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DESIGN AND ANALYSIS OF BRAKING SYSTEM OF SAE CAR

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Abstract

The following report encompasses the design of braking system. The system is one based on hydraulic braking and designed for a one-man performance racing vehicle. The objective of the system is to convert the kinetic energy of the vehicle into thermal energy, allowing the vehicle to decelerate optimally and safely. The design includes three major categories: calculation and evaluation of the hydraulic system in order to select callipers and master cylinders, the design of the pedal box, and the design of the rotors. The results and findings of the proceeding report rendered a symmetrical front and rear braking system. The Wilwood GP 200 two piston floating calliper and the Wilwood flange mount master cylinders with a ¾' inch bore was selected for the system. An optimized pedal box was designed consisting of two pedals, (Acceleration and brake), eight mounting tabs for the two pedals and master cylinders, and two electrical sensors supports (one for the brake light sensor and the other for the emergency stop switch). The pedal box was optimized for minimal mass and satisfied the design criteria (later defined in the text). Lastly, custom rotors were designed based on the optimal material choice, analysing thermal effects and considering cost.

Keywords: *calipers, wilwood, Master cylinder, Rotors*

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1. INTRODUCTION

The braking system was designed as a hydraulic system for a one-man performance racing vehicle. Generic parameters of the vehicle that were used in the system design were a total vehicle weight, including the driver, of 300 kg and a maximum velocity of 80km/hr. Further details of parameters will be defined in the report corresponding to the braking system. The three main categories/ components designed and analyzed for this system are the following: calculation and evaluation of the hydraulic system in order to select calipers and master cylinders, the design of the pedal assembly and the design of the brake disc. For the first component, the overall functionality and free variables of the braking system will be analyzed for optimum calipers master cylinder and brake disc size selection. The second main component of the design encompasses the pedal assembly design. This design includes the gas pedal, brake pedal and master cylinder orientation as well as the throttle senor and emergency stop placement and orientation. The third and final main component involves the design of the brake disc. The main focus on the design of the brake disc was material selection as well as the geometry of the brake disc. The constraints and limitations based on the competition rules of these components will be defined later in each of their sections. The objective of these major component designs waste minimizes the weight for the lightest design possible while at the same time designing to all of the mechanical and thermal conditions that the system would be subjected to. In the pedal assembly and brake disc design sections the optimization processes will be discussed as well as the finite element analysis that supports the final design. In the final sections of the report the manufacturing of the pedal assembly and brake disc will be discussed.

1.1 Basics of Hydraulics

The principle behind any hydraulic system is simple: forces that are applied at one point are transmitted to another point by means of an incompressible fluid. In brakes we call this brake fluid of which there are a few different varieties, but more on that later. As is common in hydraulics the initial force which is applied to operate the system is multiplied in the process. The amount of multiplication can be found by comparing the sizes of the pistons at either end. In braking systems for example, the piston driving the fluid is smaller than the pistons operating the brake pads therefore the force is multiplied helping you to brake easily and more efficiently. Another convenient characteristic of hydraulics is that the pipes containing the fluid can be any size, length or shape allowing the lines to be fed almost anywhere. They can also be split to enable one master cylinder to operate two or more slave cylinders if needed.

1.2 Objective

1) The brakes must be strong enough to stop the vehicle with in a minimum Distance in an emergency.
2) The driver must have proper control over the vehicle during braking and the vehicle must not skid.
3) The brakes must have good ant fade characteristics i.e. their effectiveness should not decrease with constant prolonged application.
4) The brakes should have well anti wear properties.

Braking system is one of the important safety components of a vehicle. It is mainly used to decelerate vehicles from an initial speed to a given speed. A friction based braking system is a common device to convert kinetic energy into thermal energy through a friction between the brake pads

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and the rotor faces. Because high temperatures can lead to overheating of the brake fluid, seals and other components, the stopping capability of a brake increases with the rate at which heat is dissipated due to forced convection and thermal capacity of the system. The braking system was designed as a hydraulic system with two master cylinders, one for the braking of the front two tires and one for braking of the rear two tires. Attached to each master cylinder are two universal mount calipers, one located at each of the tires for a total of four calipers for the system, as well as four rotors or brake disc.

The brake system in a Formula student is next to the wheel suspension and the choice of tires, the most important component. Especially with the winding course of the Formula student events the acceleration and stopping for fast lap times has special importance. The most efficient the brake system decelerates the vehicle; the longer can be range speeds. Ideally, the car will direct to the curve at full speed, delayed with maximum friction potential of the vehicle and in the top of the curve must start with the reacceleration.

The flow of the braking system is as follows: the driver exerts a force on the brake pedal, the brake pedal channels that force to the master cylinders, thus displacing the braking fluid in the master cylinders. The displaced fluid then exerts a pressure on each of the calipers allowing the calipers pistons to exert a clamping force on the rotors. Therefore, the input of the system is the driver's applied foot force and the output is the clamping force of the calipers exerted on the rotors. Therefore, for this braking hydraulic system design the main objective is to design the system so that the driver has to place a substantial, but not excessive force on the brake pedal in order to completely lock the tires of the vehicle. The system was designed for this foot force to approximately be 2000N. This value was decided upon by referencing other design teams and Dr. Richard Gross.

2. BRACKING FORCE CALCULATIONS

1) Coefficient of friction between tire and road μ_B=0.8
2) Height of mass Centre above road H=300mm
3) Weight of vehicle =300kg (inclusive driver)
4) Wheel base L=1550mm
5) Length of front axles (L_f) and rear axles (L_r) from C.G

   L_f=852.5mm, L_r=697.5mm
6) Radii of tireR_{ft}&R_{rt}=260.35mm
7) Radii of friction forces R_{FCP} =R_{RCP} =100mm
8) Considering wildwood caliper GP200
9) Piston area =1.23in^2=793.547mm^2
10) Number of piston =2
11) From these values the force transferred was determined:

$$F_T=\frac{\mu_B.W.H}{L}$$
$$=\frac{0.8\times300\times9.81\times300\times10-3}{1550\times10^{-3}}$$
$$F_T=455.69N.$$

The normal forces applied to the front and rear tires during breaking:

Front: $$F_{NF} = \frac{WL_R}{L}+F_T$$
$$=\frac{300\times697.5\times10^{-3}}{1550\times10^{-3}}+455.69$$

Rear: $$F_{NR}=\frac{WL_f}{L}-F_T$$
$$=\frac{300\times652.5\times10^{-3}}{1550\times10^{-3}}-455.69$$
$$=1190.481N.$$

From the suspension engineer the radii of the tires for both the front and rear were given as 250.3mm. Given the hollow rim design, where the brake disc sits inside of the rim with the calipers mounted, the rotor's diameter was restricted. The inner diameter of the rim was approximated to 278.4mm and the calipers clearance was approximated to be 40mm, (given a final selection has not been made yet). After modelling the rotor, calipers and hub in SolidWorks the largest brake disc diameter that could fit the assembly was 209.55mm. This was diameter used, as the largest rotor diameter would produce the greatest torque from the calipers, for a lower line pressure, during braking. Thus, the friction forces for the front and rear calipers could be solved for:

$$F_{FCP}\ R_{FCP} - \mu_B F_{NF} R_{FT} = 0$$

$$F_{FCP} = 3650.107N$$

Friction forces generated by rear two calipers

$$F_{RCP}\ R_{RCP} - \mu_B F_{NR} R_{RT} = 0$$

$$F_{RCP} = 2479.53N$$

At this stage in the design, it was necessary to select a calipers and brake pad. The number of pistons in the calipers and the coefficient of friction of the brake pads were the free variables for these selections. After researching on the internet possible calipers choices and drafting a calipers selection matrix (see below) based on number of pistons, weight, size and cost, the Wilwood universal mount was selected:

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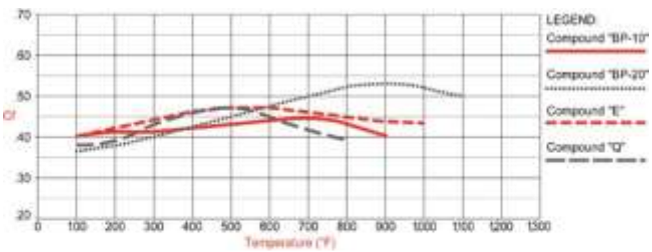
Company/ Manufacturer	Make/ Model	Body Material	Piston Material	Wieght (lbs)	Cost	Disk OD (in)	Disk thickness (in)	Piston Diam (in)	Piston Diam 2(in)	Piston Area (in^2)	Piston Area 2 (in^2)	# of Pistons
AP Racing	Aluminium - Lug Mount - CP2576	Aluminum	Aluminum	2.4		10.5118	0.38189	1.625984		2.08		2
AP Racing	Aluminium - Lug Mount - CP2577	Aluminum	Aluminum	2.4		10.5118	0.38189	1.751969		2.41		2
AP Racing	Aluminium - Lug Mount - CP3176	Aluminum	Aluminum	2.4		10.5118	0.38189	1.5		1.77		2
AP Racing	Aluminium - Lug Mount - CP3177	Aluminum	Aluminum	2.4		10.5118	0.38189	1.41732		1.58		2
AP Racing	Aluminium - Lug Mount - CP3178	Aluminum	Aluminum	2.4		10.5118	0.38189	1.251969		1.23		2
AP Racing	Aluminium - Radial Mount 130 ctrs - Solid Disc - CP6120	Cast Aluminum Alloy	Aluminium	3.3		11.1024	0.5	1.751969		2.41		2
AP Racing	Aluminium - Radial Mount 130 ctrs - Solid Disc - CP6121	Cast Aluminum Alloy	Aluminium	3.3		11.1024	0.5	1.5		1.77		2
AP Racing	Aluminium - Radial Mount 130 ctrs - Ventilated Disc - CP6126	cast Aluminium alloy	Aluminium	3.3		11.0236	0.7007874	1.751969		2.41		2
AP Racing	Aluminium Lug Mount - CP3696-6E0	Aluminium alloy	Aluminium	1.8		10.5118	0.279528	1.625		2.07		2
AP Racing	Aluminium Radial Mount - CP3676	Aluminium alloy	Aluminium	2.4		10.5118	0.38189	1.625984		2.08		2
AP Racing	Aluminium Radial Mount - CP3677	Aluminium alloy	Aluminium	2.4		10.5118	0.38189	1.751969		2.41		
AP Racing	Aluminium Radial Mount - CP4586	Aluminium alloy	Aluminium	2.4		10.5118	0.38189	1.41732		1.58		
AP Racing	Aluminium Radial Mount - CP4596	Aluminium alloy	Aluminium	2.4		10.5118	0.38189	1.251969		1.23		
AP Racing	Forged Aluminium Billet - Radial Mount - Ø36mm Bores	aluminium alloy	Aluminium	2.4		11.811	0.629921	1.41732		1.58		
AP Racing	4 Piston - Cast Body - 30mm Wide Disc - CP5090	cast aluminium alloy	Aluminium	5.1		11.0236	1.1811	1.5	1.75	1.77	2.4052819	4
Wilwood	Billet Dynalite Billet Dynalit	Aluminum	Stainless	1.8	115	13	0.38			2.4		2
Wilwood	Dynalite Single IIIA	Aluminum	Stainless	2	89	13	0.25			2.4		2
Wilwood	Dynapro Single	Aluminum	Stainless	2.3	107	13	0.19			2.4		2
Wilwood	Narrow Dynapro Lug Mout	Aluminum	Stainless	4.1	192	12.7	0.25			1.58		4
Wilwood	Forged Narrow Dynalite	Aluminum	Stainless	4.1	140	12.7	0.25			1.98		4
Wilwood	Narrow Dynapro Lug Mount	Aluminum	Stainless	4.1	192	12.72	0.25			1.98		4
Wilwood	DynaproLug Mount	Aluminum	Stainless	3.8	160	13	0.25			3		4
Wilwood	Forged Dynalite	Aluminun	Stainless	3.4	135	13	0.25			3		4
Wilwood	Narrow Dynapro Lug Mount	Aluminum	Stainless	4.1	163	12.72	0.25			3		4
Wilwood	Billet Dynalite - Side Inlet	Aluminum	Stainless	3.7	168	13	0.25			4.8		4
Wilwood	Dynapro Lug Mount	Aluminum	Stainless	3.8	160	13	0.25			4.8		4
Wilwood	Forged Dynalite	Aluminum	Stainless	3.4	135	13	0.25			4.8		4
Wilwood	Narrow Dynapro Lug Mount	Aluminum	Stainless	4.1	178	12.72	0.25			4.8		4

Chart-1: Comparative Caliper Matrix

After selecting Wilwood calipers, a Wilwood brake pad was chosen that would fit the style and size of the calipers. The Polymatrix Compound, Wilwood brake pad BP 10 smart pad was chosen for its overall high coefficient of friction. Based off of the company's performance chart (see below) between temperatures of 100°F & 700°F the average coefficient of friction was approximated as 0.6215.



