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Structural Deformation & Temperature Analysis of An Automotive Disc Brake

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Abstract – *The research study focuses on analyzing the temperature of rotors made from different materials used in the front disc brake of the APACHE RTR 160 motorbike. The objective is to evaluate their performance under severe or hard braking conditions with continuous braking until the vehicle comes to a stop, assuming no wheel locking occurs. The materials considered for the disc brake rotor are Grey Cast Iron, Ductile Cast Iron, Aluminium Metal Matrix Composite, and Martensitic Stainless Steel. The investigation utilizes a model developed using ABACUS software and performs dynamic temperature displacement explicit analysis. The simulation results are compared analytically for a solid rotor. The study aims to identify the most suitable material for the disc based on factors such as cost, maximum temperature reached, disc mass, heat distribution, temperature distribution, hot spots, and temperature gradients. During severe braking, there is a significant rise in temperature due to the short duration of braking, resulting in higher braking power and rapid heat generation. The majority of the braking power is absorbed by the disc brake system, with the thermal energy mainly absorbed by the disc and pad surfaces in contact. This requires assessing whether the brake system materials and components can withstand the fast temperature increase without failure. The analysis follows the principle of energy conservation, where the energy possessed by the vehicle at the start of braking should be mostly absorbed by the disc brake system, with minimal heat loss to the surroundings due to the brief braking time. The study considers the distribution of heat energy between the disc and pad, considering properties such as thermal conductivity, density, specific heat, and thermal diffusivity. The findings indicate that the maximum temperature occurs at the frictional contact region of the disc and pad, with higher temperatures on the disc surface compared to the inner portions. The temperature in the contact region follows a saw-tooth pattern, increasing to a maximum value and then decreasing over time, consistent with experimental results.*

Keywords- *Disc brake, thermal analysis, energy conservation principle, ANSYS*

INTRODUCTION

The braking system is an important component of a car's safety features since it helps to maintain safe stopping distances and regulate speed. Due to its dependability, simplicity of maintenance, quicker response time, and higher heat dissipation capabilities compared to drum brakes, mechanical brakes, such as disc brakes, are frequently employed in modern cars. Disc brakes are hydraulic brakes that have a rotor or disc, caliper, brake pads, master and slave cylinders, brake fluid, rubber sealing rings, and hoses as their primary parts.

The brake pads are enclosed in the caliper, which is normally constructed of lightweight Aluminium alloy, and are tightly fastened to a stationary element of the car. The rotor or disc is connected to the wheels and spins with them. Retaining pins and spring plates are used to hold the brake pads, which are made of materials with low thermal conductivity, between the piston and the disc. When the brakes are applied, the brake fluid is pressurized by the master cylinder, which also serves as the brake fluid reservoir. Through hoses, the pressurised fluid is delivered to the slave cylinders inside the caliper, which subsequently activate the brake pads, causing friction against the disc and slowing the car down. Because of the frictional forces between the disc and the pads when braking, a lot of heat is produced. The disc surfaces normally absorb between 90 and 95 percent of the heat, and the distribution of heat relies on the material qualities of the disc and pads. To survive the highest temperatures produced by elements such vehicle weight, speed, fluid pressure, and the thermal qualities of the materials, it is essential to choose the right material for the disc.

The interaction of thermal and structural features of the disc brake system is investigated in this research study using a dynamic temperature displacement explicit analysis. The system's thermal behavior is important because high temperatures and thermal loads can cause brake fade (a decrease in the coefficient of friction) and a larger disc thickness variation (DTV). The analysis first considers heat transmission through the brake disc and pads, and as the braking duration lengthens, it also considers heat transfer to the nearby components by convection and radiation.

METHOLOGY

To analyze the structural deformation and temperature distribution of an automotive disc brake, a multi-stage study methodology was used. First, using software like ANSYS, a thorough CAD model of the disc braking system is produced. The model consists of parts such the disc, pads, solid support, redesigned hub piece, and back-plates that are made to resemble the Apache RTR 160, disc brake in terms of size and shape.

The next stage is to give each disc brake system component the right material qualities. According to desirable properties like a high coefficient of friction, thermal stability, wear resistance, thermal conductivity, and strength, the materials used to make the rotors are chosen. The appropriate components are then given the material characteristics gleaned from the literature review, including Young's modulus, thermal conductivity, specific heat capacity, Poisson's ratio, density, coefficient of thermal expansion, and thermal diffusivity.

