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"A Study on Disc Brake Design & Analysis Using Topology Optimization Technique"

Nishant Singh
M.Tech (Machine Design, Millennium Institute of Technology, Bhopal)
Asst. Prof. Amit Kaimkuriya
(Department of Mechanical Engineering, Millennium Institute of Technology, Bhopal)

Abstract— In automobile the new models are proposed every day, the motive of today automobile industry is to produce vehicle with better performance and which is light in weight. In this, studies shows that disk brake are mostly used in automobile because the stopping power of disk brake is better than other commonly used brakes, but conventional disc brake are very bulky and heavy in weight. The present study focuses on the optimization of disc brake using topology optimization without affecting the basic performance of the disc brake. The ANSYS software is used for topology optimization. This topology optimization gives information of the part of material to be removed without affecting of disc brake performance. In this study, it was observed that 29% overall weight of disc brake is reduced without effecting disc brake performance.as the weight is reduced the overall cost is also gets reduced.

In this paper a different topology design is implemented to reduce the weight of disc rotor and for good wearing conductivity. Hopefully this paper will help everyone to understand analysis of disc brake rotor and how disc brake work more efficiently, which can help to reduce the accident that may happen. Modeling was done using CATIA V5R20 software and Static Analysis and Topology optimization was done using ANSYS 19 software.

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Keywords- Disc brake; Finite element analysis; Finite element method.

I. INTRODUCTION

Brakes are very important mechanical mechanism on vehicle such that the safety and the life of the human being is mainly depend on the present barking system on the vehicle which get applied during the emergency condition. Estimation made by someone that for an average one person apply at least 50000 time brakes on vehicle in one year. As per the definition, the brakes are the mechanical mechanism which absorbs the energy in the form of heat during the stopping of the vehicle. The main purpose of the brakes is to resist the motion of the vehicle using frictional force.

A. TYPES OF BRAKE SYSTEM

1) Hydraulic braking system

The hydraulic brakes are mainly work on the basis of hydraulic pressure which works on the principle of Pascal's law.

Pascal's law: The pressure exerted anywhere in a mass of confined liquid is transmitted undiminished in all directions throughout the liquid. Applied in hydraulic lifts, hydraulic brakes etc.

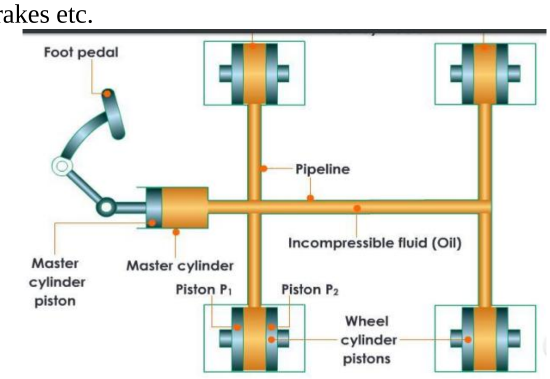

Figure: 1 hydraulic brake system

2) Air braking system

Air brakes use compressed air to make the brakes work. Air brakes are a good and safe way of stopping large and heavy vehicles, but the brakes must be well maintained and used properly. Air brakes are really three different braking systems: service brake, parking brake, and emergency brake.

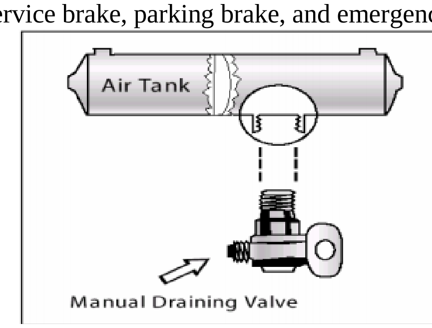

Figure: 2 Drain system

3) Based on type of rotor discs

(a) Disc Brake Rotors.
(1) Based on mounting

Hub mounter disc: The other name of the disc brake rotors are brake discs and it is generally mounted on the hub and it rotate with the wheel. During the clamping brake pads the friction generate between the brake pad and the rotor disc. This friction only reduce the speed of the vehicle and dissipate intense amount of heat to the atmosphere. It is recommended that the strength of the rotor disc should be high such that it can withstand under high temperature. The material used for manufacturing the disc rotor is cast iron and many other materials are adopted to overcome the heat dissipation from it.

Figure: 3 Hub mounted disc brake

Peripheral rotor disc.: These types of rotor discs are generally mounted on the rim of the vehicle wheel and the calliper is attached from inner side of the rotor disc.

Figure: 4 Peripheral rotor disc

4) Based on design

(a) Non ventilated/solid disc rotor: Non ventilated rotors are also called solid rotors. These types of rotors do not possess ventilated structure as it do not dissipate the heat generate due to the friction between the brake pads and rotor disc. The construction of the solid rotor disc is the single solid disc with flange at the middle portion.

