

Design and Analysis of Automotive Disc Brake using FEM

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Abstract - The aim of this project is to design and analyze the automotive disc brake of Bajaj Pulsar 220F. Disc brakes are subjected to very high contact pressure and intensive heating of friction surface. The effectiveness of braking system depends on friction coefficient, stability during braking and wear of friction material. Heat dissipation is important parameter in braking which affects the life of the brake disc. In this work 3D models of disc brakes of Bajaj Pulsar 220F having same diameter with different shape and pattern of holes are prepared in CATIA software. The objective of this work is to investigate and analyze the temperature distribution of rotor disc during braking operation using ANSYS Workbench. Application of specified braking torque leads to generation of heat flux. The heat flux generated and heat transfer coefficient taken into consideration are numerically analyzed, which are then used to calculate the rotor rigidity, maximum temperature rises on the brake disc. The work uses the finite element analysis techniques to predict the temperature distribution on discs. The static and thermal analysis under different boundary conditions is performed on the disc brake using ANSYS software. The maximum temperature observed and stresses developed in the different types of disc brake models are compared. The influence of variation of shapes and patterns of holes on heat dissipation of brake disc is studied and the optimized disc model is selected based on the analysis results.

Key Words: Surface Friction, Friction Coefficient, Heat Dissipation, Heat Flux, Heat Transfer Coefficient, Temperature Distribution, Static and Thermal Analysis.

1. INTRODUCTION

The brake is mechanical device that retards and stop the motion of vehicle. Brake absorbs kinetic energy of the wheels and the energy is dissipated in the form of heat energy to the surrounding atmosphere. Generally, in automobile industry drum brakes and disc brakes are widely used. Disc brakes are more reliable than drum brakes. Disc brake system consists of brake caliper, friction pads and rotor. Disc brakes are subjected to very high contact pressure and intensive heating of friction surface.

The degree of reliability of the braking action is directly proportional to the firmness of the entire braking system. The parameters which mainly affects brake performance are friction coefficient, stability during braking and wear of

friction material. Ideally, during braking, the heat generated should be quickly distributed to surrounding to avoid the reduction of coefficient of friction between disc brake and brake pad. The high temperature at the contact surface may cause the brake fluid to vaporize due to overheating.

The heat from contact may also cause thermoelastic deformation which will vary the coefficient of friction at brake pad surface. This may change the contact pressure distribution at the contact surface. This may create a local high temperature gradient at the contact surface called hot spot. The localized hot spot will cause high local stress that lead to material degradation and failure. Thermal deformation can also initiate major problems such as variation of coefficient of friction of the brake pad, brake fade and vapor lock. If there is inconsistent dissipation of heat, there is intensive heating of brake disc, which causes deformation of the material which leads to failure of disc.

2. LITERATURE REVIEW

Aleskander Yevtushkendo, Michal Kuciej [1] studied a two-element model of braking process for a tribosystem consisting of the pad (the strip) sliding with the time-dependent velocity (braking at uniform retardation) on a surface of the disc (the semi-space). The dependences of temperature and thermal stresses on the boundary conditions on upper surface of the ceramic-metal strip was. Analysis investigated of the evolution of thermal stresses in the frictional elements during braking proves that when it is heated considerable normal compressive stresses occur near the contact surface.

The value of this stresses decreases with an increase of time and after some time, moment t_c the sign changes - which means that the tensile stresses take place. The time when it happens increases monotonously with increase of a time braking t_s . On the basis of achieved numerical data it is established, that the possible initiation of the superficial crack during braking can be described as the series of the following phases:

- Due to local intensive frictional heating near a contact surface the field of compressive stresses is formed;
- after the beginning of braking in the sometime moment t_c the tensile normal stresses occur near the subsurface region; and

- when these stresses exceed the tensile strength of the material, the initiation of the surface cracks is possible.

M. Boniardi, F. D. Errico, C. Tagliabue, G. Gotti, G. Perricone [2] had done research on factors affecting the life of the brake disc and effect of choosing different kind of steel, characterized by greater resistance to the temperature process. In their paper 'Failure Analysis of motorcycle disc brake' they had done various analysis, at first visual and chemical analysis were done which were followed by hardness test, fractographic analysis and test of resistance to tempering for finding the impact of various factors on the life of brake disc also the same tests are followed for the different steel material and all results were analyzed.

It is found that the life of brake disc depends upon geometrical factors such as position of holes, shape of spokes etc. as well as other factors such as chemical composition of steel, microstructure also affects the life of the brake disc.

Deepak S. Hugar, Prof. U. B. kadabadi [3] had done "Design and Thermal analysis of disc brake for minimizing temperature" and studied analysis of different shapes of slot of different vehicle Disc rotor brake. They optimized number of shapes of slot to estimate good thermal conductivity of disc brake rotor. They had done thermal analysis on real model of disc brake rotor of Bajaj Pulsar. Modelling was done using CATIA V5 R21 software and static and transient thermal analysis was done using ANSYS 15 software.

Their study provided a useful design and help to improve the brake performance of disc brake system. Minimum temperature distribution obtained in modified disc as compared to actual standard disc. They found that new disc is also safe based on strength and rigidity criterion.

W. M. Zurin W. S., R. J. Talib, and N. I. Ismail [4] in this paper, solid and ventilated type of motorcycle disc brake are being analyzed using Computational Fluid Dynamic software. The main focus of the analysis is the thermal behavior during braking for solid and ventilated disc brake. They discussed the comparison between both geometries to determine the better braking performance in term of temperature distribution. The knowledge gained from the project is to be able to understand the relationship between the brake disc geometry and braking performance in term of thermal dissipation. This knowledge can be used in future as reference for similar research and development. Solid disc brake is a flat surface geometry without having any notched or grooves at the disc. It had more contact surface area during braking compared with ventilated disc brake. However, it tends to have more localized thermo-elastic instability on the contact areas. This phenomenon may lead to brake fade and pad glazing problem because they do not have appropriate ventilated hole which can help to dissipate the braking frictional heat to surrounding. Thermal analysis of the disc brake is done by using ANSYS.

Daanvir Karan Dhir [5] studied the rise in temperature of an automotive disc brake at the time of braking and its effect on disc durability using finite element method. The heat flux generated and the heat transfer coefficient taken into consideration were numerically analyzed, which were then used to calculate the rotor rigidity, maximum temperature rises on the disc rotor. The rotor was further loaded with thermo-mechanical cyclic stresses which were used to analyze the durability and fatigue factor of safety of disc. The influence of variations in disc rotor geometry i.e. holes and airfoil vents in comparison to a simple flange type disc were studied and their effect on maximum temperature rise and disc durability has been investigated by modeling and conducting FEM techniques in Solid works and ANSYS respectively.

Abdulwahab A. Alnaqi, David C. Barton, Peter C. Brooks [6] proposed a scaling methodology to evaluate the thermal performance of a disc brake at a reduced scale. The resulting small-scale disc brake has the advantage of low cost and reduced development time. The proposed scaling methodology was validated by comparing the results for the full- and small-scale discs using a conventional brake dynamometer. In addition, a two-dimensional axisymmetric transient thermal finite element model was developed using Abaqus software to assist in the validation of the scaling methodology. The numerical simulations confirmed the equivalence between the full and small-scale disc thermal performance using the proposed scaling methodology and also gave good agreement with the experimental results

3. CASE STUDY

Case study is done on the work of mBAJA team's brake disc of Sinhgad College of Engineering [10]. This study deals with the structural analysis of rotor disc of disc brake of mBAJA ATV through finite element analysis approach using ANSYS software. A disc brake is a device by means of which artificial frictional resistance is applied to the rotating member, in order to stop the motion of a machine or a vehicle. The objective of structural analysis of rotor disc is to study & evaluate the performance under severe conditions & to suggest best combination of parameters of rotor disc like disc geometry, Wall Thickness & Material composition. In this study, an attempt has been made to investigate the effect of stiffness & strength and to analyze the stability & rigidity of the rotor disc. The modelling of rotor disc is done in CATIA software. Further structural analysis and thermal analysis is done by using ANSYS.

