Impact of Shape Optimization on Structural and Thermal Design of Brake Disc Rotor

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Abstract –*This paper is a comparative study between an experimental control disc and an experimental design disc designed using shape optimization on Autodesk's Fusion 360, both have the same design parameters, material and input structural constraints. Both discs were designed for a specific brake system and were structurally analysed by conducting a Finite Element Analysis on ANSYS 19.0. Thermal characteristics of both the discs were calculated according to their design and brake system parameters. Based on this calculation, a transient thermal analysis was conducted for 4 braking cycles with an additional thermal structural analysis to find variation of structural design under thermal loading conditions. The results from the analysis showed that the optimized disc had better structural design in absence of thermal loading with 49.95% lesser deformation. Under thermal loading it was seen that the amount of average heat flux generated was 53.2% more with an almost similar average temperature (1.37% less), concluding that the optimized disc had better heat dissipation rate. Although under thermal loading the structural design showed 19.45% higher average stress being generated with a 16.12% lesser average deformation owing to shape optimization and 9.5% lesser swept area.*

Key Words: **Shape Optimization, Topology Optimization, Brake Disc, BAJA, Thermal Analysis.**

1. INTRODUCTION

Brakes are an integral part of any moving vehicle, their purpose being to convert kinetic energy into heat energy, Brake actuation results in the vehicle decelerating and approaching a stop. Disc brakes are radial brakes which were first introduced in the late 1940's since they provided better resistance to fade, better thermal cooling tendency, were resistant to debris, water and required lesser maintenance [1]. With improved alloyed materials, design innovation coupled with the quest for peak efficiency, brake manufacturers have opted to reinvent the traditional design by using weight reduction along with additive and composites manufacturing. The purpose of weight reduction is to decrease the overall mass with an aim to improve efficiency, reduce fuel consumption and reduce material cost. For a 50% weight reduction considered for a brake rotor, the predicted energy savings is 19% [2] and for every 10% of weight reduction there was a decrease of 7% in fuel consumption [3].

1.1 Types of Brake Discs

Majorly there are two types of brake disc, Solid rotor and Vented rotor. A solid rotor is a flat plate disc with two contact surfaces with a uniform thickness between them, traditionally there used to be no cut patterns as this was done to improve the overall swept area of the disc to ensure higher brake performance. But with further studies and experiments it was seen that thermal stress induced in these discs made them far less efficient as in course of braking the brake lining due to thermal stress would expel gases which would act like a lubricant between the pads and the disc causing friction loss and premature brake fade [1]. To improve thermal stress and reduce brake fade a Vented type of disc was designed with radial cooling passage pattern present inside between the two contact surface. This pattern which would replicate the blades of a pump impeller allowed air to enter through the center and exit radially, which would greatly enhance the disc's heat dissipation capacity

1.2 Types of Disc mounting

There are two different types of brake mounting, Bolted types and Dog-Drive [1]. The prior consists of holes to accommodate disc bolts and the latter consists of radial slots which are either fitted on the hub or on the rotor hat. Discs having holes can be subjected to stress concentration due to thermal stresses whereas dog-drive slots allow disc to expand uniformly keeping it centered during high braking heat cycle

2. LITERATURE REVIEW

Shape optimization is a software-based simulation which uses optimal control theory to find shape which is varied by varying an optimal set of constraint parameters. According to Alexis et al. (2006) [4] shape optimization is a gradient based method which uses Hadamard's Boundary variation method to find the changed mesh-point displacement using a Langranian functional multiplier. Research in area of shape optimization on brake disc from Pasqual and Malcher(2020)[5] found that in a comparative study for brake disc analysis for BAJA SAE vehicle it was found that shape optimized disc resulted in better heat dissipation , Shape optimized design although efficient is not so frequently used in mass production as seen from Nizam Sudin et al.(2014)[6]study where it was found that further refinement for a shape optimized model was required to ensure possibility for production. The advantage of implementing optimization for disc was seen from Chavan et al. (2018) [7] study of Thermal Analysis of brake disc which showed that cut-pattern within a disc played an important role in braking performance. To test the validity of simulation analysis with real world data, it was seen from Vidya and Singh B. (2017) [8] Experimental and numerical thermal analysis that the variation was almost similar in nature with approximately same maximum temperature. Even though Computational Model can never perfectly predict actual conditions, a comparative study can surely give an indication of its design efficiency.

