International Journal of Engineering Applied Sciences and Technology, 2020
Vol. 4, Issue 9, ISSN No. 2455-2143, Pages 286-293
Published Online January 2020 in IJEAST (http://www.ijeast.com)

DESIGN AND ANALYSIS OF DISC BRAKE ROTOR USING DIFFERENT PROFILES

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Abstract - **The main objective of this project is to propose a new automotive brake disc rotor design for BAJAJ PULSAR 150 which will reduce the total deformation and increase the maximum heat dissipation. Here various shapes of ventilated holes in brake disc rotor is designed using AUTODESK INVENTOR 2019. The inner and outer boundaries are preserved so that the changes are made only in the intermediate patterns between the boundaries, thus the models have same structural boundary limits. The static structural analysis and steady state thermal analysis of brake disc rotor is done using ANSYS 19, which is a dedicated finite element package used for determining the temperature distribution, variation of the stresses and deformation across the disc brake profile. The assembly analysis method, is carried out for static structural analysis to increase the accuracy of result. The best of the designed brake disc rotor is to be suggested based on the magnitude of von - misses stresses, deformation, temperature, total heat flux, and weight.**

Keywords – **brake disc rotor, intermediate patterns, static structural analysis, steady state thermal analysis.**

I. INTRODUCTION

The disc brake is a device which specially use for slowing or stopping the rotation of a wheel. Repetitive braking of the vehicle going generates heat during each braking movements. The finite element method is a powerful method which are used for the numerical solutions of a wide range of engineering problems.

The disc brake is a wheel brake which slows rotation of the wheel by the friction which caused by pushing brake pads towards a brake disc with a set of calipers. Brakes convert motion to heat, and if the brakes get too hot, they become less effective, this phenomenon known as brake fade, Disc brake consisting structural steel disc bolted to the wheel hub and a stationary housing which is known as caliper.

The caliper is connected to some stationery part of the vehicle like the stub axle as is cast in two parts each part consist of a piston. In between each piston and the disc there is a frictional pad hold in position by detainments pins, spring plates. The passages are so connected to another one for bleeding. Each cylinder contains rubber-sealing ring which placed between the cylinder and piston.

The aim is to compare the structural and thermal properties of rotor disc during braking, of standard motorcycle “BAJAJ PULSAR 150” with a non-standard rotor disc and to find out the difference in structural and thermal properties.

II. METHODOLOGY

A. Design of existing model

The existing model of brake disc rotor is designed based on analytical measurements and standard values available from the standard website. The standard brake disc rotor that we consider here is from BAJAJ PULSAR 150. The brake disc rotor of BAJAJ PULSAR 150 is designed using AUTODESK INVENTOR 2019.

B. Evaluation of existing model

The designed existing model of disc rotor is analyzed structurally and thermally to study its properties using ANSYS 19.

Material used

The knowledge of materials and their properties is of great significance for a design engineer. The machine elements should be made of such a material which has properties suitable for the conditions of operation, in addition to this a design engineer must be familiar with the effects which the manufacturing processes and heat treatment have in the properties of the materials. Hence, from standard reference grey cast iron and aluminium alloys are used as brake disc materials, based on reference from vehicle reviewers and dealers site grey cast iron is chosen as the standard brake disc rotor material.

Properties of grey cast iron

1. Chemical Properties of grey cast iron According to the different grades, the grey cast iron has different chemical components, however, normally, all gray iron grades have the following chemical component range:

Carbon (C) 2.8 - 3.9%, Silicon (Si) 1.1 - 2.6%, Manganese (Mn) 0.5 - 1.2%, $P \leq 0.3\%$, $S \leq 0.15\%$. However, you cannot inspect the quality by chemical components. The iron foundries have right to adjust them as long as they can meet the mechanical properties.

2. Mechanical Properties of grey cast iron According to the different grades, grey cast iron has different mechanical properties. Their tensile strength is between 72500 psi to 188500 psi. The yield strength is between 21700 psi to 72500 psi. Grey iron castings almost have no elongation. The impact toughness is less than 11 J/cm2, so if your parts need to stand impact, then ductile iron will be the better choice. The hardness of gray iron is between 145 to 280 HBS. If customers have special requirements to the hardness, you should tell the casting suppliers, otherwise, hardness cannot be an inspection standard.

