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Research Article

On the design of rear uprights for a race car

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Abstract

This paper details the design of rear upright for a JCU Tec-NQ (JTR) Racing Motorsport, Formula Society of Automotive Engineers (FSAE) race car using Aluminium Alloy 7075-T6 through the use of a track simulator developed by RMIT Racing, forces on the upright were calculated and an upright was designed in SolidWorks. To ensure optimal design with the aid of Finite Element Analysis using ANSYS workbench, three main load cases with combination of drive cases (acceleration, braking, and left and right turn) were used to evaluate performance of the rear upright. They include fatigue of upright, single bump loads and the zero-based fatigue loading of the calliper-mounting bracket. For the fatigue loading of the whole upright, it was observed that the greatest stress throughout the four drive cases occurred in a left hand turn. For single bump loads, the upright could withstand a vertical bump load of stress value significantly lower than the ultimate tensile strength. For the fatigue analysis on the brake calliper mount, the resulting safety factor plot presented a minimum factor of three for infinite life. Other relevant analysis were presented, such as sphere of influence and exaggerated deformation which showed that the upright will experience a maximum deformation of 0.132mm during the worst case lap scenario (left hand turn). The material considered for the design of the rear upright was Aluminium Alloy 7075-T6 as it has superior mechanical properties in terms of stiffness, fatigue and tensile strength for the application compared to both the Alumec alloy and Aluminium Alloy 6061-T6 considered in this paper.

1. Introduction

The Society of Automotive Engineers Australasia (SAE-A) is the world's third oldest mobility society and was founded in Melbourne in 1927 to address the need for further education for all facets surrounding Automotive Engineering and now encompasses all mobility engineering industries in the Asia Pacific region. Annually the SAE-A hosts the Formula SAE-A event (FSAE-A) which comprises of international university student entrants participating in a three day competition focussed on the engineering, manufacture and racing of an internal combustion or electric race car up to 600cc/80kW [1].

The JCU Tec-NQ Racing Team (JTR) represents James Cook University, comprised of JCU School of Engineering and Physical Sciences students, and Tec-NQ at the FSAE-A competition. Formerly The JCU Motorsports team, the club of motoring enthusiasts debuted at the 2014 Formula SAE Australia competition in Melbourne. Unfortunately, the

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2014 car did not perform up to the high standards associated with the FSAE-A competition technical inspection and was unable to compete. Many parts on a FSAE vehicle require regular maintenance and remanufacture due to limited life cycles of critical components, as well as the need to progress with ongoing technological advancements [2] [3]. The JTR team worked on analysing, designing and implementing multiple improvements to the FSAE-A vehicle. These improvements encapsulate multiple facets and functions of the vehicle and all worked towards the penultimate goal of successfully entering a competitive vehicle in the 2015 FSAE-A event. One of the key areas of the vehicle focussed on for enhancement was the rear wheel assembly.

Fig. 1 Current JTR Upright Assembly

A typical rear unsprung mass assembly of a FSAE car comprises of the following critical components:

- Upright [4]
- Brake Calliper Assembly
- Hub Shaft
- Brake Rotor
- Bearings [5]
- Fasteners [6]

Each component has its own specific role that plays a vital part in the overall performance of the vehicle. With each differing component comes new design requirements and functionalities that need analysis and enhancement. All elements must be compatible with each other and allow all other parts full functionality. The current rear wheel assembly on the JTR team's vehicle was designed and analysed in 2012 with some minor adjustments having been made since then. The current assembly is shown in Fig. 1. A critical component of the rear unsprung mass assembly in all vehicles is the rear upright.

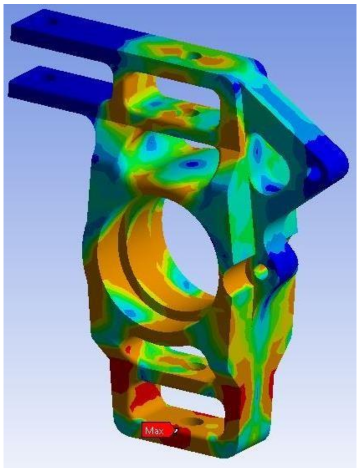


Fig. 2 Example of Finite Element Analysis conducted on the 2015 rear upright design

Rear uprights are integral components of many automotive vehicles and perform multiple key functions for the vehicle. They provide an attachment between the rear wheel assembly and the rear suspension [7], serve as a housing for the roller bearings of both the stub and hub shaft [8] and operate as an anchor point for the brake calliper. The rear uprights must be able to withstand all loads expected within the desired life cycle (strong and durable) as well as allow the car to be competitive in race conditions. Complying with the rules of the FSAE-A competition and ensuring a safe environment is achieved, must be at the forefront of all analysis undertaken.

A design audit was conducted on the current JTR upright, which included its performance under specific load conditions as well as an extensive review of relevant literature. Firstly, working on this paper brought to light the specific requirements of a rear upright on an FSAE vehicle. Following extensive FEA modelling (shown in Fig. 2), the audit showed the positive elements from the current design which could be implemented in the reworked element as well as some pitfalls which should be avoided.

This paper will specifically address the re-design of the rear uprights for the JTR team vehicle used for 2016-2017 FSAE-A competition. Using information gathered from the previous design audit, which was conducted on the current component for the 2015 competition and relevant literature. This paper will run through the design process followed to improve the rear uprights starting with a critical review of researched literature, design development and analysis, compliance with relevant standards, technical specifications and compliance of the design, mechanical drawings and costing as well as any future recommendations.

The design and analysis of a rear upright for an FSAE vehicle, when completed in its fullest, is an extremely complex process with many details needing to be taken into consideration. This practice is to ensure that the component will endure forces expected in its lifecycle while still allowing the vehicle to be competitive. First, Australian Standards and FSAE rules and regulations must be consulted to ensure that the rear uprights are not only within the confines of the competition but are also allowing the vehicle to perform under safe conditions.

