Designing of a Rear Suspension for a Race Car

Greg Wheatley, Brent Lane, Rendage Sachini Sandeepa Chandrasiri

***Abstract*- This paper was commissioned for the design and analysis of an entire rear suspension system befitting Formula Society of Automotive Engineers (FSAE) vehicle. The paper includes a literature review to gain a full understanding of the workings and design decisions applied to rear suspension in the Society of Automotive Engineers (SAE) competition. After completing the design development process, final analysis of the designed system was done to ensure the minimum two years of life requirement is met.**

It was found that due to constraints, a major design change was necessary which involves mounting the A-arms further forward on the chassis body than previous generation vehicles. This design increased the stresses present in the system compared to previous designs. As such, careful consideration had been given to the analysis aspect of the paper.

Full fatigue analysis performed individually on each component proved that the lower A-arm was the most critical component, with a predicted failure at 1466 laps. However with the given lifespan of two years, this designed procured a conservative factor of safety of above two years.

Notable mention was given to the complete development of an FSAE uniaxial force determination code. This code greatly improved the confidence in component forces and thus allowed less conservative design choices in several other aspects.

***Index Terms* - Formula Society of Automotive Engineers, Vehicles, Chassis body, Fatigue analysis**

I. INTRODUCTION

In general the purpose of a formula SAE suspension system is to increase the vehicle's performance and handling during a race [1].The suspension is utilised to ensure that all wheels remain in contact with the ground at all times during the competition. There are two key components essential to make a suspension system. First key component is the shock absorber which includes both spring and damper and second is the structural members, used to mount the shock absorber which directly joins the chassis to the uprights [2]. These structural members include; A-arms, push/pull rod, rockers, control arms and antiroll bars. Using all of these in combination results a full rear suspension system.

This arrangement of structural members allows the shock absorbers to absorb energy from the wheels so the chassis does not take the full impact of the force. The shock includes a spring and a damper which together works to absorb and then dissipate the energy created from wheel vibration at a certain rate.

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Greg Wheatley is with the College of Science and Engineering, James Cook University, Townsville, Australia (phone: +61747814137; e-mail: greg.wheatley@jcu.edu.au)

Brent Lane is with the College of Science and Engineering, James Cook University, Townsville, Australia

Rendage Sachini Sandeepa Chandrasiri is with the Department of Chemistry, University of Colombo, Colombo, Sri Lanka (e-mail: sachini.chandrasiri7@gmail.com)

The structural geometry is proportional to the number of variables that affects the vehicles performance and handling. The major objective of the literature review is to research the effect of various design parameters on the effectiveness and functionality of the rear suspension in a FSAE vehicle [3]. These findings in conjunction with a complete design audit will be analysed in the hopes of improving James cook University's (JCU) current design in preparation for FSAE vehicle competition.

The design modifications are mainly focused on possible areas of weight reduction in the overall rear suspension assembly. Another design objective is to find a method of mounting the A-arms further forward on the chassis, without compromising adjustability and allowing less restriction on the design of the differential mounting [4]. Finally, based on the literature review, it is aimed to investigate the plausibility of implementing an adjustable A-arm design that helps to improve the competitiveness in a range of events. It should be noted that other components of the current competition vehicle are being concurrently audited and all results determined are available for analysis and comparison to aid in the development of the next generation SAE vehicle for the upcoming competition.

The literature review of FSAE rear suspension is mainly targeted on particular optimisation of components and the analysis techniques utilised to verify the functionality of the designs. Additionally, it also includes current standards that are supplied by the FSAE rulebook and any additional Australian Engineering standards [5].

II. LITERATURE REVIEW

Suspension design and tuning is commonly used in SAE motorsports teams to improve both the performance and handling of the vehicle and make for a competitive team. This literature review aims to analyse the effect of various design parameters on the performance capabilities of the rear suspension. Various analysis techniques used to verify these results will also be reviewed to determine the effectiveness of analytical methods as a verification technique.

The review will firstly expand and give a more detailed overview of rear suspension components and the role they play in a FSAE vehicle. Relative standards are then discussed, to ensure that all researched designs are relevant to the current standards of the competition. A review of current designs will then be discussed, with any similarities and differences commented on, in particular, the effectiveness of each design with respect to this review's objectives of; weight reduction and forward A-arm mounting techniques. Finally, an overview of varied analysis techniques will be explored and compared with benchmarked results to effectively analyse their effectiveness at modelling FSAE suspension [6].

A suspension system design essentially depends upon the vehicle's structure and purpose. Meaning each vehicle's suspension design is custom to accommodate for its specific needs. Although each suspension system is slightly different, the process of planning designing and analysing the suspension system is fairly similar. To create an effective design, a large number of articles and pre-existing designs must be studied to gain the knowledge required to improve James Cook University Tec-NQ Racing (JTR) current FSAE suspension design. The advantages and disadvantages taken from each article will essentially reduce work time and improve future design decision making. Major design factors that will be analysed include mounting positions of components, geometry and material selection.

University of Western Australia's Renewable Energy Vehicle (REV) had one such advantageous design. The design proposal was to mount the shocks on the same chassis node to reduce the bending force caused by the shocks [7]. This design includes a push rod-rocker-shock configuration which allows flexibility in positioning the shocks [8].

The force induced on the shock in Popa's design causes bending to the lower member which weakens the structure [9]. As aforementioned the Renewable Energy Vehicle (REV) team's proposal eliminates any unnecessary bending, through mounting at a singular node. Both above designs have a relative low suspension design when compared to other existing literature. By lowering the suspension system the centre of gravity as a result becomes closer to the ground. As a consequence the moment about the roll centre of the car is reduced, which reduces the roll angle and allows faster cornering.

One fundamental part of University of Western Australia's Renewable Energy Vehicle (REV) team applied to the design was to increase the angle between the A-arm members. Increasing this angle reduces the longitude compression force on the A-arms. Since less force is going through the A-arms, less material is needed to manufacture the A-arms which reduce the weight and optimise the design change in force distribution when altering the geometry in particular the angle of the A-arm configuration.

When planning the suspension system design it is important to consider stress concentrations. The bolt thread through the member creates a stress concentration that weakens the structure. In comparison to the 2008 University of Southern Queensland (USQ)'s FSAE proposal shows the A-arms mounted directly to the kingpin through use of heim joints to reduce any aforementioned stress concentrations. In conclusion, the A-arm members should be design with minimum stress concentration and a solid mounting point.

The USQ's FSAE team ran into some troubles in the FSAE 2007 competition when the race car crashed into the barrier due to inappropriate scrub radius and a sub-optimal steering system. This flaw demonstrates the importance of correct suspension parameters in the handling ability of the car. The scrub radius primarily affects the steering and braking performance of the car. The radius size is determined by a number of factors that should be configured simultaneously. Such factors include the tyre size, camber and the pushrod horizontal angle. Therefore, the scrub radius should be considered and analysed before designing and manufacturing the new FSAE JTR vehicle.

Florida International University (FIU) had an interesting approach to determine which ratio of A-arm's length will maximise tyre performance. A suitable A-arm ratio will maximise the tyre traction with the road for a wide range of the vehicles vertical travel [10]. To avoid manufacturing a number of different A-arms, the FIU team proposed an adjustable top mount which will allow changing the length of the top A-arm.

