Concept Development of a Single Wishbone Independent Rear Suspension

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Summary

The wheel suspension has a central function in any vehicle, with its main purpose to isolate its occupants from irregularities in the road surface and at the same time provide safe and predictable handling characteristics. To give the best comfort and handling it is essential to keep the unsprung weight as low as possible, which directly rules out the possibility to use a solid axle. An independent wheel suspension attached to the body using a subframe fulfils this requirement. The independent suspension developed in this work consists of two links, one lower triangular arm, wishbone, a spring which mounted on the wishbone, one tie-rod (a transversal link) and a damper which is rigidly connected to a knuckle. The project is an attempt to show that most requirements set for a modern multi-link suspension, which in some cases consist of up to six links, can also be fulfilled with this concept at a significantly reduced cost and weight.

The recommendation is therefore to continue to develop the suspension further to the degree that hardware prototypes can be built. These prototypes should be used to evaluate the concept and serve as demonstrators for management review and a decision aid for implementation in future platform projects.
Acknowledgements

The authors would like to take the opportunity to acknowledge the contributions of experts at Saab Automobile, without whom this project would not have come this far. Foremost we would like to thank Krister Fredén, manager of the suspension design team at Saab Automobile, for making it possible to make this degree work at Saab. His support and valuable input was paramount for the project.

In enabling us to perform the suspension kinematic- and elastokinematic analysis we would like to thank Gunnar Högström and Markus D Karlsson for their input and insight in all the suspension properties and suspension modelling that had to be considered. We would also like to thank Tomas Östlund and Roger Wallgren at the chassis tuning group for their support and enthusiasm for the project.

For the vehicle integration and interfaces, Carl-Johan Andersson at the suspension design team and Kennet Pettersson, manager of platform integration as well as Morgan Börjesson at the body design team gave invaluable input during the design phase.

Concluding we would like to give special credit to Tobias Brandin, being one of Sweden’s best vehicle dynamics specialists. After all, it was his idea of the single-wishbone suspension we had the privilege to realise during this concept development design project. Tobias used the Saab in-house kinematics program Kinemat to create the first geometry and created the first layout of the suspension in Microsoft Excel.
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1 Introduction

The wheel suspension has a central function in any vehicle, with its main purpose to isolate its occupants from irregularities in the road surface and at the same time provide safe and predictable handling characteristics [1,2,4]. A suspension essentially limited to one degree of freedom [1] (with the exception of steering on a front suspension), allowing vertical movement of the wheel. This vertical motion is controlled by a spring and damper. The spring temporarily stores the energy caused by the wheel and/or body motion, and this energy is then dissipated as heat by the dampers.

To give the best comfort and handling it is essential to keep the unsprung weight as low as possible [7]. An independent wheel suspension attached to the body using a subframe fulfills this requirement as well as it provides an individual control of each wheel. Independent rear suspensions are what most modern cars are using today, but with very different designs. As new cars are developed the rear wheel suspension tends to become more and more sophisticated, meaning, they consist of four, five or in some cases even six links. These multi-link suspensions can theoretically lead to an optimal wheel control [1] but they are difficult and expensive to develop. The independent suspension developed in this report consists of two links and a fixed shock absorber or damper. The project is an attempt to show that most requirements set for a multi-link suspension can also be fulfilled with this concept at a significantly reduced cost and weight.

The design of the rear wheel suspension developed in this project is based on some general coordinates where the suspension parts should be attached to the subframe and the knuckle, which is originally designed by Tobias Brandin. The final coordinates and design is developed through this 10 credit thesis work.

At a suspension team meeting at Saab during the project, the question was put forward what to name the suspension. The name which was voted was a “single-wishbone independent rear suspension” (SWIRS).

2 Goals in the development process

The goals for this rear wheel suspension project are:

- Focus on best-in-class road-noise isolation
- Light weight
- Low cost
- Suspension that fit cars with front-wheel drive (FWD) and all-wheel drive (AWD). The latter implying that also rear-wheel drive (RWD) could be possible with the concept.
- Scalable to fit cars from entry level such as Opel Vectra, to premium such as the Saab 9-5.
• Make room for the retractable roof for convertibles
• Provide excellent handling and comfort.
• Low/ wide load floor

3 Terminology

This chapter focuses on explaining the various terms used throughout the report to describe a suspension sub-system characteristic. The various terms can be divided into static, kinematic, elasto-kinematic and dynamic properties.

- Static Properties: Properties as they are laid out in a stationary, time-independent, loaded condition (called design height).
- Kinematic Properties: Describing the wheel motion throughout the wheel travel depending on the position and layout of the various pivots of the suspension.
- Elasto-kinematic Properties: Properties, which are dependent on a bushing and component compliance under various load conditions. These properties are assumed at steady-state conditions (braking, accelerating, cornering, etc.) [1]
- Dynamic Properties: The dynamic properties of the vehicle assess the transient behaviour of the vehicle but are not within the scope of this project. The inertial forces due to accelerations resulting from various manoeuvres are used to optimise the damping settings for the suspension.

It is the aim of the report to also give some discussion as to how certain characteristics contribute to vehicle performance.

3.1 Suspension Component Terminology

Below follows a brief description of components and terminology, which are not directly related to specific suspension characteristics, used in the report

3.1.1 Subframe

The subframe or cradle is the part where all the inner joints of the suspension links are attached. The subframe itself is attached to the body directly or through bushings [1,2,3]. A subframe is used on vehicles who have a self-supporting body structure (common on modern passenger cars), as opposed to a frame chassis (common on trucks). The purpose with an isolated subframe is to isolate vibrations from the suspension so it does not transfer into the passenger compartment, which otherwise would lead to increased noise. The subframe also has the purpose of simplifying the assembly of the suspension to the car body, the complete suspension can be assembled and pre-aligned at the supplier which makes the assembly at the main factory plant simple and fast.
3.1.2 **Knuckle**

At the knuckle, the wheel and all of the outer attachment points of the suspension are mounted [11]. The purpose of the knuckle is to translate the longitudinal and lateral forces that the wheel is exposed to to the components of the suspension. Also the brake calliper, brake shield and wheel bearing are attached to the knuckle.

