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Design and analysis of brake system for FSAE race car

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Keywords: vehicle dynamics, brake system, topology optimization, thermal analysis, MATLAB, CAD modelling, FEA

Abstract

The objective of this research paper is to design, simulate and compare components of a race car braking system. The Racecar is an FSAE car that is designed around the rules and regulations of the FSAE rulebook, the main aim of this project is to make components lightweight and improve their performance as compared to their OEM counterparts. The braking system involves the mathematical calculation of pedal ratio, brake torque, heat generated in brake discs, and required clamping force using MATLAB to achieve peak deceleration after which these values would be used to design and simulate the components. The paper presents some innovative new ideas applied in an FSAE car and also involves designing techniques like topology optimization which was done using Altair inspire. Finally, all the components were designed in Solidworks, and simulations like Factor of safety, von mises stress, and strain were performed using ANSYS 18.1. The paper also compares the work of other authors as well and explains the differentiating factors between our and their design.

Nomenclature

w	weight of the vehicle in N
Φ	weight distribution at the front axle
μ	coefficient of friction of tire
μp	coefficient of friction between pad and disc
d	Deceleration in G's
R	radius of Tire
h	Height of CG
l	Wheelbase length
C	Clamping force
Ar_{cp}	Area of per piston
F_{mc}	Force on Master Cylinder
F_{app}	Force applied
B	Brake Torque
P	pressure in brake lines
N_t	normal load on one tire after weight transfer
F_t	Tangential force
R_d	radius of the brake disc
n	Number of pistons in the Caliper
FSAE	formula society of automotive engineers
CAD	computer-aided design
OEM	original equipment manufacturer

1. Introduction

Brakes are a crucial part of any automobile, the performance of automotive braking systems are subjected to high loads during short intervals. For designing a race car brake assembly there are various technical requirements, such as the components must be as lightweight as possible for improving acceleration or deceleration and better handling characteristics, since we are concerned with braking all the components should be designed or selected to operate at peak deceleration. A racecar brake system consists of several components such as the brake pedal, brake lines, calipers, discs, they should be strong enough to withstand high structural and thermal loads and deliver peak performance without compromising reliability. These components need to be designed in such a way to ensure they are feasible to manufacture, easy to access and assemble. If optimized properly small improvements in weight and braking distance can reduce lap times in seconds.

In this paper, the design and analysis of a race car braking system are being discussed. The race car is an FSAE car that is designed around the rules and regulations for FSAE. The braking system involves the mathematical calculation of max achievable deceleration, pedal ratio, brake torque, and required clamping force to achieve peak performance after which the actual designing and simulation part starts. The paper also presents some unique features such as the dual pedal ratio brake rail, allowing the driver to have a wider range of pedal feel and deceleration rates depending upon their deriving styles. New design techniques like topology optimization have also been used, topology optimization is a computational methodology that produces such organic-looking shapes intended to optimize the component layout within given space constraints for a given set of loading and boundary conditions to maximize the performance of the component.

2. Literature review

Various research papers related to designing and analysing various parts under the Braking system were studied and discussed below. In some papers, the author has focused on how to calculate brake pedal geometry and how to design a pedal box for an FSAE car keeping in mind driver safety, ergonomics, performance, and manufacturability [1]. Some papers worked on the design and analysis of pedal assembly for an FSAE car and helped to identify various design constraints for designing the pedals, such as maintaining proper distance between each pedal from an ergonomics and ease of assembly point of view [2]. In a paper published by the University of Akron [3], the author worked on determining the maximum deceleration then designing components like pedal assembly and brake disc, a detailed explanation was also provided by the author for each step and challenges for designing the pedal box and brake disc for an FSAE car was also explained. For the master cylinder and brake caliper decision matrices were made and factors for selecting these components was also explained.

Designing a two-part custom caliper is not new among formula student teams reason being, easy to design and manufacture. According to Sergent, N *et al* [4] calipers must be stiff, light and heat resistant Brake caliper requires acceptable stress and deflection under multiple load. A slight crack can result in instant brake fluid leak and caliper malfunction therefore, a stiff design provides better load distribution which ensures shorter brake pedal travel, ride quality and vehicle safety. Various papers, explain the process of designing and manufacturing was explained by Ingale *et al* [5], such as the type of seals, materials, and components used to make a custom caliper. Nikhil Pratap Wagh [6] in their paper explained the design and analysis process of a brake caliper, the paper describes FEM analysis such as structural and thermal which showed various stress points and hinted at possible ways of failure as well. Paper presented by Golhar, S. P *et al* [7] designed a hydraulic brake caliper for ATV Racing competitions. The proposed design was a 2-piston monoblock caliper. The result of this was an optimized brake caliper that was lightweight and provided adequate braking torque for efficient braking in a hostile racing environment.

Research paper by Kush Soni *et al* [8] the authors discussed various calculations for brake force, brake torque, brake disc, and brake bias. Also, caliper selection and design and analysis of brake discs are performed. Mohd. Usama, Gagan Singhal [9] gave an insight on developing a desired braking system for a high-performance race car for the competition. Various calculation like brake forces, clamping force, and heat dissipation by disc is discussed in this paper.

While designing brake discs various points have to be kept in mind as explained by Thuresson, A. [10] the paper explains different modes of failure one of the common reasons being improper geometric design which gives rise to thermal judder and results in failure due to conning, butterfly or corrugated effects. the paper also explains detrimental effects of hot spots which is a result of non uniform heat distribution in the disc which causes stress concentration and ultimately failure.

2.1. Knowledge gained and gaps in the literature

After reviewing the literature, the following information was obtained

- Various conditions that govern the designing of the Braking system were understood and the analysis procedure was also studied.
- The braking system is very closely related to the safety of the automobile and hence must be designed to uphold even in the worst possible scenarios.
- All the necessary calculations involved in the design were also reviewed.

The aforementioned works despite being informative on various topics but lagged in the following departments

- Designing for better packaging
- Cost-effective development of parts
- Optimizing Design for lighter parts/assemblies
- Proper mathematical calculations for each of the components
- thermal simulations for disc lagged stress inclusion as that would have provided more realistic results.

Hence, the objective of this paper is to design and develop a lightweight assembly as compared to existing OEM brakes components, to improve a race car's performance by reducing the unsprung mass which would improve the handling of the car. This would also reduce the polar moment of inertia which would make the car more agile on turns and hence improve lap time.

2.1.1. OEM parts pose the following problems

- Generally Heavy
- Require additional packaging space (are Bulky)
- Are not flexible according to designers preferences
- Some are not cost-effective when compared to manufacturing in-house.

Following the aforementioned points, the designing and analysis of the parts will be done with the following objectives

- To reduce the overall weight of the Braking System
- To improve on the existing braking systems used on Formula Student racecars
- To provide consistent braking.