Generally having a slight negative camber is good practice if the given track includes many sharp corners. The most suitable curve is chosen once the track properties are known since each curve will yield different performance. This highlights the importance of an easily adjustable system for a competitive system in the FSAE competition. Adjustable A-arms allow for a vehicle to have advantages in each event, not just one.

The first and most important consideration taken when selecting the suspension system material is its strength to weight ratio. Aluminium, composites and carbon tubes are examples of such materials that possess a good ratio. However, these materials are generally far more expensive and more difficult to process [11]. As the weight ratio of mild steel fairly good compared to Aluminium composites and carbon tubes. Chromalloy and steel are easy to handle and relatively cheap. The FSAE reports reviewed for the literature review either had Chromalloy or mild steel suspension system members. A comparison of several possible material selections can be seen below in Table I.

Table I: Advantages and disadvantages of different materials

Material	Strength per unit weight (kNm/kg)	Advantages	Disadvantages
Mild Steel	> 32	Baseline material requiring no additional design Easy to weld Good workability	Mild steel tube not readily available locally in small quantities
AISI 4130 (Alloy Steel)	> 50	High strength Easy to weld Can be sourced for a reasonable price	Requires interstate delivery Material weakens when welded FSAE rules state minimum tube size [12]
Composite	>75	Very high strength to weight ratio	Requires proof build quality Very expensive Needs monocoque designs Requires mechanical fastening to main hoop
Aluminium	> 60	Good strength to weight ratio High workability	Requires Mechanical fastening to the main hoops Best used in monocoque designs Difficult to source locally

This literature review is aimed to find methods, through analysis of both current and existing rear suspension designs, of weight reduction to improve performance of

JTR's next competition vehicle. Major contributions to this were found in using a dynamic loading history, rather than a 'worst case scenario' history, as this confidence in forces can be used to both determine the life of each component, but additionally can be used to reduce the desired factor of safety as more information is known about the system.

Furthermore, through the analysis and comparison of previously existing designs, it can be concluded that for an effective future design, the JTR team should give large consideration to the geometry of each component and any consequences these ensue. Material selection must also be evaluated in conjunction with the team's future goals.

Through the analysis of all reports and past designs as outlined in this review, in conjunction with the relevant rules, it is aimed to use the information to make improvements to the JTR FSAE vehicle for the upcoming competition. In particular a weight reduction of 5% is desired for the next generation suspension. This is intended to be achieved through the use of a less conservative load case analysis, as detailed in load cases. Furthermore the A-arm design is proposed to be mounted further forward on the chassis in an asymmetrical design as to allow more freedom of design for differential mounts, in turn resulting in a larger possible drive sprocket. Finally, an adjustable system is aimed to be implemented to increase JTR's competitiveness in multiple events for the next competition.

III. DESIGN DEVELOPMENT

As a basis for the design development for the JTR rear suspension system, previous designs both JTR based and externally based were investigated. However given the removal of the rear of the chassis, further investigation into approaches taken by other SAE teams were conducted, including moving the wheels forward, or sweeping the A-arms back. As such, when developing the initial concepts for the JTR team large considerations must be taken into account of the strengths and weaknesses of all reviewed designs, in order to select a style which best meets the needs of the JTR team.

The development of the final design took an iterative approach, with concepts being created and refined based on constraints, design goals and benefits or weaknesses of each by using three major design concepts for the rear suspension design with its associated explanation.

The rear sweeping A-arm concept 1, allows fewer restrictions on the rear drive shaft as the entire rear box section of the chassis is removed. This not only decreases the overall weight of the vehicle, but allows the inclusion of a higher ratio drive sprocket on the design. It should also be noted that given this design, other design team's including rear uprights do not need to make any adjustments to their design to accommodate the rearward A-arms.

Although greatly increasing freedom on other components, quick calculations using the MATLAB code developed in the design audit proved that this design greatly increases the axial forces through each member when compared to the current design. As such it may be a requirement to strengthen the components via increasing tubular wall thickness.

The neutral A-arm concept 2 was designed as to decrease the larger axial forces that concept 1 would be expected to face. Having a neutral rear member, decreases this stress, and makes for an overall more compact and lighter design. Having shorter A-arm members also reduces the risk of bending and/or buckling if contact occurs. Additionally this design decreases the wheel base of the vehicle, allowing for better handling.

To implement this design however, a conjunction of either forward pick up points for the A-arms on the upright or greater half shaft angles must be implemented.

The larger box section concept 3, is an adaptation to the existing design. Given that a design criterion as specified by JTR was the inclusion of a larger drive sprocket, this concept maintains the rear box section but increases its dimensions as to fit this new drivetrain.

Although providing less stress in the A-arms and easier manufacturing and mounting of the rear suspension, this design would only increase weight to the vehicle, making it less competitive. Taking each of the above concepts, a critical pro's and con's list was developed as seen below Table II. This was created through both team discussion and third party opinions.

Table II: Advantages and disadvantages of each design concept

Concept	Advantages	Disadvantages
1	Decreased Overall Weight No major design changes necessary for other components	Larger axial forces through A-arm members Greater pushrod angle
2	Easier Packaging Overall weight reduction Only slight increase in stress	Requires redesign of uprights Puts more angle on CV joints
3	Easily manufacturable Lower A-arm stress	No weight reduction Possible restrictive for future designs

From the above three final concept designs, an effective method of distinguishing the most appropriate for implementation on the next generation vehicle had to be established. One such method is the use of a weighted decision matrix. Table III below shows the matrix developed for the rear suspension, including both design factors and their associated weightings. It should be noted that as per the design goals, larger weightings were given to weight reduction and adjustability, with load bearing capabilities deeming the largest weighting as an incentive for the chosen design to easily meet the desired life requirements.

Table III: Design Decision Matrix

Design Concept	Manufacturability	Cost	Efficiency	Adjustability	Weight reduction	Packaging	Load capacity
Weight	7	6	8	8	7	9	
1	7	5	7	7	5	7	289
2	7	8	8	8	7	8	292
3	8	6	5	2	8	9	288

According to the results of Table III the most appropriate design concept is design 2. However upon consultation with the rear upright design group, it was determined that rim clearance with the upright was insufficient in this design, due to constraints imposed on the upright team as a result of new hubs being purchased for the JTR team

As a result, a compromise was made to implement design 1 for the rear suspension, with slightly forward mounted pick-ups on the uprights be designed as to reduce the angle imposed on the A-arms.

Before the initial analysis of the system, small refinements to the concept were made to optimise the design including determination of the best forward a-arm member angles. Given that the differential and tyre were not to be moved, there was no freedom given to the design of the rear members, however the front members could be attached at any point on the chassis.

As such, the maximum angle before fouling occurred with the rim was determined to be 62.5^0. From here the following optimisation was made using the MATLAB code that was implemented in the design audit. It was found that to minimise the uniaxial forces through each member the angle should be made as large as possible, as such an angle of 60^0 to the perpendicular was chosen.