3. BRAKE DISC MATERIAL SELECTION

Gray cast iron is generally used for disc rotors since they have good friction and wear resistance, although for our case the disc designed will be used for a BAJA specific All-Terrain vehicle and hence the structural requirements are also limited to its use. Considering the availability, machinability and cost, Stainless Steel AISI 430 was selected as the disc material having:
Density = 7750kg/m^3
Specific Heat Capacity = 460 J/Kg.K
Thermal Conductivity = 26.1 W/m.K

4. BRAKE DISC DESIGN

In order to analyze the disc with proper boundary conditions. The disc is designed in accordance with the brake system parameters which are further linked to the vehicle dimensions. Based on this input parameters a brake system is selected which will determine the final braking performance of the vehicle. The disc designed will be only considered for Rear Circuit.
Design Parameters:
Mass of the vehicle (M) =210 Kg
Wheelbase (W.B) = 1.384 m
Centre of Gravity (C.G) = 0.45 m
Constant deceleration (D) =0.68g
Tire rolling radius = 0.2921 m
Weight Distribution ratio = 45:55

The system parameters were:
Pedal ratio = 7:1
Master Cylinder Bore diameter = 19mm
Pedal effort = 45 kgf
Piston Diameter = 31.242mm

A MATLAB model was used to calculate the required braking torque considering the dynamic weight distribution and tire rolling radius. The torque required to stop the vehicle at 0.68g of deceleration was found to be 134Nm for the rear circuit. Chart 1 shows the variation of Brake torque required with varying deceleration.

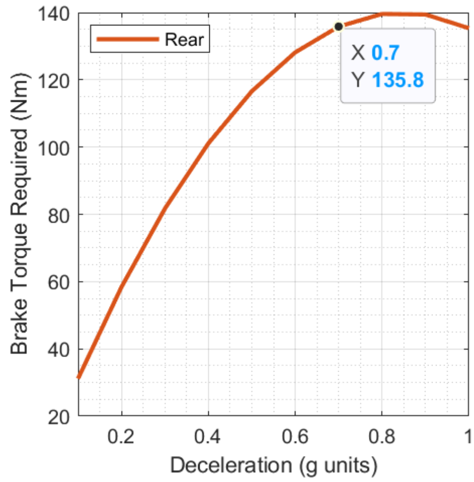

Chart -1: Brake torque required vs Deceleration

The MATLAB model was further used to calculate the brake torque generated considering the caliper used was single piston floating caliper. Chart 2 shows the Brake torque generated when the effective radius was 0.08m. The Clamping force produced was 8355.4 N

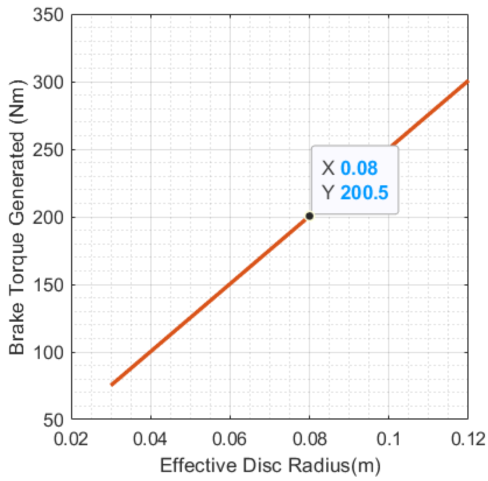

Chart -2: Brake Torque generated vs Effective radius of the disc

In order to maintain a safety factor of 1.5 an effective radius (Re) of 80mm was selected. Uniform Wear Theory is considered to calculate the effective radius is

$$Re = \frac{1}{4}(O.D + I.D) \tag{1}$$

From the design parameters a specific flat plate brake rotor of Outer Diameter (O.D) 190mm, Inner Diameter (I.D) 130mm, Pitch Circle Diameter (PCD) 100mm and Thickness (T) 4mm with 4 M10 holes 90º apart was selected.

4.1 Experimental Design Disc based on Shape Optimization

Initial disc of OD 190 mm was selected and shape optimization simulation was carried out in Autodesk's Fusion 360 software. The load conditions found from the system design being 200 Nm of torque were applied for all 4 holes collectively in anticlockwise direction, outer thickness of disc was fixed and 4 cuboidal regions per hole were preserved up to 20mm diameter. The target mass was set to below or equal 40% of total mass with maximum stiffness criteria. The result generated by the simulation are shown in Figure 1.

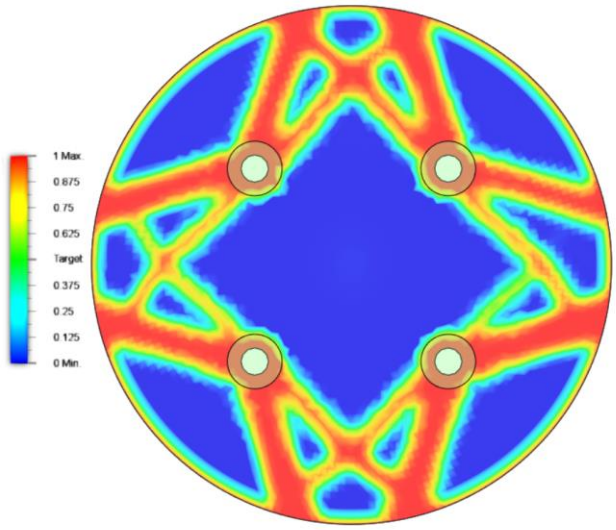

Fig -1: Shape Optimization result on Fusion 360

4.2 Interpreting the Result

Based on the load conditions and mass criteria the software determines the critical load path within the structure which is responsible for distributing stress throughout the disc. The target range of 40% mass ratio developed was represented in Figure 2.

