Design, Fabrication & Testing of Braking System & Wheel Hubs of FSAE Vehicle

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Abstract— The following project encompasses the Design of 2019 Vidyavardhini's College of Engineering And Technology FSAE vehicle braking system and wheel hubs. 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 using a hydraulic braking system. The results and findings of the proceeding report rendered a symmetrical front and unsymmetrical rear braking system. An optimized pedal box was designed consisting of three pedals, (throttle, brake and clutch), twelve mounting tabs for the three pedals and master cylinders, and two electrical sensors supports (one for the inertia switch 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). Later custom brake disc and hubs were designed based on the optimal material choice, after analyzing thermal effects.

Keywords: FSAE Vehicle, Braking System, Wheel Hubs

I. Introduction

The braking system is 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 220 kg and a maximum velocity of 100kmph. 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 box 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 and its geometry. The second main component of the design encompasses the pedal box design. This design includes the throttle pedal, brake pedal, clutch pedal and master cylinder orientation as well as the throttle sensor and emergency stop placement and orientation. 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 was to minimize 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 box s the optimization processes will be discussed as well as the finite element analysis that support the final design. In the final sections of the report the manufacturing of the pedal box, brake disc and wheel hubs will be discussed.

II. Objective

- 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.
- To minimize 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.
- To carry out necessary analysis on components using various soft-wares and compare it with theoretical data.
- To calculate Forces acting on components, braking distance, disk temperature and analyze the effect on every component.

III. Analytical Calculations and Design

A. *Design Calculation for Brakes:*

$\sum F_x$

$F_{BF} + F_{BR} = ma$

$\sum F_Y$

$R_{NF} + R_{NR} = W$

Moment at CG

$\sum M_{CG} = 0$

$R_{NR}*x + F_{BR}*h – R_{NF}*(W_B-x) + F_{BF}*h = 0$

$(R_{NR} + R_{NF})*x – R_{NF}*W_B + (F_{BR} + F_{BF}) = 0$

$Wx – R_{NF}*W_B + \mu Wh = 0$

$R_{NF} = (Wx + \mu Wh)/W_B$

$R_{NF} = W – R_{NF} = \{ W*(W_B – x) - \mu Wh \}/W_B$

FOR

$x = 0.7m$ $m = 280kg$

$h = 0.3m$ $W = 280*9.81 = 2746.8N$

$\mu = 1.4$

$W_B = 1.525m$

$R_{NF} = 2746.8*0.7 + 1.4*2746.8*0.3/1.525$

$R_{NF} = 2017.32N$

$R_{NR} = 2746.8*(1.525 – 0.7) – 1.4*2746.8*0.3/1.525$

$R_{NR} = 729.48N$

∴ $F_{BF} = \mu R_{NF} = 2824.28N$

$F_{BR} = \mu R_{NR} = 1021.27N$

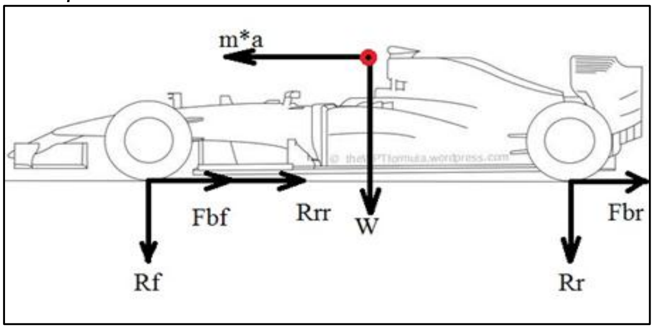


Fig. 1: Forces Resolved

B. Line Diagram

$F_B = F_{BF} + F_{BR} = MW$
$F_{RR} = f_r*W$
$\sum F_X$
$ma = (W/g)*a = F_{BF} + F_{BR} + F_{RR}$ -----(1)
$\sum F_Y$
$R_F + R_R = W$ -----(2)
Taking moment about CG we get;
$(F_{BF} + F_{BR})*h + F_{RR}*h – R_F(W_B – x) + R_R*x = 0$
$R_F*W_B = (R_F + R_R)*x + \mu Wh + f_r Wh$
$R_F = \{Wx/W_B\} + \{Wh(\mu + f_r)/W_B\}$
$R_R = W – R_F = \{W*W_B/W_B\} – \{Wx/W_B\} – \{Wh(\mu + f_r)/W_B\}$
$R_R = \{W(W_B – x)/W_B\} – \{Wh(\mu + f_r)/W_B\}$
W = 280*9.81 = 2746.8N
W_B = 1.525m
h = 0.3m
x = 0.7m
V_{max} = 80kmph
Tire pressure ‘p’ = 20psi = 1.38bar
$f_r = 0.005 + (1/p)[0.01 + 0.0095(V/100)^2]$
f_r = 0.01665
R_F = 2026.18N
R_R = 719.82N
F_{BF} = 2836.652N
F_{BR} = 1007.748N

