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Abstract

This report is a culmination of investigative efforts regarding the possible conversion of a mono-shock suspension system to a dual-shock to achieve the objective of gaining cornering performance for a 1997 formula Renault single-seat race car. Theories have been derived from the relevant research conducted, along with details of the implemented methodology and the data obtained has then been used in conjunction with data from previous observations of this car to show the effects of factors regarding rising/falling rate motion ratio, roll centre migration and effects these have on the handling balance. Mathematical principles was also used in order to help determine these factors. From here a comparison of rising and falling rate motion ratio was derived from which conclusions and recommendations were created for future work should this project be taken further.

1. Introduction

1.1 Generic Introduction

In terms of vehicle performance (especially within motorsport) it is imperative that the suspension design is suitable for that vehicle's application. In order to derive a dual-shock suspension system onto a chassis originally designed for a mono-shock to see whether performance gains can be made with regards to vehicle dynamics, great care must be taken as the slightest mistake may alter suspension characteristics for the worse. But before progress can be made and aims & objectives are derived with background research of both systems will prove beneficial to help get a greater depth of understanding of how both these systems function in select situations. Another possibility to consider is the potential of using alternate designs of the original setup and whether they can be replicated for the given case study.

Suspension Design is no easy task and so it is essential to consider various possibilities with regards to performance & outlay amongst other factors of suspension design. By looking at the methods used by suspension design engineers in industry is key in order to derive a suitable system for its intended application. Using a "think big, think small" approach, (Thorby, 2008) implies that by having sub objectives is the common approach needed in order to derive a suspension system. These objectives are:

- Packaging- Looking where it sits on the chassis but more importantly whether it meets rules/regulations.
- Motion Ratio- altering motion ratio fundamentally affects how the vehicle handles.
- CG height- centre of gravity height on the chassis and how this alters.
- Geometry- suspension geometry i.e. wishbone mounting points.
- Cost – cost must be relevant given its application.

Suspension Design has come a long way due to the ever decreasing cost of computers and geometry software needed which can be operated at all levels of motorsport from clubman to F1. In particular the clubman racer may use this method as it may prove advantageous as complex suspension behaviour principles both mathematical & mechanical are calculated by the software and not the person, which saves a lot of time and effort to previous methods of using string computers like the one illustrated in (Staniforth, 2006).

"When we talk about suspension geometry it means the broad subject of how the unsprung mass of a vehicle is connected to the sprung mass. These connections not only dictate the path of relative motion, they also control the forces that are transmitted between them" (Milliken & Milliken, 1995, p 607)

The purpose of this project is to investigate the possibility of changing the Mono-shock suspension system to a dual-shock suspension system for a single seat racing car. The case study of this project will be a 1997 formula Renault RC97 made by Tatuus currently competing in the Monoposto racing club classics 2000 class.

1.2 Case Study



Figure 1: Picture of the 1997 Renault RC97 racing at Spa-Francorchamps, Belgium.

The initial approach to this investigation is to primarily use mathematical formulae in conjunction with kinematic software to determine handling behaviour changes caused by iterations on the bell crank this helps towards achieving the second aim. On a structural basis a cad model of the proposed design shall be constructed from which hard points will be measured and then used for the kinematic model. This Approach shall be used in conjunction with the overall aims which are found in the next chapter.

These aims have been derived due to the fact that mono-shock systems although seem fairly simple in theory. In reality however they seem to be fairly complex in practice, (Rowley, 2003) implies that there are many people in industry who claim to know how they work, but very few who actually know how they work. As a result it is deemed beneficial to convert to dual-shock due to factors which are discussed in Chapter two of this report.

These two aims will be achieved through the completion sub-objectives prioritising relevant suspension factors in the same approach as (Thorby, 2008). This project is critical to industry as it may offer the opportunity to obtain more suspension data via the research of mounting a Dual – shock system to a mono-shock chassis which may prove advantageous to the operator of a given vehicle.

1.3 Aims & Objectives

Due to the lack of data available of this vehicle, it would be beneficial to derive a more adjustable suspension system. In the case of the RC97 the rear suspension system is a push-rod actuated suspension system which uses a specifically designed rocker often referred to as a bell crank to transmit the vertical movement of the wheel through an angle (commonly found at 90 degrees) which then acts upon the damper unit. The geometry of bell cranks are considered to be a major influence of motion ratio which is the relationship between wheel and spring displacement a key factor to handling balance. (More detail of this shall be covered further on in this report)

The aims of this project are to:

- Derive a dual-shock system for the RC97 using relevant methodology (solidworks & wingeo 3).
- Assess and predict factors in order to determine the effect of rising/falling rate motion ratio has on the cornering performance of the vehicle.

To achieve the aims previously stated, a series of sub-objectives regarding factors relevant to suspension design. As mentioned by (Thorby, 2008) it is beneficial to prioritise factors regarding suspension design in order to identify the conflicts between them. To summarise the purpose of this project is to whether a conversion to dual-shock would be advantageous in gaining cornering performance.

2. Background Research

2.1 Research Strategy

Before research can be conducted, a suitable research strategy plan must be formulated to ensure that research is both time-efficient and effective in order to develop theories relevant to the project aims. Due to the nature of this project research will begin with quantitative research of secondary research sources relevant to the project to first give an overview of how the suspension systems operate including key components and the benefits these components possess, and from this theories have been derived to predict characteristics of the dual-shock conversion.

According to (www.surrey.ac.uk, 2016) quantitative research gives the advantage to test theories or hypothesis which is important as the results taken from this report should reflect what the literature is saying.



Figure 2: Picture of Research strategy implemented for this project (www.edresearch.nmsu.edu, 2016)

Figure 2 shows a diagram of the research method used. To start with initial planning is conducted through quantitative research from which theories have been derived. This strategy has been the main outlay for this project.

2.2 Mono-shock Suspension

When assessing mono-shock systems it became clear that in terms of adjustment and setup they're not as compliant due to one damper effectively controlling two wheels simultaneously but a major dis-advantage to mono-shock is that there is no damping in roll. Both wheels are connected so they both move in bump and rebound. The force acting on the push rods acts upon the rotating link which then transmits through the damper and the other wheel. In a cornering situation force is applied to the outside wheel which raises the inside wheel, in some cases the inside wheel lifts off the floor completely resulting in a loss in cornering grip a characteristic which is not desired (as demonstrated in figure 3). Roll is governed by Bellville washers which are fitted on the rotating link, effectively the amount of washers applied will determine the amount of roll permissible by that setup. (Staniforth, 2006) suggests that Mono-shock suspension systems make use of the tyre spring rate which compensates for a stiffer spring setup. Simply the tyre deflects more than the coil spring. (In industry these systems are described mono/dual shock but the correct term is mono/dual damper. For the reason of continuity the descriptive noun of both systems shall be mono/dual shock to save the reader any confusion)



Figure 3: Picture showing a F1 car cornering with inside wheel lifted off the ground. (Dilworth, 2011)

Given the case of the RC97 which is a small single seat racing mono-shock offers good damping characteristics at the front where packaging is limited.



Figure 4: Picture showing mono-shock layout with centrally mounted damper. (www.tezzaworld.com/, 2015)

Figure 4 shows a common mono-shock design the bottom of the damper is mounted on top of a stack of bellville washers which govern roll. These bellville washers squish as roll is applied (the same principle as a clutch diaphragm) and pre-load can be adjusted manually via two bolts at either end of the shaft. (Staniforth, 2006) explains that race-cars with an originally stiff front/rear setup became a

stiff front/soft rear setup due to the increasing influence of aerodynamics, this alters the rake angle at the rear which will be affected by a soft suspension setup as speed is applied. Due to this factor many small junior formulas such as formula Renault and formula ford use a mono-shock system at the front because of this.

Visually speaking mono-shock setups are fairly simple given their use of a single damper in conjunction with equal length pushrods acting about a central point as previously explained, which makes them ideal for single seat race cars due to their compatibility and relatively low weight due to the use of fewer parts which affects factors such as centre of gravity & overall weight. According to (Rowley, 2003) Mono-shock suspension require a higher spring rate and damping rate in comparison to their dual-shock counterparts, a common reason why they're much stiffer in comparison to dual shock arrangements. (Rowley, 2003) also goes onto say that Mono-shock systems effectively separate bump from roll which in theory makes it fairly easy to mathematically calculate effects in either bump or roll but difficult to model both factors simultaneously. This theory is also stated by (Braun, 1999) who informs that mono-shock separate & bump roll better than a dual-shock setup with the use of bell cranks.

A key point to add there are mono-shock designs which incorporate anti-roll methods alternative to what has been looked at which use an extra rocker that rotates centrifugally above the base of the damper, this rocker has two links connected to both push rods and as force is applied at one the rocker rotates pulling the push rod of the opposite wheel upwards in order to resist roll. According to (Van Berkum, 2006) this design is limited due to the amount of space available with regards to single seat racing cars. Given the case of the RC97 this setup would be particularly difficult to derive due to this along with the fact this system would prove difficult to model on Wingeo 3 a key factor of aim 1 & aim 2.

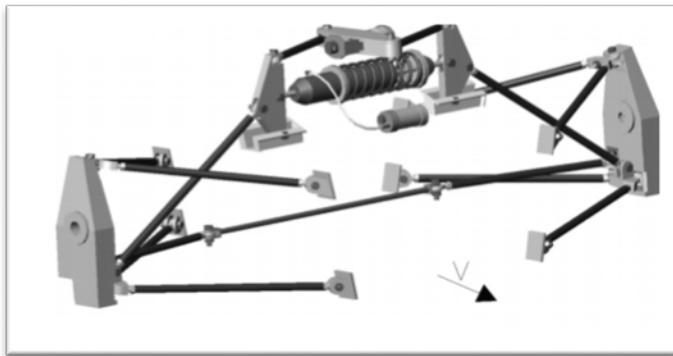


Figure 5: Diagram showing cad model of rotating link anti roll suspension of a formula student car (Van Berkum, 2006)

2.2 Mono-Shock Bellville Washer Shuttle

The Bellville washer shuttle (commonly referred to as a shuttle) is the component that determines the permissible amount of roll desired via the compression of numerous dished washers which compress and release as lateral forces are applied via the push rods.

With regards to the RC97 previous experiments regarding the mono-shock shuttle has been conducted. Testing factors like washer stiffness for example has been conducted, and according to the results obtained by (Smith, Stanford & Cutlan, 2015) the washer stiffness used on the RC97 is approximately 11208 N/mm.



Figure 6: Picture of a Bellville washer being tested on a spring compression tester (Smith, Stanford & Cutlan, 2015)

Another factor to consider is the variation or order the washers are installed. Although as implied by (Rowley, 2003) that greater roll stiffness can be obtained by using more bellville washers, even greater roll stiffness characteristics can be achieved by altering the configuration of washers within the shuttle assembly as shown in figure 7.

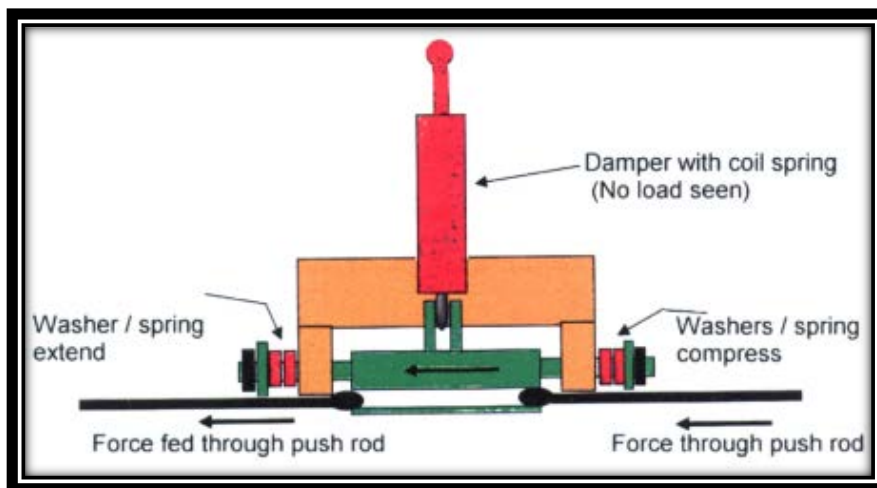


Figure 7: Diagram illustrating function of Belleville washer shuttle (Pugh, 2008)

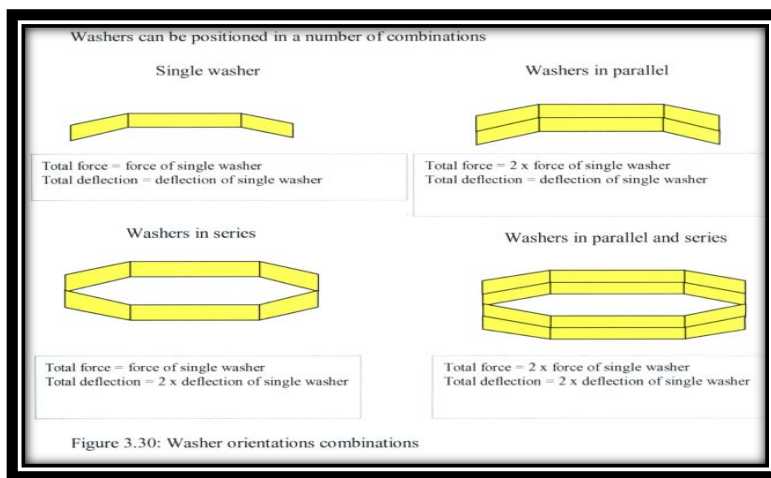


Figure 8: Diagram showing various washer configurations (Pugh, 2008)

Figure 8 shows the basic configurations for belleville washer shuttle in order to gain greater roll stiffness characteristics. Ultimately shuttle stiffness as shown by (Pugh, 2008) is calculated by:

$$\text{Shuttle Stiffness} = \frac{\text{Individual washer stiffness} * \text{sets of washers in parallel} * \text{friction factor}}{\text{sets of washers in series}}$$

Using this principle shuttle stiffness were calculated for the Rc97. According to (Smith, Stanford & Cutlan, 2015) depending on configuration significant variations in stiffness figures can be achieved ranging from 1245 n/mm and in excess of 28000 n/mm as shown in figure 11.

Outer Diameter 31.5mm	Inner Diameter 16.3mm	Thickness 2mm	Conical Height 0.5mm		
				Single washer N/mm	11208.11749
	Stack Orientation	Stack Constant	Effective Spring Constant N/mm	Effective Deflection mm	Stack Length mm
<<<<<<<	1-1-1-1-1-1-1-1-1-1	0.1111111111	1245.346388	4.5	22.5
<<<<<	1-1-1-1-1-1	0.1666666667	1868.019581	3	15
<<<<	1-1-1-1-1	0.2	2241.623498	2.5	12.5
<<<	1-1-1-1	0.25	2802.029372	2	10
<<	1-1-1	0.3333333333	3736.039163	1.5	7.5
<	1-1	0.5	5604.058744	1	5
<<<<<<<	2-2-2-2-2-2	0.4	4483.246995	2.5	25
<<<<<	2-2-2-2	0.5	5604.058744	2	20
<<<<	2-2-2	0.6666666667	7472.078326	1.5	15
<<<	2-2	1	11208.11749	1	10
<<<<<<<	3-3-3	1	11208.11749	1.5	22.5
<<<<<	3-3	1.5	16812.17623	1	15
<<<<<<<	4-4	2	22416.23498	1	20
<<<<<<<<<	5-5	2.5	28020.29372	1	25

Figure 9: Table showing stiffness characteristics with regards to washer configuration (Smith, Stanford & Cutlan, 2015).

2.3 Dual-Shock Suspension

Dual shock suspension is the more common assembly in modern day race cars. Fundamentally they use two dampers mounted in board on the chassis which are connected to each other via an anti-roll bar situated towards the front of the vehicle. To transmit the movement of the wheel from the pushrods to the damper, the relationship between the two are governed by bell cranks and how both the pushrod and damper are mounted on the bell crank.

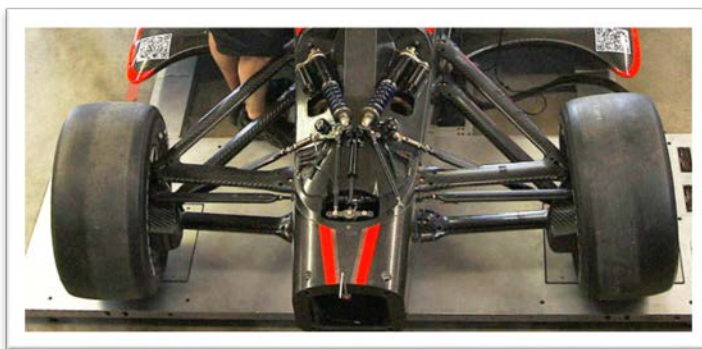


Figure 10: Picture of a dual-shock setup from an indycar. (<http://www.racecar-engineering.com/>, 2012)

By introducing a second damper on the front will allow more adjustment to setup suspension settings which are not only easier but more precise in comparison to mono-shock systems. A key

factor to dual-shock is that it is possible to have damping in a roll situation via the use of an anti-roll bar.

In comparison to mono-shocks with regards to weight dual-shock systems are slightly heavier due to the necessary components needed but dual shocks do not need a robust lower mounting bracket. When referring to mono-shocks the lower mounting bracket which not only mounts the damper to the chassis but also the roll stack of bellville washers making it fairly robust as shown in figure 4, so it is fair to say the weight gain in comparison will only be marginal but the dynamic advantages are considerable. With regards to Roll resistance dual-shock systems offer higher roll resistance due to the wider spring base of its mono-shock counterpart.

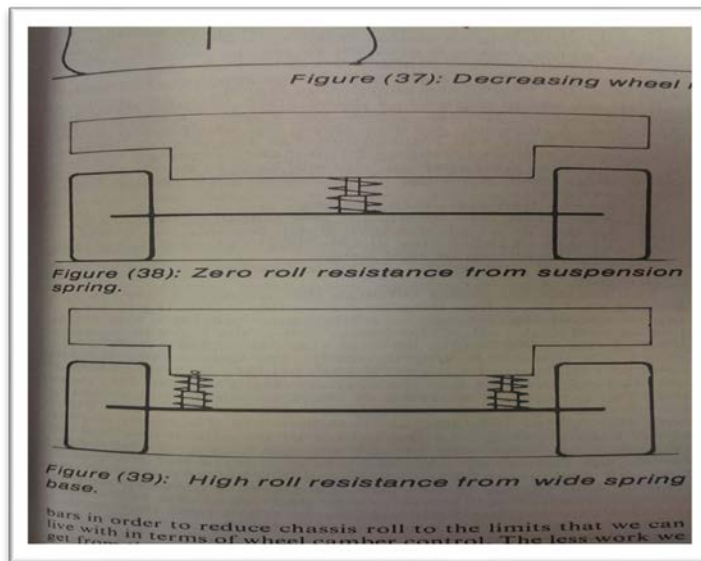


Figure 11: Picture of illustration showing difference of roll resistance between mono-shock and dual-shock. (Smith, 1978)

2.4 Bell Cranks

A key component a dual-shock incorporates is the bell crank which transmits the force of the pushrod which then pivots the bell crank and the force acts upon the damper thus dampening the impact.