Graph-1: Wilwood Brake Pad Performance Graph

The normal forces of the callipers then could then be solved:

$$F_{FCP} = \mu_{CP} F_{NFCP}$$

$$3650.107=0.6215\times F_{NFCP} = 5873.06N$$

$$F_{NFCP} = \frac{FRCP}{\mu cp} = 3989.59N$$

The pressure in both the front and rear braking lines could be solved:

$$P_R = \frac{F_{NRCP}}{\# \text{ of front caliper pistons} * A_{RCP}} = 3.7\times10^6 N/mm2$$

$$P_R = \frac{F_{NRCP}}{\# \text{ of front caliper pistons} * A_{RCP}} = 2.51\times10^6 N/mm$$

At this step in the design is necessary to select a set of master cylinders. The bore area of the master cylinder is the free variable in this selection. The bore area of the master cylinder, in addition to the pressure in each of the lines, dictate the total applied foot force needed to lock the tires during braking. After researching on the internet and drafting a master cylinder selection matrix, (see below), the wildwood flanged mount master cylinders with a 3/4" (19.04mm) diameter bore were selected for both the front and rear systems. This larger bore size was chosen because the selection rendered an applied foot force of just less than 667 N, which was the target applied force. Have to minimal of an applied foot force would cause the braking system to be very sensitive, thus making braking and overall operation not optimal for the driver.

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Master Cylinder Component Matrix	Part #	Price-each	Stroke in.	Bore in.	Length (end of rod to center of bearing)	MC Area
Tilton						
	77-Series	395-340	1.1	1	5.66	0.785
	77-Series	395-340	1.1	0.9375	5.66	0.690
	77-Series	395-340	1.1	0.8750	5.66	0.601
	77-Series	395-340	1.1	0.8125	5.66	0.518
	77-Series	395-340	1.1	0.7500	5.66	0.442
	77-Series	395-340	1.1	0.7000	5.66	0.385
	77-Series	395-340	1.1	0.6250	5.66	0.307
Brembo						0.000
	BRE-XA3G144	827.45	.906	.669		0.352
	BRE-XA3G145	827.45	.907	.75		0.442
	BRE-XA2L2A8	827.45	.908	.886		0.617
	BRE-XA2L2A9	827.45	.909	.789		0.489
AP Racing						0.000
	CP7855-88PRTE	291.547	1.1	.5	6.13	0.196
	CP7855-89PRTE	291.547	1.1	.59	6.13	0.273
	CP7855-90PRTE	291.547	1.1	.625	6.13	0.307
	CP7855-905PRTE	291.547	1.1	.66	6.13	0.342
	CP7855-91PRTE	291.547	1.1	.7	6.13	0.385
	CP7855-92PRTE	291.547	1.1	.75	6.13	0.442
	CP7855-93PRTE	291.547	1.1	0.8125	6.13	0.518
	CP7855-94PRTE	291.547	1.1	0.8750	6.13	0.601
	CP7855-95PRTE	291.547	1.1	0.9375	6.13	0.690
	CP7855-96PRTE	291.547	1.1	1.0000	6.13	0.785
Alcon USA						0.000
	MAR52xxHM101MBB	865.19		1	6.34	0.785
	Charted DWG	865.19		.9375	6.34	0.690
	Charted DWG	865.19		.875	6.34	0.601
	Charted DWG	669.97		.812	6.34	0.518
	Charted DWG	669.97		.85	6.34	0.567
	Charted DWG	669.97		.7	6.34	0.385
	Charted DWG	669.97		.625	6.34	0.307

Fig -1: Master cylinder selection matrix

In order to calculate the applied foot force, the last free variable that needed to be selected was the mechanical advantage of the pedal, from Equ.14, b/a. This factor is determined upon how the master cylinders are mounted to the pedal. This leads to the pedal box geometry design. The main constraints of this design were prescribed from the team's lead frame engineer, who requested that the surface area space of the pedal box be reduced as much as possible. Due to the allotted width and for the comfort of the driver, the frame engineer requested that the width of the pedal box be no more than 13 in (330.2). Further details of the pedal box design will be discussed later in the text. Based on the frame engineer's requests it was decided that the most optimal way to mount the master cylinders would be at an angle. A kinematic diagram was drafted of the set up (see below) where the lengths a & b are denoted as well as, the distance between the pivot point of the pedal and the master cylinders, d_f, the distance the pedal or foot of driver travels l and l_0 and , the original master cylinder stroke and the displaced master cylinder stroke.

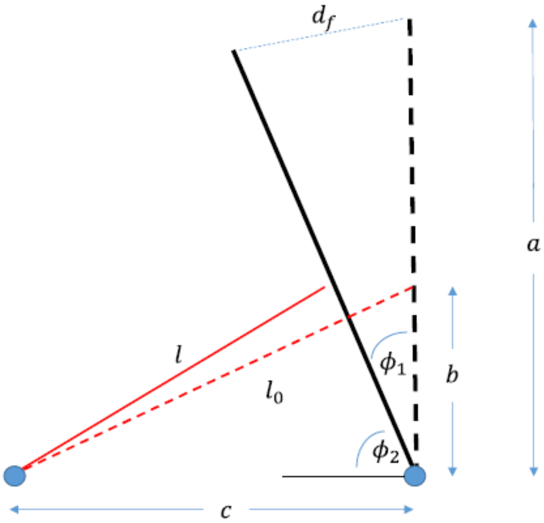

Fig -2: Kinematic diagram of brake pedal and master cylinder

Applying the Law of Cosines and Pythagorean Theorem to the above diagram the following equations can be derived:

$$D_f = a^2 + a^2 - 2a^2\cos\phi_1$$

$$L_0^2 = c^2 + b^2$$

$$T^2 = c^2+b^2-2cb\cos\phi_2$$

As recommended by Dr. Richard Gross, the value of the displacement of each calipers piston should approximately be 1.5mm. Therefore, the only remaining free variables in the pedal box geometry are lengths a, b, c.

