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FORMULA SAE REAR SUSPENSION DESIGN

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RESEARCH ARTICLE

ABSTRACT: The purpose of this report is to investigate the design development and evaluate the structural and functional performance of the proposed system for the 2019 competition. With the review of the literature surrounding rear suspension systems, FSAE standards, analysis techniques and important design parameters, the foundation for the proposed James Cook University (JCU) 2019 rear suspension system is established.

This paper is highlighted the development and analysis path undertaken in the construction of rear suspension system befitting a Formula SAE vehicle. Formula SAE is an international student competition centered on the design, construction and racing of an internal combustion vehicle. All parts are designed via SolidWorks and FEA testing is incorporated using ANSYS to test out various loads under different scenarios in racing. Main components including beam axle, trailing arms, brackets, spring and damper are covered in this paper. The design is focused on providing a low cost and easy to manufacture design which operate for infinite life cycles.

KEY WORDS: Rear suspension design, Dependent suspension system, Formula SAE suspension, steering, design, finite element analysis

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PROJEKAT ZADNJEG OSLANJANJA FORMULA SAE

REZIME: Svrha ovog rada je da prikaže istraživanje, razvoj i procenu strukturnih i funkcionalnih performansi sistema predloženog za takmičenje 2019. godine. Pregledom literature o sistemima zadnjeg oslanjanja, FSAE standarda, tehnika analize i važnim projektnim parametrima, formirana je osnova za predloženi sistem zadnjeg oslanjanja Univerziteta James Cook (JCU) 2019.

U ovom radu je prikazan razvojni put i analiza koja je urađena u konstrukciji sistema zadnjeg oslanjanja vozilu Formule SAE. Formula SAE je međunarodno studentsko takmičenje fokusirano na dizajn, konstrukciju i trke vozila sa motorom sa unutrašnjim sagorevanjem. Svi delovi su dizajnirani pomoću SolidWorks-a, a FEA testiranje opterećenja u različitim scenarijama trka je urađeno pomoću ANSIS-a. U radu je prikazan razvoj glavnih delova, uključujući: krutu osovinu, vođice, nosače, oprugu i amortizeru. Fokus dizajn je na niskim troškovima proizvodnje i jednostavnom proizvodnom procesu sistema koji treba da ima beskonačan životni ciklus.

KLJUČNE REČI: Razvoj zadnjeg oslanjanja, Zavisan sistem oslanjanja, Oslanjanje Formule SAE, Upravljanje, dizajn, Analiza konačnim elemenatima

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1 INTRODUCTION

he Formulae Society of Automotive Engineers (FSAE) competition is bounded by a strict set of rules that must be adhered to by all competing teams. These rules are enforced to ensure the safety of all involved, whilst also placing limits on vehicles, creating an even playing field. Moreover, other constraints including considerations of material, geometry and manufacturing processes should also be taken into account [1].

A motor vehicle's suspension system creates the harmony between road and the driver orchestrating all the components of the chassis that need to work together, the suspension system stabilizes the vehicle attitude during accelerating, breaking and cornering while isolating road's roughness from passenger compartment. Suspension system is the link between the wheels and the chassis, transmitting the weight of the vehicle on to the wheels. A suspension system must keep the wheels in proper camber, resist chassis roll to an extent and keep the tires in contact with the road surface with minimal load variations for the vehicle to handle in a desirable manner.

Selecting the correct suspension design for an application is of crucial importance and usually compromises are required, whether it is in the geometry to allow for more space in the car or in the stiffness of the ride to accomplish the necessary handling characteristics – planning a suspension system is a significant task while designing a desired car. There are various factors which are required to be analyzed in the development for the design of the suspension of a motor vehicle. Some major factors include, independent or dependent systems, camber, and toe, roll center etc.

According to the FSAE Design Event Score Sheet [1], with the majority of marks associated with the dynamic performance of the car, therefore, therefore, it is imperative that the suspension of the vehicle is a competitive solution and can withstand the entirety of the events.

With the review of the literature surrounding rear suspension systems, FSAE standards, analysis techniques and important design parameters, the foundation for the proposed James Cook University (JCU) 2019 rear suspension system is established. The purpose of this report is to investigate the design development and evaluate the structural and functional performance of the proposed system for the 2019 competition.

The rest of this paper is organized as follows. In Section 2, independent and dependent suspension systems are introduced. Preliminary considerations, including design constraints and load conditions, and design development are proposed in section 3. Numerical analysis of the proposed system is presented in Section 4. Finally, in Section 5, conclusions are presented.

2 SUSPENSION SYSTEM

The main objective of the current work was to design a low cost and easy to manufacture suspension system with an appropriate performance. Considering this, the optimum choice was between a dependent system and an Independent system and to save money and cost. There are a range of suspension systems utilized by various teams in the FSAE competition. Suspension systems can be categorized into two main systems, namely independent suspension and dependent suspension.