3. Methodology

The design process for the braking system was initialized with the acquisition of various parameters like wheelbase, CG *(Center of gravity)* location, coefficient of friction, etc (table 1). This is followed by calculation and various steps as shown below

As shown in the above flowchart (figure 1) after getting the max deceleration, the braking force and torque are computed by taking the maximum permissible brake disc radius where the caliper maintains a gap of 5 mm from the wheel rim. For the brake caliper, the aim is to design a 2 piston caliper for the rear axle of the car so calculations, assumptions and disc designs are explained only for the rear axle. After getting the disc radius the required clamping force, pressure in brake lines and pedal ratio can be calculated. these calculations were done by keeping in mind the FSAE rules and various other constraints such as geometric and human limitations which are denoted as system constraints in the above flowchart. Once the calculation part is over material selection is done based on a performance matrix and finally, CAD modelling and FEA simulation are performed to ensure the working of the design within safety limits. To prove the benefits of the designs presented in this paper a comparison is also done with other existing designs.

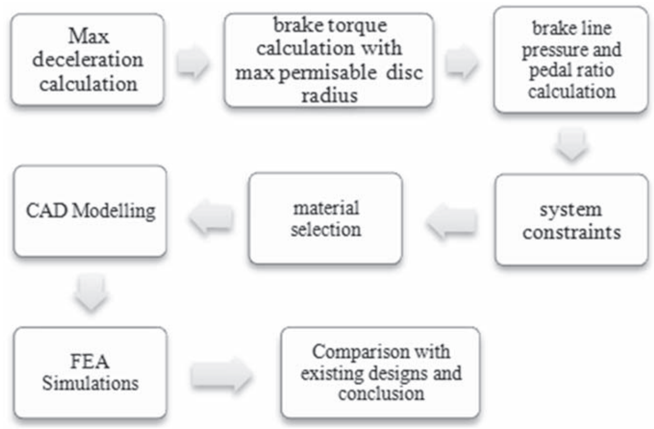


Figure 1. Design process flowchart.

Table 1. Vehicle parameters.

Parameter	Value
car weight	300 kg
static weight distribution *(Front/Rear)*	47/53
Tire Radius	0.235 m
CG height	0.3 m
Wheelbase	1.55 m
Tire coefficient of friction (dry condition)	1.6
coefficient of friction between pad and disc	0.4

3.1. Calculations

3.1.1. Max deceleration calculation

Maximum Deceleration is an important variable in the designing of a brake system as it determines the amount of load transfer and hence quantifies the amount of clamping force that will be required on each wheel assembly.

Since a tire is not a rigid structure it is a deformable body and has a chemical adhesion with the track surface so the variation of coefficient of friction with normal load isn't linear. Therefore, to compute the max deceleration the below Simulink model (figure 2) was used. In this model, the tire data is entered as provided by the manufacturer then the continuous loop is iterated until we get our max deceleration value. This loop includes equations for weight transfer & corresponding lateral and longitudinal forces transform functions [11] which further gives the maximum achievable deceleration value of 1.6 g's.

$$Weight\ \ Transfer\ \ (Wt) = (w \times d \times h)/l \tag{1}$$

$$Braking\ \ force\ \ (at\ \ front/rear\ \ axle) = (((1 - \Phi) \times w) \pm Wt)) \times \mu \times 9.8 \tag{2}$$

3.1.2. Brake torque and disc radius calculations

Brake torque specifies the amount of torque required to stop the tire from rotating and thus bring it in locking condition. Disc Radius helps quantify the amount of clamping force to be generated by each caliper, as greater the radius lesser the clamping force. There are some necessary calculations required to set boundary conditions for the disc, firstly to find the brake disk and then the radius of the disc. The brake disc radius was selected by taking the maximum permissible radius where there is at least an 8 mm gap between the 10-inch rim and the closest brake caliper surface. By doing so we got:

outer diameter $=$ 166 mm

effective disc radius $(R_d) =$ 146 mm, disc width $=$ 40 mm

$$Nt = \frac{(w \times \phi) \pm Wt}{2} \tag{3}$$

$$(Brake\ \ Torque)\mu p \times C \times Rd = \mu \times Nt \times R \tag{4}$$

From equations (3) and (4) [12] we also get clamping force for the caliper and disc simulation

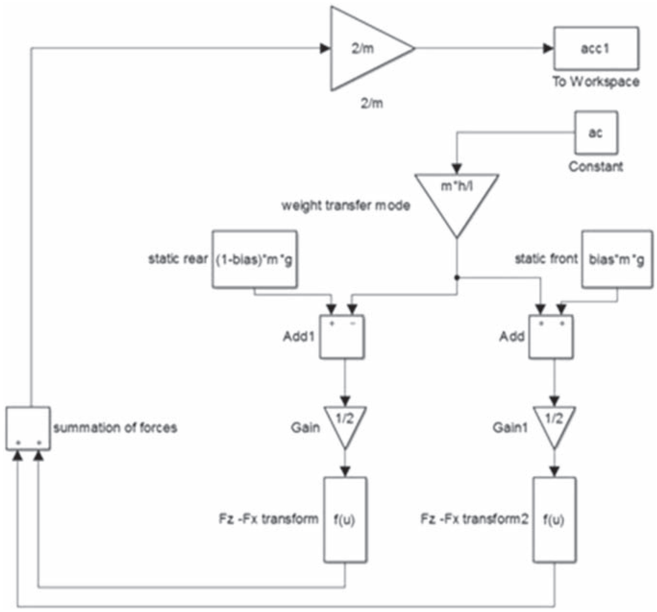


Figure 2. Simulink model for maximum deceleration.

3.1.3. Brake pedal geometry design

Since the amount of force required on brake pedal is very high we require a leverage mechanism to multiply the input force hence pedal ratio geometry is introduced to multiply the input force so for ex- if the pedal ratio is set at 5 it means that the output force is 5 times the input force. There exist various arrangements in which the brake pedal, Master cylinder, pivots, etc can be arranged. In this case, we decided to use an angled arrangement of the pedal geometry(a basic arrangement is shown in figure 3(b)) as it is very compact and lightweight as compared to other arrangements as explained by *Puhn F* [12], But it's one disadvantage is a varying pedal ratio with travel. It is necessary to minimize the deviation as even a slight deviation in the pedal ratio will affect the input force of the master cylinder eventually leading to inconsistent braking force. Various arrangements were iterated using CAD software and MATLAB to reach the decided values of pedal ratio and then micro-adjustments were done to minimize the deviation in the pedal ratio.

The final decided angled arrangement of geometry mentioned in figure 3 was selected as it provided the following benefits

- Compact Packaging
- Minimal deviation in the value of Pedal Ratio Iterations of various pedal geometries are done with the MATLAB code shown in figure 4 to help better visualize the change in the Pedal ratio and deviation in its values. According to *Limpert, R.* [11], a driver can comfortably generate an input pedal force of 445 N. So the value of 445 N was taken as input force to approximate the pedal ratio value to generate the required brake line pressure for producing the calculated clamping force (equation (4)) to generate max deceleration of 1.6 g's. The mathematical formula used for calculating the pedal ratio is shown below.

$$Pedal\ Ratio = \frac{L \text{ x } mcl}{b \text{ x } m \text{ x } \sin(C)} \tag{5}$$

As shown in figure 3(b) in equation (5) [12], L = length of pedal.

