Design Analysis and Optimization of Disc Brake Assembly of A 4-Wheeler Race Car

Avijit Singh Gangwar
B.E. Automobile Engineer
Manipal Institute Of Technology

Abstract-A disc brake is a wheel brake which slows rotation of the wheel by the friction caused by pushing brake pads against a brake disc with a set of calipers.This paper deals with the design analysis of the brake systems of a 4 wheel racecar. We have extensively designed and carried out the design analysis regarding separate parameters of the disc brake system involved in the car. For the later stages, we have optimized the working of the disc brake by optimizing the parameters in question and then did a comparative study of the 2 designs analyzed.

Keywords— Disc brake,optimization.

I. INTRODUCTION

The brake disc is usually made of cast iron, but may in some cases be made of composites such as reinforced carbon–carbon or ceramic matrix composites. This is connected to the wheel and/or the axle. To stop the wheel, friction material in the form of brake pads, mounted on a device called a brake caliper, is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc. Friction causes the disc and attached wheel to slow or stop. Brakes convert motion to heat, and if the brakes get too hot, they become less effective, a phenomenon known as brake fade.

The brake disc is the disc component of a disc brake against which the brake pads are applied. The material is typically grey iron, a form of cast iron. The design of the disc varies somewhat. Some are simply solid, but others are hollowed out with fins or vanes joining together the disc's two contact surfaces (usually included as part of a casting process). The weight and power of

the vehicle determines the need for ventilated discs. The "ventilated" disc design helps to dissipate the generated heat and is commonly used on the more-heavily-loaded front disc.

II. METHODOLOGY

The following steps were implemented in designing the disc-brake system for the given race-car:

Specifications relevant to the brake system design on the car were collected.

i. Weight
Car: 240kg
Driver: 65kg
Total weight (w): 305kg

ii. Deceleration: 1.5g
iii. Static rear weight(w_r): 173.85kg
iv. Static front weight(w_f): 131.15kg
v. Tire diameter:515.62mm
vi. Coefficient of friction between ground and tire: 1.5
vii. Coefficient of friction between tire and rotor: Depends on the material of the rotor and pad which was 0.45
viii. Wheel base(b): 1610mm
ix. Height of centre of gravity(h): 280mm
x. Actual weight transfer:
   - Weight transfer (wt): (w * a * h)/ b = 780.534N
   - Dynamic front weight: w_f + wt = 2017.116N
   - Dynamic rear weight: w_r −wt= 924.934N

xi. Master cylinder bore diameter (a_m): 15.9mm
xii. Calliper bore diameter (d_c): 28.45mm
xiii. No. of pistons: 2 rear and 2 front
xiv. Pad width: 25.4 mm

Some parameters had to be assumed:

i. Force on brake pedal: 40kg or 392.4N
ii. Pedal ratio: 6:1

The following parameters were calculated using the relevant equations:

i. Master cylinder bore area: (3.14 * a_m)/4 = 197.83mm^2
ii. Caliper bore area: (3.14 * d_c)/4 = 635.29mm^2
iii. Force on balance bar: force on pedal * pedal ratio : 2354.4N
iv. Biasing on balance bar assumed to be: 60 : 40
v. Force on master cylinder:

- Front : force on balancebar* front bias=1412.64N
- Rear : force on balance bar * rear bias = 941.76N

vi. Operating Pressure:

- Front : Force on front master cylinder / Master cylinder bore area = 7.14 N/mm2
- Rear : Force on rear master cylinder / Master cylinder bore area : 4.76 N/mm2

vii. Clamping force:

- Front : Front operating pressure * caliper front area * no. of pistons * coefficient of friction between rotor and pad =4082.72N
- Rear : Rear operating pressure * caliper rear area * no. of pistons * coefficient of friction between rotor and pad :2721.82N

viii. Torque:

- Front: (Dynamic front weight/2) * 9.81 * (tire diameter/2) * coefficient of friction between road and tire =399692.44Nmm
- Rear: (Dynamic rear weight/2) * 9.81 * (tire diameter/2) * coefficient of friction between road and tire =178842.87Nmm

ix. Effective rotor diameter:

- Front : 2 * (front tire torque/ front clamping force) =195.80mm
- Rear : 2 * (rear tire torque/ rear clamping force) = 131.41mm

x. Total rotor diameter:

- Front : effective front diameter +pad width = 221.20mm
- Rear : effective rear diameter+ pad width =156.81mm

xi. Braking force: total weight * deceleration * 9.81 = 4488.075 N

xii. Stopping distance assuming test speed of 60kph = $(v^2) / (2ag)$ = 9.438m

III. CAD MODEL

Using the calculations listed above a CAD model "Fig. 1" of the existing design of the brake disc was generated using CATIA.