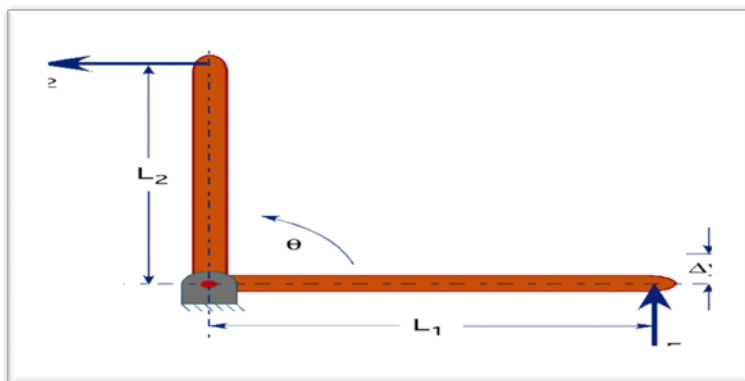


Figure 12: Diagram showing a basic bell crank. (www.daerospace.com, 2013)

Figure 12 shows a basic bell crank design. Force from the pushrod (F_1) at point (a) acts about the pivot point (θ) which then moves the bell crank and creates a force (F_2) at point (b) and it is this force which acts upon the damper. By manipulating the lengths of both members of the bell crank along with the angle between the two will affect the suspension behaviour in both bump and

rebound. (Rowley, 2008) implies that by altering bell crank parameters will cause the following effects to suspension behaviour. These effects are:

Parameter	Change of Parameter	Effect of Change
[1]. Push Rod Attachment (A)	Move closer to spring attachment (B)	Increase of initial rate but cause a rise in rate as force is applied.
[2]. Push Rod Attachment (A)	Move away from spring attachment (B)	Create a lower initial rate but rate will rise slightly as force is applied.
[3]. Push Rod Attachment (A)	Move Attachment further from pivot point	Create a lower initial rate which will increase as force is applied.
[4]. Push Rod Attachment (A)	Move attachment closer to pivot point	Create a higher initial rate which will fall as force is applied.

Table 1: Table showing effects of changing bell crank Parameters. (Rowley, 2008)

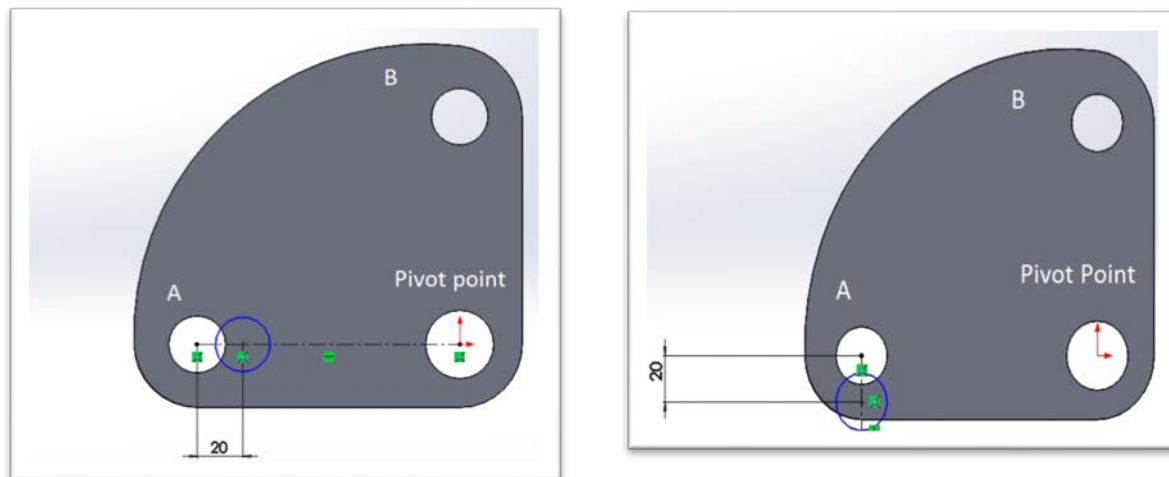


Figure 13 & 14: Bell crank drawing showing iteration [1] & [3] according to Table 1.

Figures 13 & 14 are demonstrations of the physical alterations of mounting points at the bell crank in relation to motion ratio. In scenario [1] (Rowley, 2008) states that this alteration will create an increase of initial rate which will rise as force is applied. Scenario [3] however will create a lower initial rate which will rise as motion ratio is applied. The same effects will be created alterations were made at point B instead of point A.

2.5 Motion Ratio

Motion ratio as previously stated is one of the most important factors of suspension design. It is defined as the relationship between vertical wheel displacement and spring displacement, and the behaviour of this characteristic is one that has a major influence of suspension performance. (Rowley, 2003) gives an example of motion ratio where 1" of wheel displacement causes 0.496" of spring displacement resulting in a given motion ratio of 0.496. This value could then be used to calculate the force created via (force * distance) if the spring rate is known.

A key point to add regarding Motion ratio analysis is that (Rowley, 2003) highlights specifically that the use of three dimensional software such as Wingeo 3 is imperative to accurately determining the motion ratio.

When referring to research conducted on the Rc97 by (Smith, Stanford & Cutlan, 2015) the motion ratio of the front suspension has an average rate of 0.497 in roll and approximately 1.2 when the system is in bump.

Motion Ratio (Roll)			
	Wheel	Shuttle	Ratio
mm	0	0	0
	2	0.89	0.445
	4	1.9	0.475
	6	2.88	0.48
	8	3.76	0.47
	10	5	0.5
	12	6.1	0.508333
	14	7.2	0.514286
	16	8.5	0.53125
	18	9.4	0.522222
	20	10.5	0.525
Average Motion Ratio			0.497109

Figure 15: table showing motion ratio of shuttle in roll for RC97 (Smith, Stanford & Cutlan, 2015)

2.6 Rising & Falling Motion Ratio

As briefly discussed in chapter 2.4 by altering the mounting points of both the pushrod and suspension on the bell crank will have an effect on the motion ratio.

By placing in particular the push rod mounting point above or below the pivot axis as mentioned by (Rowley, 2008) will either give a rising rate or falling rate of motion ratio depending on which way the push rod mounting is placed relative to the pivot axis. (Pashley, 2008) informs that rising rate motion is considered to be a requirement of modern racecar design. The motion ratio rises as the vehicle goes into bump meaning spring displacement exceeds wheel displacement which has the advantage of lower variations of ride height aiding in improving stability due to the lower migration of roll centre.

Falling rate is the exact opposite of rising rate where motion ratio falls when going into bump causing greater wheel displacement. This increase in wheel displacement will have a knock-on effect of the roll centre migration and thus producing decreased stability in comparison to rising rate. A graph showing this can be seen in figure 16.

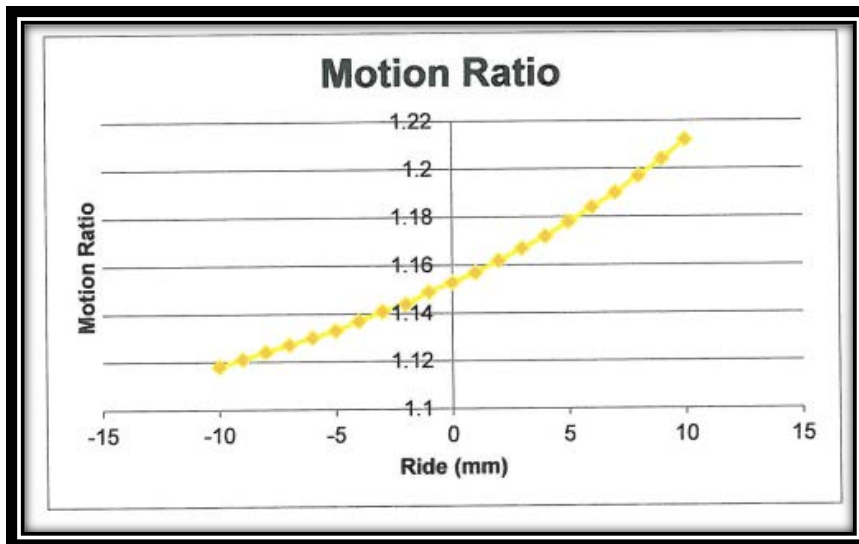


Figure 16: graph demonstrating falling rate.

2.7 Anti-Roll Bars

The anti-roll bar is the component which prohibits regulated body-roll which may affect a vehicles cornering performance. Giving the case of the mono-shock setup on the RC97, roll is governed by the rocker shaft located underneath the damper which uses Belleville washers to determine the amount of permitted roll in the system (more washers = less roll).

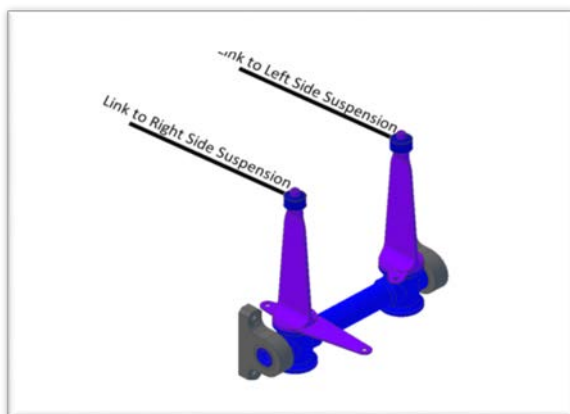


Figure 17 Diagram showing a blade type anti roll bar. (www.thinkfastengineering.com, 2012)

According to (Staniforth, 2006) Anti-roll bars do three things which are firstly resist roll due to the suspension wanting to permit rise at the outer wheel and fall at the inner wheel in a cornering situation. Roll is governed by the amount of twist along the anti-roll bar (similar to torsion beam suspension). Secondly the anti-roll bar acts as a third spring which is in conjunction between the two, and finally It determines the amount of weight transfer between the inner and outer wheels characteristic crucial to obtaining an optimum setup.

(Staniforth, 2006) also enlightens that a correctly fitted and dimensioned anti-roll bar will provide effects up to 10 times greater than that of simply fitting a stiffer spring. With regards to the RC97 blade type anti-roll bars are used which are connected via a link directly off the bell crank at the pushrod end. The motion travelling through the bell crank pulls this link which acts upon the anti-roll bar which lowers body roll.

Figure 17 shows a conventional blade type anti roll bar, notice the angle of these blades can be adjusted to give the desired amount of roll stiffness on the axle it is fitted to. By adjusting the angle of these blades is to increase or decrease the amount of surface area a force can act upon in that plane due to the fact pressure = force / area ($P = F/A$). By increasing area (A) with the same force decreases P acting upon the area in question causing minimal deflection.

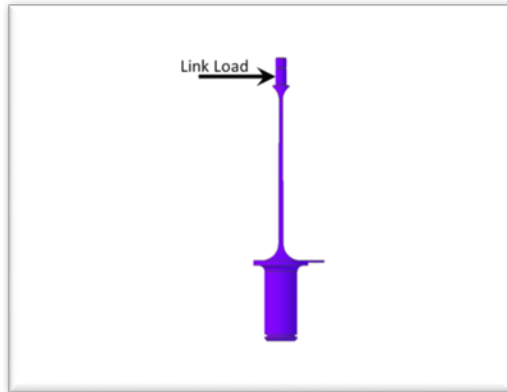


Figure 18: Diagram of anti-roll bar blade being subjected to a force. (www.thinkfastengineering.com, 2012)

Figure 18 is an example of setting up an anti-roll bar on its softest setting. The blade is perpendicular to the force being applied through it resulting in a minimal surface area in that plane which will cause the member to deflect more in comparison to a stiffer setting where the blade runs parallel with the force applied.

Another factor to consider with anti-roll bars is the length of the spring base. (Smith, 1978) enlightens that the less work from the dampers in roll resistance, the stiffer the anti-roll bar must be. This rather important due to the fact that the greater the spring displacement per degree of chassis roll will result in the lateral forces created in roll will be dampened by the shock. (Smith, 1978) also informs that a lack of damping in this situation will result in an action called “roll rock back” where the sprung mass will oscillate in roll.

2.8 Roll Centre

The roll centre is defined by (Rowley, 2003) as the point in space where the chassis rotates when subjected to an external force. Alteration of this factor will have a considerable effect on the vehicles cornering capability both positive and negative. (Shim & Velusamy, 2006) inform that by raising the roll centre height will decrease the level of roll angle produced in cornering

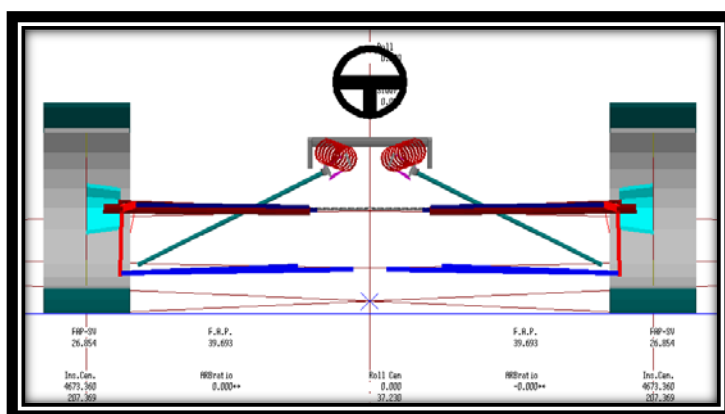


Figure 19: Basic winged 3 model showing roll centre.

Figure 20 is a demonstration confirming that by raising the roll centre height has a dramatic effect in lowering roll angle when cornering. (Shim & Velusamy, 2006) imply that by increasing the roll centre height by 20% lowers the roll angle by approximately 0.5° for an SUV (Sports Utility Vehicle) in the effort to lower the vehicle susceptibility to an accident via rollover.

As Instructed by (Adams, 1992) who explains that roll angle is dependent on the distance between the roll centre and the centre of gravity. Reducing this set distance will lower the roll angle but a higher roll centre causes increases in jacking effects and more erratic suspension characteristics. (Adams, 1992) also goes onto say that raising the roll centre height can reduce camber gain when the vehicle goes into bump & that the most successful cars tend to have a roll centre situated 25mm below the ground and 75mm above ground.

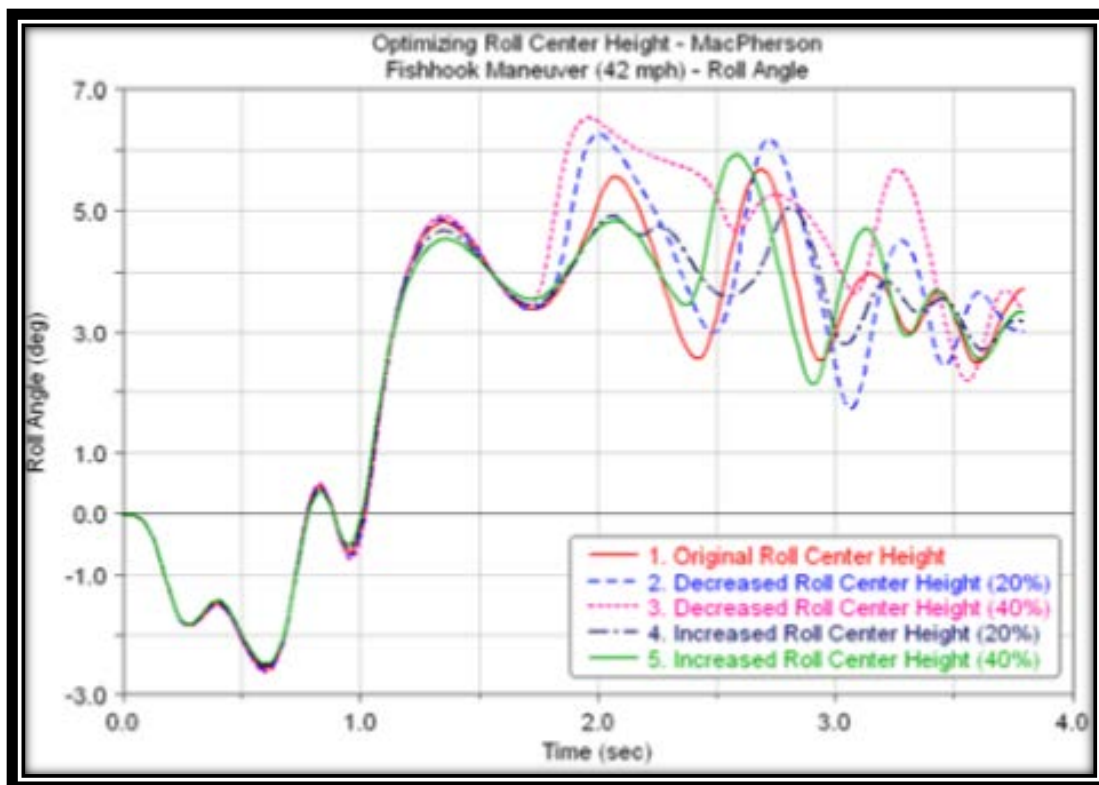


Figure 20: Graph showing roll angle vs variation in ride height (Shim & Velusamy, 2006)

2.9 Weight/Load Transfer

Load transfer occurs when the vehicle is subjected to either cornering or braking forces. (Milliken & Milliken, 1995) inform that Lateral load transfer occurs when the centrifugal forces created in a cornering situation oppose the lateral acceleration forces produced by the tyres. Primarily speaking weight transfer is governed by three major factors which are:

- Centre of Gravity Height.
- Roll centre heights (Both front and rear).
- Roll stiffness between front and rear.