2. 1 Independent rear suspension

Independent suspension refers to the rear suspension system of the vehicle which allows each wheel to move independently from one another. Independent rear suspension (IRS) systems are preferred in vehicles where comfort and performance are major requirements. The MacPherson Strut [2] and the Double wishbone [3] are two commonly configurations; however, the most common IRS setup used in racing situations such as FSAE is double wishbone [4-6]. A double wishbone uses two wishbones where each wishbone is made of two single arms which have their two ends joined together to create a V shape and a trailing arm to attach the upright to. Factors such as control arm length and mounting, camber and toe settings, roll center, pitch center and center of gravity should be considered thoroughly to construct reliable and effective rear suspension [7, 8].

Independent suspension involves complex geometries, increased costs and maintenance. Designing a successful double wishbone can greatly improve the vehicles handling, traction, and stability and ultimately dictate the success of the vehicle. However, there are many moving parts which must be accurately and precisely engineered so that geometries are exactly as designed. An error in the geometry can have detrimental effects on the vehicle’s performance and driver’s safety. A well rounded system may take multiple stages of development and multiple redesigns which can be very costly and time consuming. With increased components comes increased weight and costs. The wishbones, pushrods and sway bar will take up tremendous space at the back of the vehicle and will add a substantial amount of weight to the car. With space being already scarce on FSAE vehicles, and to improve performance; a smaller lighter unit will need to be developed. This will be a costly venture which may not be beneficial as shortened geometries can lead to reduced effectiveness.

2. 2 Dependent rear suspension

Dependent suspension systems have a rigid linkage between the two tires, such that any movement of one wheel is translated, and thus the forces are also translated from one wheel to another. Non-independent suspension system has the advantages of simple structure, low cost, high strength, and easy maintenance. As mentioned before, the aforementioned independent systems all have better handling and more effective but the many of these advantages have to be overlooked due to the cost effectiveness of beam axles. Therefore we decided to go with dependent system, more specifically, a solid/rigid axle.

There are different types of suspension setup including leaf springs [9], there-link [10] and four-link [11] systems. The solid axle consists of the basic dependent system where the two rear wheels relate to a form of a rigid beam, so that when one-wheel encounters and irregularity on the track surface, the other wheel is directly affected. This type of suspension is usually found on the rear of a rear-wheel drive and is therefore a live axle. The translation of forces between the connected wheels causes the camber angle to be consistent regardless of the travel of the suspension. A solid axles’ fore and aft location is constrained by either trailing arms, semi-trailing arms, radius rods or leaf springs. The lateral location is constrained by either a paint hard rod, a Scott Russell linkage or a Watts linkage. Solid axles have two instant center axes, one for parallel bump motion and the other for roll, these axes will move with changes in ride height [12].

The principal advantage of the solid axle is the simplicity causing it to be very space efficient and relatively cheap to manufacture, also it is extremely strong and durable, making it a suitable fit in high load environments. Drawbacks for this type of suspension is that it does not allow each wheel to move independently in response to bumps and the mass of the beam is part of the unsprung weight of the vehicle which can further reduce ride quality. The cornering is inferior than other suspension designs as the wheels have zero camber angle gain during body roll, furthermore, front solid axle suspension is unusually sensitive to any lack of concentricity in the hub and wheel assembly which can cause a side to side oscillation.

3 DESIGN APPROACH

The design of the suspension system was focused around three key objectives, it was essential to achieve infinite life cycles, high strength to weight ratio, and to reduce the total cost to manufacture and assemble the vehicle. The design process of the vehicle suspension has been based on iterative experimentation approaches, where design variables (e.g. material, geometry, damping, etc.) have been altered, and re-analyzed the system until acceptable design criteria's have been achieved.

3. 1 Preliminary considerations

Load cases

It is essential to consider multiple load cases for different scenarios of the vehicle as forces in each suspension member are different given the geometry and orientation of the system. Force analysis is an integral part of any FSAE team and must be completed as part of the FSAE Structural Requirements Certification Form (SRCF) [1]. Whilst there are a number of load scenarios that may be considered for analysis, of which would be subjected to the car during static and dynamic events, only the critical load cases will be analyzed. These critical load cases include linear acceleration, braking and critical cornering in this work. Understanding the forces and stresses within the suspension components allows effective design iterations to be made to later redesign geometry for a better performance.

Numerous assumptions need to be made to ensure the accuracy of force calculations and the transmission of forces through the vehicle components. It is essential to assume that the rigid links connect the center of mass (COM) to the tire contact points and the resulting force acts through the suspension geometry. Therefore, the forces generated by the different load cases transmit through the tire contact patch to the upright and to the mounting surfaces of the rear axle. Whilst the forces are transmitted to the mounting face between the upright and rear axle, the location of the force is the center of the tire. Hence throughout the analysis, remote forces at this location are used to account for the force and moment generated. Additionally, as the coefficient of friction between the tire and the track is constantly changing due to the heat generated at the contact point, the coefficient was 0.8 which was verified throughout literature. This value was used to determine the frictional force on the wheels for the different load cases.

For the linear acceleration and braking load cases, specialized formulas were used to develop loads based on the vehicle parameters in table 1. The calculations for these different load cases provide an estimation of the forces acting in each direction on the suspension and would require physical testing to improve accuracy.