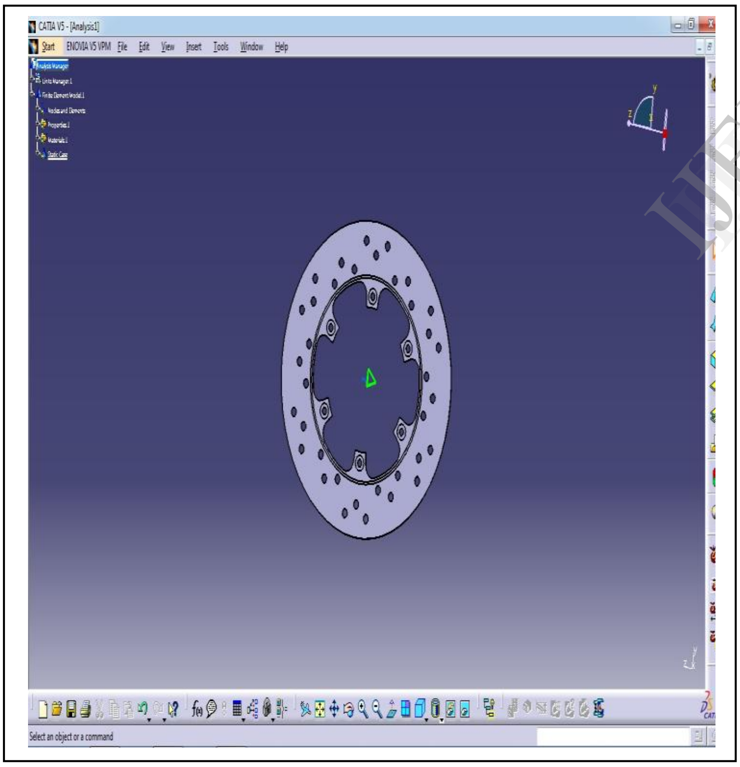

Fig. 1 – Cad Model Of Disc

IV. ANALYSIS

The inside surfaces of the six holes of the mounting points on the brake disc are constrained "Fig. 2". The retarding torque as calculated above is applied on the surface of the disc.

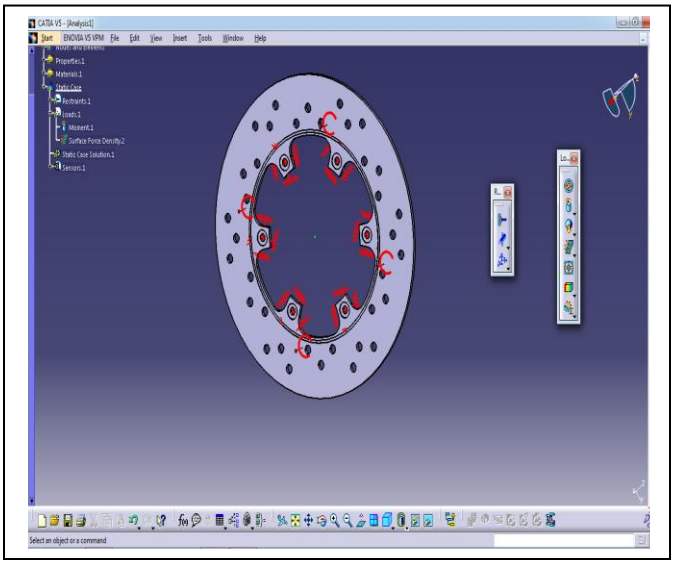

Fig. 2 – Force And Constraints

Using steel as the material of choice the structural analysis of the brake disc is performed and the following results are computed.