Finite element analysis (FEA) software, especially ABACUS, is used to run numerical simulations. The simulations include an initial step and a dynamic temperature displacement step with a total runtime of 1.6 seconds. To accurately depict how the disc brake system behaves, boundary conditions, loads, and interactions are created based on actual operating circumstances. The pads and disc come into contact with each other through friction, simulating the braking system and guaranteeing optimal component interaction.

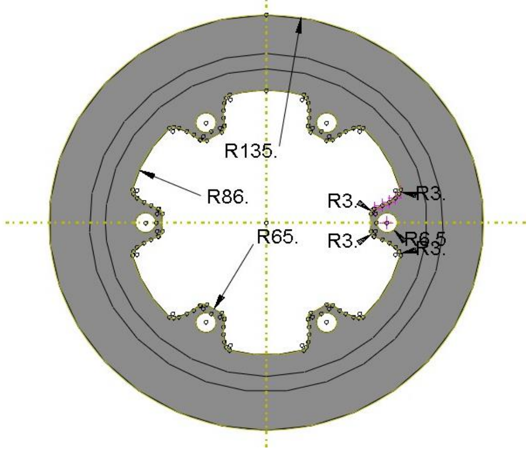
MATERIAL	Young's Modulus (E) GPa	Thermal Conductivity (K) W/m-k	Specific Heat Capacity (C) J/Kg-K	Poisson's Ratio (μ)	Density (ρ) Kg/m^3	Coefficient of Thermal Expansion (α)-(10^{-5})	Thermal diffusivity $\lambda = K/\rho C$ mm^2/s
DUCTILE CAST IRON	160	32	506	0.26	7200	1.1	8.81
GREY CAST IRON	135	48	460	0.27	7200	1.15	14.49
ALUMINIUM METAL MATRIX COMPOSITE	130	145	796	0.25	2900	1.2	62.81
MARTENSI-TIC STAINLESS STEEL	190	18	560	0.29	7800	1.1	4.12
PAD MATERIAL	70	3	1300	0.24	2100	1.2	.732

Thermal boundary conditions are used to analyze the temperature distribution within the disc braking system. For instance, the pads and disc are given a starting temperature of 35°C. In order to mesh the model, the appropriate elements are used, such as linked temperature displacement elements for the pads and disc and stress elements for the solid support, hub, and back-plates. at order to maximize computational efficiency, the mesh refinement is done to ensure a fine mesh at the contact region between the pads and disc while preserving a coarser mesh elsewhere. Both structural and thermal evaluations are included in the investigation and outcomes. To evaluate the loads and deformations the disc brake system experiences while braking, structural analysis is done.

While this is going on, the thermal analysis examines how temperature is distributed and how heat is transferred, considering things like conduction, convection, and radiation. Sensitivity assessments are carried out to determine how various factors, such as material characteristics, braking circumstances, and cooling techniques, affect the effectiveness of the disc brake system. The accuracy and dependability of the suggested approach are validated by comparing the numerical simulation results with analytical calculations. Discussion of the results and observations from the structural deformation and temperature investigation highlights the functionality of various rotor materials. The essential contributions and relevance of the established technique are summarized in the study's conclusion in order to better understand and analyze the behavior of automobile disc brakes. In addition, the study's limitations are acknowledged, and additional research topics are offered.

We are able to import or create geometry in the ABACUS software. The disc brake model was totally created within ABACUS for this investigation. An FEA model of the physical issue was converted into a geometric model using ABACUS tools. The disc, hub, brake pads, solid support, and back-plates are all part of it. Dimensions in millimeters are presented along with sketches of the disc, pad, hub, and combined disc-hub portion in 3D.

Fig. 1: Disc sketch with dimensions



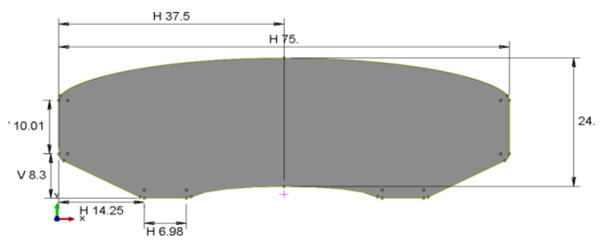


Fig. 2: Pad sketch with dimensions

DESIGN

Table 1: Parameters used in Designing.

