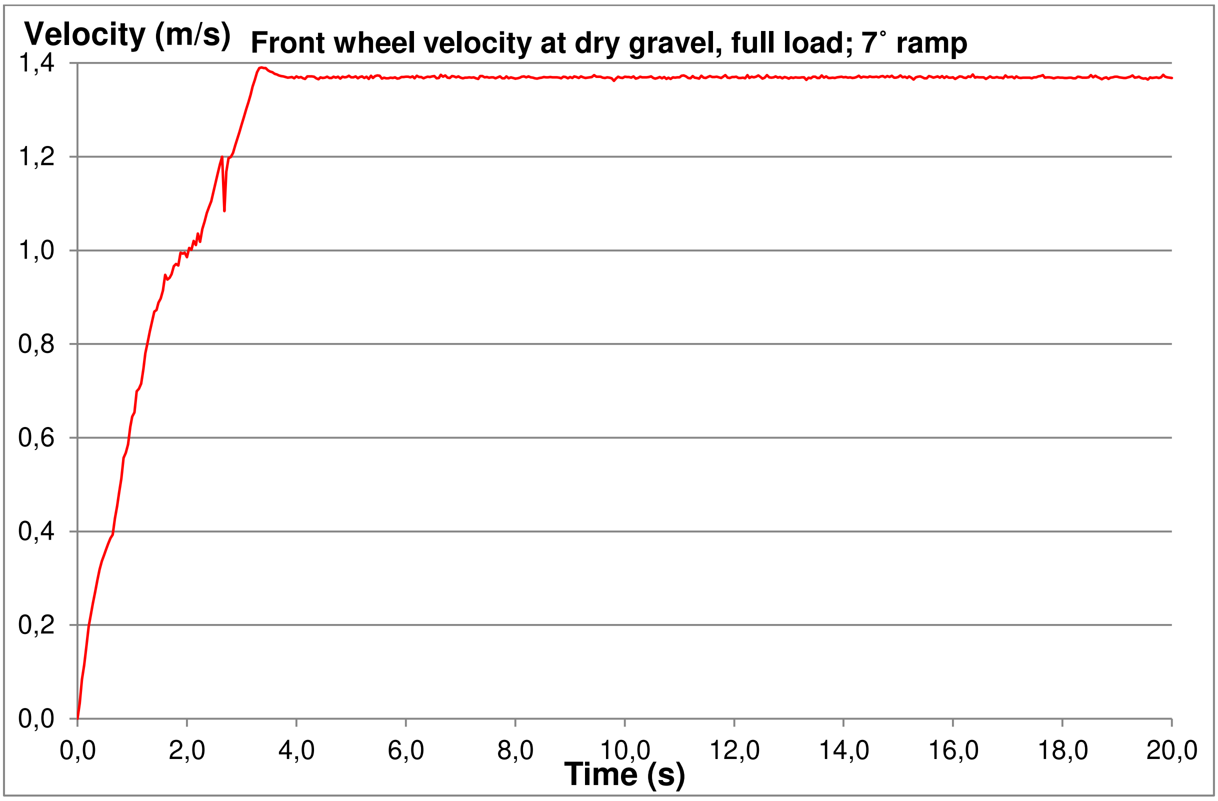

Figure 7.24: Front wheel velocity on 7˚ ramp with full load at dry gravel

Figure 7.24 shows that the auto rickshaw managed to drive up the slope and reached the velocity of 5 km/h after 3,3 seconds. However the acceleration became unsteady when the rear wheels came onto the ramp at 2,7 seconds into the simulation. With the result from the first and third ramp tests a conclusion could be drawn that the auto rickshaw would manage to drive up a ramp with seven degrees inclination if the friction coefficient is around the values for “dry gravel”. However if the friction values would be around those for “wet earth road”, the auto rickshaw probably would not manage to drive up the ramp.

Test 4 - 7˚ inclination with minimum load on wet earth road

A test with only the driver seated in the auto rickshaw was carried through to derive what kind of surface the seven-degree ramp should have for the vehicle to be able to drive up on. The first surface to be tried out was the wet earth road since it got the lowest friction values of the chosen cases.
The parameters used were:

- Minimum load
- Ramp angle: 7 degrees
- Velocity: ~5 km/h
- Friction coefficients:
    - Static: 0,55 (wet earth road)
    - Dynamic: 0,4 (wet earth road)

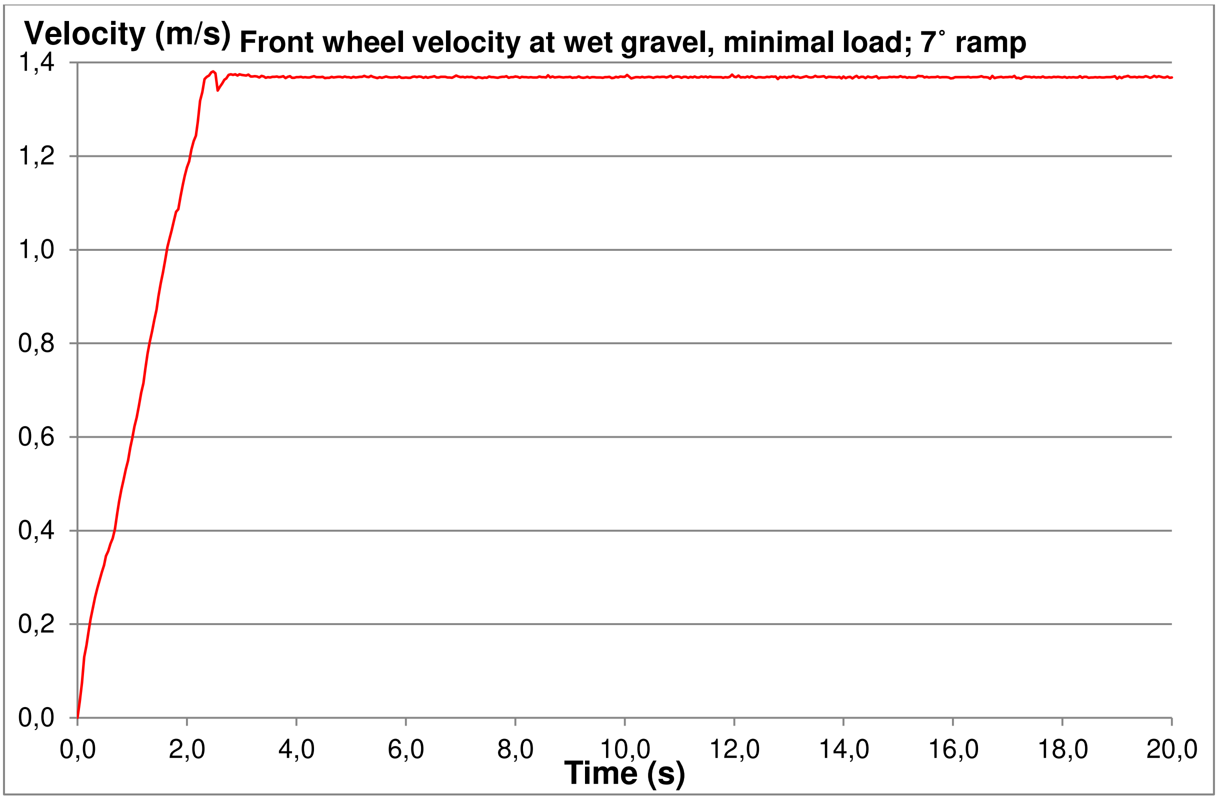

Figure 7.25: Front wheel velocity on 7˚ ramp with minimal load at wet gravel

As shown in Figure 7.25 the vehicle could drive up the ramp under these conditions. The acceleration was rather quick; it only took the auto rickshaw 2,4 seconds to reach the predetermined 5 km/h.

Test 5 - 7˚ inclination with full load on dry asphalt

A final ramp test with full load was performed to cover most possible road conditions. This test used friction values for dry asphalt (see Appendix D).
Parameters used this time were:

- Full load
- Ramp angle: 7 degrees
- Velocity: ~5 km/h
- Friction coefficients:
  - Static: 0,9 (dry asphalt)
  - Dynamic: 0,75 (dry asphalt)

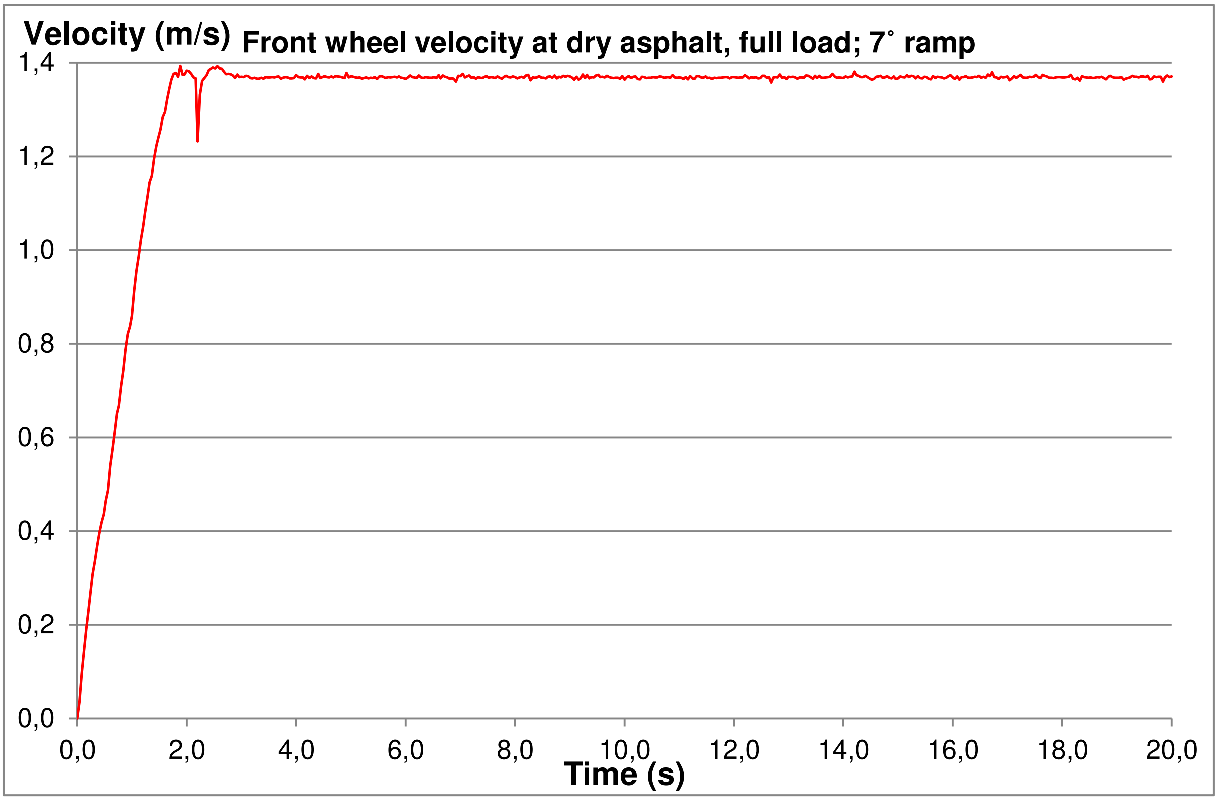

Figure 7.26: Front wheel velocity on 7˚ ramp with full load at dry asphalt

Figure 7.26 shows the vehicle's velocity as it drives up the ramp with no problem. Only when the rear tires comes onto the ramp a small decrease in velocity occurs which is regained quickly. The auto rickshaw's acceleration in this test was faster than in the previous test as it reached 5 km/h after 1,8 seconds.

7.5.8.2 Skid pad tests

To study the cornering performance of the auto rickshaw skid pad tests were conducted. A skid pad test is a test where the vehicle drives around in a circular track to obtain how much the vehicle will sway or how high velocity the vehicle can manage before turning over or skidding out of the radius. The skid pad tests were performed with the guidance of ISO 1982, No. 4138 standard which is a Steady-State Circular Test Procedure. According to ISO 4138 from the TC 22 /SC 9 (Vehicle dynamics and road holding ability) Secretariat a common radius of the skid pad is 40 m [71]. [9]

These tests were conducted with a full loaded vehicle i.e. four passengers and a driver since that will be a common case when the auto rickshaw is operating as well as there is a more distributed weight through the whole vehicle. To get more perspective additional tests were conducted with smaller radius as 10 m and 5 m as well as with minimal load. These tests were simulated with both a sway bar and without a sway bar, which then was compared.
All skid pad tests were done with the friction coefficients for dry asphalt since that surface has the highest friction coefficient and therefore will allow the auto rickshaw to manage a higher velocity before it loses its grip. Consequently the auto rickshaw will turn over rather than lose its grip and slip out of the skid pad radius. The velocity when the auto rickshaw will turn over is more significant than when it loses its grip since that scenario is more crucial. Thus the friction coefficients used in all test were:

- Friction coefficients:
  - Static: 0,9 (dry asphalt)
  - Dynamic: 0,75 (dry asphalt)

Test 1 - 40 m radius right turn with full load

The first skid pad test was based on the ISO standard test, with the 40 m radius skid pad. This test was performed without a sway bar to see how the suspension acts as well as what velocity the auto rickshaw could manage in one lap while making a right turn. To acquire a 40 m radius the steering angle was calculated with equation (5) to approximately minus three degrees.

