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Design and Optimisation of Disc Brake Rotor

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Abstract:* This paper deals with design and optimisation of brake rotor of a conventional disc brake used in an All-Terrain Vehicle or ATV. Temperature rise, weight, deformation and stress generation are the factors which are considered for optimisation of the design. For optimisation physical model of brake rotor is designed using CAD software on which analysis (Thermal and Static Structural) is done. The aim is to optimise the disc so that it has efficient cooling along with being light weight and durable.

***Keywords:* Ansys, Thermal Analysis, CAD, Structural Analysis, Design optimisation, all-terrain vehicle**

I. INTRODUCTION

Brakes are the most integral part of the vehicle while considering safety. Any vehicle with good and efficient brakes has fewer chances of accident in comparison to its counterpart with not so good braking. In automotive world, we have several different types of brakes (mainly drum and disc brakes). Use of disc brake has been increasing day by day as they provide better performance with lesser stopping distance. One of the main component of disc braking system is the brake rotor that works against the motion of the wheel to stop the vehicle. Rotor are also designed according to various factors affecting braking performance. In this research brake rotor is optimised to achieve better braking efficiency along with lighter weight of disc.

II. FACTORS AFFECTING DESIGN

A. Heat Dissipation

Holes are provided in a rotor for heat dissipation. During braking a high amount of frictional force is applied on the rotor by the pads resulting in heat generation. This heat can have a negative effect on thermal properties of rotor. Thus, proper dissipation of heat is required

B. Temperature Rise

Too high temperature rise of also effects rotor as higher the temperature it achieves higher will be chances of change in thermal properties and lower the cooling rate.

C. Deformation

The amount of force applied by the brake pads on the rotor is called clamping force. This force acts normally to the area of application and is responsible for wear of rotor. Thus, rotor gets deformed during application. The amount of deformation needs to be as minimal as possible.

D. Stress

Application of opposing forces and moments generates stress on rotor. Stresses generated should be as minimal as possible because higher stresses will lead to eventual failure of rotor along with inefficient braking due to disruption of shape.

III.METHODOLOGY

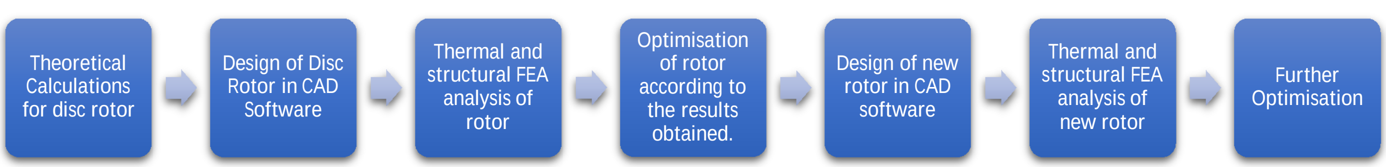

Figure I: Methodology

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IV. CALCULATIONS

S. No.	Assumption
1	Mass of atv (m) = 250 Kg
2	No. of wheels = 4 However, we will be optimising front disc only
3	Initial Velocity (u) = 16.67 m/s
4	Final Velocity (v) = 0 m/s
5	Brake rotor diameter = 0.21 m
6	Axle weight distribution (y) = 0.50
7	Kinetic energy absorbed by disc (f) = 90%
8	Acceleration due to gravity (g) = 9.81 m/s^2
9	Coefficient of friction in pads = 0.4
10	Radius of tyre = 0.2667 m
11	Pedal force (F) = 200 N
12	Pedal ratio = 5:1
13	Diameter of piston in master cylinder (Ø1) = 19mm
14	Caliper pad area = 0.025 m^2

Table I: Assumptions

S. No.	Calculations
1.	Kinetic Energy = 0.25*y*f*m*(u-v)^2 0.25*0.40*0.9*250*(16.67)^2 = 6,252 J
2.	Deceleration μmg = mα $\therefore$ μg = α Hence, deceleration α = 0.4g α = 0.4*9.81 α = 3.924m/sec^2
3.	Frictional Force F = m*α F = 250*3.924 F = 981 N
4.	Frictional Torque T= F*radius of tyre (R) T = 981*radius of tyre (R) T= 261.63 N-m This is the torque required to stop the vehicle. Now, we will calculate Braking Torque generated by hydraulic system
5.	Force in Master Cylinder $F_{mc} = F_{BP}$*Pedal ratio F_{mc} = 200*5 F_{mc} = 1,000 N
6.	Pressure created in master cylinder $P_{mc} = F_{mc}/A_{mc}$ $P_{mc} = 1000/ (\pi/4*(0.019)^2)$ $P_{mc} = 3.6 \times 10^6$ Pa or 3.6 MPa
7.	Caliper Force $P_{master\ cylinder} = P_{caliper}$ $F_{caliper} = P_{caliper}$*Caliper Pad Area $F_{caliper} = 3.6 \times 10^6 \times (0.025)^2$ $F_{caliper}$ = 2,250 N

