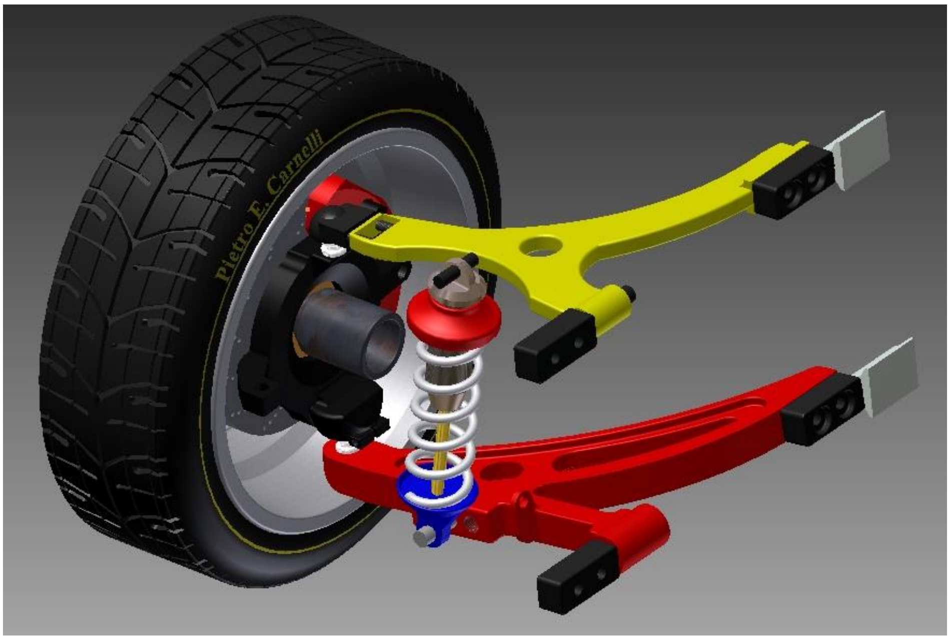
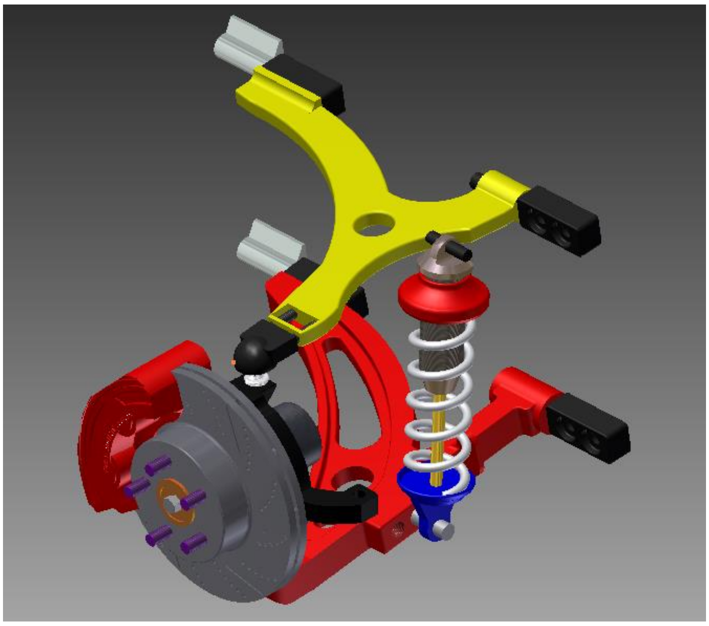
Coursework II
Suspension Assembly





Pietro E. Carnelli

1. *CAD MODELLING*

1.1 Introduction

The double wishbone suspension assembly presented in this report is designed for a high performance sport car, namely the second generation Subaru Impreza WRX STi.

Suspension systems in cars arose from dealing with rough roads. Measurements published in 1936 suggest that on an average road in the UK there were about 236 ‘bumps’ (irregularities in the surface of around 0.025m) over a length of a mile[1]. Not only did these ‘bumps’ create noise as cars with virtually non-existent suspension systems drove over them, but they also drastically reduced the working life of the car components as they were subjected to constant vibrations (and therefore repeated cyclic loading). It is worth noting that a well adjusted and designed suspension system greatly increases the life span of components by damping the vibrations, reduces the noise made from the transmission, improves ride comfort for the passengers and crucially improves cornering speeds.

The main advantages of the double wishbone suspension system are its ease of adjustability and computer modelling. Adjustability allows the driver to fine tune the suspension to the driving or race environment, this is very crucial when a hundred milliseconds can make the difference from starting first and last on the grid. The relative ease of modelling the system allows for better understanding of the loads and therefore allows a more efficient design, often achieved by reducing the sprung and un-sprung weight of the system. However due to the greater number of parts and complexity of each, high costs are normally an associated disadvantage. Double wishbone suspensions are normally not used on everyday cars. Due to their high cost and set up complexity, especially when there is a drive axle. Manufactures tend to revert to the less complex simpler and in some cases a more ‘compact’ Macpherson strut suspension design. Nonetheless a double wishbone has greater adjustability and load bearing abilities.