The following parameters are the significant ones used in this test:

- Full load
- Steering angle: ~-3 degrees
- Turning radius: ~40 m

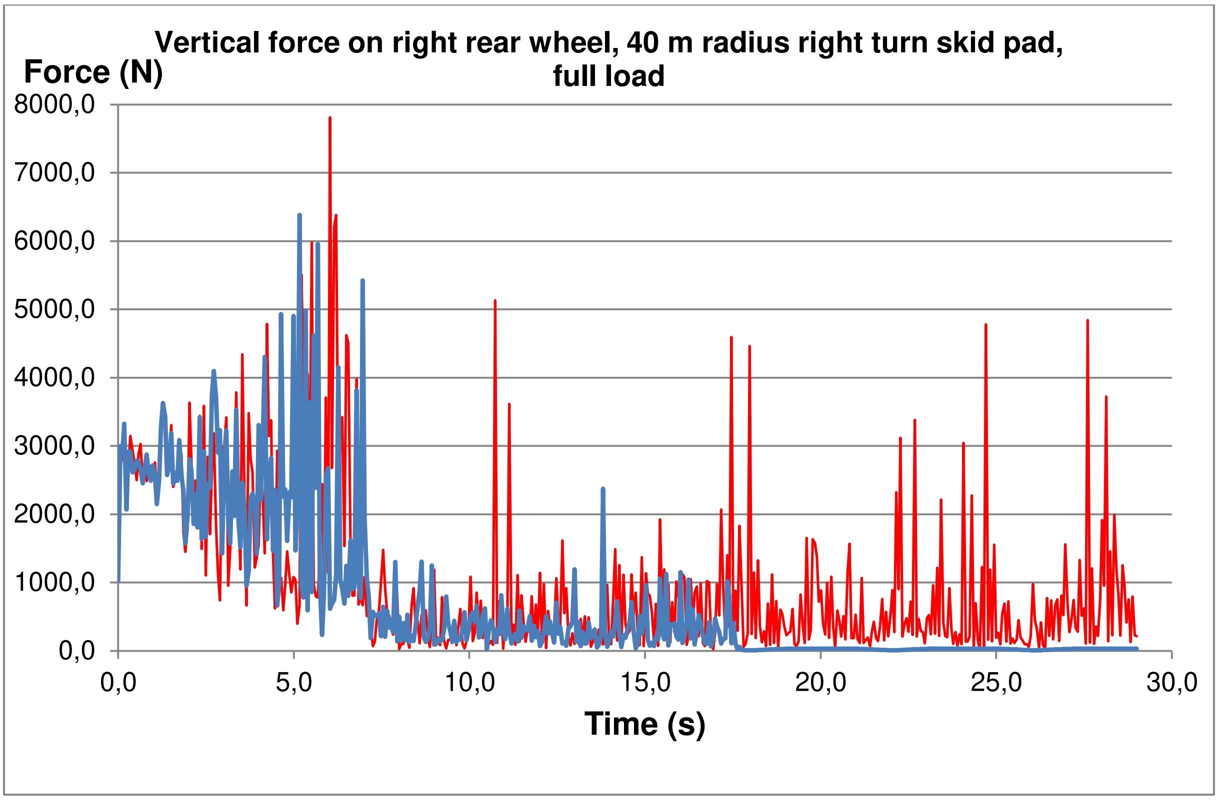

Figure 7.27: Vertical force on inner wheel with full load on 40 m radius skid pad

Figure 7.27 shows the normal force exerted by the ground on the inner wheel. The graph compares a bad run; i.e. when the auto rickshaw tilts, with results from when the auto rickshaw managed driving a full lap. The thick blue line represents the force when the auto rickshaw tilts at 40 km/h and the red when it manages the lap at 39 km/h. As seen in the plot the blue line reaches very close to zero normal force at 17,7 seconds which implies that the auto rickshaw turned over to its side. On the other hand the normal force at 39 km/h represented by the red line keeps relatively steady around 20 N and thus the vehicle does not turn over.

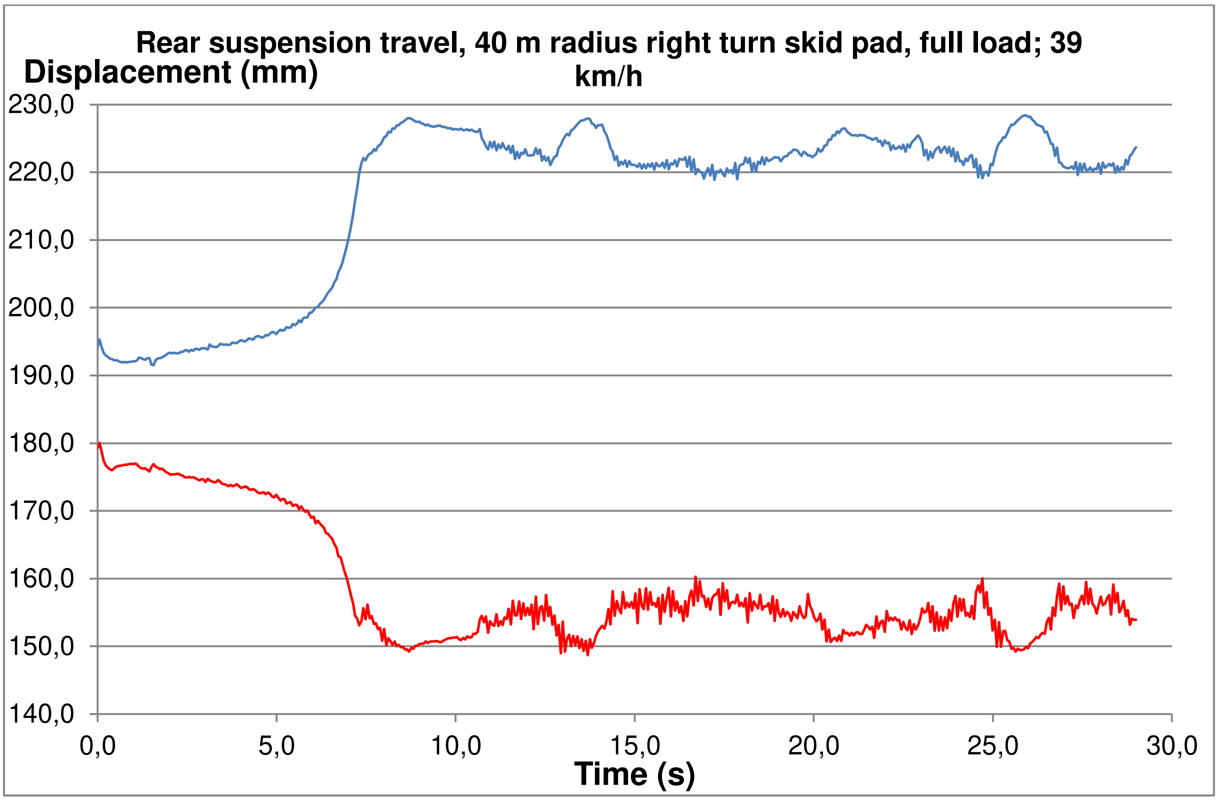

Figure 7.28: Rear suspension displacement with full load on 40 m radius skid pad at 39 km/h

Figure 7.28 shows the rear suspension travel without a sway bar inserted at 39 km/h. As the auto rickshaw made a right turn more force was distributed to the outer wheel; i.e. the left wheel. The suspension travel for the outer wheel was compressed 34 mm while the inner suspension i.e. right side, was extended 33 mm at the most. The difference between right and left suspension travel depended on the suspensions position in steady state.

Result:

- Velocity: 39 km/h
- Suspension travel:
    - Left = Outer: ~34 mm
    - Right = Inner: ~33 mm

Test 2 - 40 m radius right turn with full load including sway bar

This test was performed with the same circumstances as the previous but with the difference that a sway bar now was included.
The first task was to figure out which velocity the auto rickshaw could manage with the sway bar added. As the vehicle managed 39 km/h in the previous test, the speed limit simulations for the auto rickshaw when it would turn over started from that velocity. That because the auto rickshaw including a sway bar should be able to withstand a higher velocity before tilting.
The specific parameters used in this test were:

- Full load
- Steering angle: ~-3 degrees
- Turning radius: ~40 m

A screenshot of the auto rickshaw driving at 40 km/h on the 40 m radius skid pad are shown in Figure 7.29. The figure shows that the vehicle rolls as it corners.

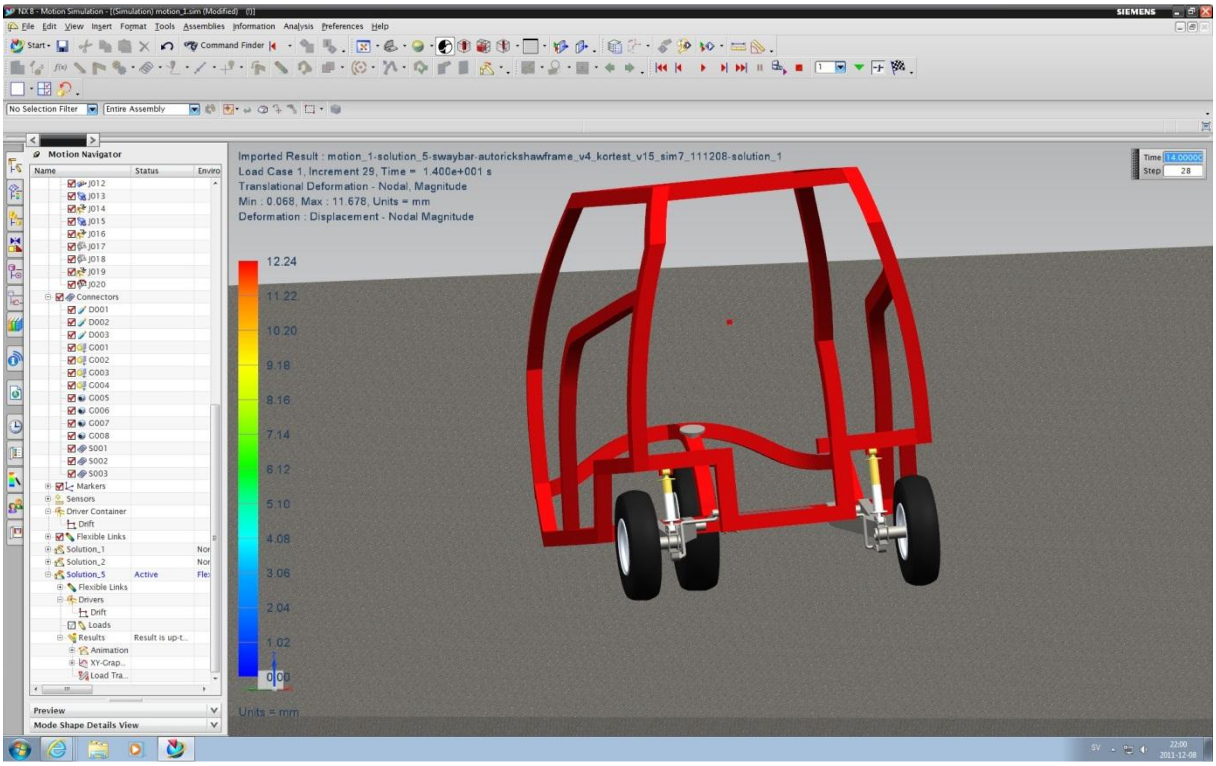

Figure 7.29: Rear view of auto rickshaw driving at 40 km/h on 40 m radius skid pad