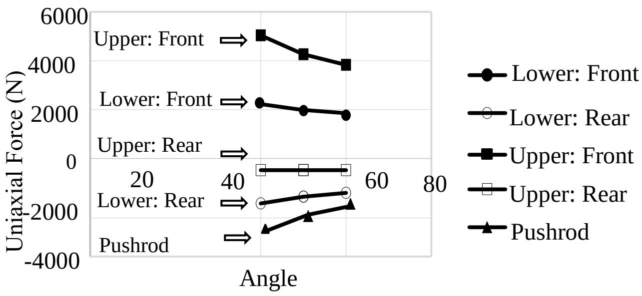

Fig. 1. Optimal angel of A-arm for right turn

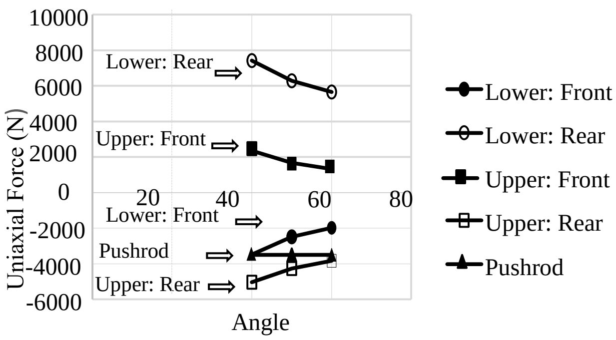

Fig. 2. Optimal angle of A-arm for left turn

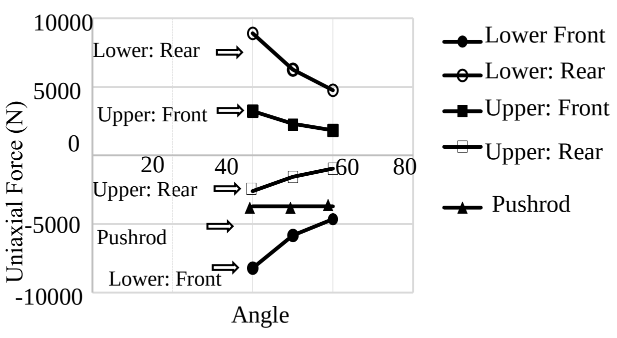

Fig. 3. Optimum angle of A-arm for accelerating

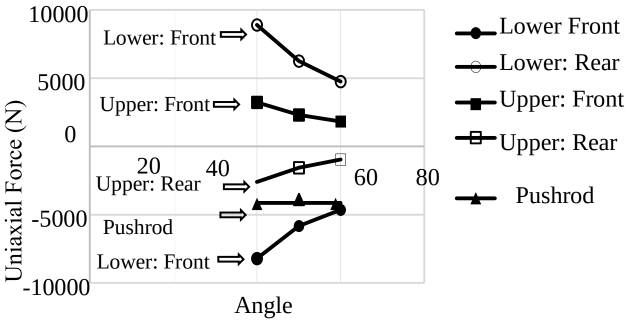

Fig.4. Optimum angle of A-arm for breaking

IV. ANALYSIS

When performing an analysis on any system, choosing correct and life like load cases is imperative in obtaining accurate analysis results. Load cases must be chosen such that they mimic the real-world applications of a system. With regard to this design project, load cases had to be developed that simulated what loads a rear suspension system would undergo during FSAE events [13]. After consultation with the JTR team and research conducted into track racing the following load cases were developed as;

Accelerating
Braking
Left Cornering
Right Cornering

Given that four load case scenarios had been develop, appropriate methods needed to be implemented to determine the reaction forces acting on the rear suspension. This solution was developed using a variety of software packages all used in conjunction to eventually develop axial reaction forces that could be imported into analysis package ANSYS

As discussed in the literature review, Tracksim, as developed by RMIT is a software package capable of exporting the COM accelerations acting on a vehicle at any point on any given track. To determine the forces acting on the rear suspension for the four aforementioned load cases this software was applied.

Table IV below provides a list of all specifications required for the Tracksim and the associated values input. It should also be noted that the track data including, corner radii and straight lengths, was taken from the 2001 Endurance track, as this data was already made available within the software [14]. Additionally, given that similar track specifications are required year to year and that current track details are not yet published, this assumption was deemed valid.

Table IV: Tracksim specifications

Vehicle mass	400 kg
Maximum power	50 kW
Transmission efficiency	100%
Base Accel grip limit	1.4g
Base lateral grip limit	1.4g
Base breaking grip limit	1.4g

Note that the transmission efficiency was given as 100% as to be conservative in the accelerations obtained. Additionally, the Tracksim code was altered as to convert the given accelerations into more appropriate forces at the centre of mass, for each 0.5m increment of the track.

These forces were then translated to contact patch forces through the use of an EES code as developed in a previous thesis.

By utilising key geometrical aspects of the vehicle, including COM height, wheel base and track width, and the forces at the COM can be translated to each wheel base contact patch. Additionally, the code assumes a roll centre that is on the ground. As detailed in technical specifications and compliance of design, this assumption is not entirely accurate, given the rear roll centre is approximately 10mm above the ground, but is close enough to assume reasonable confidence in the results obtained.

Given that appropriate load cases and their associated overall forces had been determined, a method of translating these into uniaxial component forces was required. This could have been achieved through basic hand calculated trigonometry, however, given that exact dimensions of the final design had not finalised, it was determined that producing a universal force solver code was more appropriate, as this would quickly and effectively be used to optimise designs. Coding package MATLAB was used for the development and implementation of these calculations.

The code works by the user inputting the unknown unit vector forces and their respective position, in the given scenario, this was location of the member nodes of the rear suspension along with their uniaxial direction (from chassis to upright which was transferred to a unit vector) with respect to the MATLAB codes defined positive co-ordinate system of; z into the page, x from right to left, and y directly upwards.

From this point the code would then create a matrix with the number of columns equivalent to the sum of the number of unknown components and the known components, and the number of rows equivalent to sum of all nodal positions for both knowns and unknowns plus three. Following the creation of this matrix the code would then loop through all positions in this matrix applying specific calculations dependent on the row number of the matrix. These specific calculations define the first three rows as the sum of the forces in the x, y, and z and each subsequent three rows relating to the moment calculations about each nodal location for all the unknowns and known in x, y and z for all forces.

There are several issues that this matrix creates and these are mainly to do with the repetition of moment calculations (due to multiple unknown component forces occurring at the same node) thus resulting in the potential for linearity in a result if this issue were not addressed. The next section of code is then dedicated to refining this matrix by discarding useless information and isolating important information. This is done first by looping through each consecutive moment calculation comparing with all other moment calculations and setting any repeats to three rows of zeros. The following loop then ran through every row in the matrix comparing with the condition of zero equals zero which would otherwise cause linearity in the result and isolating the rows which do not meet this condition. The reasoning behind this order of events is due to the necessity of the removal of this zero equals zero condition (e.g. all components having zero force in the z which may well have a legitimate solution but the matrix entering a line of zeros in the sum of the forces calculations which would result in linearity) and also the necessity of the removal of repeated moment calculations about the same point.

After the above section in the code has been completed the result is a refined matrix where any possible form of computational errors having been removed and the only remaining locations of linearity/error being within the problem itself. The last section of code then separates the known columns with the unknown columns into two separate matrices (essentially taking the knowns to the other side of the equals sign). Then discards the lower rows until and matrix relating to the number of unknowns is square, inverses this and computes the result. All subsequent code is then formatting this information for the users benefit.