After running a parametric sweep on these variables to find the optimal combination of the three, the following lengths were selected:
b= 240mm
a=45mm
c=80mm
Pedal ratio=5.33:1
Forces on both master cylinder
$F_{FMC} = P_F A_{FMC} = 1053.47N$
$F_{RMC} = P_R A_{RMC}\ 714.65N$

Based upon the selected pedal box geometry and master cylinder setup the mechanical advantage and applied foot force can be solved for:

$$aF_{FOOT} = b(F_{FMC} + F_{RMC}) = 361.66N$$

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Fig -3: Master cylinder CAD model

Fig -4: Balance bar CAD model

3. DESIGN CRITERIA

The constraints of the pedal box design are as follows:

1) The pedal box needed to be designed around the geometry constraints set in the results from the braking system calculations. That is the master cylinder mount height and the distance between the master cylinder pivot and the brake pedal are fixed.
2) The team's frame engineers also asked that the total width be no more than 330mm.
3) The length of both the gas and the brake pedal need to be approximately the length of a human foot (~ 9.5in).
4) The competition rules designate that the brake pedal must be able to withstand a force of 2000 N.

The objective of the pedal box design was to satisfy all of the constraints mentioned above and optimize for minimal weight.

A recommendation from the team members was that the pedal box be easily adjustable in the lateral direction, that is be comfortable for a taller driver as well as a smaller driver. Thus, the first stage in the design was this adjustment. The first draft was to have a tension line with retractable shafts in the base plate of the pedal box. That is the driver would push a button or grip a handle causing tension in the line that the shafts, which connected the pedal box to the floor of the vehicle, would retract and the pedal box could be slide into a new set of holes making it closer or further away from the driver. It was decided that this design would be too time consuming and a more robust design was desired. The second draft was similar to the first, however, omitted the tension lines and quick release pins would be used in place and the pedal box could be slide and adjusted just as before. However, this design was dismissed as an even more rigid design would be favorable. The third and chosen design was simply adjustable pedal heads. Where the pedal would have extended mounts to accommodate additional holes and quick release pins would be used for easy adjustment.

The rest of the design encompassed mounting of pedals, mounting the master cylinders and mounting the positive stop for acceleration and clutch and brake over travel switch to the pedal box.

The pedal box was designed with a single base plate; that is, all of the mounts would be fixed to this single plate. The brake pedal and master cylinder mounts were designed to be symmetrical for ease the applied foot force would be supported. The gas pedal however, would only be supported by the mounts and therefore were designed to be taller, approximately 200 mm height. The brake over travel switch mount needed to be positioned where, under the circumstance that there was a leak in the brake line and there was no longer substantial pressure in the line, the pedal would strike the switch at approximately 90% displacement of the master cylinders.

Fig -5: Mount plate CAD model

Fig -6: Pedal CAD model

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Fig -7: Brake Pedal CAD model and actual component

Fig -8: Brake pedal bushing Cad model and actual

4. MATERIAL SELECTION

In recent year, the material competes with each other for existing and new market. Over a period of time many factors that make it possible for one material replace to another for certain application. The main factors affecting the properties of the materials are strength, cost and weight. In automobile industries it is mandatory to look for cheap and lightweight materials and which should be easily accessible. The constituents of a composite are generally arranged so that one or more discontinuous phases are embedded in a continuous phase. The discontinuous phase is termed the reinforcement and the continuous phase is the matrix. A brake pedal in motor vehicles has the task of providing the driver's command through foot leg on master cylinder of the brake system in a vehicle during stopping or reducing speed of a vehicle. At present, brake pedals are largely made from metal but composite clutches and accelerator pedals are already successfully utilized in automotive vehicles. This study is mainly concentrated on variable material for the conceptual design brake pedal profile, which is achieved 78% lighter in weight compared to the present metallic pedals and 64% light weight compared to aluminium. According to General Motors specifications, the maximum load of 2700 N was applied to brake pedal. It is assumed that this is a possible panic load on a brake pedal.

The material selection process of the pedal box was based around the initial constraints of the design mentioned prior, but more specifically, the component of the pedal box that would be subjected to the most stress, the brake pedal. Based on the type of loading the pedal box would be subjected to it was modelled as a beam in bending and was decided that the desired material needed to have the following mechanical properties:

1) High Strength and be stiff
2) Good toughness
3) Machinability

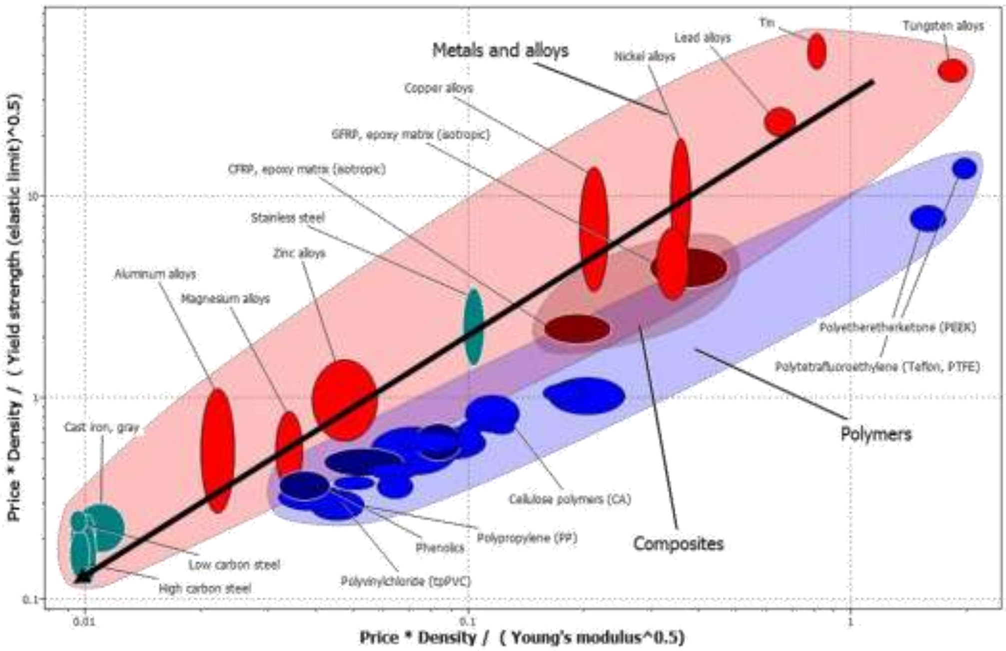

Fig -10: Material properties

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Analysis of this chart rendered the optimal choice was, based on the material properties that were most dependent on functionality and taking cost into consideration, carbon steel, cast iron or an aluminium alloy. Since out of the three, the carbon steels alloys are the suitable, this material group was selected for the design. Mild steel is quite heavier but considering strength it is best for this use.

5. FINITE ELEMENT ANALYSIS

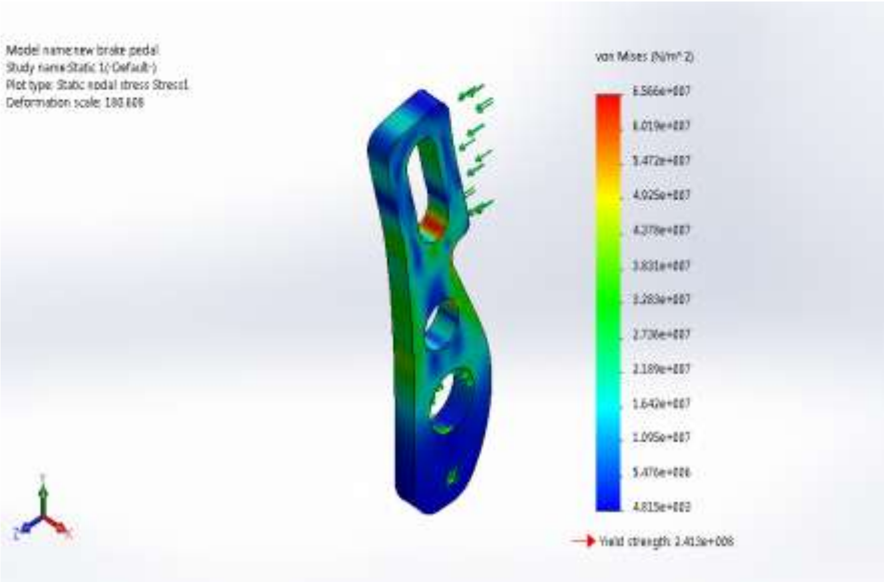

Fig -11: Brake pedal- VON: von Mises Stress

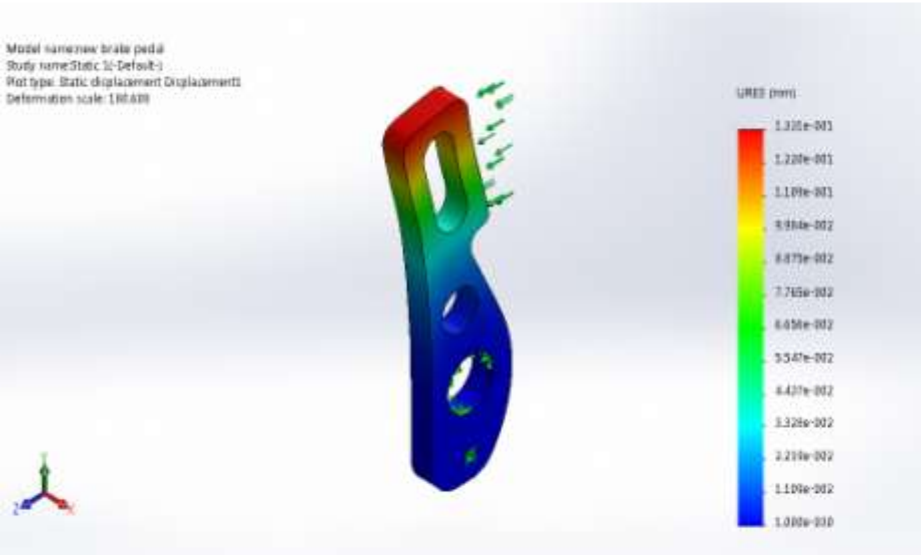

Fig -12: Brake pedal- Displacement

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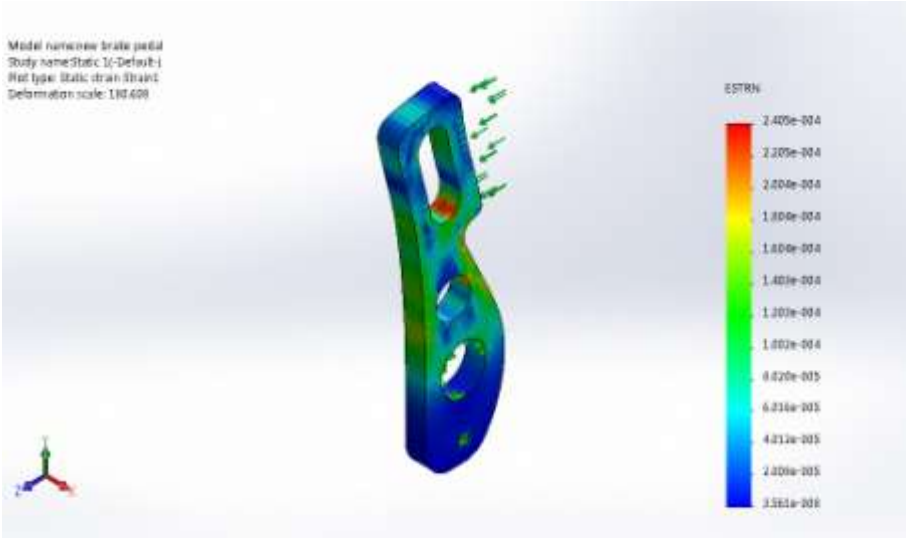