C. Calculations for Front Tyre

Fbp = Fd × {L ÷ y}
= 400 × {210 ÷30}
= 2400 N
Force distribution at balance bar
Fbp1 = x/l*Fbp=1400N (balance bar is 6 mm offset from center towards front)
Fbp2 = (l-x)/l*Fbp=1000N
Fbp = force output of brake pedal assembly
Fd= Force applied to the pedal pad
L1 = Distance from the brake pedal arm pivot to the o/p rod of M/C
L2 = Distance from the brake pedal arm pivot to brake pedal pad
Fbp1 = Force on master cylinder 1
Fbp2= Force on master cylinder 2
l =effective length of balance bar

1) Master Cylinder

Pmc = Fbp1 ÷ Amc
= 1400 ÷ {(3.142 ÷ 4) × 15.875^2}
= 7.07311 N/mm²
Pmc = Pressure at master cylinder
Amc = Area of master Cylinder
Pcal = Pmc
Pcal = Pressure at calipers

2) At calipers part 1

Fcal = Pcal × Acal
= 7.07 × {(ð ÷ 4) × 31.75^2}
= 5615.18 N
Fcal = One sided linear mechanical force generated by calipers.
Acal = Area of calipers.

3) At Brake pad

Ffriction = F clamp × ìbp × 2
= 4492.14 N
Ffriction = Frictional force
ìbp = Coefficient of friction between brake pad and road

4) At Rotor

Tr = Ffriction ×Re
= 4492.14 × 0.08
= 359.37 Nm
Tr = Torque generated by rotor
Re = Effective radius of rotor
Tt = Tw = Tr
Tt = Torque found on Tyre
Tw = Torque found on wheel

5) At Tyre

Ftyre = Tt/Rt
= 359.37/0.2286
= 1572 N
Ftyre = 1572 each front tyre S
Ftyre (L, R) = 3144.1 N

D. Calculation for Rear Tyre

1) At calipers part 1

Fcal = Pcal × Acal
= 5.05 × 706.85 = 4010.8N

2) At Brake Pad

Ffriction = F clamp × ìbp × 2
= 4010.84 × 0.4 = 3208.67N

3) At Rotor

Tr = Ffriction ×Re
= 3208.67 ×0.110 = 336.9 N-m

4) At Tyre

Ftyre = Tt/Rt
= 336.9/0.2286
= 1473.8 N
Ftyre = 1473.8 N (Both rear tyres)

E. Stopping Distance and Time Calculation

1) Deceleration of vehicle = $F_{total}/m_{vehicle}$
= [(2*1572)+1473.8]/220
= 20.99 m/sec^2 = 2.139g
2) Stopping Time = $(v*m_{vehicle})/F_{total}$
= (33.33*220)/4617
= 1.587 sec.

F. Design Calculation for Hubs

1) Fatigue Strength Calculation

Required Min. Fatigue Life

Distance for each event	Km
Endurance	22
Autocross	4
Skid Pad	2
Testing Phase	3
Distance on tracks	400
Total	431

Req. Min. no. of cycles = 431000/(peri. Of wheel with tyre)
= 431000/1.4363
= 300076.58 cycles