In order to benchmark this code against that of the analytically determined results found in the audit the geometry used in that analysis was inputted into the program and computed, the differences in the results were found to contain less than 1% error (0N to 10N difference at most) which may have been due to rounding error incurred through the analytical analysis conducted in audit. It should also be noted that it was also benchmarked with several smaller problems which were used to not only develop the "Matrix Key" stated above but also to determine the flexibility and accuracy of the program which was found to be 100% accurate for all of these smaller solutions (note: rounding error in these smaller problems was actively reduced thus explaining the increased accuracy). By implementing the above procedure, with the contact forces determined from a combination of TrackSim and the EES code, the following uniaxial forces for each component were determined under each load case condition as in Table V, VI, VII and VIII

The following section details the process involved in the analysis of all rear suspension systems. Finite element analysis package ANSYS was used for as the basis for all computational analysis. The general procedure used was as follows;

Import component geometry into static structural analysis.

Generate reasonable mesh.

Define all contacts and supports.

Using the aforementioned forces, insert remote forces and run analysis for each load case.

Following the above procedure, the obtained stressed were imported into excel. To determine the fatigue history plots. The stresses at each point on the track, relating to one of the four load cases ratio against the largest stress. These plots vary for each component and can be seen in the subsequent sections. Note that one cycle on the fatigue plot is equivalent to a single lap of the endurance course. Additionally, given the large range of motion for all components, large deflection analysis was defined.

From the design audit completed on the previous generation vehicle it was found that the Lower A-arms were the critical member. As such large design consideration was given to this component when performing the analysis on the system.

Table V: Uniaxial force through components during acceleration

Component	Push Rod	Lower A arm		Upper A-arm		Rocker	Control Arm	Tab
		Front	Rear	Front	Rear			
Axial Resultant Force (N)	-3.41 E +03	-2.57 E +03	1.74 E +03	2.57 E +03	-1.34 E +03	-3.41 E +03	1.78 E +03	-3.41 E +03

Table VI: Uniaxial force through components during braking

Component	Push Rod	Lower A-arm Front	Lower A-arm Rear	Upper arm Front	Upper A-arm Rear	Rocker	Control Arm	Tab
Axial Resultant Force (N)	-1.43E+03	3.86E+03	-783.798	-599.012	301.8169	-1.43E+03	-1.00E+03	-1.43E+03

Table VII: Uniaxial force through components during left cornering

Component	Push Rod	Lower A-arm Front	Lower A-arm Rear	Upper arm Front	Upper A-arm Rear	Rocker	Control Arm	Tab
Axial Resultant Force (N)	-4.49E+03	5.17E+03	-3.35E+03	508.2514	1.22E+03	-4.49E+03	-8.38E+02	-4.49E+03

Table VIII: Uniaxial force through components during right cornering

Component	Push Rod	Lower A-arm Front	Lower A-arm Rear	Upper arm Front	Upper A-arm Rear	Rocker	Control Arm	Tab
Axial Resultant Force (N)	-915.448	-1.41E+03	3.67E+03	865.0401	-1.95E+03	-915.448	8.38E+02	-915.448

Initial analysis performed on the first iteration of design showed that a top mounted gusset plate as shown in Figure 5 below produced large stresses at a singularity caused by both the lack of a weld simulator and the inability to place a weld on both sides of the gusset plate.

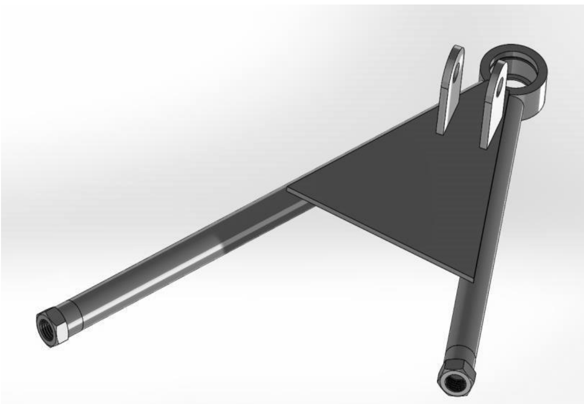

Fig.5. First iteration of lower A-arm

This was refined by centring the gusset plate in the A-arm cavity as seen in Figure 6. This allowed a weld bead to be placed on both sides of the plate and reduce the stress concentrations at the singularities.

This iteration was for the analysis as it severely reduced the stress and increased the overall life of the component.

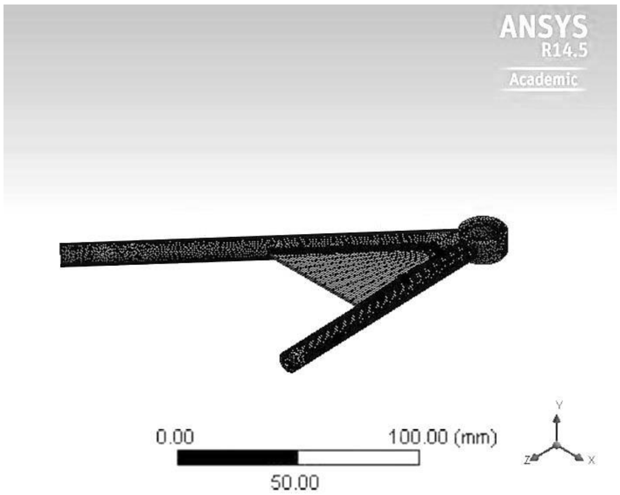

Fig.6. Final A-arm design

The analysis of the lower A-arm consisted of creating an appropriate mesh, and refining this until confidence in the results was ensured. An arbitrary mesh resolution of 3mm was initially implemented before refining this in the critical sections and reducing the size in the non-critical areas to decrease computational strain as in Figure 7.

Fig.7. Mesh of lower A-arm

Additionally, fixed supports were attached to the heim joint ends of the A-arms to simulate a conservative chassis connection. In real world scenario there is a small amount of play at these joints and as such would not be fixed, but this ensures a slightly conservative analysis.

A remote force was inserted; using the MATLAB developed values, at the centre of the spherical housing, simulating the load imposed by the bearing onto the lower A-arm. This was varied for each load case, Figure 8 below indicates the location of both the fixed support and force. It should be noted that large deflection was enabled as to resemble the large motion that a suspension system undergoes.

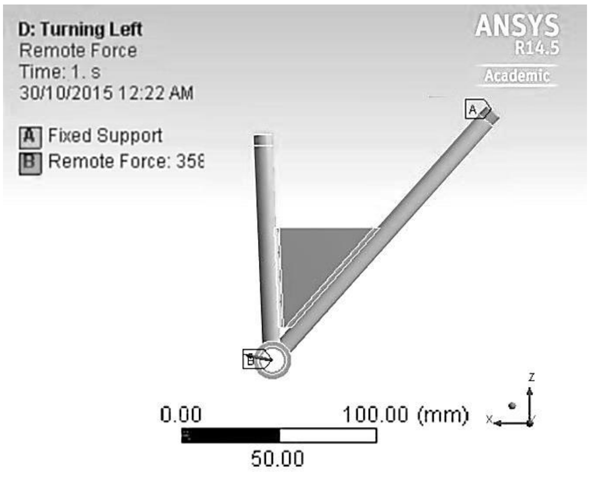

Fig.8. Supports and Forces on lower A- arm

Given that all the inputs were specified, static structural analysis was performed on the system. Figure 9 and Table IX below shows the location of the maximum stress and associated stress. It should be noted that the location of the maximum stress was at the same location for each load case, occurring at the connection of the two members. This stress concentration could be reduced with the inclusion of an appropriate weld, however this proved extremely difficult to implement in the CAD model due to its complex geometry at that point.