3.1.3 **Wishbone**

A wishbone is a suspension component, which has three or four attachment points [8]. Two are attached to a subframe and one or two are attached to the knuckle. A three-point wishbone (often called A-arm) can transfer forces in the plane of the three attachments. A four-point control arm (often called H-arm) can also transfer torque about the lateral axis. A suspension can have one or two wishbones. The wishbone can be replaced with two links (see description of a tie-rod) or a rigidly mounted shock absorber [1]. The suspension in focus for the report has one three-point wishbone per side.

3.1.4 **Tie Rod**

The tie rod is a suspension component but with two attachment points [1,2,3], one at the knuckle and the other at the subframe (or to a steering gear in the case of a front suspension). A tie-rod can only transfer forces in the direction of the rod. Its purpose is to stabilize the knuckle/wheel during lateral forces, the suspension will allow the knuckle to move under impact of these forces, and the tie rod then controls the toe angle during this movement.

3.1.5 **Spring**

The spring’s lower end is mounted on the wishbone (in the SWIRS) and its upper end to the body. The spring carries the weight of the body and allows vertical motion of the wheel [1]. By altering the stiffness and height of the spring the handling of the vehicle can be altered. A stiffer spring leads to reduced body movement and improved handling up to the limit where the wheel no longer can follow the irregularities in the road surface and starts to bounce [1]. A softer spring leads to more comfortable characteristics of the vehicle as less force is required to move the wheel.

3.1.6 **Shock Absorber/Damper**

The shock absorber or damper, is a vital part of the suspension which purpose is to absorb the energy stored in the spring during vertical movement of the suspension [1,4]. The force created by the damper is a function of speed, compared to the spring which force is a function of displacement. On the Single Wishbone IRS (as on the common McPherson suspension), the shock absorber is fixed to the knuckle enabling it to also carry lateral and longitudinal forces. This results in additional challenges of the side-loads causing friction and additional wear in the damper, but has the advantage of
removing two links or one wishbone from the suspension. Another limitation of a fixed
damper concerns the wheel motion, which is more deeply discussed in the sections
below.

3.1.7  Jounce Bumper

Before the suspension is totally compressed a rubber or polyutherane bushing is the
final stop for the suspension movement [3,4]. The jounce bumper is typically placed
either in the centre of the spring or as a part inside the shock absorber.

3.1.8  Rebound Bumper

The opposite of jounce, when the suspension is totally stretched out a rubber bushing
inside of the shock absorber stops the rebound travel [3,4].

3.1.9  Anti-roll Bar

This is a part of the suspension that strives to prevent the body to roll during cornering.
It is a transversal torsion bar attached as far out, close to the knuckles as possible, to
connect both left and right side of the suspension [11]. During cornering when the body
rolls and outer wheel travels towards jounce and inner wheel to rebound, the tension
created in the anti-roll bar strives to decrease this roll. The design specification of the
anti-roll bar is a compromise between single-wheel-impact comfort and an acceptable
roll movement of the body [3]. The stiffer the anti-roll bar is made, the worse will the
lateral comfort be. In the case of a too stiff anti-roll bar, when one side of the vehicle
travel over a bump, undesirable head toss of the passengers will be a result [11].

Additionally the anti-roll bar has an important role in balancing the load distribution
between the front and rear suspensions in a cornering situation [3]. The stiffer the rear
anti-roll bar is the more load is transferred to the rear suspension, thus leading to more
oversteer. The opposite is valid for a weaker anti-roll bar, thus leading to more
understeer.

3.1.10 Wheel Offset

The wheel offset is the distance between the wheel centre line of the rim and the hub
flange, which is mounted against the wheel [3].

3.1.11 Unsprung weight

The unsprung weight of the suspension is the tire, wheel, rotor, calliper, knuckle and
half the weight of the suspension links [1,2,4].

3.1.12 Sprung weight

The sprung weight is the part of the vehicle that is sustained by the springs [1,2,4].
3.2 Static Properties

3.2.1 Design Mass
Design mass is the most common load condition from which most properties are defined. The definition can vary from various car manufacturers, but is for GM; 2 adults + 70 kg luggage and ½ full tank [9].

3.2.2 Curb Mass
Curb mass is the unloaded vehicle. Usually the wheelhouse openings are designed for the best fit in curb position because it is in this condition the car is most commonly viewed in a showroom or at a parking lot.

3.2.3 GVM, Gross Vehicle Mass
Indicate the maximum loaded condition for the vehicle. All structural dimensioning is done to GVM.

3.2.4 Toe-in
Viewed from above, toe-in is the angle between the vehicles x-axis (anti-parallel to the driving direction) and the x-axis of the wheel, see figure 1. It is positive when the front end of the wheel is turned towards the vehicle, [10,11].

![Figure 1 - Toe-in](image)

3.2.5 Camber
Viewed from the rear camber is the angle between the wheels z-axis and a vertical to the plane of the road, see figure 2 and 3. It is negative if the top of the wheel is inclined inwards the car. Negative camber results in a pre-load of the wheel in the lateral direction [4] resulting in an improved steering response and lateral grip, [11]. Too much camber though, leads to increased tire wear why the amount of camber is usually less than 1½-degree [2] at design weight.
Single Wishbone Independent Rear Wheel Suspension – Concept Development

Figure 2 - Camber Angle

Figure 3 - Negative & Positive Camber
3.2.6 Body Roll Centre

When the car takes a turn the body rolls around the body roll centre [1,2,4,11], which is the height of the intersection between a line from the tire contact point to the instantaneous centre of the suspension and the centre line of the car. The location of the body roll centre is determined by the design of the suspension. The momentum that causes the car to roll is dependent of the distance between the body roll centre and the vehicle's centre of gravity. For a passenger car the body roll centre should be close to the ground to avoid significant changes in track width during jounce [11], this would otherwise lead to poor course stability when the car travels on bad roads. Nevertheless, a too low roll-centre height will lead to increased roll of the vehicle due to the increased distance to the centre of mass, a compromise for the specific vehicle which is to be designed is required, see figure 4.