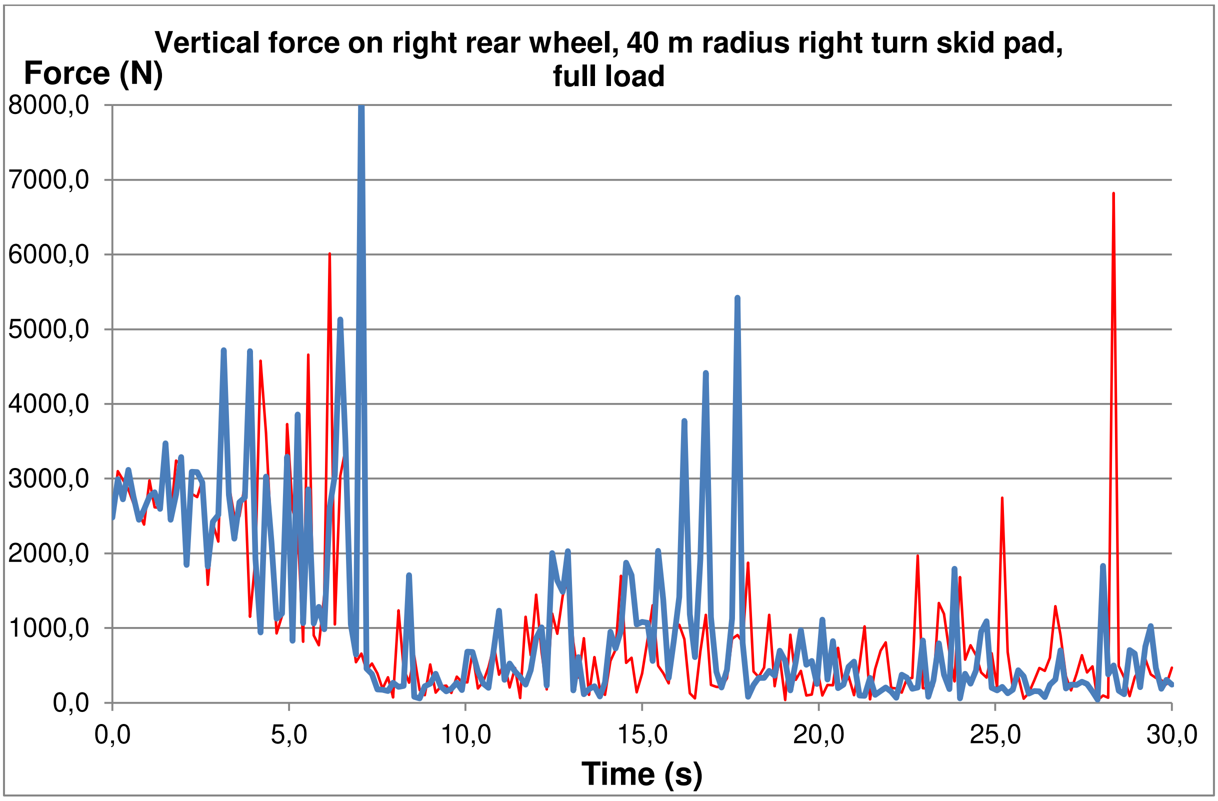

Figure 7.30: Vertical force on inner wheel including sway bar with full load on 40 m radius skid pad

Figure 7.30 shows the normal force on the inner wheel for both 39 km/h and 40 km/h which became the threshold velocity the auto rickshaw could manage. The procedure to find the threshold velocity was the same as in the previous test.
The red line represents the normal force at 39 km/h and the thick blue line represents the normal force at 40 km/h. The figure shows that the normal force at 39 km/h had a significant margin to zero normal force which resulted in another test with one km/h higher velocity. The velocity of 40 km/h gave a normal force very close to zero which resulted in the conclusion that the vehicle would not manage any higher velocity than that. The result from the 39 km/h test was then used to compare the suspension travel with the previous test where there was no sway bar included.

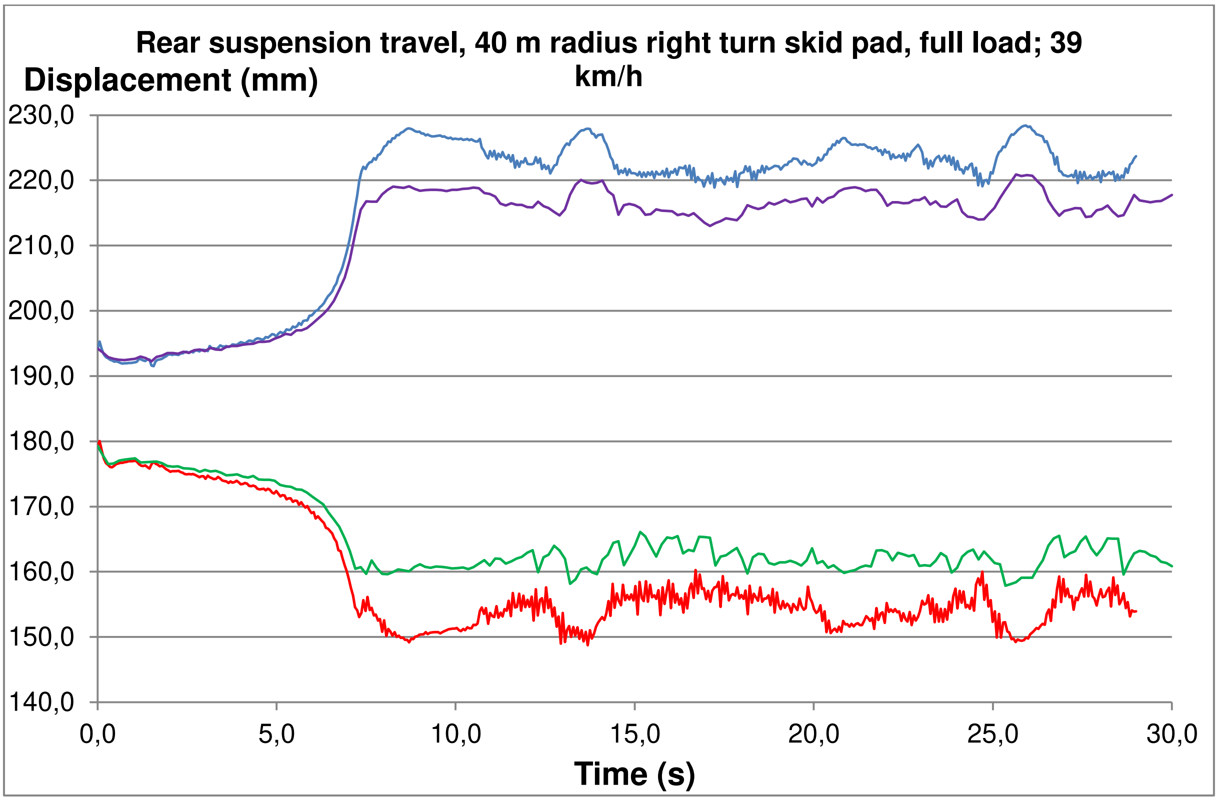

Figure 7.31: Rear suspension displacement with full load on 40 m radius skid pad at 39 km/h

A comparison was made between the suspension travel with and without sway bar. With the maximum suspension travel deduced from Figure 7.31 a comparison could be made for each side. The red and green lines represented the left suspension travel while the blue and purple represented the right suspension travel. The red and blue line is the suspension displacement without the sway bar.

The suspension travel with the sway bar resulted in:

- Velocity: 39 km/h
- Suspension travel:
  - Left = Outer: ~24 mm
  - Right = Inner: ~24 mm

With the result a difference could be calculated. For the left suspension a 29 % decrease was altered by the sway bar since it declines from 34 mm to 24 mm. The sway bar improved the right suspension with 27 % as the suspension travel decreased from 33 mm to 24 mm. I.e. the anti-roll bar reduced the sway by ca. 30 % for the rear suspension seen as one system.

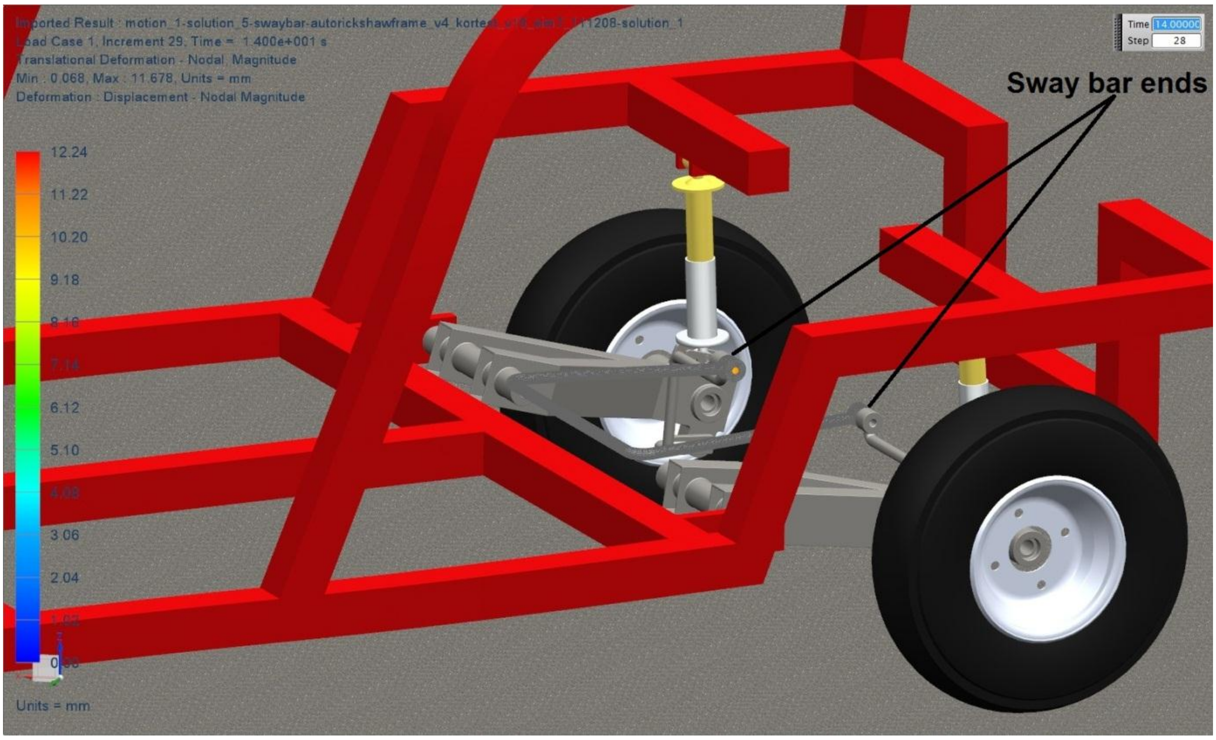

Figure 7.32: Sway bar movement in auto rickshaw when driving at 39 km/h on 40 m radius skid pad

Figure 7.32 shows the sway bar when the auto rickshaw corners at a velocity of 39 km/h on the 40 m radius skid pad. The picture is taken as a screenshot when running the simulation in NX. The coloured staple to the left informs about the displacement of the sway bar at the end points.

Test 3 - 10 m radius right turn with full load

This test was done as the previous ones but with a radius of 10 m for the skid pad. Another steering angle of 11 degrees was needed to acquire the 10 m radius.
The following parameters were used in this test:

- Full load
- Steering angle: ~-11 degrees
- Turning radius: ~10 m

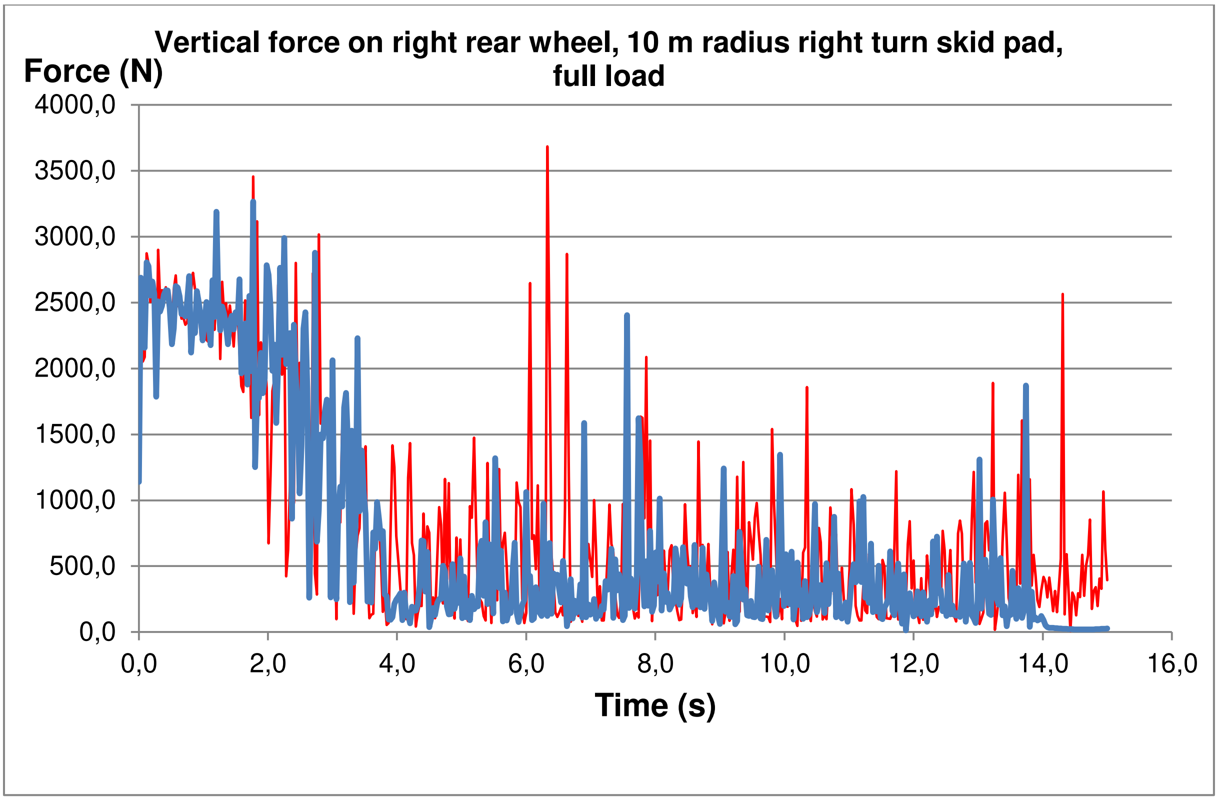

Figure 7.33: Vertical force on inner wheel with full load on 10 m radius skid pad

Figure 7.33 shows the normal force on the right rear wheel at both 20 km/h and 21 km/h. The thick blue line represents the normal force at 21 km/h and the red line represents the normal force at 20 km/h. When the auto rickshaw drives with a velocity of 21 km/h it begins to tilt at 14 seconds, while at 20 km/h the normal force remained a bit over the zero limit. This test showed that the auto rickshaw without the sway bar could manage a velocity of 20 km/h in a 10 m radius skid pad.