Handling performance properties for the rear of a vehicle such as camber, caster, and toe [9] are a function of both the upright geometry and adjustable suspension components

[10]. These characteristics greatly influence the way in which a vehicle will handle, and as such are critical to take into account during the design of an upright.

Camber, caster, and toe values can be observed in literatures [9] however such values are only approximate guidelines applicable mostly for standard automotive vehicles (i.e. monocoque road going vehicles). Extensive track testing or open wheel class racing experience is necessary to achieve the best results from the given system. It should be noted that each individually designed vehicle would likely have differing optimal upright adjustments. In a typical case, a formula FSAE vehicle would be set up with an amount (3-3.5 degrees roughly) of rear negative camber at rest in order to grip the road surface under high lateral load during cornering. While this could technically affect the vehicle's grip as it accelerates, this is likely not a problem due to the low power levels associated with these vehicles. Rear toe would likely be set at a neutral level or very slight amount of toe in to accommodate high speed stability. Caster is a less critical component for the rear of the vehicle, however a small amount of positive caster may be effective to provide the rear wheels with a trailing enhancing stability.

A number of methods for determining and analysing loads on FSAE uprights have been considered through various forms of literature [3] [7] [9]. Load cases are crucial to accurately audit previously designed uprights as well as designing a new improved upright. If the loads are not taken under very careful consideration analysis cannot be accurate and the upright may be over engineered or inadequate and cause serious injury or death to the FSAE driver and/or surrounding people upon failure. Also once the load cases have been carefully considered the analysis of those loads must be accurate and conclusive.

In order to obtain conclusive data, first load cases should be calculated using the track simulator. These loads then should be verified through on track testing using a three axis accelerometer at the centre of mass (COM) of the vehicle [11] or alternatively a suspension travel recorder. Once the load cases have been verified and are understood there are a number of methods of computationally analysing the upright as has been previously mentioned.

The software of choice in studied reports was ANSYS Workbench. Although, there are a number of different methods of applying loads to the uprights within this Finite Element Analysis package. The first method studied was to assume rigid links between the COM and the tyre patch [12] and compute reaction forces using sum of forces and moments (simultaneous equations). A code has previously been developed for the JTR FSAE team by Lachlan Plumb [13] using Engineering Equation Solver.

The second method is to model the four wheels in ANSYS as simple shapes and place them at the correct positions relative to the COM. Then to apply force in three directions at the COM (modelled as a point mass) and find the tyre patch forces using a probe.

Material Selection is an extremely vital decision that has to be made early on in the design process to guarantee that the upright will perform what is required from it. Physical properties of the material can influence many characteristics of the component that will not affect its individual performance but will affect the performance of the vehicle as a whole. Mass of the upright needs to be kept to a minimum so as to not affect the vehicles acceleration, but at the same time the element needs to have desirable strength [14] and stiffness [15] to avoid failure during competition. Material selection, in a competition such as FSAE, can ultimately be majorly influenced by costings (both time and money). Raw material and manufacturing costs can be the difference between the ideal substance and the ‘next best’ option [16]. When studying researched literature it was found that most FSAE rear uprights are designed from either Steel Alloy or Aluminium Alloy. Further research showed that high tensile Aluminium Alloys such as 6061-T6, 7075 or Alumec [17] proved to be the superior option due to their lower density and sufficient strength and stiffness. Manufacturability and manufacturing techniques can also be a major influence on upright design. While processes such as CNC machining may be ideal to create an extremely accurate design, monetary or time limitations may stand in the way. A combination of other cheaper common manufacturing processes such as water jet [18] [19] cutting, lathing or milling [20] may be a suitable comparative option which will create a component just as desirable as the other will.

Following the research and design/analysis process of an FSAE design team from Old Dominion University it can be seen that there are a number of routes which can be taken to achieve the final result of a functional and competitive upright. This specific team can be seen to have made quite a few assumptions in its analysis phase that may have influenced the accuracy of its results further on. The choice to ignore deceleration and only computationally analyse the component in a 'worst case scenario' may have resulted in an over engineered part that may actually hinder the overall performance of the car. A safety factor of 4.82, while obviously proving that the part is dependable, may be over compensating and further design changes to the upright may be able to be completed.

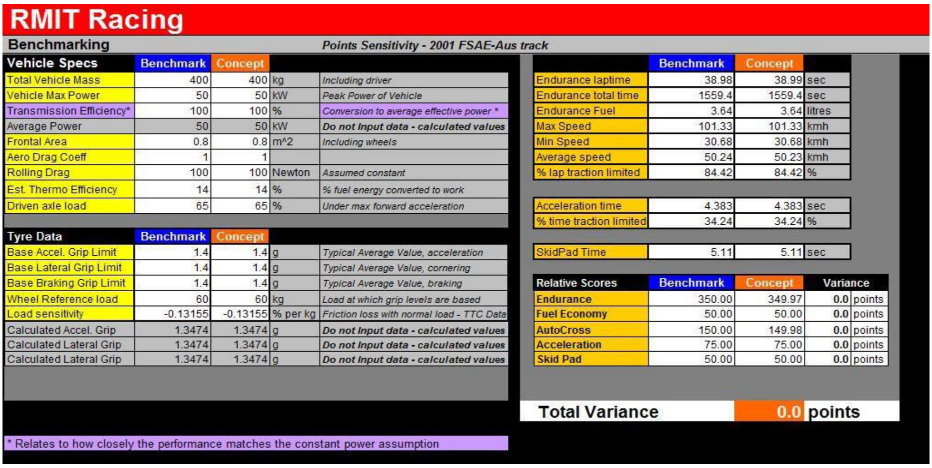
The previous design was constructed to have a geometry that was able to perform as both the front and rear uprights, which enhanced the repeatability of the component and would cut down on manufacturing and material costs but it did not allow for the differences in the functionality of the two differing uprights. The rear upright does not undergo the same stresses or perform the same processes as the front uprights mainly due to steering and the centre of mass of the vehicle. The rear upright will be reconditioned and analysed to provide better functionality and longevity in the new design.