Figure: 5 Solid/non ventilated disc rotor

(b) Ventilated discs rotor:

These types of disc rotor manufactured such that there is gap between the two discs and the ribs are provided between the two discs. The gap between them helps to dissipate the heat generated by the friction force applied between the rotor and the brake pads. When the air pass between the two discs it remove the heat by providing the air cooling hence prevent the fading of the brake

Figure: 6 ventilated disc brake

II. LITRETURE REVIEW

(Manthan Vidiya 2017) studied the theory on brakes through thermal analysis, and calculated the energy conversion of kinetic energy of the car to heat energy from brakes. It was done calculate the convection currents due to air flow on the car to find out the rise in temperature of the disc brakes. The analysis was performed in software and the results were obtained in form of graphs and table data. These results were then compared with the actual result data obtained from the car using sensors, hence its reliability and accuracy was checked.

(Qifei Jian 2017) studied the transient field of temperature in automobile under hard braking conditions. In this experiment a test was carried out professionally to study the conditions in hard braking of automobile and results were obtained for circumferential and radial directions. These results when compared to the simulated results using FEA, it was found to be equal.

(Prof. Swapneel D. Rawool 2017) performed thermal analysis using steady state analysis on a disc rotor of a two wheeler and evaluated the performance of braking under conditions of hard braking. The loading conditions, mathematical inputs, and calculations of various parameters were based on assumptions. For the design solid works 15 was used and for analysis Ansys 14.5 was used.

(Vijay Kumar B P 2016) performed by designing the model in CATIA V5, used Hypermesh for meshing and obtained results using Ansys software. Different parameters such as stress, deformations, strain analysis, vibration, thermal analysis etc. were calculated. The study object was a disc brake from a two wheeler. When brakes are applied the rotor undergoes friction with the friction material and hence the vehicle is stopped. This evolves heat energy due to absorption of kinetic energy from the brakes.

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(Vijay dadi 2015) In the present paper material and design optimization was performed on the rotor disc of the disc brake. Three different materials were used for disc which includes CI, SS and aluminium alloy. The analysis was performed on two proposed model one is solid model and other is ventilated model. In structural analysis it was found that aluminium ventilated disc performed better with minimum displacement of 0.260121mm and for equivalent stress SS ventilated Disc performed better with stress value 134.06Mpa, during thermal analysis temperature produced during the braking for ventilated CI disc was less with value 187.5K the thermal gradient was observed less for ventilated aluminium disc with value 185.02 K/mm. The result shows that heat dissipation was more in ventilated aluminium rotor disc.

(Mr. Nilesh R. Farande 2014) In the research modal analysis was performed using the ANSYS software on the drilled wear disc brake rotor. The model was created using CAD software. The analysis was performed for two model i.e. Specimen-1,2.In the result it was analyzed that for first six natural frequencies the thickness of the specimen was inversely proportional to the vibration while for 7^{th} frequency directly proportional to it with slight difference. It was also revealed that increase in flange hole diameter decreased the vibration for first 6 frequency. For specimen 2 there was reduction in natural frequency with increase in the diameter of the ventilated hole.

(Viraj Parab 2014) In the research comparative analysis was performed between two materials in disc brake rotor. For the analysis transient thermal analysis was performed and the rotor. The material taken for comparative analysis was the stainless steel and the cast iron. In the result it was found that on behalf of deformation cast iron performed better but in case of stress stainless steel performed better. The obtained results were safe as per the design criteria.

III. OBJECTIVE

1. To optimize the design of the disc brake rotor using topology optimization.
2. To reduce the weight of disc brake rotor without effecting the performance of the rotor disc.
3. To reduce the manufacturing cost by removing the unnecessary material from it.

IV. EXPECTED METHODOLOGY

The procedure followed in this research work of Piston is as follows;

A. Modeling

For the present work CATIA V5 is used for model design. CATIA V5 mainly used for design. CATIA supports multiple stages of product development (CAX), including conceptualization, design (CAD), engineering (CAE) and manufacturing (CAM). CATIA facilitates collaborative engineering across disciplines around its 3DEXPERIENCE platform, including surfacing of components & shape design of mechanical parts, electrical system and component design, fluid flow engineering and electronic systems design, mechanical engineering and systems of general engineering. Finite Element Analysis

B. Analysis

ANSYS is metaphysics software used for performing various finite element and computational fluid flow based analysis simulations to check the working of various engineering components. There are basically two types of analysis which are commonly performed using Ansys software. These analyses are FEM i.e. finite element method and CFD i.e. computational fluid dynamics. In this analysis virtual models of components or systems are created to perform the actual analysis using the similar boundary conditions as used in the live environment.