Mcl = master cylinder length

b = distance between pivot point and balance bar

m = distance between pivot point and master cylinder C = angle between b and m

After various iterations, two geometries were selected, the graphs of pedal ratio versus angle of travel for the two geometries are shown in figure 5, which shows the deviation achieved between max and min values of pedal ratio when the brake pedal is fully actuated. Two pedal ratios are shown to incorporate the concept of dual pedal ratio discussed in the upcoming section. Following are the graphs showing the deviation achieved after the brake pedal is fully actuated.

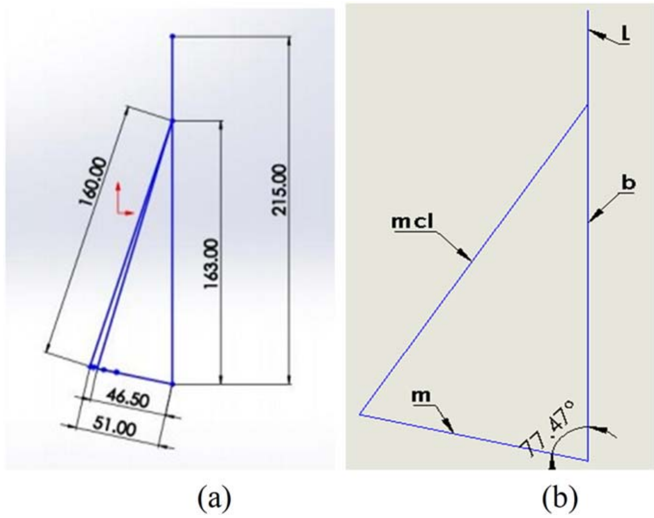


Figure 3. Brake pedal geometry (a), labelled diagram of pedal geometry (b).

```
clc
clear all
l=input('length of pedal-');
db= input('distance between pivot and bias bar-');
dm=input('distance between pivot and master cylinder-');
ai= input('initial angle-');
af= input('final angle-');
a=(ai:-0.01:af);
A=(a/180)*(22/7);
mcl=sqrt(dm^2+db^2-2*dm*db.*cos(A));
pr=l.*mcl./(db*dm*sin(A));
disp(pr)
plot(a,pr)
xlabel('Angle of pedal in degree');
ylabel('Pedal ratio')
```

Figure 4. MATLAB code for pedal ratio iteration.

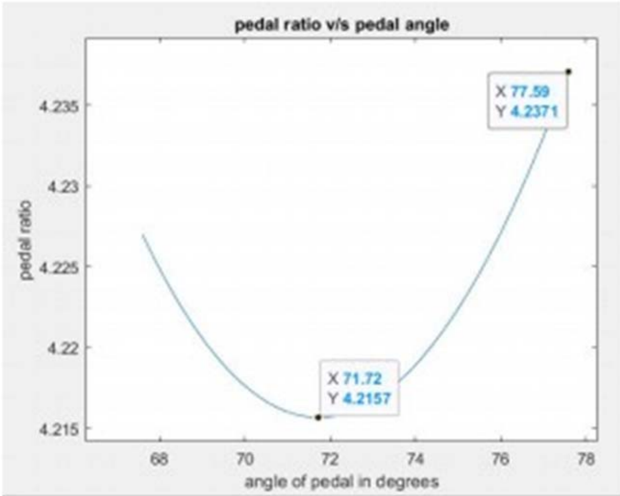
3.2. Material selection

The factors influencing the material selection for Brake pedal and Caliper are:

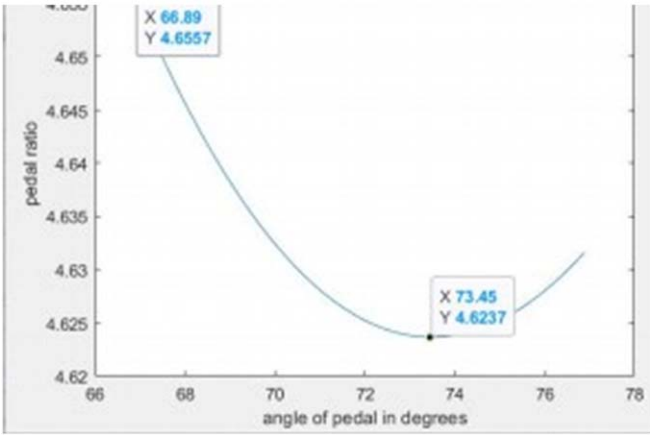
- Density: Affects the weight of the component
- Yield strength: Defines the strength of the component and hence the load-bearing capacity
- Machinability: Affects the cost of machining and eventually manufacturing.

And for the brake disc the required properties are:

- Strength: Defines the strength of the component and hence the load-bearing capacity.
- Thermal Conductivity: the property of a material to allow the flow of heat and disperse it in surrounding air.
- Specific Gravity: affects the weight of the component
- Wear Rate: defines volume lost per unit distance, which is important for brake discs.
- Specific Heat: defines the amount of energy required to raise the temperature by one degC.
- Friction Coefficient: it is the coefficient of friction between the disc material and brake pad the higher the value the better the material.



Pedal Ratio:- 4.2,Deviation :- 0.0214



Pedal Ratio:- 4.6,Deviation :- 0.03

Figure 5. Pedal ratio versus angle of travel graph.

Based on these factors a Performance matrix was made to compare the materials, each material property was scaled out of 100 and summed up to get the final score. to better represent these performance matrix graphs are shown below.

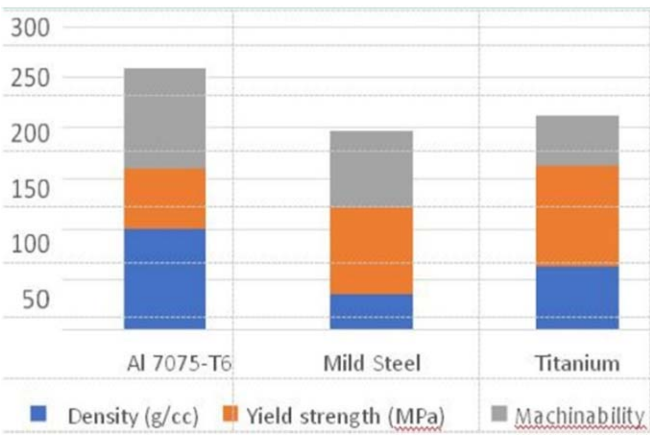


Figure 6. Material selection matrix for caliper and pedal assembly.