2. Design Analysis

When designing a structural component for a high performance vehicle that is intended to be used in racing conditions, the accuracy of load cases cannot be of higher importance. Racing requires every component of the car be pushed to its limits for a specified design life whilst proving a design which is as minimalistic and light as possible. Thus the more that is known about the loads acting on the component that is being designed the more competitive the car can be.

There is currently no track data available for the design of the JTR Motorsports FSAE vehicle therefore other theoretical tools must be heavily relied on. It should be noted that throughout this design it was expected that the car would be undergoing track testing to obtain accelerometer and possibly strain gauge data. This data is of vital importance in verifying the design of the rear uprights since all design work has been carried out with no experimental benchmark to compare to. To simulate the loads acting on the JTR vehicle, an excel spreadsheet developed by RMIT Racing that acts as a track simulator was utilised.

As it can be seen in the Fig. 3 there are a number of user inputs which are required for the track simulator to calculate forces on the vehicle. The total vehicle mass, max power, transmission efficiency, frontal area, aero drag coefficient, rolling drag, thermal efficiency and driven axle load are the vehicular inputs. The grip limit of the tyres in the acceleration braking and lateral conditions are also required. The layout of the track is also required every half meter including radius [21] of corners and length of straights.



RMIT Racing

Benchmarking — Points Sensitivity - 2001 FSAE-Aus track

Vehicle Specs	Benchmark	Concept		
Total Vehicle Mass	400	400	kg	Including driver
Vehicle Max Power	50	50	kW	Peak Power of Vehicle
Transmission Efficiency*	100	100	%	Conversion to average effective power *
Average Power	50	50	kW	Do not Input data - calculated values
Frontal Area	0.8	0.8	m^2	Including wheels
Aero Drag Coeff	1	1		
Rolling Drag	100	100	Newton	Assumed constant
Est. Thermo Efficiency	14	14	%	% fuel energy converted to work
Driven axle load	65	65	%	Under max forward acceleration

Tyre Data	Benchmark	Concept		
Base Accel. Grip Limit	1.4	1.4	g	Typical Average Value, acceleration
Base Lateral Grip Limit	1.4	1.4	g	Typical Average Value, cornering
Base Braking Grip Limit	1.4	1.4	g	Typical Average Value, braking
Wheel Reference load	60	60	kg	Load at which grip levels are based
Load sensitivity	-0.13155	-0.13155	% per kg	Friction loss with normal load - TTC Data
Calculated Accel. Grip	1.3474	1.3474	g	Do not Input data - calculated values
Calculated Lateral Grip	1.3474	1.3474	g	Do not Input data - calculated values
Calculated Lateral Grip	1.3474	1.3474	g	Do not Input data - calculated values

* Relates to how closely the performance matches the constant power assumption

	Benchmark	Concept	
Endurance laptime	38.98	38.99	sec
Endurance total time	1559.4	1559.4	sec
Endurance Fuel	3.64	3.64	litres
Max Speed	101.33	101.33	kmh
Min Speed	30.68	30.68	kmh
Average speed	50.24	50.23	kmh
% lap traction limited	84.42	84.42	%
Acceleration time	4.383	4.383	sec
% time traction limited	34.24	34.24	%
SkidPad Time	5.11	5.11	sec

Relative Scores	Benchmark	Concept	Variance	
Endurance	350.00	349.97	0.0	points
Fuel Economy	50.00	50.00	0.0	points
AutoCross	150.00	149.98	0.0	points
Acceleration	75.00	75.00	0.0	points
Skid Pad	50.00	50.00	0.0	points

Total Variance 0.0 points

Fig. 3 RMIT Track Simulator

It can be seen on the right side of the Fig. 3 that RMIT integrate the points awarded at competition for each event into the track simulator. This is a very strategic method of optimizing the teams' competitiveness and is also carried out by other teams such as Monash Motorsport.

This track simulator is designed to provide the accelerations that act on the centre of mass (COM) of the vehicle every half meter around the input track. The Rear Suspension Design Team at JCU using energy methods converted these accelerations to forces.

As previously discussed, a method for converting forces at the centre of mass of the JTR vehicle was developed by Lachlan Plumb using basic sum of forces and moment calculations with the car geometry. Lachlan wrote these methods into an Engineering Equation Solver code in order to solve a large number of simultaneous equations [13]. This code was used to convert the COM forces provided by the track simulator to tyre patch reaction forces. It should be noted that the EES code that was used assumes no body roll or tyre deformation which is justified due to the fact that maximum load on the upright when both of these factors have reached maximum and no longer deform. The resulting forces that the EES code calculated for the rear right patch are shown in Fig. 4, where Force (N) is plot against Distance (m). The resultant reaction forces RD_X (blue line), RDY (brown line) and RD_Z are represented in the graph in Fig. 3.

Before using RMIT Racing's track simulator it was studied to ensure a better understanding. There were a number of assumptions that should be noted. The first and seemingly most important assumption was that the vehicle was either under maximum acceleration braking or cornering at any time. This means that the car is thought to be 'coasting' around each corner and all braking and acceleration is done whilst the car is straight. This is not completely out of scope to a real situation since most of the acceleration and deceleration will be done before the car enters the corner. Although, there will be a small amount of braking left at the beginning of the corner and the driver will begin accelerating after reaching the apex of the corner. The result of this assumption is that the forces during cornering could be slightly larger than provided by the track simulator, thus an appropriate safety factor should be used until track data can benchmark the forces on the upright. The resulting calculations if lateral and forward accelerations are taken into consideration simultaneously prove over complex and were not carried out.