C. DESIGN CALCULATION

The brake pedal:

Brake pedal is a mechanical component used in the brake system of motor cycles and cars where the driver foot is used for applying the pressure to stop the vehicle in the running condition. In this system the increase in force is always equal to the multiplication of lever ratio used in the levels of the brake pad assembly.

$$F_{\text{brake pedal}} = F_d \times [L_1 / L_2]$$

Where,

F_{bp} = the force output of the brake pedal assembly

F_d = the force applied to the pedal pad by the driver = 370 N

L_1 = the distance from the brake pedal arm pivot to the output rod clevis attachment

L_2 = the distance from the brake pedal arm pivot to the brake pedal pad

$$(L1/L2 = 4)$$

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The incompressible liquid is assuming in master cylinder of disc brake and its fixed hydraulic vessels, the pressure calculation of master cylinder will be denoted in formula

$$P_{mc} = \frac{F_{bp}}{A_{mc}}$$

Where

P_{mc} = the hydraulic pressure generated by the master cylinder.

A_{mc} = the effective area of the master cylinder hydraulic piston = 0.000285 m^2.

Brake fluid, brake pipes and hoses: Assuming no losses along the length of the brake lines, the pressure transmitted to the calipers will be equal to:

Pcal = Pmc

Where,

Pcal = the hydraulic pressure transmitted to the caliper.

The caliper, Part I: The one-sided linear mechanical force

Generated by the caliper will be equal to:

Fcal = Pcal × Acal

Where,

Fcal = the one-sided linear mechanical force generated by the caliper.

Acal = the effective area of the caliper hydraulic piston(s) found on one half of the caliper body = 0.0007068 m^2.

The caliper, Part II: The clamping force will be equal to, in theory, twice the linear mechanical force as follows:

$F_{Clamp} = F_{cal} \times 2$

Where,

FClamp = the clamp force generated by the caliper.

The brake pads: The clamping force causes friction which acts normal to this force and tangential to the plane of the rotor. The friction force is given by:

$F_{friction} = F_{Clamp} \times \mu_{bp}$

$F_{friction}$ = the frictional force generated by the brake pads opposing the rotation of the roftor.

μ_{bp} = the coefficient of friction between the brake pad and the rotor = 0.4 (assumed).

The rotor: This torque is related to the brake pad frictional force as follows:

Tr = Ffriction × Reff

Where,

Tr = the torque generated by the rotor.

Reff = the effective radius (effective moment arm) of the rotor (measured from the rotor center of rotation to the center of pressure of the caliper pistons).

This torque generated by the rotor will be equal to the torque required to stop the vehicle. In this report, they follow

- Mass of the vehicle = 300 kg.
- Maximum velocity of the vehicle = 80 km/hr or 22.22 m/s.
- Stopping Distance = 11.69 m.
- Tire Size = 23 in diameter that is 584.2 mm with 7 mm thickness
- Disc flange or thickness = 16 mm.
- 50-50 wheel bias that is equal braking force is generated in all the 4 wheels of the vehicle.

Total force generated during braking to stop the car,

F = m×a, a = deceleration during braking = $v^2/2s$ = $22.22^2/2$ x 11.69 = 21.12 m/s^2

F= 300 x 21.12

F= 6336 N.

Torque required stopping the vehicle,

$T_r = F/4 \times R_w$

T_r = 6336/4 x 0.2921

T_r = 462.54 N-m.

As mentioned in above formulae,

$F_{bp} = F_d \times (L_1/L_2)$

F_{bp} = 370 x 4

F_{bp} = 1480 N.

P = Fbp/Amc

P_{mc} = 1480/0.000285

P_{mc} = 5192982.456 Pa

$P_{mc} = P_{cal}$ = 5192982.456 Pa

F_{cal} = Pcal × Acal

F_{cal} = 5192982.456 x 0.0007068

F_{cal} = 3670.4 N.

Clamping Force = $2F_{cal.}$

F_{clamp} = 7340.8 N.

$F_{friction}$ = Frictional force generated on the rotor during braking process,

$F_{friction}$ = 7340.8 × 0.4

$F_{friction}$ = 2936.32 N

Torque generated by the rotor during braking = $F_{friction} \times R_{eff}$ = 462.54

Therefore, the effective rotor radius R_{eff} = 0.1575 m.

Thus, the Effective Rotor Radius is 0.1575 meters that is 6.2 inches or 157.5 mm. And thus, the effective diameter is 315 mm.

Based on this effective diameter, the outer diameter of the disc is decided to be 381 mm and the inner diameter to be 125 mm.

Kinetic Energy developed during braking,

KE = ½ mv^2

KE = ½ x 300 x $(22.22)^2$

KE = 74059.26 J

Total Braking Energy/Heat required for the vehicle is equal to the

Total Kinetic Energy generated by the vehicle,

Thus Heat (Q) generated,

Q_g = 74059.26 J

Since assumption of 50-50 wheel bias is made, this heat will be equally distributed in the 4 wheels of the car, thus equally distributed in the 4 rotors. So, heat generated in 1 rotor, Q_g = 18514.815

Now, the stopping time of the vehicle will be velocity/deceleration,

t = v/a

$t = 22.22/21.12$

$t = 1.05$ sec.