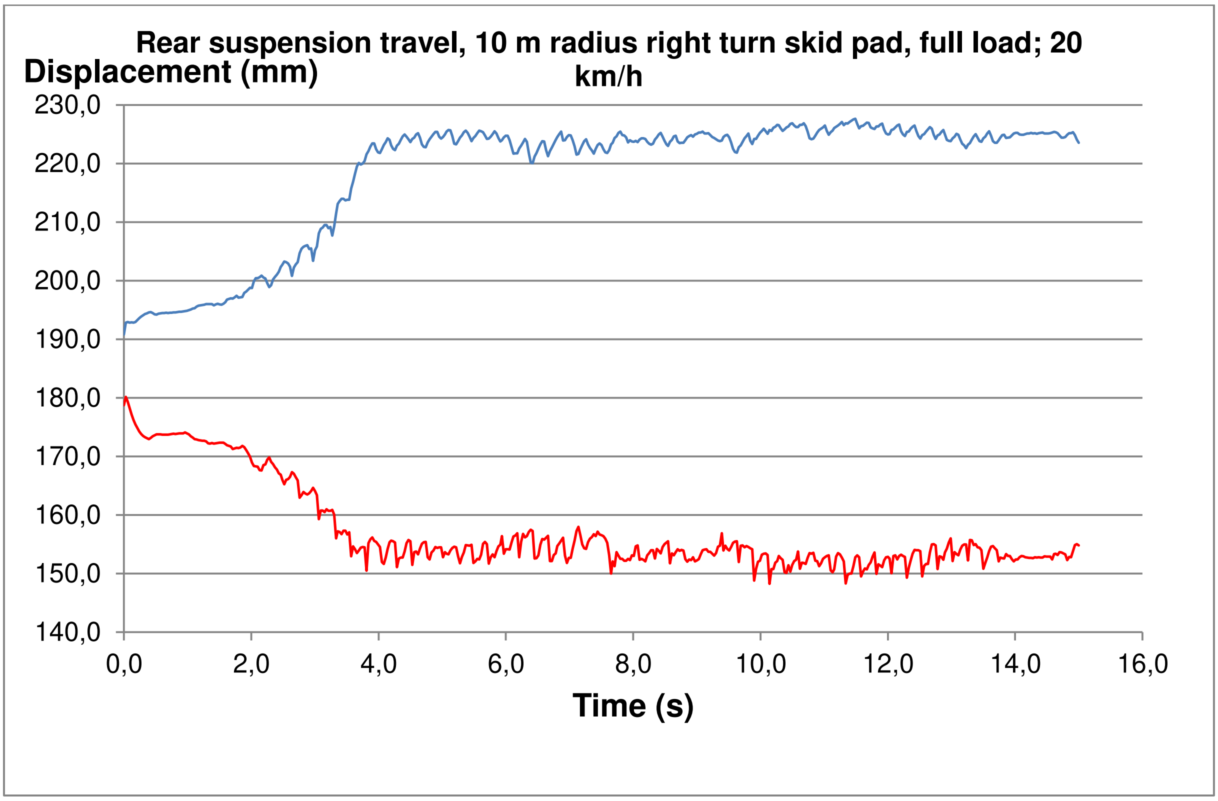

Figure 7.34: Rear suspension displacement with full load on 10 m radius skid pad at 20 km/h

Figure 7.34 shows that the suspension travel was almost the same as in the first skid pad test, i.e. around 30-35 mm for the left suspension and around 30-35 mm for the right suspension. The suspension travel landed on approximately 34 mm for the left suspension and around 32 mm for the right suspension. This declares that the suspension will allow a suspension travel of around 33 mm before turning over and crash.

Result:

- Velocity: 20 km/h
- Suspension travel:
    - Left = Outer: ~34 mm
    - Right = Inner: ~32 mm

Test 4 - 10 m radius right turn with full load including sway bar

This test was performed with the same conditions as in "test 3" but with the sway bar included. The maximum velocity the auto rickshaw could manage and the rear suspension travel at 20 km/h were searched for. The suspension travel had to be deduced at 20 km/h since the result was going to be compared with the result from "test 3" in the same way as "test 1" and "test 2" were compared.
These following parameters were used:

- Full load
- Steering angle: ~-11 degrees
- Turning radius: ~10 m

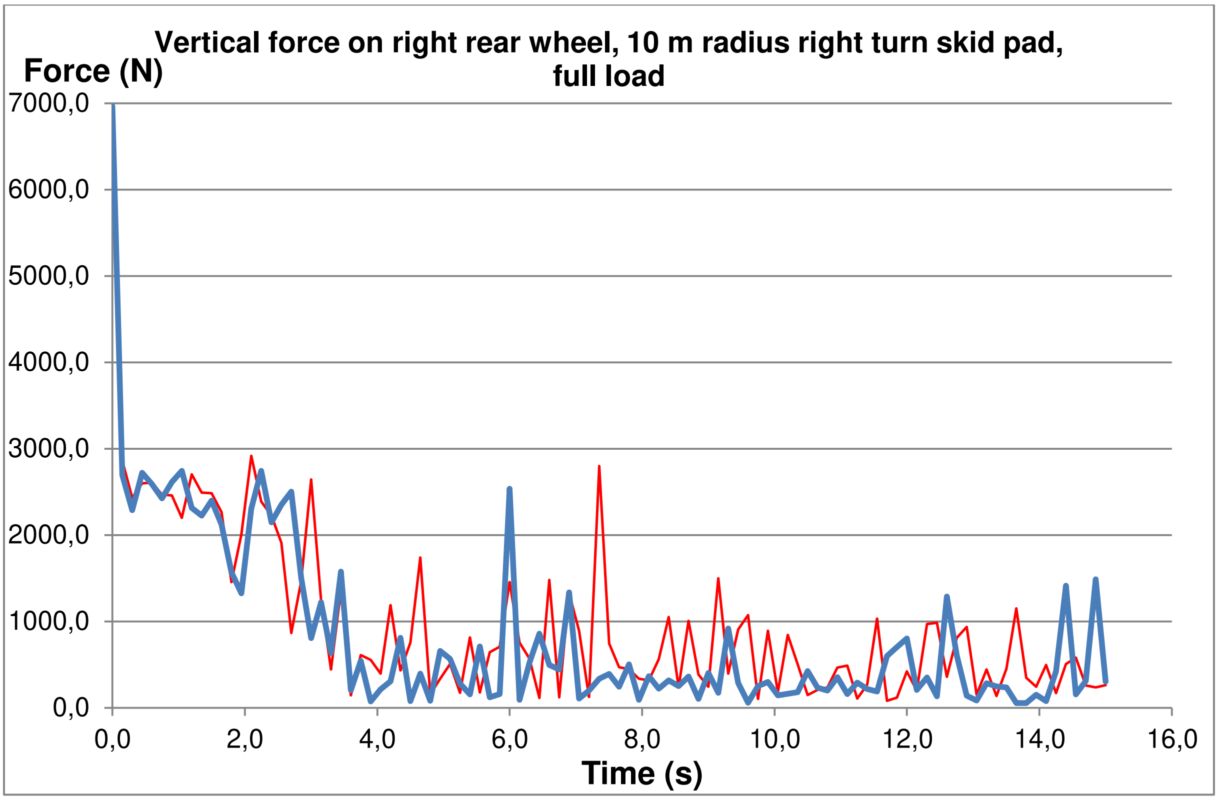

Figure 7.35: Vertical force on inner wheel including sway bar with full load on 10 m radius skid pad

The high rise of the curves shown in Figure 7.35 is due to the larger instant steering displacement of 11 degrees which results in elevation of the vehicle. Starting the simulation would thus result in the auto rickshaw falling from a height. In this case with an initial displacement of 11 degrees the vehicle would raise approximately 5 mm. Figure 7.35 shows the normal force on the right rear wheel where the thick blue line represents the force at 21 km/h and the red line represents the force at 20 km/h. The thick blue line indicates that the auto rickshaw manages a velocity of 21 km/h but not any higher since the normal force is close to critically low but never reaches zero.
With the maximum suspension travel deduced from Figure 7.36 a comparison could be made for each side.

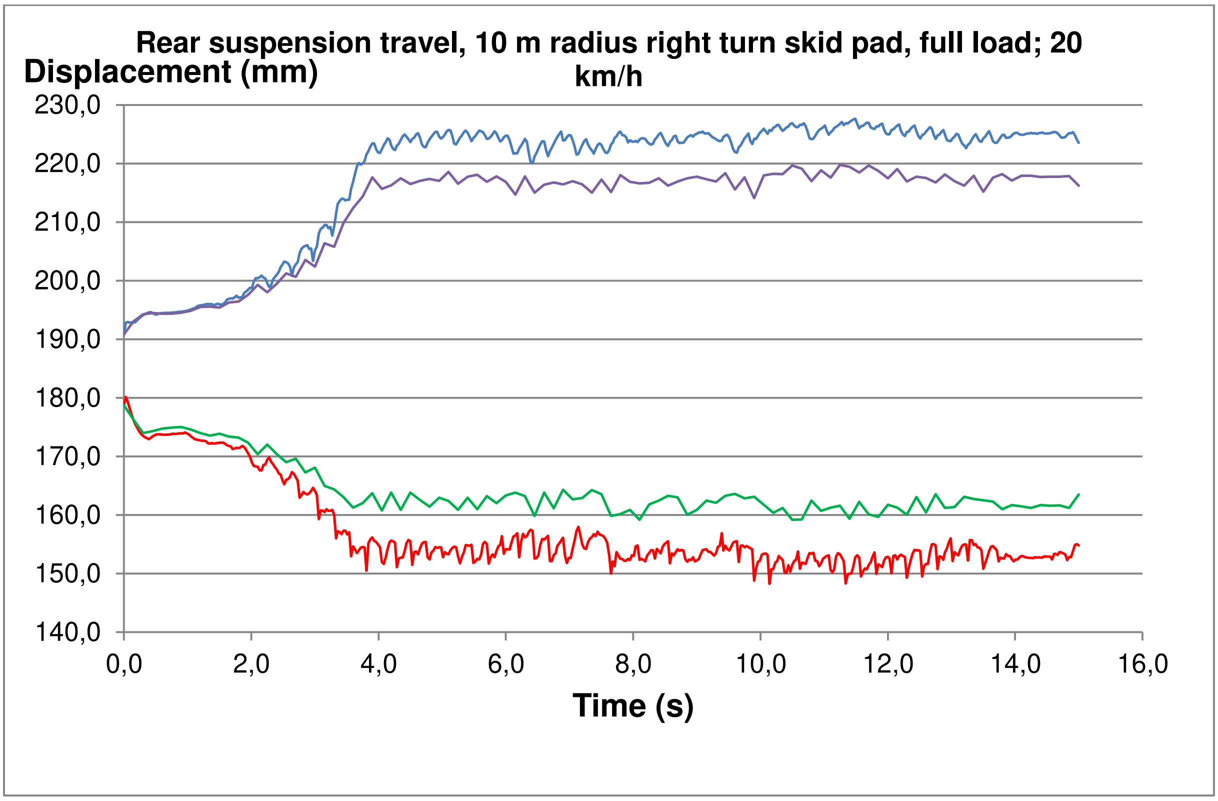

Figure 7.36: Rear suspension displacement with full load on 10 m radius skid pad at 20 km/h

Figure 7.36 shows the suspension travel for left and right, with and without sway bar. The red and green lines represented the left suspension travel while the blue and purple represented the right suspension travel. The green and purple line is the scenario with the sway bar.
The suspension travel with the sway bar resulted in following:

- Velocity: 20 km/h
- Suspension travel:
  - Left = Outer: ~23 mm
  - Right = Inner: ~24 mm

The difference between suspension travel with and without a sway bar, i.e. the decreasing sway, was calculated with the same method as in "test 2".
The sway bar reduced the left suspension travel by 32% and the right suspension travel by 25 %. That would decrease the sway by approximately 30% for the rear suspension seen as one system.

Test 5 - 5 m radius right turn with full load including sway bar

With both test 2 and 4 it was shown that the sway bar would decrease the suspension travel around 30%. An additional test was conducted to find the velocity limit for the auto rickshaw when cornering in a 5 m radius skid pad. Therefore a test including the sway bar was made on the 5 m radius skid pad.