1. Deformation results

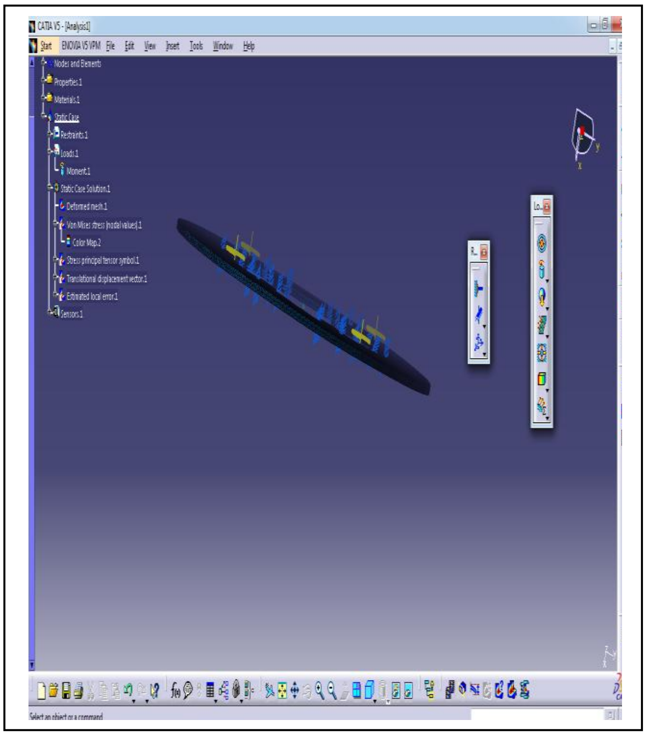

Fig. 3 – Deformation (Steel)

2. Von Mises results

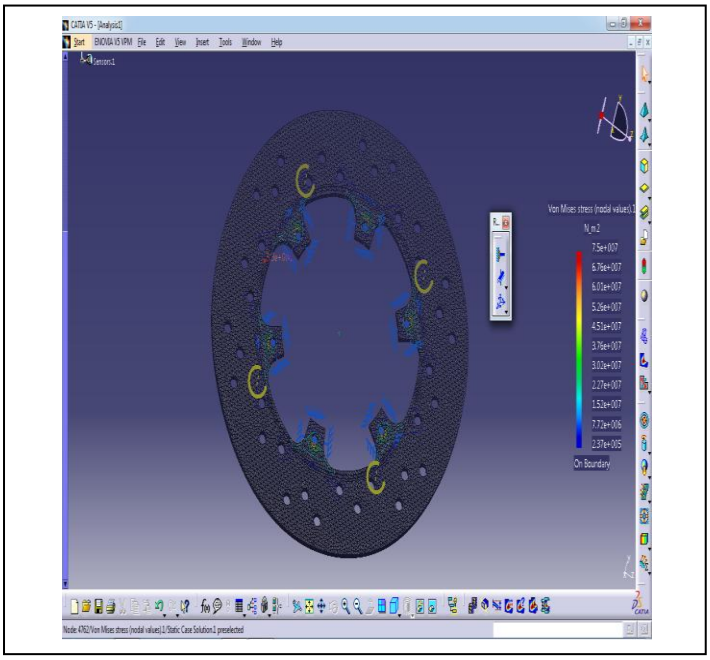

Fig. 3– Von Mises (Steel)

3. PRINCIPAL STRESS ANALYSIS

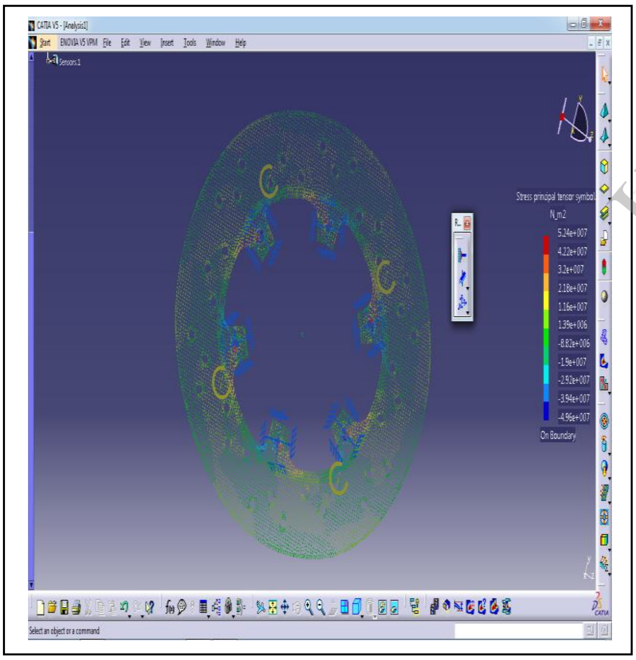

Fig. 4- Principal Stress (Steel)