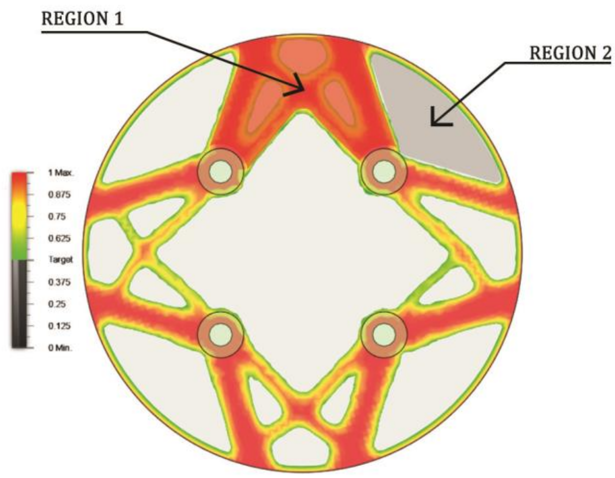

Fig -2: Target mass result

The converted load path consists of two symmetric regions. Region 1 is the overlapping load path between every adjacent hole and Region 2 is the area above every hole. Target mass based on load path was converted to a mesh body disc as seen in Figure 3.

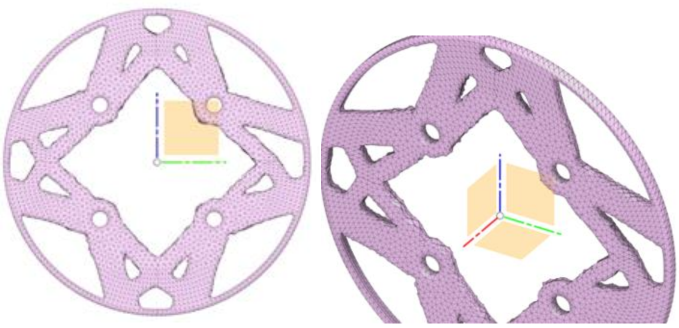

Fig -3: Promoted Mesh

This promoted mesh was used to design a disc such that Region 1 would have an intersecting section and Region 2 would have a triangular section. Figure 4 represents this design implementation with two straight lines in Region 1 and a triangular section for Region 2.

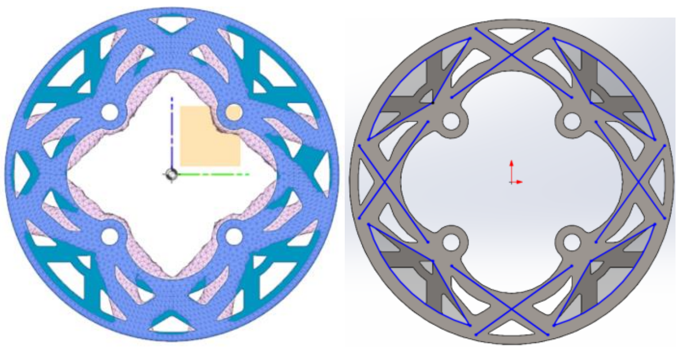

Fig -4: Optimized Disc

4.3 Experimental Control Disc Design

A brake disc was designed which would act as an Experimental Control. The purpose of this disc was to replicate the weight reduction and initial design parameters without implementing any design innovation or shape optimization. A drilled and slotted disc was designed considering the disc parameters and mass. Figure 5 shows this control disc

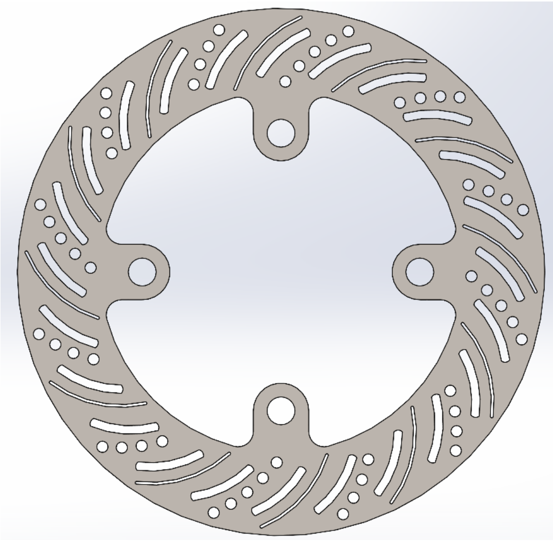

Fig -5: Experimental Control Slotted and Drilled Disc

Table -1: Experimental and Control disc comparison

Sr. No.	Para-meters	Design Disc	Control Disc	Variation with respect to Control
1.	Material	Stainless Steel (Density= 7750 kg/m^3)	Stainless Steel (Density= 7750 kg/m^3)	0%
2.	Mass (g)	425.74	426.27	0.1243 % less
3.	Volume (m^3)	5.4934E-005	5.5002E-005	0.1236 % less
4.	Surface Area (mm^2)	38594.74	44054.43	12.39 % less