1.2 Details of Suspension Assembly

Figure 1.2.1 below shows the major components of the suspension assembly. Non-realistic colours were chosen for the suspension components in order to make it simpler to identify parts. The whole assembly pivots about four separate hinge pins (black) and located at the rear extremities of the two wishbones. This allows the end user to mount them at different angles to the car main frame (chassis) in order to achieve desired kick up, bump steer and caster settings. The ride height and droop settings are also adjustable by moving the shock spring height adjuster up and down the thread on the shock absorber body. Camber is adjustable via changing the length of tie rods at the end of the upper wishbone. The shock absorber inclination is adjusted via the three mounting holes provided on the lower wishbone.

Table 1.2.1 Shows current suspension settings used in the CAD model.

Setting Type	Value
Toe	0.5 degrees toe in
Ride height	685mm (to the top of the wheel arch)
Camber	-2 degrees
Castor	2 degrees

[1] Bastow. D. (1987) *Car Suspension and Handling*, 2nd edition, Pentech Press London.

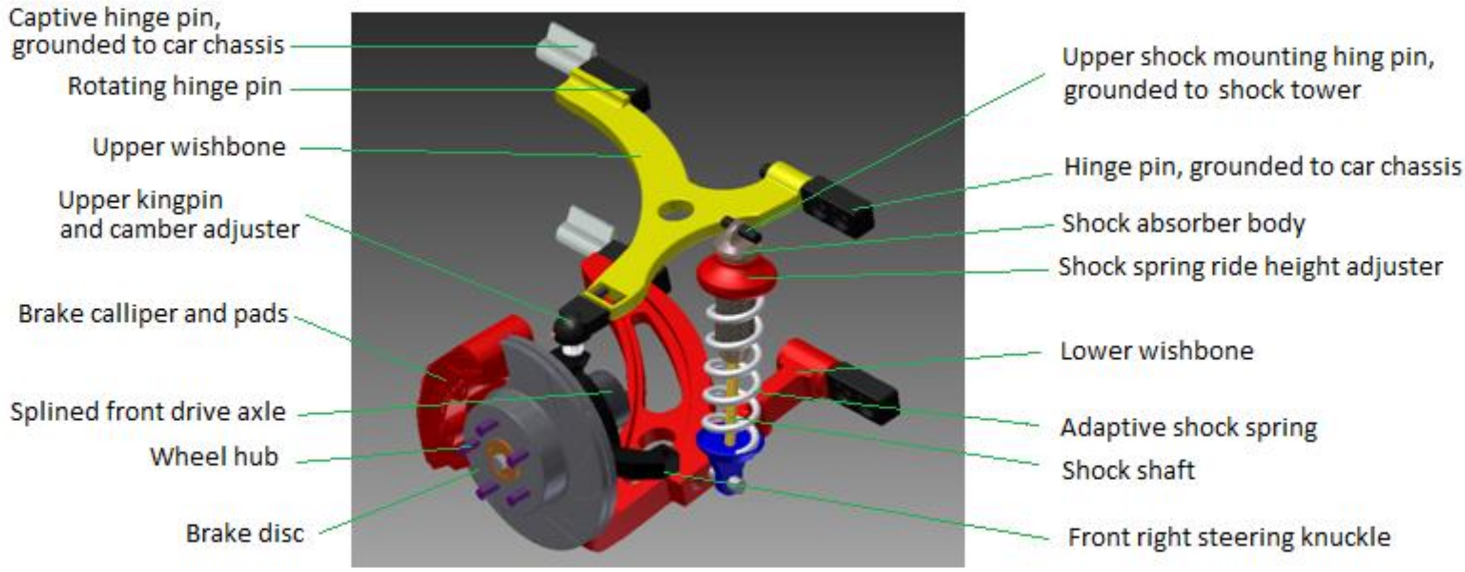


Figure 1.2.1 CAD image of complete suspension assembly without wheel and tyre.

Figure 1.2.2 below shows details of the transimssion system to the front wheel. A splined drive shaft inserts itself into an inner splined rotating hub (purple) which itslef mounts the wheel and vented brake disc (gun metal, and all held by the wheel nuts). This goes through two tapered roller bearings (blue) which are held tight inside the steering knuckle (black) by the spindle end nut (located behind the wheel dust cap).