Mechanical Properties of grey cast iron:

Mechanical Property	Value	Unit
Young's Modulus	118	GPa
Tensile Strength	310	MPa
Yield Strength	575	MPa
Poisson's ratio	0.30	-

Tab. (1)

Structural analysis

In structural analysis the deformation in steady state boundary conditions is analyzed. The brake torque, clamping force, brake pad area is given as an input boundary condition. The resulting output like total deformation, directional deformation, von-misses stress, max principal stress, etc., have been analyzed.

Thermal analysis

In steady state thermal analysis, the properties like temperature distribution, amount of heat transferred is analyzed from the boundary condition like, surface temperature during braking, convection occurring in brake disc, etc.

C. Alternative models

To improve the efficiency of brake disc rotor the alternative design models are proposed preserving the inner and outer boundary. Thus the changes are made on the intermediate faces between the inner and outer boundary and in some cases on exterior surface edges too.

Design of alternate models

The alternate models are made using holes, cuts, slots, etc., and making the patterns on different sets on different radii with increased and decreased hole radii the alternative models are made. The alternate models are designed using AUTODESK INVENTOR 2019.

Analysis of alternate models

The alternate models are individually analyzed for the calculated structural boundary conditions and the same as for thermal boundary condition. The alternate models are analysed both structurally and thermally using ANSYS 19.

Result comparison

The results are tabulated based on the models and their structural output results followed by the thermal output results.

The Structural analysis output results to be tabulated are total deformation, von misses stress, etc, The Thermal analysis output results to be tabulated are temperature, total heat flux, etc,

The best model is selected from the results which is better from all background values.

III. CALCULATIONS

A. Initial conditions and assumptions

To make the calculation perfect the initial condition are taken from dealer site and the relevant assumptions are made as follows

- The total weight of the vehicle is assumed to be 300 Kg.
- The vehicle is assumed to travel at a maximum speed of 100kmph, i.e.,v= 27.77m/s
- The axial weight distribution is taken as 0.5
- The coefficient of friction is assumed to be 0.5
- The effective radius is taken as, R eff = 0.12 m

- The Kinetic energy to absorbed is taken as 0.9
- The standard hydraulic pressure is taken as 1 Mpa
- The coefficient of friction is same for brake pad and rotor, i.e., μI = μO.
- The ambient temperature is taken as 23 °c
- The vehicle is said to stop using 1 brake caliper, i.e., the stopping distance is taken as 50 m
- The brake pad's total coverage angle is measured to be 42.5°
- The vehicle has varying leverage and actuation based on driving condition so, a FOS of 2.5 is taken into consideration for single stop surface temperature rise.
- The tangential clamping force between the brake pad and rotor on inside is equal to outside, i.e., FTRI = FTRO, FRI = FRO.

B. Calculations

Structural calculations

Brk pad cont area, A = (π(r1)^2 - π(r2)^2) * θ/360
= (π(120)^2 - π(95)^2) * 42.5/360
= 1993.49 ≈ 2000 mm^2 = 0.002 m^2

Norm force on inn side, FRI = ((P max/ 2) x A)
= ((1 * 10^6)/2 *0.002)
= ((1000000)/2 * 0.002)
= (500000 * 0.002)
= 1000 N

Tang react force on inn side, FTRI = μI x FRI
= (0.5 * 1000)
= 500N

Tang react force on outside, FTRO = μO x FRO
= (0.5 * 1000)
= 500N

Tang clamping force, FT = FTRI + FTRO
= 500 + 500
= 1000 N

Brake torque, TB = FT * Reff
= 1000* 0.12
= 120 N-m

Thermal calculations

Braking time, d = (u + v) /2 * t
50 = (0+27.77)/2 * t
13.885 t = 50
t = 3.6 s

kinetic energy, K.E = γ k * (m (u-v)^2) / 2
= (0.5) (0.9) * ((300)(0-27.77)^2) /2
= (0.5) (0.9)*(115675.5)
= 52053.98 J

Braking power, Pb = K.E / t
= 52053.98 / 3.6
= 14459.44 W

Max. contact area, A1 = π(r1)^2
= π * (120)^2
= 45238.93 mm^2

Min. contact area, A2 = π(r2)^2
= π * (95)^2
= 28352.87 mm^2

Net disc contact area, A = A1- A2
= 45238.93 - 28352.87
= 16886.06 mm^2
= 0.01688 m^2