The following parameters were used in this test:

- Full load
- Steering angle: ~-24 degrees
- Turning radius: ~5 m

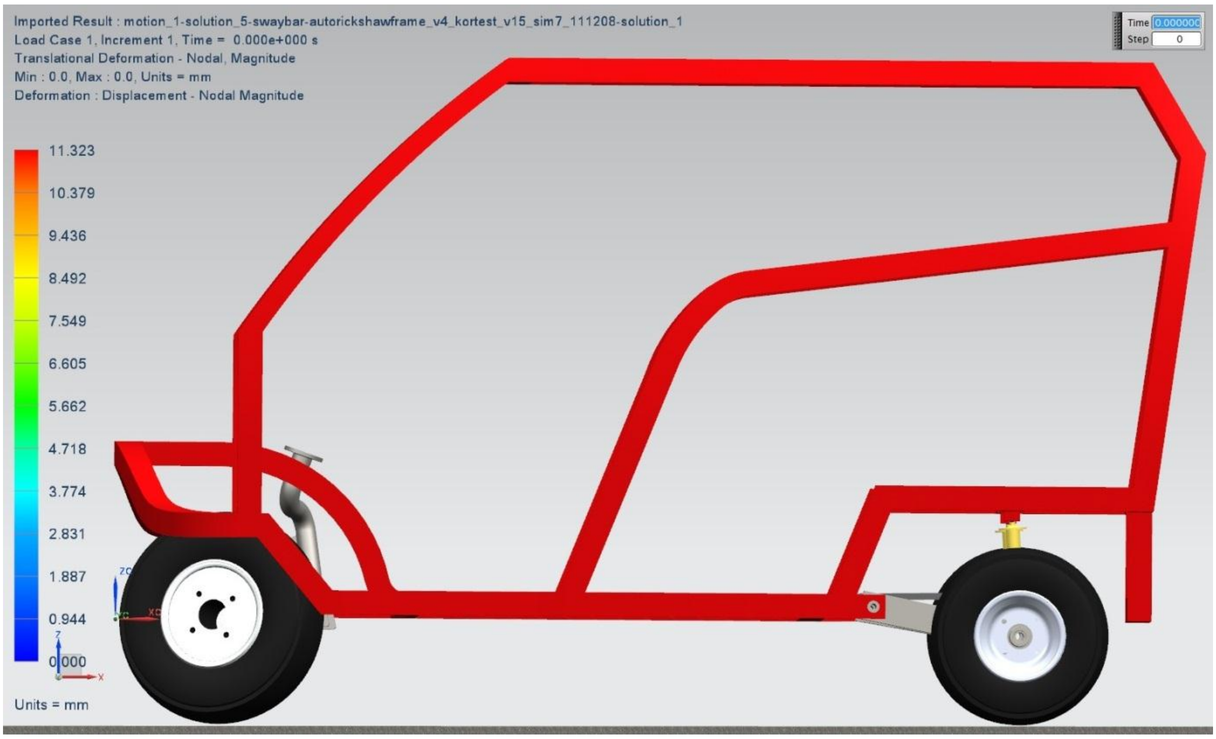

Figure 7.37: Auto rickshaw seen from the side at start of simulation for 5 m radius skid pad test

Figure 7.37 shows the elevated auto rickshaw which occurs when the steering angle of 24 degrees is added. The elevation of the wheels, which in this case was 6 mm, are shown.

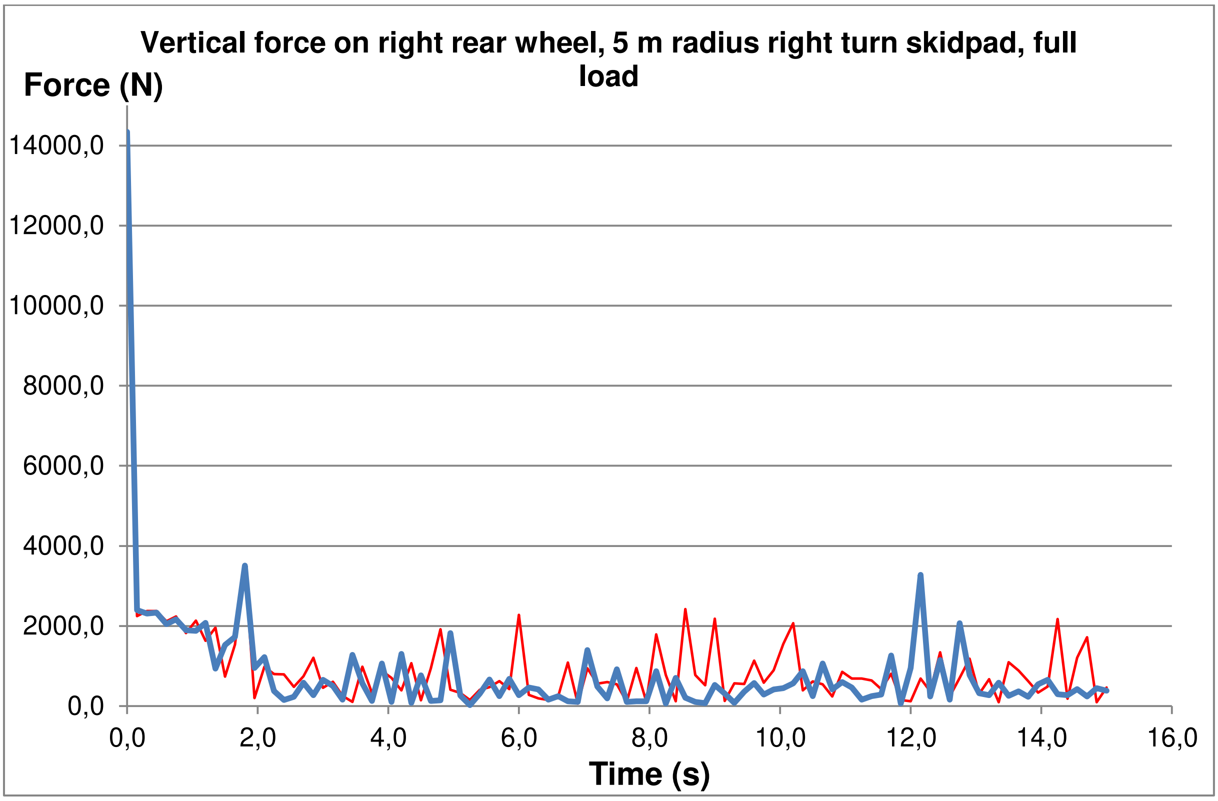

Figure 7.38: Vertical force on inner wheel including sway bar with full load on 5 m radius skid pad

Figure 7.38 shows the normal force of two different scenarios, one at 14 km/h; blue, and the other at 13 km/h; red.

A simulation was made with a speed of 13 km/h and at this velocity the normal force was high enough to maintain a stable auto rickshaw. A second simulation showed that the vehicle could manage a velocity of 14 km/h for a whole lap. The thick blue line in Figure 7.38 is close to critically low which indicates that a higher velocity would result in the vehicle tipping over. In other words the auto rickshaw was able to drive at a velocity of 14 km/h in a 5 m radius skid pad.

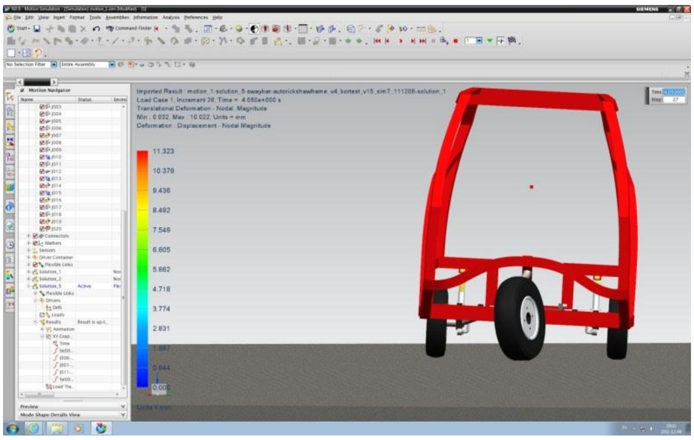

Figure 7.39: Auto rickshaw cornering in a 5 m radius with full load at 14 km/h

In Figure 7.39 the 24 degree steering angle of the front wheel are shown as the fully loaded auto rickshaw is cornering at 14 km/h in a 5 m radius skid pad test.

Test 6 - 40 m radius left turn with minimal load including sway bar

The previous tests showed that the sway bar reduced the roll tendencies when cornering at full load. To get a more thorough performance analysis of the vehicle a few more scenarios were tested. These scenarios were made with minimal load; i.e. only the driver seated in the auto rickshaw. As the centre of gravity will be moved more to the right and forward than when it is fully loaded a left turn should be more crucial than a right turn.
In this test the sway bar was included since it is utilised in the final model of the auto rickshaw.
The test was conducted with the guidance of the ISO standard.
The following parameters were used in this test:

- Minimal load
- Steering angle: ~3 degrees
- Turning radius: ~40 m

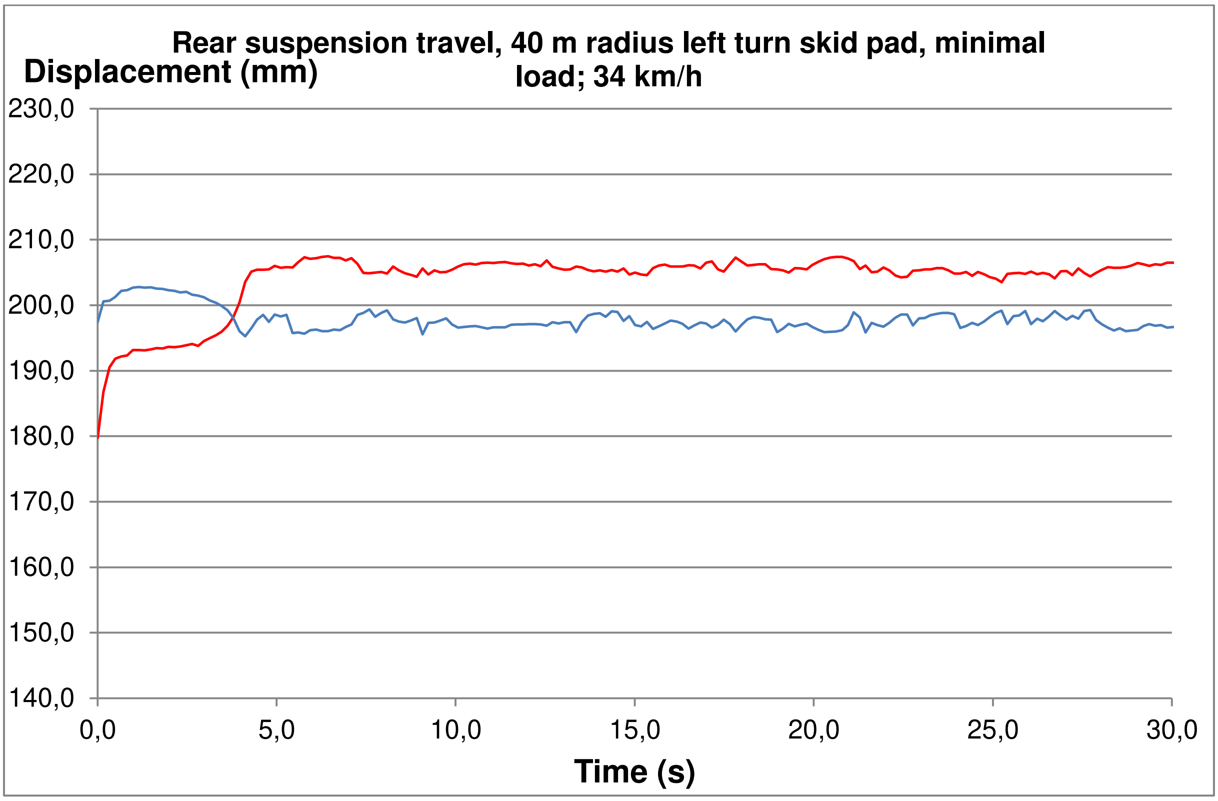

Figure 7.40: Rear suspension displacement with minimal load on 40 m radius skid pad at 34 km/h

The result received from the first minimal load test are shown in Figure 7.40. The suspension travel at the maximum velocity the vehicle could handle were 25 mm for the left suspension and approximately 1 mm for the right. Only a low movement of the right suspension occurred because of the far forward position of the centre of gravity which originates from the low sprung mass. The left suspension travel is represented by the red line and the right suspension travel is represented by the blue line.