INFERENCES

The yield strength for structural steel is 250 MPa and highest stress developed on the disc as per the von mises diagram is 75MPa. The deformation of the disc as can be seen in the diagram above is negligible. So based on this data we can infer that the existing model of the disc utilized in the disc brake of a racecar is safe and good to use.

V. OPTIMIZATION

The following objectives were kept in mind while optimizing the disc:

1. To minimize the price of the disc.
2. To increase safety of the disc.
3. To minimize material usage of the disc
4. To minimize size of the disc.

The optimization of the existing model of the disc keeping the above objectives in mind can be carried out in the following two methods :

1. By changing the material of the disc by keeping the dimensions of the disc same.
2. By changing the dimensions of the disc by keeping the material same for the disc.

Both these methods were carried out to obtain positive results which are stated below.

1. CHANGING MATERIAL OF THE DISC

The material chosen for the existing model of the disc was structural steel. The material chosen for optimizing the disc was chroma which is an alloy of aluminium and chromium. The size of the disc while choosing chroma was kept constant.

Results :

a) DEFORMATION RESULT :

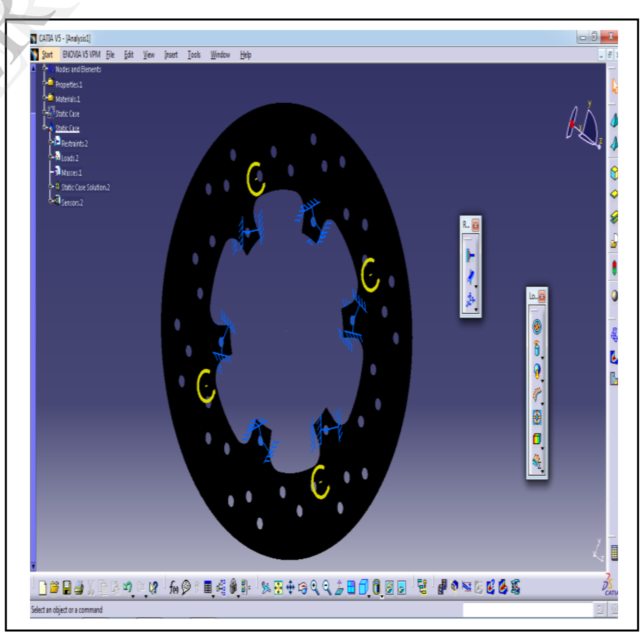

Fig. 5 – Deformation (Chroma)

b) VON MISES RESULTS

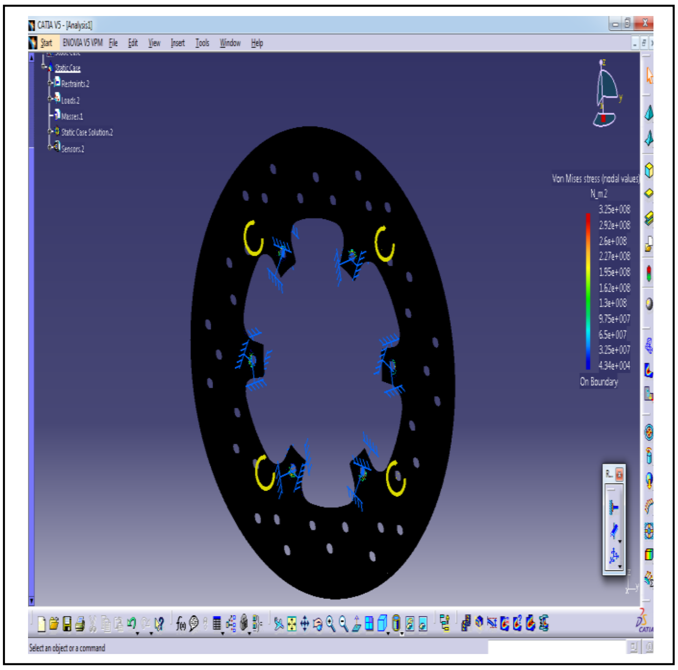

Fig. 6 – Von Mises (Chroma)