3.2.7 King Pin Inclination

The amount of camber compensation on a suspension system that uses a fixed damper is partially decided by the inclination of the damper. The angle between the wheels centre line and the intersection line between the upper mount and the outer wishbone mount is the king pin inclination [8,10,11]. As the wheel travels towards jounce an increase in negative camber will be achieved, see figure 5.
3.3 Kinematic Properties

3.3.1 Wheel Travel

Wheel travel is denoted jounce when the wheel moves upward, and rebound for vertical motion in the opposite direction.

3.3.2 Anti Lift

The anti lift is the definition of the rear wheels suspension’s ability to reduce the amount by which the tail rises when the brakes are applied [8,10]. The design of the suspension controls the anti lift. The vertical force on the suspension is measured when the brakes are applied which results in a longitudinal force at the tire patch. The formula for calculating anti lift is “(wheel base/ height of CG)*(vertical force/longitudinal braking force)*0,3” [4] where 0,3 is the amount of brake force, 30%, operating at the rear wheels.

3.3.3 Anti Squat

Anti squat is the definition of the rear wheel suspension’s behaviour to reduce the rear end of the car to move down, sink, during acceleration. Anti squat is only valid on a driven rear axle, in this case with the AWD system [2,3,6]. During acceleration when longitudinal forces are applied to the wheel centre the resulting vertical force on the suspension is measured. The vertical force is related to the load transfer from the front to rear axle. The formula for calculating anti squat is “(wheel base/ CG height)*(vertical force/traction force)*0,5” [4] where 0,5 is 50% of total tractive force.
3.3.4 Wheel Travel / Spring Travel Ratio

The spring should be placed as close to the wheel centre as possible to give the best possible wheel travel / spring travel ratio [10]. Assuming the outer control arm point to have a ratio of 1:1, a simplified method to calculate the ratio is A/B. The optimal ratio is 1:1 since this means the vertical force from the wheel goes directly to the body without going through the suspension. If the spring is placed further away from the wheel centre more loads will pass through to the inner wishbone mount, resulting in reduced comfort, increased cost and weight.

![Diagram of Spring Ratio]

Figure 6 - Spring Ratio

3.3.5 Wheel Travel / Damper Travel Ratio

The most optimum damper ratio is often regarded to be 1:1 for the same reason as for the spring ratio discussed earlier [10]. The higher the ratio, the more force is required for the same damping force on the wheel. If the ratio is lower than 1:1 though, the speed of the damper increases, generating more heat. Likely to be more important than the ratio though, is the stiffness of the connection (see the damper stiffness paragraph).

3.3.6 Roll Centre Height Variation

Roll centre height variation is a ratio describing how much the roll centre varies in height with wheel travel. It is desirable to have the front and rear roll centre variation as close to each other as possible for an even roll motion [7,10,11]. A McPherson front suspension has a ratio of about 2:1. If the rear suspension would only have 1:1, the front suspension will naturally roll more than the rear.
3.3.7 Ride Steer

The ride steer is the slope (derivative) of toe angle as a function of wheel motion and is measured in deg/100 mm. As the rear wheel travel towards jounce the toe in should increase thus giving the car a slightly more characteristic of being understeer, safe handling [2], although this should not be too much because at the same time the car should feel active, sporty to handle for the driver.

3.3.8 Ride Camber

Camber compensation during cornering; when the car is turning the body rolls. If the camber angle would be constant during wheel travel it would lead to more positive camber during the vehicles turn and therefore less grip [3], see figure 7. To obtain maximum grip an increased negative camber is favourable during the vehicles roll, see figure 8. Ride Camber is the slope of the camber curve as a function of wheel travel.

![Figure 7 - Without Camber Compensation](image7.png)

![Figure 8 - With Camber Compensation](image8.png)

3.4 Elasto-kinematic Properties

3.4.1 Lateral Force Compliance Steer

During cornering a lateral force is applied to the suspension through the tires contact patch. This force will lead to alteration of the toe in angle of the outer wheel. The wheel will turn around its dynamical centre that’s decided by the stiffness of the bushings in the suspension, the unit are deg/kN [10]. Most car manufacturers strive towards toe-in on the rear wheels and toe-out on the front wheels for the car to understeer during on-the-limit cornering for safe handling, see figure 9.
3.4.2 Lateral Force Compliance Camber

The force that affected the toe in during high speed turning is also affecting the camber due to the deflection of the bushings [10], suspension components and the wheel bearing. Camber compliance has a large influence during cornering by reducing the lateral grip [1] and should therefore be minimized. This movement is measured in deg/kN.

3.4.3 Brake Force Steer

During braking of the vehicle a longitudinal force will be applied to the suspension at the tyre contact patch. As the bushings of the suspension allow the wheel to travel
rearward when brake force is applied and the distance between outer and inner mount of the wishbone is shorter than of the tie-rod, the rearward movement of the wheel will result in an increased toe-in angle [2]. Toe-in on the rear wheel is especially important during braking while cornering when the outer wheel is most loaded. This movement is measured in, deg/kN, see figure 11.

