

# Design and Optimization of Perimetric Disc Brake Rotor

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**Abstract**— In today's era, brake designers of the greatest automotive companies are being pushed to their limits in search of an innovative design that leads to better safety of passengers, increased driver comfort and an ergonomically designed quick response system. Important parameters which decide the braking performance of a vehicle are deceleration, stopping distance, stopping time. The aim of this research is to propose a new automotive brake rotor for a sport bike which will increase the maximum deceleration of the vehicle thus reducing stopping distance and time. Also its performance is compared with conventional disc rotor and proposed perimetric rotor design to evaluate and compare the stress and deformation pattern of the rotors under extreme loads. Finite element analysis is employed for both static structural and thermal stress analysis. Temperature load was applied on the rotors and the temperature distribution was analysed considering cooling parameters (convection). An attempt is also made to suggest a best combination of material and thickness used for disc brake rotor, which yields a low temperature variation across the rotor, less deformation, and minimum von mises stress.

**Index Terms** — Disc Brake, Brake Force, Deceleration, CAD, ANSYS, Optimization, Structural Analysis, Thermal Analysis

## 1 INTRODUCTION

An automotive brake is a device by means of which artificial frictional resistance is applied to moving vehicle, in order to slow it down or stop it completely. In the process of performing this function, the brakes absorb kinetic energy of the moving vehicle which is then dissipated in the form of heat to the surrounding. The brake system should have following basic requirements:

- It must be strong enough to stop the vehicle with in a minimum distance.
- The driver must have proper control over the vehicle during braking and vehicle must not skid.
- The brakes must have well anti fade characteristics i.e. their effectiveness should not decrease with constant prolonged application.

### 1.1 Disc Brake

In Disc brake Fig. 1. a force is applied by driver on brake pedal which is then multiplied by lever ratio and pressurize the fluid inside master cylinder. This pressure is then transferred to slave cylinder at caliper via. brake line. Slave cylinder then apply force on two brake pads on opposite sides of a spinning rotor attached to the wheel hub. Disc brake pads are mounted in a caliper that sits above the spinning rotor.

This forces the inboard pad against the rotor, and the reaction force moves the outboard side of the caliper inward so that both pads grip the rotor.

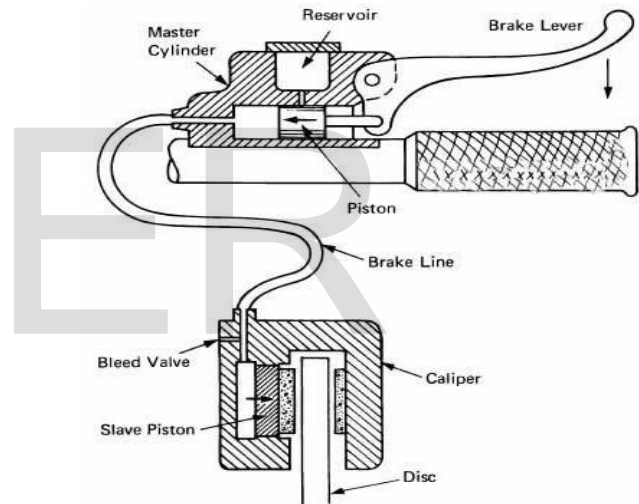


Fig. 1. Components of a disc brake system

All the friction components of a disc brake are exposed to the airstream, which helps to cool the brake parts and maintain braking effectiveness during repeated hard stops from high speeds.

Because their advantages far outweigh any disadvantage, disc brake have become the universal choice as the front brakes on all cars, sport bikes and light trucks built since the 1970s.

## 2 NEED OF A NEW SYSTEM

In conventional braking systems, the brake-disc is mounted onto the inner rims of the wheels via the mounting holes. These mounting holes are located near the inner diameter of the brake-disc. During braking, when the brake callipers clamp onto the brake-disc, the braking torque is transmitted from the brake-disc to wheel-hub at its inner diameter where

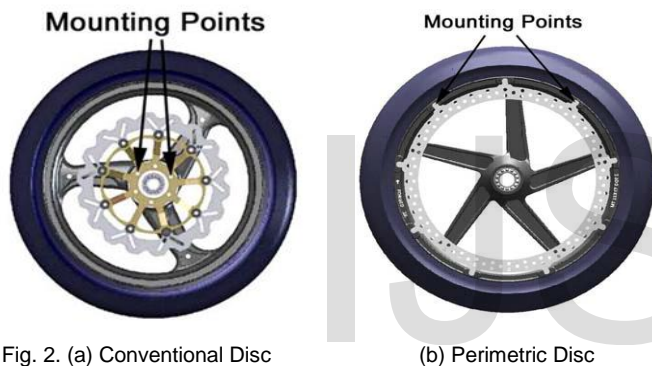
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the mounting holes are located. The braking torque is then transferred from the rigidly fixed mounting holes to the wheel-hub at its inner diameter, through the spokes to the rim of the wheels and eventually to the tires.

Now, for a sport bike, performance is of main consideration. To increase the deceleration with a conventional system two pair of disc-caliper is required at the front wheel which will increase weight, complexity and total cost of the system and thus bike as a whole. So the need is to come up with a cost effective system without compromising with performance.

### 2.1 Perimetric Disc Brakes

A new perimetric brake-disc, Fig. 2. recently developed works by mounting the brake-disc onto the outer rim of the wheel. In the new perimetric brake system, the braking torque is directly transmitted from the brake-disc to the outer rim of the wheel and finally to the tyres. So when the same force is applied onto the brake-discs, the perimetric brake system is able to produce a larger braking torque due to its larger diameter as compared to a conventional brake-disc.



In the Perimetric brake system, the braking torque is transferred directly to the outer rim, thus the braking stresses are eliminated from the spokes. Therefore, the rim can be lighter as it only needs to withstand the normal riding forces associated with the motorcycle.

With the Perimetric brake system’s unique design, it allows better utilization of braking force and a light weight design for the wheel; thus showing great potential to replace the conventional braking system.

