# Structure Optimization of Disc Brake

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**Abstract**

Transient Thermal and Structural Analysis of the Rotor of Disc Brake is aimed at evaluating the performance of disc brake rotor of a bike under severe braking conditions and there by assist in disc rotor design and analysis. An investigation into usage of new materials is required which improve braking efficiency and provide greater stability to vehicle. This investigation can be done using ANSYS software along with the modelling software CATIA V5.

In this project, “Structure Optimisation of the disc brake”, the action force, friction force and brake torque on rotor disc are studied by the basic formulae of disc brake. The aim is to compare between the rotor disc of a standard motorcycle “Bajaj Pulsar” and a non-standard rotor disc to find out the relationship value between brake torque, rotor disc dimension etc.

**Keywords:** Disk Brake, Rotor, Thermal Analysis, Structural Analysis, Transient Analysis.

## 1.Introduction

Disc-style brakes development and use began in England in the 1890s. The first calliper-type automobile disc brake was patented by Frederick William Lanchester in his Birmingham, UK factory in 1902 and used successfully on Lanchester cars. Compared to drum brakes, disc brakes offer better stopping performance, because the disc is more readily cooled.

When hydraulic pressure is applied to the caliper piston, it forces the inside pad to contact the disc. As the pressure increases, the caliper moves to the right and cause the outside pad to contact the disc. Braking force is generated by friction between the disc pads as they are squeezed against the disc rotor. Since disc brakes do not use friction between the lining and rotor to increase braking power as drum brakes do, they are less likely to cause a pull. The friction surface is constantly exposed to the air, ensuring good heat dissipation, minimizing brake fade. It also allows for self-cleaning as dust and water is thrown off, reducing friction difference.

## 2.Specifications Of Standard Disc Brake Rotor:

In this project study standard of two wheeler name “Bajaj” model Pulsar Factor;

| | |
|---|---|
| Rotor disc dimension | = 240 mm, |
| (Rotor disc material | = Gray cast iron), |
| Rotor thickness | = 10 mm, |
| Pad brake area | = 2000 $mm^2$ |
| Pad brake material | = Asbestos, |
| Coefficient of friction (Wet) | = 0.08-0.12, |
| Coefficient of friction (Dry) | = 0.2-0.5, |
| Maximum temperature | = 250 ºC, |
| Maximum pressure | = 1 MPa |

## *3.Details of Models:*

1) Original disc brake has been 6 holes dia 8 mm arranged equally. There are 36 holes Surrounding disc Dia 8 mm arranged equally. (Refer Fig.1). 2) New Disc 1: Original disc brake has been reduces 6 holes dia 6 mm. There are 36 holes Surround Dia 8 mm arranged equally. Original disc brake has been added with 18 cut-sections & changes central structure. (Refer Fig.2). 3) New Disc 2: Original disc brake has been 6 holes dia 8 mm arranged equally same. Original disc brake has been added with cut section inlet & outlet airflow is large & small respectively. The thickness has been 5mm. (Refer Fig.3). 4) New Disc 3: 15 Vanes have been arranged. 15 Elliptical shapes arranged between Vanes clockwise Inlet of air flow & outlet of air flow between the vanes is same. (Refer Fig.4). 5) New Disc 4: There are 6 holes having dia 6 mm and 18 cut-sections which changes the central structure. (Refer Fig.5). 6) New Disc 5: 60 surrounding holes having dia 7 mm arranged equally. and 6 supporting holes of 8 mm dia. Original disc brake has been added with different cut-sections & which changes central structure. (Refer Fig.6).