Item	Values
Inner disc diameter (ID)	172mm
Outer disc diameter (OD)	270mm
Disc thickness	4.8mm
Angular velocity	68rad/s
Hydraulic pressure	0.3MPa
Stopping time	1.6s
Retardation rate	$9.375 m/s^2$
Vehicle mass	200Kg
Initial speed	15m/s
Area of disc	$0.032m^2$

Braking Energy Calculation:

1. Braking energy = $\frac{KmV_1^2}{2} = \frac{1.05 \times 200 \times 15^2}{2}$ = 23625 J

Taking 95% of the energy possessed by the vehicle to be converted into the thermal energy of the disc brake system.

2. Energy converted into thermal energy
= 0.95 × 23625 J = 22445.75 J

Front Brake Calculation:

1. As weight transfer occurs to the front axle during braking, more braking is applied to the front brakes for better control and stability. Taking 70% of the braking action on the front brake:
Energy dissipated at the front brakes = 0.7 × 22445.75 J = 15710.625J

Power Calculation:

With uniform deceleration, the following power calculations are made:

1. Total braking power at initial moment (t=0) =
$Kma (V_1 - a_t)$ = 1.05 × 200 × 9.375 × 15 = 29531.25W
2. Total average braking power = $\frac{KmaV_1}{2}$ = 14765.625W
3. Front brake initial braking power =
29531.25 × 0.95 × 0.7 = 19638.28W
4. Front brake average braking power = 9819.14W

Braking power on the disc:

1. Braking power at initial moment to be dissipated by the disc = 18656.357W
(95% goes to the disc as assumed)
2. Average braking power to be dissipated by the disc = 9328.18W
3. Braking power on one side of the disc surface (at t=0) = 9328.17W
4. Average braking power on one side of the disc surface = 4664.1W

Heat Flux:

Heat flux $= \frac{\text{Braking Power}}{\text{Swept area of disc}}$

$q_{(0)}$"(initial heat flux) =

$$\frac{\text{Initial braking power}}{\text{Swept area of disc}} = \frac{9328.17}{0.032} = 291505.98 W/m^2$$

q_0"(average heat flux) =

$$\frac{\text{average braking power}}{\text{Swept area of disc}} = \frac{4664.1}{0.032} = 145753.12 W/m^2$$

CONVECTION HEAT TRANSFER COEFFICIENT

The convection heat transfer coefficient is calculated as follows:

Re= Reynolds number= $\frac{\rho a V D_{out}}{\mu a}$

Where, $\mu = 1.77\times 10^{-5}\ \frac{kg}{m-s}$

ρ_a=1.2 kg/m^3 V=15 m/s D_{out}=0.270m

Re=$\frac{1.2 \times 15 \times .270}{1.77 \times 10-5}$=$2.74\times10^5$ (hence turbulent flow)

Therefore, convection heat transfer coefficient is calculated from $h_R=0.04(\frac{K_a}{D_{out}})\ Re^{0.8}$

Where, K_a=Thermal conductivity of air

$h_R=0.04\times(\frac{0.026}{0.27})(2.74\times10^5)^{0.8}$

h_R =86.6W/m^2-k

ANALYTICAL TEMPERATURE VALUES

In this part analytical temperature is calculated for the MSS material in the temperature profile with highest temperature region.
h_R =86.6W/m^{2}k
L=0.0024m
K=18W/m-k

$$\frac{h_R L}{K} = \frac{86.6 \times 0.0024}{18} = 0.011546$$

From the table of transcendental roots,
$\lambda_1 L = 0.11$

$$a_t = \frac{K}{pC} = \frac{18\ \times 3600}{7800\ \times 560} = 0.01483\ \frac{m2}{h}$$

t=0.76s=0.000211h

ts=1.6s=0.00044 h

$q_{(0)}''$=291505.98W/m^2

q_0''=145753.12W/m^2

$$\theta_0(z,t)\ =\ \frac{q''0}{\mathrm{hR}}\ [2(\frac{\theta \mathrm{hR}}{q''0}\ -\ 1)\sum_{n=1}^{\infty}\frac{sin(\lambda_n L)}{\lambda nL + \cos(\lambda nL)\sin(\lambda nL)}e^{(-a_t\lambda_n^2\,t)\,cos(\lambda_n Z)\,+\,1)}]$$

$\Theta_i= T_i-T_a=0$

Z=0.0024m

$\lambda_1=\frac{\lambda_1 L}{L}=\frac{0.11}{0.024}$=45.83

As discussed previously that only the n=1 term is sufficient to accurate results, so

$$\boldsymbol{\theta_0(z,t)}=\frac{145753.1}{86.6}[2(-1)\frac{\sin(0.11\ \times 57.3)}{0.11+\cos(0.11\times 57.3)\sin(0.11\times 57.3)}$$
$$e^{(-0.01483\times 45.83\ \times 0.000211)\cos(6.3)+1}]$$

$\Theta_0(z,t)=T_0(z, t)-T_\infty$= 21.47 K

After obtaining the above $\Theta_0(z,t)$ using constant heat flux, varying heat flux equation is applied to obtain the final temperature values.