(www.imps4ever.info, 2014)

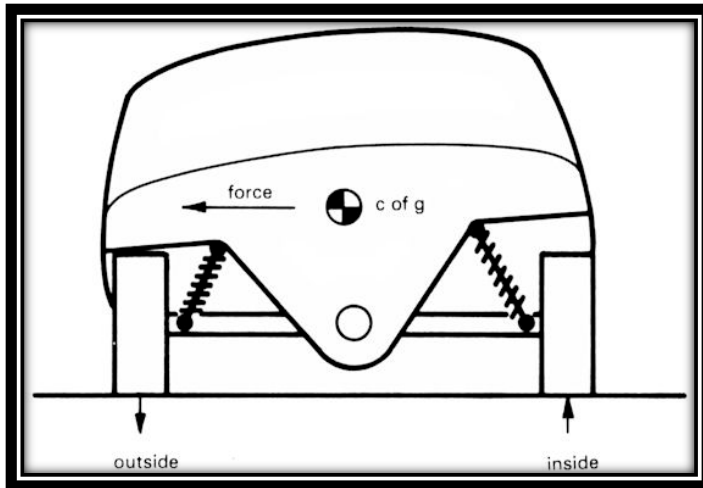


Figure 21: Diagram showing load transfer (www.imps4ever.info, 2014)

Figure 21 shows basic load transfer during cornering. As the vehicle turns the vehicle begins to lean outwards and by doing so changing the load on both wheels. The outside wheel gains load while the inside wheel loses load which if excessive enough will have an impact on the handling balance of the vehicle. Fundamentally speaking load transfer is influenced by mass, lateral acceleration, COG height and track/wheelbase hence the formula as explained by (Milliken & Milliken, 1995):

$$[1]. WT \text{ lat} = \frac{M * Ay * H}{T}$$

$$[2]. WT \text{ Long} = \frac{M * Ax * H}{WB}$$

2.10 Roll Stiffness

Roll Stiffness is the factor that defines a given axle's ability to resist roll when cornering. Roll stiffness is important as this can have an influence to the handling balance of the vehicle. (Deakin, Crolla, Ramirez & Hanley, 2000) imply that by altering load transfer via altering roll stiffness characteristics can either lower understeer by lowering load transfer at the front and increasing load transfer at the rear and the opposite to counter oversteer if needed.

Roll stiffness is influenced by factors such as track length, wheel rate & spring rate, hence the formula:

$$Roll \text{ Stiffness} = \frac{Track^2 * wheel \text{ rate}}{2}$$

3. Methodology

3.1 Justification of Methodology

Before the data can be collected a suitable methodology of design of experiments was derived by researching methods used by suspension system designers. With the ever increasing influence of CAD within design it would be beneficial to start by constructing a CAD model of the chassis using data taken from a measuring exercise conducted previously. As discussed by (Gaffney & Salinas, 1997) who enlightens at the fact that using a kinematic software package such as wingeo 3 in conjunction with CAD software is the most easiest method of suspension design, due to the ease of moving suspension points which can then be edited within CAD.

As previously discussed (Rowley, 2003) emphasizes the importance using Wingeo 3 in suspension design to accurately determine motion ratio & roll centre characteristics in both static and dynamic situations.

There are however more sophisticated ways of suspension design which shall be discussed later on within this report.

3.2 The Chassis (CAD Model)

Upon the purchasing of the RC97 the vehicle was purchased as a package meaning the entire vehicle would require assembling of all major components, this gave the opportunity to conduct a measuring exercise of the chassis should the need for additional modification arise in the future.

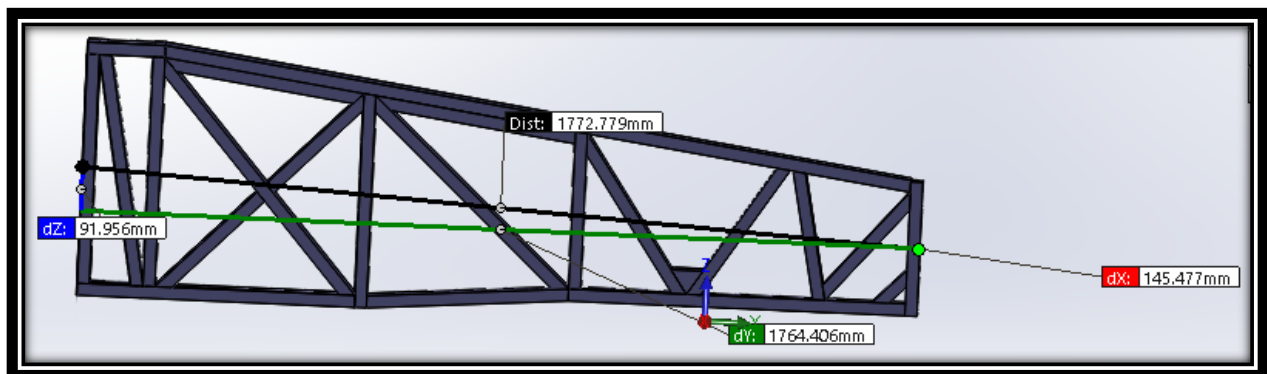


Figure 22: Picture 1 showing CAD model of Chassis.

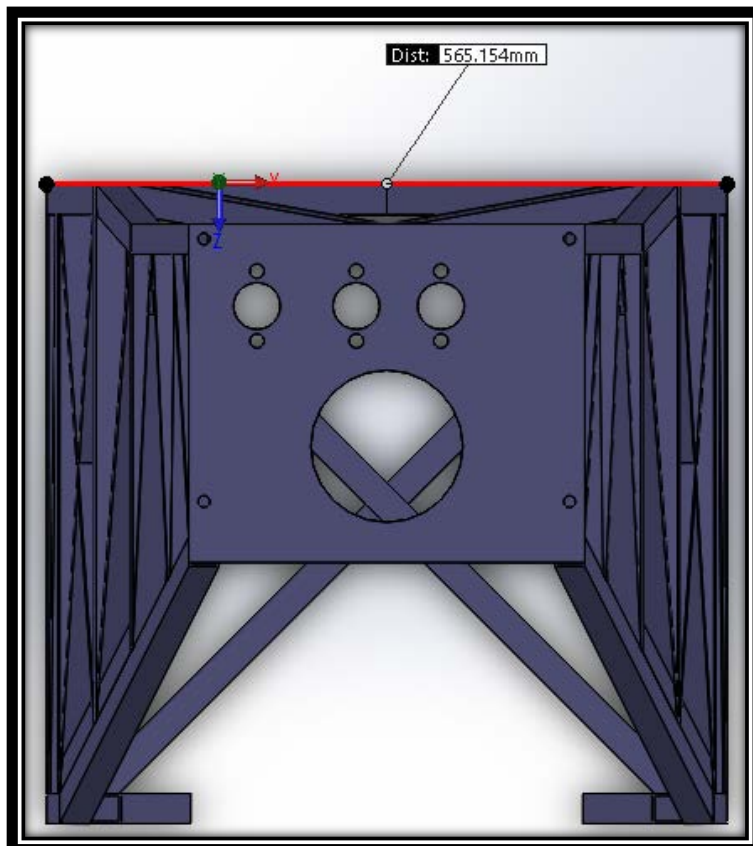


Figure 23: Picture 2 showing CAD model of the chassis.

Figures 22 & 23 show the cad model of the RC97 chassis resulting from the initial exercise. All measurements were taken using handheld measuring equipment such as Vernier calipers, height gauges, steel rules and micrometers.

Live measurements boast the advantage of being obtained fairly quickly with reasonable accuracy of data but a considerable amount of time and man-hours were sacrificed in order to conduct this observation, although this opportunity was considered too advantageous to pass away. A key point to add is that this CAD model is also being used to construct a new chassis so that the team will eventually be running two cars with the latter being used for experimental purposes and data acquisition.

3.3 Chassis Components

3.3.1 Wishbones

Like the chassis, dimensions were taken of the wishbones in order for them to be modelled both on CAD and wingeo 3. It is important that the dimensions are as accurate as possible given the equipment provided as the geometry of the upper and lower wishbones define the track and this will have an influence on Camber angle in bump and also turning radius.

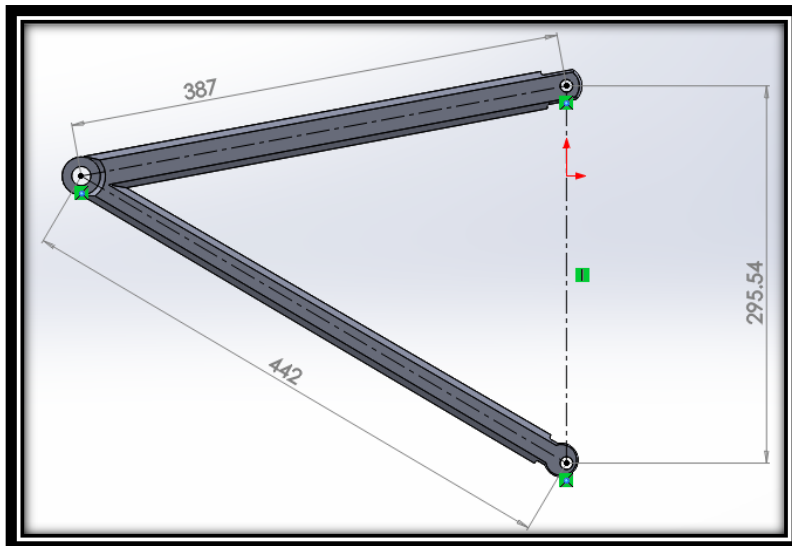
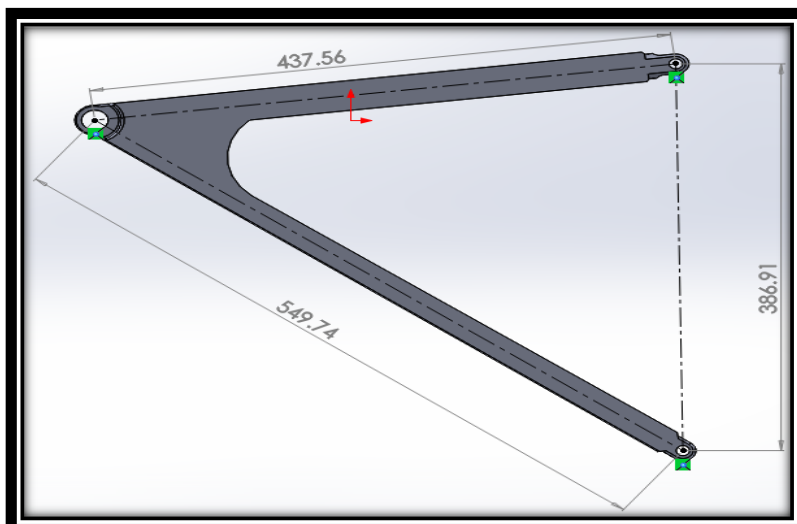


Figure 24 & 25: Cad Model of the upper wishbone & lower wishbones



3.3.2 Clevises

The clevises are the component which provide a safe attachment of the wishbones to the chassis. A clevis is machined from one piece of material (in this case steel) which has a centrally mounted spherical bearing around which the wishbone can pivot and rotate as the wheel interacts with the road surface.

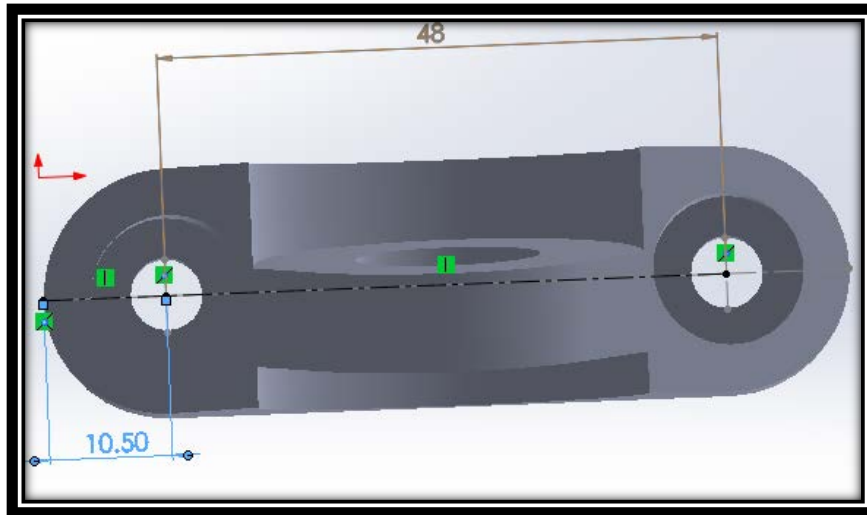


Figure 26: CAD model of the RC97 clevis.

This CAD design of clevis has been derived and are currently being used on the RC97. The tapered angle of the bearing mount gives less chance of the wishbone clashing with the clevis in comparison to the previous straight bearing mount point which was more prone to this.

To allow the free movement of the wishbone and thus correctly attaching them was achieved by mating simple spherical bearings using a tangential mate between the outer edge of the bearing and the inner edge of the bearing mount on the clevis. From here the wishbone was then mated to the bearing using both concentric and distance mates. The concentric mate aligned the holes of both the wishbone and the bearing and the distance mates defined the set distance so that when it is moved vertically simulating tyre interaction with the road surface, these two separate components would not clash.

3.3.3 Bell Crank

As a basis the bell crank design was based upon the bell cranks used on the rear suspension system so it was imperative that the bell crank is measured as accurate as possible. The bell crank is the most influential with regards to motion ratio as the geometry defines the motion ratio & also its characteristic (rising/falling or neutral).



Figure 27: Picture of bell crank being measured.

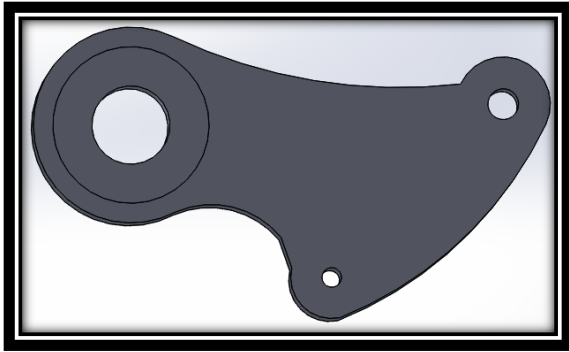


Figure 28: CAD model of the RC97 bell crank.

From here a second bell crank was derived which had multiple mounting points for both push rod and damper mount and doing so changing the motion ratio. When referring to the literature and the advantages of rising rate motion ratio, the new bell crank is modelled around exploring different rates of rising rate motion ratio via placing the push rod mounting points above the pivot axis as instructed by (Rowley, 2008) in order to achieve a rising rate motion ratio.

3.3.4 Other Components Modelled

To accommodate the new layout there were minimal modifications to the chassis. New additional members to install both the dampers and bell cranks were derived to achieve the desired position on the chassis where these components will be mounted to.

A key consideration was to get the bell crank pivot as close to the lower pushrod mount point as possible so that the pushrod only moves linearly and not at angle. The Suspension mounting points were placed as far apart as possible to give greater roll resistance characteristics as shown in figure 11. Also the majority of welded joints relevant to the suspension have been mounted so that they are in compression and not tension so there is minimal chance for welds to fail.

An anti-roll bar was then modelled to fit at the front of the chassis just above the brake and clutch master cylinders as this is an ideal place to mount this component given the limited space at the front along with appropriate mounting points (these shall be discussed further in the results section of this report).

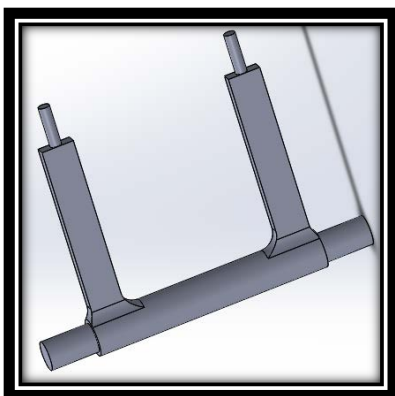


Figure 29: CAD model of anti-roll bar.

3.4 Wingeo 3 RC97 Model

3.4.1 Taking Hard-point Measurements from CAD Model

At this point the CAD model has been completed and from here measurements were taken of the hard points of the suspension system via the measure feature on Solidworks. The measure feature defines a point in space in the X, Y & Z axis, exactly what was needed to derive the kinematic model on wingeo 3.

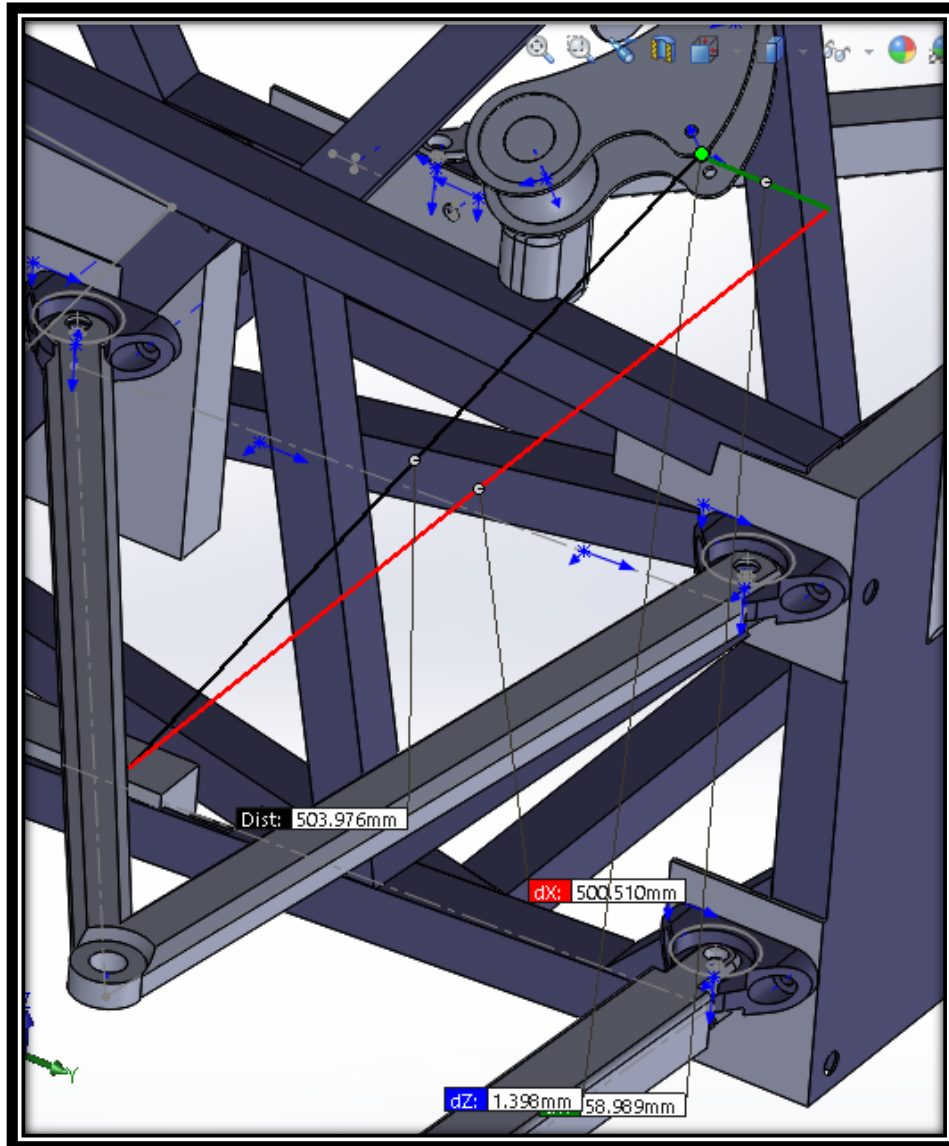


Figure 30: Picture showing hard points measurements being taken using solidworks measure feature.