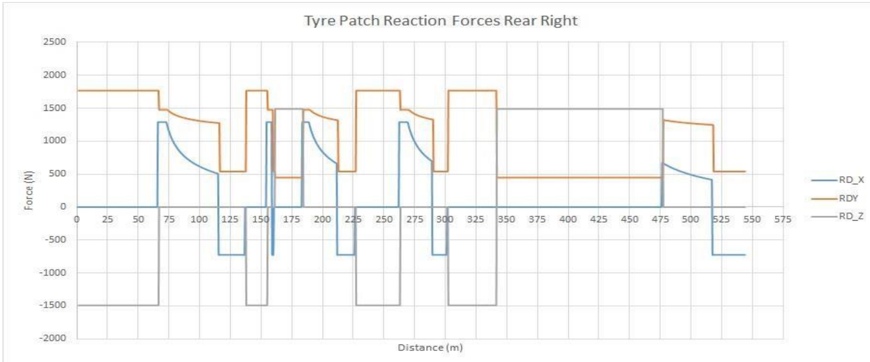


Fig. 4 Plot of Resulting Tyre Patch Reaction Forces for the Rear Right Tyre from Engineering Equation Solver

The second assumption that the track simulator makes is that the car will always drive in the centre of the track. This assumption causes the cornering loads that the simulator provides to be conservative (larger than a real racing scenario) since a 'racing line' will be much smoother. Since in practice the corner on the track is not followed in the centreline and a smoother path is taken, the forces on the tyres and all other components will be slightly less than those given by the track simulator.

Previous uprights were designed to withstand the worst-case scenarios such as maximum turning, braking and acceleration whilst hitting a bump to ensure that the upright will not fail under worst-case conditions. Although analysing fatigue on drive case such as this can result in an over conservative design that is stronger than necessary for its design life and heavy.

Instead of considering the worst possible load case to design the 2016 JTR rear uprights the track simulator was used in an attempt to produce fatigue history data for loads on the upright throughout one lap of a FSAE endurance track. The method of obtaining load cases used in this design provides potential for more competitive component design, specifically the upright. In addition to the fatigue load, cases a bump force of 2g was analysed for static failure to ensure any large bumps that the vehicle hits occasionally do not alter the structural integrity of the upright. The final load case, which was considered, was the zero-based fatigue loading which the brake calliper mounts must endure.

2.1 Finite Element Analysis

In order to design an upright that is strong enough to withstand the considerable forces of FSAE racing there were three main load cases, which were considered. These load cases included fatigue of whole upright, single bump loads and the zero-based fatigue loading of the calliper mounting bracket.

2.1.1 Fatigue Analysis

The most critical and complex load analysis that was conducted on the 2016 rear upright design was the fatigue of the entire upright. In order for the upright design to be successful, it must last the design life of two years including track testing and competition days.

The fatigue analyses was conducted using the maximum acceleration, max braking and maximum cornering (left and right) as previously discussed. The resulting forces used for each load was a reaction from the road onto the tyre contact patch. These forces were given as X, Y, Z components which were entered into ANSYS at the tyre contact patch as a remote force acting on the bearing [22] housing. The origin of the axis system is situated at the centre of the tyre patch, axis X is horizontal forward-backwards, axis Z is positioned vertically top to bottom, and axis Y is positioned horizontally from side to side.

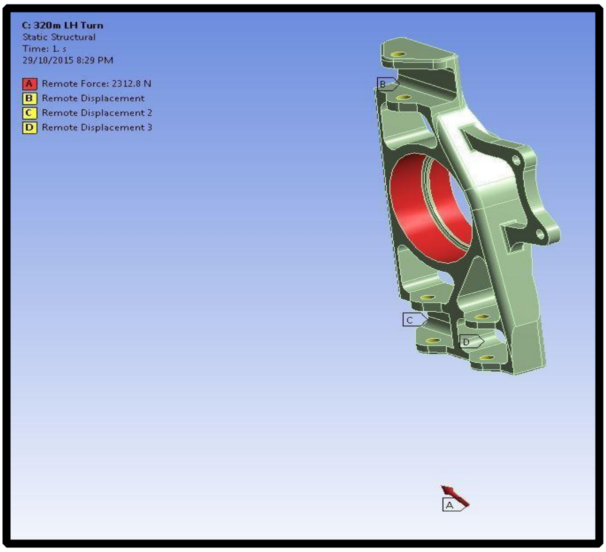


Fig. 5 Boundary Conditions and Applied Force in ANSYS (Left turn)

The difficulty was using forces that change magnitude in all three dimensions, as the vehicle navigates the track, to develop a fatigue analysis within ANSYS Workbench. Thus, a method of producing fatigue load history data for one lap around a previously used FSAE endurance track was developed and will be further discussed below.

A static structural analysis was conducted for the rear right upright (acceleration, braking, left and right turn) using all four drive cases on the upright. The tyre patch forces, which were given from the track simulator and EES code, were applied as a remote force acting on the bearing housing [23].

With the forces acting through the bearing housing, it was necessary to apply appropriate boundary conditions to each bolthole for the a-arms, control arm and the calliper mount. This was achieved through thorough consideration of the boundary conditions of each connection. The top a-arm only resists movement in the x and z directions as there is no push or pull rod attached to it. The bottom a-arm resists movement in all three directions (x, y, and z) since there is a push rod, which translates the vertical force on the wheel to the suspension. The control arm only resists toe angle movement thus only resists movement in the z – direction. Fig. 6 shows the installation with upper and lower a-arms.

Once forces and boundary conditions were accurately applied in ANSYS for each drive case, a critical load case was found. The comparison of the four drive cases can be seen in Fig. 7.

It can be seen that the greatest stress (approximately 85.05MPa) throughout the four load cases (accelerating, braking, left and right turn) occurred in a left hand turn. Thus, the stress at this critical point was found throughout the other three load cases and is shown in Table 1. Negative stress values were found for the right turn, this was found using a maximum principle stress probe in ANSYS at the critical location with positive values indicating tension thus negative indicating compression.