Table 1 Vehicle parameters

Parameter	value
Linear acceleration	1 g
Braking	1 g
Mass of car	400 kg
Wheelbase	1630 mm
Centre of Mass Height	350 mm

Coefficient of Friction	0.8

Linear acceleration

An important load case to consider for the FSAE vehicle is routine linear acceleration which occurs frequently during the dynamic events. From ref [1], the acceleration event involves the evaluation of the vehicle's acceleration in a straight line on flat pavement across 75m. For this load case it is assumed the acceleration is purely linear as it travels down the track. The maximum acceleration of the vehicle for critical acceleration is considered to be $1.019\ g$ which corresponds to $10\ m/s^2$. As purely linear acceleration is being analysed for this load case without any affect from braking, the force is considered a zero-based, cyclic load. During linear acceleration, the $\mathrm{F_z}$ (forward direction) component of the rear suspension is due to the tractive forces on the rearward tires. The $\mathrm{F_x}$ component of the vehicle is zero as the acceleration is purely linear, and no cornering is experienced. In addition, forces in the y direction are comprised of the static weight of the vehicle and driver on the wheels and the dynamic weight transfer from front to rear caused by the acceleration. These forces can be calculated with the equations proposed by Smith [8] (see Appendix A).

Braking

Another very common occurrence during the racing of the FSAE vehicle is specifically linear braking, experienced when slowing down [1]. The maximum braking force is a function of the deceleration in the z component, which was considered a maximum of 1 g without the wheels locking up. Like the linear acceleration, the F_{x} component will be zero as there is no cornering. The force in the y direction is due to the static weight on wheels less the dynamic weight transfer as the braking occurs and the weight shifts forward. Using the vehicle parameters in table 1, the braking forces for each component are determined from the formulations in ref [8] (see Appendix A).

Critical Cornering

To incorporate all the forces acting on the car at once during a scenario, a load case has been developed for the critical cornering whilst braking. This case can be considered a worst-case scenario and is essential for analysis to determine the performance of the suspension under loading configurations. This scenario is subjective, with some papers considering this to be the forces experienced while cornering at maximum deceleration, see Figure 1 [13], while others consider it to be merely a maximum cornering effect, whereas the suspension is fully compressed on one side and fully relaxed on the other [14].

The free body diagram in Figure 1 displays the information for the applied loads during critical cornering of the car. There are numerous assumptions associated with this load case, which stipulate that the vehicles performance is $1.2\ \mathrm{g}$ cornering acceleration and $10\ \mathrm{m/s^2}$ straight line acceleration. In addition, it is assumed that rigid links connect the COM to the tire contact points and the resulting force acts through the suspension geometry.

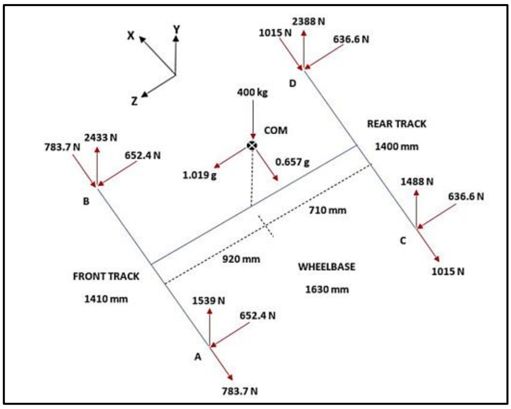


Figure 1 Critical load case [13]

From this set of information, the critical forces were identified, and forces produced from the tires to the rear axle and the direction of the forces were used to develop the load case. During cornering, the weight naturally transfers from the outside to the inside wheel, which creates a moment with respect to the origin.

On the other hand, the suspension will resist this moment through the various components designed to counteract the motion and hence generate the reactive force. As the vehicle takes alternative corners, the direction of the loads changes and hence the load is analyzed as a fully reversed, cyclic load.

Table 2 Summary of load case forces

Load Case	F_x	F_y	F_z
Linear Acceleration	0	1528.28 N	1222.4 N
Braking	0	658.76 N	54861 N
Critical Cornering	1015 N	2388 N	636.6 N

From the critical loading case of the entire vehicle, the loads for the rear suspension quarter can be applied to the geometry. The 1.019 g force acts on all components whilst the X, Y and Z forces from the tire act on the mounting surface of the rear axle. With the loading specifications sourced directly from the client and calculated using reliable sources, the accuracy of the loading conditions are assured. A summary of the loads in each direction acting on the rear axle during critical load cases is presented in Table 2. These forces are used for the FEA investigation to analysis the performance of the rear suspension and inform design changes.

3. 2 Constraints of the design

The design of any component should be undertaken with the goals of satisfying the technical requirements set by FSAE 2019 rules (e.g. geometry restrictions on the size of wheelbase track and ground clearance, fasteners, etc.) [1] and constraints arise from facility limitations and considering the main objective of the design including low cost and easy to manufacture suspension system. For this purpose, the following considerations were performed.

Material selection is one of the first key factors for all FSAE teams when designing components for the suspension system. The first and most important consideration taken when selecting the suspension system material is its strength to weight ratio. Aluminum, 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 [13]. Due to this a compromise is generally made and therefore common materials used for structural members are Chromalloy and mild steel. These material's strength to weight ratio is still fairly good compared to Aluminium composites and carbon tubes. Chromalloy and steel are easy to handle and relatively cheap. A comparison of several possible material selections can be seen below in Table 3. Therefore, we decided to set 4130 Steel for the beam axle, trailing arms and mild steel for the brackets and weld-in bung.