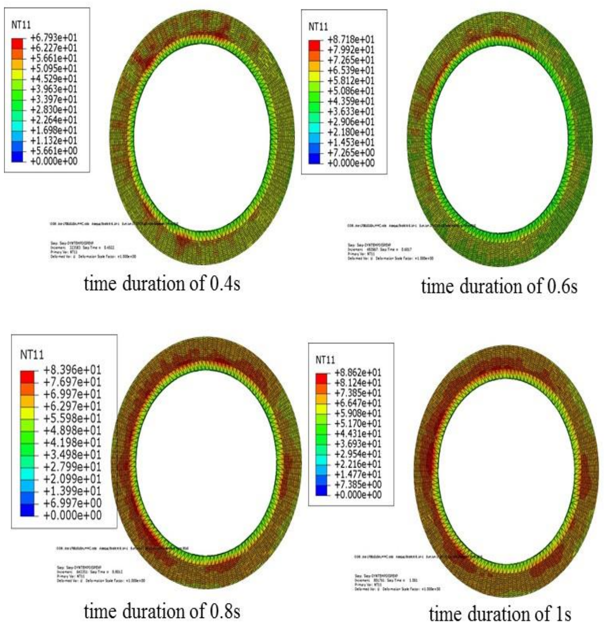
$$\Theta(z, t)=\frac{291506}{145753}\times 21.47-\frac{291506\times 3600}{1.6\times 86.6}[0.000211-2(\frac{\sin(6.3)}{0.11+\cos(6.3)\sin(6.3)})(\frac{\cos(0.3)}{0.01483\times 45.832})$$
$$(1-e^{(-0.01483\times 45.832\times 0.000211)})]$$

$\Theta(z,t)$=42.94+44.68K
$\Theta(z,t)=T(z, t)-T\infty$=87.62K
T(z, t)=273+35+87.6K=395.62K=122.6ºC

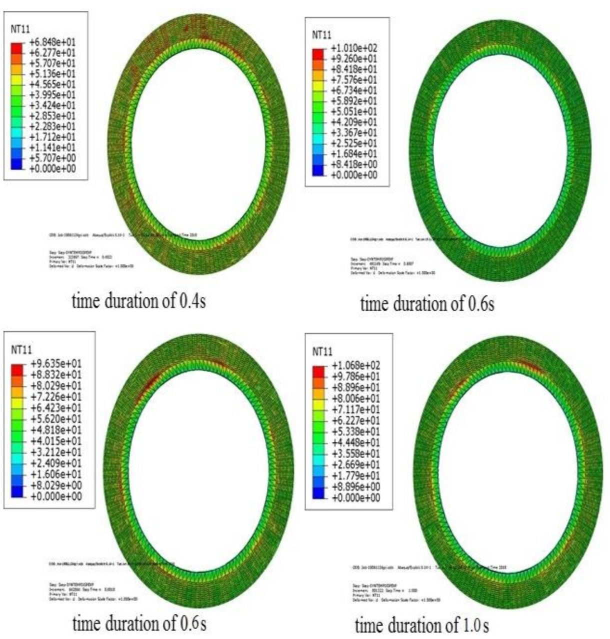
CONCLUSION

This research work analyses a disc brake by considering various materials, including ALMMC, DCI, GCI, and MSS. For each material, the study looks at temperature gradients, maximum temperatures reached, temperature fluctuation curves, and temperature rise during braking. Discussion follows the results that were reached through simulation and analytical techniques. Indicated by a larger decline at the start of braking due to a decrease in the coefficient of friction at high temperatures, there is