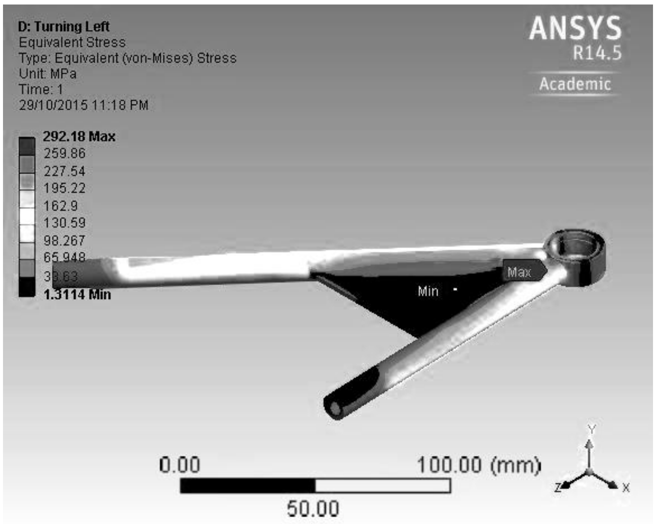

Fig. 9. Critical load case

Table IX: Lower A-arm Stress Values

Load Case	A-arm stress (MPa)
Accelerating	154
Braking	267
Left cornering	292
Right cornering	157

From this stage a fatigue analysis could be performed on the system. This consisted of developing an appropriate load history, by taking an endurance track and determining which load case was being undertaken at each point on the track and making its stress a ratio against that of the most critical load case, left cornering. This developed the following fatigue history as in Figure 10 below;

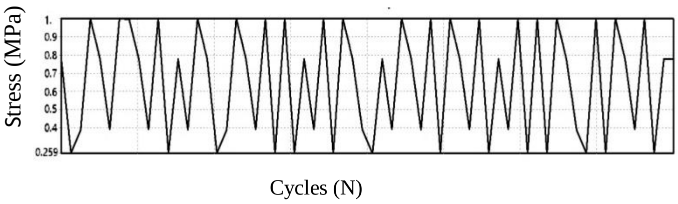

Fig.10: Lower A-arm fatigue history plot

Additionally, a fatigue factor, K_f, had to be determined for the fatigue analysis, this process is given in Table X below.

Performing this analysis produced the following life distribution in the lower A-arms. According to Figure 11 It was found that the minimum life expected is 1466. This value will be discussed further in a later section of the paper.

The upper A-arm analysis was very similar to that of the lower, with all fixed supports and applied forces at the same locations. The upper A-arm also had the same fatigue factor of the lower. However, the ratio for the fatigue history plot differs.

The maximum location of the stress also occurs at the same point, due to a large stress concentration at the connection of the two members

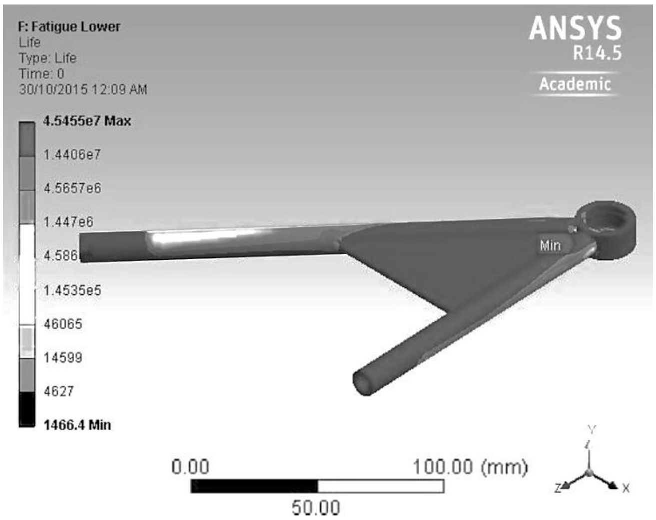

Fig. 11. Life span of lower A- arm

Table X: Fatigue Factors for Lower A-arms

Fatigue Factor	Symbol	Value	Comment
Loading	C_a	0.9	All reactions are axial
Size Factor	C_b	1	Already accounted in C_a
Surface Factor	C_c	0.79	Machined mild steel
Temperature Factor	C_d	1	T<350°C
Reliability Factor	C_e	0.814	99% as is easily replaced part and want A-arms to fail before other more expensive components
Modifying Factor	C_f	1	Accounted by ANSYS
Miscellaneous Factor	C_g	1	Assume no other factors
Total	K_f	0.57	

The history plot and the associated life of the components are given in following Figures 12 and 13.

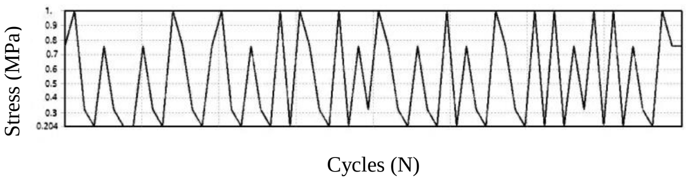

Fig.12. Upper A arm fatigue history plot

The rocker, being an integral part of the suspension process, was analysed critically and to a conservative manner. It was modelled using AISI 1020 steel as an alternative to the HA250 grade that has been specified for purchase. The following procedure was undertaken to analyse the rocker under the specified load cases.

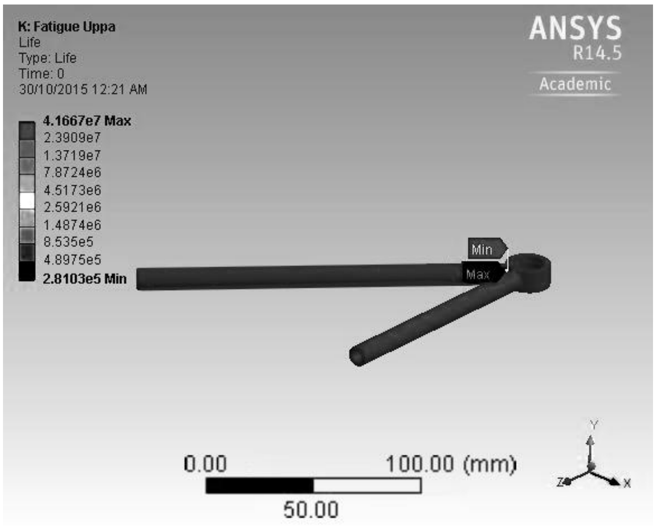

Fig. 13. Life span of upper A-arm

It should be noted that although the rocker would ideally be modelled with a spring-to-ground contact, as it is how occurs on the vehicle, this was not performed as the model failed to accurately represent the stresses under these conditions This issue was unable to be rectified and as such, more conservative analysis of making the rocker fixed above the spring mounts was conducted.