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8.	Clamping Force $F_{clamping} = 2 \times F_{caliper}$ $F_{clamping} = 2 \times 2250$ $F_{clamping} = 4{,}500$ N
9.	Frictional Force $F_{friction} = \mu \times F_{clamping}$ $F_{friction} = 0.4 \text{ x } F_{clamping}$ $F_{friction} = 1{,}800$ N
10.	For 210 mm diameter of front disc Torque being produced to stop the vehicle $T_{braking} = F_{friction}$*Rotor diameter $T_{Braking} = 378$ N-m
11.	Deceleration a' = $F_{Friction}$/Mass of vehicle a' = 3600/250 a' = - 14.4 m/s^2
12.	Time of application V = u + a'T T = 1.15 sec
13.	Stopping Distance d = u^2/ (4*a') d = (16.67)^2/ (4*14.4) = 4.82 m
14.	Braking Power P_{BP} = K.E./Time P_{BP} = 6252/1.15 P_{BP} = 5,437 W
15.	Heat Flux Hf = P_{BP}/Area of disc Hf = 5437/0.042 Hf = 1,29,452 W/m^2

Table II: Calculations

Since we have assumed weight distribution of 50:50
Heat flux acting on a single disc can be calculated by 129452/4 = 32,363 W/m^2
Also, since disc is cooled by forced air flow, convection current is taken as = 230 W/m^2 (standard value)

V. CAD DESIGN AND ANALYSIS OF ROTOR

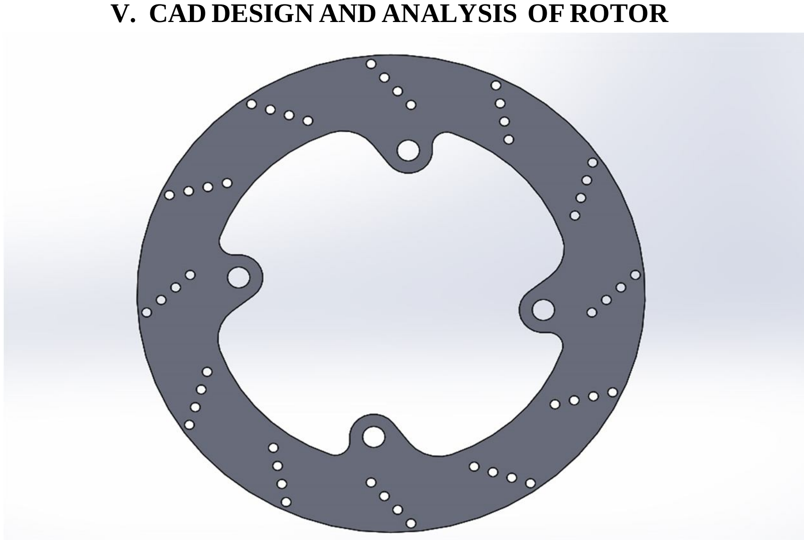

Figure II: Rotor CAD Design

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S. No.	Dimension	Value
1.	Material	Stainless Steel 420
2.	Disc Type	Fully Circular
3..	Rotor Diameter	210 mm
4.	Rotor Thickness	3 mm
5.	Hole Diameter	4 mm
6.	Weight of Disc	435 grams