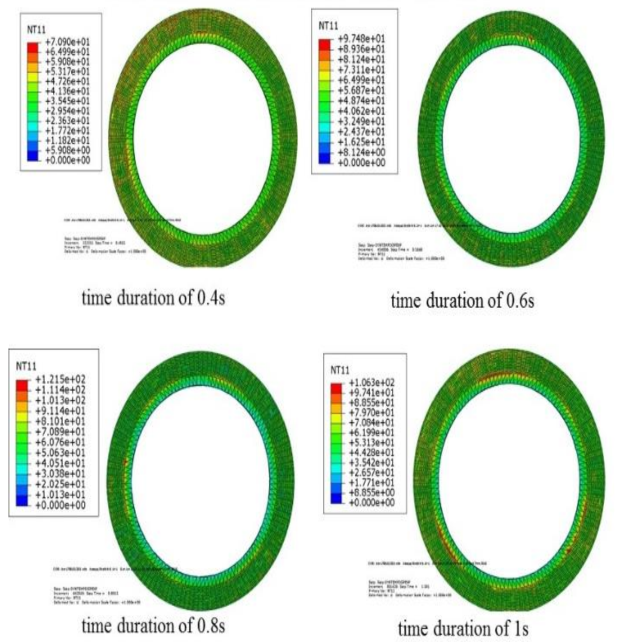
Temperature profile of ALMMC after specific time duration



Temperature profile of GCI after specific time duration



Temperature profile of MSS after specific time duration



Temperature profile of DCI after specific time duration

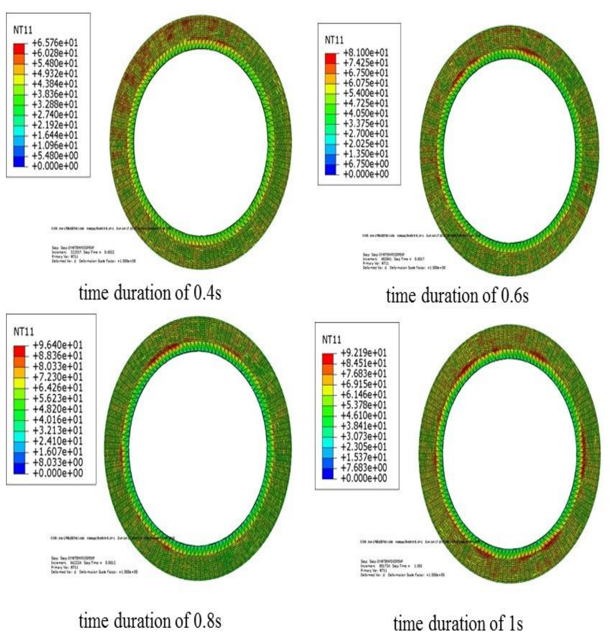
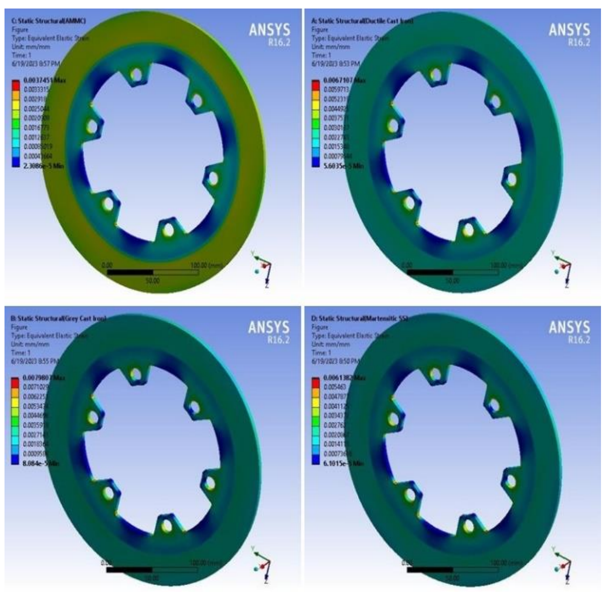


Table 2: Maximum Temperature Values at Different Instance of Time

Materials	Time			
	0.4s	0.6s	0.8s	1s
ALMMC	67.93 ºC	87.18 ºC	83.96 ºC	88.62 ºC
DCI	65.76 ºC	81 ºC	96.4 ºC	92.19 ºC
GCI	68.48 ºC	96 ºC	96.35 ºC	106.8 ºC
MSS	70.99 ºC	97.48 ºC	121.5 ºC	106.3 ºC

Structural Analysis:

Decrease in rotational kinetic energy of the combined disc-hub portion with time. Based on the conservation of kinetic energy, the hub's density is changed to maintain energy equivalence while the disc's characteristics are left unaltered in the numerical analysis methods. In the simulation, the convection heat transfer coefficient is calculated to be 86.6 W/m2-K, considering the minimal conduction into the hub from the quick braking. Based on the static structural analysis results and the provided boundary conditions of an angular velocity of 68 rad/s, hydraulic pressure of 0.3 MPa, disc thickness of 4.8