As illustrated in figure 30 solidworks measure feature gives measurements of all three axis and the overall distance. Depending on the measurements reference points were used and the position of the desired point (in this case the pushrod mount on the bellcrank) which then gave the measurement for a given axis.

For the x axis the reference point for measurements for X were taken from the centre of the hub which for the RC97 aligns with the outboard hub mount of the lower wishbone and the Y axis measurements reference point was taken from the centre of the front facing plate as illustrated in figure 31.

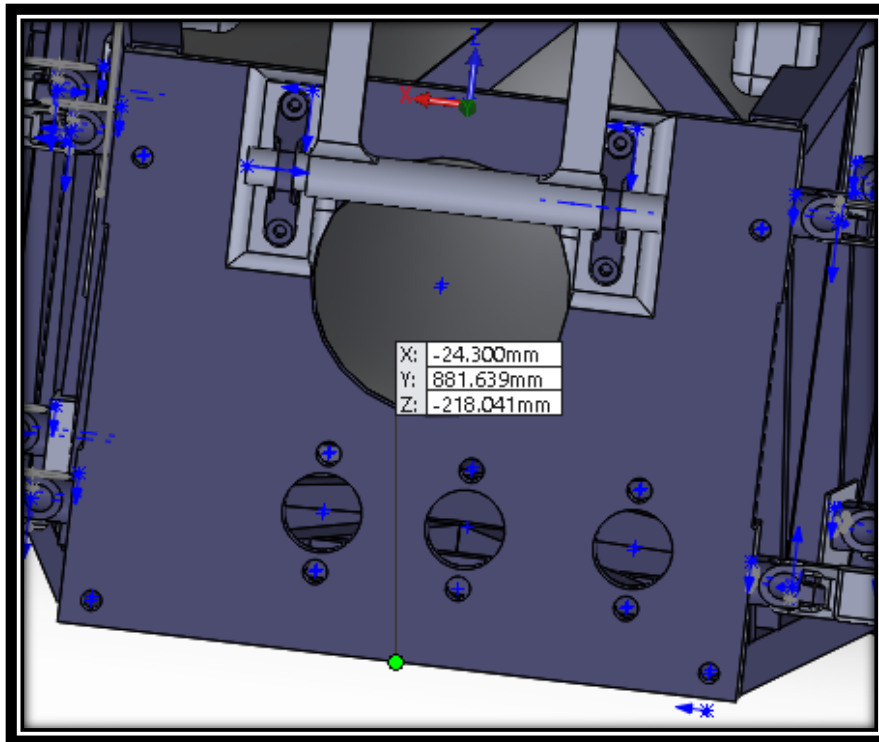


Figure 31: Picture showing Y axis measurement reference point (highlighted in green).

3.4.2 Rear Wingeo Model

In order to fully satisfy the aims for the project a wingeo 3 model of the rear was also derived in order to create a front suspension system which complimented the rear with regards to factors such as roll stiffness, roll centre and motion ratio.

Measurements of the rear suspension system hardpoints were taken from (Pugh, 2008) who conducted a full chassis measuring exercise and then transferred these measurements onto wingeo 3 to derive a kinematic model of the rear. This was advantageous as this saved the necessity of a second measuring exercise which would require a couple of hours to complete, coupled with the fact that the RC97 is stored in a separate garage off campus which would require it to be brought down again saving time.

3.5 Data Collection from Wingeo 3

3.5.1 Summarising Wingeo data

To get an idea of the suspension systems characteristics generic data was taken of both suspension systems to understand how the vehicle behaves in bump, rebound and roll. As Pashley (2008, Page 32) emphasises that by testing the vehicle in the following situations will give a generic idea of how the vehicle behaves these tests are:

- 25mm bump with 0 roll.
- 25mm bump with 1 degree of roll.
- Normal ride height with 1 degree of roll
- 25mm bump with 1 degree of roll.

From here comparatives shall be drawn to show characteristics regarding roll centre migration in both bump rebound and roll to signify which motion ratio is more stable rising or falling and thus confirming the validity of the literature review.

3.5.2 Calculating Centre of Gravity Height above the Roll Axis (HSRA)

By introducing a new front suspension system this has caused a change in roll centre height and as a result of this change the HSRA value must now be recalculated via taking moments. HSRA is important as this value is used to find the overturning moment from which the roll angle can be defined for a given lateral acceleration.

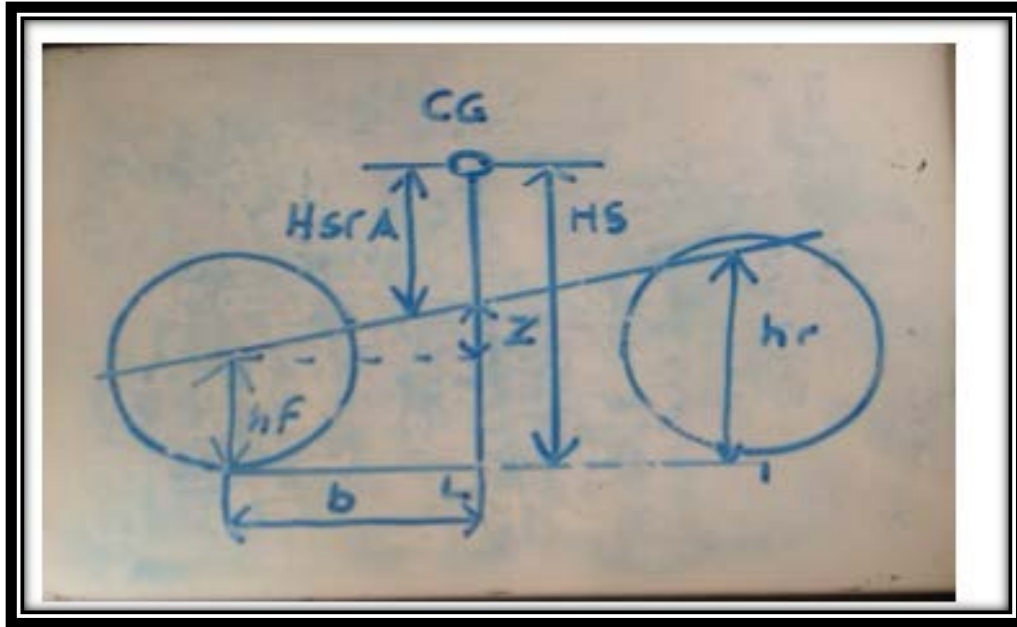


Figure 32: Diagram showing theory for calculating HSRA (Smith, Stanford & Cutlan, 2015).

Using the theory shown in figure 32 the following formulas were derived which are:

$$[1.] \frac{Z}{b} = \frac{Hr - Hf}{L}$$

To calculate HSRA correctly equation [1.] is re-arranged to make Z the subject which gives:

$$[2.] Z = B * \frac{Hr - Hf}{L}$$

Using the new Front roll centre height (Hf), calculations were made to find Z which is then used to calculate HSRA via the formula:

$$[3.] HSRA = HS - Z - Hf$$

HS is the centre of gravity height taken from (Pugh, 2008) as being 0.32m. Z was found to 0.044m and Hf of the new suspension system at the front is read at 0.03812m (38.012mm) via the Wingeo 3 Model.

3.5.3 Calculating Roll stiffness

For roll stiffness calculations to be conducted, the wheel rates must first be determined. Wheel rate is calculated via the formula:

$$[4.] \text{Wheel rate} = \text{Spring rate} \left(\frac{N}{m} \right) * \text{Motion Ratio}^2$$

For the purpose of calculations spring rates are equal front to rear at a rate of 800 lbs/inch or 140101 N/m with the motion ratio value being read from the wingeo 3 model. This is conducted for both front and rear suspension systems.

With wheel rates calculated roll stiffness front and rear can now be calculated using the formulas:

$$[5.] \text{ Roll stiffness} = \frac{\text{Track}^2 * \text{Wheel rate}}{2}$$

$$[6.] \text{ Roll stiffness front} = \frac{\text{Front Track}^2 * \text{Front Wheel rate}}{2}$$

$$[7.] \text{ Roll stiffness rear} = \frac{\text{Rear Track}^2 * \text{Rear Wheel rate}}{2}$$

3.5.4 Calculating Roll angle

At this point factors such as roll stiffness's, HSRA have all been calculated thus far allows the roll angle to be calculated for a given lateral acceleration. The given formula for roll angle is given by:

$$[8.] \text{ Roll angle} = \frac{\text{Moment of overturning } (MS * Ay * HSRA)}{\text{Roll stiffness Total (front + rear)}}$$

Ms is the total sprung mass taken from (Smith, Stanford & Cutlan, 2015) as being 483.5 kg and the lateral acceleration Ay is 1g which is the equivalent to 9.81 m/s. the initial answer is given in radians which is then converted to degrees by multiplying 180/π. A key point to add is that by halving the lateral acceleration it became possible to calculate the roll angle at 0.5G or 4.905 m/s which equates to half the roll angle at 1g.

3.5.5 Iterative Process

With roll angle found this figure was punched into wingeo 3 and then the whole calculation process was then repeated. The objective of this iterative process was to find the true roll angle.

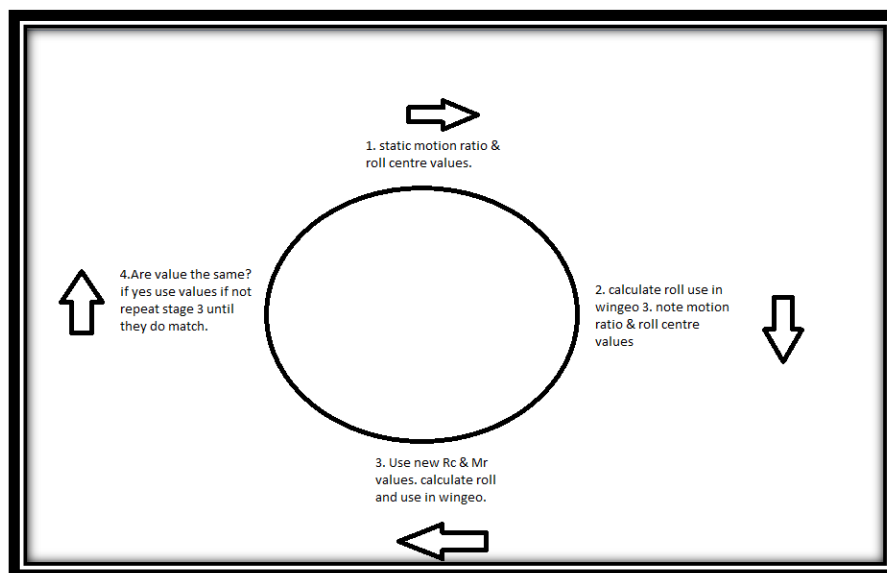


Figure 33: diagram showing iteration process to obtain true roll angle.

The true roll angle was obtained using the process indicated in figure 33. Due to the minimal change in roll angle this iteration was only repeated once due to the fact wingeo 3 works to three decimal places. On the second iteration the results obtained were rounded off to 3 DP which worked out as the same as the previous value and by doing so validating the original roll angle value.

3.5.6 Calculating Ride Deflection

Ride deflection (pitch) is also calculated to as any difference in pitch will signify changes in loads at the front wheels and this load change will have an effect on the load split front/rear which according to (Deakin, Crolla, Ramirez & Hanley, 2000) will have an effect on the handling balance.

Ride deflection is calculated by the formula:

$$[9.] \text{ Ride deflection (mm)} = \frac{\text{Weight Transfer}}{\text{Front Wheel rate}}$$

With weight transfer being:

$$[10.] \text{ Weight Transfer long} = \frac{Ms * Ax * Hs}{WB}$$

Using the sprung mass (Ms), Longitudinal acceleration (Ax), Centre of gravity height (Hs) all dividing by the wheelbase (WB). A demonstration of this can be viewed in the appendix to calculate accuracy of cornering simulation data.

3.6 Corner Simulation

The calculations relating to pitch and roll angle are necessary in order to simulate a cornering situation. By doing this it gives a good indication of how the suspension system behaves through a multiple of corner situations including braking cornering and accelerating. Through each section of the corner data regarding motion ratio, roll centre migration, camber, spring displacement have all been recorded and compared between rising and falling rate motion ratio in the aid of increasing stability at the front end.

Cornering Scenario	Lateral acceleration roll (x)	Lateral acceleration pitch (y)
1. braking	-1 g	0g
2. braking & cornering	-0.5g	0.5g
3. cornering	0g	1g
4. cornering & accelerating	+0.5g	0.5g
5. accelerating	+1g	0g

Table 2: table showing cornering simulation and their respective scenarios.

3.7 Load Split Calculator

With the wheel rates known for both rising and falling rate setups, it was then possible to determine the wheel loads via monitoring the wheel displacement on wingeo 3 for each section of the corner simulation. Then by using a Rowley load split calculator gives the opportunity to see how the load splits affect handling balance and to see if the suspension system causes the vehicle to understeer/oversteer. Given the case of the RC97 monoshock systems tend to be prone to understeer due to excessive front roll stiffness causing the inside wheel to lift resulting in a loss of grip (as shown in figure 5).

Static				48.3%				
				LF+RF				
				3650.0				
				24.2%	LF	RF	24.2%	
Front Split	LF-RF	0.0	1825				1825	
				LF+LR	50.0%	3777		
				(LR-RR) - (LF-RF)	0.0	25.8%		
Rear Split	LR-RR	0.0	1952		LR	RR	1952	
				LR-LF Split	127.0	3904.0	127.0	RR-RF Split
				Wedge	50.0%	LR+RR		
				LR-RF Split	127.0	51.7%		

Figure 34: Picture showing a Rowley load split calculator.

3.8 Bell Crank Design

Besides assessing the conversion a new adjustable geometry bell crank has been derived based upon the advantages rising rate possess as indicated by (Rowley, 2008) and (Pashley, 2008).

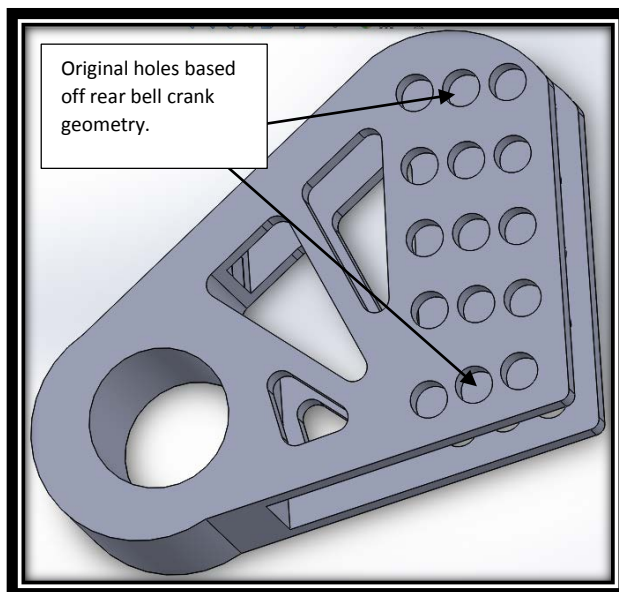


Figure 35: Picture of multiple geometry bell crank concept.

Figure 35 shows the bell crank concept derived from the original. The top two rows consist of mounting points solely for the lower suspension mounts and the remaining 3 rows are for the pushrod mounting points. By having this many variations of motion ratio would be beneficial to finding the optimum values needed to achieve the best setup for a given circuit. The mounting points highlighted in figure 35 are the ones mirrored off the bell cranks used at the rear.

By having bell cranks of the same geometry front and rear will give the same different rates of change to motion ratio thus giving a more balanced setup in theoretically speaking. Due to the limited access to the university engineering workshops it was not possible to create these bell cranks and conduct live testing to test this theory although the theory behind it is logical and may be used

in the near future. (Data regarding motion ratio changes caused by different mounting points can be viewed in the appendix)

3.9 Methodology Summary

To conclude this chapter of the report and for clarification the proposed methodology used for the obtaining is as follows:

1. Derive CAD model of chassis and relevant suspension components and assemble.
2. Take measurements of hard points to be used for the kinematic model & create via Wingeo 3.
3. Take basic via Wingeo 3 data to show systems basic characteristics for both front and rear.
4. Calculate roll angle & pitch for both rising and falling rate suspension systems. Place these values in wingeo to simulate a cornering scenario and take data regarding motion ratio and roll centre migration.
5. Calculate change in wheel loads via wheel displacement use data in Rowley load split calculator, assess how wheel load changes handling balance with regards to rising and falling rate suspension.

4. Results & Analysis

4.1 CAD Model of RC97

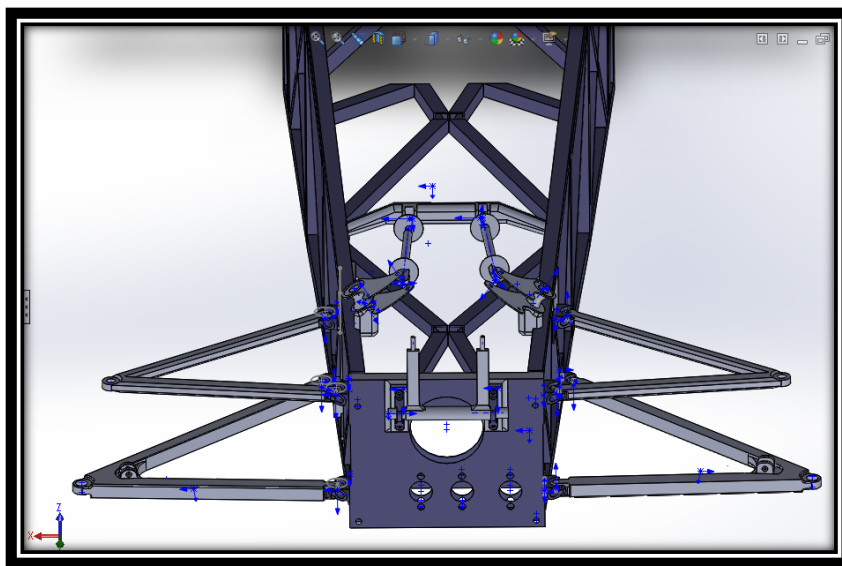


Figure 36: Screenshot of CAD model of new dual-shock suspension system layout.