Table 3 Material Advantages and Disadvantages [13]

Material	**Advantages**	**Disadvantages**
Mild Steel	• Baseline material requiring no additional design • Easy to weld • Good workability	• Mild steel tube not readily available locally in small quantities.
AISI 4130	• 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
Composite	• Very high strength to weight ratio	• Requires proof build quality • Very expensive • Needs monocoque designs • Requires mechanical fastening to main hoop
Aluminium	• Good strength to weight ratio • High workability	• Requires Mechanical fastening to the main hoops • Best used in monocoque designs • Difficult to source locally

Additionally, according to all relevant FSAE standards, ensuring no component of the suspension clashed with external components of the vehicle is essential. This involved designing the suspension to avoid the drive train, differential and CV shafts whilst static and during vertical travel of the suspension. To ensure this, regions specifying the locations where the brackets could be mounted to the frame to support the trailing arms should be identified accurately.

Based on budget limitation, the type and size of the shocks to be used for the suspension was specified in the preliminary design. Along with an effective suspension geometry, selecting the correct springs and shock absorbers is essential to maximize the tire contact with the road surface. Consequently, it will result in an increase in traction of the vehicle which will enable the car to travel at higher speeds more safely [15]. It is decided to use Penskie 7800 shocks with with a spring length of 200 mm and spring stiffness of 57 N/mm.

Moreover, to reduce the overall weight and consequently the cost, it essential that the geometry of the suspension is compact and many external components are avoided, yet structurally sufficient. In addition, the proper movement of the suspension system in response to dynamic loads is critical to ensure the optimal performance of the vehicle. This involves ensure the roll centre is adequately design with consideration of the trailing arm geometry. Further, the beam axle must be allowed to effectively rotate as it travels to allow the transfer of loads thought the suspension. This ensures that the rod ends and trailing arms are not placed in significant bending, which is to be avoided. Finally, the suspension system could be design to operate as required for dynamic events. Whilst under loads caused by acceleration, braking and cornering, the suspension is required to support the load of the vehicle whilst maximizing the tire contact. To allow this movement, trailing arms are required to be mounted to the frame and the rear axle which support the wheels.

This was the basis for the preliminary design and further development was achieved through iteration processes based on informed analysis.

3. 3 Design development

Beam axle

There were two options for the design of the beam axle, a single beam across the rear chassis connecting to the two-wheel uprights, or a double beam side by side which are smaller in diameter. The latter option was a method to reduce the space taken vertically to avoid any potential clashes in geometry, however, after incorporating the wheel diameters, it was found that there was enough clearance to just use one beam with a larger diameter.

The initial design consisted a 50 mm diameter beam with 3 mm thick walls. Multiple brackets were created for this design and attached to the beam geometry to start testing. There were two options to adjust toe as follows. The first one was to incorporate ball joints, which are two plates with ball bearings roaming free between them. For this purpose, the hub shaft would have to be thicker and too many parts would need to be altered for a simple toe adjustment on race day. Another possible option was the use of shims which were incorporated in the wheel hub because of ease of implementation.

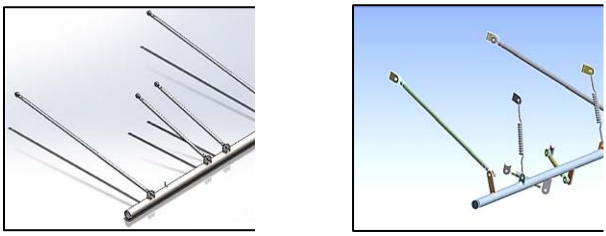
After testing, it was found that the diameter and thickness of the beam should be updated to 30 mm and 6 mm respectively to tackle the increased forces when under critical loads. The weight difference between the two beams were considered negligible relative to the increased strength of the beams. It should be noted that the mounting brackets have been tilted 30 degrees towards the front of the car to avoid obstructing the CV shaft when it under full compression.

Moreover, the weld-in bund was developed to allow integration between the upright and the beam axle. The weld-in bung is welded to the axle and to a bracket which is then bolted to the upright to support the wheel assembly. The rigid connection between the upright and the axle allows the tire contact forces to transmit through the wheel assembly and are dispersed through the circular tubing and suspension components.

Training arms

Figure 2(a) shows the preliminary geometry of the mild steel trailing arms attached to the beam axle, it used a parallel geometry for the arms and connected to the beam using bolts. The arms were imported to the main assembly for the vehicle and it was at this point a glaring issue was identified.

SolidWorks has a very convenient feature to observe points such as instantaneous centers and roll centers. Not that the roll center is the point where the loads experiences at the tire patches act on the sprung mass of the system. Its position with respect to the center of gravity dictates how much rolling moment the system will experience while cornering. Essentially, lowering of the roll center will result in a higher rolling moment while cornering and influences the turning radius of the vehicle. Further effects of a higher rear roll center include, higher responsiveness in cars when coming in and out of corners, and advantageous to use on high grip conditions, such as tracks, to avoid traction rolling. It was found that with the parallel arm geometry with equal lengths, the roll center remained close to the ground.