### 3 OBJECTIVE OF PRESENT STUDY

The present investigation is aimed to:

1. Design a perimetric disc and compare its performance with conventional disc brakes in terms of deceleration, stopping distance and time.
2. Perform static stress and thermal analysis on three selected material for disc and compare results.

### 4 DESIGN SPECIFICATION OF BRAKE SYSTEM

For this project study Yamaha R15 sport bike is chosen. Reason being it is one of the most popular and affordable sport bike in India and is easily available for study.

TABLE 1 shows the specifications of the bike which will be required for determining performance parameters. These will be same for conventional as well as newly designed perimetric brake system.

TABLE 1 Specification : Yamaha R15 version 2.0

	FRONT	REAR
M/C Diameter	11 mm	12.70 mm
M.C motion ratio factor	0.8	0.8
Caliper Diameter	25.40 mm (Dual Piston)	32.00 mm (Single Piston)
Caliper motion ratio factor	0.7	0.7
Pedal Ratio	5 : 1	5 : 1
Pedal Force	90 N	100 N
Tire Diameter	575.79 mm	613.79 mm
Centre of Gravity (horizontal)	737.69 mm	607.31 mm
Centre of Gravity (height)	603.90 mm	
Wheelbase	1345 mm	
Curb Weight of vehicle	136 kg	
Driver Weight	60 kg	

### 4.1 Conventional Disc Brakes

- Disc Diameter = 267 mm (Front)  
= 220 mm (Rear)
- Pad Rubbing Diameter = 265/210 mm (Front)  
= 200/138 mm (Rear)

### 4.2 Perimetric Disc Brakes

- Disc Diameter = 393.97 mm (Front)  
= 220 mm (Rear)
- Pad Rubbing Diameter = 393.97/338.97 mm (Front)  
= 200/138 mm (Rear)

### 5 CALCULATING PERFORMANCE

Driver applies a force on the brake pedal during braking, which is known as pedal force (Fp). This pedal force when multiplied by pedal ratio (Pr)(mechanical advantage) gives force on the piston of master cylinder (Fmc). Motion ratio factor is used to incorporate friction losses.

$$F_{mc} = F_p \times P_r \times 0.8 \dots\dots (0.8 = M.C \text{ motion ratio factor})$$

$$Wt. \text{ transfer} = (a \times mass \times C.O.G \text{ (ht)} / wheel \text{ base})$$

Pressure developed (Pmc) in the brake fluid due to pedal force is calculated by dividing Fmc by cross sectional area of master cylinder piston (Amc).

Now as per the specification in TABLE 1 and given formula, performance parameters are calculated for both conventional disc brakes and newly designed perimetric disc brakes and comparison is done as shown in TABLE 2.

$$A_{mc} = \pi/4 \times (dm)^2$$

$$P_{mc} = F_{mc} / A_{mc}$$

This pressure is then transmitted by brake fluid to piston of caliper. Force at the caliper piston (Fcal) is thus calculated by multiplying pressure (Pmc) to cross sectional area of caliper piston (Acal).

$$A_{cal} = 2(\pi/4 \times (dc)^2) \dots\dots\dots (2 \text{ pistons})$$

$$F_{cal} = P_{mc} \times A_{cal} \times 0.7 \dots\dots(0.7 = caliper \text{ motion ratio factor})$$

Since there are 2 pads, clamping force by pads onto disc (Fclamp) is twice of the (Fcal).

$$F_{clamp} = 2 \times F_{cal} \dots\dots\dots (2 \text{ pads})$$

Friction force (Ffriction) between pads-disc surface is obtained by multiplying Fclamp to coefficient of friction between pad-disc surface (μ).

$$F_{friction} = \mu \times F_{clamp} = \dots\dots\dots (\mu \text{ pads} = 0.35)$$

Braking torque (TB) is calculated by multiplying Ffriction to mean effective radius of disc (Rdisc).

$$Braking \text{ torque (TB)} = F_{friction} \times R_{disc}$$

Braking Force (FB) which the tire applies on ground is obtained by dividing Braking torque (TB) with Radius of tire (Rtire).

$$F_B = T_B / R_{tire}$$

$$Total \text{ Braking Force} = F_{Br} + F_{Bf} \dots\dots\dots (front + rear)$$

Total braking force when divided by mass of vehicle gives maximum deceleration (a)

$$a = Force / Mass$$

Now, assuming deceleration as constant over the entire braking period stopping distance and time is calculated using newton's equations of motion.

Stopping Time (100kmph - 0):

$$v = u + at$$

$$t_s = u/a \dots\dots\dots (v = 0)$$

Stopping distance:

$$s = u.t_s + \frac{1}{2} a.t_s^2$$

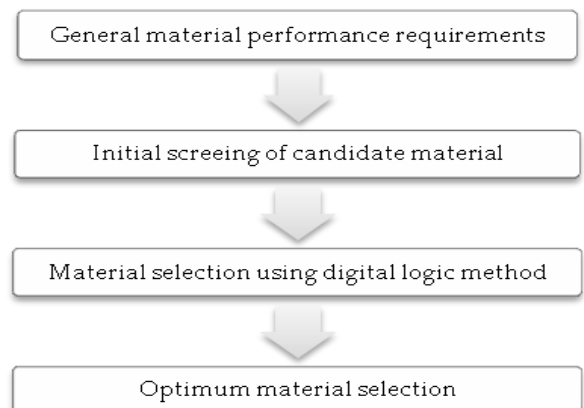
TABLE 2 Performance parameters of Brake system

	CONVENTIONAL	PERIMETRIC
Braking Force	1116.54 N	1536.73 N
Deceleration	5.69 m/sec <sup>2</sup>	7.84 m/sec <sup>2</sup>
Stopping time (100-0 kmph)	4.88 sec	3.54 sec
Stopping distance (100-0 kmph)	67.76 m	49.18 m
Stopping time (50-0 kmph)	2.44 sec	1.77 sec
Stopping distance (50-0 kmph)	16.95 m	12.30 m
Weight transfer	53.46 kg	70.33 kg
Brake bias	72 : 28	81 : 19

Above table clearly shows that if we replace front conventional disc with perimetric disc there will be 38% increase in braking force which results into increase in deceleration by 38%. Stopping time and stopping distance equally decreases by 27%. For a performance bike these percentage changes are quite significant.