![Figure 11 - Brake Force Steer](image)

### 3.4.4 Aligning Torque Compliance Steer

The lateral displacement of the tire contact patch combined with the compliance camber, see 2.4.2, due to lateral forces longitudinal forces are being applied inside of the centre plane of the wheel. This effect causes an increase in tire aligning torque in driving wheels [10]. In a rear wheel drive design this torque has an understeer effect during acceleration while cornering. The displacement of the tire patch of the rear wheels will also contribute to an understeer effect during braking while turning. This effect is measured in deg/kNm, see figure 12.
3.4.5 Wind-up stiffness

Wind-up stiffness describes how much rotation of the knuckle a certain brake force gives [1]. Since rotation of the knuckle gives a change in the geometry, it will have a steering effect. The aim is therefore to have as high wind-up stiffness as possible. If wind-up stiffness is coupled to longitudinal flexibility (as is the case on many suspensions), there is a compromise between how much toe-change is allowed and the longitudinal flexibility.

3.4.6 Damper stiffness

Damper stiffness is measured as how much deflection a certain damper force results in without changing the wheel or body motion. Since the function of the damper is to absorb energy stored in the spring because of body and/or wheel motion, it is desirable to have the coupling between the wheel and the body as rigid as possible. If not, body and wheel control is compromised, resulting in reduced comfort and handling performance [8,6].

3.4.7 Damper Side Load

The damper side load should be as low as possible [1]. It is measured at the top mount of the damper. Any side load on the damper affects the friction inside the damper, high side load results in high friction which leads to reduced damping sensitivity, and therefore it should be as low as possible.
4 Discussion

4.1 Benchmarking

Studying different solutions that are already at hand gave valuable background to the assessment of the investigated concept. Some key design guidelines were a result of a benchmarking study done by the Saab NVH department [5] on rear wheel suspensions on cars in the premium segment such as Audi A4/A6 (trapezoidal link), BMW 5/6/7-series (trapezoidal link), Volvo S60/V70/S80/XC90 (4.5-link) and Mercedes D/E/S (5-link). One thing they all have in common is that they all use a subframe for attachment of the links. The use of an isolated subframe gives a good isolation for vibration/noise that otherwise would be transferred into the passenger compartment, but it is also a matter of cost. The reason for the improved isolation in the discussed case is that of double-mass isolation [5]. The different levels of sophistication between the models regarding the wheel suspensions were obvious. Entry models such as Volvo S40/V50, Opel Vectra, VW Golf 5, Ford Focus use a simpler type of independent rear suspensions where some links are attached to a non-isolated subframe and some are attached directly to the body, thus resulting in more road noise to be transferred to the passenger compartment. Nevertheless these latter type of suspensions still have similar performance with regard to handling and ride comfort. The focus for the single-wishbone suspension was on improving the road-noise isolation without increasing the complexity of the suspension. This resulted in a fully isolated subframe with a minimum of links.

The single-wishbone suspension is a derivative from a McPherson damper suspension, which is a very common suspension for both front and rear suspensions. Nevertheless, it seemed to be more uncommon with the spring located on the control arm. Vehicles using this concept that were found in the survey are illustrated beneath.
4.1.1 Mercedes Sprinter Front Suspension (1995)

4.1.2 Mercedes 190 Front Suspension (1982)
4.1.3 Autobianchi Rear Suspension (1969)

4.1.4 Ford Escort Rear Suspension
4.2 Development tools used in the Project

Throughout the project different tools have been used, primarily the CAD program Unigraphics (UG), and the rigid body dynamics simulation program MSC.Adams Car. UG has been used to develop 3D animated models of the suspension. It was also used to perform kinematic motion tests of the suspension to confirm that no collisions occurred between the parts within the model during the suspensions travel from jounce to rebound. With MSC.Adams, kinematic and elasto-kinematical calculations were performed. MSC.Adams involved extensive amount of work to evaluate the final coordinates for the suspension, the characteristic of the suspension regarding toe in and camber during different load conditions as lateral force compliance steer, lateral force compliance camber, brake force steer, aligning torque compliance steer, etc..

4.3 Concept development

4.3.1 Basic Layout

The first concept is developed from a basic design that consists of one lower triangular arm, wishbone, the spring that are mounted on the wishbone, one tie-rod (a transversal link) and a damper of McPherson type that also can control lateral and longitudinal forces. McPherson suspensions are commonly used as a front suspension where the spring is mounted on the damper, above the wheel. McPherson damper suspensions are also used on many rear suspensions but then often with two transverse links and one trailing link. This suspension is also called a tri-link suspension. [1,3]

The main advantage with the McPherson damper is that it is cost and weight effective but also that the rigid connection between the wheel and the damper (since the damper is bolted unto the knuckle) which results in an excellent wheel control on poor roads. The latter is well known among rally car engineers, which is why the concept is commonly used for all four wheels. The disadvantage on the other hand is that the motion of the wheel is directed by the inclination of the damper, thus imposing limitation wheel forces. Because the damper has a linear motion (in the direction of the damper piston), and the wishbone and tie-rod a circular motion (around their inner pivots), the wheel will have a non-linear camber motion. The amount of camber-compensation is limited to the damper side-view inclination, which again is limited to the amount of available space between the body structure and the wheel envelope, see figure 14. The improved control of the wheel motion is also the reason why road-racing cars as well as many rear-wheel suspensions reviewed the use of an upper control arm for improving the amount of camber compensation, thus using the damper only for vertical control.

The disadvantage is that the rear-view inclination of the damper (called the king-pin inclination) and the installation of the spring on the damper intrude too much into the luggage compartment, making it unsuitable for wagon-type cars. For example the Ford Mondeo uses a McPherson rear wheel suspension on the sedan and a 4-link suspension
on the wagon. Installing the spring on the wishbone puts the spring below the luggage compartment. Additionally the damper has been moved behind the wheel, making it possible to lower the damper. This installation is both favourable for the luggage compartment size as well as it enables the stowage of a convertible top-stack.

4.3.2 Transversal Leaf spring

The second study that was made involved the use of a transversal leaf spring instead of a steel spring. This would provide a very good wheel travel/spring travel ratio since the spring would be mounted almost directly above the wheel centre. But it proved to be very difficult with the packing with this type of unconventional design and therefore it was cancelled.