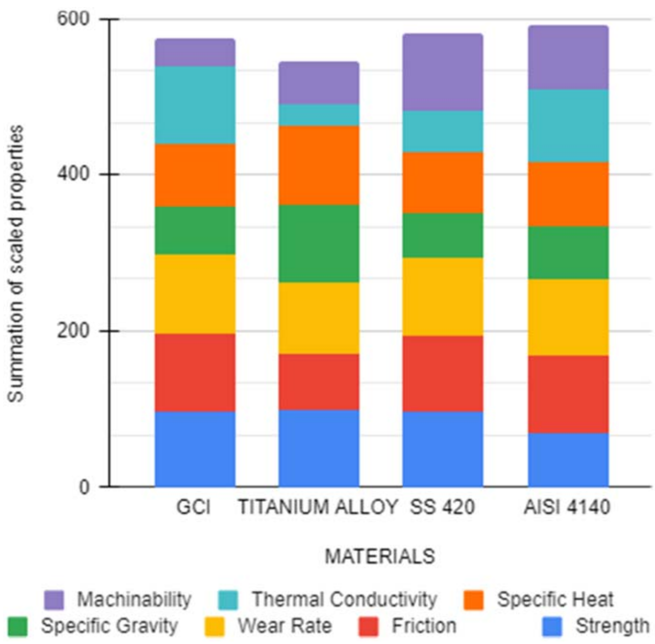


Figure 7. Material selection matrix for brake disc.

The above graph (figure 6) was used to select materials for the brake caliper and pedal assembly, three potential materials were selected namely Al 7075 T6, mild steel, titanium. Titanium has a better strength-to-weight ratio which would make it an ideal material for the application but machining of titanium is difficult and costly, therefore the decision matrix helped to conclude that Al 7075-T6 was the best material available to manufacture the Brake Pedal and Caliper. Similarly, the factors that influence the material selection of Brake Discs were noted and a comparison matrix was made from this decision matrix, we saw two materials that were equally good which were GCI(grey cast iron) and AISI 4140 (figure 7), for brake disc application GCI would have been a better option because of its slightly higher specific heat and conductivity value, but machining of GCI is a bit difficult because of its relatively higher hardness and low toughness. Thus, it was concluded that AISI 4140 was the best material available for the Brake Discs.

3.3. Pedal assembly design

The main objectives behind designing the Pedal assembly were

- To mount the assembly on a chassis member instead of a baseplate for better stress distribution
- To make the overall assembly lighter

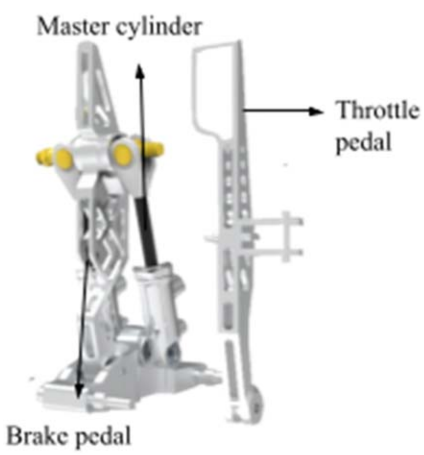


Figure 8. pedal assembly.

- To implement the concept of a dual pedal ratio
- To enhance adjustability
- To be cost-effective

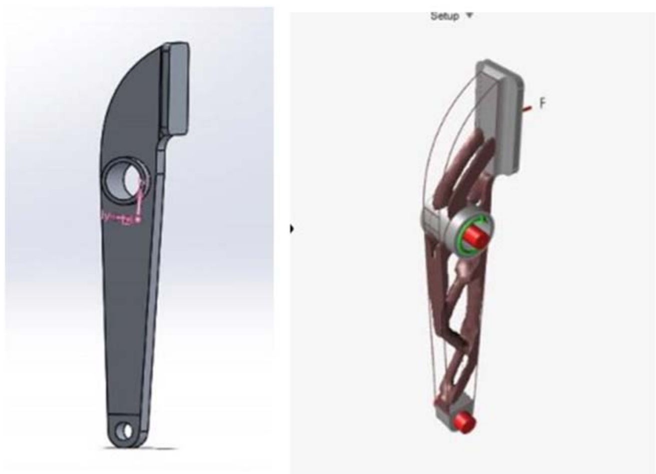
The pedal consists of brake and throttle pedal as shown in figure 8, clutch pedal is not shown as the race car uses pneumatic shifting but for comparison purposes clutch pedal is also considered in a three-pedal assembly. The overall weight measured in CAD software (Solidworks 2018) of the finished custom assembly was found to be **676 grams** (not including the tabs required to mount the assembly to the chassis, master cylinder, and retraction springs). The process of designing and simulating is explained in the upcoming sections which started by computing the pedal ratio as explained in section 3.1.3.

3.3.1. Brake pedal design

The decided Pedal Geometry from figure 6 is used as the mainframe to design the Brake pedal. According to the FSAE rule book, the brake pedal should be able to withstand a force of 2000N so the design needs to be strong enough to bear this load and have sufficient stiffness, some other Constraints while designing the brake pedal were:

- Pedal length is decided according to Driver's feet length
- Sufficient space is for the fit and assembly of the Brake Bias.
- Sufficient strength and stiffness to withstand 2000N of force
- feasible for manufacturing.

Following the above points, various iterations were made, and ultimately, a general structure was designed, on which topology optimization was done using Altair Inspire Software to obtain the above-shown result in figure 9. By performing topology optimization [13] we were able to observe the load paths (figure 9(b)) and provide strength in those regions while keeping the weight as minimum as possible. Since the topology optimized result is non-machinable it would require advanced manufacturing technologies like additive manufacturing (3D printing) which are very costly especially considering Metal 3D printing. So it was decided to study the structure of the topology optimization and embody it into a machinable form. Thus, resulting in a finalized brake pedal shown in figure 10. The final pedal made from Al 7075 T6, a detailed decision matrix for material selection is explained in a later section, the final pedal weighted only **125.14 grams**.



(a) (b)

Figure 9. Initial pedal structure(a), Topology optimized result(b).

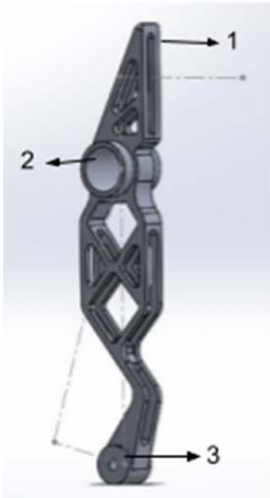


Figure 10. Finalized Brake Pedal.