### 6 MATERIAL SELECTION FOR BRAKE DISC

In this investigation, the stages of material selection method are shown below using a flow chart.



**6.1 General Material Performance Requirements**

Structural materials used in brakes should possess some combination of properties such as good compressive strength, higher friction coefficient, wear resistance, light weight, good thermal capacity and should be economically viable.

**6.2 Initial Screening of Candidate Material**

Following section will describe the potential candidate materials which can be used for brake rotor application. Based on the above mentioned properties, potential candidate materials for automotive brake disc are selected as:

1. Gray cast iron (GCI)
2. Ti-alloy (Ti-6Al-4V)
3. 7.5 wt% WC and 7.5 wt% TiC reinforced Ti-composite (TMC)
4. 20% SiC reinforced Al-composite (AMC 1)
5. 20% SiC reinforced Al-Cu alloy (AMC 2) [AL 2014 T6]
6. Stainless Steel AISI Type 420

**6.3 Material Selection using Digital Logic Method**

The digital logic method can be employed for the optimum material selection using ranking. As a first step, the property requirements for a brake rotor were determined based on previous discussion. The number of properties (N) and the total number of decisions, i.e.  $N(N - 1)/2 = 10$ , TABLE 3. The weighting factor for each property, which is indicative of the importance of one property as compared to others, was obtained by dividing the numbers of positive decisions given to each property by the total number of decisions.

TABLE 3 Decisional numbers

Decision Numbers										
	1	2	3	4	5	6	7	8	9	10
Compressive Strength	0	0	0	1						
Friction Coefficient	1				1	0	1			
Wear Resistance		1			0			1	1	
Thermal Capacity			1			1		0		0
Specific gravity				0			0		0	1

TABLE 4 Weighing Factors for Brake Disc

Property	Positive Decision	Weighing Factor
Compressive Strength	1	0.1
Friction coefficient	3	0.3
Wear resistance	3	0.3
Thermal capacity	2	0.2
Specific gravity	1	0.1
Total	10	1.0

From TABLE 4 it is found that Friction coefficient and wear resistance have the highest weighting factors followed by thermal capacity, whereas the least important properties are compressive strength and specific gravity hence, obtained lower weighting factor.

**6.4 Optimum Material Selection**

In order to complete the Digital Logic method, the next step is to scale the properties of the materials based on their respective weighting factor and the scale value. TABLE 5 shows the required properties of materials under consideration. For the present application, materials with higher compressive strength, friction coefficient and thermal capacity are more desirable and highest value is rated as 100. Their scaled values are calculated using the following equation.

$$\text{Scaled Value of Property} = \text{Numerical Value of Property} * 100 / \text{Maximum Value in the List}$$

Since lower wear rate and specific gravity are desirable for the automotive brake disc, therefore, their lowest value is considered as 100 and scaled values are calculated using equation.

$$\text{Scaled Property} = \text{Minimum Value in the List} * 100 / \text{Numerical Value of Property}$$

Finally, material performance index ( $\gamma$ ) is calculated as:

$$\gamma = \sum \beta * \Omega$$

Where  $\beta$  is the Scaled Property (TABLE 6) and  $\Omega$  is the Weighing Factor (TABLE 4).

TABLE 5 Properties of candidate materials for brake disc

Material	1	2	3	4	5
	Compressive Strength (MPa)	Friction Coefficient ( $\mu$ )	Wear Rate ( $\times 10^{-6}$ mm <sup>3</sup> /N/m)	Specific Heat (J/kg <sup>o</sup> C)	Density ( $\rho$ ) (g/cm <sup>2</sup> )
GCI	1290	0.41	2.36	460	7.2
Al 2014 T6	761	0.44	2.91	920	2.8
Ti-6Al-4V	1070	0.34	246.3	580	4.42
TMC	1300	0.31	8.19	510	4.68
AMC	406	0.35	3.25	980	2.7
SS420	1480	0.43	2	460	7.6

TABLE 6 Scaled value of properties of each material and corresponding Performance Index

Scaled Properties						Performance Index ( $\gamma$ )
1	2	3	4	5		
GCI	87	93	84	47	38	75
AMC2	51	100	69	93	96	84
Ti6Al-4V	72	77	0.8	59	61	48.44
TMC	87	70	24	52	57	53
AMC	27	79	61	100	100	74.7
SS420	100	97	100	47	36	82.1

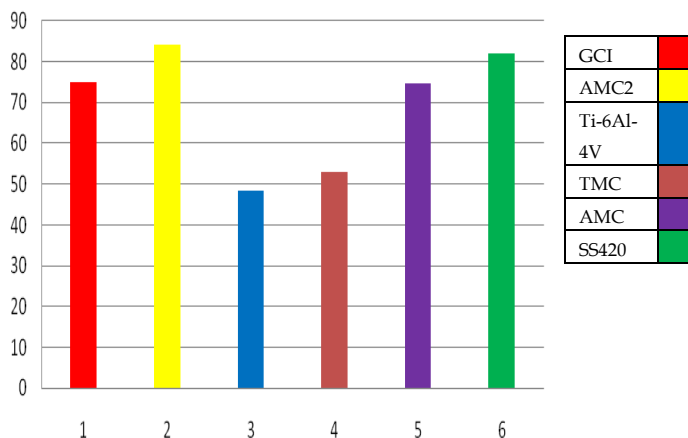


Fig. 3. Performance index comparison

On the basis of Performance Index (Fig .3.), we have chosen Grey Cast Iron (GCI), AL 2014 T6 (AMC 2) and Stainless Steel AISI Type 420 as our material of highlight.

## 7 FINITE ELEMENT ANALYSIS

The analysis and comparison study is done through modelling and simulation using the respective software: CATIA V5 and ANSYS Workbench 14.5. The brake-discs is simulated under three different loading conditions: two static tests and one thermal analysis. The two static cases are Torsional strength simulation and Lateral strength simulation. A Residual stress simulation is carried out to find temperature distribution and induced stress and deformation.