The Control disc designed would replicate same mass and volume although owing to shape optimization the surface area of the designed disc was found to be 12.4 % lesser as compared to the non-optimized disc

5. STATIC STRUCTURAL DESIGN VALIDATION USING FEA

Both the discs were validated by conducting a Finite Element analysis on ANSYS 19.0 software. The Boundary conditions involved were, Moment of 200 Nm anticlockwise was applied to both surfaces with all 4 M10 holes were kept fixed

5.1 Experimental Design Disc Analysis

Stainless Steel was selected for the material geometry with material properties matching AISI SS430. Patch Conforming Method was used for meshing tetrahedron elements. The Nodes created were 46034 and elements formed were 25010. Figure 6 and Figure 7 represent Total Deformation and Equivalent Stress respectively

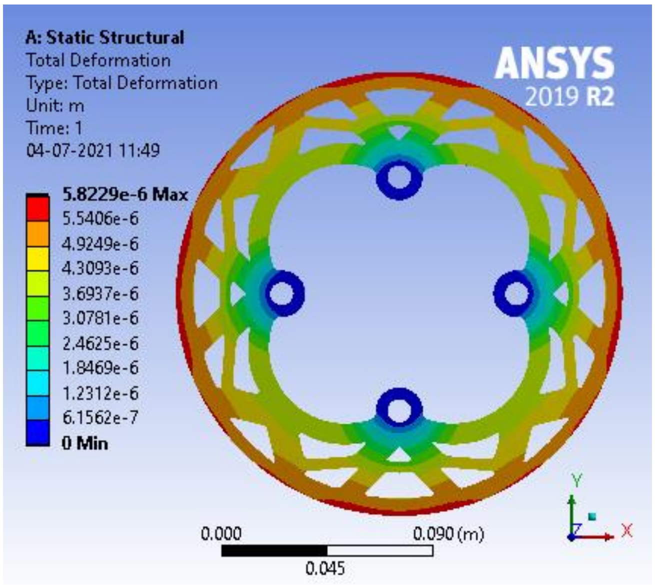

Fig -6: Total Deformation of Design Disc

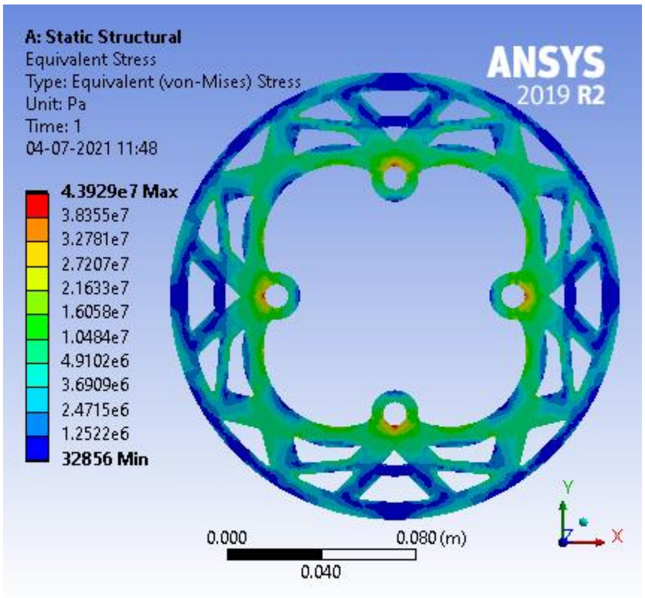

Fig -7: Equivalent Stress of Design Disc

5.2 Experimental Control Disc Analysis

Patch Conforming Method was also used to generate mesh with the final Nodes being 30420 and elements generated being 15237. Figure 8 and Figure 9 represent the Total Deformation and Equivalent Stress for Control Disc respectively