3.1 Rotor disc

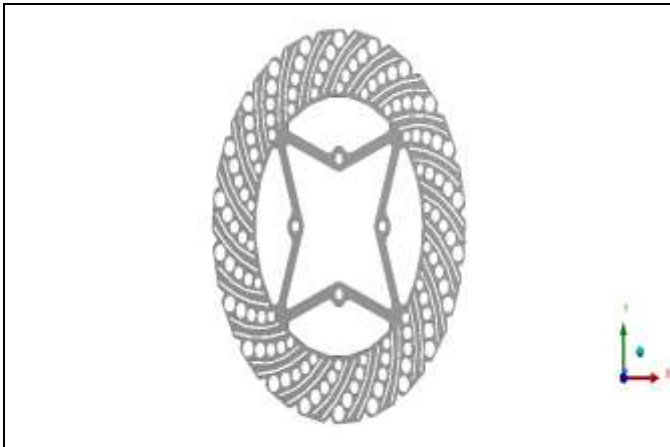


Fig -1: Model of brake disc

The static structural analysis is performed on brake disc by considering the boundary conditions which are torque applied on the disc and the mounting points of the brake disc are fixed and the stress and the deformation are evaluated. The thermal analysis is performed on brake disc by considering the boundary conditions which are heat flux applied on the disc and convection through the disc and temperature is evaluated.

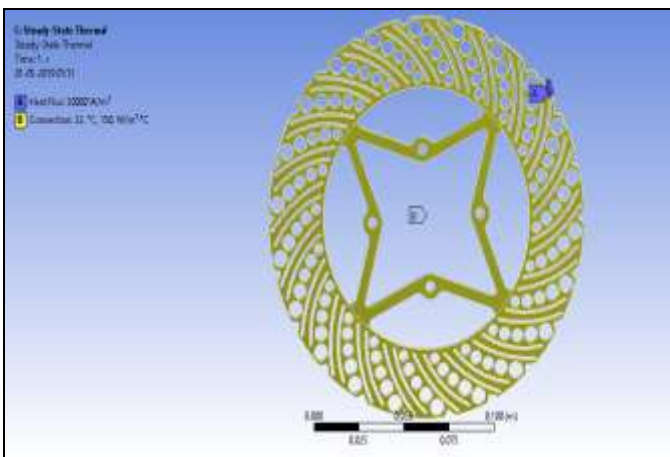


Fig -2: Boundary conditions for thermal analysis

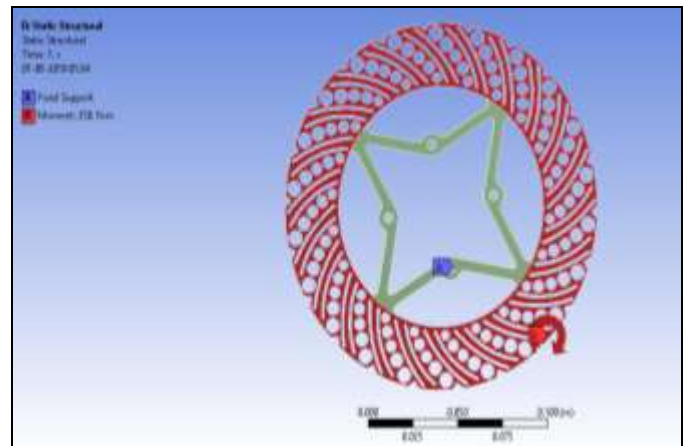


Fig-3: Boundary conditions for structural analysis

3.2 Results

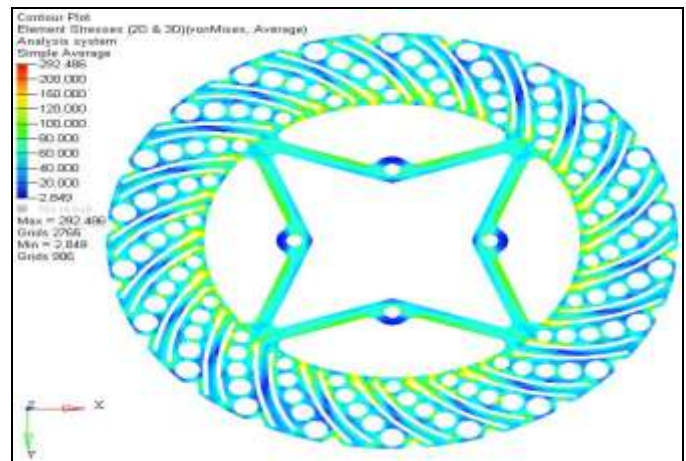


Fig -4: Stress distribution of the brake disc model

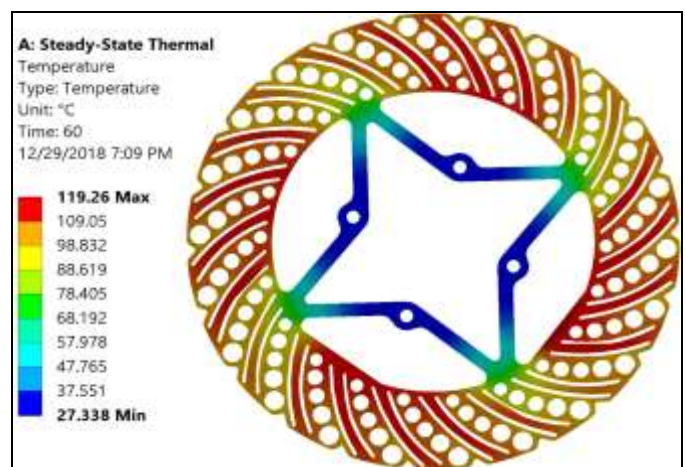


Fig -5: Temperature distribution of the brake disc model

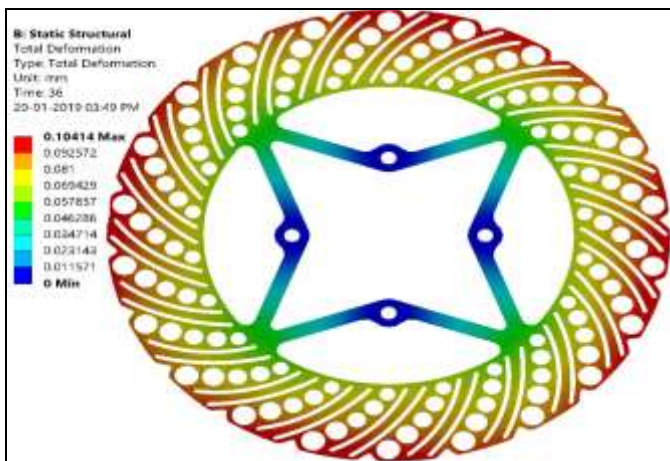


Fig -6: Total deformation of the brake disc model

3.3 Conclusion

The braking system is the backbone of an ATV, determining the best parameters several analyses is done by using Ansys. The structural analysis is done to determine the strength and sturdiness of the disc, along with it deformation, stress, strain, displacement and factor of safety is analyzed to provide the best suitable disc quality for better performance. Heat flux is applied on disc during thermal analysis to determine the heat dissipation. The single stop temperature analysis provides to understand the instant temperature occurred in disc. The brake disc is suitable for efficient performance of BAJA ATV. The disc is enough sturdy and safe for high-duty performance and the master cylinder, rotor positions provides all 4 wheels locking simultaneously, within minimum stopping distance.

Hence, the same procedure can be used in design and analysis of automotive brake disc in our project work.

4. CALCULATIONS

Brake systems are designed for maximum speed to ensure maximum safety, Bajaj Pulsar 220F has maximum speed of 134 kmph. The constraints for Thermal Analysis and Structural Analysis are calculated by assuming that the rider is driving the vehicle at maximum speed and suddenly applying brakes to retard and stop the motion of the vehicle. While calculating for the constraints, several assumptions are considered are as follows [4]:

- All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux.
- The heat transfer involved in this analysis takes place only by conduction and convection. Heat transfer by radiation can be neglected as it is negligible
- The disc material is considered as homogeneous and isotropic.
- The domain is considered as axisymmetric.