The primary focus is to simulate the performances of the brake-discs using ANSYS which is developed based on the finite element method (FEM). In order to obtain accurate and reliable results for the simulations, the meshing of the brake-discs must be properly carried out. Meshing is performed to discretize the geometry created into small pieces called elements.

The elements used for the mesh of the model are tetrahedral three-dimensional. Now, there are various element sizes available in ANSYS. Selection of element size should be done so as to get optimum results in lesser computational time. Lesser is the element size, more accurate will be the results. In our study, torsional simulation is performed on conventional disc for different element size ranging from 1 - 10 mm and induced deformation is checked.

The following graph is obtained:

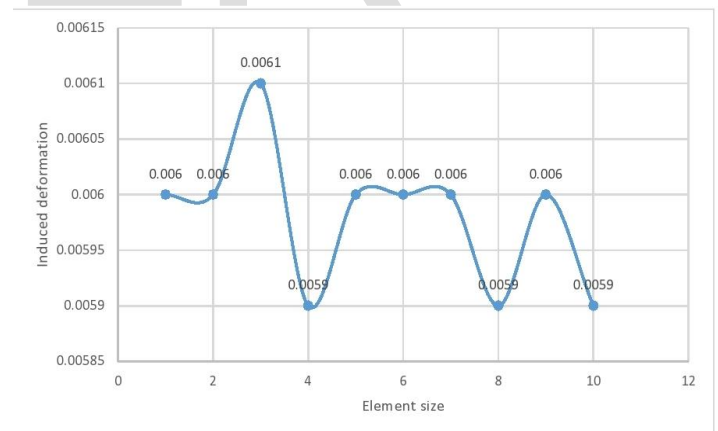


Fig. 4. Graph plot of element size vs induced deformation

From Fig .4. it is observed that the variation in induced deformation is negligible for element size of 5, 6 and 7mm. So 5 mm is selected.

### 7.1 Static Test - Lateral Strength Simulation

In this simulation, a lateral force is exerted on the disc to check for any distortion in the disc. This test studies the behaviour of the brake-disc when it encounters lateral loading during usage, for example a person kicking the disc, stones or other

similar materials coming into contact with the brake-disc during riding. If it is distorted, the contact surface between the brake pad and brake-disc will become uneven, which may result in vibrations that affect the performance of the brake.

Assumptions:

1. The lateral load is exerted onto the brake-disc through the brake pad.
2. A force of 1000 N is enough to test the behaviour of the brake-disc under lateral loading. (The same value is used in industry standard tests.)
3. The force is purely transmitted in the lateral direction.

The nodes within a rectangular shape on the surface of the brake-disc are chosen, simulating the dimensions of a brake pad. A total force of 1000 N is applied in the axial direction on the nodes. The mountings holes are then constrained to give zero displacement and the solution is then obtained.

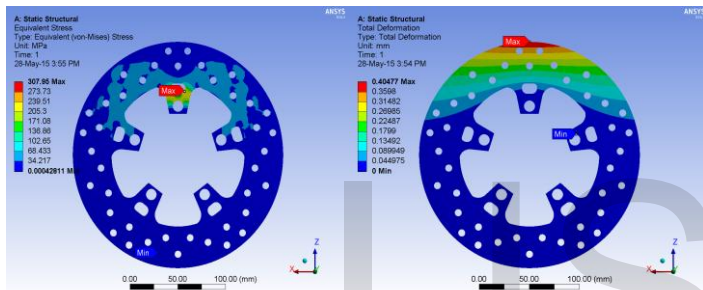


Fig. 5. Stress and deformation in rear conventional disc (SS-420)

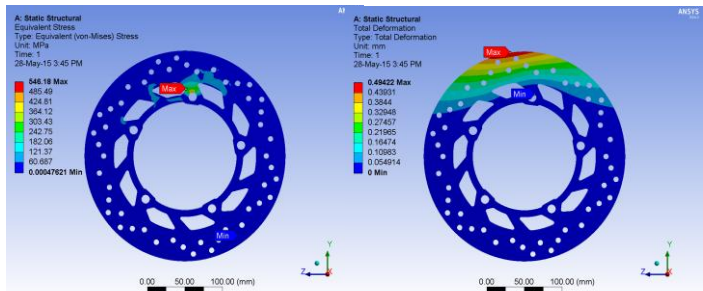


Fig. 6. Stress and deformation in front conventional disc (SS-420)

SS-420 is the material used in conventional brake rotors. Now, static lateral analysis is carried out for perimetric disc for 3 different materials i.e. Stainless steel, Grey cast iron and AMC 2 and multiple iteration is done to find the best combination of material and thickness that can sustain forces within a safe deformation limit. And finally 5 mm for Stainless steel (SS-420), 6 mm for Cast iron and 6.5 mm for Aluminium 2014 T-6 is selected. Results are as follows:

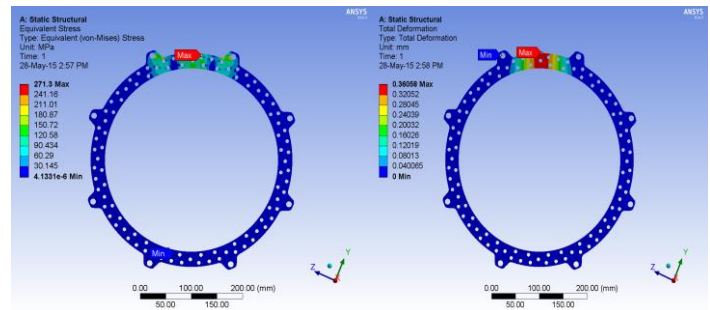


Fig. 7. Stress and deformation in perimetric disc of SS-420

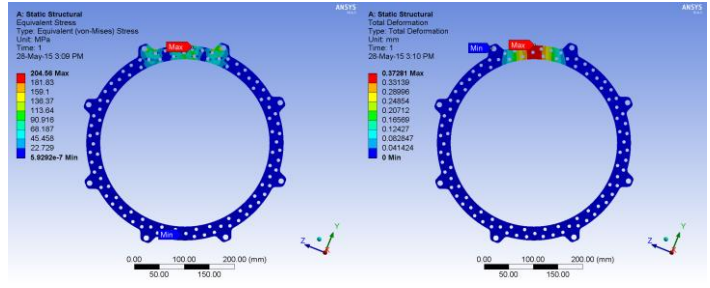


Fig. 8. Stress and deformation in perimetric disc of Grey cast iron

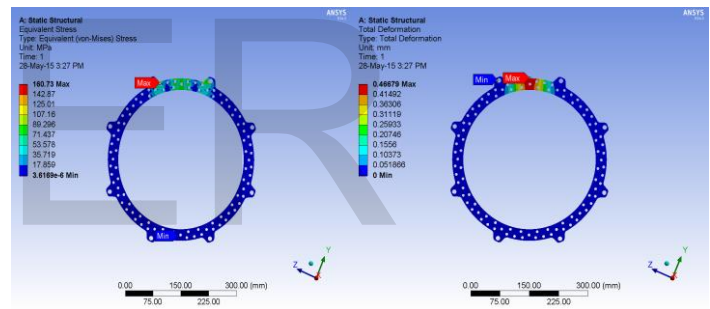


Fig. 9. Stress and deformation in perimetric disc of Al 2014 T-6

For the Perimetric brake-disc, the stress concentrates on the piercing holes and adjacent mounting holes where the lateral load is applied.

### 7.2 Static Test - Torsional Strength Simulation

In this simulation, a torsional force is exerted on the brake-disc, simulating the torsional force that a brake-disc experiences during hard braking. This is similar to actual braking conditions, when the rider comes to an emergency stop while travelling at high speed. It is important to ensure that the brake-disc is able to withstand the high torsional stress generated due to the brake-pads clamping down on the spinning brake-disc. In this test, it is important to check for any distortion, especially near holes (mounting or piercing holes), where there might be stress concentrations during braking.

Assumptions:

- 1) The maximum torque is concentrated on a single brake pad that is in contact with the brake-disc.

2) The maximum weight of the motorcycle is the same for the perimetric brake-disc and the conventional brake-disc: which is equal to the cumulative weight of the motorcycle and driver.

3) Maximum friction coefficient between the wheels and the ground is assumed to be 0.8.

The nodes within a rectangular shape on the surface of the brake-disc are chosen, simulating the dimensions of a brake pad. A torsional force of 1194 N and 773.84 N is applied on the perimetric and conventional brake-disc (onto the chosen nodes) respectively. This value of torsional force is calculated based on the maximum deceleration that the motorcycle would experience during hard braking. The mountings holes are then constrained to give zero displacement and the solution is then obtained.

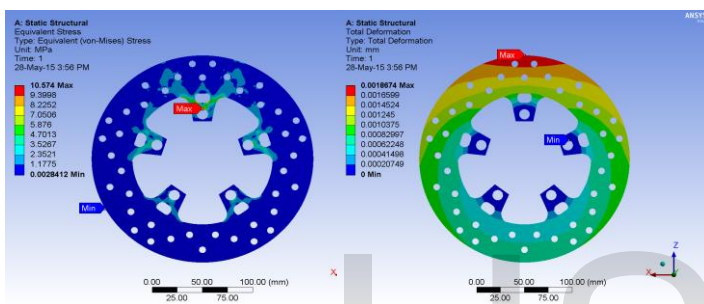


Fig. 10. Stress and deformation in rear conventional disc (SS-420)

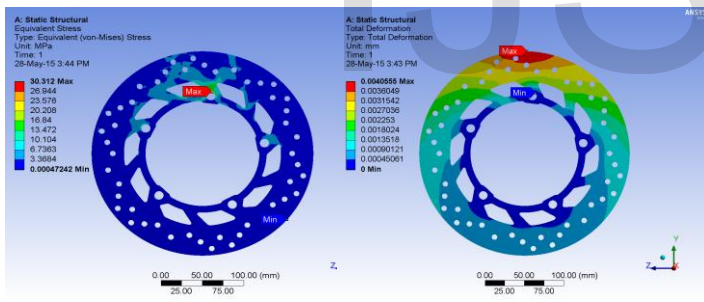


Fig. 11. Stress and deformation in front conventional disc (SS-420)