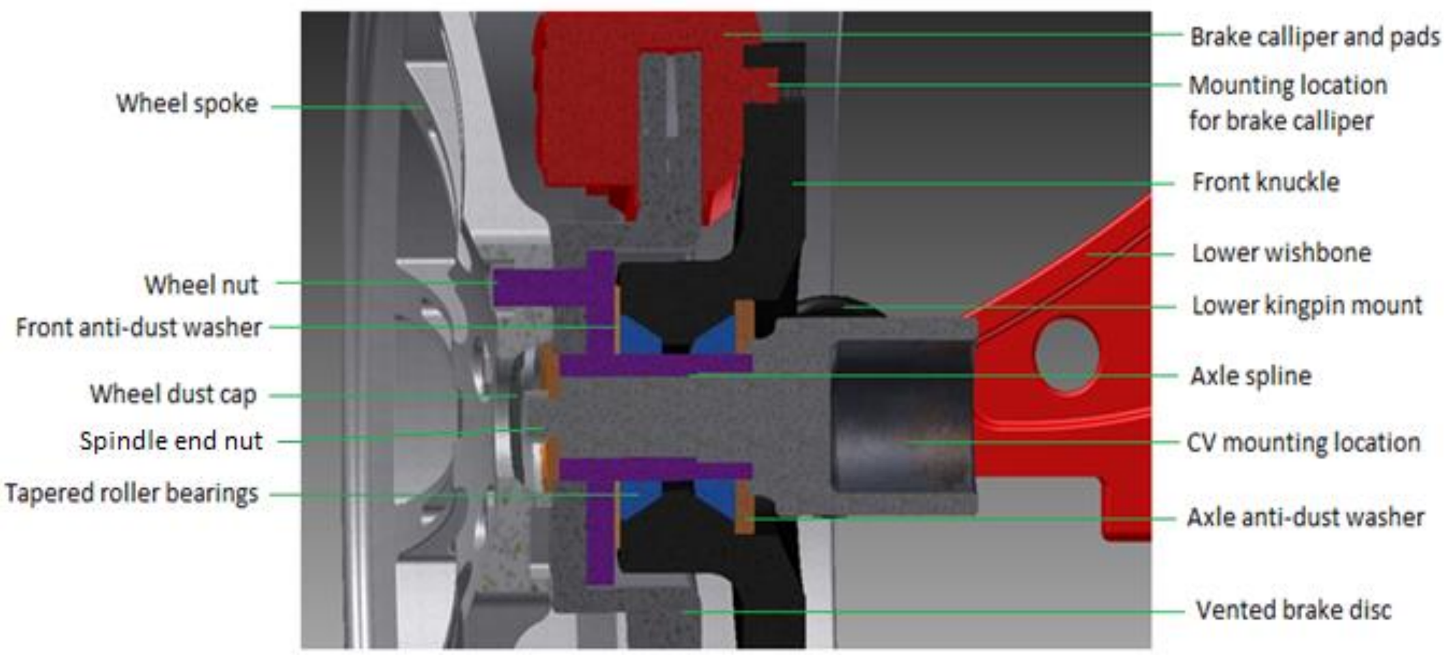


Figure 1.2.2 CAD image showing half section of transmission system.

More components are detailed below, such as the vented disc brake (figure 1.2.3) and the steering knuckle (figure 1.2.4). The disc brake design has essentially two separated faces (by about 6 mm) with a 'vane' running in between. This vane was designed in order to aid cooling of the disc brake by displacing hot air as the entire drive system rotates. A pattern of six through holes repeated twenty times also aids with cooling, lightening and gives more bite (more friction) as they pass through the pads. Series of grooves were also cut into the brake disc as this again helps with increasing friction between the disc and the brake pads and cooling, allowing hot gases to escape between the rotor and the brake pad.

The steering knuckle or hub carrier is also shown below. Attention has been taken to make sure there is a tight fit with the tapered roller bearing and that the kingpin angles are correct (2 degrees). Two ball joints mount the knuckle to the upper and lower wish bones whilst allowing them to rotate. It also crucially allows for uninterrupted travel across the suspension travel and steering sweep.