- Inertia and body force effects are negligible during the analysis.
- The disc is stress free before the brake application.
- In this analysis, the ambient temperature and initial temperature is set to 22 °C.
- All other possible disc brake loads are neglected.
- Only certain parts of disc brake rotor experience convection heat transfer such as the cooling vanes area, the outer ring diameter area and the disc brake surface.
- Uniform pressure distribution generated by the brake pad onto the disc brake surface is considered.

Numerical Method

Disc brake model: Bajaj Pulsar 220F

Material: Stainless Steel (SS410)

Outer Diameter: 260 mm

Inner Diameter: 130 mm

Mean Diameter: 195 mm

Nomenclatures

q_0	Average braking power in Watt
t	Stopping time
A_1	Piston area of brake calliper
μ	Coefficient of friction between brake pad and calliper
P_{max}	Maximum pressure in calliper
u	Top speed of vehicle
a	Vehicle deceleration
w	weight of the vehicle
D	outer diameter of disc
d	inner diameter of disc

Disc Thickness: 5 mm

Effective rotor radius: 97.5 mm

Top Speed: 134 Kmph=37.22 m/s [13]

Kerb Weight = 155 kg [13]

Assuming weight of passengers=160 Kg ... (for safety)

Piston area of brake caliper = $A_1 = 2000 \times 10^{-6} \text{ m}^2$

For vehicle safety purpose deceleration is 21 ft/s = 6.4 m/s² [2]

Deceleration = 21 ft/s = 6.4 m/s²

$$\text{Stopping Time} = \frac{V}{a} = \frac{37.22}{6.4} = 5.81s$$

The average braking power applied on all wheels is given by,

$$\text{Average braking power} = q_o = \frac{u \times a \times w}{2 \times g}$$

$$= \frac{37.22 \times 6.4 \times 315 \times 9.81}{2 \times 9.81} = 37517.76 \text{ W}$$

Average braking power applied on each wheel of front side,

$$q_o = 37517.76 \times 0.6 \times 0.5 = 11255.33 \text{ W}$$

The dynamic weight transfer during braking on front and rear axle is 60:40 percent of total vehicle weight respectively [1].

Hence,

Weight distribution on front wheels=0.60

Since only one side of brake disc is considered=0.50

$$\text{Swept area of rotor} = \frac{\pi}{4} (0.24^2 - 0.18^2)$$

$$= 0.0198 \text{ m}^2$$

$$\text{Heat Flux} = \frac{q_o}{A \times t} = \frac{11255.33}{0.0198 \times 5.81} = 97837.24 \text{ w/m}^2$$

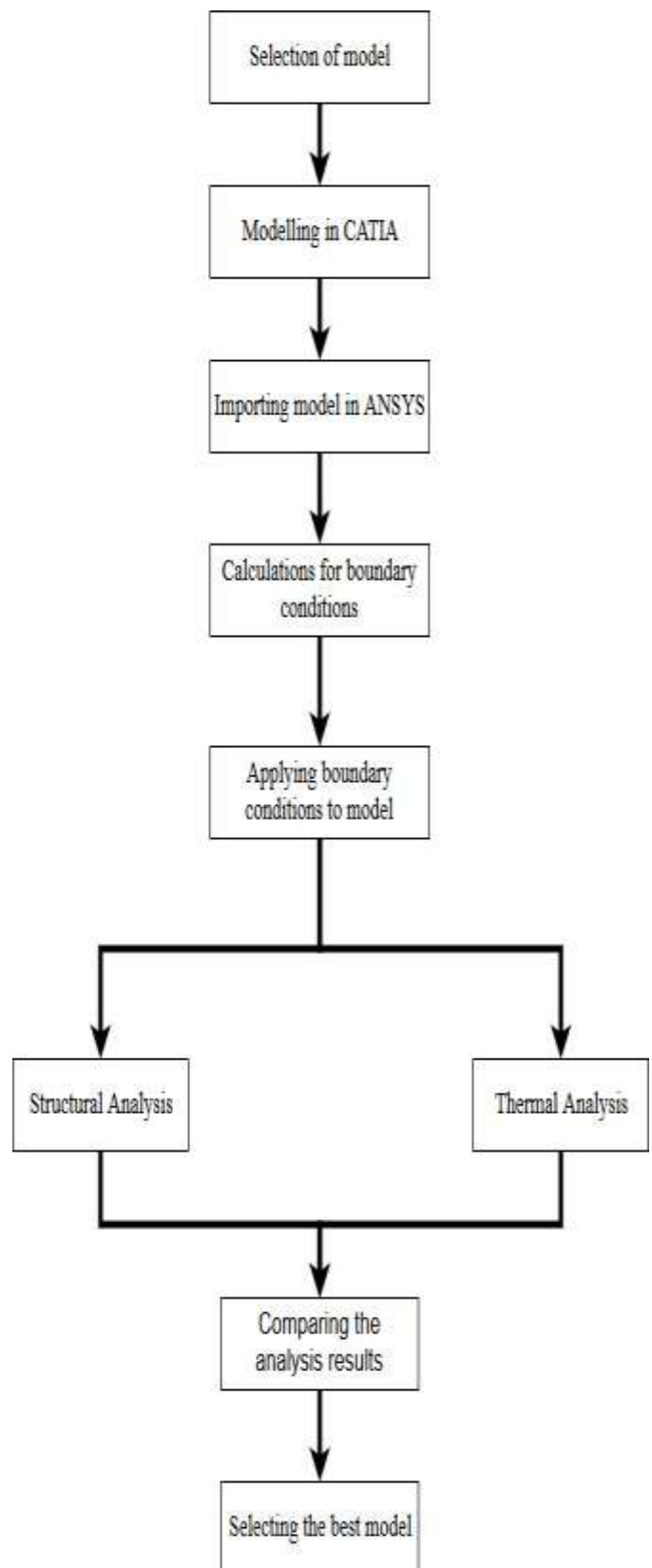
Convective heat transfer coefficient = 200 w/m² [11]

Tangential force between pad and rotor=

$$\mu \times \frac{P_{max}}{2} \times A_1 = 0.5 \times \frac{1 \times 10^6}{2} \times 2000 \times 10^{-6} = 500N$$

$$\text{Braking Torque} = 500 \times 0.195 = 48.75Nm$$

5. DESIGN AND ANALYSIS



5.1 Modeling in CATIA

Disc structure significantly defines the air flow characteristics, different types of disc rotors with different shapes and patterns of holes are developed in CATIA software, for better heat dissipation. Fig.7 is the CATIA model of actual disc which is to be modified. Five different CATIA models are produced by variation of geometry of holes, Shape of slots and pattern of the holes