Table III: Disc Properties

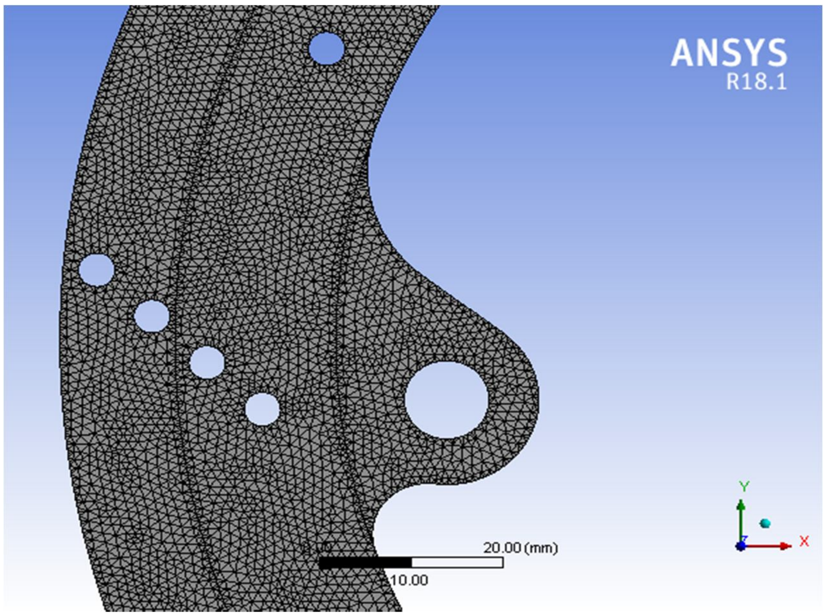

Figure II: Mesh Model of Rotor

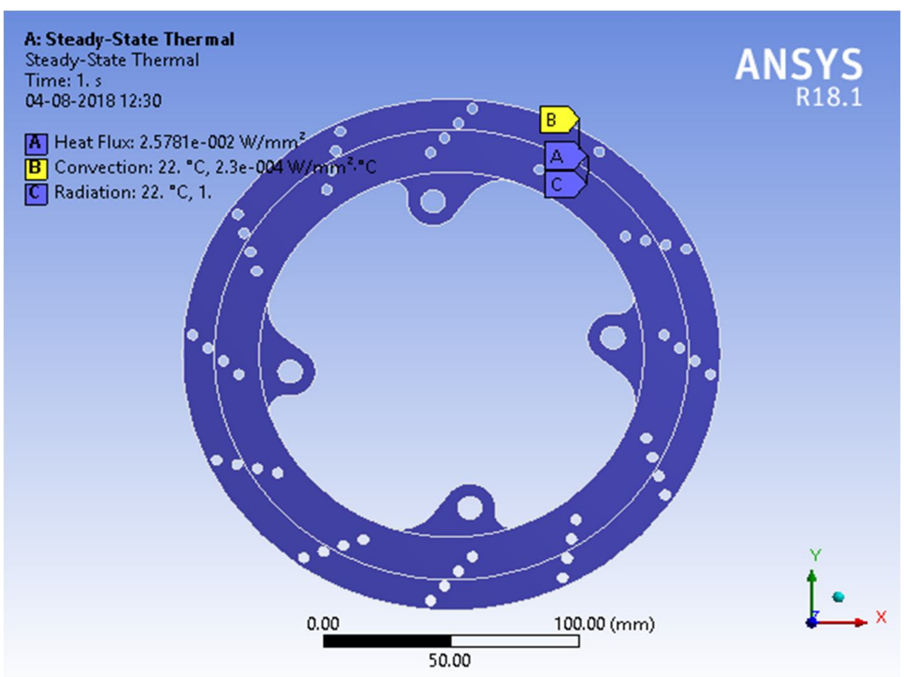

Figure III: Heat Transfer Acting on Rotor

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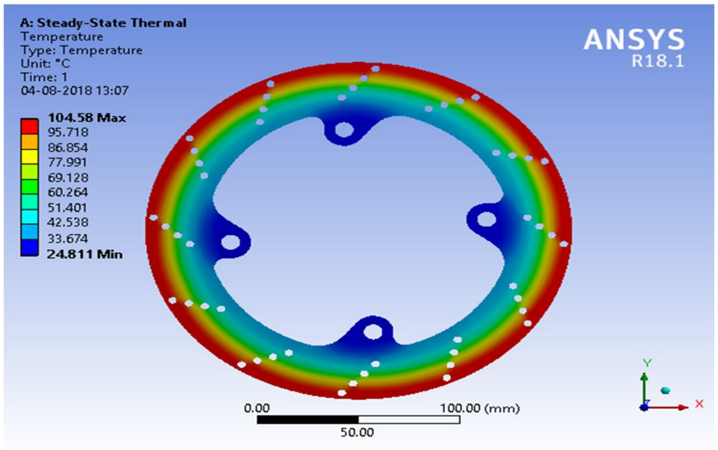