4.3.3 Installation of the damper

Different dampers will be used on this suspension and for package considerations the largest one, a nivomat damper was used. This damper will be an option for the customer, which dimensions are described in appendix C. The nivomat damper is a self-levelling damper provided by ZF Sachs. By placing the damper as low as possible on the knuckle, a low point for the upper damper mount were obtained leaving great room for packaging of the convertible top, also, the low installation means that it makes it possible to give the damper a larger camber angle resulting in an increased king pin inclination, which gives better handling when the body rolls during high speed turns.

Figure 14. Tire Envelope Clearance
Another requirement is the load through space between the upper mounts of the damper, on an ordinary McPherson damper where the spring is attached on top of the damper the load through space requirement would not have been met. On this design were the damper and spring is separated and the damper has a low installation it meets the requirements of load through space, see figure 14. The control of the negative camber in this rear wheel suspension design is partially controlled with the inclination of the damper. During a high-speed turn where the body rolls the suspension on the outside travels to jounce and the tire patch are reduced due to the deflection of the tire, as a result the handling of the car is severely degraded. Giving the damper a larger negative camber angle at installation, see figure 15, that leads to a greater king pin inclination, camber compensation, this motion can be controlled giving the tire the best possible angle for grip during the suspensions travel. The more the wheel travels towards the jounce stop the greater the negative angle of camber is produced by the suspension, this gives the suspension the possibility of achieving high lateral g-forces during high speed manoeuvres when the body rolls.

4.3.4 Installation of the spring

On an ordinary McPherson damper the spring is attached on top of the damper. This gives a very high upper damper mount installation and that is not desirable for packaging reasons. On this design the damper and spring are separated. As were discussed in the section of installation of the damper the same general requests are favourable here, the spring should be placed as close to the wheel centre as possible thus giving the best possible wheel travel / spring travel ratio. The longer distance between the spring and the wheel centre more loads will pass through to the inner wishbone joint and further to the subframe, resulting in reduced comfort. As the damper is located behind the wheel centre and the drive shaft for the AWD cars, the only free space left is on the control arm. Here the spring is placed as close to the rim and knuckle for obtaining the best possible travel ratio. The spring has the same inclination as the damper. For supporting the upper part of the spring a bracket needs to be designed,
when the jounce stop is located inside the damper the force on the body is only limited to what the spring produces.

### 4.3.5 Wishbone

The design of the wishbone is made with regard of that it will be able to control both lateral- and longitudinal forces. The forward wishbone mount will be exposed for both these forces. To achieve the best possible control of the longitudinal forces the forward wishbone mount should be placed as close to wheel centres x-plane as possible. The outer and inner wishbone mount will be exposed to most lateral forces that mean they must be stiff in the y-direction for good control of the wheels movement. For best possible isolation against road noise the wishbone is attached to a subframe instead of directly to the body, see figure 15. The outer mount is placed on the knuckle, very close to the wheel centre. Together with the damper the wishbone controls the suspensions camber angle at the suspensions travel. Due to that the outer mount is located lower in z-plane than the inner mount a difference in wheelbase will occur during wheel travel against the jounce stop. This together with the angle of the damper leads to increased negative camber of the wheel during jounce, which is favourable.

### 4.3.6 Tie rod

During jounce of the rear wheel, the wishbone and tie rod controls the wheels movement around its z-axis [1], see figure 15. This is achieved due to that the rear-view inclination is greater than that of the wishbone, thus the outer point of the tie rod moves along a wider circle than the outer joint of the wishbone. As the wheel moves towards its jounce stop an increase in positive toe angle is the result. During braking and high-speed manoeuvres of the vehicle the tie rod helps to sustain an acceptable control of brake force steer and lateral force compliance camber. The dynamical centres z-axis where the wheels are turning around can be tuned by the stiffness of the tie rods bushings.
4.3.7 Subframe

The subframes developed in this report are to be used as guidelines at production of possible prototypes, not as a final design for mass-production. Two different types of subframes were developed, fwd and AWD. There is a difference due to all the additional components that an AWD system involves. The components that is added is mainly the attachment points for the differential, see figure 17, but they are still very similar which helps to keep the costs down. On the fwd subframe a very simple and light design could be used, see figure 16, the main issue during development of this was to make room for the fuel tank. The coordinates were the subframe is to be attached are based on an ongoing development project, see figure 18. The bushings that are used for attaching the subframe to the body are soft in the z-axis for best comfort and stiffer in the y-, and x-axis for better handling. By altering the stiffness of these bushings the toe in of the suspension can be tuned. When the suspension is under influence of lateral or longitudinal forces the subframe will also be affected and move accordingly to this. The main thing is to keep the bushings as soft as possible in the z-axis direction for best possible comfort and isolation but without decreasing the handling characteristics.
Figure 15 - Front Iso-view FWD

Figure 16 - Front Iso-view AWD
5 Results

5.1 Static Properties

All of the following parameters have been calculated using MSC.ADAMS and Matlab (see Appendix B), see table 1 for a complete summary of the design factors. For calculations in MSC.ADAMS a new model of the suspension was made, with this model different load cases were evaluated and the model modified until the targets given in table 1 were met as far as was possible. These targets are typical targets derived from previous experience, benchmarking and vehicle level targets done at Saab Automobile.