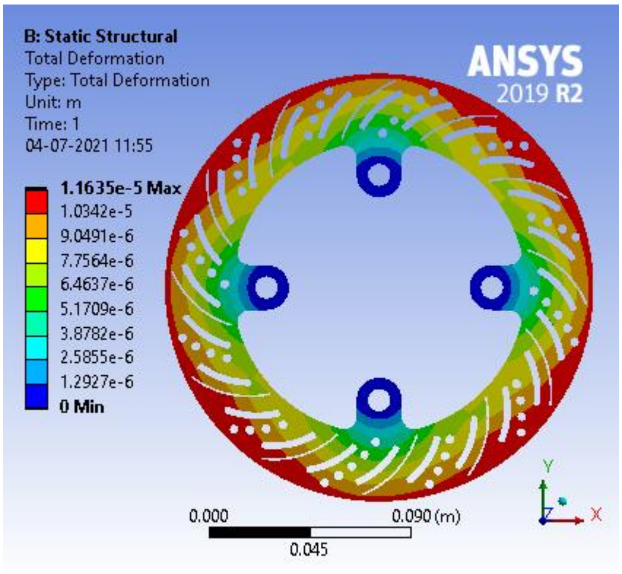

Fig -8: Total Deformation of Control Disc

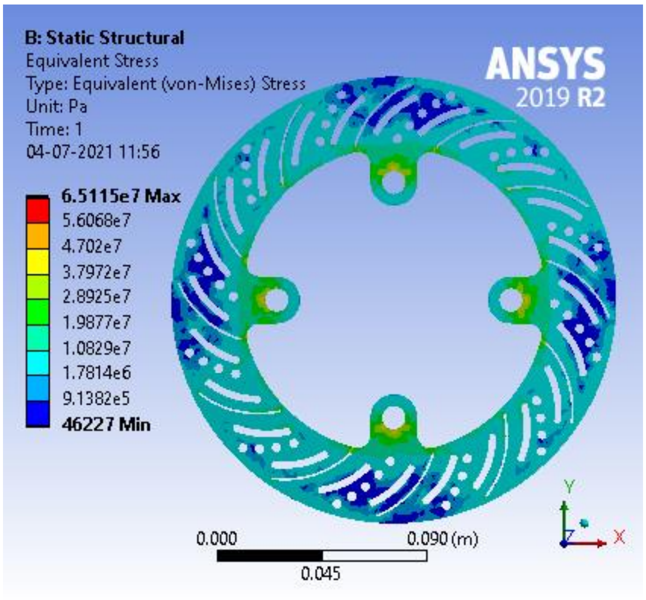

Fig -9: Equivalent Stress of Control Disc

5.3 Static Structural Results

From the results I observed that the Design disc underwent lesser deformation, it had less stress formation and had a higher Factor of safety. Owing to lesser deformation the amount of plastic deformation is also limited leading to lesser microscopic cracks being formed, thus improving the disc's long term fatigue life. In Figure 7 it was also seen that the Region 2 of the disc had lesser stress concentration as compared to stress in region 1, which further suggested that shape optimized weight reduction based on the load path was efficient. Table 2 represents the comparison between the two discs and their variation

Table -2: Static Structural disc comparison

Sr. No.	Para-meters	Design Disc	Control Disc	Variation w.r.t control
1.	Maximum Deformation (m)	5.8229E-6	1.1635E-5	49.95% less
2.	Average Deformation (m)	3.87E-06	8.33E-06	53.54% less
3.	Maximum stress (Pa)	4.3929E+7	6.511E+7	32.56 % less
4.	Average stress (Pa)	4.79E+06	7.38E+06	35.09% less

5.	Strain Energy(J)	2.5365E-5	4.0702E-5	37.68% less
6.	Factor of Safety (Min)	4.7121	3.179	48.22 % More

6. THERMAL DESIGN VALIDATION

During braking the friction pads restrict the heated disc from expanding, the disc in response applies a reaction force on the pads, Due to this reaction force, an additional stress is generated within the disc known as thermal stress. During a hard stop the outer surface tries to expand but the cold interior prevents it from doing so and hence the outer surface undergoes compression with the interior under-going tension. As the disc cools, thermal stress also minimizes due to temperature equalization which further causes the outer surface to undergo tension [1]. This continued cycle of repeating stresses cause formation of cracks which leads to premature disc failure and warping. In order to consider this effect, thermal analysis is conducted to validate the disc's design under thermal loading conditions. Assuming all kinetic energy is dissipated to heat energy during deceleration, the amount of Braking power generated as per the disc design and system parameters is found using Limpert's [9] brake power equation for deceleration on level surface

$$E = \frac{M}{2} \times K \times (V_1)^2 \quad , \text{Nm} \qquad (2)$$

E= Braking Energy
M= Mass of Vehicle (kg)
V1= Maximum velocity (m/s)
K= Correction Factor for Rotating Masses

Braking power (P) is Rate of change of energy upon time

$$P = \frac{dE}{dt} \quad , \text{Nm/s} \qquad (3)$$

Deceleration is taken as constant, therefore velocity V(t) is given by

$$V(t) = V_1 - AT \quad , \text{m/s} \qquad (4)$$

A=Deceleration
T=Time Elapsed

From eq. 3 and eq 4 we get brake power generated [9]

$$P = KMA(V - AT)$$

Considering a case where this brake disc will be used for a BAJA all-terrain vehicle, here let
M=210 Kg
V1=50 km/hr. = 13.8 m/s
A=0.68g = 0.68*9.81 = 6.67m/s²
T=0 to 2 seconds with an increment of 0.5
K=1 (Continuous Variable Transmission)

Considering an all-wheel outboard braking system. Brake power per disc rotor is further obtained by multiplying power equation by static weight distribution ratio. Chart 3 represents the variation of brake power during 1 cycle of brake actuation obtained from a MATLAB thermal model.