Fig -13: Brake pedal-Strain

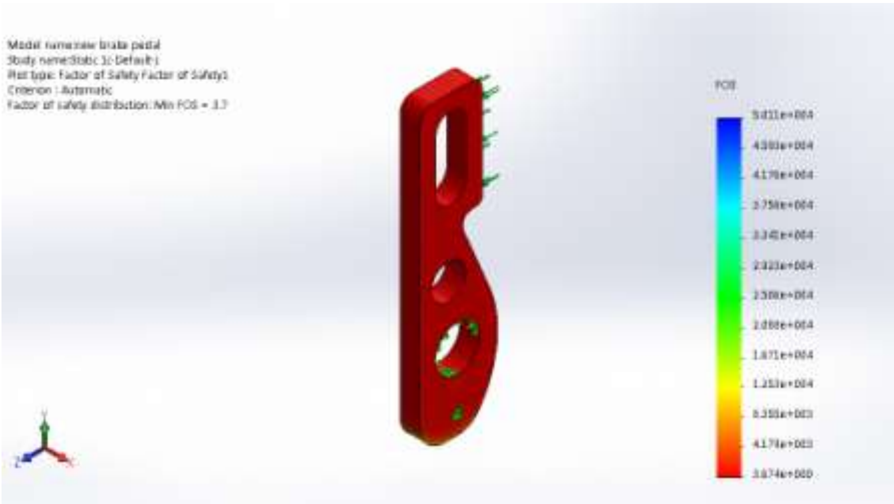

Fig -14: Brake pedal-Factor of safety

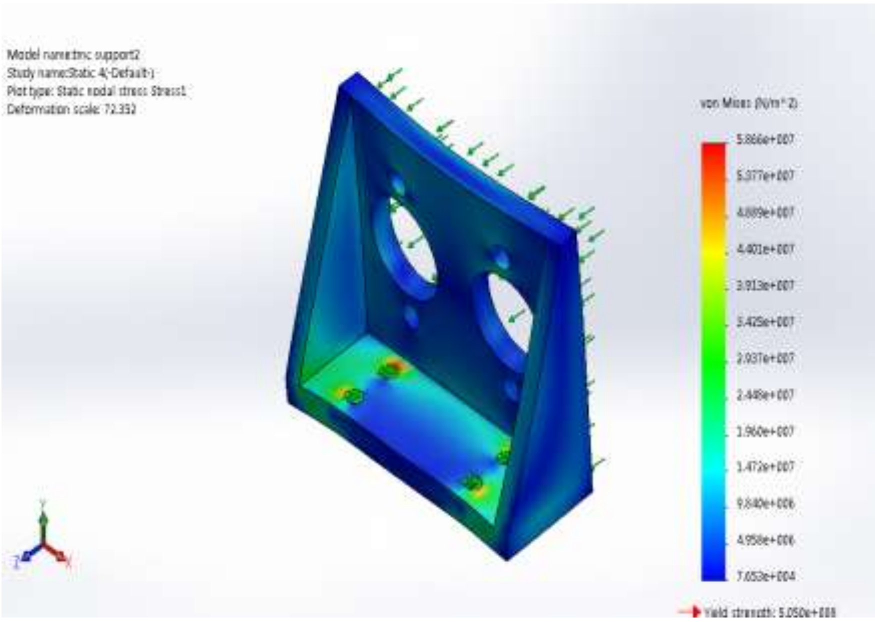

Fig -15: Base plate - VON: von Mises Stress

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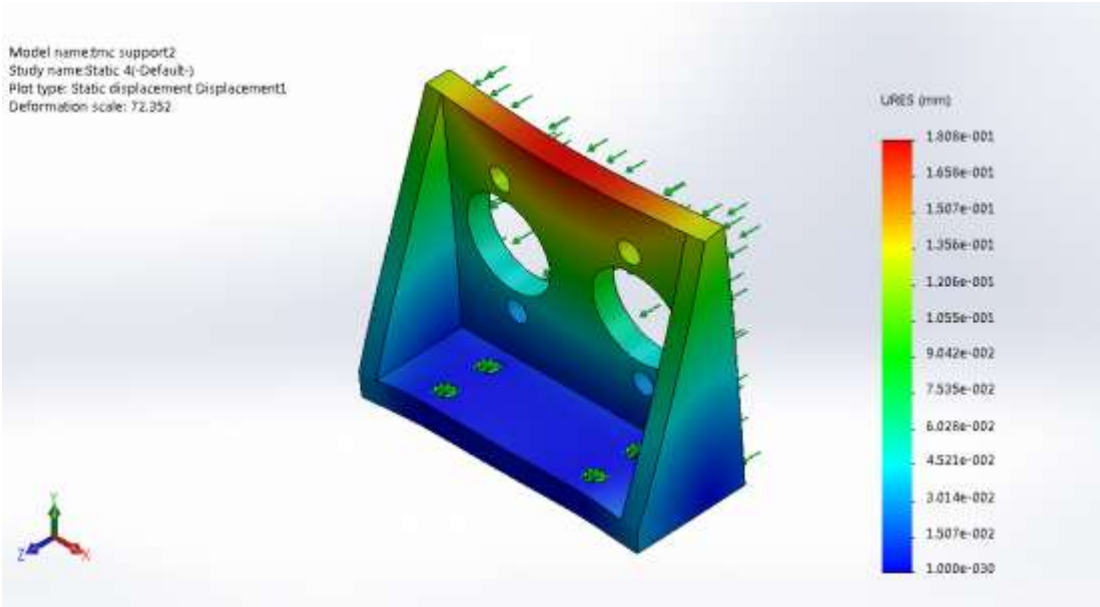