Hence, power generated in one rotor

$P = Qg/t$

$P = 18514.815/1.05$

$P = 17633.16$ Watts.

Thereby, we can calculate the heat flux through one disc rotor with 0.381m outer diameter and 0.125m inner diameter.

Heat flux = $4 \times P/3.14 \times (Do^2\text{-}D_i^2)$

Heat flux = $4 \times 17633.16/3.14 \times (0.381^2 - 0.125^2)$

Heat flux = 173408.3233 Watts/m^2.

D. MATERIAL PROPERTIES

Generally the material can used in disc brake is gray cast iron because its good tensile strength and thermal conductivity, In addition, the excellent properties of Grey Cast Iron have made it one of the most widely used alloys. Its properties are as follows:

Table 1: Properties of gray cast iron.

Density (kg/m3)	7200
Young's Modulus (GPa)	125
Poisson's ratio	0.25
Thermal Conductivity (W/m-K)	54.5
Specific Heat (J/kg-K)	586
Coefficient of friction	0.25

E. STEP OF WORKING

1) Collecting information and data related to the hydraulic press plate.
2) A fully parametric model of the hydraulic press plate is generated using CatiaV5
3) Model obtained in Step 2 is analyzed using ANSYS 15.
4) Manual calculations are done.
5) Finally, we compare the results obtained from ANSYS

F. STEPS OF ANSYS ANALYSIS

The different analysis steps involved in ANSYS are mentioned below.

1) Pre-process

The model setup is basically done in pre-processor. The different steps in pre-processing are

2) Building the model

The CATIA provides the following approaches for model generation: Creating a solid model within CATIA. Every design starts with the conventional calculations by applying various fundamentals of design. The basic model of disc brake is creating in following step.

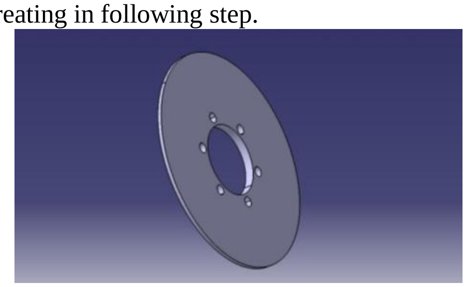


Figure 6 Final base model of Disc brake in CATIA V5

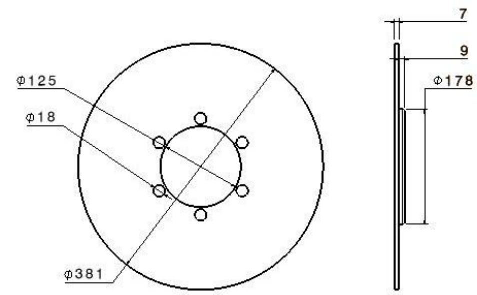


Figure 7 Sketch of Basic Disc Brake model

After the analyze of basic model. The new mo0del is proposed

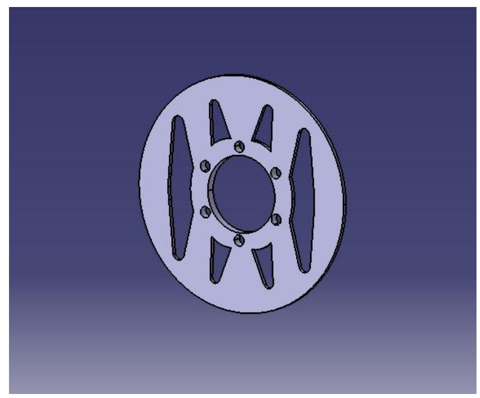


Figure 8 Final topology optimized disc brake model

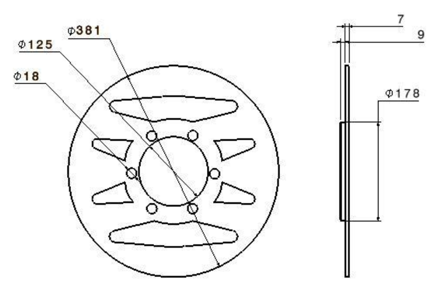


Figure 9 Sketch of topology optimize Disc brake model

G. Meshing

ANSYS Meshing includes intelligent, general-purpose, automated high-performance type of product. It delivers the most suitable work for exact, proficient Multi physics arrangements. A work appropriate for a particular investigation can be created with a solitary mouse click for all parts in a model. For the master client who needs to tweak on it give full controls over the alternatives used to create the work are accessible. The energy of parallel preparing is consequently used to decrease the time you have to wait for mesh generation.

Creating a mesh in the imported geometry is an important step in ANSYS analysis as the size of the finite element is decided by the mesh properties. Finer the mesh is, more accurate are the results.

1) Meshing of base model

After the given meshing in ANSYS the number of element is 13888 and number of Nodes is 27527.