Figure IV: Temperature Rise in Rotor

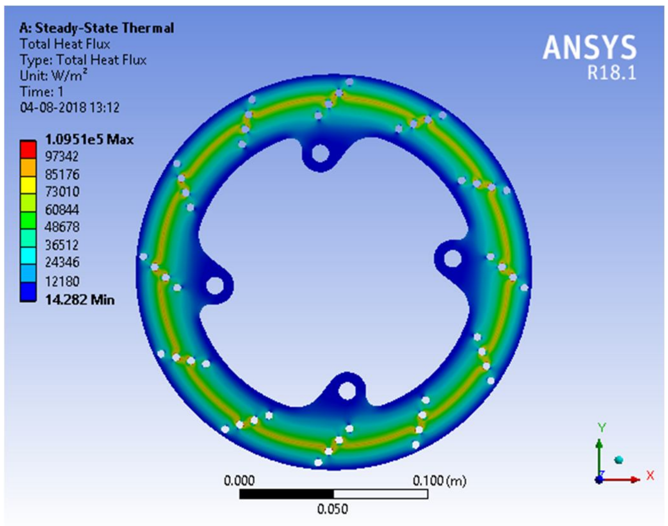

Figure V: Heat Flux through Rotor

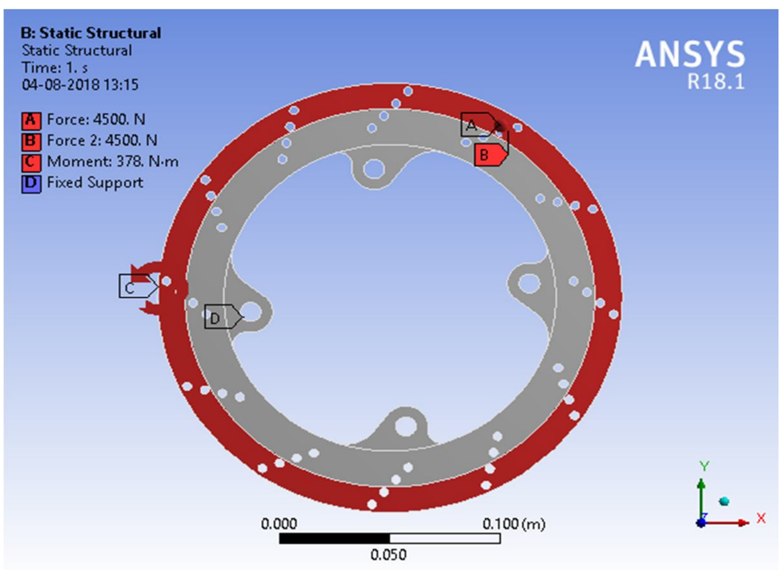

Figure VI: Forces Acting on Rotor

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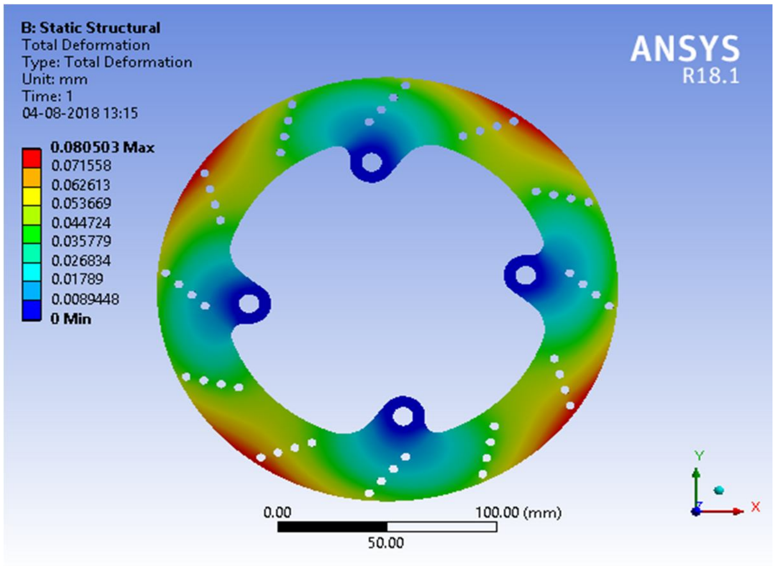