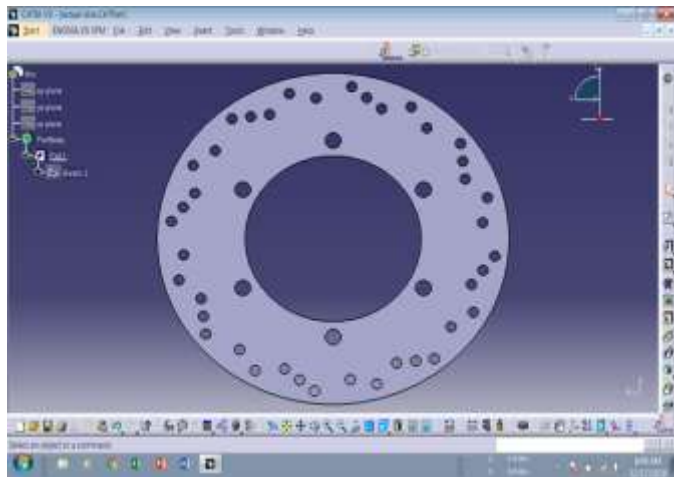


Fig -7: Actual Disc model

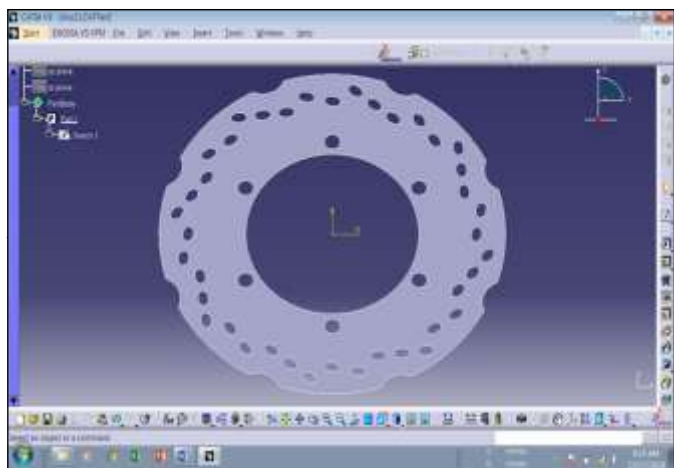


Fig -8: Disc model 1

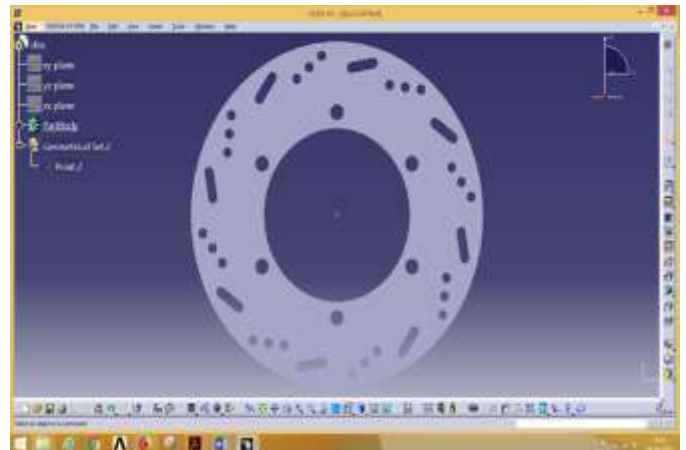


Fig -9: Disc model 2

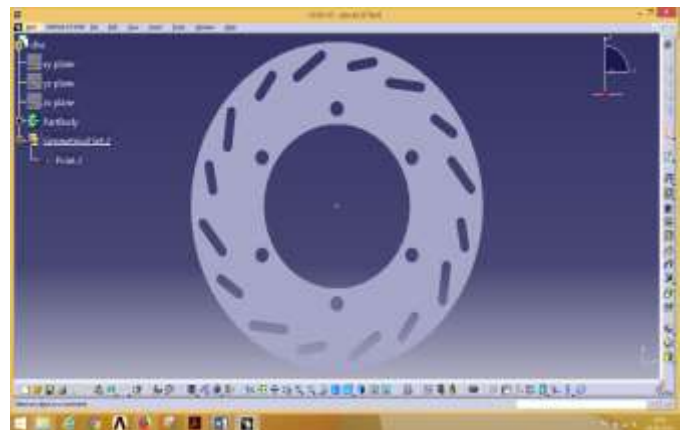


Fig -10: Disc model 3

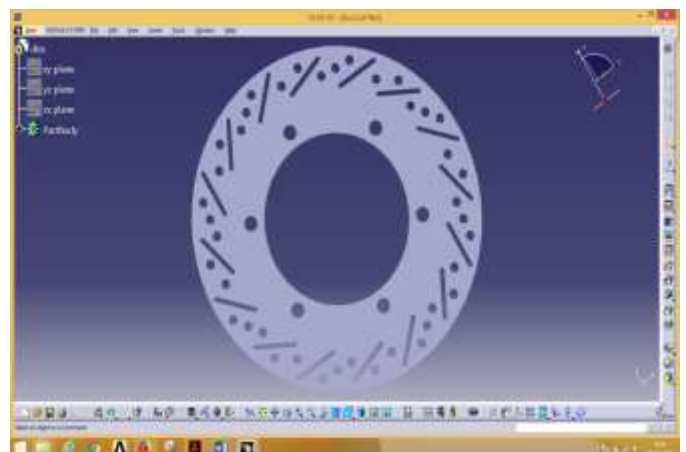


Fig -11: Disc model 4

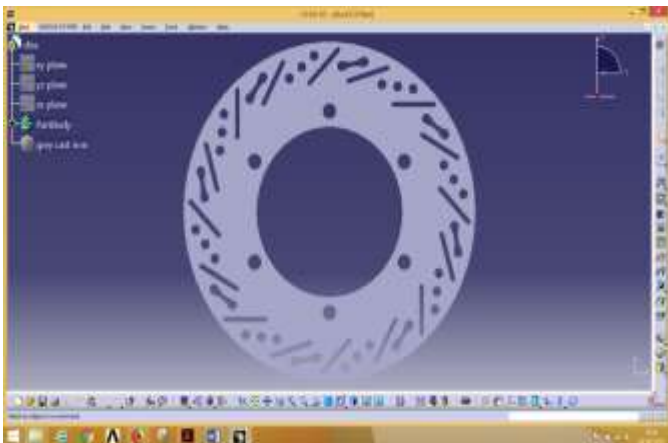


Fig -12: Disc model 5

5.2 Meshing of the disc

Meshing is important to simulate the boundary technique of problem. Mesh quality has great influence on the results. Mesh size decides the numerical convergence of solution. If the mesh size is poor then it will not give proper results. Fine mesh gives numerical convergence with accurate results.

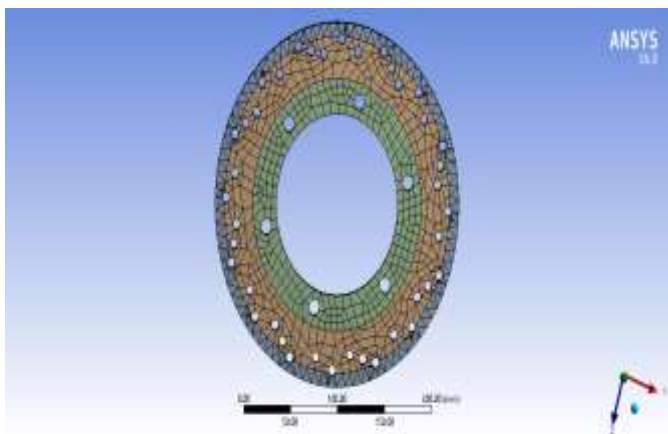


Fig -13: Meshing of the brake disc

5.3 Loading and boundary conditions

For thermal analysis, the temperature distribution depends upon the heat flux entering the disc through both sides of the disc and wall heat transfer coefficient. For analysis the initial and boundary conditions are introduced in the thermal module of ANSYS.

For thermal analysis of brake disc, due to friction between brake calliper and brake pad the heat is generated and due to this heat flux is applied on model of brake disc prepared in ANSYS. The value of heat flux calculated for the maximum speed is 97837 W/m², and the ambient temperature is 220C. These constraints are used for thermal analysis of brake disc model. Also, heat loss due to radiation takes places but the

value is very less as compared to heat flux value, thus it is neglected.

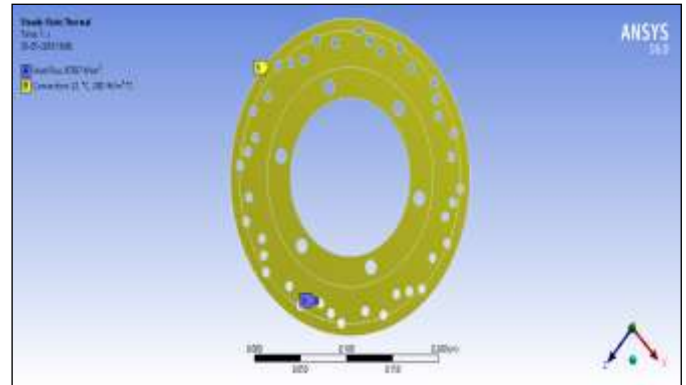


Fig -14: Boundary conditions for thermal analysis

For structural analysis, temperature and corresponding stress in disc brake vary under freeway driving conditions. For analysis the initial and boundary conditions are introduced in structural module of ANSYS. The calculated torque is 48.75Nm and the fixed support constraint is applied on disc model.