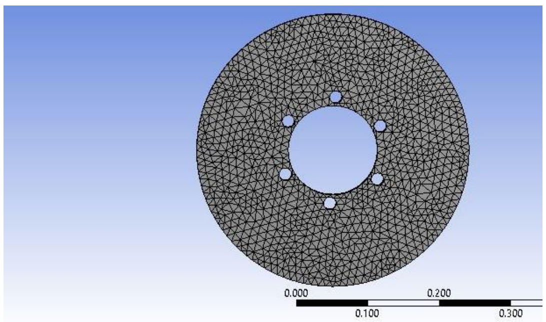


Figure 10 Meshing of base model

Table 2 Nodes & Element

Number of Nodes	15745
Number of Elements	7753

2) Meshing of Topology optimize model

After the given meshing in ANSYS the number of element is 13888 and number of Nodes are 27527.

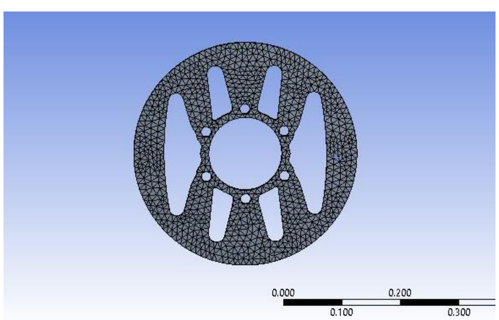


Figure 11 Meshing of topology optimize model

Table 3 Nodes & Element

Number of Nodes	12141
Number of Elements	5471

H. BOUNDARAY CONDITION

The next step in the static structural ANSYS analysis is to apply the boundary conditions. All six hole are fixed in applying fixed support and the after applying a force, this force are applying in disc brake in to opposite surfaces of disc brake. The total applying force is 2936.32 Newton is apply in top and bottom of disc brake. The above analysis is study in total deformation equivalent stress and strain analysis. Giving 27.63 rpm rotational velocity for disc brake by clockwise direction run calculation and monitoring the solution. Since Also since the disc has to be fixed at its centers, fixed supports are given to the hub bolts and the inner portion of the entire inner circle. Thus, overall there are 4 initial boundary conditions given to the disc rotors model or geometry before proceeding to the solution.

1) Applying fixed support

All six hole and centre part of disc brake are fixed before run simulation

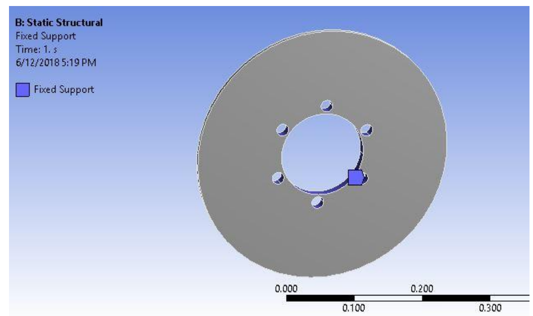


Figure 12 Fixed support in base model

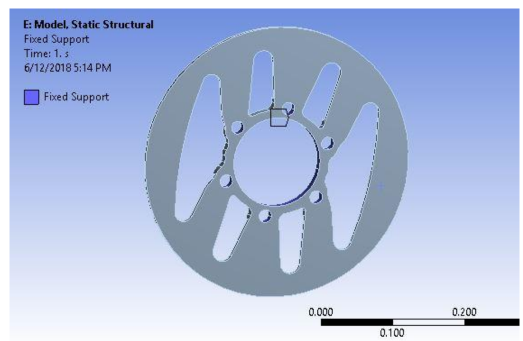


Figure 13 fixed support in base model

2) Applying force on disc brake

a) For base model

After applying fixed support 2936.32 Newton force is applied in both the face of disk brake

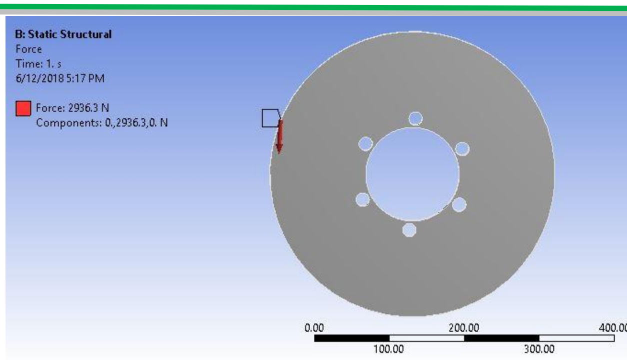


Figure 14 Applying force in top face in downward direction on disc brake

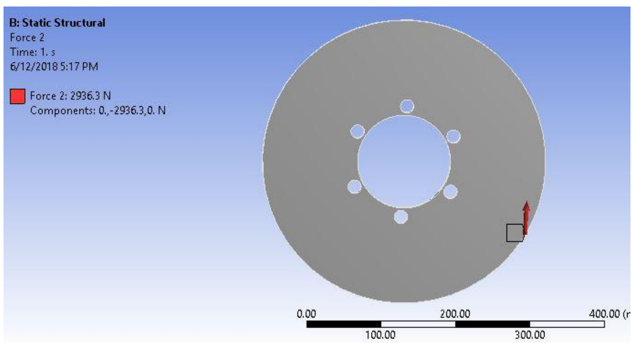


Figure 15 Applying force in bottom face in upward direction on disc brake

b) For topology optimize model

After applying fixed support 2936.32 Newton force is applied in both the face of disc brake