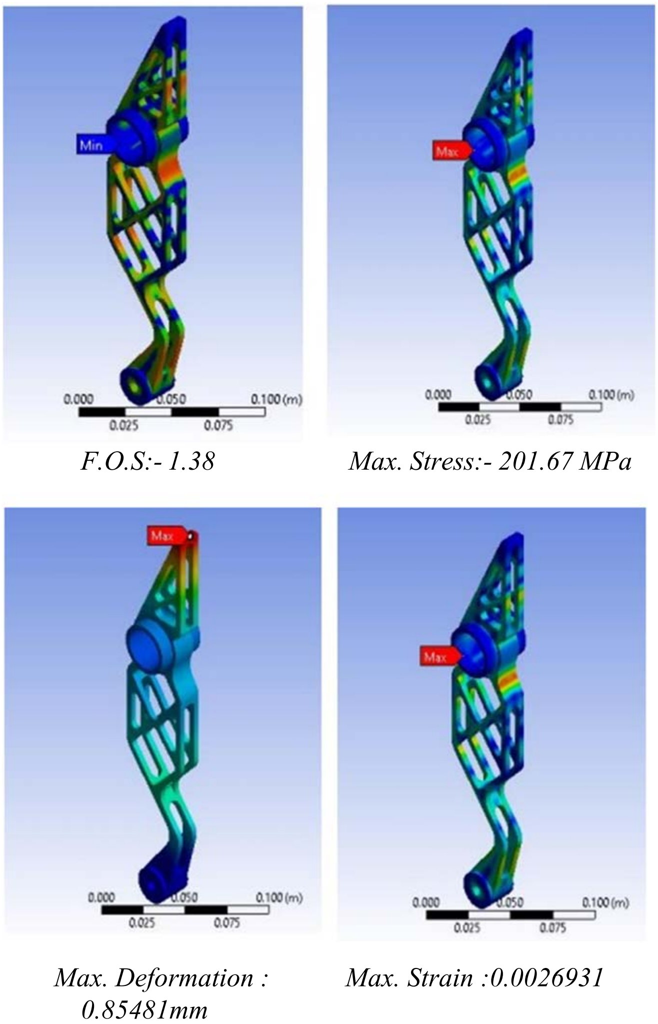


Figure 11. Structural simulations.

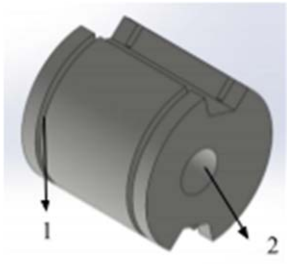


Figure 12. Eccentric(1- circlip groove,2- master cylinder mounting point).

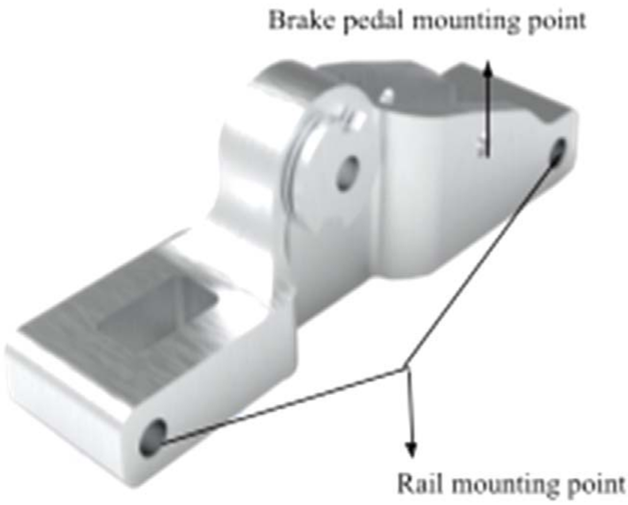


Figure 13. Assembled Brake Rail.

as shown in figure 10 various positions have been labelled in the picture which represent the following points:

1. pedal face
2. master cylinder mounting point
3. pedal pivot point

3.3.1.1. Simulation

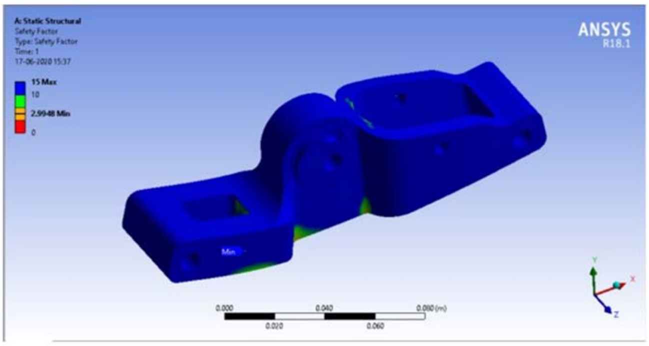
The pedal is simulated by applying a force of 2000N (as mentioned in the FSAE rule book) on the pedal face and giving the boundary conditions of Cylindrical support for balance bar bearing and fixed supports at the bottom to mount the pedal. The following results are obtained for the F.O.S, stress, deformation, and strain simulations (figure 11). From the simulation results, we can see that because of topology optimization the stress is distributed all over the pedal body and there are no stress concentration points which leads to higher fatigue life as well; there are very few blue regions which proves the effectiveness of the technique and allow the pedal to be so lightweight.

3.3.2. Brake rail design

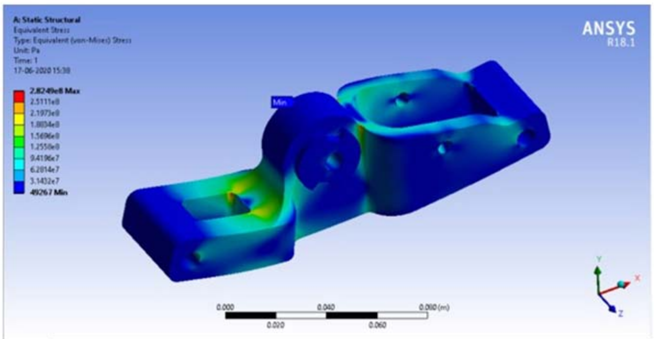
Presently F1 cars have advanced technologies that allow the Pedal ratio to be changed actively. This was taken as an inspiration to design the Brake rail. The availability of multiple pedal ratios allows the driver to have a preference over the spectrum of deceleration that he will experience while braking based on driving style of different drivers. Thus to initialize the concept of multiple pedal ratios two-pedal ratios were decided as shown in figures 3(a) and 5 which shows the two pedal geometries and graph of pedal ratio deviation versus angle of travel respectively. An eccentric was designed to act as a pivot point for the Master Cylinder and provide room for the pedal ratio adjustment. Pedal Ratio can be changed by adjusting the Eccentric(figure 12) in two different fixed positions. The lateral movement of the Eccentric is eliminated by providing grooves for Circlips. This way of implementation of multiple pedal ratio was not only effective but also lightweight and much more compact than an F1 car's assembly. This flexibility in braking performance aims to accommodate different drivers with different braking styles resulting in more consistent, precise, and controlled braking. The final assembly is shown in figure 13.

3.3.2.1. Simulation

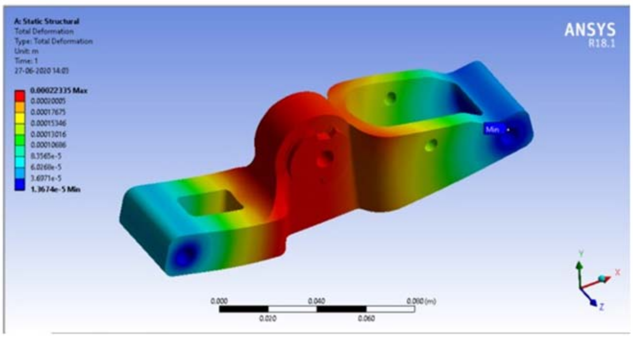
The force calculations for the Brake Rail are done by using the Pedal ratio and the boundary conditions applied were fixed support at the mounting points and an input force of 9200 N at the master cylinder mounting point at the eccentric. Static structural analysis was performed to get a measure of FOS, stress, and deformation (figure 14).