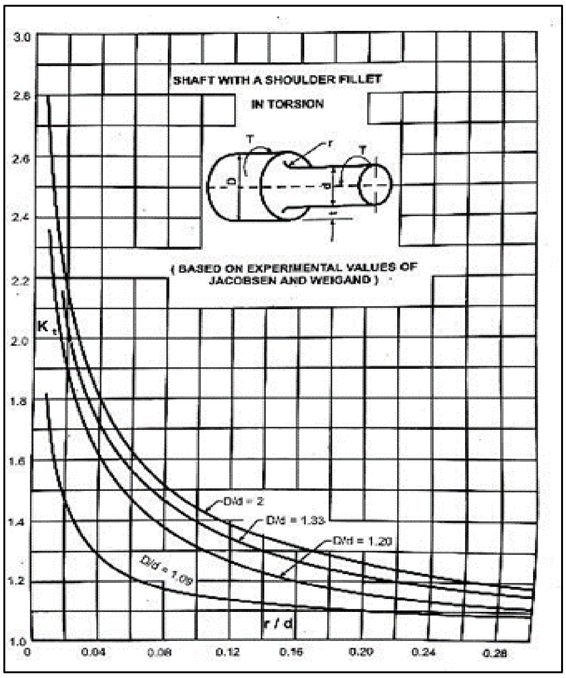


Fig. 2: Stress Concentration Factor vs. r/d ratio of shaft in torsion

Theoretical fatigue strength Calculation

Surface Roughness Factor

$Ka = a1(Su)^{b1}$

$= 4.51x(1200)^{-0.265}$

Ka = 0.68

Size Factor

$Kb = 1.51\ (d)^{-0.157}$

$= 1.51x\ (140)^{-0.157}$

Kb = 0.69

Reliability Factor

Kc = 1-0.08 (Za)

= 1-0.08 (1.645)

….assuming 90% reliability (Za=1.645)

Kc = 0.868

Loading Factor

assume Kd = 1

Temperature Factor

assume Ke =1

Fatigue Stress concentration Factor

$Kf = [1+(K_t-1)]/[1-(e/r)^{1/2}]$

Kf = 0.958

Endurance Limit

$Se^{!} = 0.5\ (Su)$

= 0.5 (1200)

$Se^{!} = 600\ N/mm^2$

Modified Endurance Limit

$Se = Ka*Kb*Kc*Kd*Ke*Se^{!}/Kf$

$= 254.811\ N/mm^2$

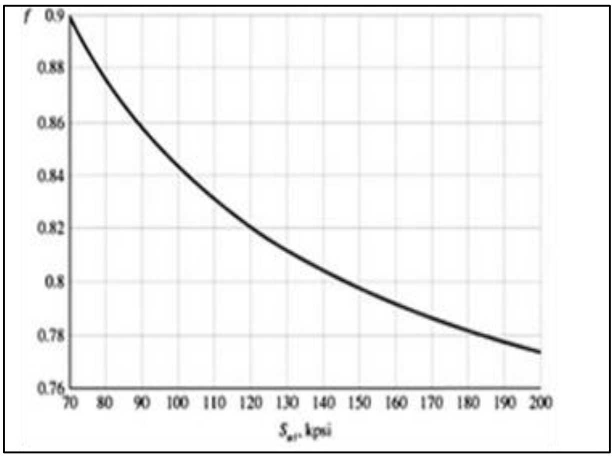


Fig. 3: Fatigue strength fraction vs. Ultimate strength of the material

Fatigue Strength at 300000 cycles

Fatigue Strength Coefficient (a) = $(f*Su)^2/Se$ = 3526.943

Fatigue Strength Exponent (b) = (-1/3) ln [(f*Su)/Se] = -0.437

By Basquin Equation

$Sf = a\ (Nf)^b$

$Sf = 14.252\ N/mm^2$

2) Bump Force Calculation

Braking Torque Brake Pedal force

1) Force applied on pedal is assumed to be 600 N
2) Pedal ratio = 7:1
3) fmax = force*pedal ratio
   = 400*7
   = 2800 N
4) Hydrostatic pressure (Pr) = $fmax/(\pi/4)*D^2$
   $= 2800/(\pi/4)*(31.8x10^{-3})^2$
   $= 3525.444\ N/mm^2$
5) $Fmax = fmax*(D/d)^2$
   $= 2800*(0.0318/0.009525)^2$
   = 31209.176 N
6) Torque acting on disc

T = Fmax*μ*Re*no.of pistons per caliper

$= 5168.239x10^3$ N-mm (rear disc)

$= 2321.962x10^3$ N-mm (front disc)

Bump Force Calculation

1) Newton's 2nd Law

F = ma

= 220*1.2*9.81 (a = 1.2g)

= 2707.56 N

2) Total Force acting on each wheel

$F_t = T/Re + F/4$

Rear wheel = 49898.21 N

Front wheel = 29701.39 N

G. Brake Discs Thermal Calculation

1) Kinetic Energy
   $K.E = 0.5*m*v^2$
   $= 0.5*220*27.77^2$
   = 84876.543 J
2) Heat Flux = (K.E* η)/(Δt*a*n)
   Where, K.E = Kinetic Energy = 84876.543 J
   η = efficiency of brake pad = 0.6
   Δt = braking time = 3 sec

a = area of brake pad = 1.0870×10^{-3}
n = no. of pads = 2
Heat Flux = 7808042.298 W/m^2

3) Prandtl's Number (Pr)
$Pr = (1007 * 1.983 \times 10^{-5})/0.024$
$= 0.8320 > 0.6$

4) Reynold's number (Re)
$Re = (v*x)/\nu$
$= [27.77*(2\pi r)]/2 \times 10^{-5}$
= 540900.5 (Front Disc)
= 601970 (Rear Disc)

5) Nusselt number (Nu)
$Nu = 0.0296 * Pr^{1/3} * Re^{4/5}$
= 1074.38 (Front Disc)
= 1170.38 (Rear Disc)

6) Film coefficient (h)
$h = (Nu*x)/k$
$= 66.19\ W/m^2\ {}^0K$
$= 64.79\ W/m^2\ {}^0K$

IV. Concept of Balance Bar

In order to obtain 4 wheel locking brake force must be applied on each wheel proportionately.