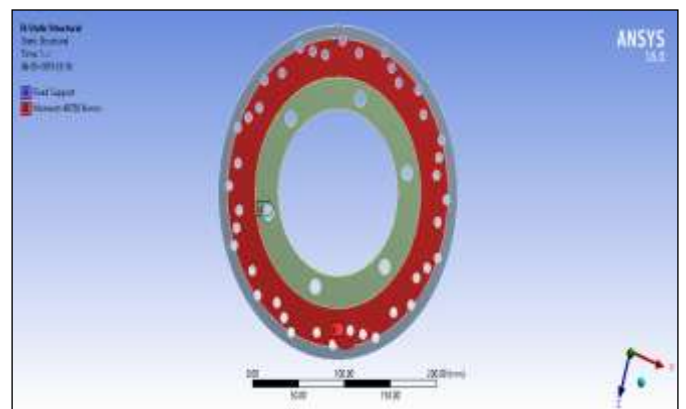


Fig -15: Boundary conditions for structural analysis

5.4 Thermal analysis

Understanding the thermal performance of automotive disc brake is the factor in developing a scaling methodology that replicates the real-world conditions. In this section thermal analysis of disc brake is presented, in order to understand the temperature distribution over the brake disc. The heat flux generated by the friction between the pad and disc is required.

The following assumptions apply:

- The kinetic energy of the vehicle is converted to thermal energy due to friction at the sliding surface without any energy loss during the braking event [4].

- The heat flux generated by friction at the interface between the pad and disc is transferred to the brake pads and the disc according to their respective thermal properties [5].
- All brake parts are in steady state condition before braking commences.

The steady-state thermal analysis is used to determine temperatures, thermal gradients, heat flow rates, and heat fluxes in an object that are caused by thermal loads that do not vary over time. A steady-state thermal analysis calculates the effects of steady thermal loads on a system or component. The temperature distribution results in thermal stresses that can cause the failure [2]. In such cases temperature from the transient thermal analysis for thermal stress is evaluated.

Steady-state thermal analysis is performed on all discs by applying temperature constraints. The heat flux of 97837 W/mm² is applied on circular cross-section of the disc in contact with the piston cylinder and the ambient temperature is taken as 295K.