Heat flux, q = Pb/A
= 14459.44 / 0.01688
= 856601.89 W/m^2

Max temperature, $\mathrm{Tmax} = \frac{0.527 * q * \sqrt{t}}{\sqrt{(\rho * c * k)}} + \mathrm{Tamb}$

$= \frac{0.527*856601.89*\sqrt{3.6}}{\sqrt{(6600*460*50)}} + 296$

$= \frac{0.527*856601.89*1.897}{\sqrt{151800000}} + 296$

$= \frac{856361.18}{12320.71} + 296$

= 69.505 + 296 = 365.505 K
= 92.505 ° c ≈ 93 ° c

Considering, FOS = 2.5
= 2.5 * 93 = 232.5 ≈ 250 °c
T max = 250°c

IV. DESIGN OF DISC BRAKE ROTORS

A. Standard brake disc

The standard brake disc is designed containing 8 mm holes on circular patterns along the radii.

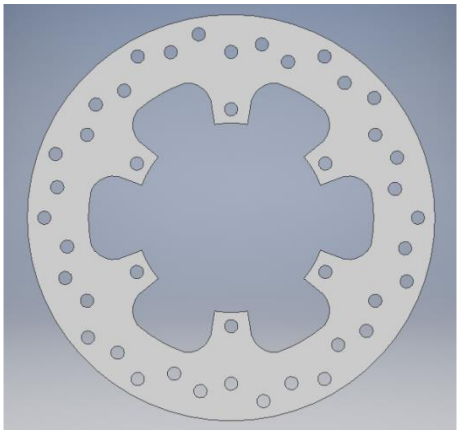


Fig (a) standard brake disc 1

B. Alternate brake disc

The alternate brake disc is designed containing 8 mm holes angularly on circular patterns.

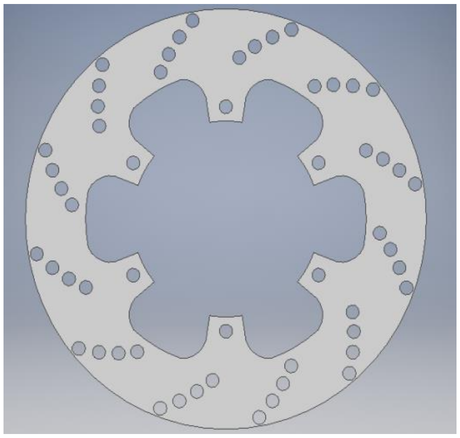


Fig (b) alternate brake disc 2

C. Alternate brake disc 3

The alternate brake disc is designed containing 8 mm holes angularly on circular patterns, with circular notch on the circumference of disc.

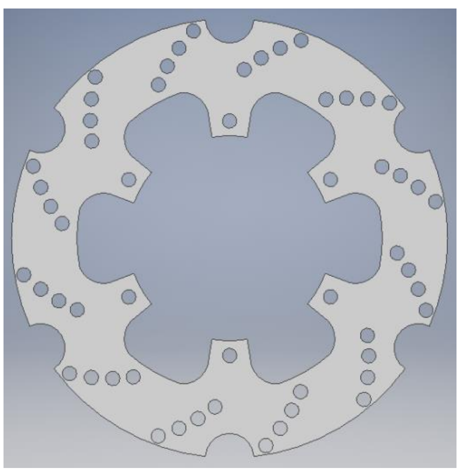


Fig (c) alternate brake disc 3

D. Alternate brake disc 4

The alternate brake disc is designed containing 8 mm holes on circular patterns along the radii.

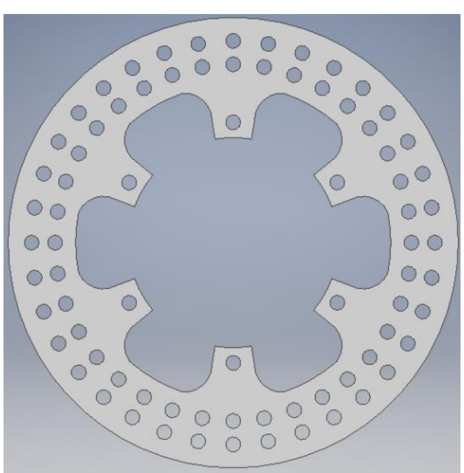


Fig (d) alternate brake disc 4

E. Alternate brake disc 5

The alternate brake disc is designed containing 8 mm holes angularly on circular patterns, with V notch on the circumference of disc.