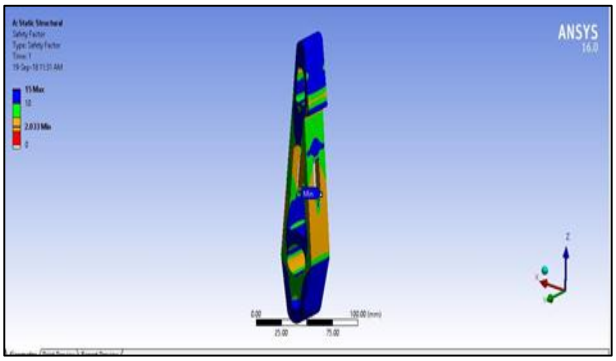
While the car is in motion the weight of car (engine, driver's weight, and all moving parts) is acting on rear side of vehicle, but when sudden braking takes place the C.G of car moves forward and hence the weight is distributed in some ratio. This is called weight distribution ratio (ideally 50:50).

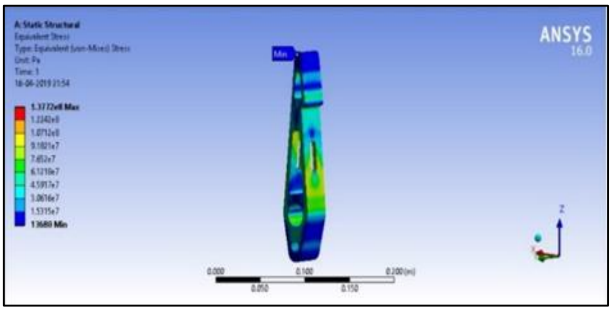
By calculation we concluded that the weight distribution of vehicle when in motion was 40:60 and during braking changes to 60:40. Hence, it was required to provide more braking force on front side. This is done by using a very simple yet quite effective mechanism called 'Balance Bar'.

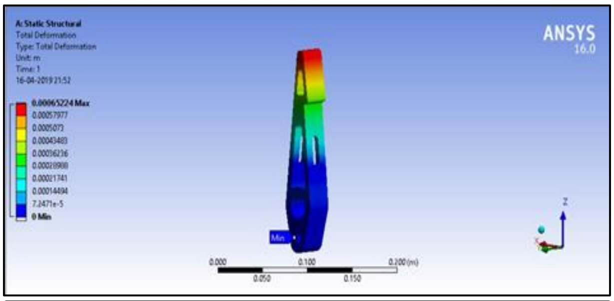
Balance bar is a unit which is used to distribute the force applied by driver's foot in proportion to respective master cylinder. It is a cylindrical beam which has a spherical bearing so that the bar is able to move freely.

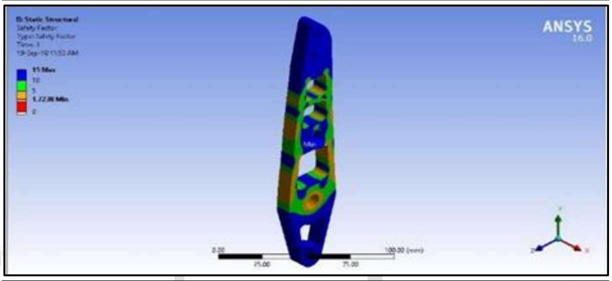
SPHERICAL JOINT POSITION	FORCE DISTRIBUTION (%)		FORCE ACTING		PRESSURE DEVELOPED in Master Cylinder (bar)	
12.6 mm right from center	40	60	960	1440	134.47	202.08
Centered	50	50	1200	1200	168.40	168.40
6.30 mm left from center	55	45	1320	1080	185.24	151.56
25.2 mm left from center	70	30	1680	720	235.77	101.04