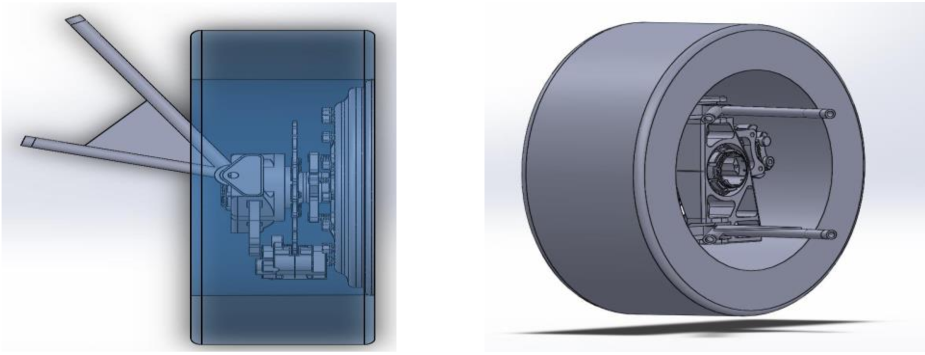


Fig. 6 Installation of rear upright with upper and lower a-arm in rear wheel assembly and Upright successfully connected to upper a-arm in rear wheel assembly (swept forward 60 degrees).

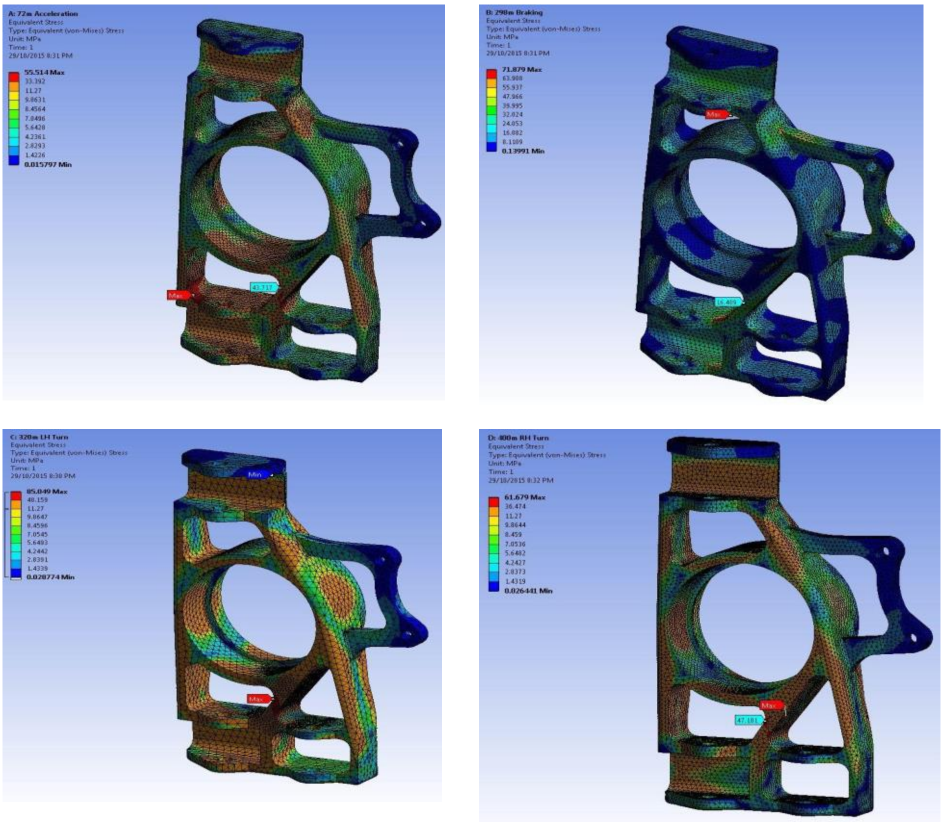


Fig. 7 Static Structural FE Analysis for Acceleration, Braking, Left and Right Turn

Table 1. Resulting Stress at Critical Point

Drive Case	Action	Dist (m)	Force X (N)	Force Y (N)	Force Z (N)	Stress at Critical Point (MPa)	Ratio of Maximum Stress
1	Acceleration	72	1291	1476	0	43.717	0.5140
2	Braking	298	-727.3	540.3	0	16.409	0.1929
3	LH Turn	320	0	1768	-1491	85.049	1
4	RH Turn	400	0	447.2	1491	-47.181	-0.5547

Table 1 was also used to find the ratio of maximum stress between each drive case in order to develop a load history. Once the ratio of maximum stress was found for each loading case it was matched against the track simulator data in order to provide appropriate ratios for each load as the vehicle navigates the track. The resulting load history data for one lap can be seen in the graph in Fig. 8.

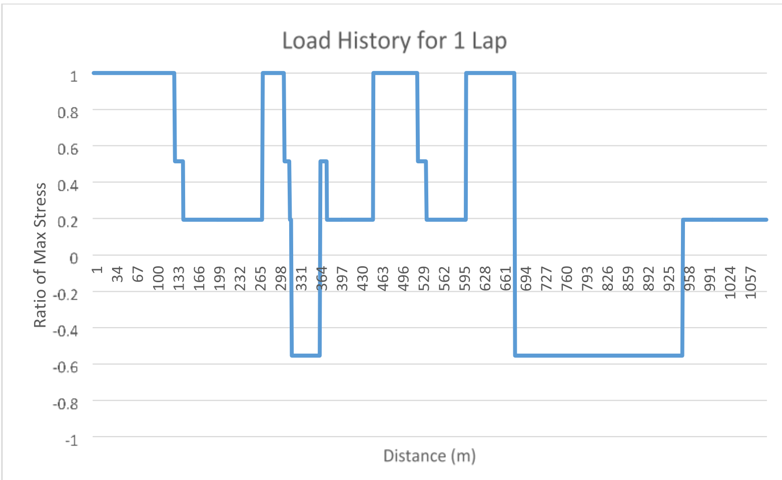


Fig. 8 Load history ratio plot (one lap)

The history data was then imported as a .DAT file into an ANSYS Fatigue Tool for the left hand turn case. This is not a perfect method of analysing fatigue for a force which changes in three dimensions but after consultation with JTR Motorsport alumni it was deemed acceptable until further improvements are made.