F.O.S: 2.9



Max. Stress : 1.625 x 102 MPa



Max. Deformation : 9.148 x 10-5 mm.

Figure 14. Structural simulation brake rail.

3.3.3. Throttle and clutch pedal design

The throttle pedal was designed under the following constraints

- Pedal length should be decided according to the driver's feet length
- The pedal should be lightweight
- Provide sufficient stiffness to avoid flexing in extreme conditions
- Should provide sufficient torsional strength as the cable is attached to a bracket on the pedal

After iterating the design of pedals it was noted that many throttle pedals possess similar profiles in side view.

On doing a structural analysis it was seen that the said profile provided a good load/stress transfer path and hence was incorporated in the design(figure 15).

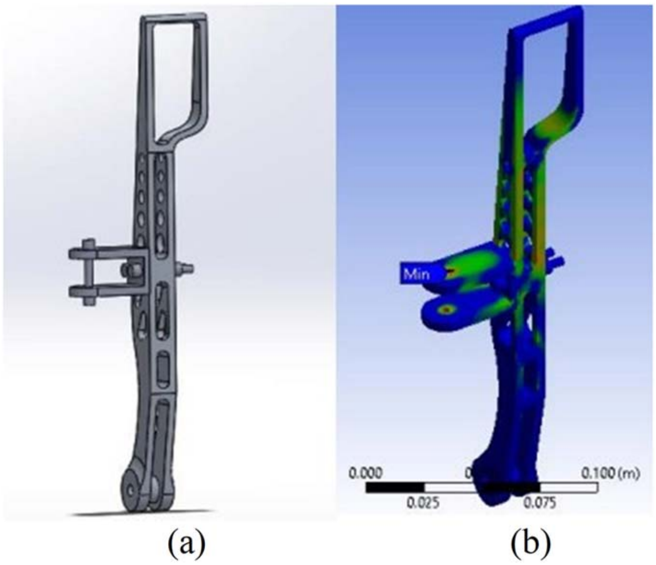


Figure 15. Throttle/Clutch pedal(a), F.O.S = 1.3 (b).

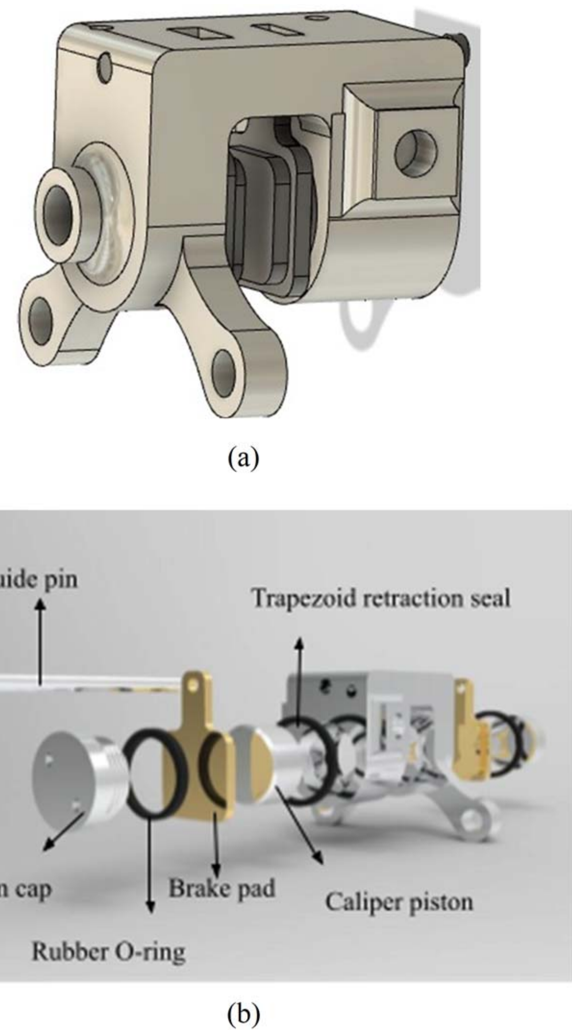


Figure 16. Final custom caliper (a), exploded view of the caliper (b).

3.3.3.1. Simulation

The Throttle pedal and clutch pedal were simulated with the input force as 445 N as given by Limpert, R. [11] and giving the fixed support at the pivot and at the cable holder bracket which is connected to the throttle body at intake.

3.4. Brake caliper design

The constraints for Brake caliper design are generally derived from the stock caliper taken as a reference which is the Wilwood PS1 caliper. The design majorly focuses on reducing the weight without having any adverse effect on its strength.

The design constraints for caliper are

- Type of caliper (i.e. Opposed Piston type, two-piston- one on each side, fixed with axial mounting)
- Caliper replacement is immediate hence the mounting points must be the same
- The caliper should be accommodated within the confined space of the wheel rim and the brake disc with sufficient clearances.
- The caliper should be able to withstand the pressure of 750.73 psi.
- The caliper should have two bleed screws on opposite sides so that it can be mounted on any side of the car.

A major change to achieve the above goals was to make the caliper monoblock (figure 16) instead of a 2-piece to enhance the resistance to bending moments improving the overall structural strength and also making caliper lightweight. The calculation for the clamping force generated by the caliper can be found from equations (4) or (5) below.

$$C(Clamping\ Force) = n \times Arcp \times P \tag{6}$$

from the equation (6) [11] total force was found to be 5081.62 N

3.4.1. Simulation

The calculated pressure was applied to the piston housing area and fixed support was applied at the mounting point for static structural analysis to obtain the FOS, stress, strain, and deformation results. The critical task in the caliper was designing the bridge because it is the region where the majority of bending and compressive stress occurs [14]. Having less stiffness there will result in a softer pedal feel as deflection increases and with the periodic load, it will ultimately fail due to lower fatigue strength. From the simulation results, it can be seen how stress is distributed uniformly around the bridge balancing good stiffness, FOS, and low weight, which are the benefits of having a monoblock caliper.

Having a monoblock caliper usually results in less weight as there are fewer parts involved in assembling the caliper such as bolts at the junction of the two halves, having a monoblock caliper also eliminates any chance of leakage at the junction area [15]. From the simulation results, we found the results of F.O.S, max strain , max stress and max deformation as mentioned in figures 17–20 respectively. from past experiences in designing calipers which was of a two-part caliper we achieved a F.O.S of 1.45 for that design and the part lasted more than one season even though it had poor load distribution than this caliper (figure 17) so, it was decided that the current value of F.O.S is in an acceptable range and safe to use.