Figure 36 shows the proposed suspension system layout with a proposed mounting point for the anti-roll bar taking inspiration from figure 12. The distance between the lower and upper push rod mount are equal meaning that the pushrod acts vertically and not through an angle. The dampers are of a basic model due to the fact only the points of mounting points are the main priority for the assembly.

4.2 Wingeo 3 Model for the RC97

4.2.1 Front Rising Rate Suspension System

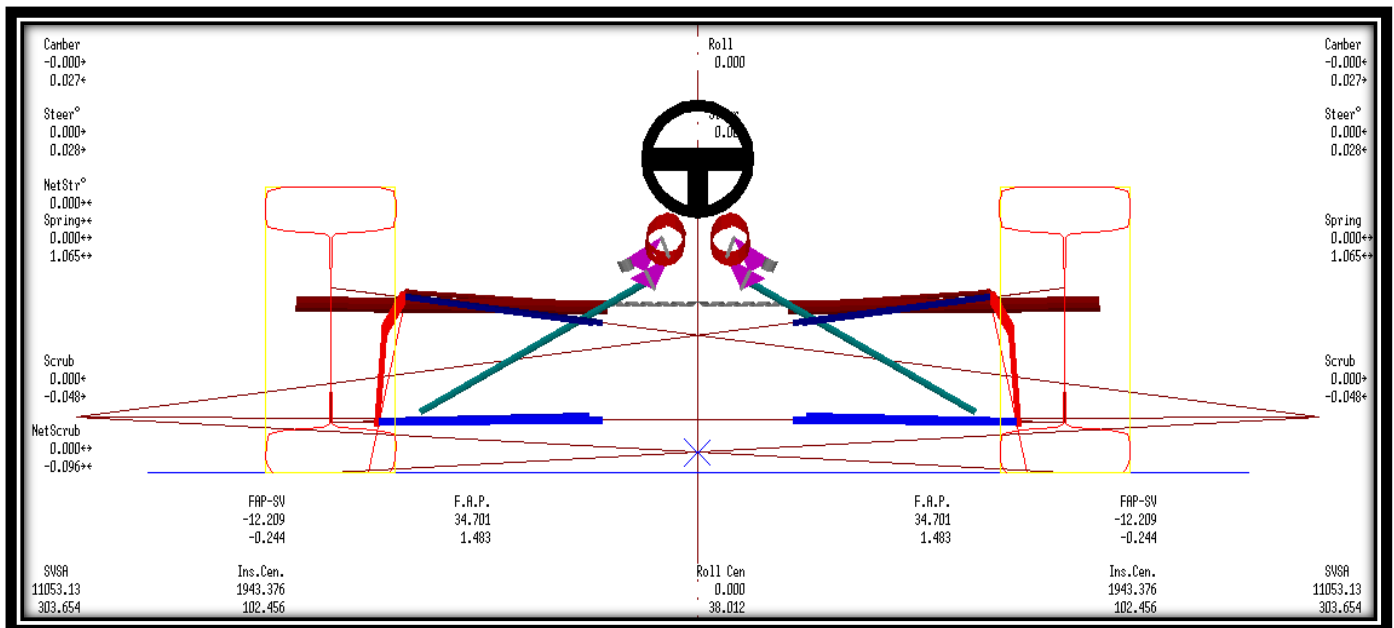


Figure 37: Rising rate wingeo 3 model for the RC97 dual-shock front suspension system.

Figure 37 is the kinematic model for the dual-shock front suspension system proposed for the RC97. Initial motion ratio is 1.065 which in comparison to the mono-shock motion ratio of 0.495 taken from (Smith, Stanford & Cutlan, 2015) is over twice as high.

Initial roll centre height is taken as 38.012mm which when referring to (Adams, 1992) who specifies a sufficient roll centre height range between 75mm above and 25mm below the ground as being an appropriate range with regards to racing cars.

Hub measurements were taken from (Pugh, 2008) to give the vehicle the same camber characteristics in bump and rebound. Using wingeo 3 the distance between the upper and lower ball joints of the wishbones were 228.468mm. A key point to add is that all measurements were raised by 20mm to coincide with the monoposto racing club regulations regarding a minimum ride height of 40mm. Figures 40, 41 & 42 are a culmination of the measurements taken from the CAD model.

Both sides		X fore-aft	Y width	Z vertical	Check distance
Lower A-arm forward	A	-19.335	187.101	96.049	A to B= 437.561
Lower ball joint	B	0.000	624.232	94.540	C to B= 530.661
Lower A-arm rearward	C	336.571	214.043	102.680	A to C= 356.986
Upper A-arm forward	D	46.334	188.632	275.696	D to E= 387.480
Upper ball joint	E	23.457	572.532	323.008	F to E= 471.926
Upper A-arm rearward	F	318.572	207.143	277.048	D to F= 272.870
Wheelbase, track, tire diameter		2520.000	1440.000	525.000	Rollout1649.336
Camber, Toesteer, Toe span		0.000	0.000	711.201	HubTrak 720.000
Tire contact patch		0.000	720.000	0.000	B to E= 235.416

Figure 38: Table showing wishbone measurements for Wingeo model.

Left	Right	BOTH	Suspension	Steering	AUXILIARY	Swaybar	3rd	Pivot	Halfshaft	Clearance	Fixture	Sketch	Origin	Notes
Both sides			X fore-aft	Y width	Z vertical	Check sums		Kingpin axis 12.751 degrees						
Spring upper on chassis	U	248.020	62.298	442.175	U to V= 213.343		Scrub radius 74.375							
Spring lower on suspension	V	36.390	64.325	415.270	Incline 85.692		Caster 5.862 degrees							
Attach_Spring is:		4 is Pivot Bell-cra		☒ Want_Spring True		Caster trail 9.707								
Shock upper on chassis		0.000	0.000	0.000	U to V= 0.000		Upper A-arm 387.465							
Shock lower suspension		0.000	0.000	0.000	Incline 0.000		Lower A-arm 434.431							
Attach_Shock is:		4 is Pivot Bell-cra		☐ Want_Shock False										
Poten upper on chassis		0.000	0.000	0.000	U to V= 0.000		Track at ground 1440.000							
Poten lower suspension		0.000	0.000	0.000	Incline 0.000		Track at hub 1440.000							
Attach_Data_Pot is:		0 is None		☐ Want_Data_Pot False										

Figure 39: Table showing shock mounting point measurements for Wingeo model.

Left	Right	BOTH	Suspension	Steering	Auxiliary	Swaybar	3rd	PIVOT	Halfshaft	Clearance	Fixture	Sketch	Origin	Notes
Both sides			X fore-aft	Y width	Z vertical	Check sums		Kingpin axis 12.751 degrees						
Pivot axis one	P	60.460	144.769	390.128	P to Q= 20.181		Scrub radius 74.375							
Pivot axis two	Q	60.460	136.571	371.687	Incline 66.032		Caster 5.862 degrees							
Push/Pull rod pivot	M	28.989	94.007	354.576	M to W= 511.006		Caster trail 9.707							
Push/Pull rod attach	W	26.698	544.007	112.456	Incline 28.282		Upper A-arm 387.465							
Attach_Push_Pull is:	3 is Lower A-arm		☒ Want_Shock_Pivot True		Lower A-arm 434.431									

Figure 40: Table showing Bell crank measurements for Wingeo model.

From the measurements shown in figures 38, 39 & 40 tests runs were conducted to obtain initial data in accordance with (Pashley, 2008) who specifies doing this in order the get a generic idea of how the suspension system behaves.

ride height mm	motion ratio	RC height mm	camber angle
25	0.776	74.876	0.653
20	0.82	64.476	0.525
15	0.869	60.09	0.395
10	0.925	52.718	0.265
5	0.989	45.359	0.133
0	1.065	38.012	0
-5	1.156	30.676	-0.134
-10	1.269	23.352	-0.27
-15	1.418	16.038	-0.407
-20	1.626	8.735	-0.545
-25	1.96	1.441	-0.685

Table 3: Data of rising rate suspension change with regards to ride height (no roll)

Table 3 shows motion ratio, roll centre & camber characteristics for the rising rate system in both bump and rebound. Motion ratio rises rapidly in bump from 1.065 to 1.96 over 25mm of wheel travel giving an average motion ratio of 1.415 which when compared to the mono-shock data obtained by (Smith, Stanford & Cutlan, 2015) is an increase from 1.233. The motion ratio characteristic can be viewed within figure 41 which signifies an exponential increase in motion ratio as the vehicle goes into bump, indicating a rising rate being used for this setup.

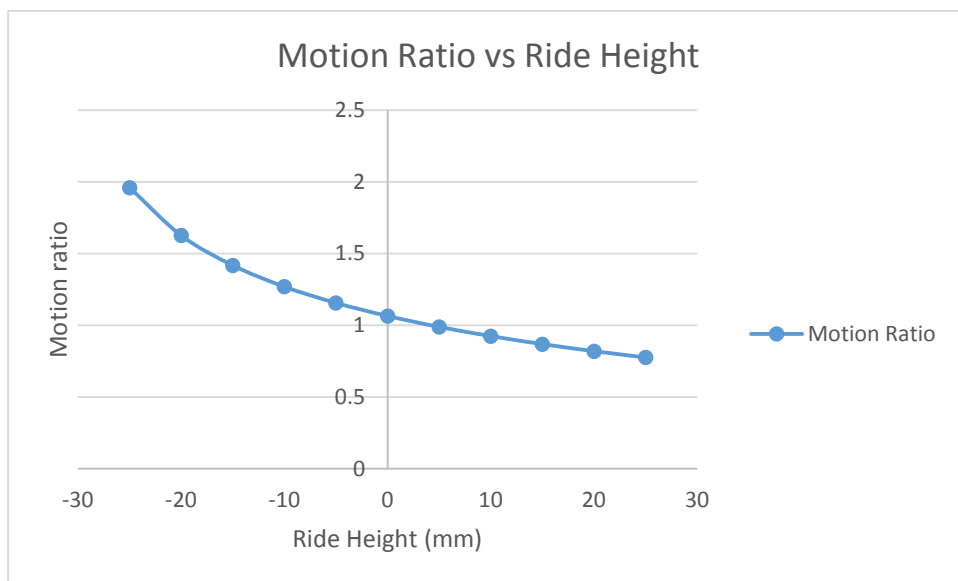


Figure 41: Graph showing motion ratio for rising rate suspension system.

4.2.2 Falling Rate Suspension System

The same process for rising rate within chapter 4.2.1 was repeated to conduct assessment for a falling rate suspension for the front. Using the previous wingeo model a falling rate was achieved by altering the geometry of the bell crank in accordance with (Rowley, 2008) who specifies that by placing the push rod mount (shown in green) on the bell crank below the pivot axis will cause the motion to drop as force is applied giving a falling rate of motion ratio (as shown in figure 42).

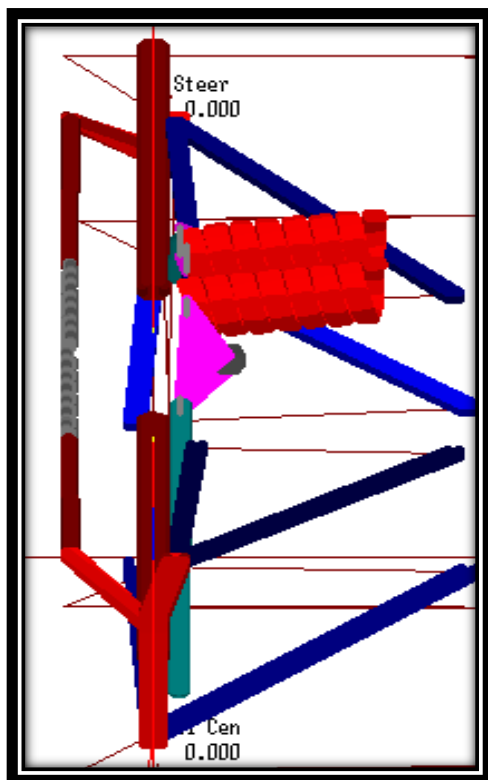


Figure 42: Picture of falling rate system bell crank geometry.

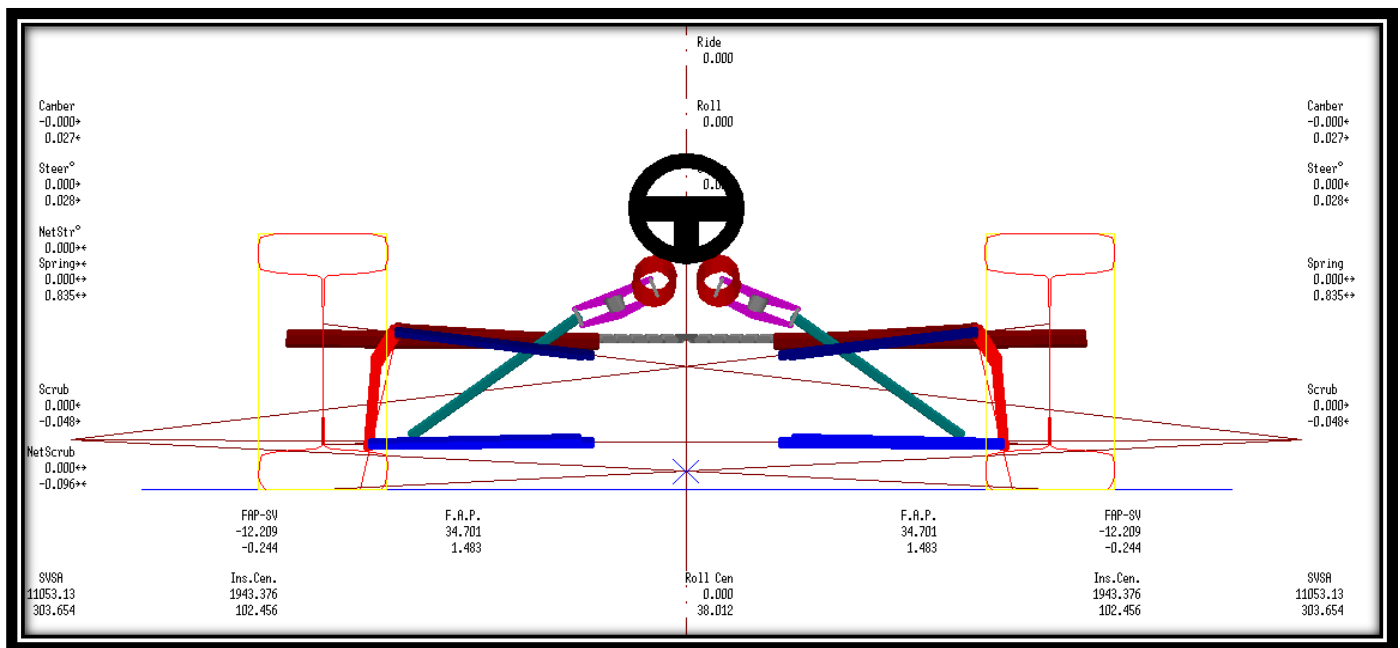


Figure 43: Wingeo model of the falling rate system.

As shown by figure 43 the falling rate system has an initial motion ratio of 0.835 which is approximately 0.2 lower than its rising rate counterpart. Roll centre height is still 38.012mm as there has been no alteration to wishbone mounting points. The only alterations have been to the bell crank pivot axis & push rod mount which have been moved 20mm in the x axis and 120mm on the y axis respectively.

Ride height mm	Motion Ratio	RC height (mm)
25	5.897	74.876
20	1.553	67.476
15	1.177	60.09
10	1.007	52.718
5	0.905	45.359
0	0.835	38.012
-5	0.784	30.676
-10	0.745	23.352
-15	0.713	16.038
-20	0.687	8.735
-25	0.666	1.441

Table 4: Table showing motion ratio characteristic with regards to ride height.

Table 4 show the motion ratio characteristic for the falling rate system. In bump the average motion ratio is approximately 0.719 which is almost half that of its rising rate counterpart. In rebound this setup shows significant levels in motion ratio indicating that this system is more stable at rebound. Due to the nature of racing cars with aerodynamic load this would not prove advantageous as force is applied the motion ratio decreases and by doing so increases the wheel displacement giving lower stability.

Ride height mm	Motion Ratio	roll centre mm	Camber
25	0.774	178.457	1.271
20	0.81	172.417	1.016
15	0.851	166.387	0.762
10	0.898	160.367	0.508
5	0.953	154.357	0.254
0	1.02	148.356	0
-5	1.107	142.365	-0.254
-10	1.229	136.382	-0.508
-15	1.437	130.408	-0.762
-20	2.057	124.442	-1.017

Table 5: Table showing data for rear suspension.

According to table 5 the rear suspension system has an average motion ratio of 1.37 in bump which when compared to rising rate front suspension is lower. By having lower motion ratio at the rear hence lower stability may cause the balance to sway towards oversteer which would be beneficial as the RC97 monoshock system has the tendency to understeer due to its excessively stiff nature.

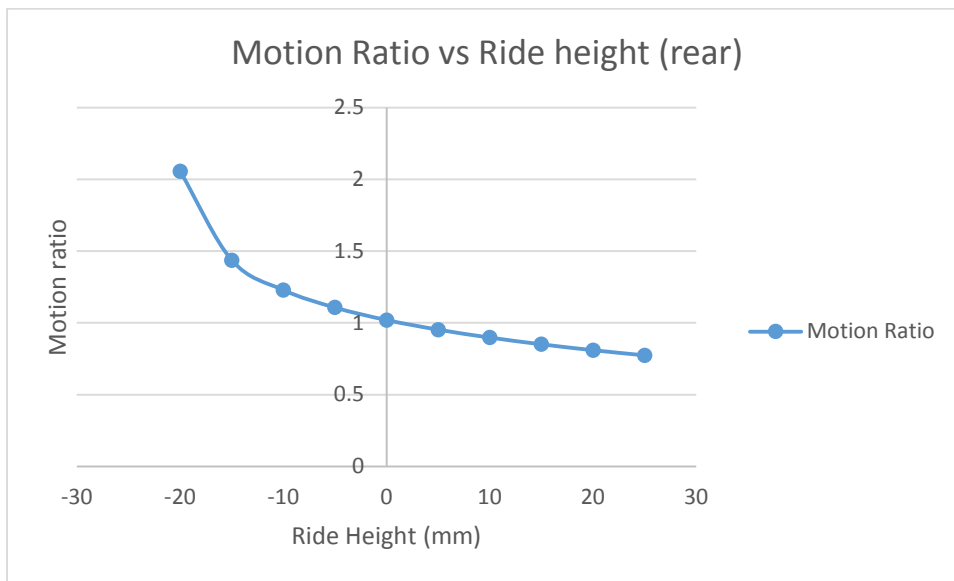


Figure 46: Graph showing motion ratio characteristic for the rear suspension system.