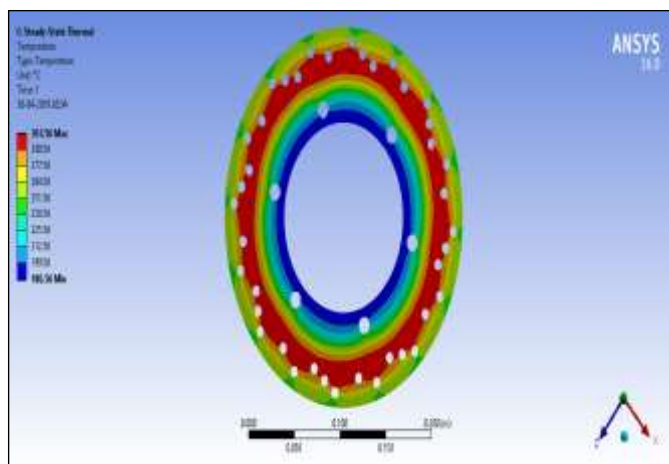


Fig -16: Temperature of actual brake disc model

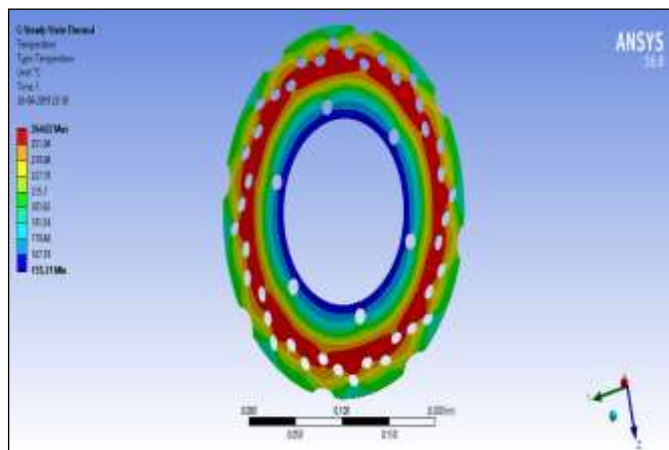


Fig -17: Temperature of the brake disc model 1

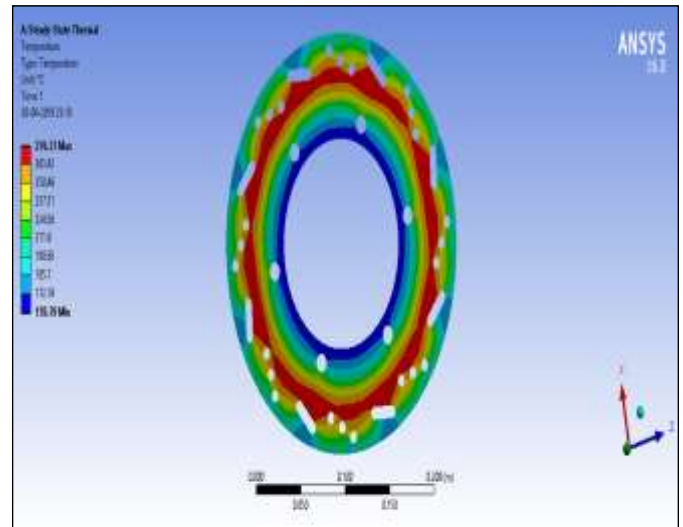


Fig -18: Temperature of the brake disc model 2

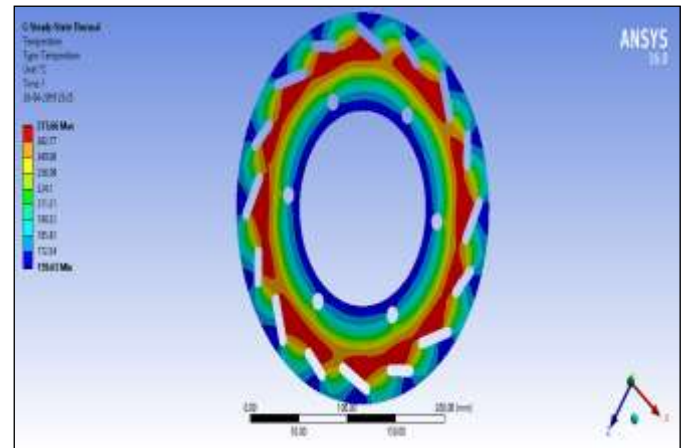


Fig -19: Temperature of the brake disc model 3

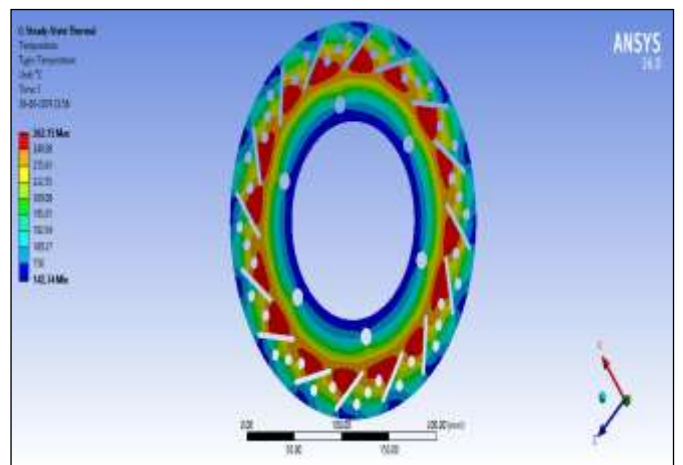


Fig -20: Temperature of the brake disc model 4

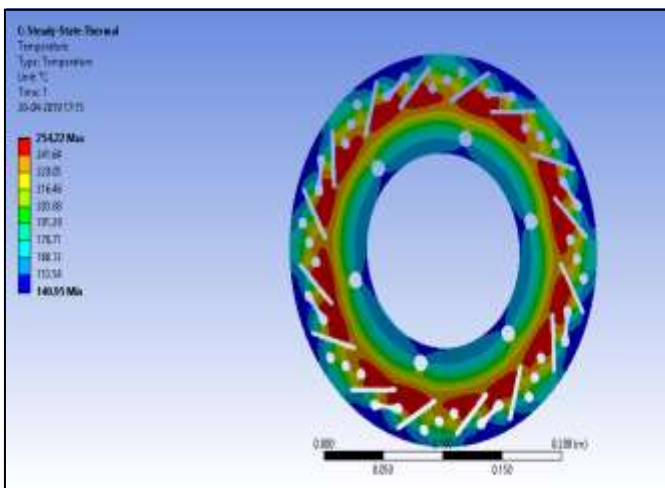


Fig -21: Temperature of the brake disc model 5

Thermal analysis is performed on different brake disc models and the temperature distribution is studied for different disc models. The actual brake disc of Bajaj Pulsar 220F has a smaller number of holes and the maximum temperature rise is observed in actual brake disc model and it was 303.56°C. In brake disc model 1 the holes were changed to elliptical shape and the cut-outs were given on the outer diameter of brake disc to increase the area available for heat dissipation thus the maximum temperature in brake disc model 2 was 264.02°C. In brake disc model 2 the combination of rounded rectangular holes and circular holes were designed. The maximum temperature observed in the brake disc model 2 was 276.37°C. In brake disc model 3 all holes were of the rounded rectangular shape and the maximum temperature observed in this disc is 275.66°C.

In brake disc model 4 linear slots were provided along with the circular shaped holes. The area available for heat dissipation in brake disc model 4 was more as compared to other disc models and the maximum temperature observed in the brake disc model 4 was 262.15°C. In brake disc model 5 dumbbell shaped holes were provided along with linear slots and circular shaped holes. The maximum temperature observed in this disc was 254.22°C which is lowest among all other models and hence it is the best brake disc model among all other models.

The maximum attained temperature in the other disc models is less than the actual disc due to better cooling and heat dissipating characteristics. Disc 1,2,3,4 and 5 have a greater number of drilled holes and circular holes which increase the contact surface area of the disc with the air, thus increasing the rate of heat dissipation through convection. Disc model 5 shows lowest maximum temperature among all other models.

5.5 Structural analysis

Static structure analysis is the most common application in FEM. Structural loads on brake disc, geometry of brake disc, support conditions and material properties of brake disc are determined to perform accurate structural analysis. Static analysis determines the displacement, stress, strain, force in structure or component caused by loads that do not induce inertia and damping effects. This project deals with the study of stress, deformation on different brake disc models under static condition. After completion of finite element model, it must constrain and load must be applied to the model. Torque of 48.75Nm is applied on the swept area of brake disc and the bolt holes are fixed.