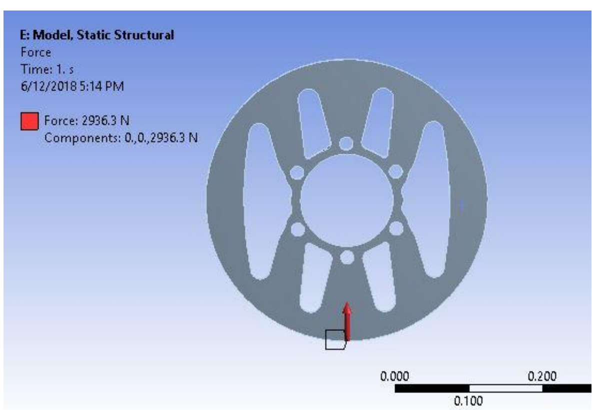


Figure 16 Applying force in top face in upward direction on disc brake

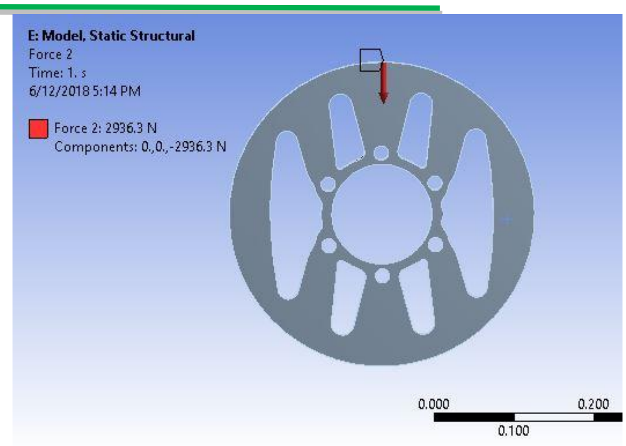


Figure 16 Applying force in bottom face in downward direction on disc brake

I. ANSYS analysis of disc brake base model

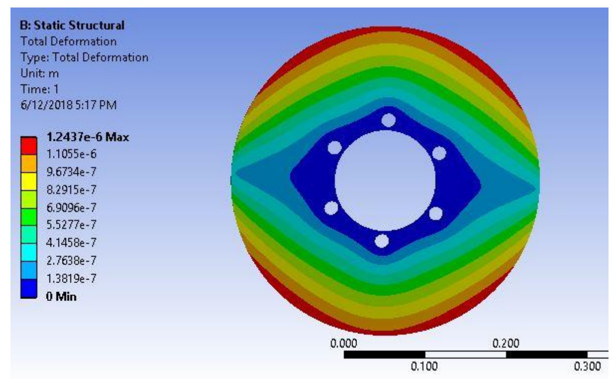


Figure 17 Total deformation of disc brake base model

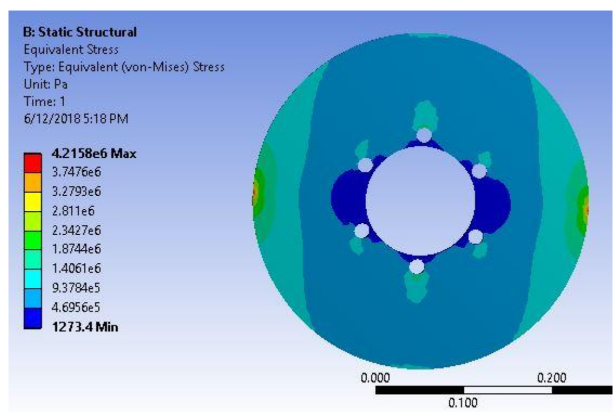


Figure 18 equivalent stress of disc brake base model

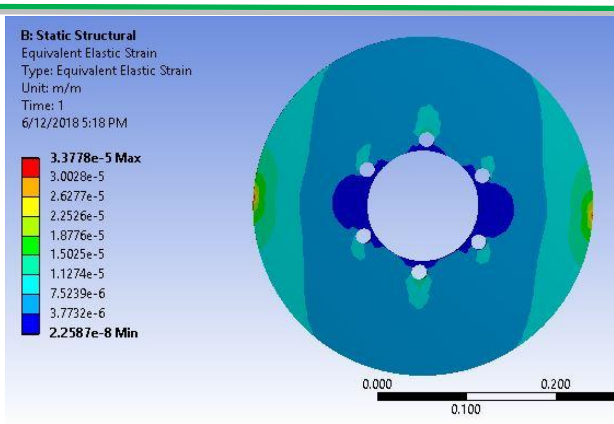