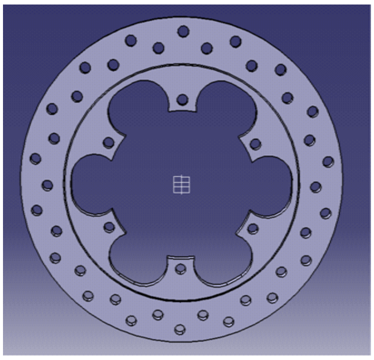

Fig.1 Baseline Original disc

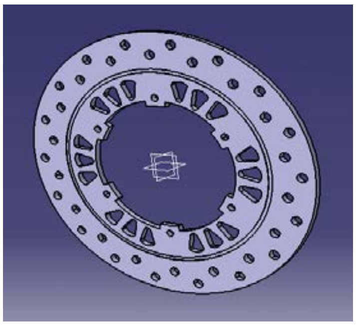

Fig.2 New Disc 1

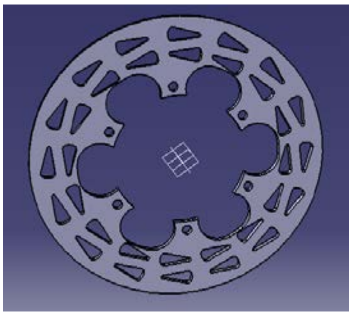

Fig.3 New Disc 2

Fig.4 New Disc 3

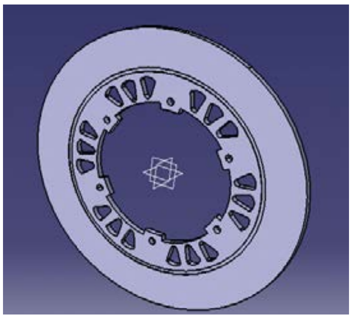

Fig.5 New Disc 4

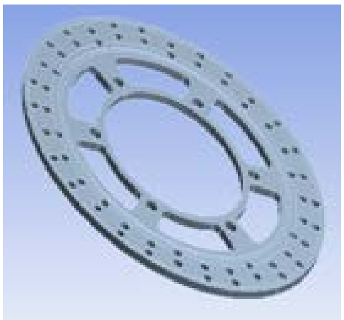

Fig.6 New Disc 5

### 3.1Boundary conditions and loading for structural analysis:

In this project study about stress, deformation on rotor disc under condition of static equation. After completion of the finite element model it has to constrain and load has to be applied to the model. User can define constraints and loads in various ways.

This helps the user to keep track of load cases.

- Force along x-axis
- Fixed inside surface of hole.
- Total tangential forces 1000 N.
- Material properties reference grey Cast Iron.

### 3.2Boundary Condition for thermal analysis

In this project study about thermal analysis is carried out on disc brake under condition of static equation.

Maximum outer Temperature = 250°C,
Maximum inner Temperature = 40°C

## 4.Results And Discussion

### *4.1Result for structural analysis:*

Table 1:- Transient thermal analysis of original disc

| Disc Type | Stress ($e^7$) $N/mm^2$ | Deformation ($e^{-6}$) mm | Weight (kg) |
|---|---|---|---|

| | | | |
|---|---|---|---|
| Original | 2.309 | 3.99 | 0.98541 |
| Disc 1 | 1.967 | 3.829 | 0.98788 |
| Disc 2 | 2.991 | 5.680 | 0.98902 |
| Disc 3 | 2.745 | 5.342 | 0.98511 |
| Disc 4 | 1.800 | 3.519 | 1.03255 |
| Disc 5 | 2.565 | 5.850 | 0.8970 |

### 4.2Steady State Thermal Analysis-Temperature Distribution

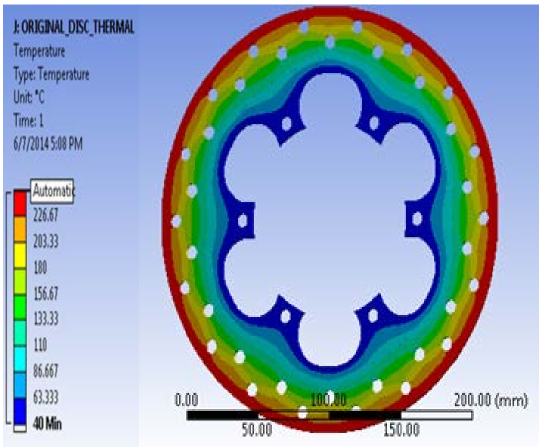


Fig.7 Thermal Analysis of Original disc

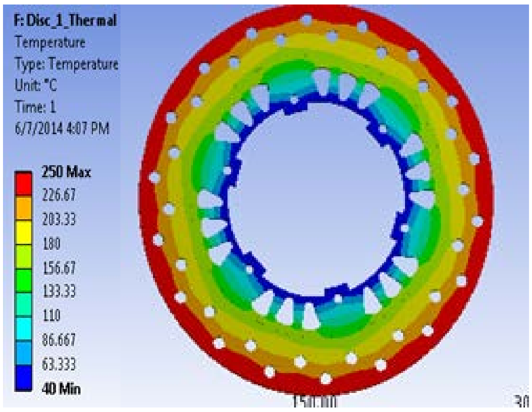


Fig.8 Thermal Analysis of New Disc1

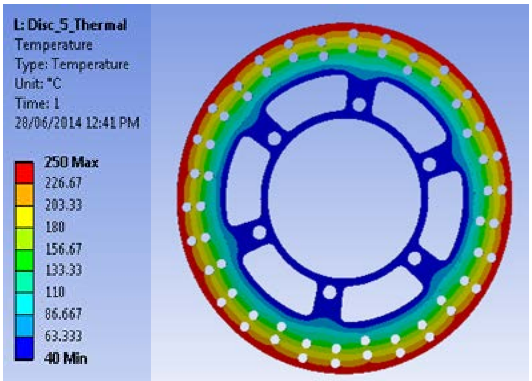


Fig.9 Thermal Analysis of New Disc 5

### 4.3Transient Thermal Analysis Result

| Sr. No | Time in min | Temp (max) [°C] ANSYS result Original Disc | Temp (max) [°C] ANSYS result Selected Disc | Temp (max) [°C] Experimental Result Selected Disc |
|---|---|---|---|---|
| 1 | 10 | 24.908 | 24.679 | 24.02 |
| 2 | 20 | 31.671 | 31.165 | 29.33 |
| 3 | 30 | 38.429 | 37.637 | 34.91 |
| 4 | 40 | 45.186 | 44.108 | 44.54 |
| 5 | 50 | 51.944 | 50.578 | 52.82 |
| 6 | 60 | 58.702 | 57.049 | 57.56 |