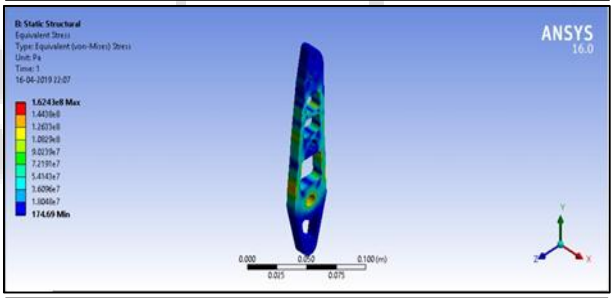
V. Results

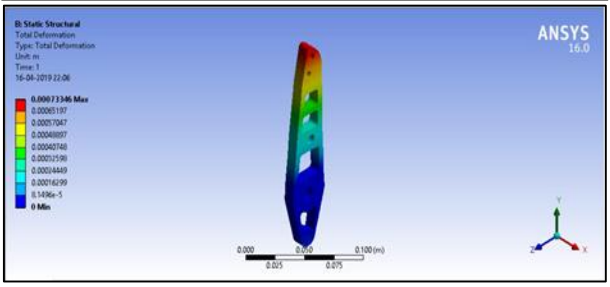


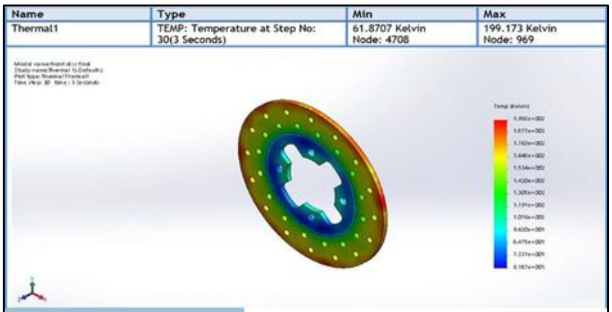




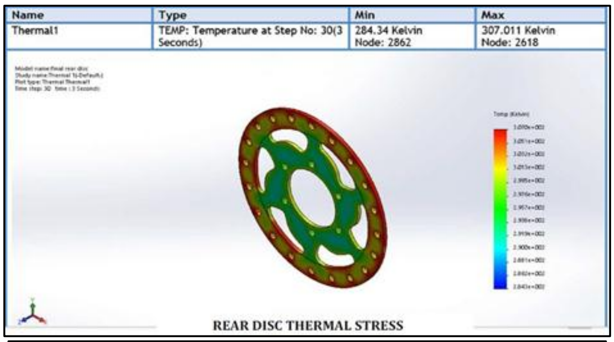


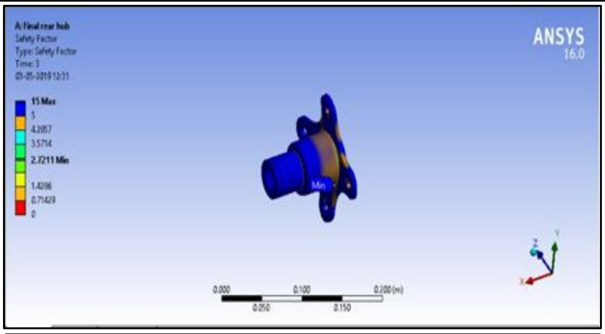


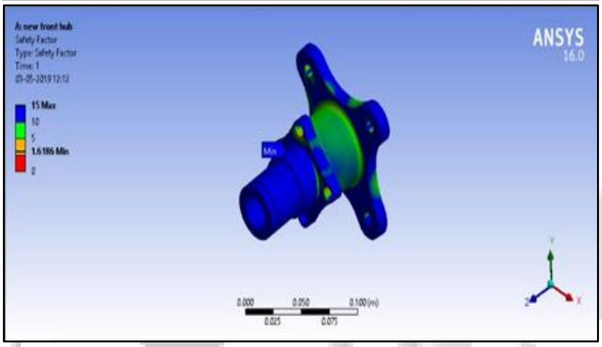


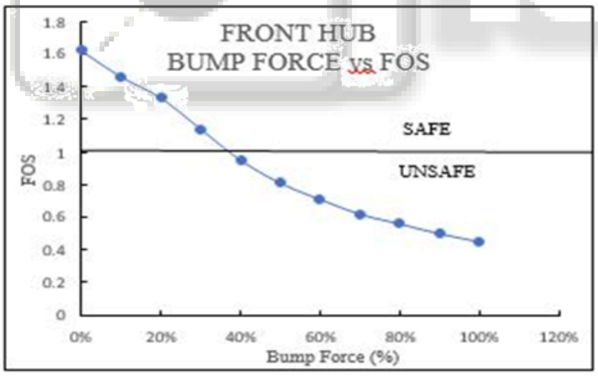


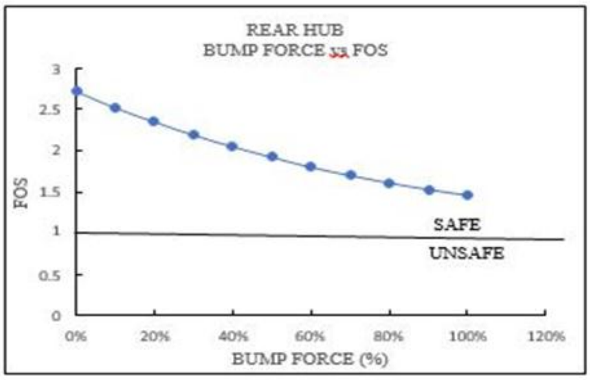
FRONT DISC- THERMAL STRESS

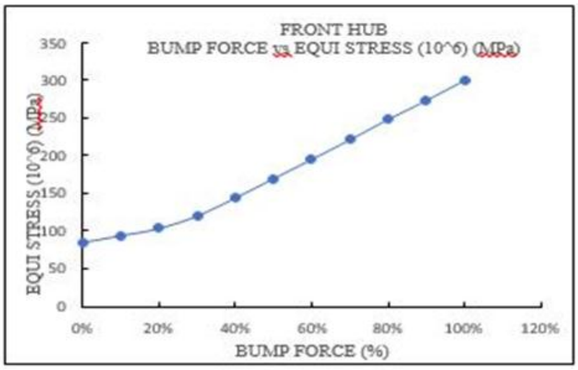

Name | Type | Min | Max