Figure 19 Strain of disc brake base model

J. ANSYS analysis of disc brake Topology optimize model

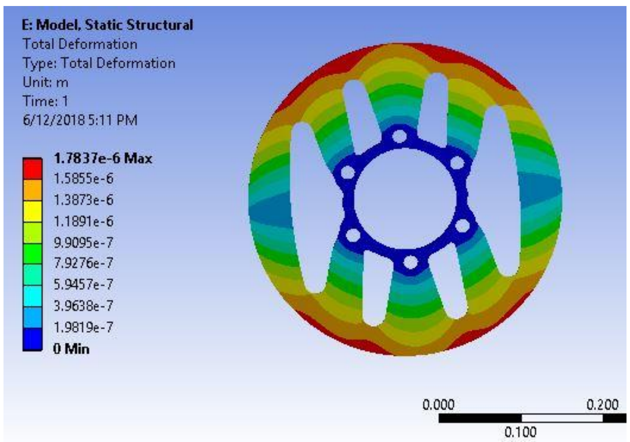


Figure 20 Total deformation of disc brake Topology optimize model

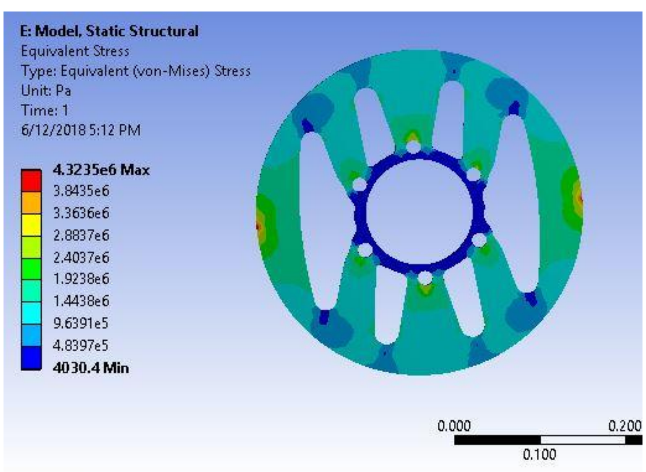


Figure 21 Equivalent stress of disc brake topology optimize model

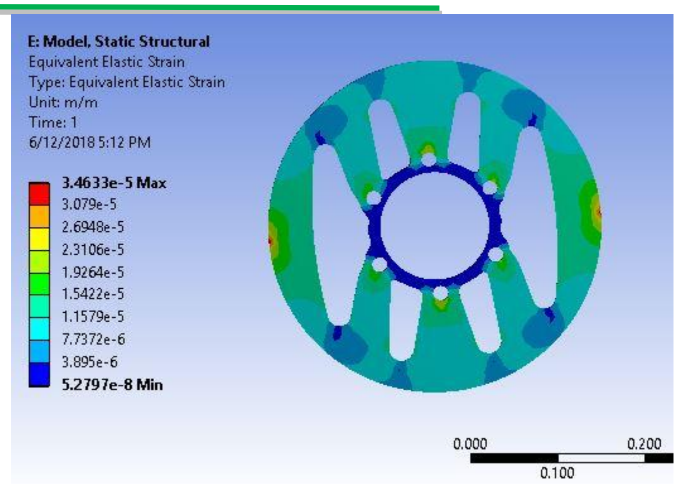


Figure 22 Strain of disc brake base model

V. RESULT

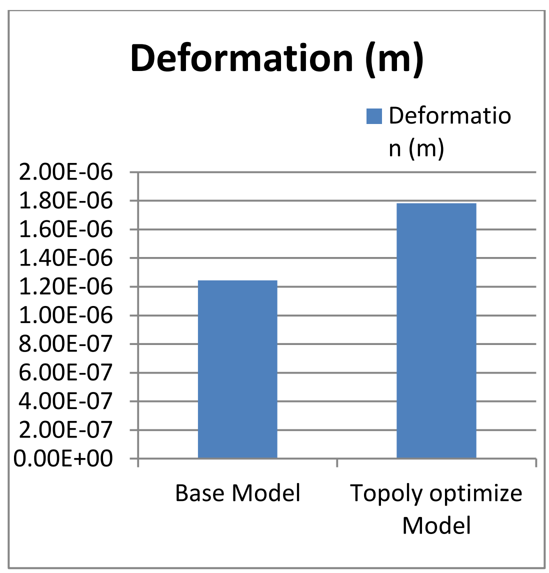
After the designing and analysis following result are obtain in ANSYS.