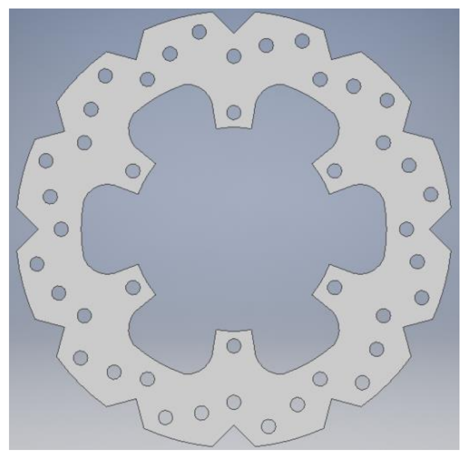
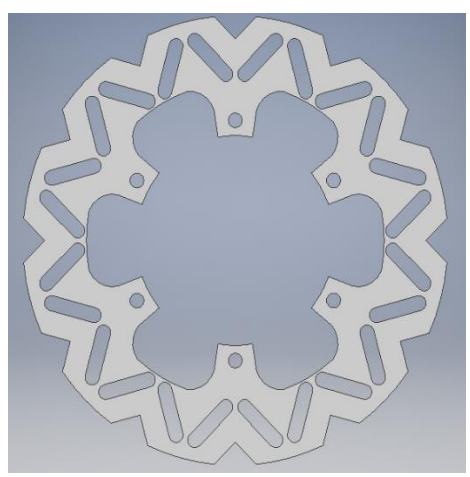


Fig (e) alternate brake disc 5

F. Alternate brake disc 6

The alternate brake disc is designed containing slots angularly on circular patterns, with V notch on the circumference of disc.



V. ANALYSIS OF DISC BRAKE ROTORS

Structural analysis

Boundary conditions

- 6 inner bolt holes - FIXED
- brake torque - 120 N-m
- tangential force - 1000 N (applied using brake pads)
- material - grey cast iron
- brake pad displacement - 0,0,free
- mesh - preferred size

Brake pads

The brake pads of area 0.002m^2 is designed on both sides of the disc using ANSYS 19 DESIGN MODELER to apply the tangential force.