Thermal1 | TEMP: Temperature at Step No: 30(3 Seconds) | 284.34 Kelvin Node: 2862 | 307.011 Kelvin Node: 2618
REAR DISC THERMAL STRESS

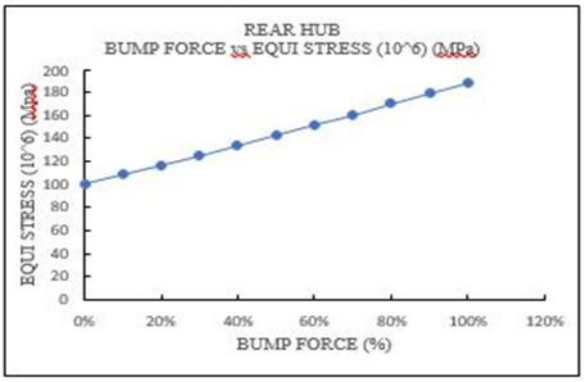

A: Final rear hub
Safety Factor
Type: Safety Factor
ANSYS
16.0

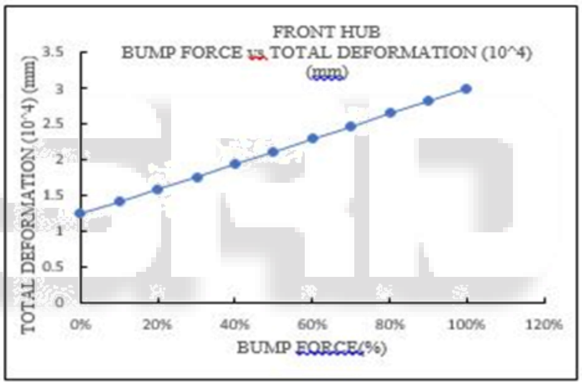

A: new front hub
Safety Factor
Type: Safety Factor
ANSYS
16.0

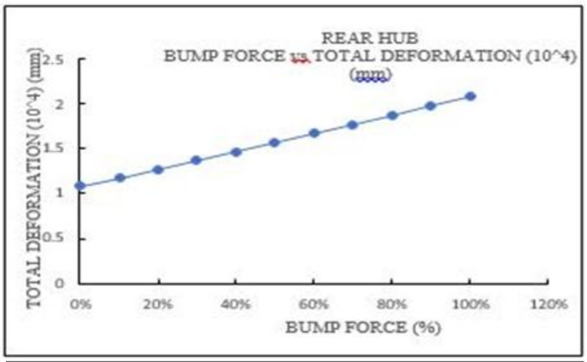

FRONT HUB
BUMP FORCE vs FOS
SAFE
UNSAFE
FOS
Bump Force (%)