It should be noted that before a fatigue analysis was conducted the fatigue sensitivity factor K_f was determined. Using "*Fundamentals of Machine Component Design*" [24] with an ultimate tensile strength for Aluminium Alloy 7075-T6 of 580MPa and a machined surface finish the surface factor was found to be 0.79. A reliability of 99% was chosen which gives a reliability factor of 0.814. All other factors that affect fatigue sensitivity were considered carefully and it was found that many of these had a value of one with ANSYS taking into account all others.

K_f = Surface Factor X Reliability Factor

K_f = **0.64306** (1)

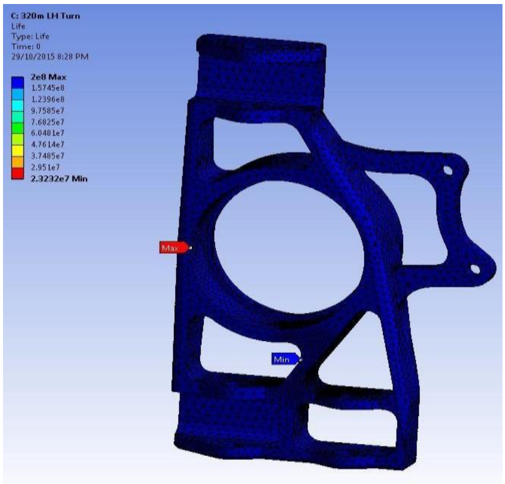


Fig. 9 Fatigue life using track history data

This fatigue sensitivity factor was implemented into the ANSYS fatigue tool (using Goodman mean stress theory). The resulting life of the upright is shown in Fig. 9 where one cycle is representative of one lap of an FSAE endurance track. As it can be seen, the critical point on the upright dropped slightly below infinite life but is still in the order of millions of laps before failure.

2.1.2 Bump Analysis

For repetitive larger forces, a static structural analysis was done to simulate the vehicle hitting a large bump, this is called bump analysis. A bump analysis was performed to ensure the upright could not only endure the fatigue of racing around a track but also a worst-case scenario of hitting a large bump (possibly a ripple strip) while turning as hard left as possible. To simulate this situation the left load case was used with a vertical force of two times that which is normally experienced in a left hand turn:

$F_y = 2 \times 1768 = 3536N$ (2)

Using double (200%) the vertical force whilst hitting a bump and turning left than that of just turning left, the resulting equivalent stress was increased from 85.049Mpa to 111.5Mpa (131%). Thus it was found that the upright can withstand a vertical bump load of 200% with an ultimate tensile strength of 580Mpa. A zero based sinusoidal fatigue load was applied to investigate the fatigue results of hitting large bump. A zero based sinusoidal fatigue load was applied to investigate the fatigue results of hitting a large bump. It was found that the upright lasted 1e9 bumps with a factor of safety of approximately one.

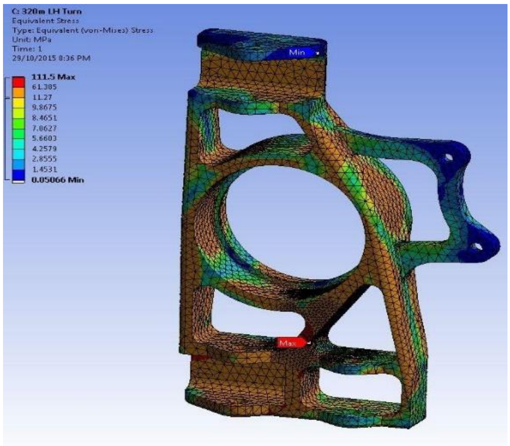


Fig. 10 Static structural equivalent stress results for bump test

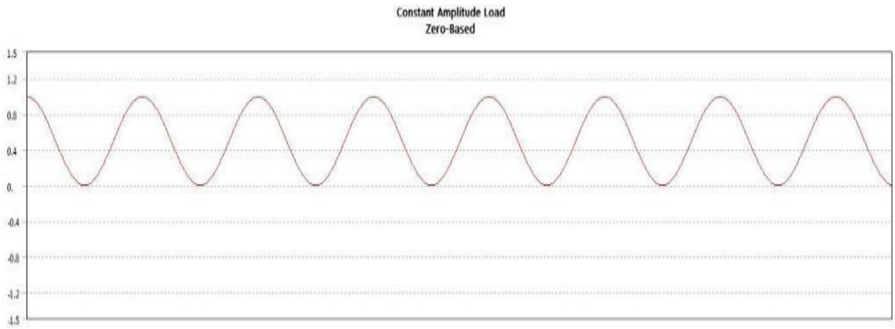


Fig. 11 Bump test zero based fatigue load

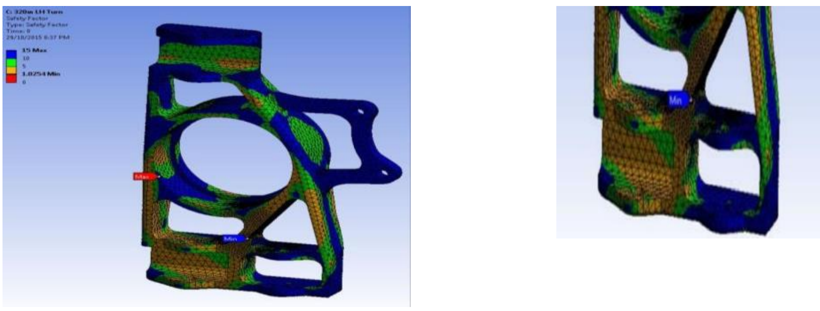


Fig. 12 Bump test fatigue safety factor also showing focus on the minimum fatigue area