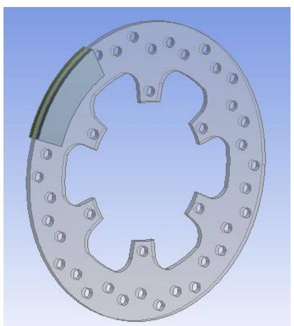


Fig. (1)
brake pad

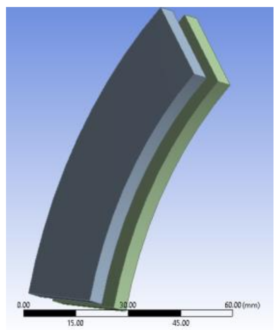


Fig. (2)
brake pad

Results

A. Standard disc 1

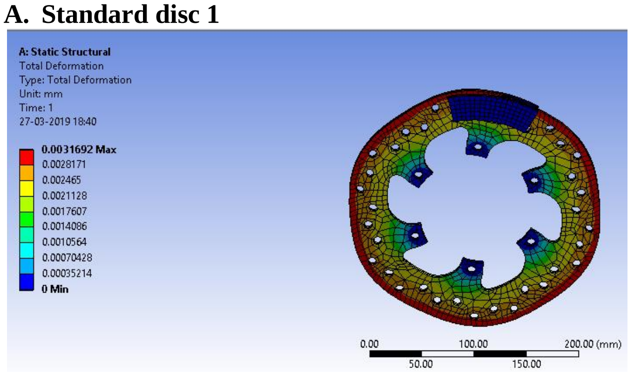


Fig. (a). Total deformation of standard disc 1

B. Alternate disc 2

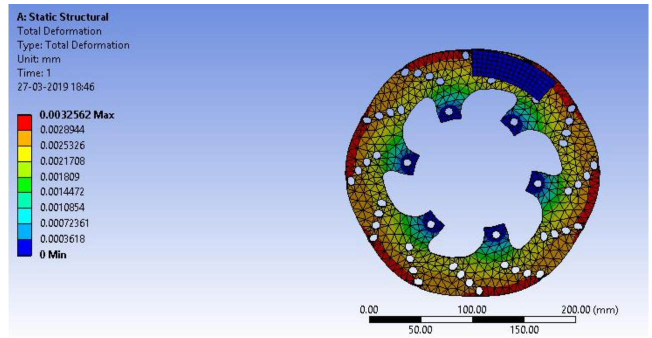


Fig. (b). Total deformation of alternate disc 2

C. Alternate disc 3

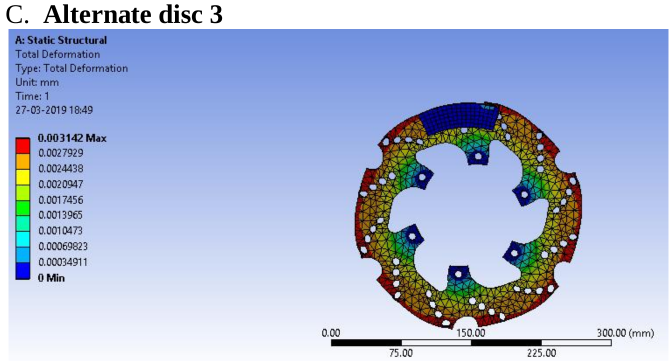


Fig. (c). Total deformation of alternate disc 3

D. Alternate disc 4

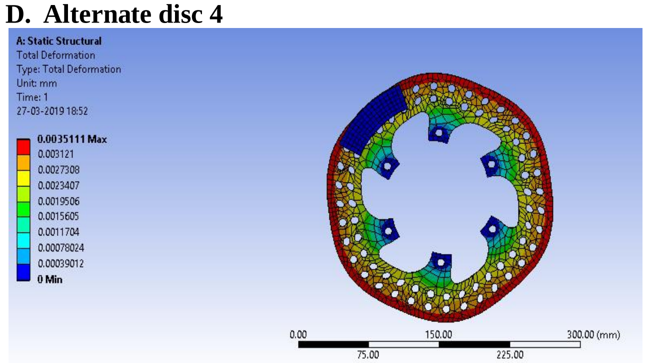


Fig. (d). Total deformation of alternate disc 4

E. Alternate disc 5

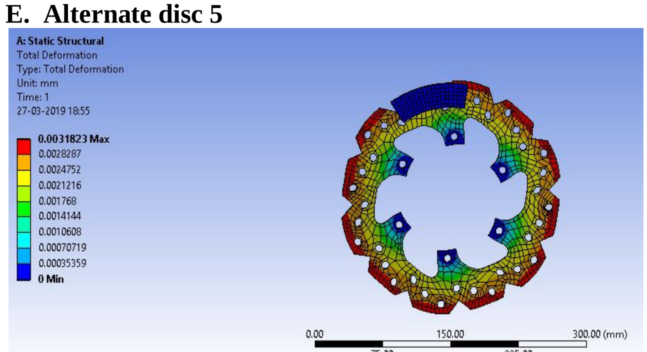


Fig. (e). Total deformation of alternate disc 5

F. Alternate disc 6

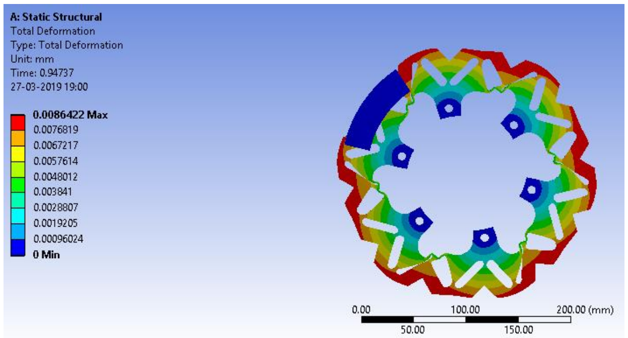


Fig. (f). Total deformation of alternate disc 6

Thermal analysis

Boundary conditions

- maximum permissible temperature - 250°c
- ambient temperature - 23°c
- convection - through non contact area in disc