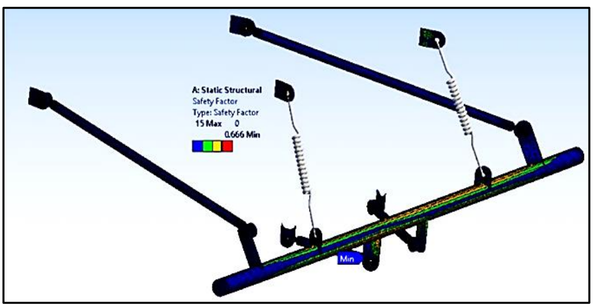


(a) (b)

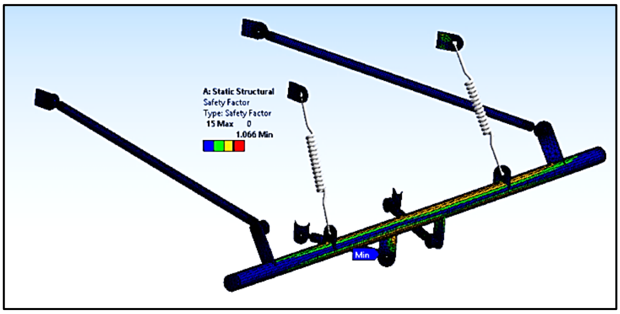
Figure 2 Trailing arm: (a) initial geometry and (b) final geometry

To combat this issue and have a higher roll center, the arms are angled in from the chassis to the beam axle. With the trailing arms angled in the design of the suspension geometry, any axial loads on the beam axle are transferred to compressive and tensile forces through the arms. This avoids undesirable bending within the trailing arms and provides a reduction in stress.

In addition to this, the lower arms are much shorter in length, causing a change in camber of the vehicle as it rolls, helping it to keep the contact patch square on the ground, increasing the ultimate cornering capacity of the vehicle. It also reduces the wear of the outer edge of the tire. FEA testing was performed on this short-long arms design, and it was found that cornering forces are higher and therefore stiffer bushings are required at the body. The arms have been designed with spherical bearings for all attachment points. This maintains all forces transferred in line with the bearings, eliminating unnecessary moments. With the thoughtful design of the suspension geometry, the trailing arms and rod ends used for the system are readily available, off the shelf components. With reduced stresses present, cost effective components can be utilized without compromising structural performance.



(a)



(b)

Figure 3 Safety factor of the space frame made of (a) steel and (b) Chromoly

Mild steel in the same wall thickness, as a Chromoly tube, is said to be almost half the strength in terms of tensile and torsional resistance. The high strength to weight ratio makes Chromoly desirable for applications where weight saving is essential, such as aerospace components and race car parts. It can easily be machined using conventional methods and the welding of the material can be performed by all commercial methods. The difficulty with Chromoly is with respect to its costing, especially for those where the budget is heavily constrained. Keeping this in mind, the design team recommends using Chromoly for the beam axle and the trailing arms, as this allows the component to have the required structural integrity while reducing the weight by 42%.

As shown in Figure 3(a), using the mild steel beam results in heavy stresses acting on the arms. The life cycle on this arms is said to be finite and this is not a desired outcome when designing a dynamic part with a great consideration to performance and safety, especially due to the fact that reusing the model in future years is a very desirable outcome. In contrast to the previous beam material, form Figure 3(b), FEA results indicate an infinite life cycle and a minimum safety factor of 1. Note that a spring was used in the FEA analysis to replace the shocks and replicate the reaction force counteracting the travel of the suspension during loading.

Bracket

When modelled with our critical forces, which is the loads calculated when the car is cornering, there are some stress concentrations on the edges of the bracket where the trailing arms are connected.

A stress concentration is defined as a high localized stress, compared to the average stress of the body, and is typically found in a region that has an abrupt geometric change. They will be in the small radii and sharp corners that are in a load path. The max von Mises equivalent stress was 311 MPa, which is well below the yield strength of Chromoly, but not low enough for our brackets to have infinite life. An investigation was undertaking to reduce the force flow around the notch to solve this issue.

As mentioned in ref (Wiley and sons), stress concentration factor, is a dimensionless factor that is used to quantify how concentrated the stress is in a material. It is defined as the ratio of the highest stress in the element to the reference stress, the graphs provide minimum radius lengths of a fillet when there is a connection between varying diameters. As presented inFigure 4, incorporating this concept of fillets smooth out the stress flow lines and along with applying a concentrated mesh at the max point provided more accurate and improved values. The new von Mises stress was 91 MPa, well below the yield strength and therefore had an infinite life cycle, rendering this design iteration as very successful change.

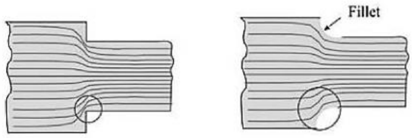


Figure 4 Stress concentration diagram

The water jet cutting was used for the brackets because it offers manufacturers flexibility that no other cutting process can offer. It is preferred over CNC milling as it is a more accurate form of manufacturing and this is vital as the filleted corners need to be as smooth as possible to mitigate any stress concentrations.