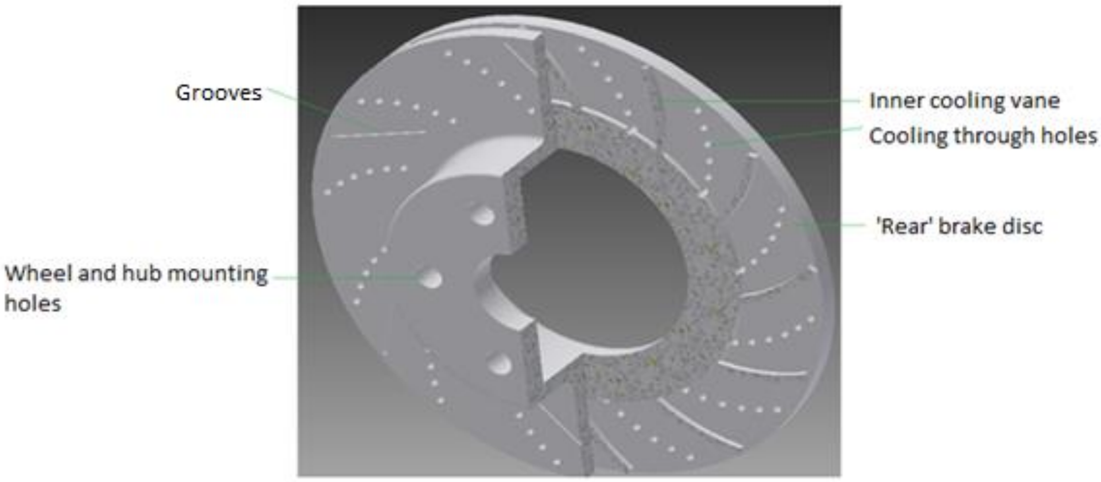


Figure 1.2.3 CAD image showing section of vented disc brake

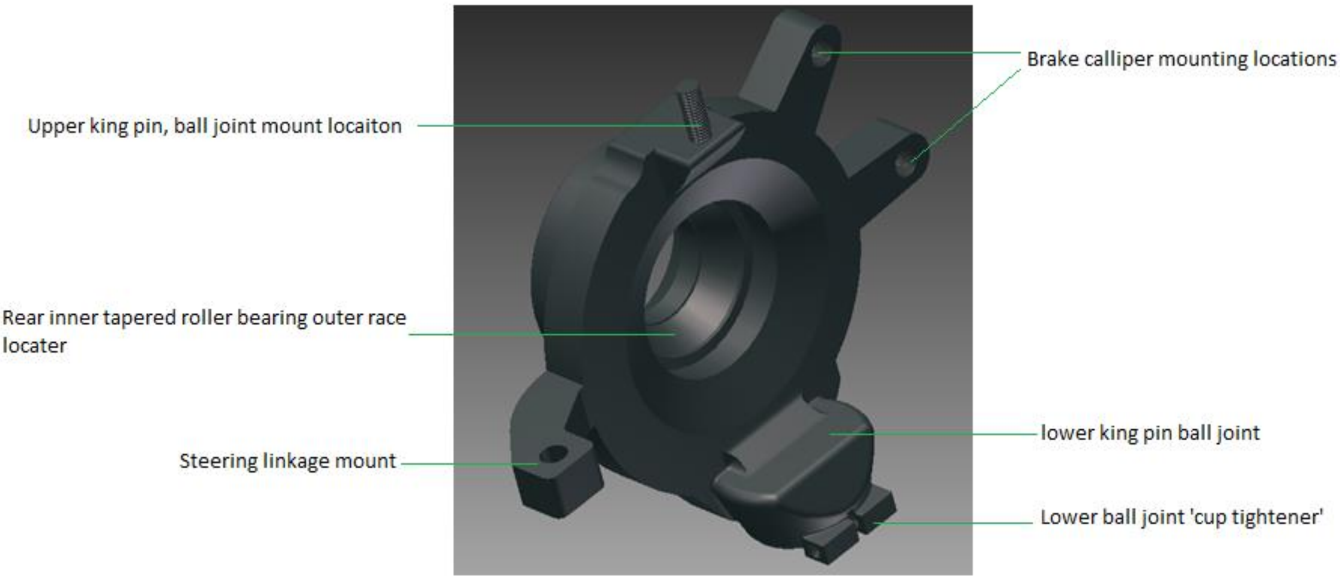


Figure 1.2.4 CAD image showing front right steering knuckle (hub carrier).

1.3 Details of Suspension Assembly Animation

The entire suspension and drive system was animated in order to show it rotating, steering and absorbing bumps in the road. Figure 1.3.1 below shows two images of the suspension system at its extremities, i.e. the wishbones moving 20 degrees above and below the normal with the wheels turning through ±40 degrees (from straight line normal). None of the components interfere with each other throughout the entire shock travel and steering sweep.