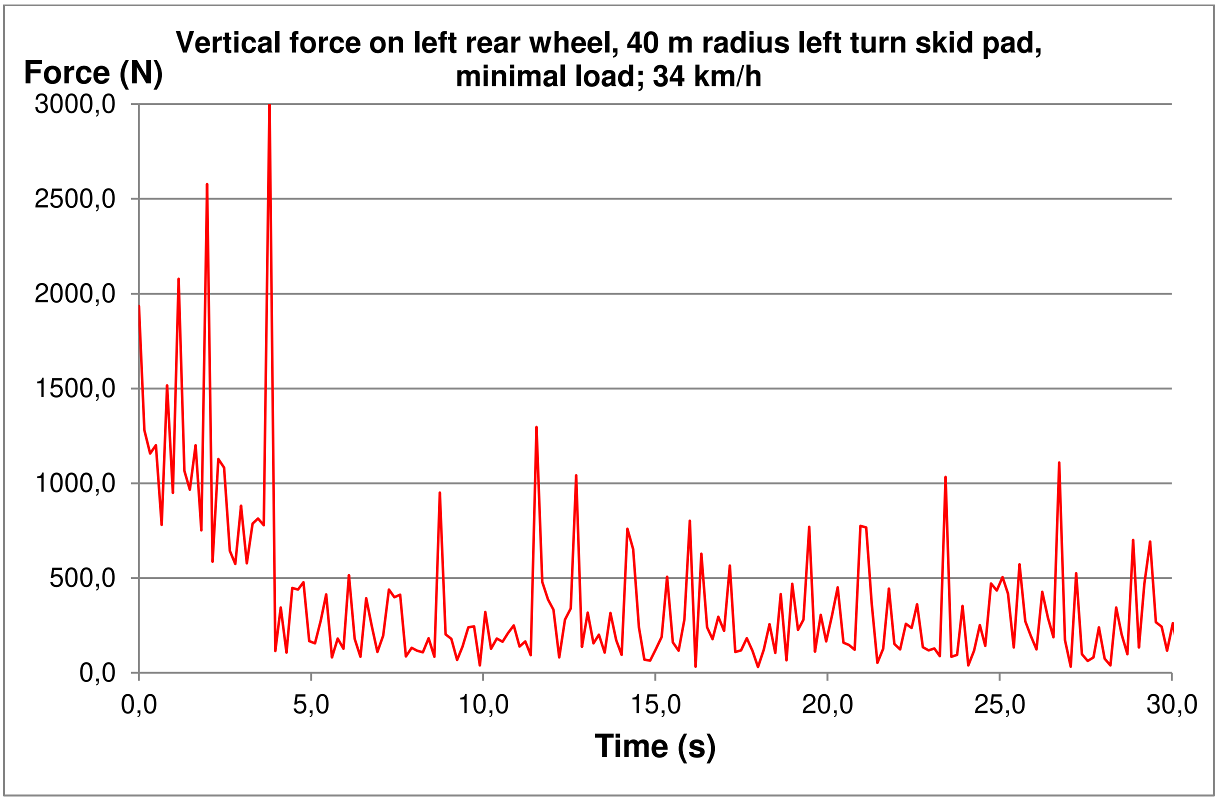

Figure 7.41: Vertical force on inner wheel including sway bar with minimal load on 40 m radius skid pad

Figure 7.41 shows that the normal force acting on the inner wheel; i.e. left wheel, was relatively close to zero at some points. The velocity the auto rickshaw could handle in a 40 m radius skid pad was found to be 34 km/h. That was 6 km/h lower than when turning right with full load in the same radius because of the more critical positioning of the centre of gravity; which are shown in Figure 7.3.

Result:

- Velocity: 34 km/h
- Suspension travel:
  - Left = Inner: 25 mm
  - Right = Outer: 1 mm

Test 7 - 10 m radius left turn with minimal load including sway bar

A similar test as "test 6" was conducted but with a 10 m radius skid pad as this radius was used in earlier simulations. The conditions remained the same as in "test 6", as well as making a left turn, except for the steering angle which had to be set to 11 degrees to clear a 10 m radius. The parameters searched for were the maximum velocity the system could manage and the suspension travel for both sides of the rear suspension.
The following parameters were used in this test:

- Minimal load
- Steering angle: ~11 degree
- Turning radius: ~10 m

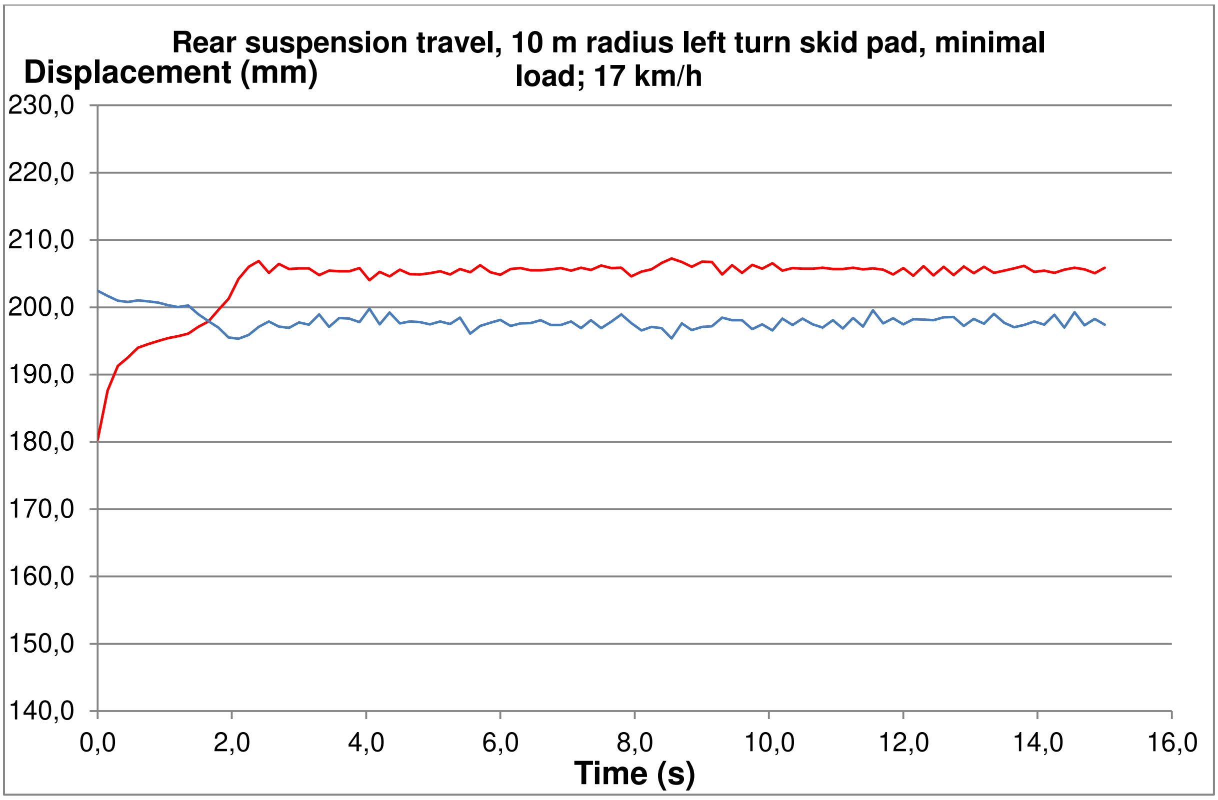

Figure 7.42: Rear suspension displacement with minimal load on 10 m radius skid pad at 17 km/h

Deduced from Figure 7.42 the suspension travel ends at 24 mm for the left suspension and 1 mm for the right suspension. These values correspond to the values from rear suspension travel in "test 6", which even more indicates that the suspension acts as it should. The red line shows the left suspension's travel and blue show the right ones.
Another observation of Figure 7.42 is that the high peak that occurs the first milliseconds. The peak is a result of the steering angle that is implemented at the start of the simulation and this steering angle would automatically raise the vehicle a few millimetres as it did in "test 4".

Result:

Velocity: 17 km/h

- Suspension:
  - Left = Inner: 24 mm
  - Right = Outer: 1 mm

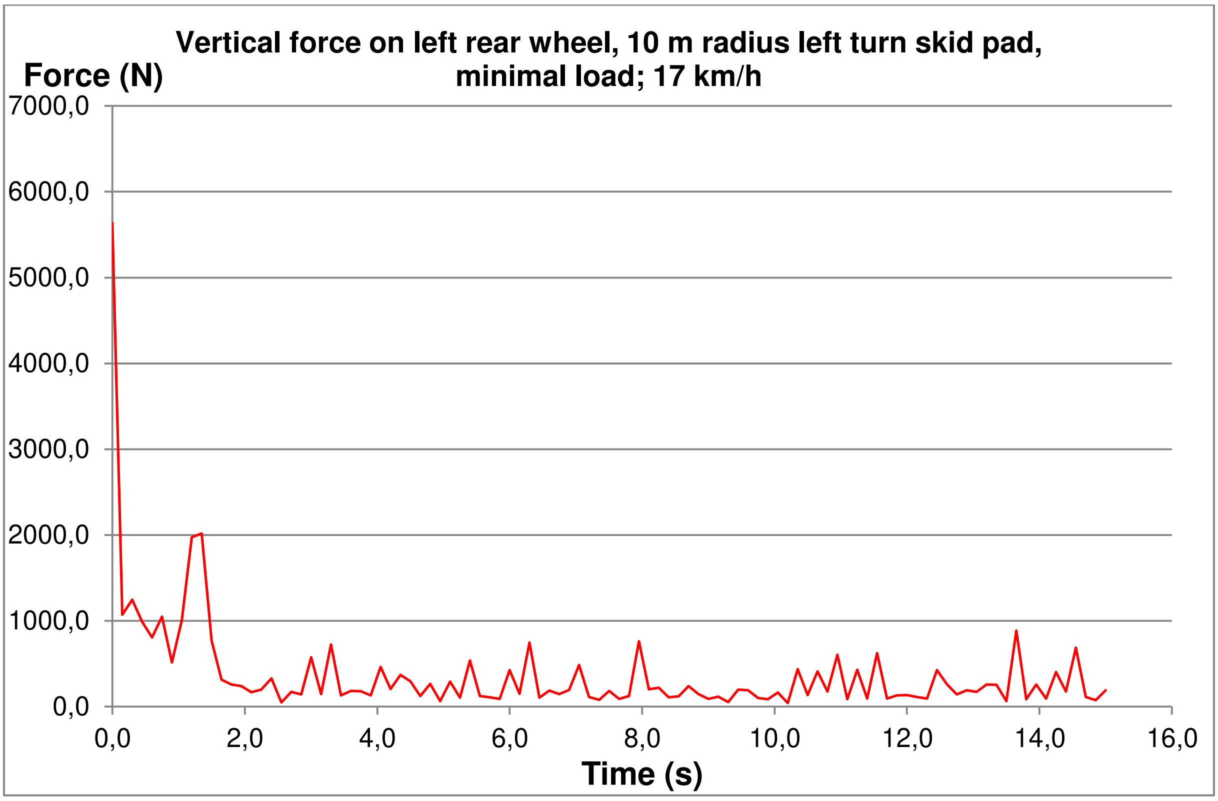

Figure 7.43: Vertical force on inner wheel including sway bar with minimal load on 10 m radius skid pad

Figure 7.43 shows the normal force acting on the inner wheel and read from the graph the normal force lands close to zero and hence a velocity of 17 km/h is the limit for this skid pad radius before tilting over.

Test 8 - 5 m radius left turn with minimal load including sway bar

Same circumstances were used in this test as in "test 6 "and "test 7" with a change of the skid pad radius which was set to 5 m. For the auto rickshaw to drive in a 5 m radius skid pad the steering angle had to be altered to 24 degrees.
The following parameters were used in this test:

- Minimal load
- Steering angle: ~24 degree
- Turning radius: ~5 m

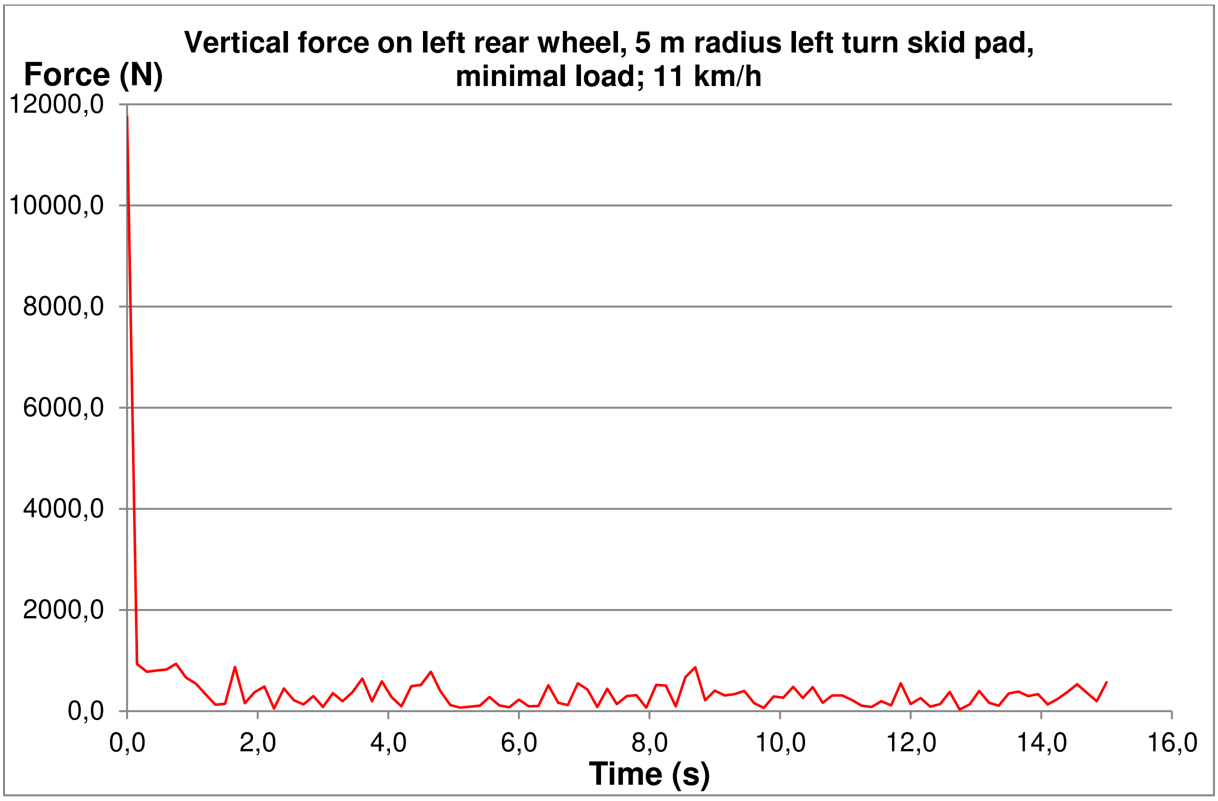

Figure 7.44: Vertical force on inner wheel including sway bar with minimal load on 5 m radius skid pad

Figure 7.44 shows that at 12,8 seconds the normal force reached its lowest value of 27,5 N; which is very close to tilting. It was thus concluded that the auto rickshaw could manage a velocity of 11 km/h before overturning. The same elevation occurred here as in "test 5", i.e. when the simulation started, the steering angle would instantly raise the vehicle a few millimetres. This elevation resulted in the high peak within the first milliseconds as shown in Figure 7.44 when the auto rickshaw impacts the ground.