c) PRINCIPAL STRESS RESULTS

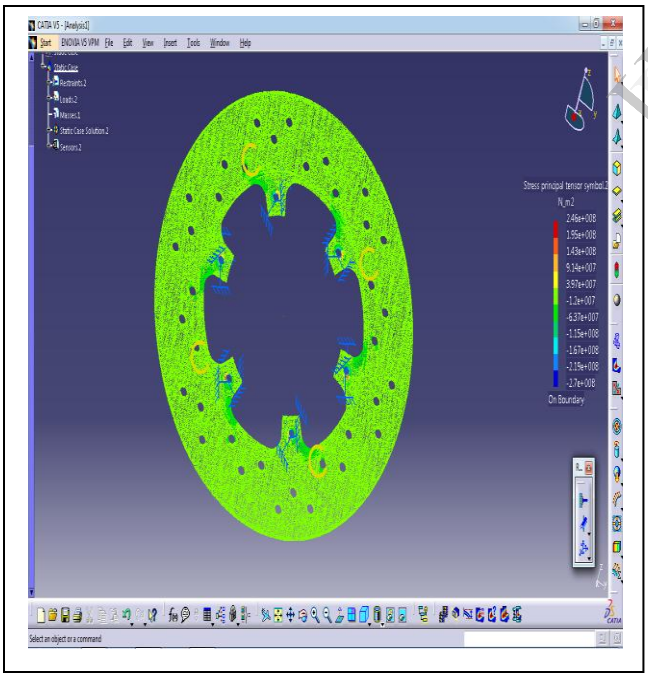

Fig. 7 – Principal Stress (Chroma)

INFERENCES

The yield strength of chroma is considerably higher than structural steel which makes more efficient in handling the stresses generated during braking.

Thus using a chroma disk we have the freedom of increasing the brake force while keeping the design structurally safe which will help in improving braking performance.

3. CHANGING DIMENSION OF THE DISC

As stated in the results above for structural steel the maximum stress generated is 75MPa which is very low compared to the yield strength which is 250MPa.
Thus reducing the thickness of the disk will keep the design safe despite of increased stress levels even after keeping a standard factor of safety.
Reducing the thickness will reduce material usage which will cut down on cost.
Overall weight optimization will also be aided. The thickness of the disc used in existing designs is 3.8mm. We reduced the thickness of the disc by 1mm to optimize the disc on the basis of our objectives. The following results are stated by making the thickness 2.8mm

1. DEFORMATION RESULT

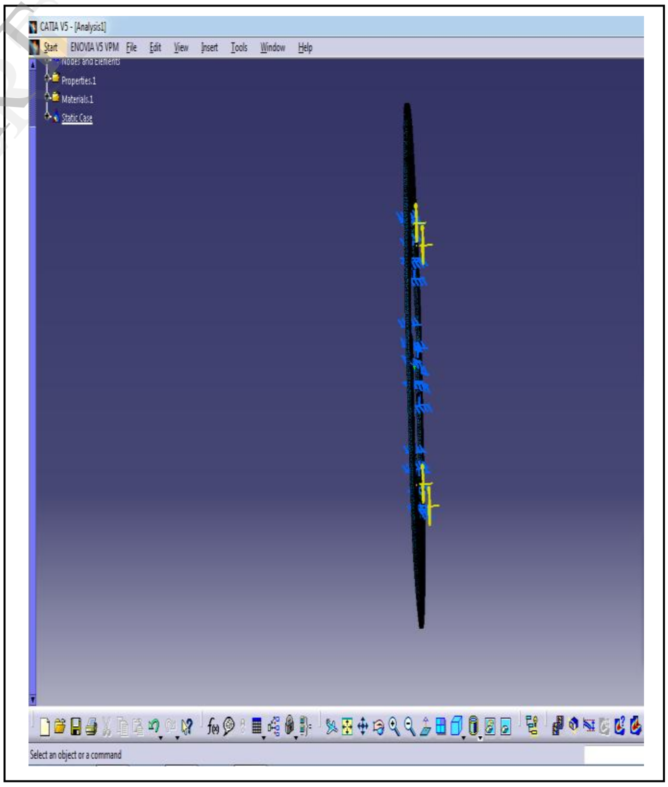

Fig. 8 – Deformation (Thin)