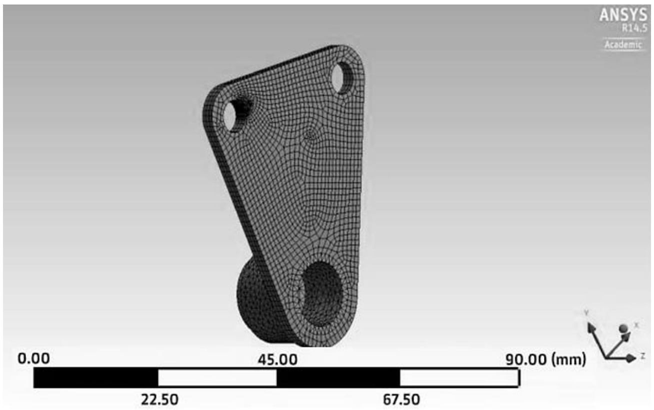

Fig. 14. Overall rocker mesh

An overall mesh resolution of 2mm was initially conducted as an arbitrary choice.

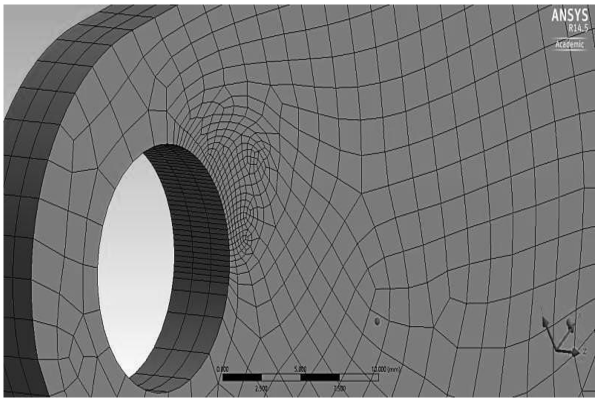

Fig. 15. Mesh resolutions at critical regions

After preliminary analysis, regions of interest were refined using spheres and line edgings. The final mesh resolutions are as Figures 14 and 15.

All contacts between the rocker plates and shaft were defined as bonded as to simulate the welds that would be present. Additionally, a frictionless support, as in Figure 16 below was inserted as to resemble the bearing that would be present.

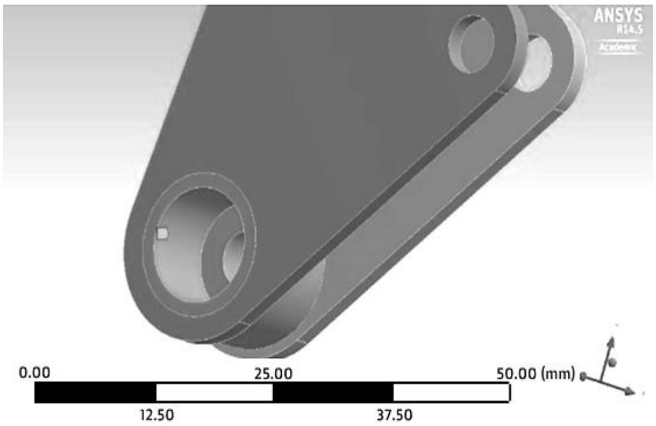

Fig.16. Frictionless support

The following Figures 17, 18 and 19 represent the static structural major supports and loads. The analysis had two components: the force by the pushrod and the countered force from the shock.

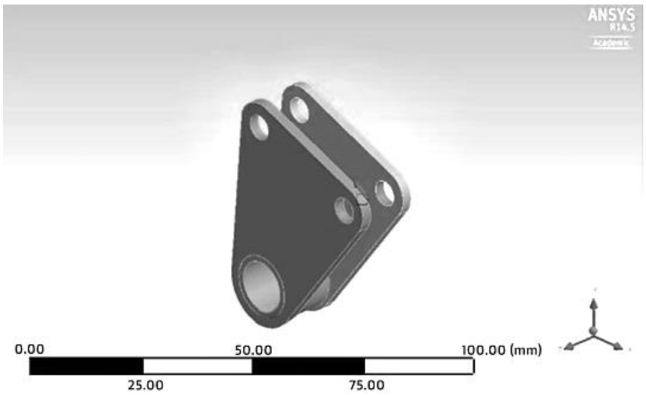

Fig. 17. Resultant Pushrod force

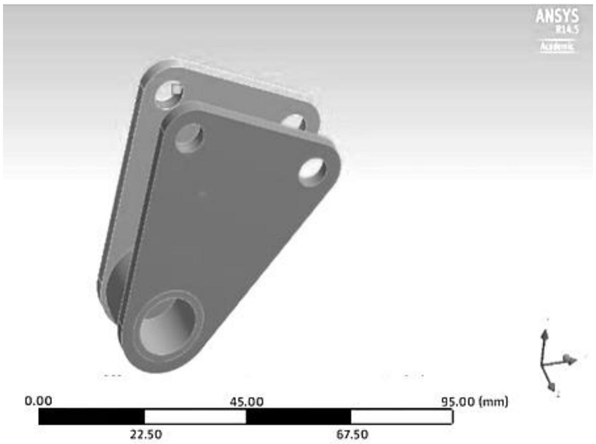

Fig. 18. Fixed Rocker Support

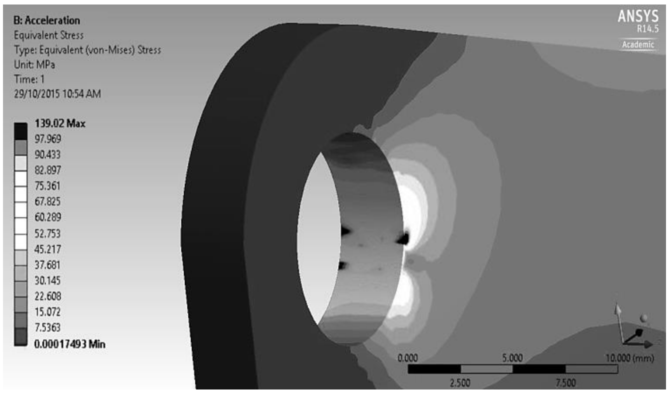

Fig. 19. Location of maximum rocker stress

Table XI: Stresses present in Rocker

Load Case	A-arm stress (MPa)
Accelerating	108
Braking	54
Left cornering	139
Right cornering	36

These stresses were then input into the fatigue analysis with the following fatigue factor in Table XII, and ran against the fatigue history as given in Figure 20 below. Note that due to the rocker only ever experiencing a compressive force from the rocker, the history plot never falls below zero.

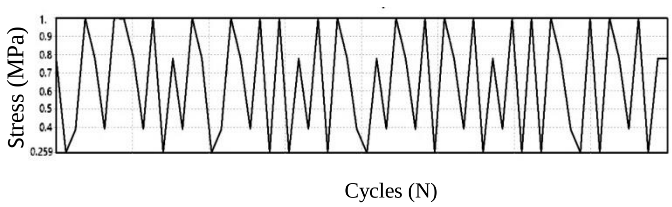

Fig. 20. Fatigue history plot for Rocker

Table XII: Rocker Fatigue History

Fatigue Factor	Symbol	Value	Comment
Loading	C_a	1	Assume Bending
Size Factor	C_b	1	Already accounted in C_a
Surface Factor	C_c	0.79	Machined mild steel
Temperature Factor	C_d	1	T<350°C
Reliability Factor	C_e	0.814	99% as is easily replaced part and do not expect rockers to fail within 2 years.
Modifying Factor	C_f	0.8	Circular cut outs
Miscellaneous Factor	C_g	1	Assume no other factors
Total	K_f	0.51	

With this given history plot and fatigue factor, an ANSYS based fatigue analysis was conducted. It was found that this component would last infinitely at a total number of laps of 4.5×10^8.