Results

A. Standard disc 1

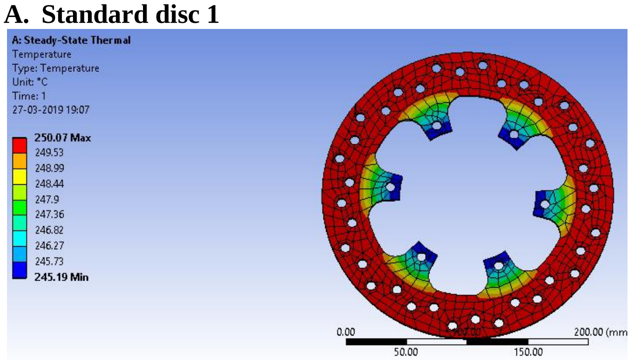


Fig. (a). Temperature of standard disc 1

B. Alternate disc 2

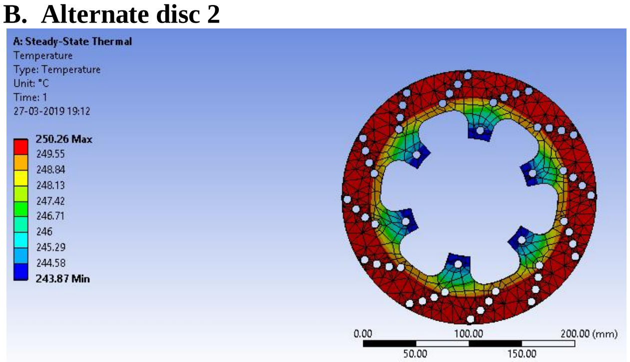


Fig. (b). Temperature of alternate disc 2

C. Alternate disc 3

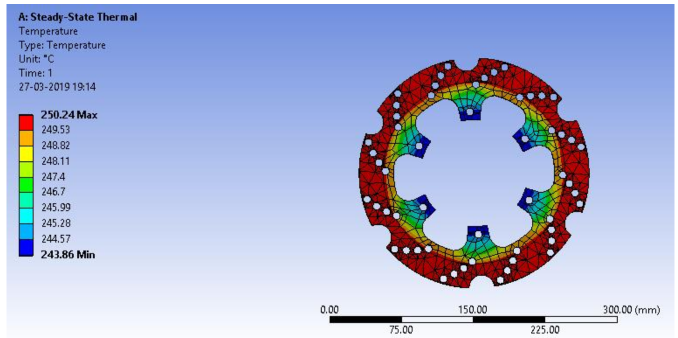


Fig. (c). Temperature of alternate disc 3

D. Alternate disc 4

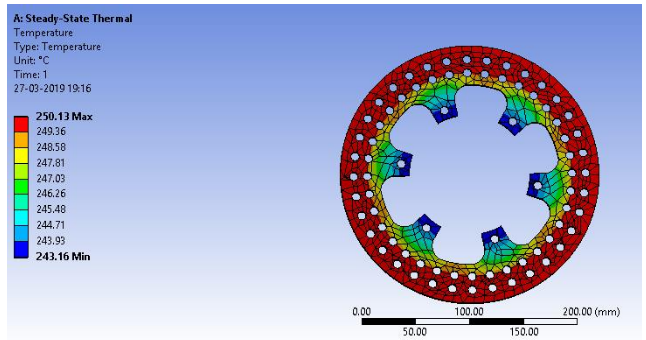


Fig. (d). Temperature of alternate disc 4

E. Alternate disc 5

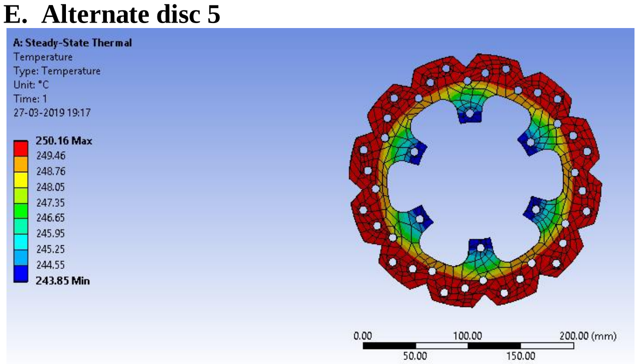


Fig. (e). Temperature of alternate disc 5

F. Alternate disc 6

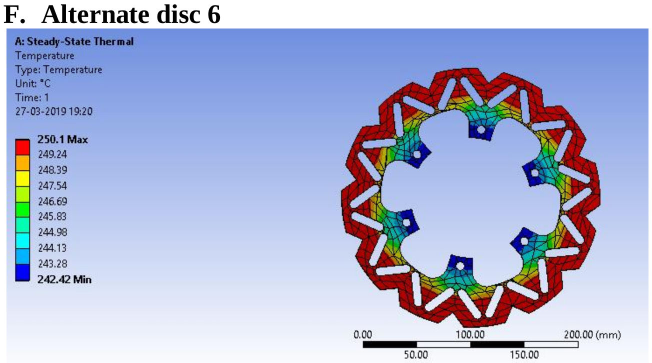


Fig. (f). Temperature of alternate disc

VI. RESULTS

A. Standard disc 1

PARAMETERS	VALUES
TOTAL DEFORMATION (mm)	0.0018638
DIRECT DEFORMATION (mm)	0.0002323
VON - MISSES STRESS (MPa)	1.085
MAX PRINCIPAL STRESS (MPa)	0.56155
WEIGHT (kg)	1.0656
TEMPERATURE (°C)	249.41
TOTAL HEAT FLUX (W/mm^2)	0.0018928

Tab. (a)

B. Alternate disc 2

PARAMETERS	VALUES
TOTAL DEFORMATION (mm)	0.0018909
DIRECT DEFORMATION (mm)	0.0000058284
VON - MISSES STRESS (MPa)	1.2035
MAX PRINCIPAL STRESS (MPa)	0.63453
WEIGHT (kg)	1.0439
TEMPERATURE (°C)	249.38
TOTAL HEAT FLUX (W/mm^2)	0.0028535

Tab. (b)

C. Alternate disc 3

PARAMETERS	VALUES
TOTAL DEFORMATION (mm)	0.0018844
DIRECT DEFORMATION (mm)	0.00000037549
VON - MISSES STRESS (MPa)	1.1924
MAX PRINCIPAL STRESS (MPa)	0.59846
WEIGHT (kg)	0.98451
TEMPERATURE (°C)	249.38
TOTAL HEAT FLUX (W/mm^2)	0.0028759

Tab. (c)

D. Alternate disc 4

PARAMETERS	VALUES