2. VON MISES DIAGRAM

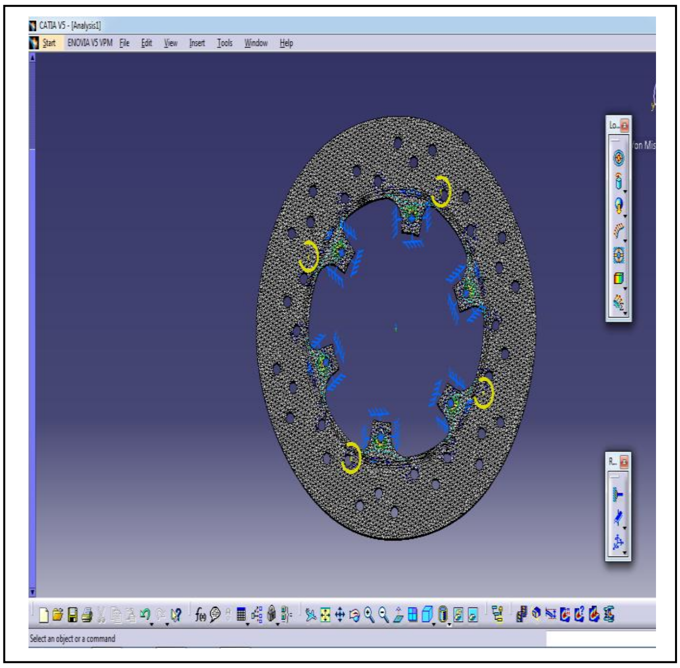

Fig. 9 – Von Mises (Thin)

3. PRINCIPAL STRESS RESULT

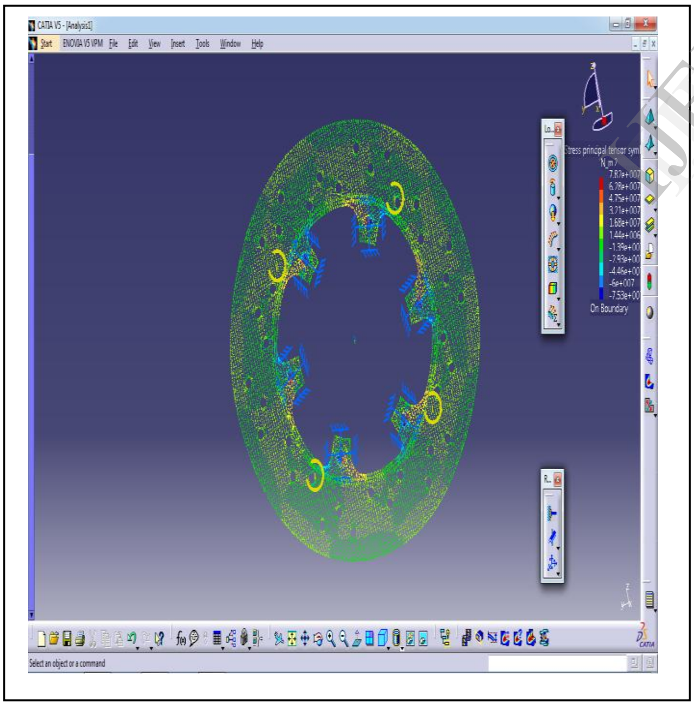

Fig. 10 – Principal Stress (Thin)

INFERRENCES

Using structural steel as material and a reduced thickness of 2.8 mm it was observed that the deformation still remained negligible but the maximum stress induced increased to 116MPa. However since structural steel has a yield strength of 250 MPa, even this result is safe after applying a standard factor of safety of 1.5 .

CONCLUSION

TABLE 1 – COMPARATIVE STUDY

Material Used	Structural Steel	Structural Steel	Chroma
Thickness	3.8 mm	2.8 mm	3.8 mm
Yield Strength	250 MPa	250 MPa	360 MPa
Max. Stress	75 MPa	116 MPa	75 MPa
Deformation	0.0125 mm	0.0178 mm	0.0125 mm

ACKNOWLEDGMENT

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