Further refinement of the rocker is recommended due to this fatigue result, however given time restraints, more focus was given to the more critical components, such as the A-arms.

The pushrod was a relatively simple component in the rear suspension system to analyse. A body mesh of 2mm was implemented, with a convergence test run where the results converged within a single iteration regardless. The meshing is shown in below Figure 21.

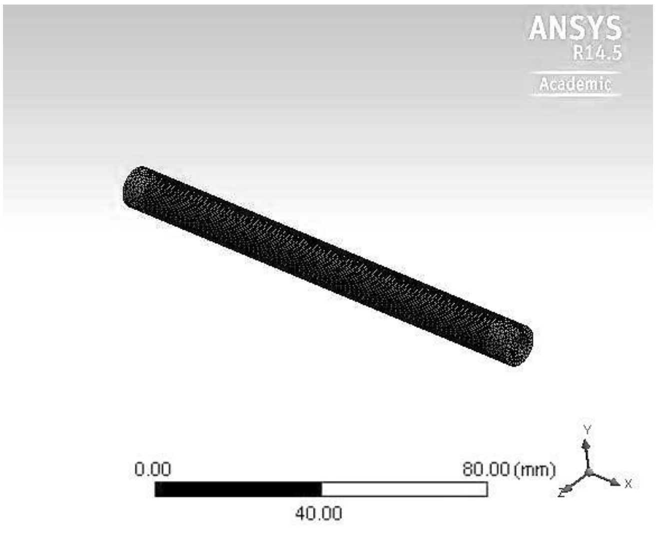

Fig. 21. Pushrod Mesh

A very basic load and support were implemented, with one end being fully fixed and an axial force being applied to the other. Running each load case through a static structural analysis resulted in maximum vonmises' stress. The maximum for each case was found to be at the adaptor, as seen in the left cornering scenario represented in Figure 22 below.

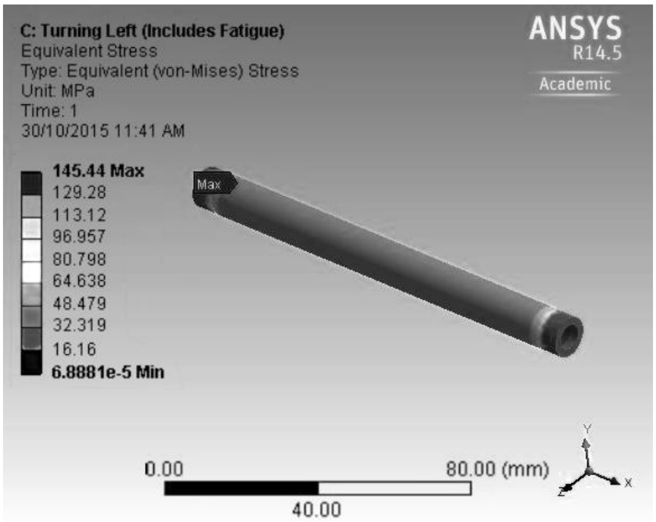

Fig. 22. Maximum stress location

All Maximum stresses are also shown in the table XIII below,

Table XIII: Maximum Pushrod Stress

Load Case	A-arm stress (MPa)
Accelerating	110
Braking	46
Left cornering	145
Right cornering	30

Following the same procedure as previous fatigue analysis to procure a history plot and using the same fatigue factor as both A-arms gives the following Figure 23 and 24

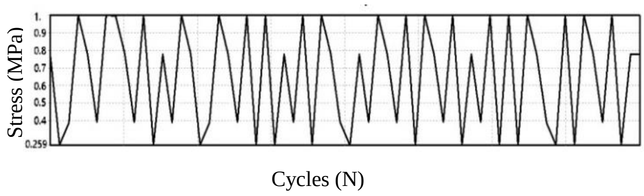

Fig. 23: Pushrod fatigue history

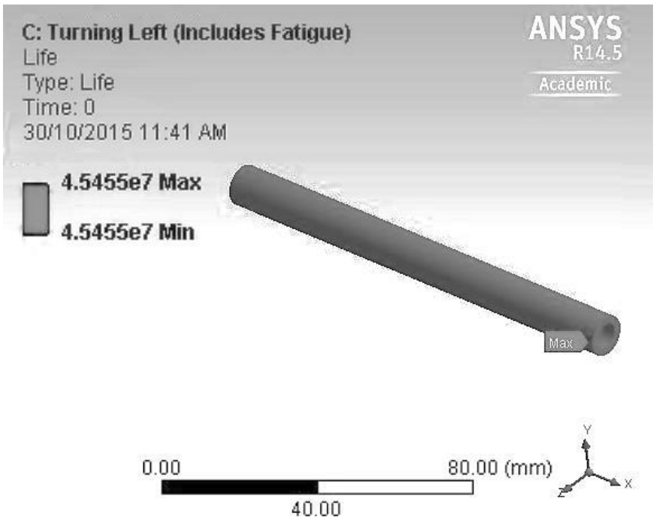

Fig. 24. Pushrod fatigue life

For the pushrod to effectively mount to the lower A-arms, a tab mount was constructed. These were welded to the lower A-arm gusset plate and attached to the pushrod using one M10 bolt. The Tabs were made of HA250 steel and 3mm wide, this grade of steel was equivalent to AISI 1020. The following process was undertaken to determine the stresses present for each load case and hence the overall life of the component.

Initially an overall 2mm mesh element size was used to locate the zone of critical stress concentration. Once the critical zone was located, a sphere of influence element size 0.2mm was used to refine the computational approximation. This reduction in the element size was found to significantly alter the stresses, and as such a convergence test was run. This greatly improved the confidence in the results given and shown in Figures 25 and 26.

The system was comprised of two independent components, the supports used were fixed at the base of each plate to represent their weldments on the gusset plate as in Figure below.

Additionally, a force equal to that of the pushrods force was implemented as shown in Figure 27 and 28.

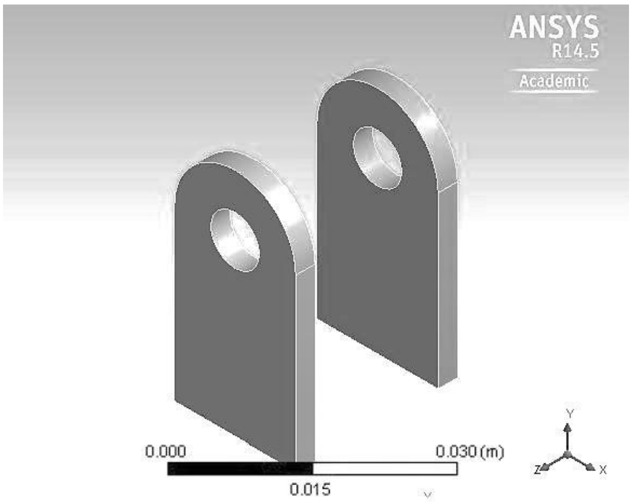

Fig. 25. Overall tab analysis

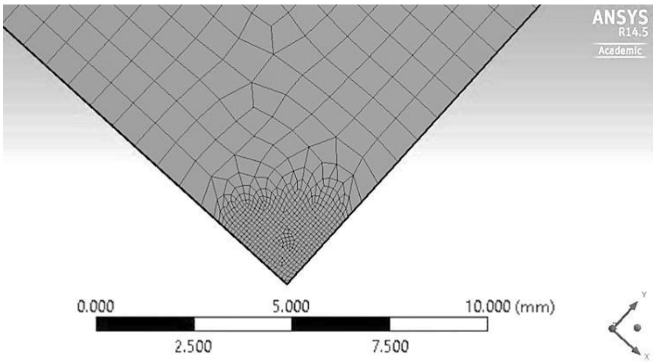

Fig. 26. Tab stress concentration

This force was distributed over the two bolt holes, mirroring the force distribution that would occur under real circumstances. Note that a full static bolt analysis will be outlined later in the paper ensuring that it would not fail.