Fig -16: Base plate –Displacement

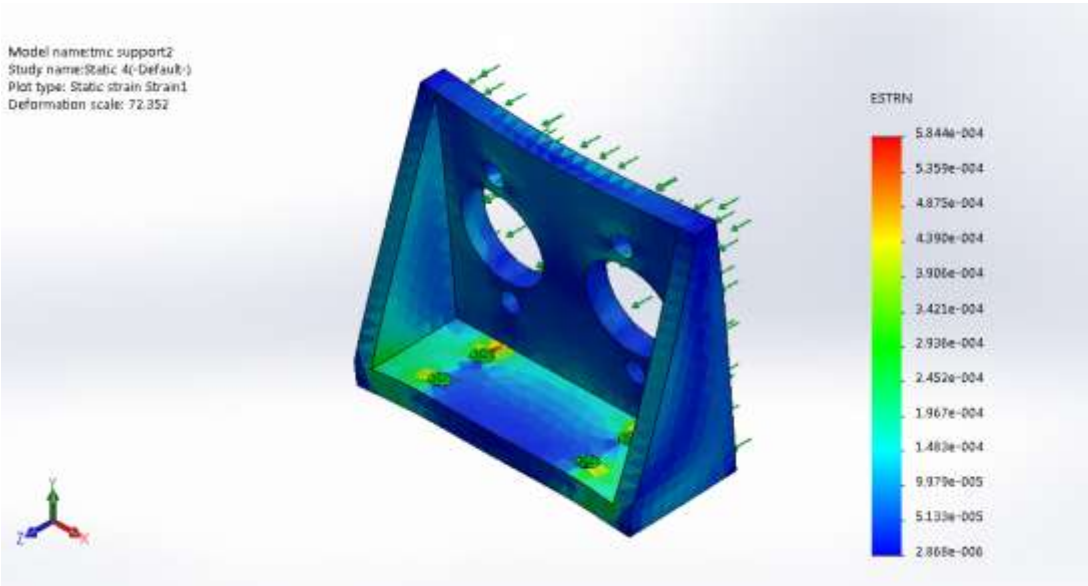

Fig -17: Base plate –Strain

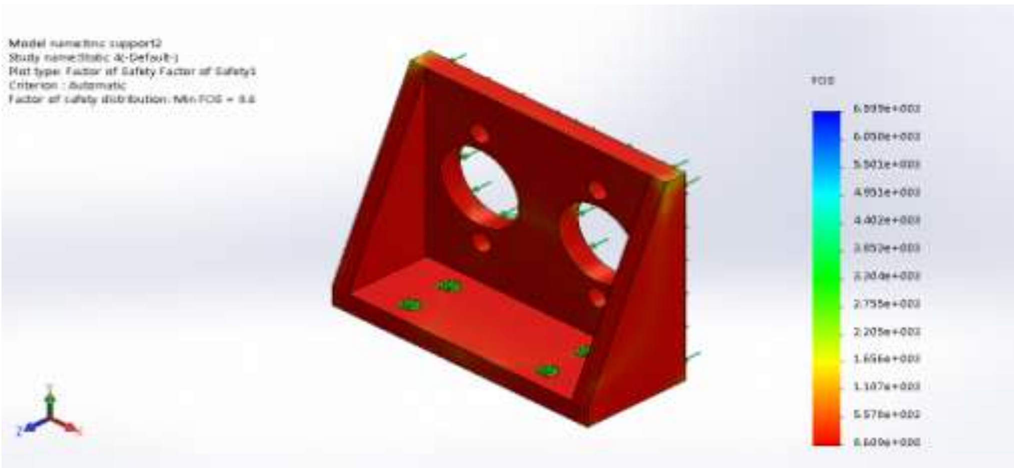

Fig -18: Base plate –Strain

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6. ROTOR DESIGN

Full Floating" means that the outer rim of the rotor (the braking surface) is allowed to move slightly with respect to the solid center of the brake disc.

Both fully floating brake disc and semi floating brake disc are constructed in two parts. A mild steel center part which is fixed to the motorcycle wheel and a stainless rotor part which the brake pads push on

1) A floating rotor will find its own optimal path through the brake caliper. This means reduced brake drag (friction) which lets you put more horsepower to the ground instead of wasting it. They also help control disc warping under hard braking by allowing the braking surface to expand and contract naturally.
2) Second, they reduce the weight of your brake rotors. This reduces unsprung, rotating, mass. This has a similar benefit to lightweight wheels: better acceleration and faster turn-in.
3) When the rotor is subjected to serious heat it expands. By allowing it to float separately from the mounting face it is free to expand and shrink again at will without being constrained by its mounting. When this expansion takes place is does so in all directions at once and it will not be constrained. If you prevent this from happening in one direction (by fixing it on its mounting face) it has no choice but to warp, so floating discs and semi-floating discs are made in two parts to allow the discs to expand and prevent them from warping. This is mainly a high-performance type brake disc
4) Floating rotors tend to have less alignment problems with pads and pads wear is more uniform because of the float.

Repetitive braking of a vehicle generates large amount of heat. This heat has to be dissipated for better performance of brake. Braking performance largely affected by the temperature rise in the brake components. High temperature may cause thermal cracks, brake fade, wear and reduction in coefficient of friction.

During braking, the kinetic and potential energies of a moving vehicle get converted into thermal energy through friction in the brakes. The heat generated between the brake pad& disc has to be dissipated by passing air over them. This heat transfer takes place by conduction, convection and somewhat by radiation. To achieve proper cooling of the disc and the pad by convection, study of the heat transport phenomenon between disc, pad and the air medium is necessary. Then it is important to analyse the thermal performance of the disc brake system to predict the increase in temperature during braking. Convective heat transfer model has been developed to analyse the cooling performance. Brake discs are provided with cuts to increase the area coming in contact with air and improve heat transfer from disc.