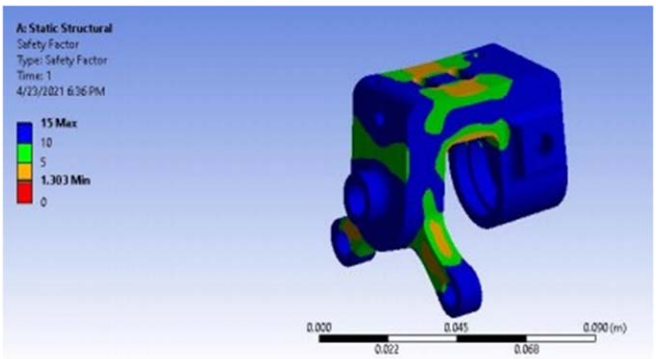


Figure 17. F.O.S: 1.3.

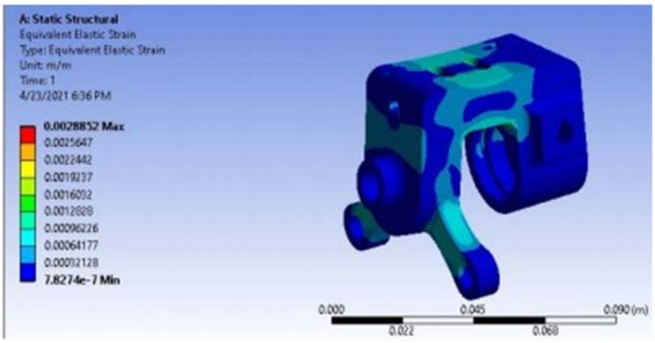


Figure 18. Max. Strain: 0.002 m m^{-1}.

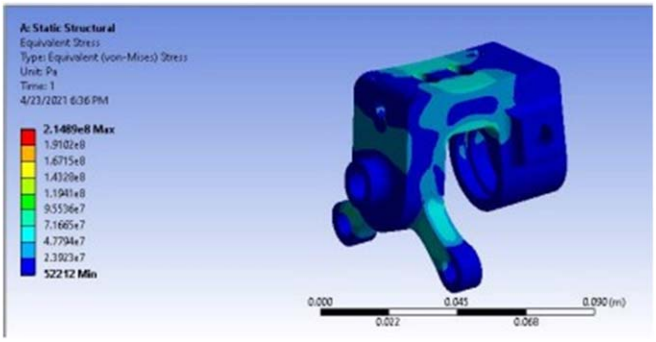


Figure 19. Max. Stress: 214.89 MPa.

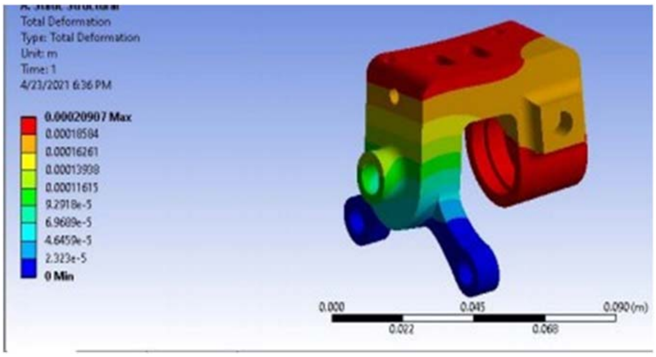


Figure 20. Max. Deformation: 0.2mm.

3.5. Brake disc design

The initial step was to choose the type of disc to be designed. Floating type disc was selected as it is better at managing thermal stresses by reducing chances of thermal juddering and also because they could accommodate any manufacturing defects in wheel assembly and work properly.

The major objectives while designing Brake Discs were

- Better thermal management
- Weight reduction
- Floating type

The constraints derived for the design of the brake disc were:

- Since the Disc will be assembled on the wheel hub inside the wheel rim, sufficient clearances are to be given in the design
- Provision of Slots and floral patterns for better thermal management as slots improve airflow and floral pattern increase disc surface area
- Giving sufficient thickness to endure the clamping and shear force developed while braking.

keeping these factors in mind the final CAD design of disc is shown below (figure 21).

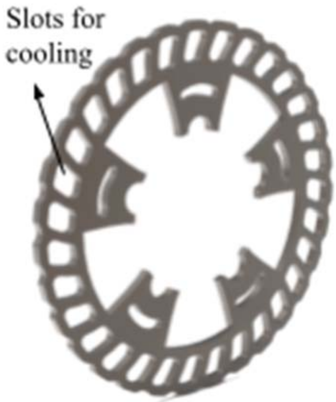
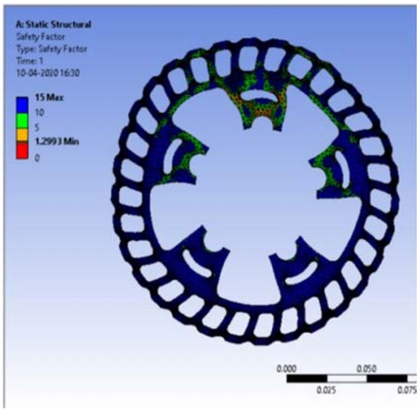


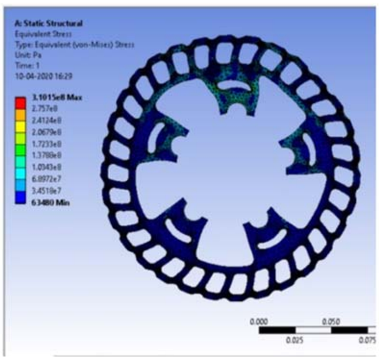
Figure 21. CAD model of the brake disc.

3.5.1. Simulation

The input forces and the heat generated were calculated using MATLAB and manual calculations. The max shear force was computed from the clamping force which was found to be 4500 N (figure 22) which was applied on the



Shear Force:- 4500 N
F.O.S:-1.29



Max. Stress :-3.10 x 108 Pa

Figure 22. Structural simulation of the brake disc.

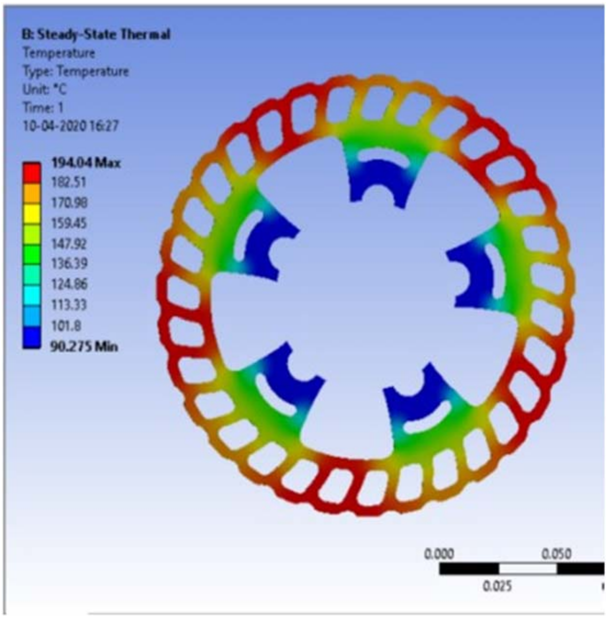
brake pad profile area, and at peak deceleration, heat flux was computed to be 18000 W/m^2 which was applied on the surface of the brake disc. After applying these loads simulation results were obtained as shown in figure 22.