Equivalent Elastic Strain for various materials

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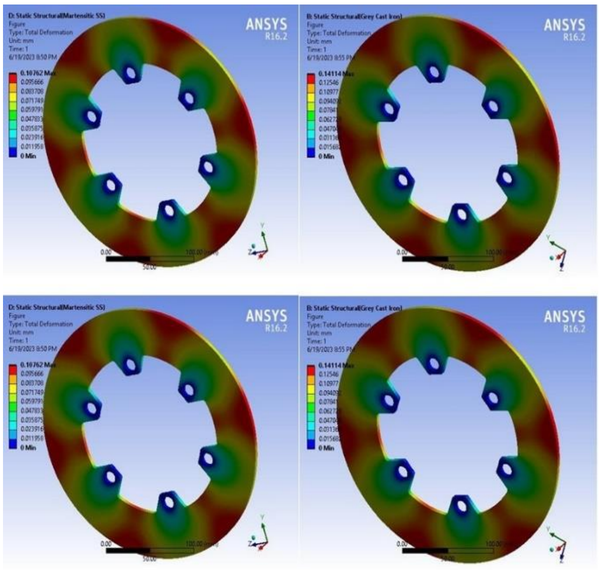
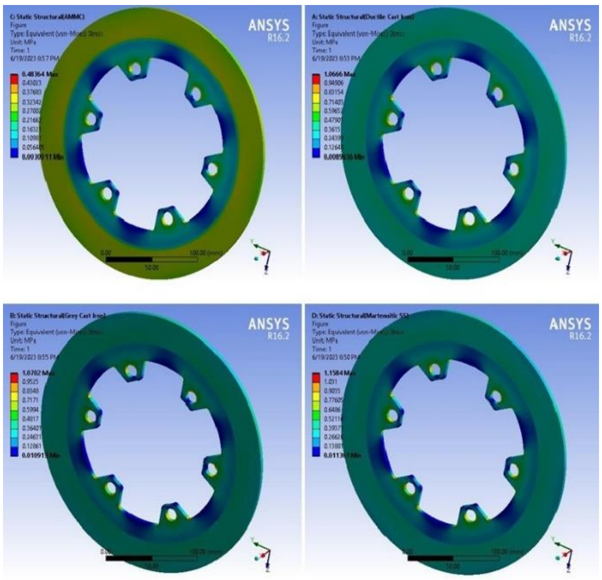
mm, and stopping time of 1.6 seconds, the following conclusion and results can be derived:



Total Deformation for various materials

1. Deformation Analysis: The disc rotor's structural analysis reveals varying levels of deformation among the materials tested. GCI exhibits the highest total deformation (0.1411 mm), followed by MSS (0.1076 mm), DCI (0.1184 mm), and ALMMC (0.077 mm). This suggests that GCI is most susceptible to deformation under the applied loads, while ALMMC demonstrates the least deformation.

2. Stress Concentration Analysis: The analysis indicates



Equivalent (von-Mises) Stress for various materials

different stress concentrations in the materials tested. MSS exhibits the highest equivalent Von-Mises stress (1.1584 MPa), followed by GCI (1.0702 MPa), DCI (1.0666 MPa), and ALMMC (0.4836 MPa). This implies that MSS experiences the most significant stress concentration, while ALMMC exhibits the lowest stress concentration.

Conclusion: The results highlight the importance of material selection in disc rotor design. GCI demonstrates higher deformation and stress levels, suggesting potential concerns for durability and structural integrity. Conversely, ALMMC exhibits superior mechanical performance with lower deformation and stress, making it a favorable choice for improved disc brake performance. MSS shows the highest stress concentration, indicating the need for careful consideration in its application. These findings provide valuable insights for optimizing disc brake design, considering the materials' mechanical properties and their response to applied loads. They can guide engineers in selecting the most suitable material and design modifications to enhance the disc brake's performance, reliability, and longevity.

Material	Highest Temperature	Total Deformation	Equivalent Elastic strain	Equivalent (Von-Mise) Stress
ALMMC	101.3°C	0.077 mm	0.003	0.4836 MPa
DCI	99.1°C	0.1184 mm	0.006	1.0666 MPa
GCI	112.2°C	0.1411 mm	0.007	1.0702 MPa
MSS	128.1°C	0.1076 mm	0.006	1.1584 MPa

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