*** More initial data can be viewed within the appendix of this report***

4.3 Roll and Pitch Calculations

4.3.1 Front Suspension System (Rising Rate)

Due to change in roll centre height caused by this conversion, HSRA (centre of gravity height above the roll axis) must be recalculated using the formula shown in chapter 3.5.2.

$$HSRA = H_s - Z - H_f = 0.32 - 0.044 - 0.038 = 0.238 \text{ m or } 238\text{mm}$$

$$Z = b * \frac{H_r - H_f}{Wb} = 1.016 * \frac{0.148356 - 0.038012}{2.54} = 0.044 \text{ M or } 44\text{mm}$$

Hr & hf were obtained using wingeo 3 as these are the roll centre height for both suspension systems. B was calculated by multiplying the wheelbase (2.54 m) by the weight split which is taken as 40/60 front to rear which gives the figure 1.016 indicating the centre of gravity is 1.016 metres from the nose of the vehicle.

Calculations:

Motion Ratio front = 1.065

Motion Ratio rear = 1.020

Sprung mass = 483.5 kg

Spring rate = 140101 n/m

Lateral acceleration = 9.81 m/s² (1g)

1. Wheel rates

$$Kw_{front} = \text{spring rate} * \text{motion ratio}^2 = 140101 * 1.065^2 = 158906.056 \text{ n/m}$$

$$Kw_{rear} = \text{spring rate} * \text{motion ratio}^2 = 140101 * 1.020^2 = 145761.080 \text{ n/m}$$

2. Roll Stiffness

$$K\phi_{front} = \frac{\text{front track}^2 * Kw_{front}}{2} = \frac{1.44^2 * 158906.056}{2} = 164753.798 \text{ nm/rad}$$

$$K\phi_{rear} = \frac{\text{rear track}^2 * Kw_{rear}}{2} = \frac{1.2^2 * 145761.080}{2} = 104947.977 \text{ nm/rad}$$

$$K\phi_{total} = K\phi_{front} + K\phi_{rear} = 164753.798 + 104947.977 = 269701.775 \text{ nm/rad}$$

3. Roll angle static

$$\text{Roll angle } \phi = \frac{\text{Moment of overturning}}{K\phi_{total}}$$

$$\text{Moment of overturning MOT} = Ms * Ay * Hsra = 483.5 * 9.81 * 0.238 = 1128.866$$

$$\text{Roll angle } \phi = \frac{1128.866}{269701.775} = 4.186 * 10^{-3} \text{ radians} = 4.186 * 10^{-3} * \frac{180}{\pi} = 0.239 \text{ degrees } 1g$$

$$\text{roll angle } \phi @ 0.5g = \frac{0.239}{2} = 0.119 \text{ degrees}$$

4. Roll angle (iterative process)

$$Kw_{front} = 140101 * 1.067^2 = 159503.447 \text{ n/m}$$

$$Kw_{rear} = 140101 * 1.022^2 = 146333.253 \text{ n/m}$$

$$K\phi_{front} = \frac{1.44^2 * 159503.447}{2} = 165373.174 \text{ nm/rad}$$

$$K\phi_{rear} = \frac{1.2^2 * 146333.253}{2} = 105359.942 \text{ nm/rad}$$

$$K\phi_{total} = 165373.174 + 105359.942 = 270733.116 \text{ nm/rad}$$

$$\text{Roll angle } \phi = \frac{1128.866}{270733.116} = 4.169 * 10^{-3} \text{ radians} = 4.169 * 10^{-3} * \frac{180}{\pi} = 0.2388 (0.239)$$

5. Pitch (deflection)

$$\text{Pitch} = \frac{\text{weight transfer}}{\text{wheel rate}}$$

$$\text{Weight transfer} = \frac{Ms * Ax * H}{WB} = \frac{483.5 * 9.81 * 0.32}{2.54} = 597.56 \text{ N}$$

$$\text{Pitch} = \frac{597.56}{159503.447} = 3.76 \text{ mm @ 1g}$$

$$\text{Pitch @1.5G} = \frac{3.76}{2} = 1.88 \text{ mm}$$

4.3.2 Front Suspension System (Falling rate)

Using the same process as discussed previously in chapter 4.3.1 roll and pitch calculations were conducted to obtain data to strike a comparison between the two.

Calculations:

Motion Ratio front = 0.835

Motion Ratio rear = 1.020

Spring Mass = 483.5 kg

Spring Rate = 140101 n/m

Lateral Acceleration = 9.81 m/s² (1g)

1. Wheel rates

$$Kw_{front} = \text{Spring rate} * \text{front motion ratio}^2 = 140101 * 0.835^2 = 97681.912 \text{ n/m}$$

$$Kw_{rear} = \text{spring rate} * \text{rear motion ratio}^2 = 140101 * 1.020^2 = 145761.080 \text{ n/m}$$

2. Roll stiffness

$$K\phi_{front} = \frac{\text{front track}^2 * Kw_{front}}{2} = \frac{1.44^2 * 97681.912}{2} = 101276.606 \text{ nm/rad}$$

$$K\phi_{rear} = \frac{\text{rear track}^2 * Kw_{rear}}{2} = \frac{1.2^2 * 145761.080}{2} = 104947.977 \text{ nm/rad}$$

$$K\phi_{total} = K\phi_{front} + K\phi_{rear} = 101276.606 + 104947.977 = 206224.583 \text{ nm/rad}$$

3. Roll angle

$$\text{Roll angle } \phi = \frac{Ms * Ay * HSRA}{K\phi \text{ total}} = \frac{483.5 * 9.81 * 0.238}{206224.583} = 5.473 * 10^{-3} \text{ radians}$$

$$5.473 * 10^{-3} * \frac{180}{\pi} = 0.313 \text{ degrees}$$

$$\text{Roll angle for } 0.5 g = \frac{0.313}{2} = 0.156 \text{ degrees}$$

4. Roll angle (Iterative process)

$$Kw \text{ front} = 140101 * 0.794^2 = 88324.714 \text{ n/m}$$

$$Kw \text{ rear} = 140101 * 1.076^2 = 162205.575 \text{ n/m}$$

$$K\phi \text{ front} = \frac{1.44^2 * 88324.714}{2} = 91575.063 \text{ nm/rad}$$

$$K\phi \text{ rear} = \frac{1.2^2 * 162205.575}{2} = 116788.014 \text{ nm/rad}$$

$$K\phi \text{ total} = 91575.063 + 116788.014 = 208363.077 \text{ nm/rad}$$

$$\text{Roll angle } \phi = \frac{483.5 * 9.81 * 0.238}{208363.077} = 5.417 * 10^{-3} = 0.310 \text{ degrees}$$

5. Roll Angle (Iterative process 2)

$$Kw \text{ front} = 140101 * 0.794^2 = 88324.714 \text{ n/m}$$

$$Kw \text{ rear} = 140101 * 1.075^2 = 161904.218 \text{ n/m}$$

$$K\phi \text{ front} = \frac{1.44^2 * 88324.714}{2} = 91575.048 \text{ nm/rad}$$

$$K\phi \text{ rear} = \frac{1.2^2 * 161904.218}{2} = 116571.037 \text{ nm/rad}$$

$$K\phi \text{ total} = 91575.048 + 116571.037 = 208146.085 \text{ nm/rad}$$

$$\text{Roll angle } \phi = \frac{483.5 * 9.81 * 0.238}{208146.085} = 5.423 * 10^{-3} \text{ rads} = 0.310 \text{ degrees}$$

$$\text{Roll angle } \phi 0.5g = \frac{0.310}{2} = 0.155 \text{ degrees}$$

6. Pitch (deflection)

$$\text{Pitch} = \frac{\text{weight transfer}}{\text{wheel rate}} = \frac{597.506}{91575.048} = 6.52 \text{ mm } 1g$$

$$\text{Pitch at } 0.5g = \frac{6.52}{2} = 3.26 \text{ mm}$$

Due to the lower wheel rates created by lower levels of motion ratio this will have a significant effect on both roll angle and pitch. Rising rate suspension gives a roll angle at 1g of 0.239 degrees which is far low to that of falling rate which gives a roll angle at the same lateral acceleration of 0.310 degrees.

This effect is also mirrored with regards to pitch as rising rate gives a pitch at 1g of 3.76 mm in comparison to falling rates pitch of 6.52 mm. what should be seen is a variation in roll centre movement which is proportional to its respective system. This is important as explained by (Adams, 1992) who specifies that by keeping the roll centre more consistent will give greater consistency of the handling. (Adams, 1992) also goes onto say that by removing any inconsistencies i.e. unpredictable changes in handling while the vehicle is in motion this will give the driver more confidence in the cars handling ability and thus the ability to run faster.

4.4 Cornering Simulation

4.4.1 Rising Rate

Scenario	Lat acceleration (x) pitch G	Lat acceleration (x) pitch mm	Lat acceleration (y) roll G	Lat acceleration (y) roll degrees
1.Braking	-1	-3.76	0	0
2.Braking/Cornering	-0.5	-1.88	0.5	0.119
3.Cornering	0	0	1	0.239
4.Cornering/accelerating	0.5	1.88	0.5	0.119
5.Accelerating	1	3.76	0	0

Table 6: cornering simulation data of rising rate suspension.

1. Braking

AX	AY	Mr left	Mr right	Spring L	Spring R	Rc mm(Z)	RC mm (y)	Camber L	Camber R
-3.76	0	1.072	1.072	-0.603	-0.603	37.54	0	0.016	-0.016

Table 7: Cornering simulation data for braking scene.

2. Braking/Cornering

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
-1.88	0.119	1.098	1.047	-2.225	-0.981	37.527	-13.46	0.060	-0.093

Table 8: Cornering simulation data for braking/cornering scene.

3. Cornering

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
0	0.239	1.117	1.017	-3.275	3.125	37.961	-26.478	0.158	-0.159

Table 9: Cornering simulation data for cornering scene.

4. Cornering/Accelerating

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
1.88	0.119	1.095	1.045	-1.770	1.430	37.830	-13.314	0.077	-0.084

Table 10: Cornering simulation data for cornering/accelerating scene.

5. Accelerating

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
3.76	0	1.081	1.081	-0.608	-0.608	37.400	0	-0.014	-0.014

Table 11: Cornering simulation data for accelerating scene.

Tables 7-11 are a culmination of data obtained in the cornering simulation for the rising rate suspension system. Motion ratio for both sides of the system remains above the value of 1 right throughout the corner which gives roll centre migration in the z direction within a range of 37.961-37.400 mm. roll centre migration in the Y direction ranges between 0 and 26.478 mm.

4.4.2 Falling Rate

1. Braking

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
-6.52	0	0.816	0.816	-1.400	-1.400	36.943	0	-0.051	-0.051

Table 12: Cornering simulation data for braking scene (falling rate).

2. Braking/Cornering

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
-3.26	0.155	0.808	0.852	-1.946	1.283	37.644	-17.429	0.088	-0.112

Table 13: Cornering simulation data for braking/cornering scene (falling rate).

3. Cornering

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
0	0.310	0.794	0.887	-3.178	3.349	37.926	-34.342	0.205	-0.206

Table 14: Cornering simulation data for cornering scene (falling rate).

4. Cornering/accelerating

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
3.26	0.155	0.813	0.858	-1.960	1.293	37.523	-17.617	0.095	-0.206

Table 15: Cornering simulation data for cornering accelerating scene (falling rate).

5. Accelerating

AX	AY	Mr left	Mr right	Spring L	Spring R	RC mm (z)	RC mm (y)	Camber L	Camber R
6.52	0	0.827	0.827	-1.421	-1.421	36.239	0	-0.040	-0.040

Table 16: Cornering simulation data for accelerating scene (falling rate).

Tables 12-16 are a culmination of data obtained from the corner simulation exercise. Motion ratio is considerably lower than its rising rate counterpart which produces lower wheel rates and by having lower wheel rates will increase the displacement of the suspension system, and more importantly roll centre affecting the stability of that suspension system. Roll centre migration in the vertical direction ranges between 36.239-37.926mm or 1.687mm which when comparing to the displacement range of rising rate of 0.561mm is over three times the size in magnitude. This increase in displacement is also seen in the Y direction. Falling rate roll centre moves within a range of 34.432mm which is 7.954mm larger than rising rate. Due to the advantages previously discussed in particular by (Adams, 1992) it would be beneficial to prioritise rising rate over falling rate in the pursuit of achieving a more stable front end.

From the data shown in figure 47 the rising rate setup gives a more consistent roll centre height across the corner which in turn produces a more stable front end of the vehicle.

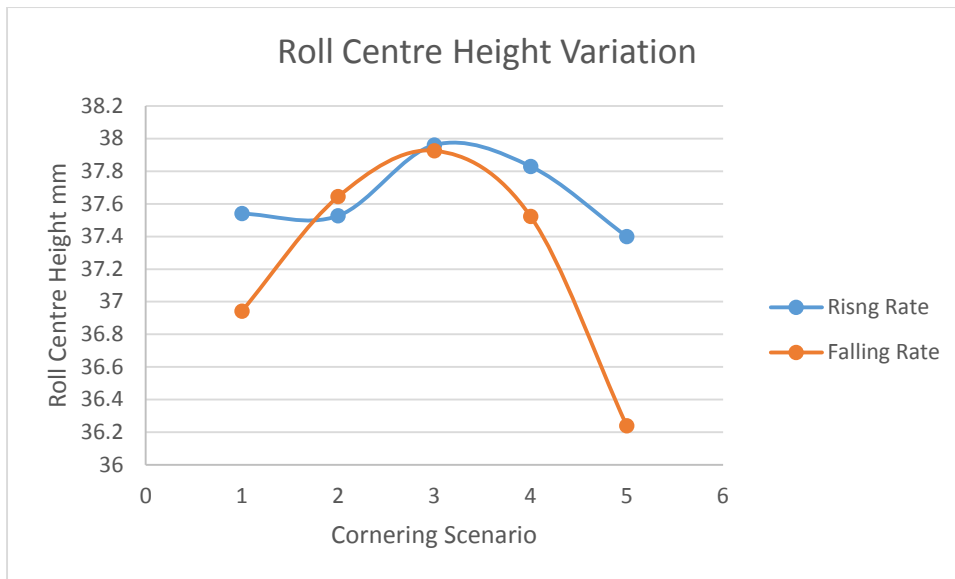


Figure 47: Graph showing roll centre height change during corner simulation.

4.5 Wheels Loads & Handling Balance

4.5.1 Rising Rate Characteristics

By using the corner simulation data for both systems with regards to pitch and roll values, it became possible to calculate using this data in conjunction with the front wheel rates of both rising and falling suspension systems. With wheel rates known using the wheel displacement option on wingeo 3 individual wheel loads were determined and then placed into a Rowley load split calculator to determine how the difference in loads affects handling balance going through the corner.

Scenario	Wheel displacement mm (Left)	Wheel displacement mm (Right)
1. Braking	+3.76	+3.76
2. Braking/Cornering	+1.495	-1.495
3. Cornering	+3.003	-3.003
4. Cornering/Accelerating	1.495	-1.495
5. Accelerating	-3.76	-3.76

Table 17: Table showing wheel displacement for rising rate during cornering simulation.

From here using the front wheel rate of 158906.056 n/m (158.906 n/mm) wheel loads were calculated which can be observed in table 17.

Scenario	Wheel Load N (Left)	Wheel Load (Right)
1. Braking	+597.486	+597.486
2. Braking/Cornering	+237.564	-237.564
3. Cornering	+477.195	-477.195
4. Cornering/Accelerating	+237.564	-237.564
5. Accelerating	-597.486	-597.486

Table 18: Table showing wheel loads for rising rate during cornering simulation.

These results were validated when comparing results obtained via the weight transfer formula. Using braking for example weight transfer is 597.56 N when compared to the value given in table 17 gives an accuracy of 99.98%. Calculations can be viewed in the appendix.

Right Turn					38.6%				
					LF+RF				
					1830.5				
			38.2%	LF		RF	0.4%		
Front Split	LF-RF	1795.0	1812.8				17.8		
					4743.1				
	LF+LR	87.2%	4133.892				609.2	12.8%	RF+RR
	(LR-RR) - (LF-RF)	-65.3	48.9%				12.5%		
Rear Split	LR-RR	1729.7	2321.1	LR		RR	591.5		
	LR-LF Split	508.4			2912.6		573.7	RR-RF Split	
	Wedge	49.3%			LR+RR				
	LR-RF Split	2303.3			61.4%				

Figure 48: Load split calculator data for rising rate system in right turn scenario (scenario 3).

Figure 48 shows the wheel load characteristic for scenario 3 (cornering). The individual wheel loads shown were evaluated using data in table 19. The wedge is 49.3% which for a right hand corner has the tendency to understeer than oversteer (less wedge = more understeer).

4.5.2 Falling Rate Characteristics

Scenario	Wheel Displacement mm (Left)	Wheel Displacement mm (Right)
1. Braking	+6.52	+6.52
2. Braking/Cornering	+1.948	-1.948
3. Cornering	+3.896	-3.896
4. Cornering/Accelerating	+1.948	-1.948
5. Accelerating	-6.52	-6.52

Table 19: Table showing wheel displacement for falling rate during cornering simulation.

As expected the displacements with falling rate are higher due to the lower wheel rates created due to lower levels of motion ratio. Wheel displacement in cornering 0.893mm higher on falling rate than its rising rate counterpart.

Scenario	Wheel Load N (Left)	Wheel Load N (Right)
1. Braking	+636.886	+636.668
2. Braking/Cornering	+190.284	-190.284
3. Cornering	+380.569	-380.569
4. Cornering/Accelerating	+190.284	+190.284
5. Accelerating	-636.886	-636.886

Table 20: Table showing Wheel loads for falling rate during cornering simulation.

During pitch wheel loads are greater due to the decrease of wheel rate. During cornering however wheel loads are lower therefore lower levels of weight transfer are taking place which may create lower levels of understeer.

Figure 49 indicates that the handling balance for falling rate again has the tendency to oversteer due to the high wedge of 53.9%. When comparing that to rising rate for example there is 4.6% difference which sways the handling balance between understeer and oversteer. Preferably an ideal wedge characteristic would be 50% giving balance through both left and right hand corners with that in mind the preferred choice of system would be the rising rate system giving that the handling balance is within 0.7% of being neutral making it more versatile. Given that the RC97 drives on clockwise circuits and thus having more right hand corners it would be beneficial to use rising rate over falling rate should this conversion be implemented.