A. Comparison deformation of disc brake

The maximum deformation of disc brake base model is 1.2437e-6 m and total deformation of topology optimize model is 1.7837e-6m

Table 4 Result of deformation

Disc brake model	Deformation (m)
Base model	1.2437e-6
Topology optimize model	1.7837e-6



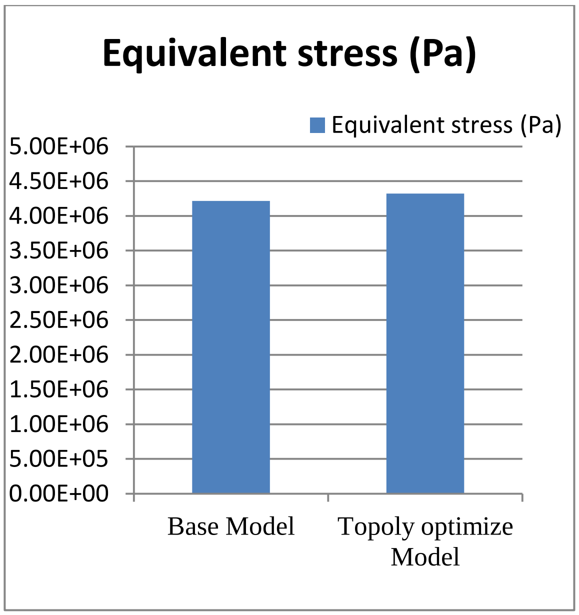
Graph 1 Comparison deformation

B. Comparison Equivalent stress of disc brake

The maximum equivalent stress of disc brake base model is 4.2158e6 Pa and equivalent stress of topology optimize model is 4.3235e6 Pa.

Table 5 Result of Equivalent stress

Disc brake model	Equivalent stress (Pa)
Base model	4.2158e6
Topology optimize model	4.3235e6



Graph 2 Comparison of Equivalent stresses

C. Comparison Strain of disc brake

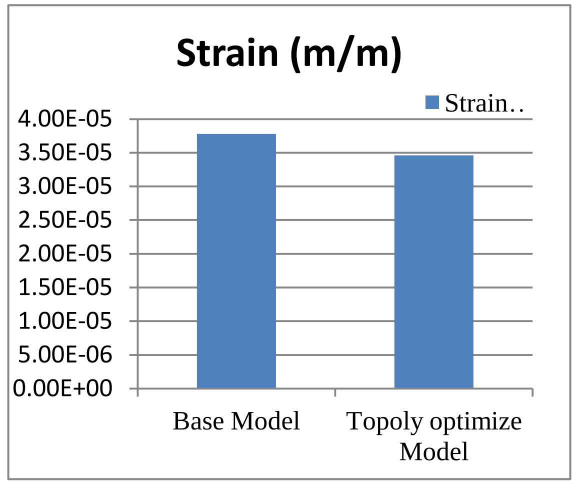
The maximum Strain of disc brake base model is 3.778e-5 and Strain of topology optimize model is 3.4633e-5.

Table 6 Results of Strain

Disc brake model	Strain (m/m)
Base model	3.778e-5
Topology optimize model	3.4633e-5

Graph 3 Comparison of strain

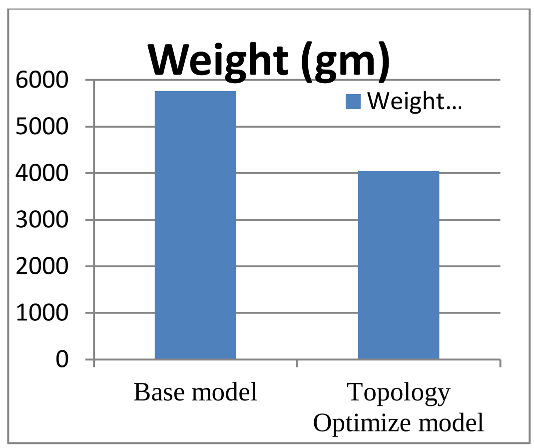
D. Comparison Weight of disc brake



The Weight of disc brake base model is 5761.3 gm and weight of topology optimize model is 4046 gm.

Table 7 Result of Weight

Disc brake model	Weight (gm)
Base model	5761.3
Topology optimize model	4046



Graph 4 Comparison of Weight

VI. COUNCLUSION

The disc brake is very necessary part of any automobile vehicle the good variety of disc brake is responsible for increasing the overall performance of automobile vehicle. The new propose model of disc brake reduction in weight and easy to design compared to traditionally used disc brakes. The advantages of removing unnecessary material in disc brake are as follows:

The overall weight of single disc brake is reducing 29.78%. Generally four disc brakes are used in single vehicle the total weight reduction of any vehicle is 6861.2 gm. These weight reductions of disc brake are increasing the overall performance of vehicle. The proposed car weight is 300kg.

The new proposed disc brake stress in under the limit of yield stress so overall design is safe. The ultimate stress of grey cast iron is 1.4e8 Pa and after applying the force the stress generated in disc brake is 4.3235e6 Pa. The new proposed design has factor of safety 32 which is under safe limit.

VII. REFRENCE

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