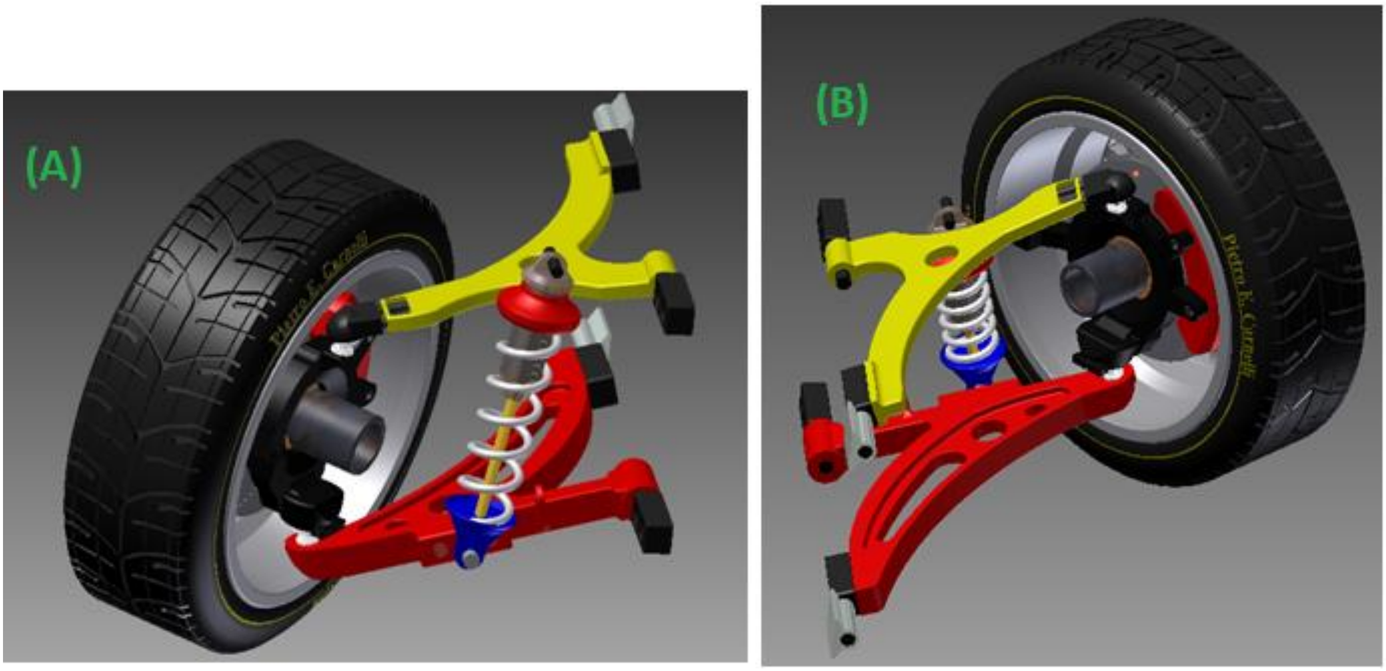


Figure 1.3.1. CAD images showing suspension animation at lowest point with right steering lock (A) and suspension at highest point with left steering lock (B).

1.4 CAD Drawing of Lower Wishbone

A fully dimensioned drawing of the lower wishbone is included with this report. It is important to note that it is divided over **two pages** as unfortunately there was no method of fitting all onto one sheet. Dimensions are in mm and are given to one decimal place. It is **not** as result necessary to manufacture to this very precise tolerance however it is given should computer modelling or load calculations be made from the drawing.

2. RESEARCH INTO CAR STEERING, SUSPENSION AND TRANSMISSION

2.1 Understeer and Oversteer

Understeer and oversteer are based on the differences in steady-state conditions where the vehicle is following a constant radius path at a constant speed with a constant steering wheel angle, on a flat and level surface. If more steering is needed at the higher speed to maintain the same turn radius, then the vehicle is said to understeer. If less steering is needed at the higher speed, then the vehicle is said to oversteer. Understeer and oversteer are defined by an understeer gradient U that is the difference between a reference steer angle gradient and the Ackerman steer angle gradient.[2]
A less rigorous way of defining understeer for example is by measuring the number of degrees more the steering wheel has to be turned per G of lateral acceleration. A car with a higher ratio will require a greater

[2] In standard terminology defined by the Society of Automotive Engineers (SAE) J670 and the International Organization for Standardization (ISO) 8855.

degree of rotation of the steering wheel to get a given change in front wheel angle (hence the more the car understeers). Typical car values are around 14 to 15:1. For the Subaru Impreza WRX STi its 37 degrees per G. It is more common for typical road cars to understeer as it is more controllable than oversteer, which in extreme cases can spin the car around. Typical causes for understeer and oversteer are listed below. It is important to note that there are different types of understeer and oversteer, especially with regard to on power and off power settings. Often, the easiest way to avoid understeer is to slow down, as this transfers the weight of the vehicle back over the front wheels, increasing traction and as a result allows them to steer the car.

- Limiting friction. This apart from set up is the main cause of understeer. This is determined as with most cases by the type of tyre compound and thread pattern used. Tyre compound, thread and pressure are the three most important characteristics of any vehicle set up. A poor choice could decrease the friction with the road surface and therefore increase understeer as the tyres struggle for grip both on and off power.

- Front toe. Toe in decreases the steering response and therefore increases understeeer entering and in the middle of the turn. Toe out increases steering response and straight line stability.

- Anti-roll bars. Thinner front anti-roll bars increases traction off power, but increases understeer. Thinner rear anti-roll bars increase rear traction and increase on power understeer.