The model was a rigid-body model with linear bushings with all the pivot points and linear bushing rates described in table 2. The fact that the model was a rigid model means that some properties, where the flexibility of the components give a large contribution, are not entirely correct. Lateral force compliance camber, is such a property where the flexibility of the damper and subframe can give significant contributions. Additionally, the non-linear behaviour, when the suspension bottoms out can not be predicted by the model. Nevertheless, most of the static design factors in figure 21 are considered as sufficiently validated, given requirements for a concept development.
5.1.1 Toe-in

Since the tolerance chain of all the suspension and body parts which influence the toe angle is too long, and the tolerance requirements is likely too strict to avoid toe-in adjustment in production. For a concept car, also the slope of the toe-curve will be adjustable. The inner mount of the tie rod is fastened with an eccentric bolt allowing adjustment in direction of the z-axis. By altering the position along the z-axis of the inner mount the slope of the toe in curve can be altered, thus producing the right toe in during jounce. The tie rod bar is designed with the possibility to adjust its length in direction of the y-axis, with similar principals as the steering rod in the front suspension. The adjustment of the tie rods length is required so correct alignment can be achieved during alteration of the wishbones inner mount, see 4.1.2. With the basic design created in this report the slope of the toe in is almost linear with a maximum of 0.19 degrees at 100 mm jounce, see figure 18.
5.1.2 Camber

To alter the camber of the suspension the distance between the dampers low point of installation compared to the vehicles centre line needs to be changed. The eccentric bolt of the inner mount of the wishbone and the adjustable length of the tie rod accomplishes this. By altering these design parameters the final tuning of the static camber can be achieved. At design height the suspension has a nominal camber angle of -1°. The slope has a slightly digressive character, giving a camber angle of -2,6° at 100 mm jounce, see figure 19. This is enough to produce satisfactory camber compensation during jounce.

Figure 19 - Toe angle versus Wheel Travel
5.1.3 Body Roll Centre

The position of the roll centre is decided by the inclination of the damper and the angle of the wishbone relative to the ground. During the design the outer and inner mount of the wishbone was moved around until the final position was found that produces the right characteristics for the suspension. The body roll centre height in this design is 95 mm above ground.

5.1.4 King Pin Inclination

The king pin inclination which is a set value and will not be possible to tune during prototype testing is 18 degrees. This is sufficient to produce the desired increase of negative camber angle during jounce, camber compensation.

5.2 Kinematic Properties

5.2.1 Anti Lift

The suspensions characteristic to produce lift during braking is reduced by 60 % of maximum lift. With 70 % of the brake load operating on the front wheels.
5.2.2 Anti Squat

With the AWD version and a ratio of tractive force of 50/50 between front- and rear-axis the squat is reduced by 22% of total squat due to the design of the suspension.

5.2.3 Wheel Travel / Spring Travel Ratio

Due to collision between the front mount of the wishbone and the fuel tank during design of the suspension, the position of the front mount was moved towards the wheel. This produced a ratio of 1:1.52 and is not a desirable ratio because it leads to decreased comfort due to more vertical force will be translated into the front and inner bushings of the wishbone during jounce. To improve the ratio the spring needs to be redesigned so it can be moved further towards the wheel centre or the front mount of the wishbone moved back to its previous position. This proved to be one of the projects more challenging problems to solve. A redesigning of the subframe proved to be the best solution, due to the changes of the subframe the front mount of the wishbone could be moved away from the wheel centre thus producing a better wheel travel/spring travel ratio of 1.44.

5.2.4 Wheel Travel / Damper Travel Ratio

The ratio of wheel travel/damper travel is 1:1.07, which is very desirable. This is obtained due to the favourable location of damper; directly attached to the knuckle.

5.2.5 Roll Centre Height Variation

The variation of the roll centre height is dependent on the angle variation of the wishbone during jounce. When the suspension jounces the roll centre is lowered. The calculated relationship between roll centre at design height and when the suspension is fully jounced is 1:2.11.

5.2.6 Ride Steer

During jounce the suspension characteristics gives an increase of toe-in angle with 0.2 deg/mm. The linearity of the toe-curve was tuned with the length of the tie-rod and the slope with the height of the inner pivot point.

5.2.7 Ride Camber

During jounce a camber increase of 2.5 deg/100mm is sustained, which gives the desirable camber compensation.

Figure 21 illustrates the non-linearity of the single-wishbone suspension camber curve. Two different versions of the suspension are compared. Version 0.17 has a 470 mm long wishbone as opposed to v0.20 and v0.21 with 420 mm. As can be seen in figure 21 the longer the wishbone, the more linear the toe-curve. Appendix D contains code for calculating the perpendicular distance between p7 (wishbone outer) and the line between p1 and p2 (wishbone front and rear inner points).
5.3 Elasto-kinematic Properties

5.3.1 Lateral Force Compliance Steer

During high speed cornering where a considerable lateral force is applied to the suspension an increase in toe in of the outer wheel is sustained by 0,029deg/kN. Thus producing a more understeer character of the vehicle. This property was tuned with the outer and inner-rear wishbone bushings. The softer these bushings, the more understeer.

5.3.2 Lateral Force Compliance Camber

The change of camber angle due to lateral force as a result of high speed cornering is decreasing with 0,11deg/kN. As previously discussed, this property is also largely influenced by the stiffness of the wheel bearing, knuckle, damper and subframe. Component flexibility was not considered during this project.

5.3.3 Brake Force Steer

When brake force is applied the longitudinal force on the suspension contributes to a larger toe in angle by 0,055deg/kN, according to the red curve in figure 22. Brake force steer was tuned by moving the wishbone inner-rear pivot point forward or backward.
The more forward this point was located, the more understeer it produced during braking.

5.3.4 Brake Bias

When the brakes are applied and the friction between left and right wheel differs there is a twisting movement of the subframe (if it is isolated) leading to a reduced toe-in to 0,01 deg/kN for single-wheel braking, see figure 22.