Figure VII: Rotor Deformation

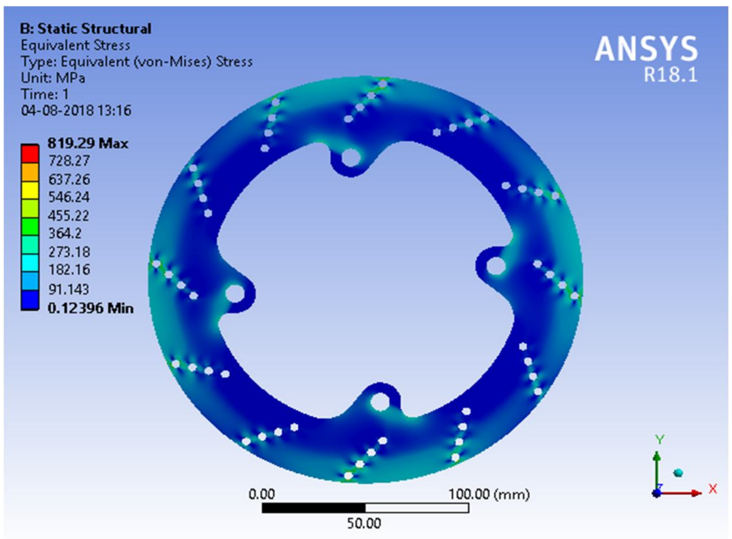

Figure VIII: Stress Acting on Rotor

S. No.	Parameter	Value
1.	Meshing Element Size	0.50 mm
2.	Heat Flux Acting on Disc	32,363 W/m^2
3.	Convection Coefficient	230 $W/m^{2\,0}C$
4.	Force on rotor	4500 N on both sides
5.	Moment on rotor	378 N m

Table IV: ANSYS Parameters

Result: The temperature rise for the rotor is high and flow of heat is relatively low. Also, stress generated on rotor's surface is quite high which will lead to less life and sooner deformation.

Thus, we can optimise this disc to reduce weight along with reduction in stress generation and increase in heat flux causing rotor to cool faster.

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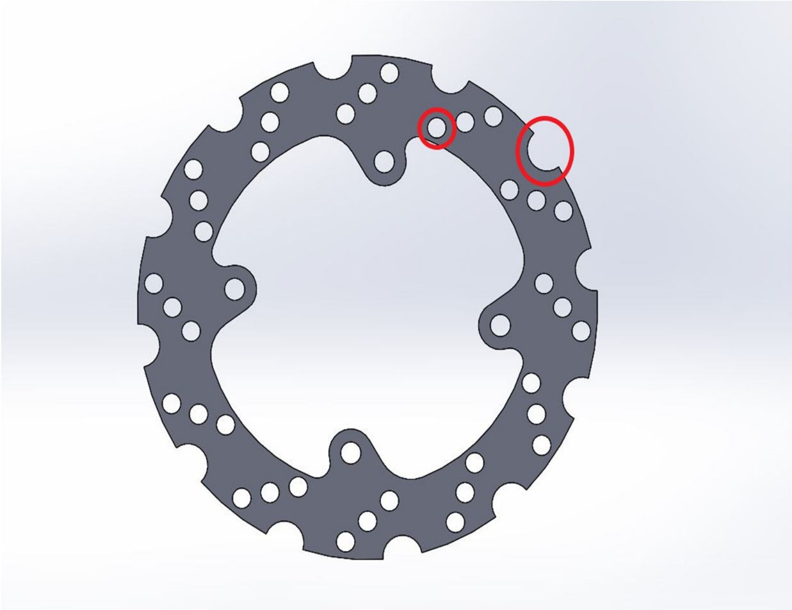

Figure IX: Optimised Rotor CAD Design

Now, we optimise the rotor by increasing holes size on surface of the disc along with cutting petals around the disc outer surface keeping in mind that it doesn't affect our pad area which can lead to failure of rotor along with inefficient braking.

S. No.	Dimension	Value
1.	Material	Stainless Steel 420
2	Disc Type	Petal
3..	Rotor Diameter	210 mm
4.	Rotor Thickness	3 mm
5.	Hole Diameter	8.5 mm
6.	Weight of Disc	360 grams