Since Braking converts the kinetic energy of the vehicle into heat energy, the kinetic energy of the vehicle is obtained by equation (7) [10]:

$$Heat\ generated(H) = \frac{mv2}{2} + \frac{I\omega 2}{2} \tag{7}$$

Here, m = dynamic weight at one tire, v = velocity of the car, I = moment of inertia of one wheel assembly, ω = angular velocity of the wheel. This was taken as the input heat energy and divided into the front and rear brake discs, so for one wheel the heat generated simulation results prove that the design works and is safe as it can withstand peak loads and manage the thermal energy keeping the temperature lower than the boiling point of the brake fluid (Dot 5.1).

For the thermal simulation (figure 23) to get a more precise result pre-stress conditions of the structural simulation was applied and the coefficient of heat transfer between disc and air was taken for flowing air at 38 km h^{-1} instead of stationary air as it is an average velocity that we observed during an FSAE endurance event.



Heat Flux:- 18,000W/m²
Max. Temperature:- 194.04 deg. C

Figure 23. Thermal simulation of the brake disc.

4. Comparison

4.1. Pedal assembly

The OEM assembly (figure 24)designed by Tilton [16] a popular choice among formula student teams was observed to be **66% heavier** than the custom-designed assembly (figure 8) which was only **676 grams** (not including the tabs required to mount the assembly to the chassis, master cylinder, retraction springs) considering the clutch pedal as well for three-pedal configurations.

the custom pedal assembly also provided additional features such as:

- Adjustable according to driver's height
- The tunable pedal ratio for achieving different pedal responses suited to different drivers.

Figure 24. OEM Tilton pedal assembly.

Figure 25. Wilwood PS1 caliper.

In comparison with the pedal assemblies (as shown in table 2) from the authors of various research papers, the assembly was **40%–65% lighter** hence complementing our objectives. The pedal assembly designed by GalbinceaN.D. [17] weighed 1.9 kg which was **65% heavier**(considering throttle and brake pedal only) as compared to our designed pedal assembly. This weight was not including the master cylinder and other sensors.

The pedal Assembly designed by *Hathcock, C., & Munden* [18] weighed 1.7 kg and was 60% (considering throttle and brake pedal only) heavier as compared to our designed assembly. This design had a very bulky throttle pedal and assembly mounting system which was the reason for its higher weight. The pedal assembly also lacked adjustability features as compared to our design. Another pedal assembly by Zarizambri Bin Ahmad [2] weighed about 2 kg and individual components like pedal face, throttle/clutch cable mounting bracket were welded together this not only made the assembly heavy but heat affected zones developed due to welding would weaken the part and result in early failure. This pedal assembly not only was heavy but also lacked features like adjustability and tunable pedal ratio.

Such significant weight reduction in the pedal assembly presented by this paper was possible due to the usage of new techniques such as topology optimization and DFMA(Design For Manufacturing and Assembly) which led to these improvements. This comparison showed the benefits of the pedal assembly design presented in this paper(figure 8) such as adjustability to accommodate drivers of different heights and tunability of pedal ratio for different brake pedal responses and a wider spectrum of deceleration.

Table 2. Pedal assembly comparison.

Parameters	OEM assembly	Custom pedal assembly
Weight	2.3kg	676g
adjustability	no adjustability	upto 50mm
additional features	no unique features	tunable pedal ratio

Table 3. Caliper comparison.

Parameters	OEM Wilwood caliper	Custom caliper
Weight	431 g	172 g
type	two part caliper	monoblock
additional features	no unique features	ambidextrous with 2 bleed ports

4.2. Brake caliper

On comparing the Brake caliper to the OEM Wilwood PS1 caliper [19] (figure 25) which is a common OEM caliper seen in formula student cars the following differences were observed-

- Wilwood PS1 calipers weighed 450 g(Approx), the custom caliper (figure 16) was **61% lighter**.
- Two Bleed ports were given to make the caliper ambidextrous were missing in the OEM caliper.
- Being monoblock the designed caliper tackles the bending moment at the bridge better than the OEM two Piece caliper and also less number of parts than OEM caliper resulted in less weight.

Caliper design by *Phad, D et al* [20] had a total weight of 398 grams which is **56%** heavier than our custom caliper and had a total deformation of **0.4 mm.** The caliper designed by Ingale S *et al* [5] had an overall weight of 372 grams which was 52% heavier and had total deformation of 0.7 mm. More details of comparison can be seen in table 3.

This comparison proved the effectiveness of the custom caliper presented by the paper as it was lightweight as mentioned previously and being a monoblock caliper much stiffer which can also be proved by looking at the deformation and stress regions & values(figures 19 and 20), the design was also safe as it had acceptable F.O.S as shown in figure 17.

5. Conclusion

The paper studies the design and analysis of an FSAE vehicle's brake system which includes pedal assembly, brake caliper, and brake disc, and also compare it to its OEM counterparts as well and presents a comparison with other existing custom designs as explained in the comparison section of the paper.

The final designed custom pedal assembly was found to be **66% lighter** than the Tilton 900 series pedal assembly and about **40%–65% lighter** than other existing pedal assemblies this was possible by applying principles of DFMA and techniques like topology optimization, the brake pedal was designed so as to withstand force upto 2000N. The pedal assembly also incorporated a novel design of dual pedal ratio brake rail through which the driver can select between his suitable pedal ratio according to the track conditions which can provide an edge over other formula student cars, in future this system can also be motorized which would result in dynamic change of pedal ratio according to any corner through a button on steering wheel, another benefit of the custom pedal assembly was that it was adjustable to accommodate drivers of different height.

The custom two-piston brake calipers had huge weight savings which only weighed **172 grams** and was **61%–50%** lighter than OEM and other designs, less weight improves acceleration but will also improve the car's handling characteristics as it decreases the polar moment of inertia of the car. The caliper can withstand the pressure of more than 750.73 psi which gives a clamping force of 5081 N and provide deceleration upto 1.6 g. The weight reduction resulted in volume reduction of the caliper which allows more air to enter around it resulting in better cooling as well.

For the brake disc, necessary calculation and iteration were made so that it meets the required targets of keeping the max temperature well below the boiling point of the brake fluid, having sufficient structural strength to withstand a shear force of 4500 N, disc design also took care of accommodating any manufacturing errors and axial forces by making a floating type disc design. These gains from different areas of the car's braking system may appear small but summing up all these gains will put a significant improvement to the car's performance. Hence, a competitive package was successfully developed which can surely reduce lap times for a race car.

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Data availability statement

All simulation and data's are available with corresponding author and can be provided if needed.

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