Right Turn					38.6%				
					LF+RF				
					1830.5				
			33.6%	LF		RF	4.9%		
Front Split	LF-RF	1361.1	1595.8				234.7		
	LF+LR	82.6%	3916.975		4743.1		826.2	17.4%	RF+RR
	(LR-RR) - (LF-RF)	368.5	48.9%				12.5%		
Rear Split	LR-RR	1729.7	2321.1	LR		RR	591.5		
	LR-LF Split	725.3			2912.6		356.8	RR-RF Split	
	Wedge	53.9%			LR+RR				
	LR-RF Split	2086.4			61.4%				

Figure 49: Load split calculator data for falling rate system in right turn scenario (Scenario 3).

4.6 Multiple Geometry Bell Crank Concept

4.6.1 Obtaining Bell Crank Data

Although the main objective of this project was to investigate the effects of using dual-shock front suspension system, a multiple geometry bell crank has been conjured to enable the team to conduct more testing in the hope of achieving better balance of setup and by doing so gaining performance.

Taking data from the original rear bell crank from the RC97 in the form of measurements and also weight so that the new bell crank can then be compared to its original counterpart. From these findings it was found that the original bell crank made from steel weighed in at 0.359kg as shown in figure 50.



Figure 50: Picture showing RC97 bell crank being weighed.

This in conjunction with bell crank calculations using measurement data and a formula to calculate motion ratio at the bell crank. As described by (www.viewfelt.com, 2015) motion ratio at the bell crank is given by:

$$MR = \frac{L1}{L2} * \cos\phi$$

Using this formula and the measurements of the rear bell crank via the CAD model the initial motion ratio was:

$$MR = \frac{93.1}{67.69} * \cos(43.25) = 1.002$$

This can be validated by assessing the rear motion ratio via wingo 3 which is given as 1.020 meaning both measurements give an accuracy of 98% the difference may be down to the accuracy of the measurements conducted using handheld tool but more importantly these were transferred to solidworks which allowed more accurate measurement of the angle between L1 and L2. This process was completed for every mounting point on the bell crank.

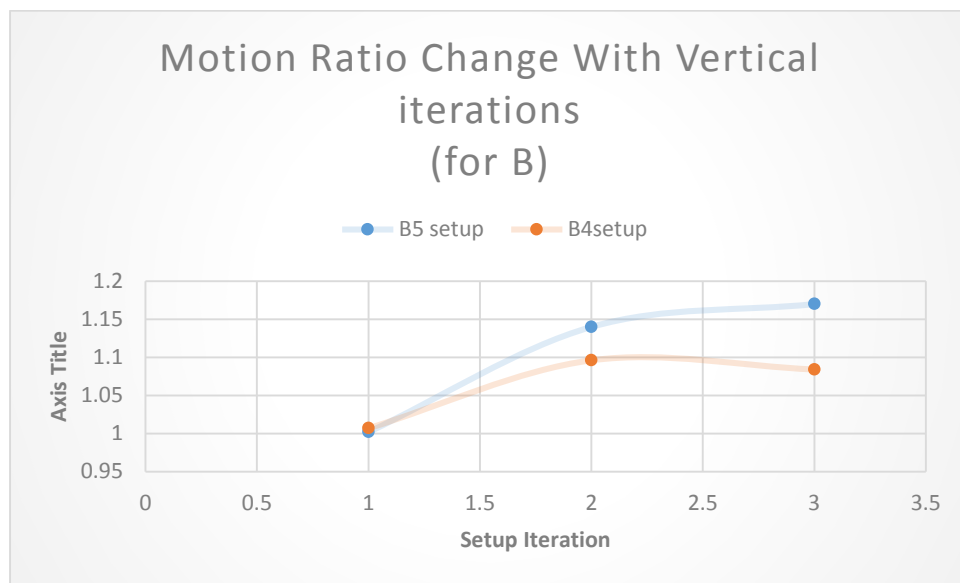


Figure 51: Graph showing motion ratios for different bell crank mounting points.

Figure 51 shows the motion ratio changes with regards to the alteration of pushrod/damper attachment point on the bell crank. The numbers B5 & B4 refer to the suspension mounting points and the corresponding 3 points shown on the graph are B1, B2 & B3. The B1-B5 marked in blue is the standard motion ratio taken from the rear as discussed in the previous chapter. As the pushrod is mounted further up the bell crank away from the pivot the motion ratio increases as predicted by (Rowley, 2008).

**** More data can be found of this within the appendix of this report. ****

4.6.2 FEA Analysis

FEA Analysis was conducted to ensure the components integrity via solidworks. The chosen material to be used for the concept is 7075 aluminium due to its high strength low weight properties as proposed by (Jawad & Baumann, 2002) who specify the use of this material to make strong but lightweight bell cranks and also its current availability as other vehicle components are currently being constructed using this material.

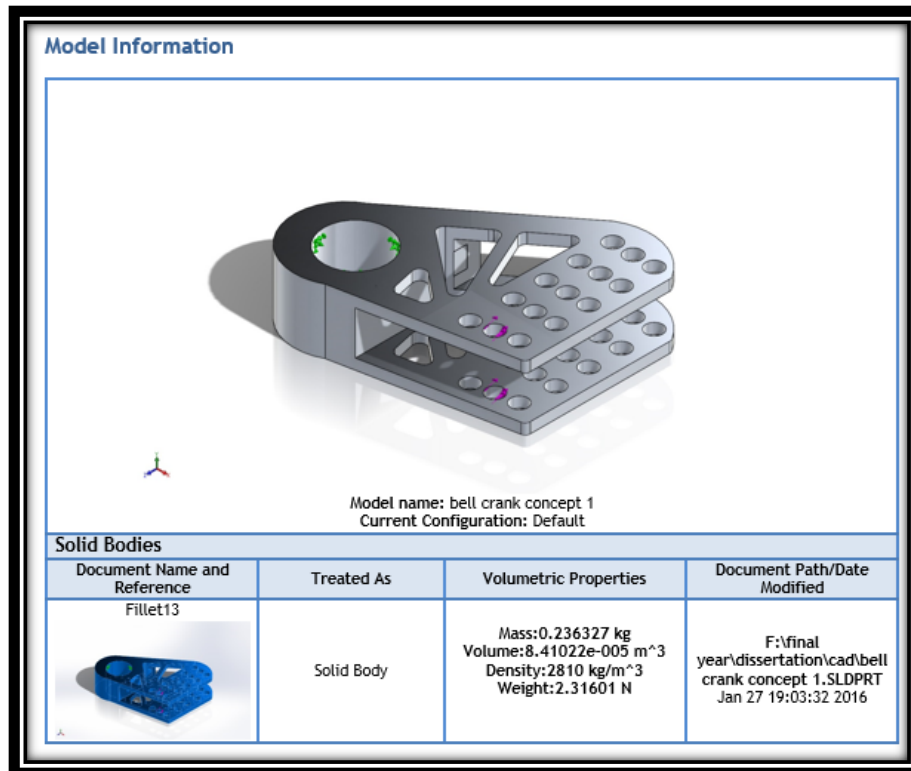


Figure 52: Screenshot of bell crank information.

According to the FEA report the mass of the bell crank is 236g which when compared to the standard rear bell cranks is approximately 123g lighter. The peak stresses in the component is 1.12705×10^8 N/m² given the components max yield strength of 5.05×10^8 N/m² gives a factor of safety of 4.48 over a maximum displacement of 0.155mm (as shown in figure 53). Triangles have been cut to lower weight and increase strength and the edges have been rounded off to minimize stress concentrations in the component.

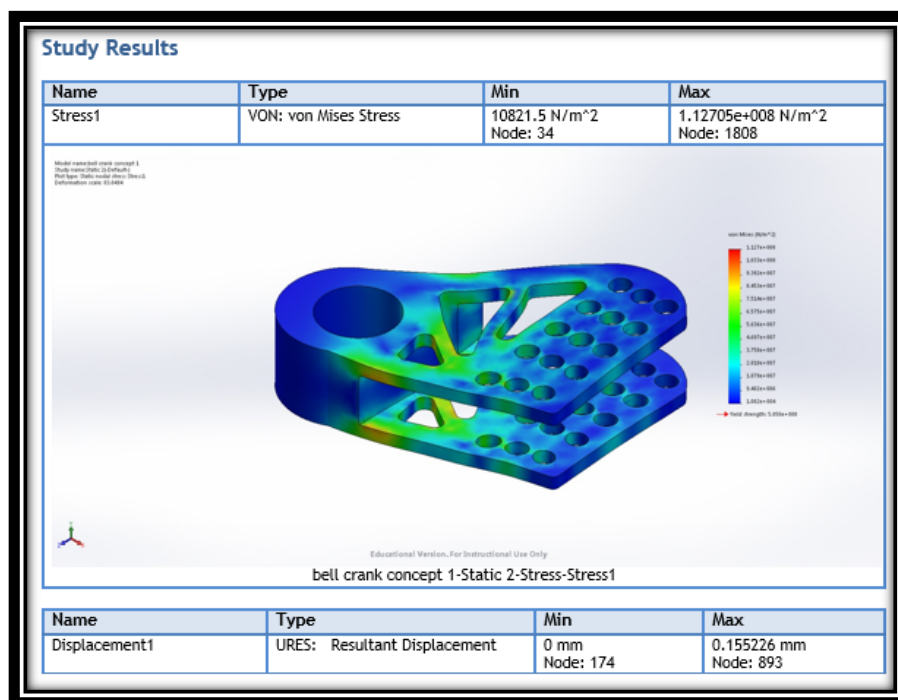


Figure 53: FEA study of bell crank concept results.

5. Conclusion

5.1 Project Findings

From the following results and the corresponding analysis has seen rising rate become the logical choice of dual shock suspension system for this conversion. Motion ratio has values higher than 1 in bump for the rising rate giving lower levels of roll centre migration which indicates a more stable suspension system. When comparing to falling rate the displacement is three times smaller, resulting in lower roll angle, pitch and wheel load distribution all have resulted from the higher wheel rates obtained due to higher motion ratio.

All these effects have resulted in a 49.3% wedge characteristic for a right hand corner for rising rate meaning the handling balance is swayed more to understeer but when comparing to that of falling which is 53.9% due to its decreased stability. Due to the RC97 competing on clock wise circuits a rising rate setup is proven to be more advantageous as this setup is only 0.7% from achieving a balanced setup rather than 3.9% to its falling rate comparison. These factors also confirm the statement of (Pashley, 2008) in that rising rate is the preferred choice of motion ratio characteristic.

The multiple geometry bell crank has been deigned shows increases in initial rate of motion ratio confirming the (Rowley, 2008) prediction. The use of 7075 aluminium has resulted in a weight decrease of 133g which could potentially result in the weight loss of unsprung mass at the rear of 266g lowering inertia characteristics of the bell crank mechanism resulting in higher suspension frequencies and thus greater accuracy of damping and improvement in handling.

To summarise the key findings of this report are:

- Rising rate has a roll centre migration three times lower than falling rate.
- Rising rate has a roll angle 0.072° lower than falling rate through a 1g corner.
- Rising rate has a pitch displacement 2.76mm lower than falling rate.
- Rising rate has a handling balance (wedge) 3.2% closer to 50% than falling rate indicating a more balanced handling characteristic.
- Bell crank concept is 133g lighter with a factor of safety of at least 4.48 lowering weight transfer at the rear by 0.69n. (calculations of this are in the appendix)

5.2 Future Work

As of this point the project and the findings have been purely theoretical using calculations and drawings to derive a dual-shock system and the effects these have regarding handling balance and cornering performance.

To continue obtaining more knowledge regarding this setup would require:

1. Live testing

Using a data logger and relevant sensors to assess factors such as wheel/spring displacement (motion ratio) along with roll and pitch movements which can then be compared to that of the results obtained within this report.

2. Anti-roll bar research

Although anti roll bars have been mentioned they have not been included in the calculations due to the main focus is the suspension system. Anti-roll bar data can be viewed within (Pugh, 2008) which may be used in initial calculations and then tested once the component has been machined.

3. Installation of different dampers

Due to the increase in motion ratio would result in the dampers being worked harder this increase in damping performance in particular the damper velocity may possibly be too excessive for the current dampers and may need the requirement to purchase dampers that can cope with the increase in damping velocities.

4. Simulator testing

Using the simulator it may be possible to see how the rising rate motion ratio at the front will affect downforce with regards to rake angle. By changing from rising to falling rate may affect the amount of overall downforce created as the front of the vehicle acts as a throat and the underneath of the vehicle acts as a venturi creating low pressure. Theoretically speaking by having a more stable front end would give a more consistent level of downforce. Data from this can then be compared to live testing data for any inconsistencies and the cause if any and by doing so acquiring more data which may be useful to the team.

5.3 Areas of Improvement

Throughout this project the main cause for concern was the accuracy of the measurements of obtained. Components like the bell crank for example were measured with great care and precision which proved to be very time costly but due to financial constraints was the logical choice for collecting data. Upon reflection of this project using electronic measuring such a faro arm may have been used to collect measurements more quickly and more accurately than by using conventional measuring tools.

Using a faro arm would give the possibility of taking measurements while simultaneously building a CAD model from those measurements making it a time efficient tool with regards to measuring and modelling. According to (www.faro.com, 2012) faro arms have been used by McLaren formula one team for the past 14 years in various manufacturing areas from aerodynamics to quality control.



Figure 54: Picture of a faro arm being used for a formula one application (www.faro.com, 2012)

(www.faro.com, 2012) also go on to inform that faro arms have also increased fidelity and accuracy besides the speed of the measuring process. Although a costly piece of equipment to have it would

have proven beneficial with regards to efficiency in the measuring and modelling process due to the modelling process taking up 7 weeks of the overall project timeline as shown in figure 55.

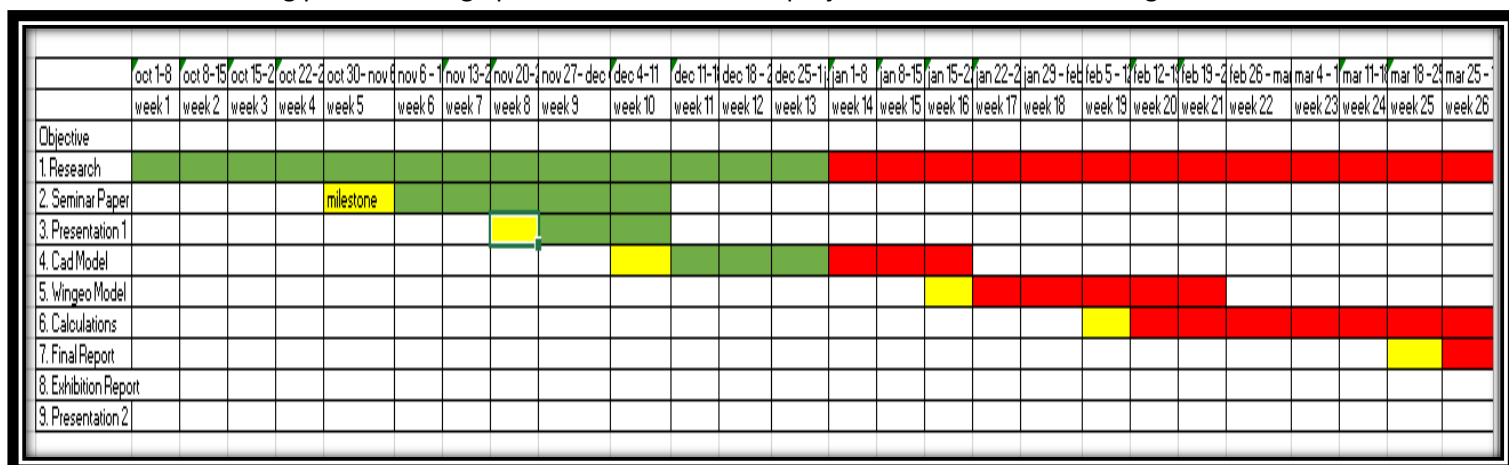


Figure 55: screenshot showing project gantt chart.