[1] Table V: Disc Properties

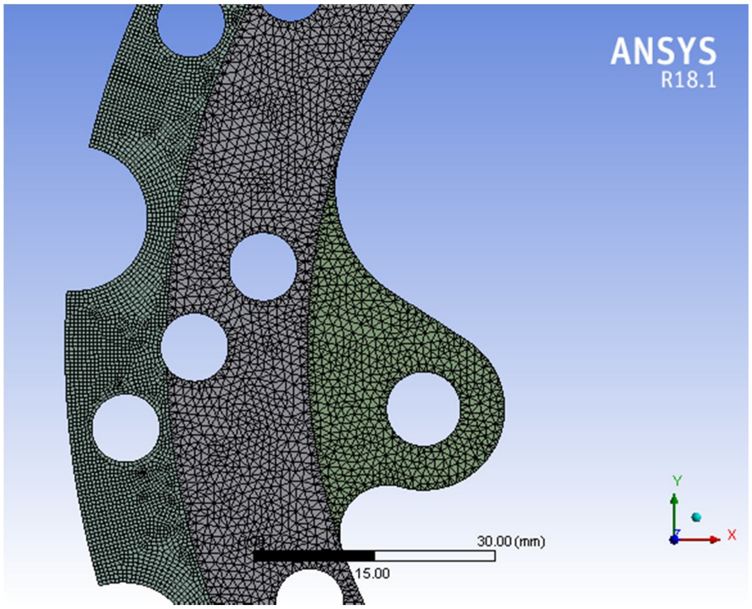

Figure X: Mesh Model of Rotor

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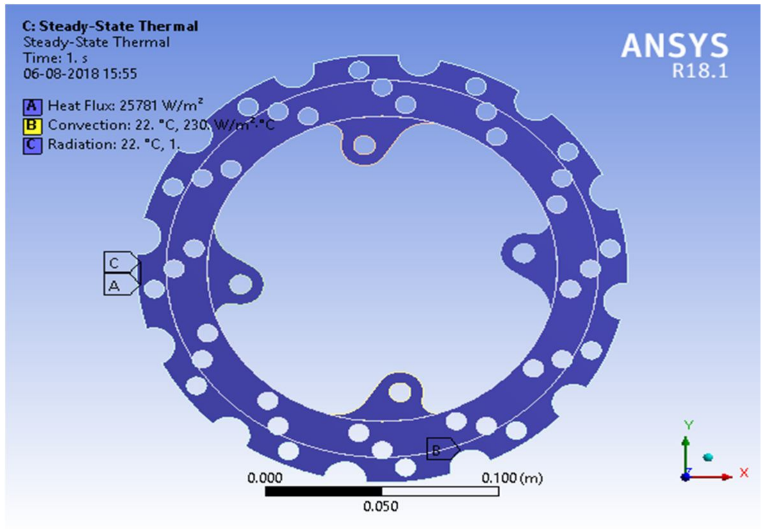