5.3.5 Aligning Torque Compliance Steer

During acceleration while cornering the aligning torque is 1,1 deg/kN. This is evaluated by plotting the changes in toe-in versus aligning torque.
5.4 Static Design Factors

<table>
<thead>
<tr>
<th>Units</th>
<th>Target</th>
<th>v0.21</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring Ratio</td>
<td>0.9-1.25</td>
<td>1.44</td>
</tr>
<tr>
<td>Damper Ratio</td>
<td>1.0-1.2</td>
<td>1.07</td>
</tr>
<tr>
<td>Roll Center Height</td>
<td>70-90</td>
<td>91</td>
</tr>
<tr>
<td>Anti-Lift</td>
<td>40-90</td>
<td>64</td>
</tr>
<tr>
<td>Anti-Squat</td>
<td>10-30</td>
<td>24</td>
</tr>
<tr>
<td>Ride Steer deg/mm</td>
<td>0.1-0.2</td>
<td>0.19</td>
</tr>
<tr>
<td>Ride Steer dev@ 50 deg/mm</td>
<td>+/- 0.02</td>
<td>0.003</td>
</tr>
<tr>
<td>Ride Camber deg/mm</td>
<td>2.5</td>
<td>2.5</td>
</tr>
<tr>
<td>Ride Camber dev@ 50 deg/mm</td>
<td>+/- 0.25</td>
<td>0.2</td>
</tr>
<tr>
<td>Camber @ GVM deg</td>
<td>&lt; 2.5</td>
<td>2.3</td>
</tr>
<tr>
<td>Roll Center Height Var mm/mm</td>
<td>1.5-2.0</td>
<td>2.1</td>
</tr>
<tr>
<td>Lateral Force Steer deg/kN</td>
<td>-0.03-0.05</td>
<td>0.04</td>
</tr>
<tr>
<td>Lateral Force Camber deg/kN</td>
<td>&lt; 0.25</td>
<td>0.11</td>
</tr>
<tr>
<td>Brake Steer deg/kN</td>
<td>0-0.05</td>
<td>0.05</td>
</tr>
<tr>
<td>Brake Bias deg/kN</td>
<td>0-0.05</td>
<td>0.01</td>
</tr>
<tr>
<td>Aligning Torque Steer deg/kNm</td>
<td>&lt; 1.3</td>
<td>1.3</td>
</tr>
<tr>
<td>Long Compliance mm/kN</td>
<td>3.5-4.0</td>
<td>3.8</td>
</tr>
<tr>
<td>Compliance WC-lateral mm/kN</td>
<td>&lt; 0.5</td>
<td>0.6</td>
</tr>
<tr>
<td>SVSAA deg</td>
<td>-3 - -6</td>
<td>-5.1</td>
</tr>
</tbody>
</table>

Table 1 - Static Design Factors

5.5 Suspension coordinates & bushing stiffness

During the development work in MSC.ADAMS the coordinates were moved around until the best locations were found. When tuning for the elastic kinematical behaviour the stiffness of the bushings were evaluated for achieving best comfort and handling, see figure 22.

<table>
<thead>
<tr>
<th>hardpoint name</th>
<th>x_value</th>
<th>y_value</th>
<th>z_value</th>
<th>Radial 1</th>
<th>Radial 2</th>
<th>Axial</th>
</tr>
</thead>
<tbody>
<tr>
<td>p01_lca_front</td>
<td>4167.0</td>
<td>-425.0</td>
<td>184.0</td>
<td>0.4</td>
<td>1.2</td>
<td>0.3</td>
</tr>
<tr>
<td>p02_lca_rear</td>
<td>4461.0</td>
<td>-296.0</td>
<td>144.0</td>
<td>10.0</td>
<td>10.0</td>
<td>1.0</td>
</tr>
<tr>
<td>p05_wheel_center</td>
<td>4468.0</td>
<td>-782.0</td>
<td>247.0</td>
<td>10</td>
<td></td>
<td></td>
</tr>
<tr>
<td>p06_tierod_outer</td>
<td>4583.0</td>
<td>-750.0</td>
<td>132.0</td>
<td>25.0</td>
<td>25.0</td>
<td>10.0</td>
</tr>
<tr>
<td>p07_lower_ball_joint</td>
<td>4458.0</td>
<td>-750.0</td>
<td>132.0</td>
<td>10.0</td>
<td>10.0</td>
<td>1.0</td>
</tr>
<tr>
<td>p08_strut_upper</td>
<td>4613.0</td>
<td>-566.0</td>
<td>660.0</td>
<td>10.0</td>
<td>10.0</td>
<td>10.0</td>
</tr>
<tr>
<td>p10_tierod_inner</td>
<td>4618.0</td>
<td>-273.0</td>
<td>148.0</td>
<td>25.0</td>
<td>25.0</td>
<td>10.0</td>
</tr>
<tr>
<td>p11_strut_lower</td>
<td>4618.0</td>
<td>-668.0</td>
<td>160.0</td>
<td>fixed to knuckle</td>
<td></td>
<td></td>
</tr>
<tr>
<td>p21_front_subframe</td>
<td>4059.0</td>
<td>-504.0</td>
<td>152.5</td>
<td>0.75</td>
<td>1.5</td>
<td>1.5</td>
</tr>
<tr>
<td>p22_rear_subframe</td>
<td>4703.0</td>
<td>-470.0</td>
<td>320.0</td>
<td>0.8</td>
<td>1.5</td>
<td>1.5</td>
</tr>
<tr>
<td>p41_spring_upper</td>
<td>4380.0</td>
<td>-615.0</td>
<td>346.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>p42_spring_lower</td>
<td>4380.0</td>
<td>-655.0</td>
<td>150.0</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2 - Kinematic Hardpoints and Bushing Linear Stiffness (version 0.21)
5.6 Proposals for Further Work

5.6.1 Transversal leaf spring

In the beginning of the concept development the transversal leaf spring were cancelled due to difficult packaging problems. But, as the development progressed some significant changes were made, such as the subframe now only goes below the differential. By doing so more space is available for a transversal leaf spring between the body and differential. The leaf spring would be attached to the body and at the top of the knuckle, thus producing a wheel travel / spring travel ratio of ~1:1.

5.6.2 Damper Side Loads

The damper side load is regarded acceptable as it is now, but if comfort is to be further enhanced some attention to lower the side load of the damper should be made.

5.6.3 Component Stiffness Analysis

The ADAMS model should be further enhanced so that also the component flexibility is included. This can be done, using finite element models.

5.6.4 Structural Strength

Using various standard load cases and a flexible model of the suspension, the structural integrity of the various components should be evaluated.