*** A full Gantt chart can be viewed within the appendix of this report***

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Appendix

1. Initial Wingeo data (rising rate)

zero ride height	roll angle	Motion Ratio left	motion ratio right	Spring disp left mm	Spring disp right	camber left	camber right	Roll centre Z (mm)	Roll centre y (mm)
	2	1.976	0.773	-35.073	22.654	1.31	-1.345	34.367	-220.87
	1.8	1.781	0.795	-30.36	20.687	1.181	-1.209	35.138	-198.904
	1.6	1.634	0.817	-26.073	18.665	1.051	-1.073	35.74	-176.899
	1.4	1.517	0.841	-22.115	16.584	0.921	-0.938	36.271	-154.861
	1.2	1.422	0.867	-18.422	14.441	0.791	-0.803	36.732	-132.793
	1	1.341	0.894	-14.951	12.231	0.66	-0.668	37.123	-110.7
	0.8	1.271	0.923	-11.668	9.949	0.529	-0.534	37.443	-88.585
	0.6	1.211	0.954	-8.549	7.592	0.397	-0.4	37.691	-66.454
	0.4	1.157	0.988	-5.575	5.152	0.265	-0.266	37.869	-44.31
	0.2	1.108	1.025	-2.73	2.624	0.133	-0.133	37.976	-22.157
	0	1.065	1.065	0	0	0	0	38.012	0

with 1 degree roll									
ride height mm	Motion Ratio left	Motion Ratio Right	spring disp left mm	spring disp right	camber left	camber right	Roll centre z (mm)	Roll centre y	
25	0.898	0.683	12.137	31.708	1.329	-0.03	74.41	-56.978	
20	0.957	0.717	7.503	28.21	1.198	-0.156	66.963	-63.061	
15	1.027	0.754	2.546	24.553	1.065	-0.282	59.518	-70.622	
10	1.11	0.796	-2.79	20.66	0.931	-0.41	52.07	-80.277	
5	1.212	0.842	-8.854	16.568	0.796	-0.539	44.61	-93.038	
0	1.341	0.894	-14.951	12.231	0.66	-0.668	37.123	-110.7	
-5	1.516	0.954	-22.067	7.614	0.522	-0.799	29.581	-136.758	
-10	1.776	1.023	-30.244	2.617	0.383	-0.932	21.92	-179.083	
-15	2.244	1.106	-40.148	-2.64	0.242	-1.065	13.963	-259.851	
-20	3.759	1.207	-54.097	-8.413	0.1	-1.2	4.942	-475.104	

2. Wingeo initial data (falling rate)

zero ride height	Roll angle	Motion ratio left	motion ratio right	Spring displacement left	spring displacement right	camber left	camber right	RC height (z)	RC lat (Y)
	2	0.665	6.822	-18.49	33.607	1.31	-1.345	34.467	-220.87
	1.8	0.676	2.076	-16.802	25.761	1.181	-1.209	35.138	-198.904
	1.6	0.687	1.559	-15.087	21.303	1.051	-1.073	35.74	-176.899
	1.4	0.699	1.322	-13.343	17.716	0.921	-0.938	36.271	-154.861
	1.2	0.713	1.178	-11.567	14.59	0.791	-0.803	36.732	-132.793
	1	0.728	1.08	-9.755	11.762	0.66	-0.668	37.123	-110.7
	0.8	0.745	1.007	-7.904	9.146	0.529	-0.534	37.443	-88.585
	0.6	0.763	0.95	-6.008	6.69	0.397	-0.4	37.691	-66.454
	0.4	0.784	0.905	-4.064	4.362	0.265	-0.266	37.869	-44.31
	0.2	0.808	0.867	-2.064	2.138	0.133	-0.133	37.976	-22.157
	0	0.835	0.835	0	0	0	0	38.012	0

25mm ride height										
1 degree of roll	roll angle	ride height (mm)	motion ratio left	motion ratio right	spring disp left (mm)	spring disp right (mm)	Camber left	Camber right	RC height (z)	RC lat (y)
	1	10	0.808	2.066	-2.108	25.689	0.931	-0.41	52.07	-80.277
	1	5	0.763	1.321	-6.03	17.69	0.796	-0.539	44.61	-93.039
	1	0	0.728	1.08	-9.755	11.762	0.66	-0.668	37.123	-110.7
	1	-5	0.699	0.95	-13.321	6.713	0.522	-0.799	29.581	-136.758
	1	-10	0.676	0.867	-16.758	2.182	0.383	-0.932	21.92	-179.083

2. Bell crank concept motion ratio variation data.

Formula: $MR = \text{length A} / \text{Length B} * \cos \theta$

$$A1-A5 = (86.1/57.87) * \cos (49.94) = 0.965$$

$$A2-A5 = (86.1/61.36) * \cos (31.75) = 1.193$$

$$A3-A5 = (86.1/66.99) * \cos (18.81) = 1.217$$

$$A1-A4 = (75.45/57.87) * \cos (41.66) = 0.974$$

$$A2-A4 = (75.45/61.36) * \cos (23.47) = 1.128$$

$$A3-A4 = (75.45/66.99) * \cos (10.53) = 1.107$$

$$B1-B5 = 1.002 \text{ (standard setup)}$$

$$B2-B5 = (93.1/71.07) * \cos (29.23) = 1.14$$

$$B3-B5 = (93.1/75.99) * \cos (17.73) = 1.17$$

$$B1-B4 = (83.55/67.69) * \cos (35.29) = 1.007$$

$$B2-B4 = (83.55/71.07) * \cos (21.27) = 1.096$$

$$B3-B4 = (83.55/75.99) * \cos (9.77) = 1.084$$

$$C1-C5 = (100.77/77.55) * \cos (38.05) = 1.023$$

$$C2-C5 = (100.77/80.86) * \cos (26.98) = 1.111$$

$$C3-C5 = (100.77/85.22) * \cos (16.68) = 1.133$$

$$C1-C4 = (92.02/77.55) * \cos (30.41) = 1.023$$

$$C2-C4 = (92.02/80.86) * \cos (19.36) = 1.074$$

$$C3-C4 = (92.02/85.22) * \cos (9.04) = 1.066$$

5. Weight transfer calculations

For bell cranks

Standard bell cranks

$$WT = \frac{296.9 * 9.81 * 0.32}{1.2} = 776.69 \text{ N}$$

Concept bell cranks

$$WT = \frac{(296.9 - 0.266) * 9.81 * 0.32}{1.2} = 775.99 \text{ N}$$

Difference = 0.7

For cornering sim

$$WT = \frac{483.5 * 9.81 * 0.32}{2.54} = 597.56 \text{ N}$$

Using wheel rates

$$WT = \text{wheel rate} * \text{displacement} = 158.906 * 3.76 = 597.48$$

=98% accurate

6. FEA report for bell crank concept.



Description

No Data

Simulation of bell crank concept 1

Date: 28 January 2016

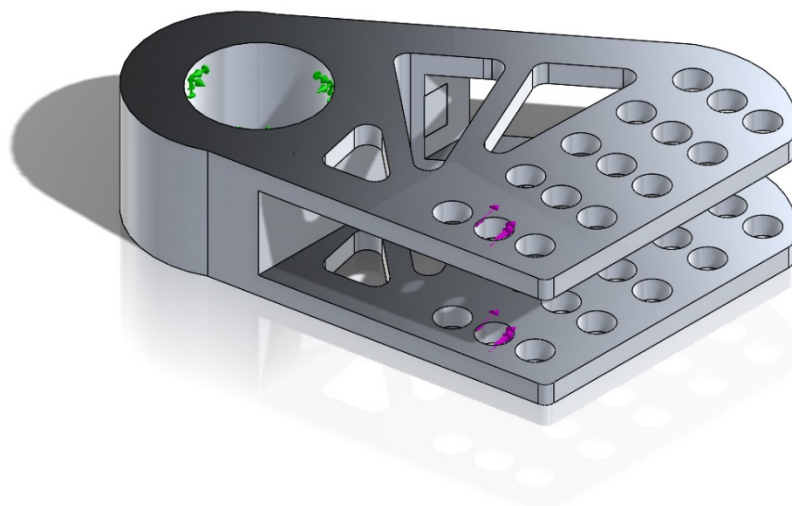
Designer: James Cooze

Study name: Static 2

Analysis type: Static

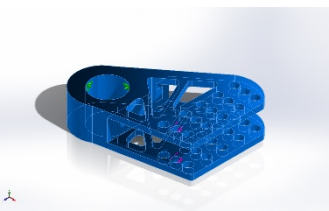
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Model name: bell crank concept 1
Current Configuration: Default

Solid Bodies

Document Name and Reference	Treated As	Volumetric Properties	Document Path/Date Modified
Fillet13 	Solid Body	Mass:0.236327 kg Volume:8.41022e-005 m ³ Density:2810 kg/m ³ Weight:2.31601 N	F:\final year\dissertation\cad\bell crank concept 1.SLDPRT Jan 27 19:03:32 2016

Assumptions

Model Information

Study Properties

Study name	Static 2
Analysis type	Static
Mesh type	Solid Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SOLIDWORKS Flow Simulation	Off
Solver type	FFEPlus
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Friction	Off
Use Adaptive Method:	Off
Result folder	SOLIDWORKS document (F:\final year\dissertation\cad)

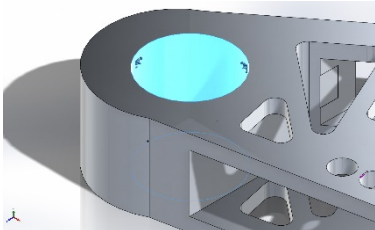
Units

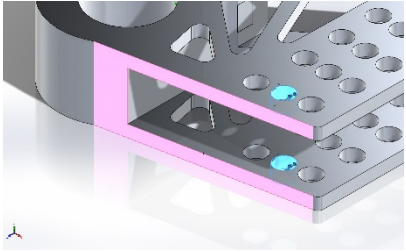
Unit system:	SI (MKS)
Length/Displacement	mm
Temperature	Kelvin
Angular velocity	Rad/sec
Pressure/Stress	N/m ²

Material Properties

Model Reference	Properties	Components
	Name: 7075-T6 (SN) Model type: Linear Elastic Isotropic Default failure criterion: Max von Mises Stress Yield strength: 5.05e+008 N/m^2 Tensile strength: 5.7e+008 N/m^2 Elastic modulus: 7.2e+010 N/m^2 Poisson's ratio: 0.33 Mass density: 2810 kg/m^3 Shear modulus: 2.69e+010 N/m^2 Thermal expansion coefficient: 2.36e-005 /Kelvin	SolidBody 1(Fillet13)(bell crank concept 1)
Curve Data:N/A		

Loads and Fixtures

Fixture name	Fixture Image	Fixture Details		
Fixed-1		Entities: 1 face(s) Type: Fixed Geometry		
Resultant Forces				
Components	X	Y	Z	Resultant
Reaction force(N)	746.024	-0.421306	3769.63	3842.74
Reaction Moment(N.m)	0	0	0	0

Load name	Load Image	Load Details
Force-1		Entities: 2 face(s) Reference: Face< 1 > Type: Apply force Values: ---, ---, -1921.23 N

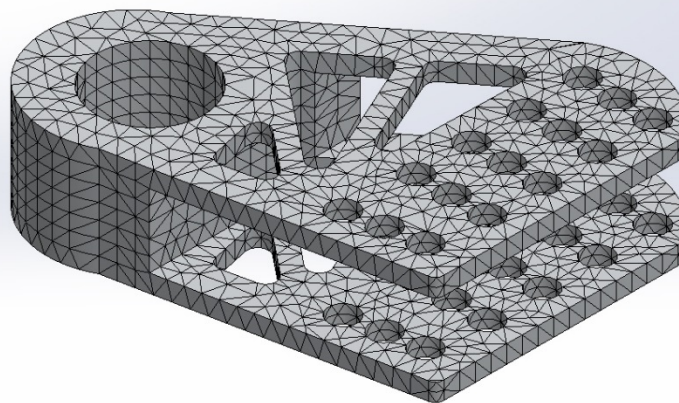
Mesh information

Mesh type	Solid Mesh
Mesher Used:	Standard mesh
Automatic Transition:	Off
Include Mesh Auto Loops:	Off
Jacobian points	4 Points
Element Size	4.38267 mm
Tolerance	0.219133 mm
Mesh Quality	High

Mesh information - Details

Total Nodes	19773
Total Elements	10864
Maximum Aspect Ratio	32.173
% of elements with Aspect Ratio < 3	97.6
% of elements with Aspect Ratio > 10	0.092
% of distorted elements(Jacobian)	0
Time to complete mesh(hh:mm:ss):	00:00:05
Computer name:	FAD-MT302-16

Model name: bell crank concept.1
Study name: Static 2(Default)
Mesh type: Solid Mesh



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Resultant Forces

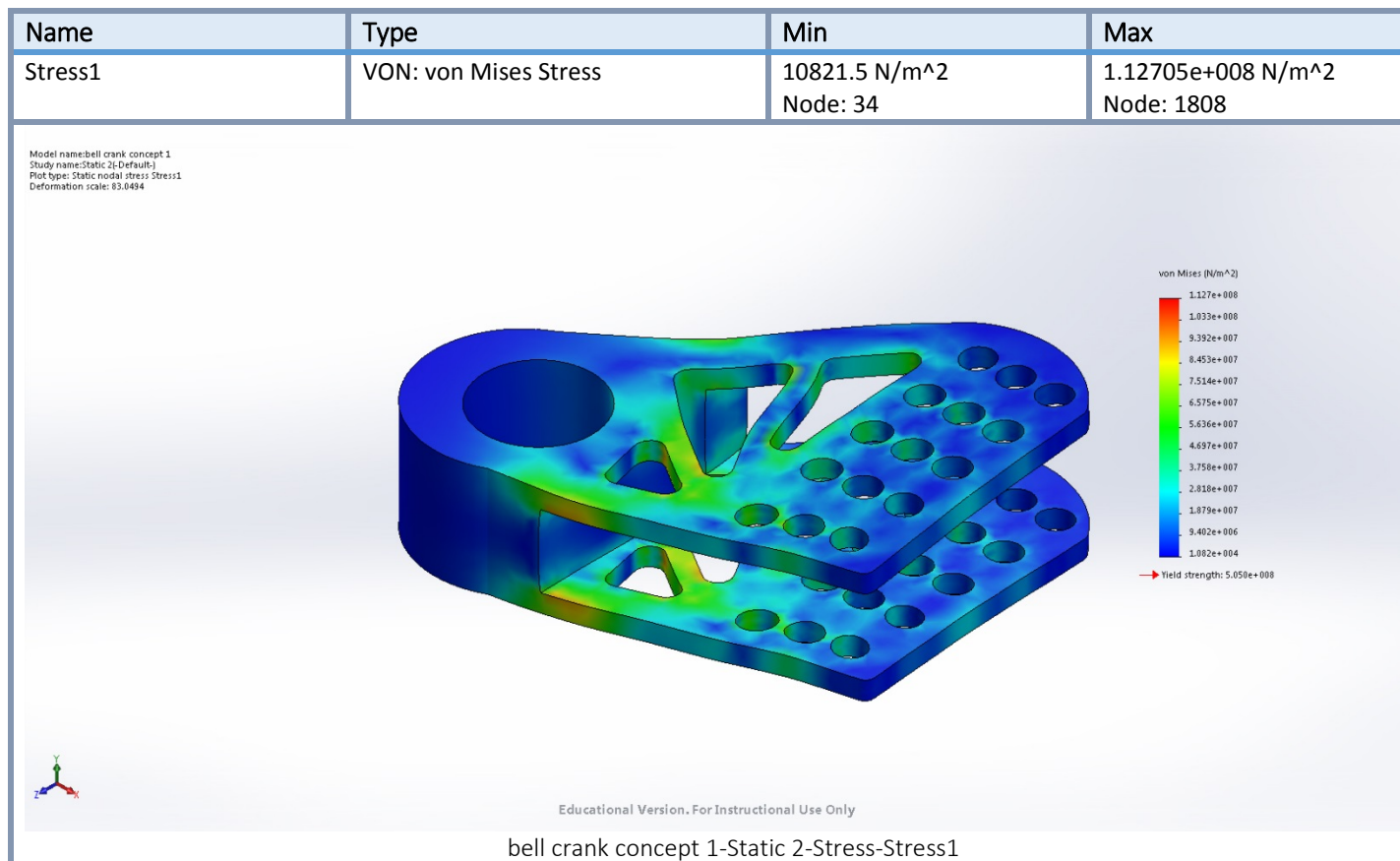
Reaction forces

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N	746.024	-0.421306	3769.63	3842.74

Reaction Moments

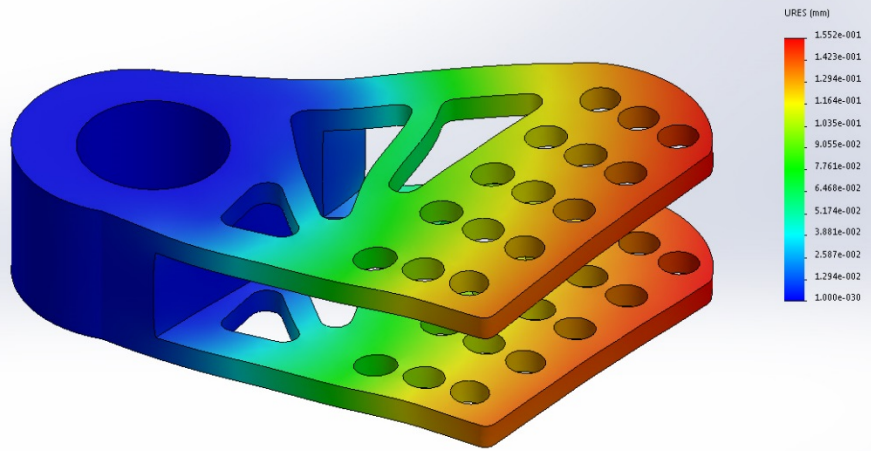
Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N.m	0	0	0	0

Study Results



Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 174	0.155226 mm Node: 893

Model name: bell crank concept 1
 Study name: Static 2j-Default-j
 Plot type: Static displacement Displacement1
 Deformation scale: 63.6494

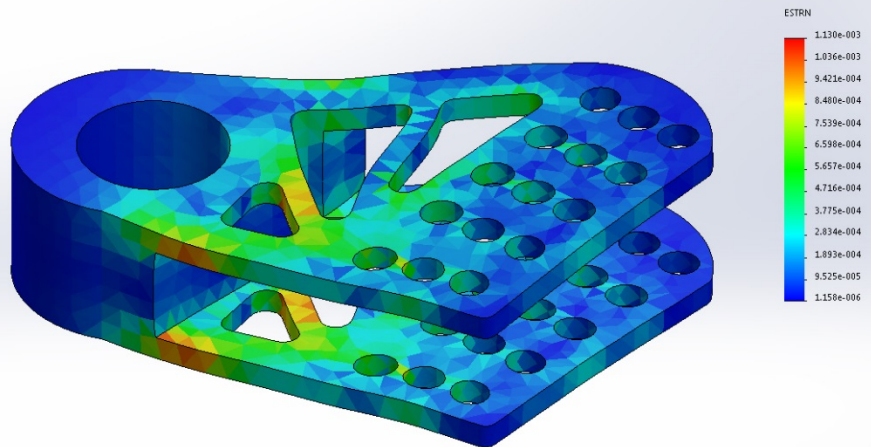


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bell crank concept 1-Static 2-Displacement-Displacement1

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	1.15847e-006 Element: 6643	0.00113028 Element: 5527

Model name: bell crank concept 1
 Study name: Static 2j-Default-j
 Plot type: Static strain Strain1
 Deformation scale: 63.6494

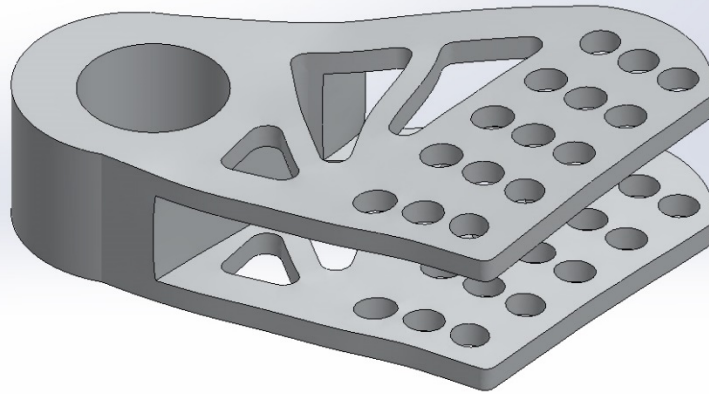


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bell crank concept 1-Static 2-Strain-Strain1

Name	Type
Displacement1{1}	Deformed shape

Model name: bell crank concept 1
Study name: Static 2-Default-1
Plot type: Deformed shape Displacement1{1}
Deformation scale: 83.6494



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bell crank concept 1-Static 2-Displacement-Displacement1{1}

Conclusion