TOTAL DEFORMATION (mm)	0.002139
DIRECT DEFORMATION (mm)	0.00000025394
VON - MISSES STRESS (MPa)	1.1645
MAX PRINCIPAL STRESS (MPa)	0.60581
WEIGHT (kg)	1.0004
TEMPERATURE (°C)	248.95
TOTAL HEAT FLUX (W/mm^2)	0.0044506

Tab. (d)

E. Alternate disc 5

PARAMETERS	VALUES
TOTAL DEFORMATION (mm)	0.0019249
DIRECT DEFORMATION (mm)	0.00000040734
VON - MISSES STRESS (MPa)	1.1911
MAX PRINCIPAL STRESS (MPa)	0.62355
WEIGHT (kg)	1.0005
TEMPERATURE (°C)	248.98
TOTAL HEAT FLUX (W/mm^2)	0.0037049

Tab. (e)

F. Alternate disc 6

PARAMETERS	VALUES
TOTAL DEFORMATION (mm)	0.0050694
DIRECT DEFORMATION (mm)	0.0000006901
VON - MISSES STRESS (MPa)	2.6767
MAX PRINCIPAL STRESS (MPa)	1.4283
WEIGHT (kg)	0.84944
TEMPERATURE (°C)	248.25
TOTAL HEAT FLUX (W/mm^2)	0.005121

Tab. (f)

VII. CONCLUSION

A clear comparison is made between the results from the tables on the basis of each domain, the DISC 2,4,5,6 is found to deform in a comparatively higher value on applying the same load. The DISC 2,4,5,6 takes up a nominally higher Von-misses stress, comparatively nominal rise in temperature, nominal heat flux, with a slight increase in weight. Thus, the DISC 3 is found to deform in a minimum value on applying the same load. The DISC 3 takes up a nominal Von - misses stress, comparatively nominal rise in temperature, nominal heat flux, with a better weight reduction. Thus, the ALTERNATE DISC 3 is proposed to use instead of the STANDARD DISC 1 brake disc rotor, used in BAJAJ PULSAR 150.

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