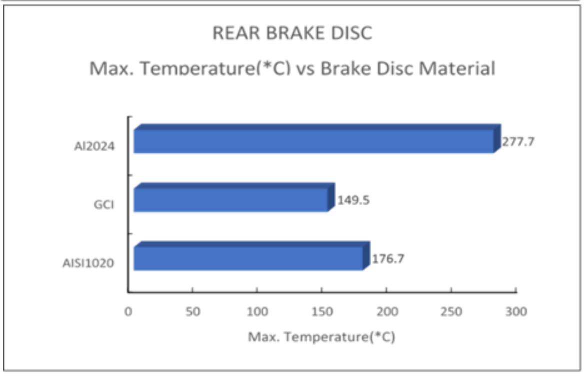

REAR HUB
BUMP FORCE vs FOS
SAFE
UNSAFE
FOS
BUMP FORCE (%)

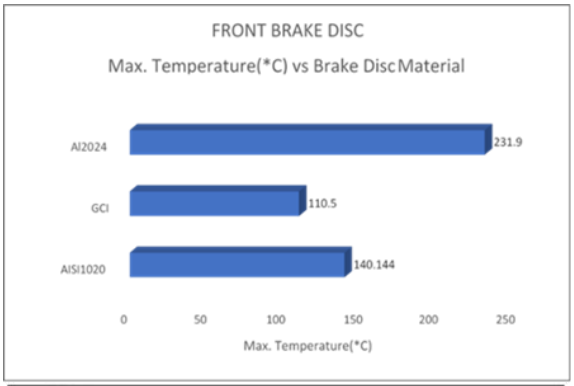

FRONT HUB
BUMP FORCE vs EQUI STRESS (10^6) (MPa)
EQUI STRESS (10^6) (MPa)
BUMP FORCE (%)

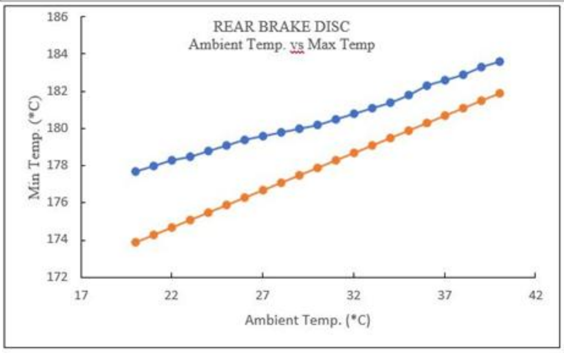

REAR HUB
BUMP FORCE vs EQUI STRESS (10^6) (MPa)
EQUI STRESS (10^6) (Mpa)
BUMP FORCE (%)

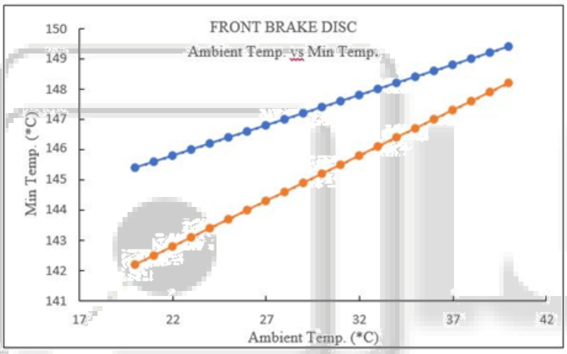

FRONT HUB
BUMP FORCE vs TOTAL DEFORMATION (10^4) (mm)
TOTAL DEFORMATION (10^4) (mm)
BUMP FORCE(%)

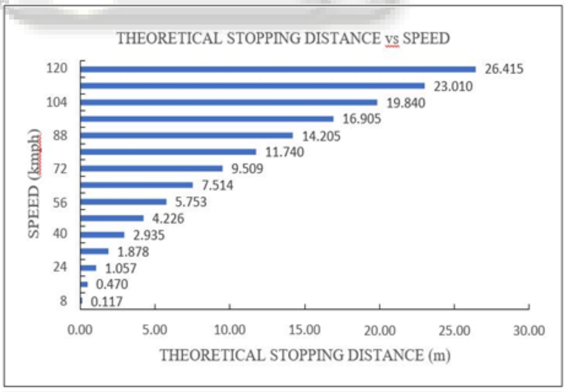

REAR HUB
BUMP FORCE vs TOTAL DEFORMATION (10^4) (mm)
TOTAL DEFORMATION (10^4) (mm)
BUMP FORCE (%)