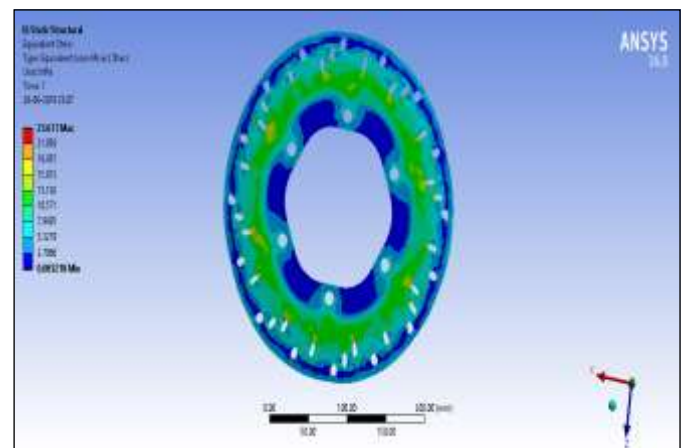


Fig -22: Stress distribution of the actual brake disc model

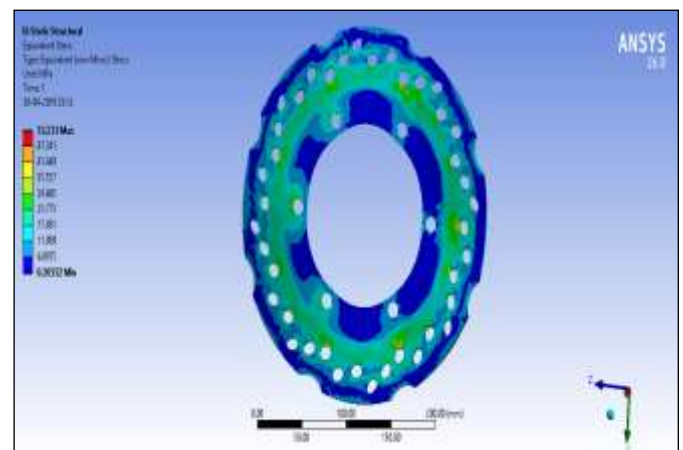


Fig -23: Stress distribution of the brake disc model 1

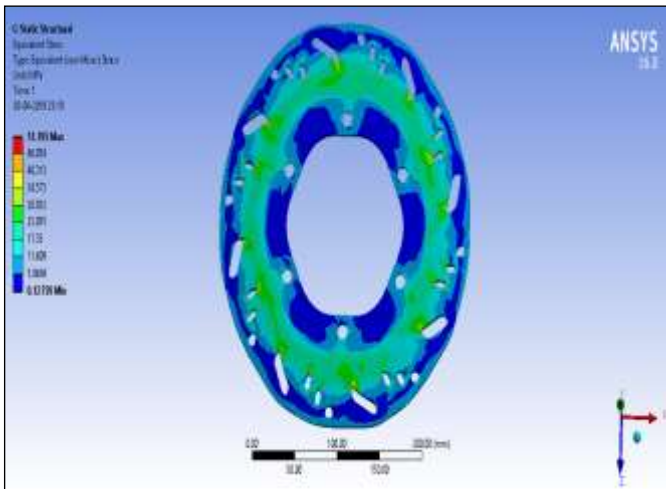


Fig -24: Stress distribution of the brake disc model 2

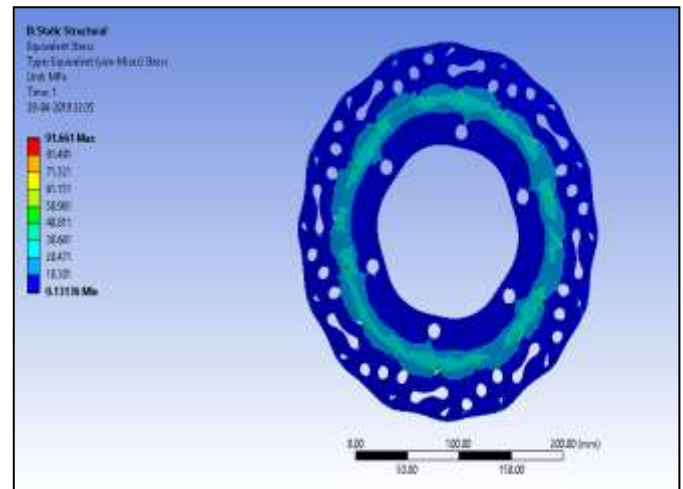


Fig -27: Stress distribution of the brake disc model 5

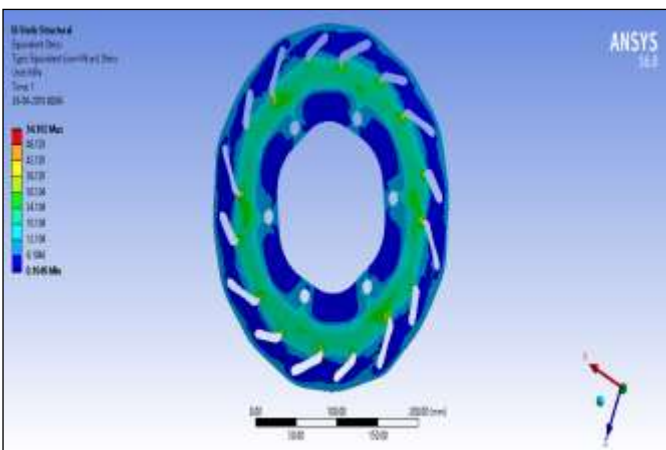


Fig -25: Stress distribution of the brake disc model 3

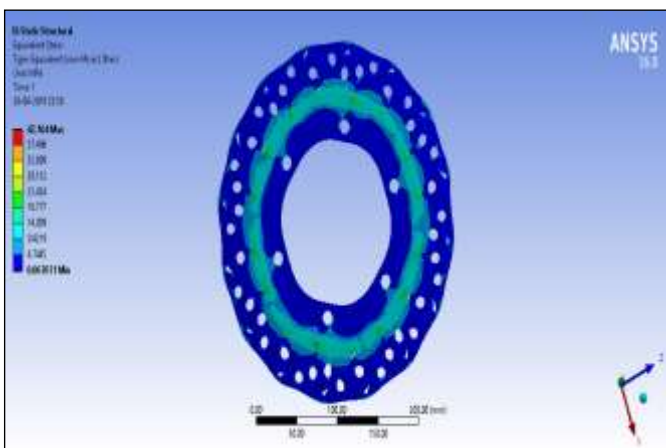


Fig -26: Stress distribution of the brake disc model 4

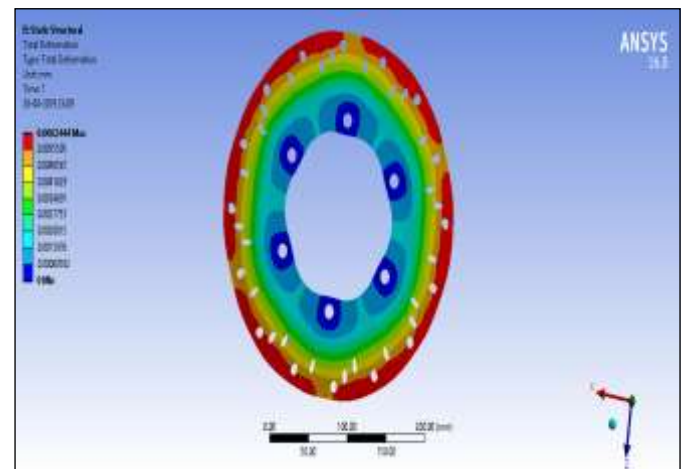


Fig -28: Total deformation of the actual brake disc model

The actual brake disc of Bajaj pulsar 220F has less number of holes. The maximum stress observed in actual brake disc is 23.67 MPa which is least among all other disc models. As the shape of holes is changed and a greater number of holes are provided area of the brake disc decreases and the maximum stress increases. Maximum stress observed in brake disc model 1 is 53.23MPa.

Maximum stress observed in brake disc model 2 is 57.79 MPa. In Brake disc model 3 the maximum stress observed is 54.10 MPa. Maximum stress observed in brake disc model 4 is 42.16 MPa. Brake disc model 5 has a greater number of holes and more pattern of holes. Disc 5 shows highest value of stress of 91.66 MPa among other models. As the maximum allowable stress for stainless steel (SS410) is 510 MPa. The modified brake disc model 5 is safe.