2.1.3 Fatigue Loading of the Calliper Mounting Bracket (Brake Calliper Analysis)

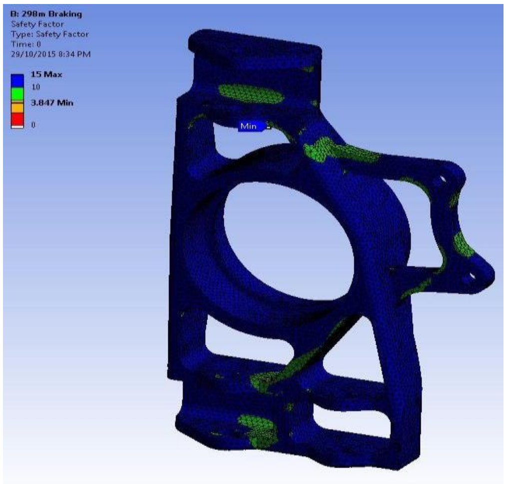


Fig. 13 Brake calliper mount fatigue safety factor

A separate fatigue analysis was conducted on the brake calliper mount to ensure that a zero-based load will not cause the mount to fail. The resulting safety factor plot is shown in Fig. 13 with a minimum factor of three for infinite life.

2.2 Mesh Refinement

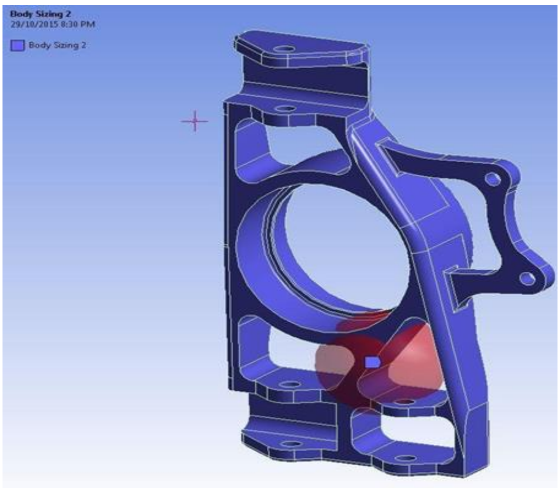


Fig. 14 Sphere of influence

A mesh of 5mm elements was used on the entire upright with further refinement around the critical point. To ensure maximum accuracy and minimal computational expense the mesh was reduced to an element size of 2mm around the critical point using a sphere of influence as shown in Fig. 14.

This methodology used simulated track data to calculate the force experienced at each tyre contact patch for the four different drive cases that the vehicle will experience assuming cases do not occur at the same time. This data was in turn used to calculate the forces being experienced by the rear upright and FEA within ANSYS workbench was used to find the resulting stresses. The zone on the upright, which was seen to encounter the most stress, was deemed critical. A fatigue analysis was conducted on this critical region by subjecting it to conditions experienced in one lap of a past FSAE endurance track. The lifecycle of the rear upright was defined by the amount of times it could complete this lap before failing. The design life of the rear upright is only within the thousands of laps throughout its time on the vehicle. Although, methods of determining the load cases throughout the analysis required the use of a track simulator that is not fully understood. Thus, the upright has not been optimised as far as initially intended. The ability to verify load cases with JTR vehicle track data will greatly increase the knowledge of the track simulator results and the accuracy associated. In order to verify the load cases a number of instruments could be used including; a 3-axis accelerometer mounted closely to the centre of mass of the car, strain gauges on suspension members and/or a suspension travel potentiometer. Once loads on the upright are found from track data, the design can continue to be optimised by removing unnecessary material.

2.3 Life

Using the process described in the design analysis section of this paper, a conservative but still realistic load life was developed for the rear upright. After conducting this analysis it was found that the constructed rear upright was able to successfully complete 20 million laps (infinite life) before failing under the experienced stresses. The critical regions of the design are highlighted in Fig. 7, shown earlier in the document, which depicts contoured regions that represent the ability to undergo differing numbers of laps before failure. Although this design life may far exceed what is actually required of the rear uprights it allows for any uncertainties or approximations made during; track simulation, contact patch forces calculations and FEA analysis. It also gives leeway for any inaccuracies during manufacture or installation and gives the component a better chance at surviving any unforeseen circumstances (crashes or impacts).

2.4 Exaggerated Deformation

The goal of an upright is to find an appropriate middle ground between two cases; be stiff enough to ensure that pre-set performance characteristics are not altered during performance while at the same time still also having the ability to provide some flex during uncommon circumstances (bumps, cone contacts) to avoid fracture. This is achieved through designed geometry and material selection. The conceptual upright design provides geometry and material selection that allows for minimal deformation under its calculated life cycle but still allows for some movement under infrequent occurrences. Fig. 15 shows that the upright will experience a maximum deformation of 0.132mm during the worst case lap scenario (left hand turn). After consultation with JTR members it was deemed that this was an acceptable amount as it would have minimal effect on pre-set performance characteristics. When applying a ‘bump’ to the computer model by doubling the load applied on the vertical axis it was seen that the deformation nearly doubled to 0.18mm under the worst case scenario (left hand turn). This shows that the design is not statically rigid and will aptly distort when required.