Shocks

The spring and damper are mounted directly to the chassis of the vehicle, and this configuration reduces unsprung weight and improves the response of the suspension system. The suspension system uses coil over springs where the springs are mounted to the outside of the damper with an adjustable preload on the spring to adjust the ride height of the car. This allows the design to be attentive on performance over comfort and directs the optimization of the handling of the vehicle. With the spring stiffness increasing the stiffness of the suspension coupled with a low center of gravity, the vehicle can be controlled significantly easier.

3. 4 Final design

Depicted below in Figure 5, the rear suspension system is designed within FSAE and Australian Standards and adheres to the predefined constraints and limitations. It is structurally sufficient and thoughtfully developed to ensure adequate scoring during FSAE static and dynamic events. As mentioned before, the final design features 30mm diameters Chromoly beam axles with a tubing thickness of 6mm. It is equipped with direct acting Penskie 7800 shocks with spring stiffness of 57 N/mm. The trailing arm geometry is positioned to optimize the position of the roll center to allow better performance of the vehicle.

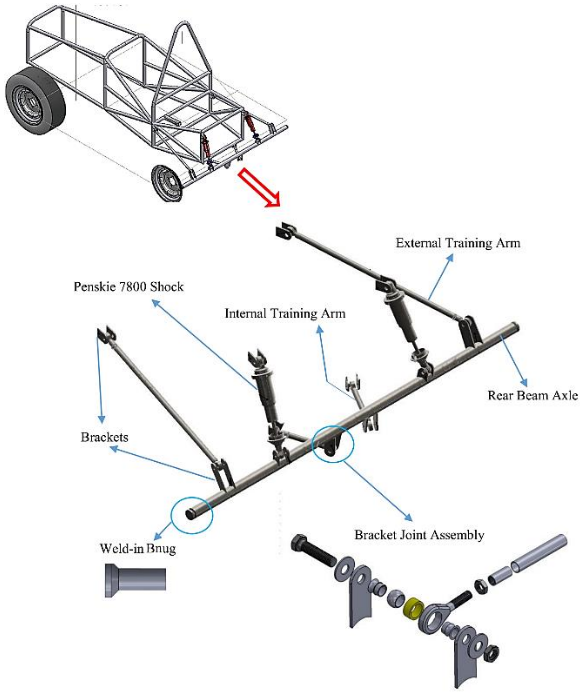


Figure 5 Final suspension design schematic

4 NUMERICAL ANALYSIS

Before manufacturing processes, Finite element analysis (FEA) can be used to represent the complex geometry and simulate the forces transmitted to the model with equivalent boundary conditions. In this way refinements can be made and rapidly assessed for a small fraction of the cost of prototyping and experimental testing. The FEA method is utilized through ANSYS static structural modelling of equivalent stress and total deformation in the vehicle's rear suspension.

Moreover, since the rear suspension components are critical elements of the vehicle system which are subjected to a cyclic loading, a fatigue analysis must be conducted with realistic endurance limit modifying factor. It was conducted in ANSYS using the fatigue tool which analysed the stress life using the Goodman equation. The cyclic loading generated by the load cases is equivalent to zero based loading for purely linear acceleration and braking and fully-reversed for the critical cornering case, with the endurance factor of 0.67 (see Appendix B) calculated from ref [16]. Given the material properties of each suspension component, this analysis provides an expected life and safety factor of the mechanism based on the limiting factors specified. To design the components for infinite life, the suspension is expected to exceed 10^6 loading cycles.

4. 1 Linear Acceleration

With the forces developed for the linear acceleration load case, the structural performance of the rear suspension was analyzed in ANSYS. Figure 6 demonstrates the stress profile throughout the beam axle and trailing arms, identifying the maximum stress location at the rod ends. The maximum Von Mises stress recorded in the rear suspension was 174.15 MPa. With the minimum fatigue limit of materials used for the components being 220 MPa for mild steel, the safety factor of the system is 1.26. Therefore, it is expected that under linear acceleration the rear suspension would operate for infinite life.

From the stress concentration, it is clear that the beam axle has area of high stress, specifically around the bracket mounting locations. However, with the beam axle tubing constructed from 4130 Steel, the strength is more than sufficient to withstand these stresses. In addition, from the FEA analysis, the vertical travel of the suspension during linear acceleration was determined to be 28.12 mm. This resulted in a counteractive spring force of 1602 N provided by the shocks.

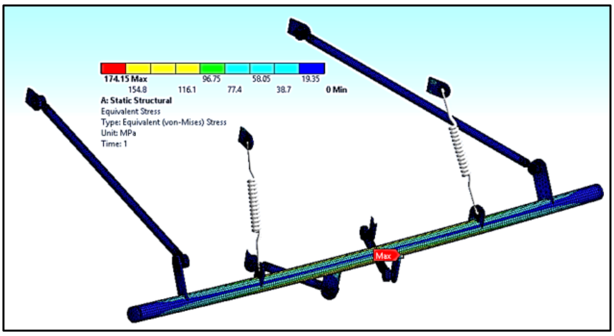


Figure 6 Stress concentration for linear acceleration load

4. 2 Braking

With the weight transfer towards the front of the vehicle during braking, it is expected that the loads and resulting stress in the rear suspension is reduced, when compared to acceleration. As proposed in Figure 7, the resulting stress profile identified a maximum von Mises stress of 157 MPa, located within the rod ends.