Figure XI: Heat Transfer Acting on Rotor

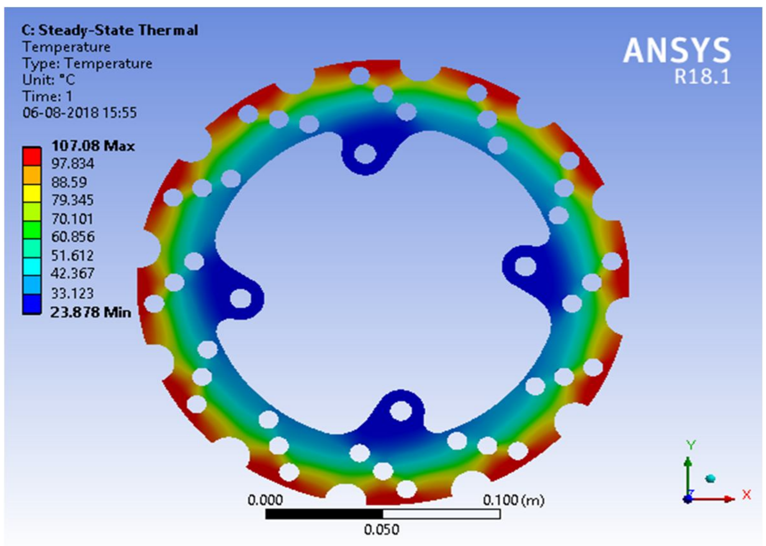

Figure XII: Temperature Rise in Rotor

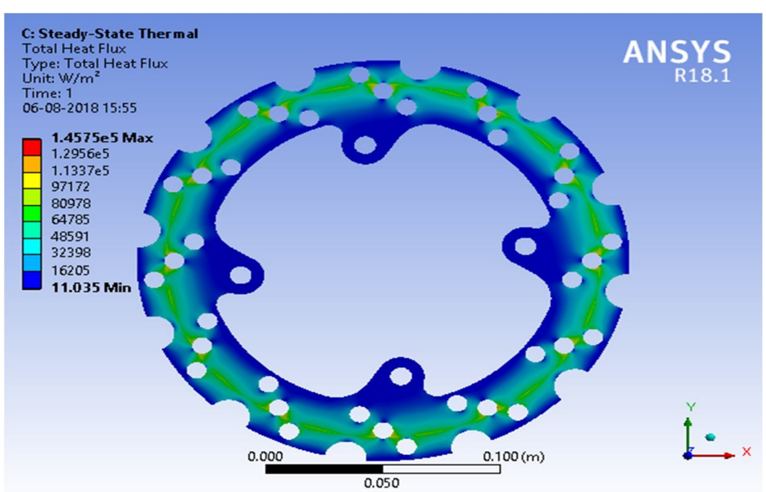

Figure XIII: Heat Flux Through Rotor

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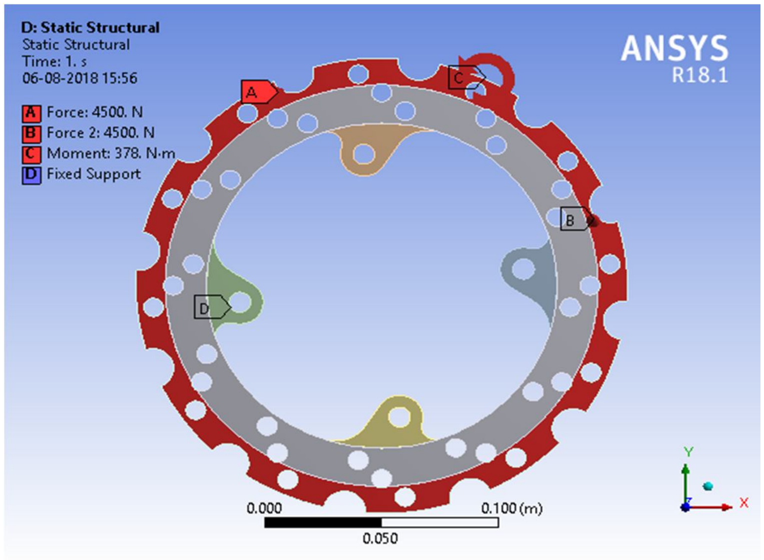

Figure XIV: Forces Acting on Rotor

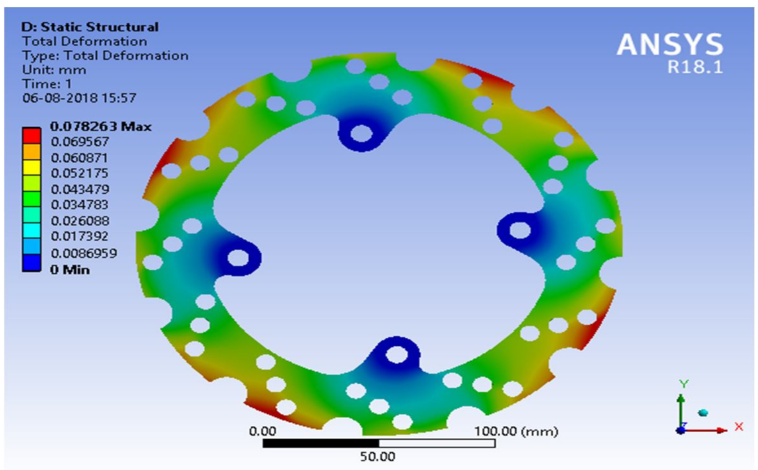

Figure XV: Rotor Deformation

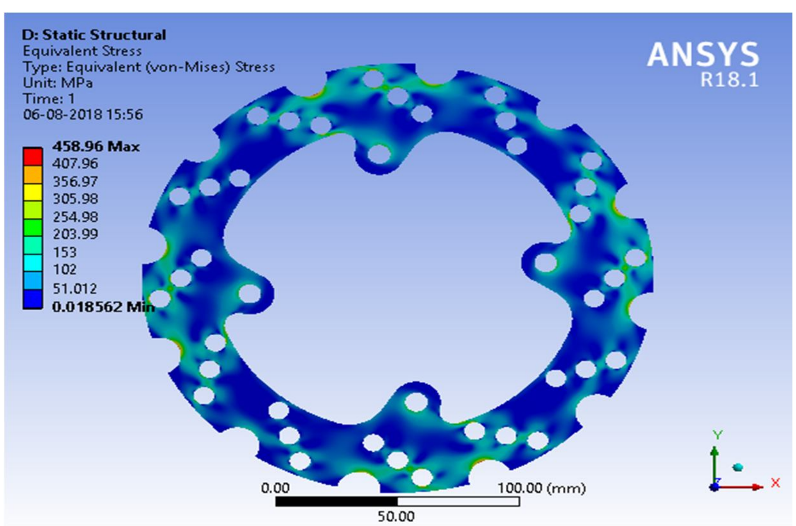

Figure XVI: Stress Acting on Rotor

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VI. CONCLUSION

Thus, by introducing petals and increasing the size of holes on surface rotor's weight is decreased along with stress generation almost reducing to half of the previous value. Also heat flux is considerably increased leading to a faster rate of cooling. Hence, our rotor is optimised without compromising the efficiency. The rotor can be further optimised but introducing more/larger holes or larger petals might result either in failure of rotor or dislocation of pads.