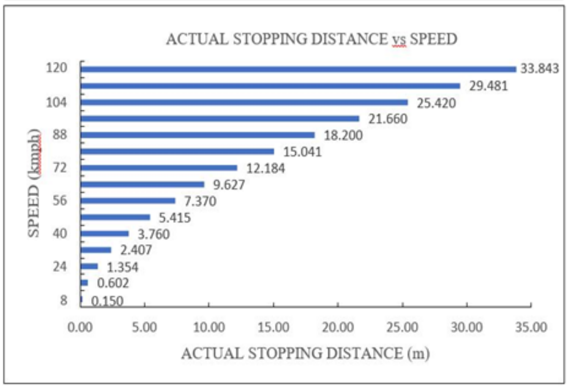

REAR BRAKE DISC
Max. Temperature(*C) vs Brake Disc Material
Al2024
277.7
GCI
149.5
AISI1020
176.7
Max. Temperature(*C)











VI. Conclusion

In this project we designed the braking system completely. The braking system is a custom designed with the help of data collected from previous years. The designing and assembly of complex components of system which were also tedious to manufacture were ordered from the market as per their specifications required.

- As Bump force increases the FOS of Hubs decreases.
- Rear Hub since subjected to more bump force is designed safer while front hub can withstand 30% bump force.
- The temperature of disc at time of braking depends upon the ambient temperature in which the car runs.
- With increase in speed of car stopping distance increases and is well within the permissible limit.
- All Four wheel lock condition is achieved.

VII. List of Symbols

- F_{BF} Front Brake Force
- F_{BR} Rear Brake Force
- m mass of the vehicle
- a acceleration of vehicle
- R_{NF} Normal Reaction Force at Front
- R_{NR} Normal Reaction Force at Rear
- W Weight of the vehicle
- x Distance between CG and front wheelbase
- h Height of CG from ground
- W_B Wheel Base
- μ Coefficient of Friction
- F_B Braking Force
- F_{RR} Rolling Reaction Force
- f_r Friction
- g Gravitational Force
- R_F Reaction Force Front
- R_R Reaction Force Rear
- V_{max} Maximum Velocity of vehicle
- p Tire Pressure
- F_{bp} Output Force of Brake Assembly
- F_d Force applied to pedal pad
- L1 Dist. from brake pedal arm pivot to o\p rod of M/C
- L2 Dist. from brake pedal arm pivot to brake pedal pad
- F_{bp1} Force on master cylinder 1
- F_{bp2} Force on master cylinder 2
- l Effective length of balance bar
- Pmc Master cylinder pressure
- Amc Area of Master Cylinder
- Pcal Pressure at calipers
- Fcal One sided linear force generated by calipers
- Acal Area of calipers
- Ffriction Frictional Force
- ibp Coefficient of friction between brake pad and road
- Tr torque generated by rotor
- Re Effective radius of rotor

- Tt — Torque found on tire
- Tw — Torque found on wheel
- Ftyre — Force on Tire
- K_A — Surface roughness factor
- K_B — Size factor
- K_C — Reliability factor
- K_D — Loading factor
- K_E — Temperature factor
- K_F — Fatigue stress concentration factor
- S_E' — Endurance limit
- S_E — modified Endurance limit
- a — Fatigue strength coefficient
- b — Fatigue strength exponent
- N_F — Fatigue strength cycles
- D — Diameter of piston caliper
- d — Bore Diameter of master cylinder
- F_{max} — Maximum hydraulic force
- η — Brake pad efficiency
- δt — Braking time
- n — no. of pads
- P_R — Prandtl's number
- Re — Reynold's number
- Nu — Nusselt number
- h — Film coefficient
- re — Effective radius of brake disc
- x — Surface area of brake disc

REFERENCES

[1] Carroll Smith, "Tune to win", year 1978.
[2] 'Racing Car & Vehicle Dynamic' William Milliken & Douglas Milliken, 1995.
[3] Global Journal of Researches in Engineering: b Automotive Engineering Volume 14 Issue 1 Version 1.0 Year 2014 Type: Double Blind Peer Reviewed International Research Journal Publisher: Global Journals Inc. (USA) Online ISSN: 2249-4596 & Print ISSN: 0975-5861
[4] Brake Bias Bar Design and Simulation on a Formula Student Car Apurv Puranik[1], Harshal Jaju[2], Tejas Karambelkar[3], International Conference on Ideas, Impact and Innovation in Mechanical Engineering (ICIIIME 2017) Volume: 5 Issue: 6
[5] Fatigue based design and analysis of wheel hub for Student formula car by Simulation Approach V Gowtham[1], A S Ranganathan[2], S Satish[3], S John Alexis[4], S Siva kumar[5]
[6] Design and Optimization of a Formula SAE Vehicle, Worcester Polytechnic Institute.
[7] Formula Student Design and Manufacturing Do and Don't by Claude Rouelle.