5.6.5 Road Noise Isolation

Using finite element models of the suspension, car body and passenger compartment, a road noise isolation assessment can be made. This should be done in order to evaluate the road noise isolation properties for this suspension.
5.7 Mass & Cost Estimations

As was mentioned in the introduction of the report, for best comfort and handling the unsprung mass of the suspensions needs to be as low as possible, during the design of this suspension it was clear that it would be significant lighter than the present rear suspension of Saab 9-3. This is mainly accomplished by reducing the number of components in the suspension and as a result of this the total cost of the suspension is reduced.

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass [kg]</th>
<th>Cost [€]</th>
</tr>
</thead>
<tbody>
<tr>
<td>cradle assy (rigid)</td>
<td>14,0</td>
<td>66,5</td>
</tr>
<tr>
<td>lower wishbone assy</td>
<td>9,0</td>
<td>23,0</td>
</tr>
<tr>
<td>toe-link assy</td>
<td>2,7</td>
<td>13,0</td>
</tr>
<tr>
<td>knuckle (steel)</td>
<td>7,6</td>
<td>20,8</td>
</tr>
<tr>
<td>shock absorber assy</td>
<td>7,5</td>
<td>37,0</td>
</tr>
<tr>
<td>spring</td>
<td>5,0</td>
<td>15,2</td>
</tr>
<tr>
<td>stabiliser assy</td>
<td>3,5</td>
<td>11,4</td>
</tr>
<tr>
<td>fasteners</td>
<td>2,0</td>
<td>8,6</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>51</strong></td>
<td><strong>195</strong></td>
</tr>
</tbody>
</table>

Table 3 - Mass & Cost Estimation

6 Conclusions

The task for this thesis work was to develop a low-complexity rear suspension toward premium level targets for ride quality, handling performance and road noise isolation combined with a high level of content (including all-wheel-drive and active systems). From this work, the conclusion can be drawn that many of these imperatives can be met at a much reduced cost and weight compared to current solutions, and with the reduced complexity, lower running costs in terms of aftermarket and quality issues.

The main challenges which still remain to investigate are how the pre-load on the damper in the concept is acceptable in terms of its affect on ride comfort and road noise isolation. Additionally, the main path for road noise transfer to the passenger compartment will likely be the damper to body attachment. This will lead to high demands on the body attachment dynamic stiffness, which is another issue to evaluate.

The recommendation is therefore to continue to develop the suspension further to the degree that hardware prototypes can be built, see 4.6. These prototypes should be used to evaluate the concept and serve as demonstrators for management review and a decision aid for implementation in future platform projects.
References


A Original Suspension Geometry
Version 0.05 created by Tobias Brandin 031208
B MATLAB code for moving inner wishbone points

The following MATLAB program was used during the development of the suspension geometry. In order to not change the kinematics while moving the points forward and backward they could be kept on the same line.

% lca.m
% 2004 (C) Matthijs Klomp, All Rights Reserved

function [newp1,newp2] = lca(d1,d2);

% Function which allows to move the front and rear lower wishbone points in the direction of the two original points

% Input d1 and d2 are distance from p1 and p2
% Two points on the line
p1=[4077 -395 200];
p2=[4432 -240 158];

% Creating a unit-vector between the two points
v12=(p2-p1)/norm(p1-p2);

% Creating new points in the direction of the vector between the two original points
newp1=p1+v12*d1;
newp2=p2+v12*d2;

disp('New P1 is:')
disp(newp1)
disp('New P2 is:')
disp(newp2)
Hello Matthijs Klomp,

I have calculated two different solutions using the data you sent Thursday last week.

Maybe at first one point regarding the data, I think the ratio of the wheel to spring has to be 1.25 instead of 0.80.

Because E2 Opel has also a ratio of 1.25 wheel travel to spring travel.

With this, the results are as follows:

Using an outertube diameter 63 mm:
- Pistonrod diameter 22 mm
- Length outerhousing 340 mm
- Nivomat compressed length 374 mm
- Nivomat extended length 557 mm

Using an outertube diameter 68 mm:
- Pistonrod diameter 22 mm
- Length outerhousing 278 mm
- Nivomat compressed length 312 mm
- Nivomat extended length 494 mm

Kind Regards,

Gerhard Boll
Project Manager

ZF SACHS AG
Nivomat Product Division
D-53783 Eitorf, Bogestrasse 50
phone: 0049 2243 12486, fax: 0049 2243 12480

gerhard.boll@sachs.de
D MATLAB code for calculating the perpendicular distance from p7 to the line p1-p2

```
% normdist_from_p7.m
% 2004 (C) Matthijs Klomp, All Rights Reserved

% Program which calculates perpendicular distance
% between three points. The points are p1, p2 and p7

% |AxB|=|A|*|B|*sin(alpha) and |B|*sin(alpha) is the height
% of the triangle.

% Version 0.17 has the following points:

p1_017=[4077.0 -395.0 200.0]; % lca_front
p2_017=[4453.0 -231.0 148.0]; % lca_rear
p7_017=[4448.0 -742.0 132.0]; % lca_outer

% Version 0.21 has the following points:

p1_021=[4167.0 -425.0 184.0];
p2_021=[4461.0 -296.0 144.0];
p7_021=[4458.0 -750.0 132.0];

h_017=norm(cross((p1_017-p2_017),
                 (p2_017-p7_017)))/norm(p1_017-p2_017);

h_021=norm(cross((p1_021-p2_021),
                 (p2_021-p7_021)))/norm(p1_021-p2_021);

disp("The perpendicular distance from p7 to p1-p2 is...")
disp("... for version 0.17:")
disp(h_017)

disp("... for version 0.21:")
disp(h_021)
```

\[
A \times B = C \\
|C| = |A| \cdot |B| \cdot \sin(\alpha) \\
h = |A \times B| / |A| \\
h = |B| \cdot \sin(\alpha)
\]