2.2 Steering Drift

Steering drift is when a vehicle has a tendency to deviate whilst moving from a straight line direction. This is often simply caused by a minor misalignment of the front wheels. It is very rare for this to occur due to minor misalignment of the rear wheels. This can be due to design or manufacture error. More commonly it is caused simply by bad vehicle maintenance and wear of critical parts or incorrect alignment of steering tube and linkages. Not to be confused with torque steer.

Through continued use of parts, such as ball joints used in steering linkages, kingpins and even suspension wishbones can become worn and therefore loose. This allows one wheel to turn at a different angle (ie not the designed Ackerman angle) relative to the opposite wheel. This means that the friction forces will be different to each wheel and therefore will start to turn the vehicle even if it is being driven along a straight path with no change in the driver inputs.

This difference in the friction forces between the two front wheels can also be attributed to other types of misalignments, such as ride height, toe and camber. Ride height changes can sometimes occur due to different pressures inside each tyre. This means that there is an unequal weight distribution between the tyres and therefore changes the contact areas each tyre has with the road, therefore the friction forces. Similarly this can happen with unequal camber settings between the two front wheels.

2.3 Sprung and Un-sprung Weight

pc0179@bris.ac.uk

Sprung weight is the total weight supported by the shock spring in the suspension system. This includes the weight of the chassis, the engine, passengers, luggage and transmission.... The un-sprung weight is the total weight of the components which are not supported by the shock spring. This includes the weight of the wheel, tyre and some of the lower components of the suspension and transmission system (depending on design).
It is generally desirable to have the lowest sprung and un-sprung weight possible. This is in part to aid the vehicles acceleration as well as reducing the amount of energy required to drive it, lowering fuel costs. Reducing the weight reduces the inertial effects of the un-sprung mass as it travels through the suspension's sweep, therefore allowing the suspension to move faster and follow the uneven surface of the road.

Automotive suspension designers need to consider the natural frequency of the un-sprung mass. The un-sprung mass will effectively oscillate as the wheel rolls over bumps and irregularities in the road surface (forced vibration). Ultimately setting up a mass-damper forced vibration system. Reducing too much the weight can reduce the un-sprung mass's strength and could potentially bring its natural frequency closer with that of the forced vibration, creating unwanted resonant responses which could ultimately damage the suspension system.

2.4 Toe and Camber Angles and King Pin Inclination

Toe angle is measured from the central axis of the wheel when straight outwards to the true orientation of the wheel, as shown below in figure 2.4.1. When travelling in a straight line it is preferable to have straight pointing wheels, however during cornering it is not as beneficial. As mentioned earlier in the understeer and oversteer section, front toe in (negative) decreases steering response entering and in the middle of the turn. More front toe out (positive) increases the steering response when entering the turn. These responses are due to the angles the tyres make with the road surface and the turning radius, this increases or decreases the rotating friction force experienced (reduced when wheels are in line with direction of travel). Rear toe in increases the forward traction and stability on power turning, as the weight transfers away from the corner the outside rear wheel will have a greater proportion of the weight. Therefore it is advantageous to have it pointing (toe in) slightly in the direction of the exit of turn.

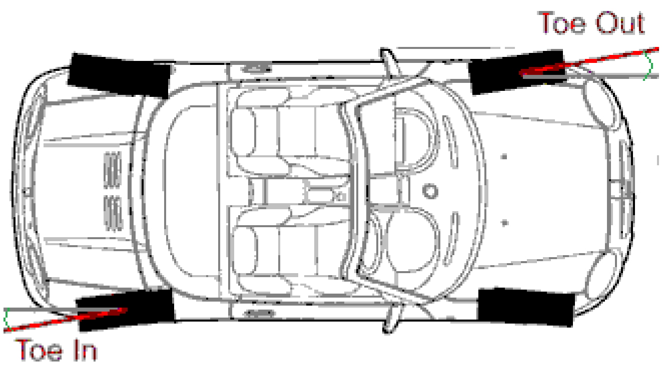


Figure 2.4.1 Image showing measurement of toe angle.[3]

[3] Adapted from image: www.mgf.ultimatemg.com

Camber angle is measured from the vertical axis perpendicular to the road to the vertical axis of the wheel, this is shown below in figure 2.4.2.

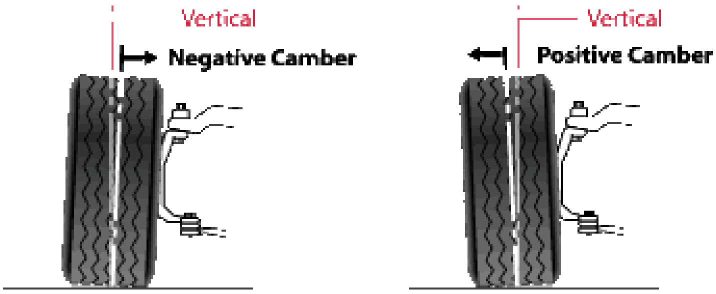


Figure 2.4.2 Image showing measurement of camber angle[4].