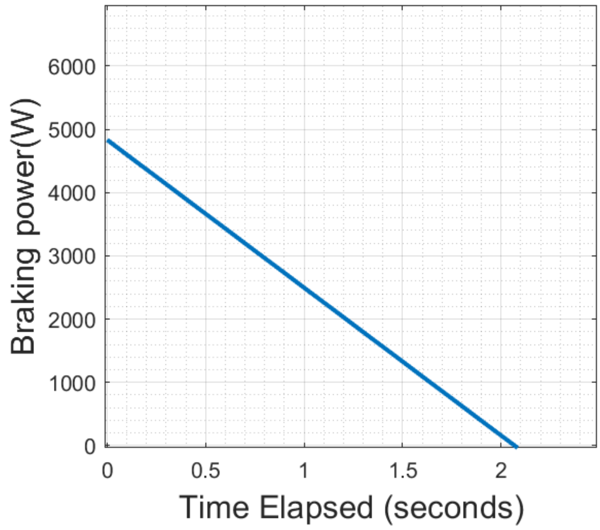

Chart -3: Brake Power vs Time Elapsed

Heat flux generated is equal to the brake power divided by the total brake pad swept area of disc.

Swept area of:

Design Disc = 0.020822m²

Control Disc = 0.02301m²

Swept area of experimental design disc was 9.5% lesser than control disc, which resulted in a higher heat flux being generated during braking. Chart 4 represents the variation in heat flux generated in both disc during complete stopping

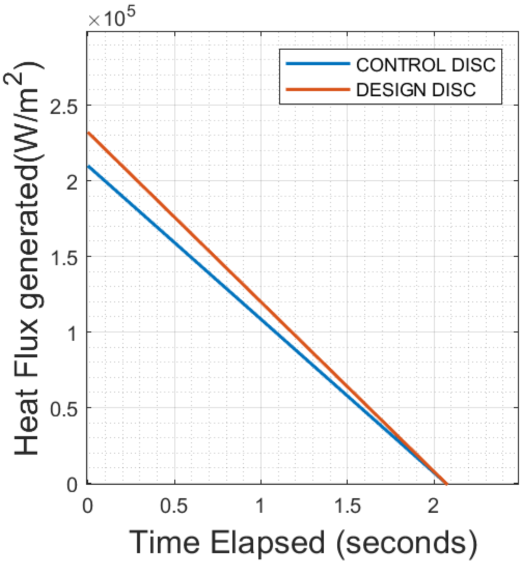

Chart -4: Heat Flux Generated vs Time Elapsed

Heat dissipation takes place by three methods namely, Convection, Conduction and Radiation. Since conduction leads to redistribution of heat rather than heat dissipation [4], it is neglected for brake disc analysis. Numerical study by Vidiya and Singh[8] found that the Convection co-efficient for brake disc travelling at 13.8m/s was 243.5 W/m$^{2\circ}$ºC. Radiation emissivity factor for a brake rotor in Dynamic regime was found to be between 0.4-0.35 for disc temperatures between 140ºC- 230ºC from Dragomir et al. [10]. The values of time dependent heat flux generated for both discs were presented in Table 3

Table -3: Heat flux values for both discs

Sr. No.	Time (sec)	Braking Power (watt)	Heat Flux (W/m²)	
			Design Disc	Control Disc
1.	0	4833	232110	209960
2.	0.5	3664.9	176010	159210
3.	1	2496.8	119910	108470
4.	1.5	1328.7	63810	57720
5.	2	160.5	7710	6970

6.1 Transient Thermal Analysis

Transient Thermal Analysis was carried out by in Ansys Workbench 2019, The model was exported to ANSYS Mechanical, material used was stainless steel, Initial temperature was set to 30ºC, Radiation was limited to pad swept area with emissivity set to 0.4. Convection was applied to entire disc with a convection co-efficient of 240 W/m$^{2\circ}$ºC. In order to properly visualize a braking scenario, Heat flux was provided on the pad swept area of the Disc, with 19 steps each of 0.5 second sub step to stimulate 4 braking cycles each with a timeframe of 2 seconds. Table 3 shows the values of heat flux used for the transient thermal analysis for both control and design disc for one cycle each. Figure 10 and Figure 11 represent the Transient Thermal analysis of control and design disc respectively