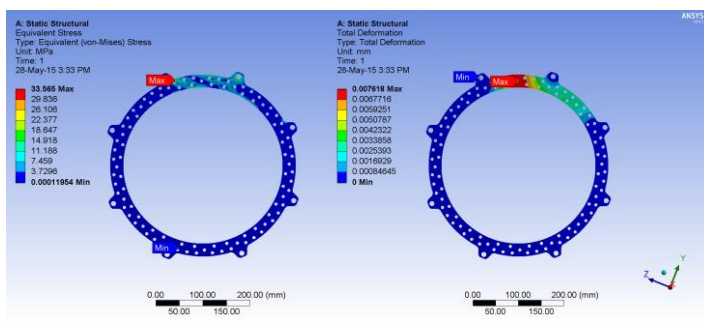


Fig. 12. Stress and deformation in perimetric disc of SS-420

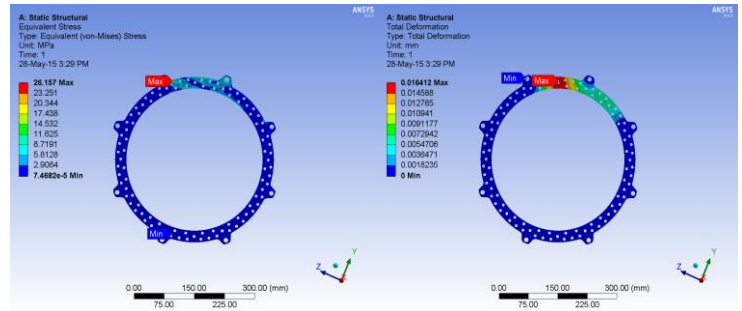


Fig. 13. Stress and deformation in perimetric disc of Al 2014T-6

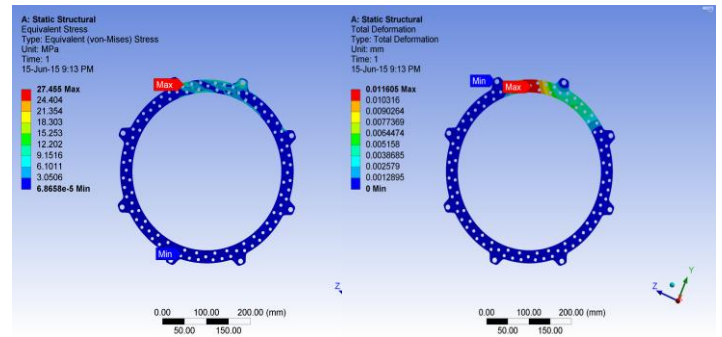


Fig. 14 Stress and deformation in perimetric disc of Grey Cast Iron

In the conventional brake-disc, the stress concentrates near the mounting holes. For the Perimetric brake-disc, the stresses concentrate around the area where the torsional load is applied; especially on the adjacent piercing holes.

### 7.3 Residual Stress Simulation

The main challenge of this project is to study the behaviour of the Perimetric brake-disc under residual stress, and compare it to the conventional brake-disc. Special attention has to be given to this test as the Perimetric brake-disc is constrained by the rim (by the mounting holes at the outer diameter of the disc). Hence during the expansion and contraction of the brake-disc, there may be problem of warping or stress concentration. In conventional brake-disc system, there is more allowance for expansion and contraction of the disc, as it is only fixed to the rim at its inner diameter. The severe thermal distortion of a disc can affect important brake system characteristics such as system response and brake judder propensity.

Assumptions:

- 1) All heat generated by the contact of the brake pad and the brake-disc goes to the rotor.
- 2) All the kinetic energy at maximum velocity is converted to heat energy.
- 3) The maximum weight of the motorcycle is the same for the perimetric brake-disc and the conventional brake-disc; which is equal to the cumulative weight of motorcycle and driver.

To validate the present models, firstly thermal analysis of disc brake is performed to find temperature variation across the disc surface. A braking application to stop vehicle from 100 kmph is taken into consideration. The cycle composed of braking time of 4.88 sec for conventional system and 3.54 sec for perimetric system. The angular velocity is assumed to decay linearly and finally become zero at 4.88 and 3.54 sec respectively for conventional and perimetric system.

Maximum Temperature generated through the surface of brake disc is calculated using thermal heat transfer and single stop temperature rise equations -

$$K.E = \frac{1}{2} * m * v^2 \quad \text{and} \quad R.E = 0.3 * K.E$$

$$T.E = K.E + R.E$$

$$\text{Surface area of brake disc } (A_d) = \pi / 4 (d_o^2 - d_i^2)$$

$$\text{Braking power } (B.P) = T.E / t$$

Where, 't' is the braking time

Now, braking power in the front disc is calculated by multiplying the dynamic brake weight transfer ratio in the front to the total braking power -

$$B.P_f = D.B.R_f * B.P$$

Hence, Heat flux through the disc surface is calculated by

$$Q_f = B.P_f / A_d$$

Now, maximum temperature generated through the frictional force in brake disc in single stop temperature rise is calculated by -

$$T_{max} = (0.527 * Q_f * \sqrt{t}) / (\sqrt{\rho * c * k}) + T_{amb}$$

Here:  $\rho$  = density;  
 $c$  = specific heat;  
 $k$  = thermal conductivity

Value of  $\rho$ ,  $c$ , can be obtained from TABLE 5.

$k$  = 154, 35 and 52 W/mK for Al 1014 T-6, SS-420 and Grey Cast Iron respectively

Using above formula maximum induced temperature along the surface is calculated which is shown in TABLE 7 .

The maximum temperature calculated along with the convection heat transfer coefficient is provided as input to the thermal analysis of brake rotor and the temperature distribution in complete brake disc rotor is found out. The value of convection heat transfer coefficient is taken as a simplified case of ANSYS 16.0 (stagnant air - horizontal cylinder) which is temperature dependent as shown in Fig .15.

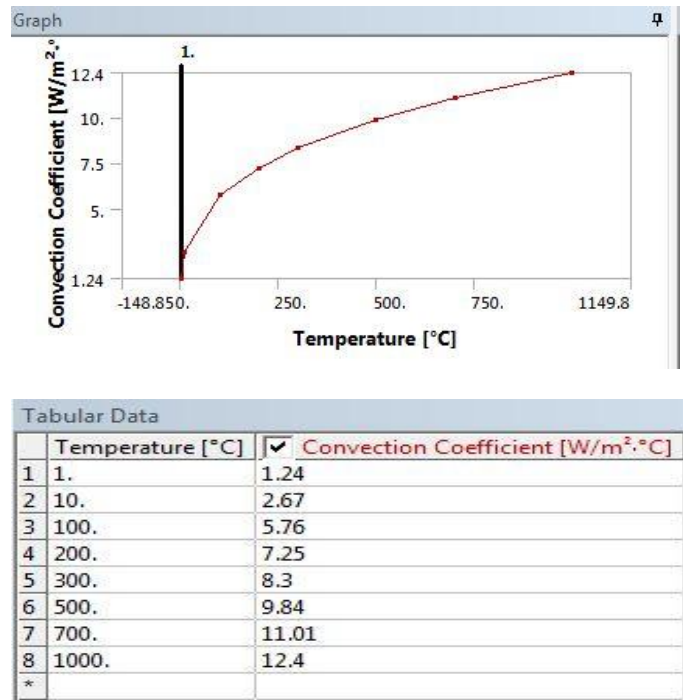


Fig. 15. Convection coefficient temperature dependent values

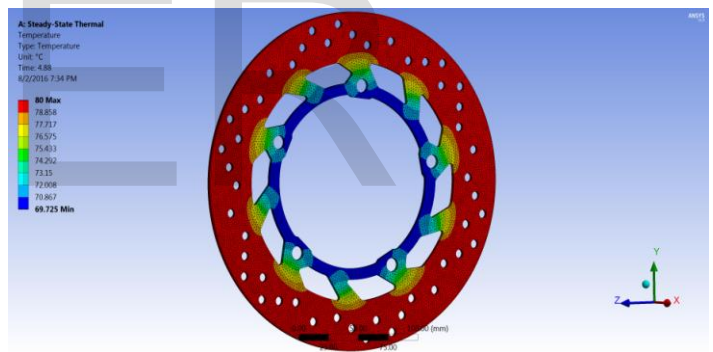


Fig. 16. Temperature distribution of conventional disc (SS-420)