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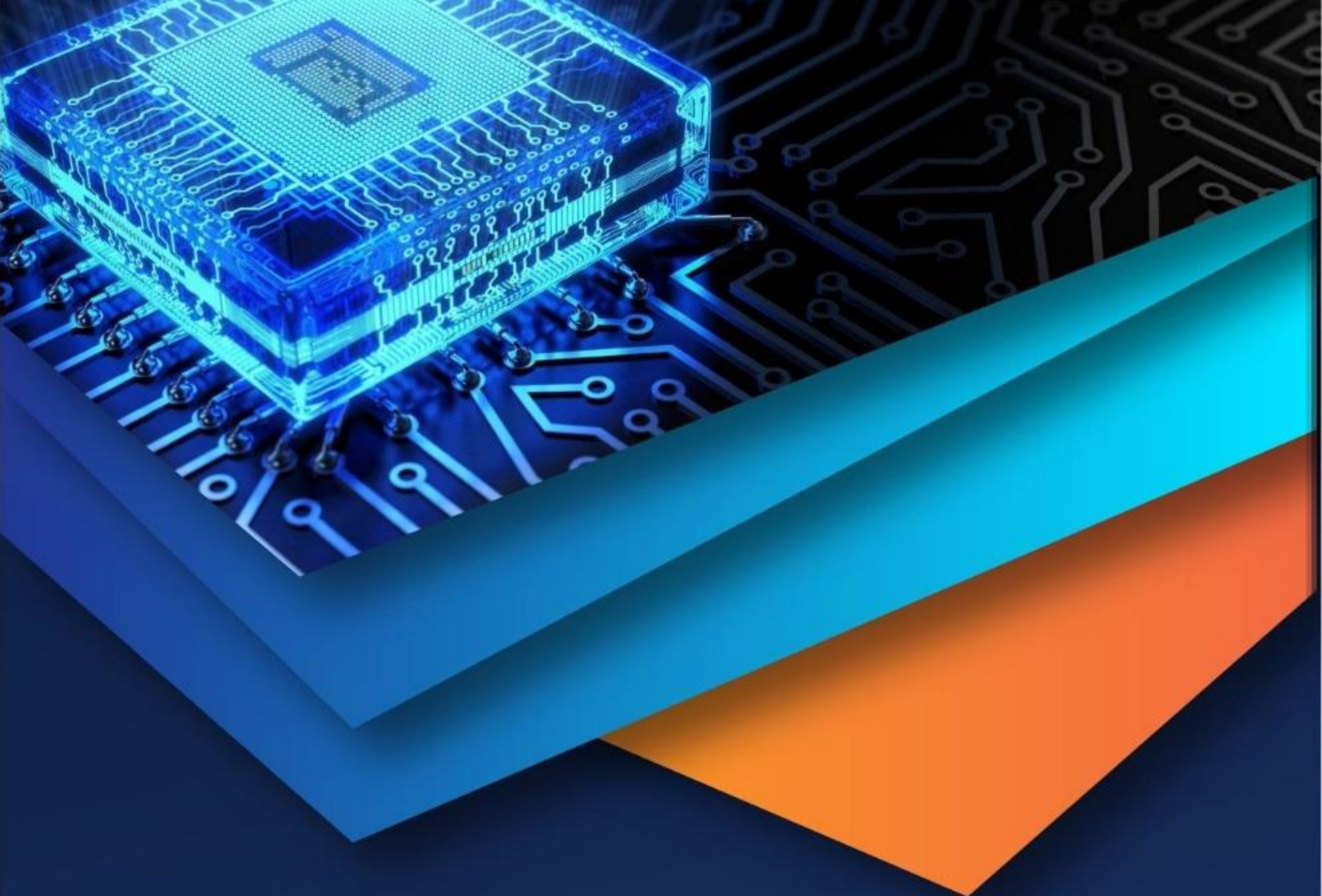

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