Throughout the suspension travel the camber angle on a double wishbone suspension often changes due to the mounting locations of the wishbones relative to each other. Camber gain is defined as the total camber change through the suspension travel, camber gain is good for when driving on very narrow and twisty tracks where there are many direction changes, however it is generally avoided. In general vehicles have negative camber on all four wheels. Negative camber provides a greater steering response as the tyre leans away from the corner (due to lateral acceleration) more of the tyre will effectively be in contact with road surface, providing greater driving traction. However there are limits to this increase in turning gain as the contact area when the vehicle is driving in a straight line is reduced making it unstable and unsafe. Too much camber also increases localised tyre wear making the vehicle unsafe under braking conditions.

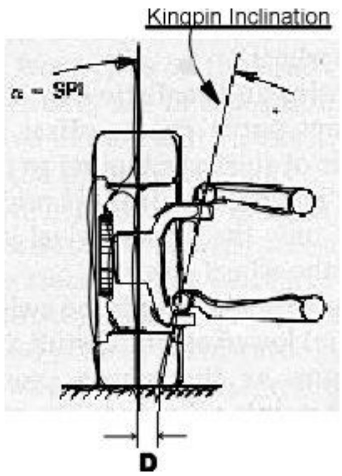


Figure 2.4.3 Image showing measurement of king pin inclination and scrub radius (D)[5]

The king pin inclination angle is measured between the vertical plane of the wheel and the two ball joints or king pins which link the front hub or knuckle to the wishbones as demonstrated above in figure 2.4.3. The weight of the vehicle tends to turn the king pin to the straight ahead position where the wheel is at its highest point relative to the chassis. The king pin inclination also sets the scrub radius, defined as the offset between the tyre's contact point with the road and the projected plane of the king pin inclination. Zero scrub radius (where D equals zero in figure 2.4.3) isolates the steering components from bumps in the road surface. A

[4] Adapted from image: www.blackboots.co.uk
[5] Source of image: www.autozine.org

larger scrub radius tends to reduce the amount of input force required to turn the wheel. However subjects the steering components to more of the forces from driving over bumps in the road.

2.5 Anti-Roll Bars

Anti-roll bars are components designed to reduce the 'roll' of a vehicle as it corners. As a vehicle turns the lateral acceleration forces the chassis of the car to tilt to the outside of the corner. This has the effect of reducing the contact area of the inside tyres, reducing grip and speed.

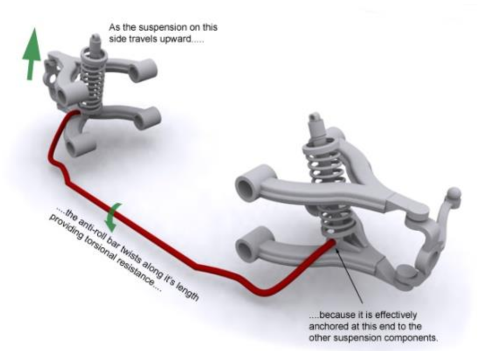


Figure 2.5.1 Image of a anti-roll bar on the front suspension[6]

Anti-roll bars are mounted to the chassis body and the ends to the lower or upper wishbone as shown above in figure 2.5.1. This partially or wholly (depending on the stiffness/thickness) couples the suspensions together. The result is that it reduces the amount the outer suspensions compression and helps maintain the inner set of wheels firmly on the road.

Too stiff anti roll bars may increase corner speed on very high traction smooth road surfaces however they seriously reduce how independent the suspension system of each wheel is. This reduces right comfort and stability as each bump is felt throughout the car rather than being absorbed by the shock absorber. Too thin anti-roll bars do not reduce the amount of roll of the vehicle as it allows the suspension to act too independently and therefore reduce its effectiveness at preventing chassis roll.

2.6 Constant Velocity Drive Shaft (CVD)

A constant velocity drive shaft allows torque (or power) to be transmitted through a variable angle at a constant speed, without a significant increase in friction or 'play'. There are several designs for a CVD however one of the most common is the Rzeppa joint (see figure 2.6.1) designed by engineer Alfred H. Rzeppa whilst he was working at the Ford Motor Company. It commonly uses six or eight spheres with an inner and outer race to provide essentially constant velocity torque transfer.