With similar loads applied, the resulting stress profile resembled that of the linear acceleration cases, with only a minor reduction in stresses. With the maximum stress significantly below any of the materials fatigue limit, the safety factor of the suspension system during braking was a minimum of 1.4. Consequently, all the components are operating within the limit of infinite life. Since the maximum stress significantly below any of the materials fatigue limit, the safety factor of the suspension system during braking was a minimum of 1.4. Consequently, all the components are operating within the limit of infinite life. Under this load case, the vehicle travel of the suspension was measured as 9.24 mm, with a resulting shock force of 527 N.

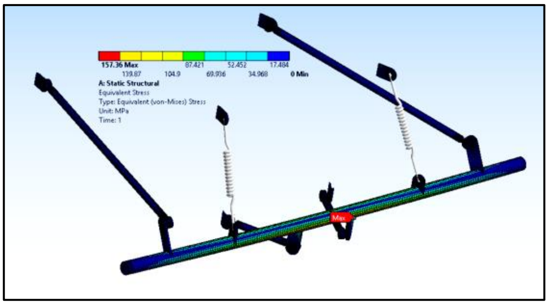


Figure 7 Stress concentration for braking load

4. 3 Critical Cornering

The final and most critical scenario considered for analysis was maximum cornering whilst accelerating, based on the load case previously developed for high-speed cornering situations. The stress profile for the suspension under this loading configuration is displayed in Figure 8.

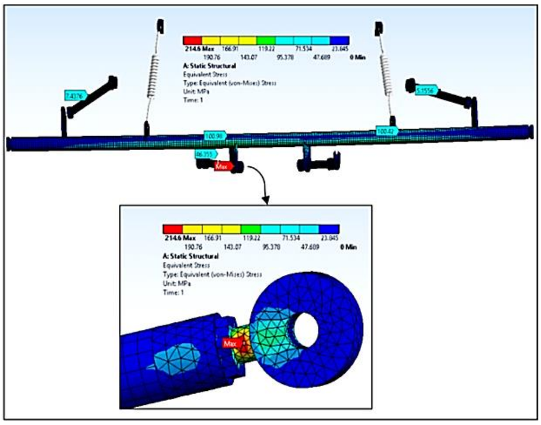
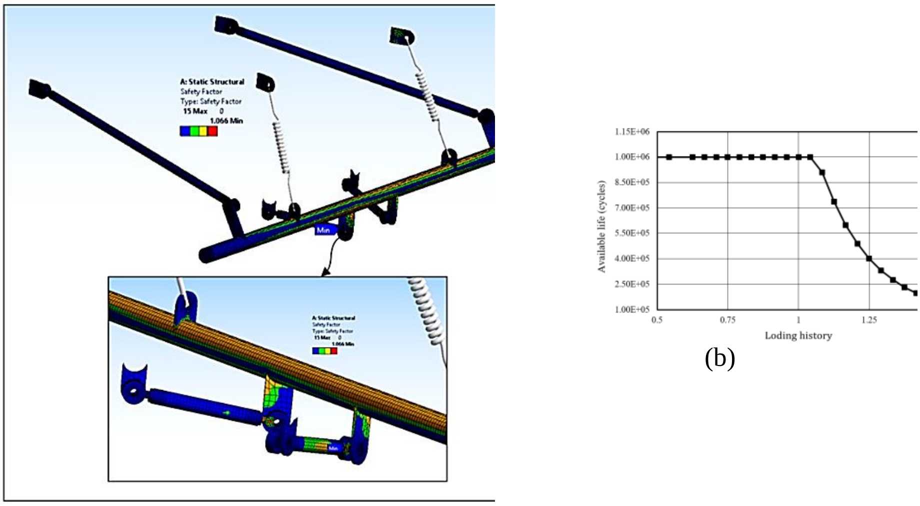


Figure 8 Stress concentration for critical cornering load

With stress probes used to demonstrate key values. This figure demonstrates the stress concentration along the Chromoly beam axle and within the trailing arms. With the maximum stress of 214.6 MPa identified in the rod ends, the other stress concentrations highlighted along the beam axle are significantly lower. In addition, with the use of a spring probe, the travel of the suspension for critical cornering was determined to be 62.27 mm, producing a force in the shocks of 3549 N.

Figure 9(a) demonstrates the safety factor of a number of suspension components, with the minimum safety factor identified in the rod ends of 1.066. For the beam axle, the safety factor ranges from 1, in areas of high stress, to 15, where the stress concentration is minimal. In addition, it was observed that the trailing arms has limited stress throughout them, leading to a significantly high safety factor approximately 15, for the majority of the length. Further analysis of the rear suspension under critical cornering involved a fatigue sensitivity analysis for the given loads. Form Figure 9(b), it was identified that for the critical cornering loading conditions, the suspension system was operating within the limit of infinite life for all components.