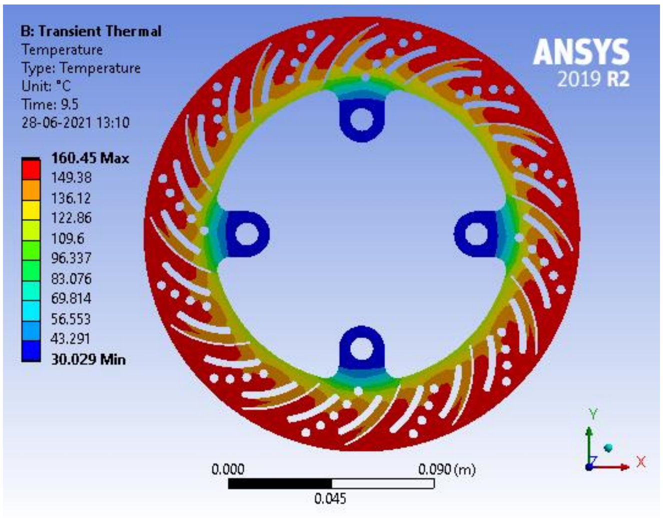

Fig -10: Transient Thermal Analysis of Control disc

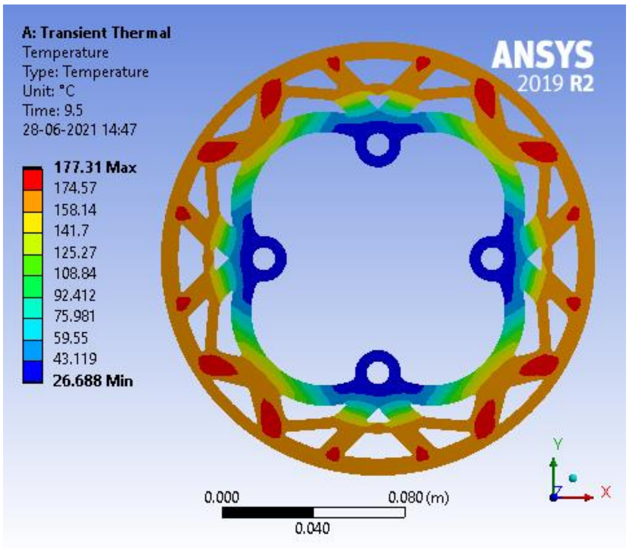

Fig -11: Transient Thermal Analysis of Design disc

6.2 Transient Thermal Analysis Results

The result which was obtained from the Analysis were compiled on MathWorks' s MATLAB software and the combination of the plots representing the variation of the heat flux and temperature change for both the discs were plotted on Chart 5 and Chart 6.

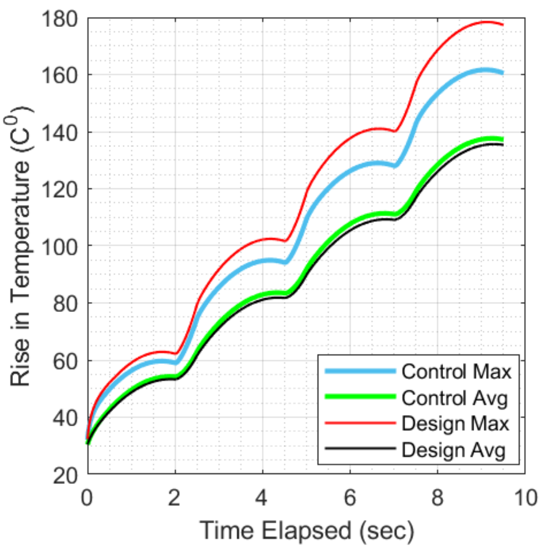

Chart -5: Rise in temperature,

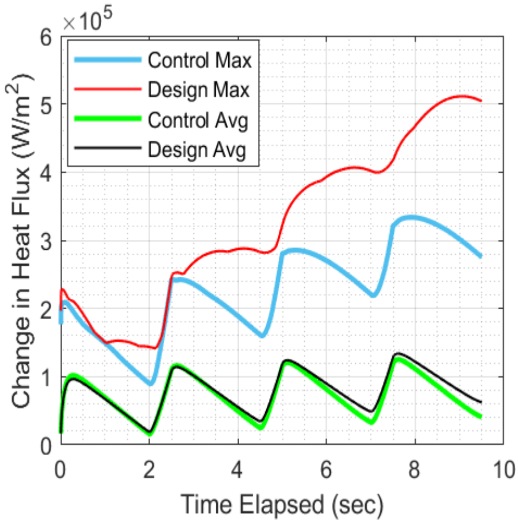

Chart -6: Rise of Heat Flux

Since the swept area of the design disc was 9.5% lesser than the Control disc the maximum temperature of the control disc was also 10.5% more than the design disc as seen on Chart 5, owning to a lesser swept area, the heat dissipation occurring in the optimized disc throughout the braking cycles was higher despite having a larger increase of heat flux with the average rise in temperature almost the same for both the discs, with the design disc even having 1.37% lower average temperature. The advantage of using shape optimized disc was that it reduced hot gases and dust particle build up during brake actuation and helped reduce brake fade owing to lesser swept area The nature of maximum heat flux rise was also steeper for the Design Disc (as seen on Chart 6) with the same average rise for both the discs proving that shape optimization had better heat dissipation and cooling tendency. Table 4 represents the variation of thermal properties of both discs

Table -4: Transient Thermal Result Variation

Sr. No.	Para-meters	Design Disc	Control Disc	Variation w.r.t control
1.	Maximum Temperature(ºC)	177.31	160.45	10.5% more
2.	Average Temperature(ºC)	135.34	137.22	1.37% less
3.	Maximum Heat Flux (W/m^2)	5.04E+05	2.75E+05	83.08% more
4.	Average Heat Flux (W/m^2)	62221	40613	53.20% more
4.	Nodes	25405	27976	-
5.	Elements	12884	13757	-