The rotors of the vehicle are the medium in which the kinetic energy from the motion of the vehicle is converted into thermal energy and then dissipated. From the braking system calculations earlier in the text the geometrical constraints of the rotors are:

1) 222.25mm (8.75in) diameter
2) Thickness between 0.25in (6.15mm), (dependent on caliper selection)
3) Based on the application the other constraints that will be imposed on design are:
4) Mechanical endurance at high temperatures
5) Ability to dissipated or with stand high temperatures

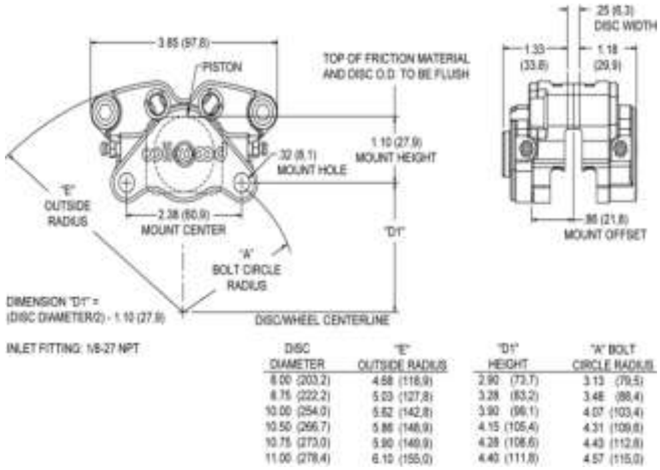

Fig -19: wildwood calipers GP200

Above fig. shows the schematic diagram, of wildwood calipers GP200 with the chart showing disc diagram with their mounting position on upright. Considering maximum available space in rim disc diameter is selected.

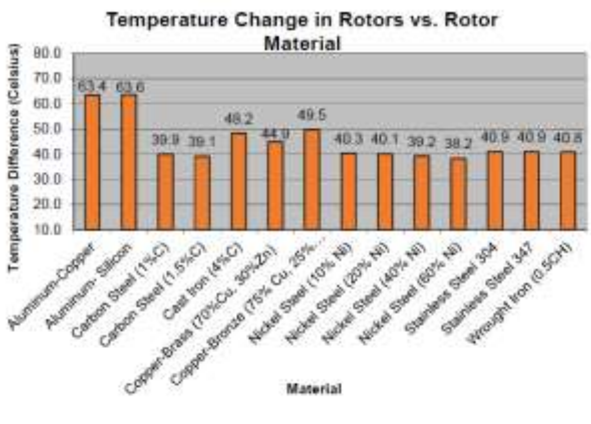

Fig -20: Rotor change temperature vs Rotor material

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Fig -21: Rotor CAD model

Based on this graph, the optimal choice is:
1) High Carbon Steel
2) Nickel Steel
3) Stainless Steel

The cheapest of these was selected for the rotor material, thus the high carbon steel was chosen. However, the final material choice was A36 Carbon Steel, low carbon steel.

7. HEAT GENERATED DUE TO FRICTION CALCULATION

In the aspect of the car accident prevention, the braking performance of vehicles has been a critical issue. The rotor model heat flux is calculated for the car moving with a velocity 27.77 m/s (100kmph) and the following is the calculation procedure.

Data:
1) Mass of the vehicle = 300 kg
2) Initial velocity (u) = 27.7 m/s (100 kmph)
3) Vehicle speed at the end of the braking application (v) = 0 m/s
4) Brake rotor diameter 0.2225 m
5) Axle weight distribution 30% on each side (γ)=0.3
6) Percentage of kinetic energy that disc absorbs (90%) k=0.9
7) Acceleration due to gravity g =9.81m/s2
8) Coefficient of friction for dry pavement μ=0.7

Step 1
Kinetic energy (KE) = $\frac{1}{2} * \mathrm{m} * \mathrm{v}^2$
= 4693.5 joules
The above said the total kinetic energy induced while the vehicle is under motion.

Step 2
The total kinetic energy = the heat generated
Qg = 4693.5 joules

Step 3
The area of the rubbing faces
A= $\Pi * (0.2225 - 0.1875)$
=0.1099 m^2

Step 4
Heat flux = heat generated / second / rubbing area
=10676.75 watts / m^2

In most rotor designs there are cuts or holes in the surface where the rotor comes in contact with the brake pad. The main intension of these cuts is to reduce brake fad during braking. As the temperature of the brake pad begins reaching its design limit a faint layer of gas from the deterioration of the brake pad forms between the pad and the rotor. The effect of this gas layer is a severe decrease in the brake pad coefficient of friction, which translates as an ineffective brake to the driver. This effect gives the driver a spongy feel to the brake pedal; as the applied foot force of the driver is use to applying barley brakes, thus hindering the deceleration of the vehicle. Having cuts or holes in the rotor creates a vent or fan like effect, which flushes out this gas layer that forms. Two side effects of having these cuts in the rotor are reduced mass and an increase in the cooling. Thus, it was decided that the rotor would have cut geometry.

9. CAD MODEL

Fig -22: Pedal Final Assembly CAD model

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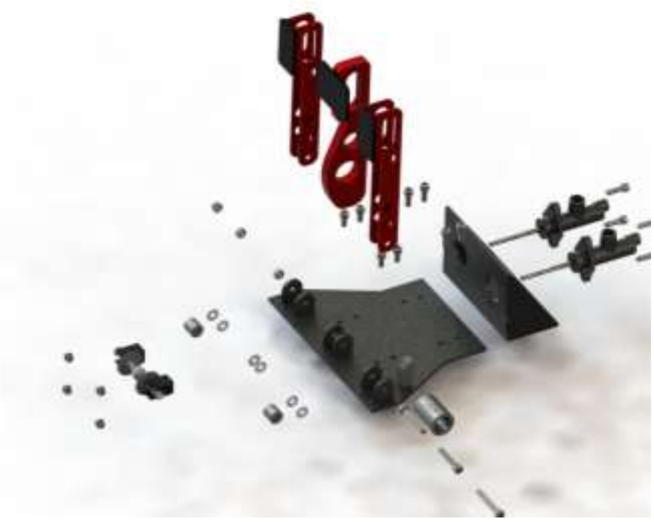

Fig -23: Pedal assembly exploded view

10. CONCLUSION

Heat flux = heat generated / second / rubbing =10676.75 watts / m^2

The pressure in both the front and rear braking lines

$$P_R = 3.7 \times 10^6 \mathrm{N/mm2}$$

$$P_R = 2.51 \times 10^6 \mathrm{N/mm}$$

Applied foot force= 361.66N

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BIOGRAPHIE

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