[6] Source of image: www.automation-drive.com/anti-roll-bar

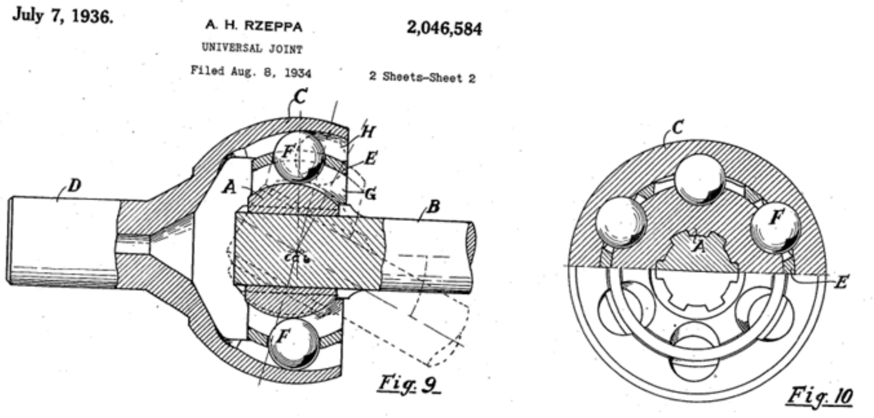


Figure 2.6.1 Image of the original US Patent (2,046,584) of the Rzeppa Universal Joint[7]

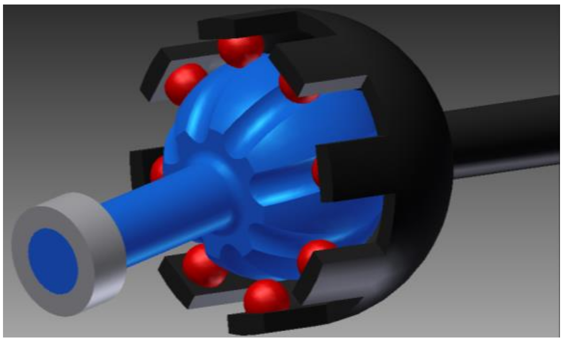


Figure 2.6.2 CAD image of a Rzeppa Universal Joint

Clearly from figure 2.6.2 as the outer race (black) rotates the spheres (red) rotate with it as the outer race has deep grooves. This in turn forces the inner race (light blue) to also rotate as it too has deep grooves preventing the spheres from moving laterally. The spheres allow the shafts to be angled as long as they remain within the grooves of each of the races. Typically a CVD can transmit power efficiently and reliably up to 45-48 degrees of articulation.

For more rugged applications such as steering columns and off road vehicles double Cardan joints are often used. double Cardan joints as shown below in figure 2.6.3 have a floating centre element to maintain equal angles between the driven and driving shafts. They are rarely used in high speed applications as the joint tends to generate additional vibrations and adds rotational mass to the drive train.

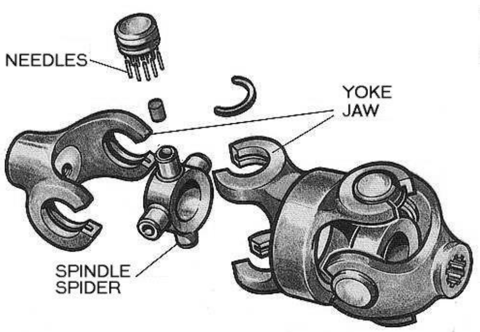


Figure 2.6.3 Image of a double Cardan universal joint.[8]

[7] Source of image: www.google.com/patents

pc0179@bris.ac.uk

3. Conclusions

Utmost care and attention was done in order to design an animated, working and realistic double wishbone suspension on CAD in this report. In conclusion to the research section, the future of suspension lies in designing active systems. It is noteworthy to remember that the most crucial aspect of any car set up is the choice of tyre compound and thread pattern. Over 90% of a car's set up and ability to handle well is determined by those two factors. So far we have only in this report mentioned and discussed passive systems where the shock absorber takes all the energy out of the bump and does not change its setting from shock to shock. Active damping is a relatively recent idea and was entirely theoretical until at least circa 1985 when Lotus started doing research and eventually designing a basic working system for its formula one car.

Active damping has neither inherent spring rate or damping force. The spring and damper are replaced by an actuator whose purpose is to reduce to a minimum chassis movements. This system proved to be quite successful on smooth level tracks where the added 45 kilos didn't prove much of a penalty as the active damping was such an improvement in increasing cornering speeds and vehicle stability. These very complicated an expensive systems are sometimes seen today on very high end cars (such as the Range Rover Evoque). It is therefore much more common to have a semi-active suspension system as it provides an ideal compromise between price and performance.

4. Bibliography

- Bastow. D. (1987) *Car Suspension and Handling,* 2nd edition, Pentech Press London.
- Giles. T. (2005) *Automotive Chassis; Brakes, Steering and Suspension,* 2nd edition, Thomson Delmar Learning
- Genta. G, Morello. L. (2009) *The Automotive Chassis,* 1st edition, Springer
- Heisler. H. (2002) *Advanced vehicle technology,* 2nd edition, Butterworth-Heinemann Publications

[8] Image source: www.motorera.com