6.3 Thermal Structural Analysis

To further validate structural validity after 4 cycles of panic brake cycle, a thermal structural analysis of the disc was further carried out in Ansys Workbench 2019. The resultant temperature from each transient structural analysis were imported further on, added boundary conditions applied to the static thermal were rotational velocity of 100 rad/s about disc center axis in anti-clockwise direction.
Parameters found initially from system design
Clamping force of Brake Caliper = 10000N
Brake Pad Area (Wilwood GP 200)= 1.83 in² = 0.00118 m²
Pressure per pad on disc = 4.234 Mpa

This pressure was applied to disc swept area on both individual surfaces and all 4 M10 bolt holes were fixed Figure 12 and 13 represent the deformation developed in the experimental design and control disc.

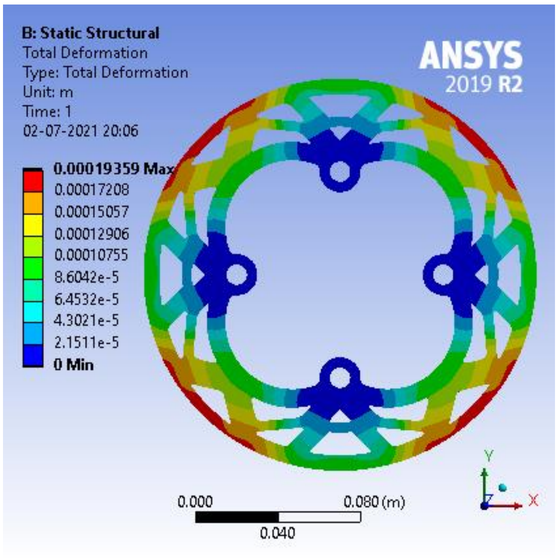

Fig -12: Thermal Structural Analysis of Design disc

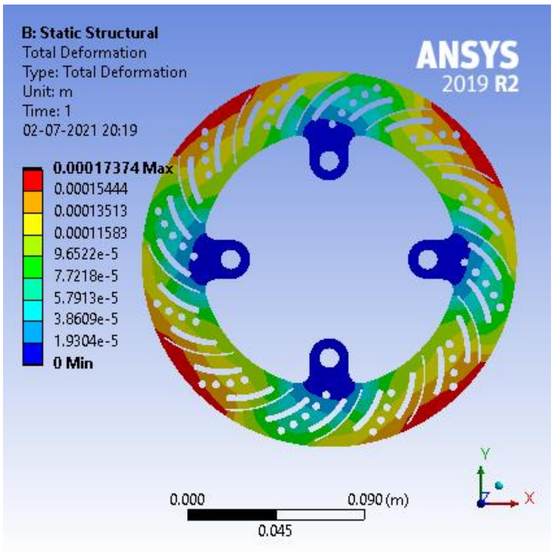

Fig -13: Thermal Structural Analysis of Control disc

6.4 Thermal Structural Analysis Result

From the results it was observed that the design disc on average had 19.45 % more stress developed with 16.12% lesser overall deformation but had a 35% lower factor of safety. Since maximum deformation occurred in region 1 of the disc which from shape optimization simulation was found to have majority of the load path, Region 2 having the most weight reduction area, the overall deformation occurred was also less as compared to the control disc. Table 5 represents the variation of thermal structural analysis results.

Table -5: Thermal Structural Result Variation

Sr. No.	Parameters	Design Disc	Control Disc	Variation w.r.t control
1.	Maximum Stress (Pa)	9.51E+08	1.43E+09	33.67% more
2.	Average Stress (Pa)	1.87E+08	1.56E+08	19.45% more
3.	Maximum Deformation (m)	1.94E-04	1.74E-04	11.42% more
4.	Average Deformation (m)	7.77E-05	9.27E-05	16.12% less
4.	Factor of Safety (Avg)	1.9906	3.0833	35.43% less

7. CONCLUSIONS

From this research study it was seen that brake disc designed using shape optimization had better static structural design as compared to control disc with 49.95% lesser total deformation and 32.56% less stress being generated , Thermal analysis conducted showed that the average heat flux generated was 53.20% more as compared to control disc along with an almost similar average temperature (design disc had 1.37% lesser average temperature) concluding that the optimized disc had better heat dissipation rate even though maximum temperature produced in design disc was 10.5% more owning to 9.5% lesser pad swept area. Thermal structural analysis conducted showed that the design disc under thermal loading underwent 19.45% greater average stress formation with 16.12% lesser average deformation. From these results it can be said that

- Shape Optimization is suitable for discs which are used for braking performance limited only for short duration of time where cooling is critical and high heat dissipation is required with spare discs being replaced after every few kilometers. example Formula SAE cars, BAJA SAE All-Terrain vehicles etc.
- For long term reliability where good efficiency in terms of structural stability under thermal exposure is required, shape optimized disc is not necessary

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BIOGRAPHIES

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