8. Error assessment

The 3D models created in Siemens NX used estimated measures that were provided from an early step in the conceptual design phase and was thus probably where the most sources of errors occurred because the measures and positions may change as the vehicle development proceeds. All parts were modelled with estimated measures but since most parts are considered as rigid bodies some measures did not affect the result.
The part that could influence the results significantly was the sway bar since it was converted into a flexible body which included many different parameters. Because the sway bar was considered as a flexible body it had to be assigned a material, and the parameters determining that material properties were in literature often given within a range instead of a fixed value. There is a wide range of different material used in sway bars depending on the purpose for it, therefore the material chosen now might not be the right one. Other features defining the sway bar was the connections and meshing which also could be sources of error. Simulations to calibrate the sway bar have not been performed and thus the sway bar may not operate optimally.
The durability and strength of the individual parts and their design have not been calculated, FEM analysed with NX or tested in any way.

Measures that could affect the calibrations were the sprung mass and the position of the centre of gravity that were calculated in MATLAB. The weight and position of the centre of gravity were determined by the vehicle's measures and the position and weight of each part inside the vehicle. As the vehicle's measures and the position and weight of each part inside the vehicle were estimated the centre of gravity would also be estimated. Since the centre of gravity is a crucial parameter in all tests even the smallest change of the estimated measures could affect the final results.
Another crucial parameter is the positions of the springs. The calculations were based on the front spring being positioned in the middle of the vehicle as of its real position which was a few centimetres to the right of the middle.

Motion Simulation in Siemens NX has a lot of parameters that can affect the simulation, among them the 3D contact which contains some crucial parameters. The values used to create a 3D contact contain friction values and other parameters that have great importance later on when a simulation is performed. With wrong values assigned the vehicle could for example start to bounce or drift away. The parameters used in this thesis arrived mostly from NX Help with recommended or default values, except for the stiffness and friction parameters which were found from literature studies. Many of the values found in NX help that were used was for common rubber and may thus differ from values appropriate for a tyre.
The maximal velocity gained from the skid pad simulation is based on driving in a constant radius and since a locked steering angle is used to acquire that radius the result may differ. One explanation can be that the radius was not constant because of wheel slip and thus allowing the vehicle to handle a higher velocity due to the lower centripetal acceleration that a wider radius induces.

The dampers used in all simulations had linear properties which will affect the results of the simulations as most dampers used in real life have non-linear characteristics.

The results only includes information from a few worst case scenarios which gave a narrowed range of data, more simulations with different scenarios would be needed to get a wider range of data to give a more thorough vision of the suspensions characteristics as a system. As the solving time was significantly raised when a larger number of steps were used, the number of steps was kept low and that might have affected the final results.
Siemens NX might not be the optimal program to use when it comes to creating vehicle simulations in the sense of dynamic aspects and therefore the results might not be as accurate as softwares especially designed for the vehicle dynamics.

No results have been validated; i.e. no parts have been constructed and tested in reality, to see that the results from the simulation are conformable with the results from real practical tests.

9. Discussion and conclusion

The concepts selection was made by use of a weight matrix which resulted in three remaining front suspensions and three remaining rear concepts. Literature studies in combination with simulations should be used to find the best possible solution of the remaining concepts. Unfortunately this could not be executed since there was a limited time frame and therefore the choice of concept was determined by only literature studies.
The choice of concept were trailing arm suspension for the rear and leading link suspension for the front.

Adams/Car is a program used for simulating and testing vehicle dynamics and thus a better program to use than any other if the goal is to retrieve vehicle performance data. Therefore it seemed as an appropriate software to use, but since Adams/Car is not designed for three wheeled vehicles a lot of difficulties occurred. All that were accomplished in Adams/Car were simple procedures e.g. modelling of suspensions and a few suspension travel simulations. Although it should be possible to modify the program so that it would fit a three wheeled vehicle there was a lack of knowledge as well as support and hence another program was used. A conclusion could be made that in the case of simulating handling/dynamic performances for a three wheeled vehicle Adams/Car should not be the first hand choice since it is designed for four wheeled vehicles.

The software that was used instead was Siemens NX since basic knowledge of the program existed but this also meant that the modelling of the suspension had to be carried out from scratch again. Siemens NX had enough functions to obtain the results that were searched for. The process before being able to create a simulation consisted of pre-work including research to figure out the vast amount of parameters and to find appropriate numerical data. The parameters used had a great impact on the results and the slightest change could affect the results a lot.
Simplification of parts and parameters were used to reduce the solving time and thereby being able to run more simulations. One of the simplifications made was the use of linear damping instead of non-linear damping, and another was to not use the more complex final auto rickshaw structure.

The goal was to find the maximal velocity the vehicle could handle when cornering without turning over instead of looking for the velocity the vehicle should have to keep the skid pad radius. The reason for that is because the point when the auto rickshaw turns over is more crucial and interesting than when it loses its grip and slips out of radius. From these skid pad simulations the body roll could also be analysed.
The results from the simulation showed that the auto rickshaw did not manage a high velocity when cornering; i.e. 40 km/h in a 40 meter radius skid pad. This limitation is probably due to the three wheels setting with one front wheel. With only one front wheel the vehicle will get more unstable compared to a vehicle with two front wheels. If the weight distribution is moved forward the vehicle will be even more unstable which can be observed by comparing skid pad "test 2" and "test 6" in the auto rickshaw performance test section.

To reduce the body roll a sway bar was added to the rear suspension. The objective of a sway bar is to decrease the roll which was shown in "test 2" as the vehicle managed a higher cornering velocity as well as lowering the displacement difference between the rear suspensions. The sway bar improved the body roll by approximately 30 % when fully loaded which allowed the vehicle to increase its cornering velocity with a few percents.
Another test was conducted to find out if the vehicle could manage an inclination of 7 degrees which is the minimum requirement the vehicle should manage. The vehicle did manage to drive up the 7 degree inclination on all tested surfaces except for wet earth road (lowest friction values tested) which the vehicle managed at an angle of 6 degrees.

The suspension solution has been verified with different simulations. Suggestions of further development can be seen in the Future work chapter.

A suspension suitable for an auto rickshaw could be a leading link front suspension and trailing arm rear suspension in combination with a sway bar. This report concludes that such a solution would have sufficient performance and still be cheap enough for the tough Indian market.
The three-wheel configuration in a delta shape has flaws when it comes to vehicle dynamics and handling as it is more unstable compared to a four wheel configuration. The three wheel set up although have some positive attributes as it is very cheap and is a combination of both a two and a four wheeled vehicle.

The objective of this thesis was to develop a vehicle suspension intended for an auto rickshaw with improved handling and dynamic characteristics. The development of an auto rickshaw suspension was accomplished and data from simulations of the auto rickshaw and its suspension was obtained.

The work resulted in an overall suspension solution including parts as leading link suspension for the front, trailing arm suspension with sway bar for the rear and suitable dampers and springs.

Future work

The program that was used in the beginning of this thesis; Adams/Car, can still be used by modifying the software so it would work for a three wheeled vehicle. The least modifications that would be needed to be able to simulate the auto rickshaw would be to alter the test rig within the software. The test rig would be needed to work for one wheel as well as three wheel simulations. The output communicator "suspension parameter array" must be able to be added to a single sided suspension to establish a working auto rickshaw suspension. Some further modifications may also be needed to make the suspensions work.
Since there exist a report [62] where engineers have accomplished performing a three wheel simulation with Adams/Car this indicates that it is possible to execute.

If possible a connection between Siemens NX and Simulink should be established as more function and more advanced features will be available. A function that can be able is to create a more advanced steering function that will keep a constant radius and thus retrieve a more accurate result on e.g. maximal velocity during a skid pad test. As a follow up of the constant radius tests an investigation on how torque steer effects would influence the steering could be made. The more advanced steering function could also be used to perform e.g. ISO lane change, brake and turn, or acceleration-brake to mention a few. From the acceleration-brake tests investigation of the anti-dive effect could be made and thus verifying that the leading link acts as it should. Even the magnitude of the anti-dive should be possible to calibrate.

Procedures that could be done in the future are FEM analyses to view durability and strength of the trailing arms and leading link. What kind of shape/thickness would be needed to withstand forces contra what would be prime from the performance point of view.
Investigations of altering different suspension parameters as toe in/out, camber, caster, trail, etc. should be made and then tested to obtain better suspension characteristics.
Since the simulations performed in this thesis are executed with linear dampers, future simulations could be carried out with dampers based on non-linear properties as that will represent a more realistic scenario.
One component that can influence the vehicle´s performance significantly is the sway bar and hence simulations with different measure i.e. thickness, width, etc. of the sway bar should be tested to get the optimal configuration.

Since the full vehicle was created with measures deriving from the first step in the design process, measures as weights, properties, etc., were estimated. These parameters will be corrected to more accurate values as the project proceeds and will therefore give better results in the future. As the correct measures and weight are presented, the cube representing the centre of gravity could be removed. Instead the weights and position of each part could be implemented, and thus adding the effect from the mass of inertia.
The actual tyres used for the auto rickshaw should be tested to retrieve their properties as i.e. stiffness, material damping, friction values, etc. to eventually acquire more appropriate values and consequently more accurate results.

More kinds of tests should be performed; e.g. all ISO standard tests, as well as the tests done in this thesis but with the calibrations mentioned above; this to see if any improvements have been made.

If more advanced simulations are going to be carried out, a computer with good solving performance should be used since simulations made in this thesis could take up to four hours on a 2,66 GHz dual core 4GB RAM standard desktop computer.

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Appendix A – Suspension evaluation

Criterias	Rear suspensions										
	Double wishbone	McPherson strut	Chapman strut	Semi-trailing arm	Trailing arm	Multi-link	Torsion bar	Twist-beam	Swing axel	Rigid axle	De Dion tube
Simplicity (Low grade = low simplicity) *1	2	3	3	4	4	1	5	4	5	5	4
Space demand (independent 65x40x40cm) (Low grade = takes a lot of space) *2	4	1	1	5	5	1	5	3	3	2	3
Transverse axle *3	No	No	No	No	No	Yes/no	Yes/no	Yes	Yes	Yes	Yes
The suspensions with transverse axles was discarded at this point of the evalution stage								Discarded	Discarded	Discarded	Discarded
Handling (Low grade = poor handling) *4	5	3	4	2	2	5	1				
Dynamics (Low grade = poor dynamics) *5	4	3	3	2	2	5	1				
Economic aspect (manufacturing cost) (Low grade = high manufacturing costs) *6	2	3	3	4	5	1	5				
Economic aspect (maintenance, durability) (Low grade = high costs for the customer) *7	2	3	3	5	4	1	5				
Grade:	**19**	**16**	**17**	**22**	**22**	**14**	**22**	**Discarded**	**Discarded**	**Discarded**	**Discarded**

Grades from 1 to 5.

*1 Suspensions that contains many movable parts are considered as complex and graded with a low score

*2 Each independent suspensions limited movable volyme, within the wheel arch, are going to be 65 cm front to back, 40 cm top to bottom and 40 cm from outside of fender to inside of the wheel arch. High score equals low volyme requirements for static condition as well as moving within nessesary tolerances with regard to suspension geometry and body stability.

*3 The platform will be flat between the wheels and the ground clearence is decided to be approximately 165 mm which makes no place for a transverse axle. With this criteria a couple of rear suspensions can be discarded.

*4 With reference to "The Automotive Chassis: Engineering Principles - Second Edition" the handling aspects was aquired. Scored depending on how the vehicle remains grip to the surface, high score equals low handling.