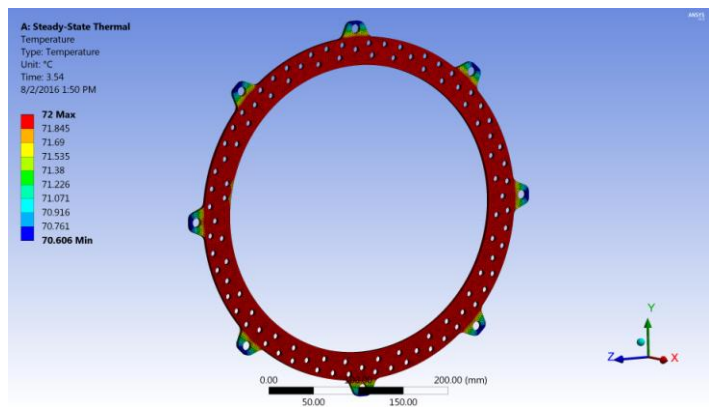


Fig. 17. Temperature distribution of Perimetric disc (SS – 420)



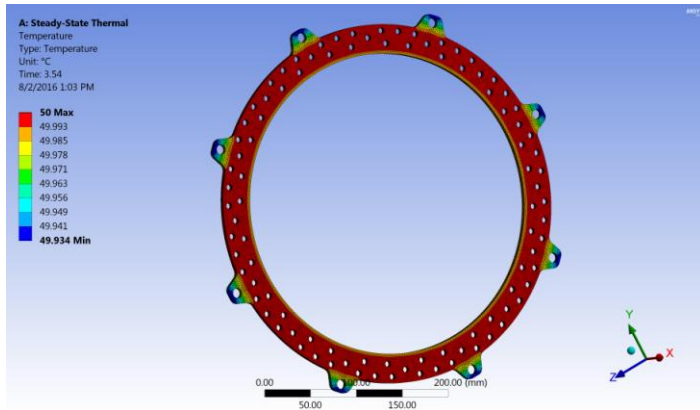


Fig. 18. Temperature distribution of Perimetric disc (Al 2014 T-6)

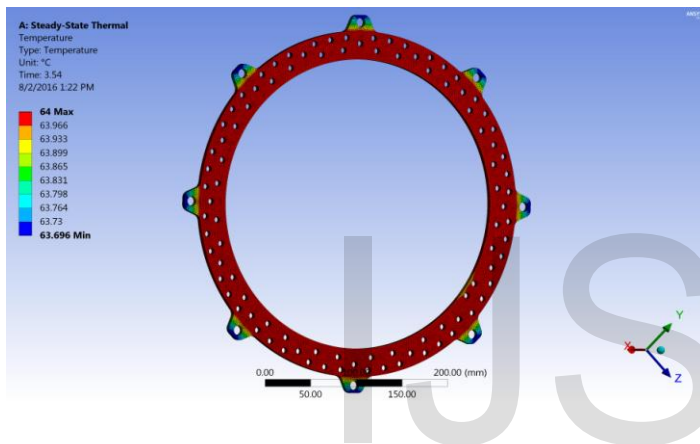


Fig. 19. Temperature distribution of Perimetric disc (Grey Cast Iron)

TABLE 7 Distribution of temperature values of Conventional and Perimetric disc of different materials

	Maximum Temperature (Celsius)	Minimum Temperature (Celsius)
Front Conventional Rotor (SS – 420)	80	69.725
Perimetric Rotor (SS – 420)	72	70.606
Perimetric Rotor (GCI)	64	63.696
Perimetric Rotor (Al 2014 T6)	50	49.934

For both types of brake-discs, the maximum temperature concentrates around the area where the load is applied; especially on the adjacent piercing holes, Fig .16 - .19. Now the temperature distribution of rotor is provided as input

to the static structural part of Residual / Thermal stress analysis and the resultant von-mises stress and deformation due to temperature change are found out. Mounting holes are provided zero displacement in all the directions.

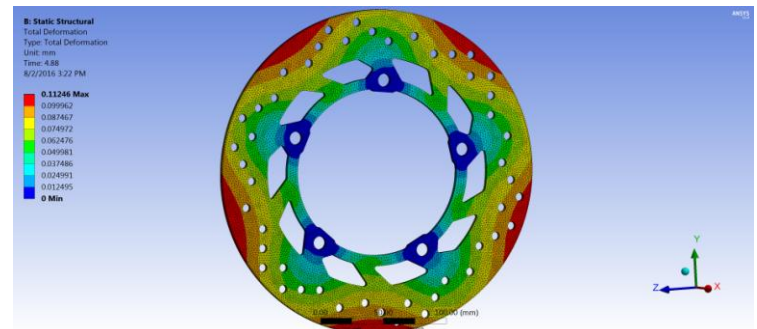


Fig. 20. Deformation and Residual stress in front conventional disc (SS – 420)

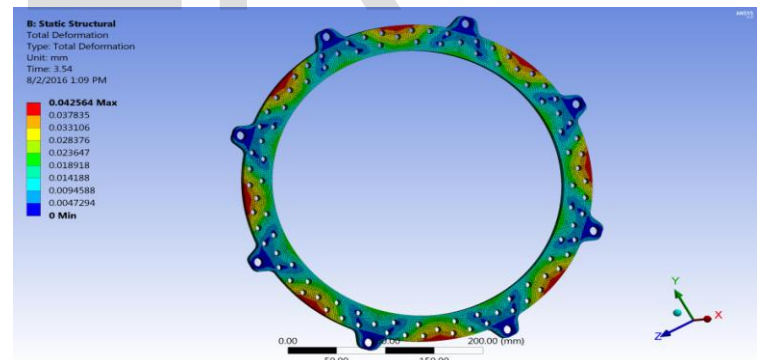


Fig. 21. Deformation and Residual stress in perimetric disc (Al 2014 T-6)