(b)

(a)

Figure 9 (a) Safety factor for critical cornering load, (b) Fatigue sensitivity graph

Through the design iterations, the mounting brackets on the beam axle and frame have been considerably angled. This ensures that the rod ends are not misaligned at a static position, but rather directly link the frame and the rear axle. This rotation of the brackets also ensures there is 12 degrees of allowable misalignment in the spherical bearing of the heim joint. This allows effective movement of the suspension system and prevents clashing of components during dynamic loads and suspension travel. Using ANSYS, the critical bolted joint was found to be withstand the forces applied while under critical loading. The M10x50 mm grade 8.8 high tensile bolt was analyzed with a 21.9 kN pretension to suit an assembly torque of 44Nm. The performance of the bolt and joint can be seen in Figure 10. The maximum stress on the bolt is 57.16 MPa which is significantly below the yield strength of the bolt which is 640 MPa.

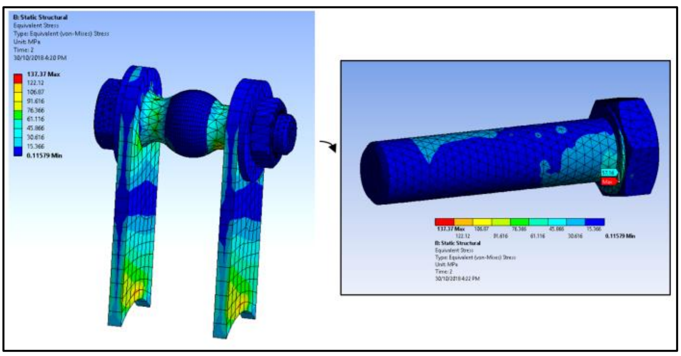


Figure 10 Bolt performance

5 CONCLUSION

This paper provides key considerations for designing and analyzing rear suspension system of a Formula SAE vehicle produced by James Cook University students. They were addressed appropriately because of the successful performance of the designed vehicle during driver's training and at the 2019 Formula SAE competition. With the proposed suspension which is a dependent one and based on the finite element analysis, the stress concentration of the proposed design is significantly low under different load cases. The dependent system transfers the load transmitted through the larger beam axle and consequently has a lower maximum stress. It is shown that since the beam axle is mounted to the upright by means of the weld-in bung, the contact surface area for the transmitted forces is significantly large. This ultimately leads to lower stress values present resulting in a higher safety factor. Utilizing a material with a high strength to weight ratio, Chromoly, allows the overall size of the beam axle to be reduced to avoid clashes with components. With the implementation of Chromoly as a substitute for mild steel and thicker material utilized, the safety factor and life cycle are significant high; under considered load conditions the proposed suspension system would operate for infinite life. With the trailing arms angled in the design of the suspension geometry, the undesirable bending within the trailing arms is avoided and a reduction in stress and higher roll center are provided. Moreover, it is proposed that the stress concentration in brackets with filleted corner is considerably lower than those with sharp corners, which means better stress distribution in the component and consequently higher life cycle for the whole system.

APPENDIX A

Forces acting on the rear suspension for linear acceleration load case:

$$\text{longitudinal weight transfer} = \frac{\text{acceleration}\times\text{COM height}\times\text{weight}}{2\times\text{wheelbase}} = 42.94\ \text{kg}$$

$$\text{vertical weight transfer} = \text{weight transfer} + \text{weight distribution} = 42.98 \times 9.81 + 1107 = 1528.28\ \text{N}$$

$$\text{friction force} = \text{vertical force} \times \text{friction coefficient} = 1528.25 \times 0.8 + 1107 = 1222.4\ \text{N}$$

Forces acting on the rear suspension for braking load case:

$$\text{longitudinal weight transfer} = \frac{\text{acceleration}\times\text{COM height}\times\text{weight}}{2\times\text{wheelbase}} = 42.94\ \text{kg}$$

$$\text{vertical weight transfer} = \text{weight transfer} + \text{weight distribution} = 1107 - 42.94 \times 9.81 = 658.76\ \text{N}$$

$$\text{friction force} = \text{vertical force} \times \text{friction coefficient} = 658.76 \times 0.8 = 548.61\ \text{N}$$

APPENDIX B

Endurance limit modifying factor for fatigue analysis:

$$endurance\ limit\ factor = k_a k_b k_c k_d k_e k_f$$

k_a: *surface modification factor*	$k_a = a \times S_{ut}^{b} = 4.51 \times 460^{-0.265} = 0.888$
k_b: *size factor*	$k_b = 1.24(0.37 \times d)^{-0.107} = 1.04$
k_c: *loading factor*	$k_c = 1$

$k_d: temperature\ factor$ $k_d = 1$ (temperature is below 400℃)

$k_e: reliabity\ factor$ $k_e = 0.753$ (reliability is assumed to be 99.9%)

$k_e: miscellaneous\ effect\ factor$ $k_f = 1$

$$\rightarrow\ endurance\ limit\ factor = 0.67$$

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