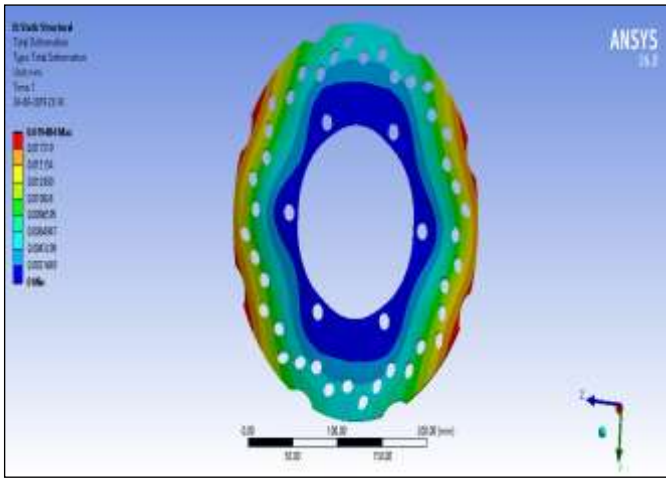


Fig -29: Total deformation of the brake disc model 1

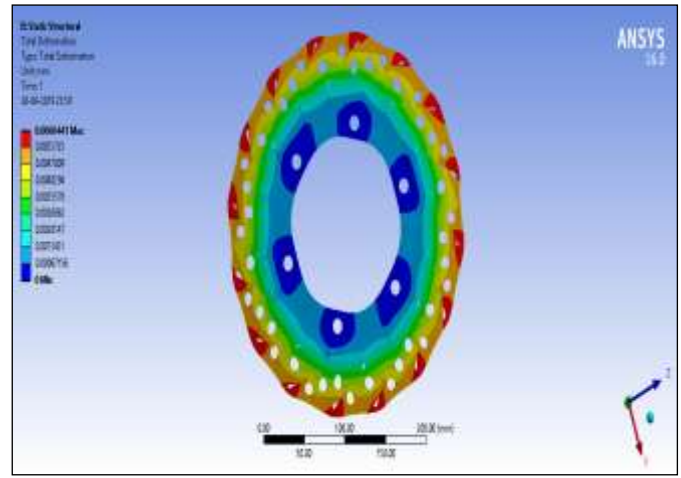


Fig -32: Total deformation of the brake disc model 4

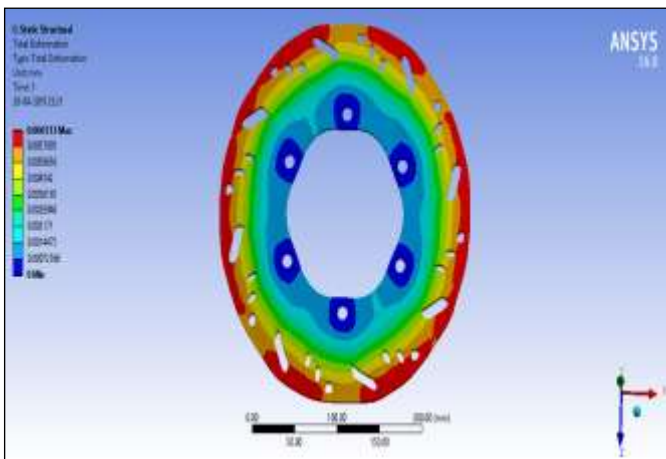


Fig -30: Total deformation of the brake disc model 2

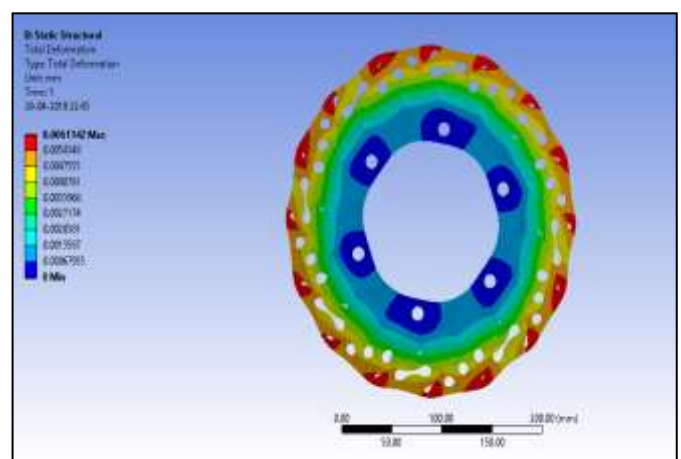


Fig -33: Total deformation of the brake disc model 5

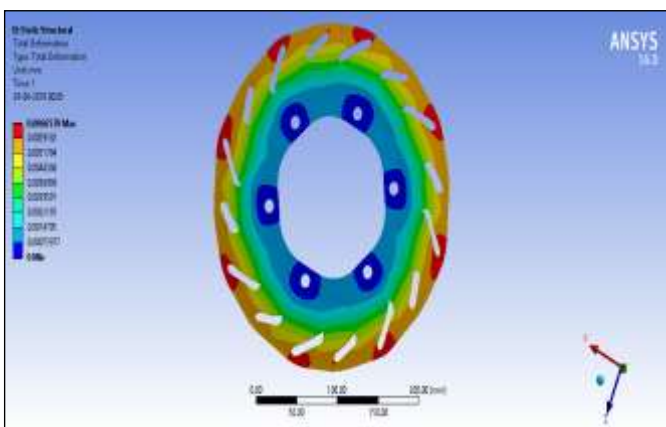


Fig -31: Total deformation of the brake disc model 3

Deformation refers to any changes in the shape or size of an object due to:

- An applied force (the deformation energy in this case is transferred through work) or
- A change in temperature (the deformation energy in this case is transferred through heat).

As deformation occurs, internal inter-molecular forces arise that oppose the applied force. If the applied force is not too great, these forces may be sufficient to completely resist the applied force and allow the object to assume a new equilibrium state and to return to its original state when the load is removed. A larger applied force may lead to a permanent deformation of the object or even to its structural failure.

The maximum deformation in actual brake disc is 0.00624mm. The loss in rigidity of rotor accounts for increased deformation is due to material removal. The maximum deformation observed in brake disc model 1, model 2, model 3, model 4 and model 5 are 0.01948mm, 0.00651mm, 0.00665mm, 0.00604mm and 0.00611mm respectively.

Table -1: Comparison of parameters between different models				
Model	Temperature (°C)	Max Stress MPa	Max deformation (mm)	Weight (Kg)
Actual Disc	303.56	23.67	0.00624	1.459
Disc 1	264.02	53.23	0.01948	1.386
Disc 2	276.37	57.79	0.00651	1.425
Disc 3	275.66	54.10	0.00665	1.385
Disc 4	262.15	42.16	0.00604	1.361
Disc 5	254.22	91.66	0.00611	1.343

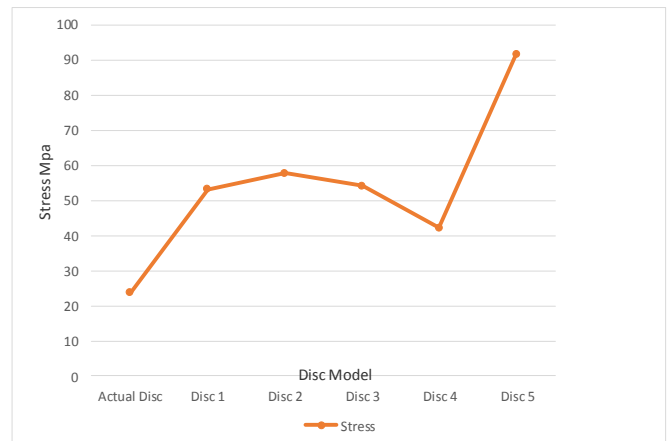


Chart -2: Stress variation in different disc models

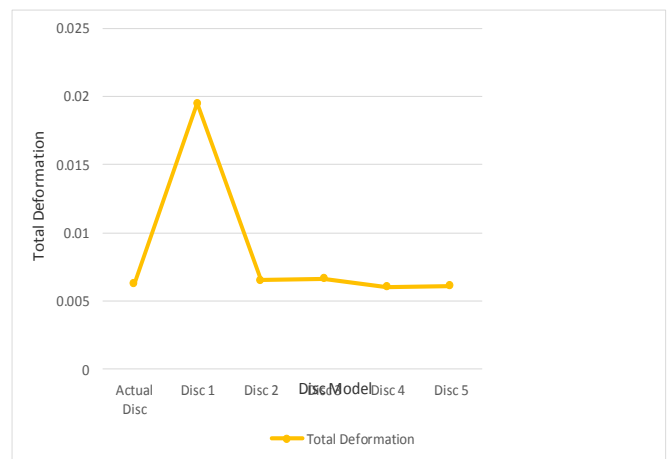


Chart -3: Total deformation variation in different disc models

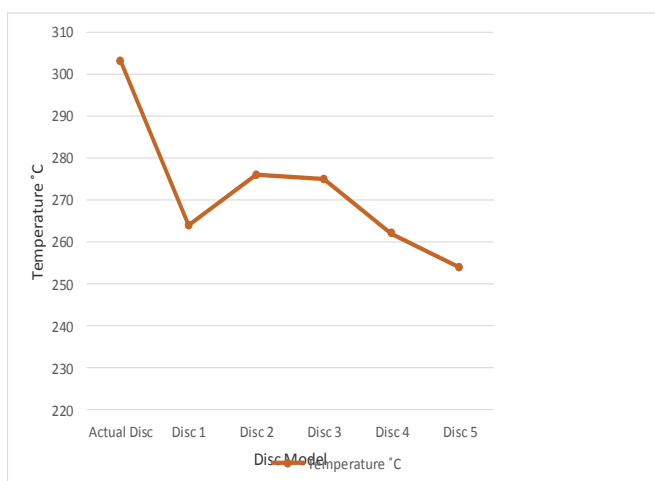


Chart -1: Temperature variation in different disc models

5.6 Optimization and Selection of final model

The variability in disc geometry plays very significant role in cooling of the disc in braking performance. The maximum temperature observed and stresses developed in the different types of disc brake models are compared. Actual disc shows highest maximum temperature of 303.56°C and weight of 1.459 Kg. Among all modified models disc model 5 has lowest maximum temperature of 254.22°C and is lightest with weight of 1.343 Kg. The maximum stress and deformation of disc model 5 is within the permissible limits.

6. RESULT

The geometric design of brake disc is an essential factor in deciding its thermo-mechanical characteristics. Discs with geometric patterns can be used where faster cooling and lightness in weight is preferable, as desired in the modern automobiles. Variation in geometric shape and pattern of holes affects the temperature distribution over the area of brake disc. As the convection area increases, heat dissipation increases. Increased heat dissipation results in reduction of maximum temperature attained in braking.

From analytical calculations it is found out that the heat flux and braking torque acting on the brake disc is 97837 W/m² and 48.75 Nm. The thermal and structural analysis is done on different disc models and the results are compared to select the best model. Brake disc model 5 is the best model.

7. CONCLUSION

- The variability in geometric design plays important role in cooling of the braking phase. Actual disc rotor has maximum weight i.e. 2.2725 Kg and has maximum temperature rise than all other models.
- Disc rotor model 2,3,4 and 5 possess less maximum temperature rise than actual rotor model due to increase in surface area of the disc exposed to the atmosphere.
- Maximum temperature in brake disc 5 is 254°C which is lowest among all other disc models which proves that the heat dissipation of the modified disc 5 is best among all other models. Disc rotor model 5 is lightest rotor i.e. 8 % lighter than actual rotor and shows minimum rise in temperature i.e. 16.17 % less than actual rotor model.

From structural analysis result of different brake disc models, the maximum stress observed is 91.66MPa for brake disc 5 which is less than ultimate tensile strength of stainless steel which is 510MPa. Hence, the brake disc 5 is structurally safe.

8. ACKNOWLEDGEMENT

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