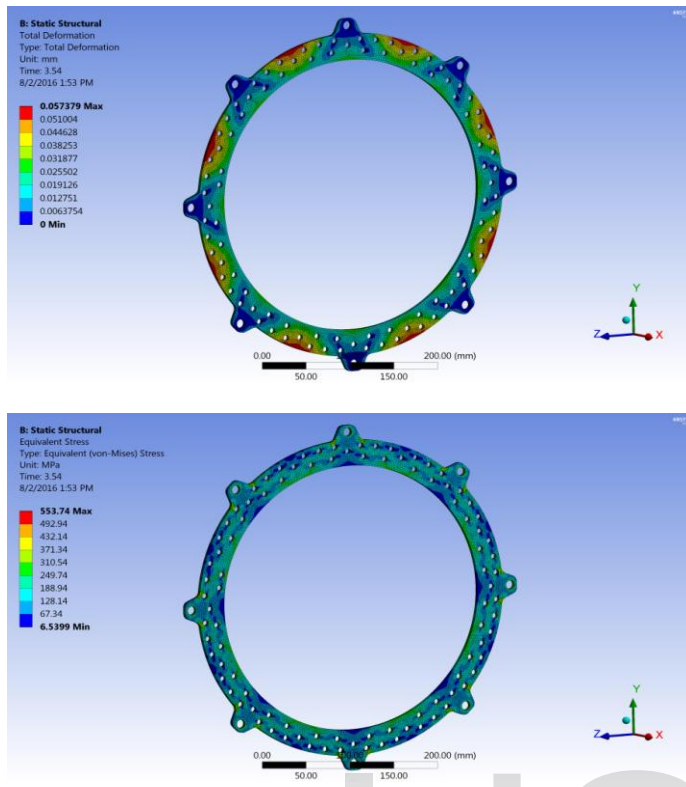


Fig. 22. Deformation and Residual stress in perimetric disc (SS – 420)

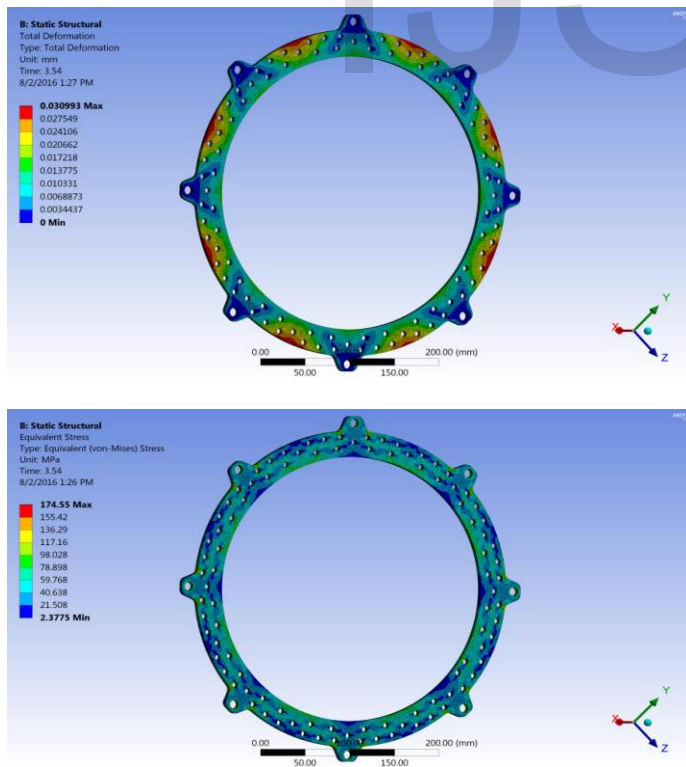


Fig. 23. Deformation and Residual stress in perimetric disc (Grey Cast Iron)

The results of the analysis for all three materials is tabulated and shown in TABLE 8 of Appendix.

### 8 CONCLUSION

For Torsional strength simulation the maximum values of the Von Mises Stress in perimetric disc is quite comparable to that in the conventional brake-disc and is much lower than the yield stress of the material, indicating that yielding is unlikely to occur during hard braking of the motorcycle.

For Lateral strength simulation maximum value of Von Mises Stress is about 50% lower in the perimetric brake-disc as compared to conventional disc. Again, the maximum stresses are much lower than the yield stress of the material, indicating that bending or yielding is unlikely to occur in the perimetric brake-disc during lateral loading.

For the perimetric brake-disc, it is noted that the maximum Residual stress value due to temperature distribution is less than the yield stress value. Also the values of both perimetric and conventional are quite comparable to each other. Hence the design is safe.

From the above study it can be concluded that perimetric brake system is better than the conventional system as it enhances the following properties as compared to the conventional system:

- Increases the deceleration of the vehicle.
- Decreases the stopping distance of the vehicle.
- Decreases the stopping time of the vehicle.
- The design is structurally and thermally safe.

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**APPENDIX**

TABLE 8 Comparison of stress and deformation values of Conventional and Perimetric disc of different materials

		Stainless Steel (SS-420)		Cast Iron		Al 2014 T-6	
		Stress (MPa)	Deformation (mm)	Stress (MPa)	Deformation (mm)	Stress (MPa)	Deformation (mm)
<b>Torsional Strength Simulation</b>	Conventional	30	0.004				
	Perimetric	33	0.0076	27	0.0116	26	0.0164
<b>Lateral Strength Simulation</b>	Conventional	546	0.4942				
	Perimetric	271	0.3605	204	0.3728	160	0.4667
<b>Residual Strength Simulation</b>	Conventional	507.87	0.11246				
	Perimetric	553.74	0.057379	174.55	0.03099	152.73	0.042564
<b>Yield Stress (Mpa)</b>		<b>600</b>		<b>200</b>		<b>414</b>	