*5 Dynamic behaviors with regards to the safety and comfort to the users as well as the suspensions ability to absorb the vertical and lateral forces.

*6 Evaluated depending on number of parts, complicated manufacturing processes. Expensive to manufacture gives a low grade.

*7 Poor durability and expensive for the customer to maintain and service gives a low grade.

Criterias	Front suspensions						
	Trailing link	Leading link	Earles fork (type of leading link)	Telescopic fork	Springer fork	Girder fork	Hossack/Fior fork
Simplicity (Low grade = low implicity) *1	4	4	3	4	2	2	1
Space demand (Low grade = takes a lot of space) *2	4	4	3	3	2	2	1
Possibility to mount on only one side *3	Yes	Yes	Yes	Yes/no	Yes/no	No	No
Only possible single sided forks were further evaluated						Discarded	Discarded
Handling (Low grade = poor handling) *4	3	4	5	3	1	2	5
Dynamics (Low grade = poor dynamics) *5	3	3	3	4	2	3	5
Economic aspect (manufacturing cost) (Low grade = high manufacturing costs) *6	4	4	4	5	4	2	1
Economic aspect (maintenance, durability) (Low grade = high costs for the customer) *7	4	4	5	4	3		2
Grade:	**22**	**23**	**23**	**23**	**14**	**Discarded**	**Discarded**

Grades from 1 to 5.

*1 Fork design that contains many movable parts are considered as complex and graded with a low score.

*2 If the solution takes a lot of space around the wheel it will get a lower grade because of the limited space in the front of the auto rickshaw.

*3 Beacuse of space specification its been decided to have one sided fork and to be the userfriendly.

*4 Light steering with less torque steer effect and anti-dive is prefered. Wheels capability to remain grip to the ground gets a higher grade.

*5 Less ungsprung mass, smaller steering vibrations and high vertical force absorbation gets higher grade.

*6 Evaluated depending on number of parts, complicated manufacturing processes. Expensive to manufacture gives a low grade.

*7 Poor durability and expensive for the customer to maintain and service gives a low grade.

Criteria	Simplicity	Space demand	Handling	Dynamics	Economic aspect (manufacturing)	Economic aspect (maintenance, durability)	Sum	Weight %
Simplicity *1	1	1	0	0	0	1	3	**8,33%**
Space demand *2	1	1	0	0	1	0	3	**8,33%**
Handling *3	2	2	1	1	1	1	8	**22,22%**
Dynamics *4	2	2	1	1	1	1	8	**22,22%**
Economic aspect (manufacturing) *5	2	1	1	1	1	1	7	**19,44%**
Economic aspect (maintenance, durability) *6	1	2	1	1	1	1	7	**19,44%**
						Total:	36	**100,00%**

*1 Simplicity is neccesary but not prioritised as first hand choice.

*2 There is limited space but enough to mount the suspension.

*3 One of the main reasons for this thesis is to improve the handling characteristics.

*4 One other main reason is to improve the dynamics, both from vehicle aspect as w ell as driver aspect.

*5 Main market target is low income earners, this demands for a low manufacturing costs.

*6 BNP per capita in India is 1176$ in 2010.

Criterias	Rear suspension						
	Double w ishbone	McPherson strut	Chapman strut	Semi-trailing arm	Trailing arm	Multi-link	Torsion bar
Simplicity (Low grade = low simplicity)	0,17	0,25	0,25	0,33	0,33	0,08	0,42
Space demand (independent 65x40x40cm) (Low grade = takes a lot of space)	0,33	0,08	0,08	0,42	0,42	0,08	0,42
Handling (Low grade = poor handling)	1,11	0,67	0,89	0,44	0,44	1,11	0,22
Dynamics (Low grade = poor dynamics)	0,89	0,67	0,67	0,44	0,44	1,11	0,22
Economic aspect (manufacturing cost) (Low grade = high manufacturing costs)	0,39	0,58	0,58	0,78	0,97	0,19	0,97
Economic aspect (maintenance, durability) (Low grade = high costs for the customer)	0,39	0,58	0,58	0,97	0,78	0,19	0,97
Grade:	**3,28**	**2,83**	**3,06**	**3,39**	**3,39**	**2,78**	**3,22**
Total result:	**Chosen**	**Discarded**	**Discarded**	**Chosen**	**Chosen**	**Discarded**	**Discarded**
The three types w ith the highest grade w ill be chosen for further evaluation.							

Criterias	Front suspension				
	Trailing link	Leading link	Earles fork (type of leading link)	Telescopic fork	Springer fork
Simplicity (Low grade = low implicity)	0,33	0,33	0,25	0,33	0,17
Space demand (Low grade = takes a lot of space)	0,33	0,33	0,25	0,25	0,17
Handling (Low grade = poor handling)	0,67	0,89	1,11	0,67	0,22
Dynamics (Low grade = poor dynamics)	0,67	0,67	0,67	0,89	0,44
Economic aspect (manufacturing cost) (Low grade = high manufacturing costs)	0,78	0,78	0,78	0,97	0,78
Economic aspect (maintenance, durability) (Low grade = high costs for the customer)	0,78	0,78	0,97	0,78	0,58
Grade:	**3,56**	**3,78**	**4,03**	**3,89**	**2,36**
Total result:	**Discarded**	**Chosen**	**Chosen**	**Chosen**	**Discarded**
The three types w ith the highest grade w ill be chosen for further evaluation.					

Appendix B – MATLAB script – Centre of mass calculations

```
%Centre of mass calculation for minimal load in X-,Y-,Z-coordinates


A = [1030, 2000, 1000];      %Coordinates for driver
B = [665, 315, 350];         %Coordinates for main battery
C = [280, 2400, 460];        %Coordinates for fuel engine
D = [665, 1400, 685];        %Coordinates for vehicle body/frame
E = [253,810,760];           %Coordinates for left seat
F = [982,810,760];           %Coordinates for right seat
G = [960,1665,670];          %Coordinates for driver seat
H = [665,2200,700];          %Coordinates for small battery
I = [700,2200,1000];         %Coordinates for dashboard

%Dotted plot on each part in the vehicle
plot3(A(1),A(2),A(3),'*')    %Plotting A on the plane
hold on                      %Keeps the plot, since it is erased after each
run
grid on                      %Makes the grid visible
plot3(B(1),B(2),B(3),'*')    %Plotting B on the plane
plot3(C(1),C(2),C(3),'*')    %Plotting C on the plane
plot3(D(1),D(2),D(3),'*')    %Plotting D on the plane
plot3(E(1),E(2),E(3),'*')    %Plotting E on the plane
plot3(F(1),F(2),F(3),'*')    %Plotting F on the plane
plot3(G(1),G(2),G(3),'*')    %Plotting G on the plane
plot3(H(1),H(2),H(3),'*')    %Plotting H on the plane
plot3(I(1),I(2),I(3),'*')    %Plotting I on the plane

%Coordinate limitation in X-,Y-,Z-direction
xlim([0,1330])
ylim([0,2746])
zlim([0,1740])
%Labeling each direction
xlabel('Width(mm)')
ylabel('Length(mm)')
zlabel('Height(mm)')

%Weights on each part
a = 49;              %Driver
b = 25.6;            %Main battery
c = 22;              %Fuel engine
d = 180;             %Body/frame
e = 10;              %Left seat
f = 10;              %Right seat
g = 5;               %Driver seat
h = 3.4;             %Small battery
i = 20;              %Dashboard
%Total mass: 325 kg
%Equations, weight on each coordinate
aA = (a*A);
bB = (b*B);
cC = (c*C);
dD = (d*D);
eE = (e*E);
fF = (f*F);
gG = (g*G);
```

```
hH = (h*H);
iI = (i*I);

%Equation, centre of mass in X-,Y-,Z-plane
S = ((aA+bB+cC+dD+eE+fF+gG+hH+iI)/(a+b+c+d+e+f+h+i));

%Plotting centre of mass on X-,Y-,Z-plane
plot3(S(1),S(2),S(3),'ro')
hold off            %After each run it will terminate the previous plot




%Centre of mass calculation for full load in X-,Y-,Z-coordinates


A = [230, 593, 1010];       %Coordinate for left rear passenger
B = [230, 1027, 1010];      %Coordinate for left front passenger
C = [1005, 1027, 1010];     %Coordinate for right front passenger
D = [1005, 593, 1010];      %Coordinate for right rear passenger
E = [960, 1573, 920];       %Coordinate for driver
F = [665, 315, 350];        %Coordinate for main battery
G = [280, 2400, 460];       %Coordinate for fuel engine
H = [665, 1400, 685];       %Coordinate for vehicle body/frame
I = [253, 810, 760];        %Coordinate for left seat
J = [982, 810, 760];        %Coordinate for right seat
K = [960, 1665, 670];       %Coordinate for driver seat
L = [665, 2200, 700];       %Coordinate for small battery
M = [700, 2200, 1000];      %Coordinate for dashboard

%Dotted plot on each part in the vehicle
plot3(A(1),A(2),A(3),'+')   %Plotting A on the plane
hold on                     %Keeps the plot, since it is erased after each
run
grid on                     %Makes the grid visible
plot3(B(1),B(2),B(3),'+')   %Plotting B on the plane
plot3(C(1),C(2),C(3),'+')   %Plotting C on the plane
plot3(D(1),D(2),D(3),'+')   %Plotting D on the plane
plot3(E(1),E(2),E(3),'+')   %Plotting E on the plane
plot3(F(1),F(2),F(3),'+')   %Plotting F on the plane
plot3(G(1),G(2),G(3),'+')   %Plotting G on the plane
plot3(H(1),H(2),H(3),'+')   %Plotting H on the plane
plot3(I(1),I(2),I(3),'+')   %Plotting I on the plane
plot3(J(1),J(2),J(3),'+')   %Plotting J on the plane
plot3(K(1),K(2),K(3),'+')   %Plotting K on the plane
plot3(L(1),L(2),L(3),'+')   %Plotting L on the plane
plot3(M(1),M(2),M(3),'+')   %Plotting M on the plane

%Coordinate limitation in X-,Y-,Z-direction
xlim([0,1330])
ylim([0,2746])
zlim([0,1740])
%Labeling each direction
xlabel('Width(mm)')
ylabel('Length(mm)')
zlabel('Height(mm)')

%Weights on each part
a = 49;             %Passenger
```

```
b = a;                  %Passenger
c = a;                  %Passenger
d = a;                  %Passenger
e = a;                  %Driver
f = 25.6;               %Main battery
g = 22;                 %Fuel engine
h = 180;                %Body/frame
i = 10;                 %Left seat
j = 10;                 %Right seat
k = 5;                  %Driver seat
l = 3.4;                %Small battery
m = 20;                 %Dashboard
%Total mass = 521 kg
%Equations, weight on each coordinate
aA = (a*A);
bB = (b*B);
cC = (c*C);
dD = (d*D);
eE = (e*E);
fF = (f*F);
gG = (g*G);
hH = (h*H);
iI = (i*I);
jJ = (j*J);
kK = (k*K);
lL = (l*L);
mM = (m*M);

%Equation, centre of mass in X-,Y-,Z-plane
S = ((aA+bB+cC+dD+eE+fF+gG+hH+iI+jJ+kK+lL+mM)/(a+b+c+d+e+f+g+h+i+j+k+l+m));

%Plotting centre of mass on X-,Y-,Z-plane
plot3(S(1),S(2),S(3),'ro')
hold off                %After each run it will terminate the previous plot
```

Appendix C – MATLAB script – Spring stiffness coefficients

```
%Distances


Lv = 2746;      %Length of vehicle (mm)
Wb = 1979.4;    %Wheel base (mm)
Wv = 1237;      %Width of vehicle (mm)
Lm = 1546.9;    %Length to centre of mass (mm)
M = 521;        %Sprung mass (kg)
Wd = 827;       %Distance between dampers (mm)
Lfwf = 264.46; %Length from front of vehicle to centre of front wheel (mm)
Llwm = 661;     %Width from left rear wheel to centre of mass (mm)
G = 9.81;       %Standard gravity (m/s^2)
Hfd = 30;       %Front spring free length (mm)
Hrd = 30;       %Rear spring free length (mm)

%Distance from spring & dampers to centre of mass
Lfwm = Lm-Lfwf;
Lrwm = Wb-Lfwm;
Wlwm = (Llwm-((Wv-Wd)/2))/Wd;

%Mass distribution
R = (Lfwm/Wb)*M;
F = Lrwm/Wb*M;
RL = Wlwm*R;

%Spring constant
Front = (F*G)/Hfd;
Rear = (RL*G)/Hrd;
```

Appendix D – Average values of coefficient of road adhesion [72]

Surface	Peak Value μ_p	Sliding Value μ_s
Asphalt and concrete (dry)	0,8-0,9	0,75
Asphalt (wet)	0,5-0,7	0,45-0,6
Concrete (wet)	0,8	0,7
Gravel	0,6	0,55
Earth road (dry)	0,68	0,65
Earth road (wet)	0,55	0,4-0,5
Snow (hard-packed)	0,2	0,15
Ice	0,1	0,07