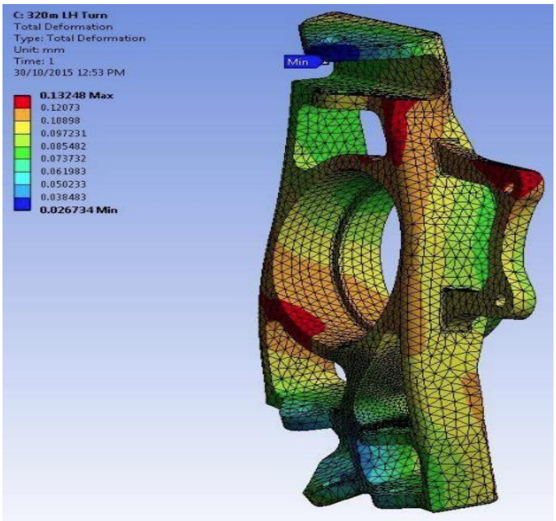


Fig. 15 Exaggerated Deformation FEA Analysis

Producing a rear upright of minimal weight has been at the forefront of most design decisions. The final design has an approximate mass of 1065 grams with an approximate volume of 3.7886×10^{-3} m^3. This showed a reduction of 11.35% in weight when compared to the current upright in place on the JTR vehicle. Relatively speaking this may not seem extremely significant, but when working with a vehicle as light as those used in the FSAE competition every reduction in weight adds up and can lead to improved lap times at competition. Weight reductions were achieved by firstly removing material that did not necessarily have a functional obligation in the design (vacancies in the upright). As well as this, relatively thin walls can be seen in the final design which noticeably reduces weight. It is important to note that weight reduction was not made at the cost of stiffness or strength and that a functional and conservative middle ground was found.

2.5 Materials Selection

The material that was used in the design of this rear upright was Aluminium Alloy 7075-T6 (based on the Aluminium 7*** series of alloys) as it has superior mechanical properties for the application. It has similar properties to the Alumec alloy, which was used in the current upright design. Aluminium 7075-T6 has superior fatigue strength and high tensile strength for high stress applications compared to the Alumec alloy and Aluminium 6061-T6. Alumec has superior wear and corrosion resistance compared to the other Aluminium alloys considered and have comparable high strength levels with Aluminium 7075-T6. Aluminium 6061-T1 is more workable compared to the rest but has reduced fatigue strength compared to Aluminium 705-T6.

Therefore, apart from the price range of Aluminium 7075-T6, which is relatively close to that of Aluminium 6061-T1 and significantly lower than Alumec, it shows preferable collection of properties required and cost implication for the rear upright design and possesses high fatigue strength and tensile strength, which are integral for the application. It should be noted that the front upright team is designing to 6061-T6 Aluminium Alloy thus fatigue was also analysed for the rear upright using this material.

As can be seen in Fig. 16, when the material was changed to 6061-T6 the number of laps until failure reduced significantly down to the order of hundreds of thousands. Thus it can be said the the use of 6061-T6 will greatly reduce the life of the upright which must be considered when making modifications to the upright design. Although the rear uprights are able to be manufactured using 6061-T6 and still last their design life in order to reduce the cost of materials when purchasing for front and rear uprights.

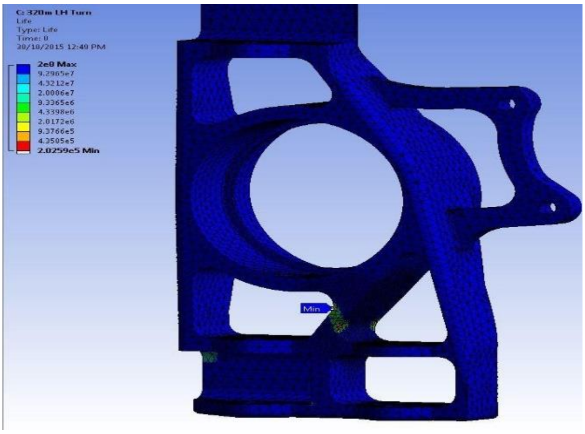


Fig. 16 Life plot of Aluminium Alloy 6061 T6 fatigue analysis

3. Conclusion

Rear uprights have a great impact on performance characteristics of the vehicle, such as toe, camber and caster, which are relevant in improving handling and posting competitive times during competition. It was seen to be successful in meeting important criteria at this design phase such as weight, bump, and fatigue strength.

Three important load cases were focused on with combination of drive cases (acceleration, braking, and left and right turn) were used to evaluate performance in the design of the rear upright, they include fatigue of whole upright, single bump loads and the zero-based fatigue loading of the calliper mounting bracket, It was discovered that:

- For the fatigue loading of the whole upright, it was observed that the greatest stress, approximately 85.05MPa throughout the four drive cases occurred during a left hand turn.
- For single bump loads, the upright could withstand a vertical bump load of 111.5MPa with an ultimate tensile strength of 580 MPa
- A separate fatigue analysis was conducted on the brake calliper mount to ensure that a zero-based load will not cause the mount to fail. The resulting safety factor plot presented a minimum factor of 3 for infinite life

Other relevant information/analysis were presented, such as sphere of influence and exaggerated deformation which showed that the upright will experience a maximum deformation of 0.132mm during the worst case lap scenario (left hand turn).

The material considered for the design of the rear upright was Aluminium Alloy 7075-T6 as it has superior mechanical properties in terms of stiffness, fatigue and tensile strength for the application compared to both the Alumec alloy and Aluminium Alloy 6061-T6 considered in this paper.

Throughout the design of the rear upright, a number of other components were undergoing design changes as well. This made it difficult to ensure compatibility with rear suspension as they collaborated with the chassis and different mount design groups. In order to compensate for the large angle, which the a-arms are swept backward the connection points on the a-arms, were offset forward 20mm. This affects the strength of the uprights as the a-arm connections are not vertically in line with the centre of the hub, in order to design an optimal upright these connections should be moved in line with the hub centre.

It should be noted for future developments that methods for accurately analysing bolt [i] on clevises would make achieving full range of motion simpler. The use of shims with a bolt on clevis would allow the camber to be adjusted.

In order to provide load cases that match future acquired track data, it is recommended that the RMIT Racing track simulator be further studied and understood. Alternatively, JTR Motorsports could develop their own track simulator that would include benchmark and concept cars along with competition points in order to prove more competitive each year.

Bearing seals were investigated and it is thought that a 'sheave seal' should be used from Timken but the supplier has not yet given a part number. Generic bearing seal dimensions were added in SolidWorks and before manufacture, the dimensions of the bearing housing in the upright should be confirmed.

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