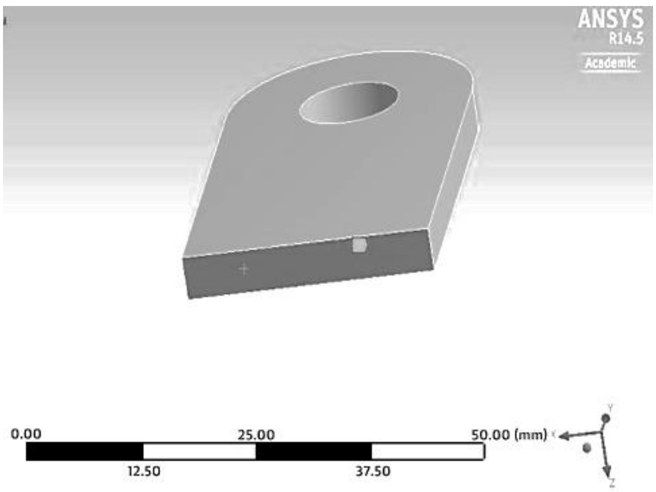

Fig. 27. Fixed support

Upon running each load case the maximum stresses were developed as shown in Table XIV.

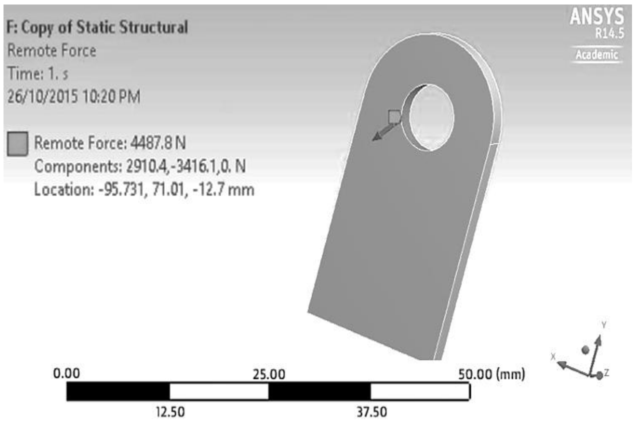

Fig. 28. Remote displacement example

Upon running each load case the maximum stresses were developed as shown in Table XIV. It was found that the maximum stress concentration (Figure 29) occurred at the tab corners and would be severely reduced with the implementation of a weld, as would be done to mount the tabs in place. But given that infinite life was achieved regardless, as shown further below, this analysis was deemed unnecessary.

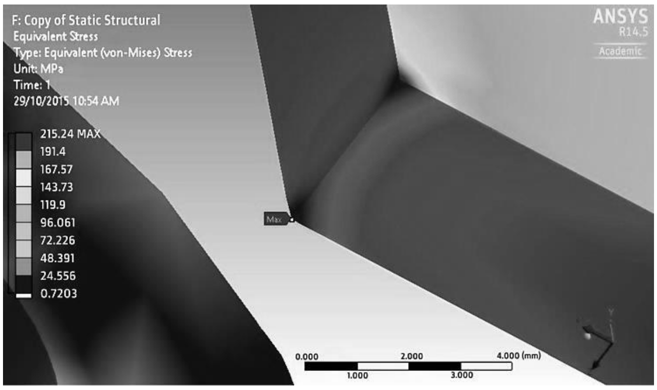

Fig. 29. Location of maximum tab stress

Table XIV: Maximum Stresses in Tab

Load Case	A-arm stress (MPa)
Accelerating	136
Braking	57
Left cornering	215
Right cornering	37

Again, based on the above maximum stresses a fatigue analysis was performed on the tab, using the below fatigue history and fatigue factor as in Table XV and Figure 30;

It was found that the entirety of the tab achieved infinite life under the given circumstances. Due to infinite life being achieved and no critical stress zones that were unaccounted for no further refinement was necessary for the tab.

From the given analysis of each rear suspension component, it was found that the designed system would be effective to be implemented in a vehicle. Moreover, that each component met infinite life with the exception of the lower A-arm, yielding approximately 1466 laps before predicted failure

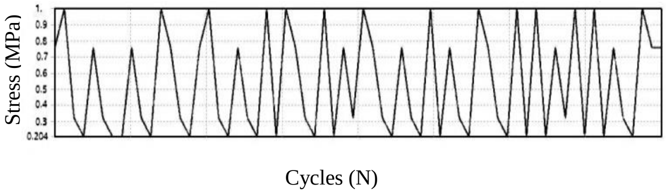

Fig. 30. Fatigue history loading of tab

Table XV: Fatigue Factor for Tab

Fatigue Factor	Symbol	Value	Comment
Loading	C_a	1	Assume Bending
Size Factor	C_b	1	Already accounted in C_a
Surface Factor	C_c	0.79	Machined mild steel
Temperature Factor	C_d	1	T<350°C
Reliability Factor	C_e	0.814	99% as is easily replaced part and do not expect rockers to fail within 2 years.
Modifying Factor	C_f	0.8	Circular cut outs
Miscellaneous Factor	C_g	1	Assume no other factors
Total	K_f	0.51	

It was estimated that given 5 track tests a year, plus each competition event would only amount to approximately 500 laps. This gives the designed system a Factor of Safety of over two years of life span.

Table XVI: Summary of the designed system

Component	Max Stress (MPa)	Minimum Life (Laps)
Lower A-arms	294	1466
Upper A-arms	75	2.81×10^5
Pushrod	145	Infinite
Rocker	139	Infinite
Tabs	215	Infinite

In addition to fatigue analysis, other forms of failures that suspension system susceptible to were analysed to ensure full confidence in the designed system.

Given that the pushrod is a relatively thin long member, which is constantly undergoing compressive stress, it is highly vulnerable to buckling failure.

Hence buckling analysis was conducted on the pushrod under its highest stress state in left cornering. It was found that the worst case loading, with the current dimensions and material selection of the pushrod at 16x1.6mm and annealed mild steel respectively, the Factor of Safety for the system buckling is 3.1. This Factor of Safety proves that the pushrod system will not fail under buckling.

V CONCLUSIONS

In conclusion, the design fits the current chassis geometry and meets the specifications assign by the FSAE competition. The computational analysis and load casing provides a significant refinement in developing and manufacturing of this novel design from the current system. The lower a-arm was found to be the most critical component in the assembly with a maximum stress of 264 Mpa and a minimum life of 1466 track laps. There is still room to refine the novel model since three of its assemblies last for infinite cycles.

For future reflection, the complete vehicle stage development should initiate at the uprights and progress to the suspension system ending at the chassis. Designing a complete chassis before the suspension system restricts the a-arms and rocker mounting points, which also limits the design from reaching maximum performance.

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