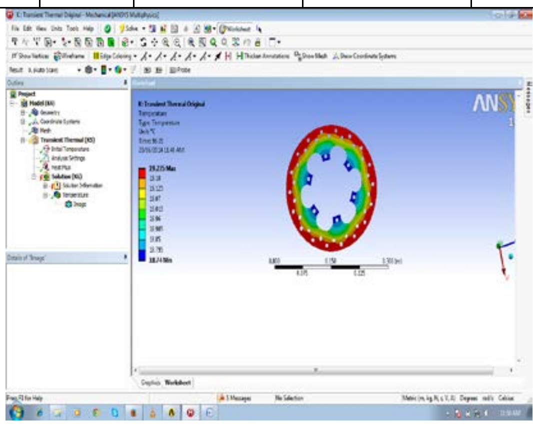

Fig.10 Transient Thermal Analysis Result -Original Disc

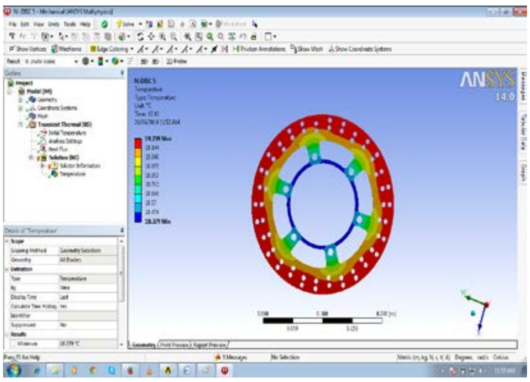

Fig.11 Transient Thermal Analysis Result - Selected disc

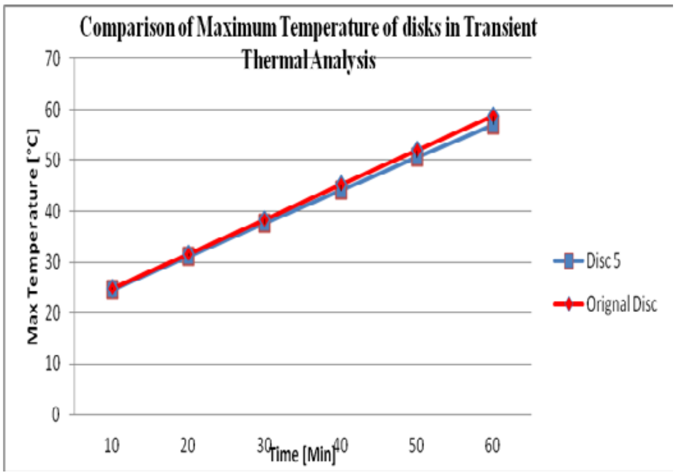


Fig 12:- Comparison of Maximum Temperature of discs in Transient

Fig.13 Experimental Setup

Table 2: Maximum temperature of disc

Table 3:- Weight comparison of the disc

| Result of | Mass (kg) | Remark |
|---|---|---|
| Original | 0.98541 | Good |
| Disc 5 | 0.8970 | Very Good |

Table 4: - Weight of selected disc

| | |
|---|---|
| Nonlinear Effects | Yes |
| Thermal Strain Effects | Yes |
| Bounding Box | |
| Properties | |
| Volume | 1.3316e+005 mm³ |
| Mass | 0.98541 kg |
| Centroid X | 1.82e-003 mm |
| Centroid Y | 2.5 mm |
| Centroid Z | 8.8514e-003 mm |
| Moment of Inertia Ip1 | 4735.2 kg·mm² |
| Moment of Inertia Ip2 | 9465.7 kg·mm² |
| Moment of Inertia Ip3 | 4734.6 kg·mm² |

Table 5: - Weight of selected disc

| | |
|---|---|
| Material | |
| Assignment | CAst Iron |
| Nonlinear Effects | Yes |
| Thermal Strain Effects | Yes |
| Bounding Box | |
| Properties | |
| Volume | 1.2122e+005 mm³ |
| Mass | 0.897 kg |
| Centroid X | 2.5 mm |
| Centroid Y | 1.2373e-002 mm |
| Centroid Z | -1.8813e-005 mm |
| Moment of Inertia Ip1 | 8656.9 kg·mm² |
| Moment of Inertia Ip2 | 4329.7 kg·mm² |

## 5.Conclusion

The present study shows that it is a useful design which improves the overall brake performance of disc brake system. It is found that the results given by the selected disc are desirable.

a) Weight of the selected disc is less as compared to original disc, this saves the material.

b) Heat generated in selected disc is less as compared to original disc but it is not showing significant effect on variation in temperature distribution.

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