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Influence of Brake Disc Material on Temperature Distribution in Ventilated Discs: A Numerical Simulation Study

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Abstract
The braking system is one of the most critical systems to ensure the safe driving performance of vehicles. Among the components of the braking system, the brake disc is the part most susceptible to wear. When the braking system is engaged, physical contact between the brake pad and the brake disc generates high pressure and temperature. If the temperature rises beyond the material's endurance limit, it can cause wear or cracking of the brake disc, potentially leading to brake system failure and safety hazards. Therefore, this paper presents a study on the heat generation process in disc brake mechanism using hydraulic drive and simulates the temperature distribution field on the brake disc using three different disc materials to determine the maximum temperature values distributed on the disc. The study is conducted on disc brake mechanism of a passenger car. The materials used in the research include Grey Cast Iron, Inconel 718, and Inconel 625. Based on the developed simulation model, the regions with high-temperature formation were identified. In general, the highest temperatures typically appear on the friction surface of the brake disc around the 2nd second of the braking cycle, then gradually decrease and end at the fourth second. The simulation results show that the brake disc made from Grey Cast Iron exhibits the lowest maximum temperature compared to the other two materials.

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Keywords: Passenger car, disc brake mechanism, material, temperature distribution field, maximum temperature

1. Introduction

The braking system in automobiles is responsible for reducing speed or stopping the vehicle as desired by the driver. The operating principle of the braking system is the conversion of the vehicle's kinetic energy into thermal energy through the friction generated on the contact surfaces during braking. Currently, disc brake systems are widely used in automobiles due to several advantages such as braking stability and effective heat dissipation. According to various studies, up to 90% of the heat generated during braking can be absorbed by the brake disc [1, 2]. The high temperature of the brake disc leads to several negative effects, such as increased wear, brake fade, and the development of cracks in the disc. Research [3] has shown that when the friction coefficient decreases from 0.4 to 0.3 during braking, the loss of braking torque can reach up to 25%. The rate of conversion from kinetic energy to thermal energy determines the braking power of the system, while the braking efficiency depends on how quickly the disc can be cooled. If the heat dissipation process is slow, the disc temperature will rise, reducing braking efficiency and possibly leading to the propagation of surface cracks. The amount of work performed during braking significantly influences the heat generated and is affected by various factors, including the pressure distribution at the contact surface. When modeling the braking system, two common assumptions are typically applied: (i) Constant pressure distribution, or (ii) Constant wear of the friction material. These assumptions depend on the hardness of the brake material, where harder materials tend to experience more uniform wear on the contact surface [4, 5]. The brake bench test and numerical simulation analysis of heat pipe ventilated brake disc (VBD) and general VBD under the condition of repeated fifteen braking and continuous downhill braking proposed to analyze the temperature field distribution of VBD during repeated fifteen braking and continuous downhill braking [6].

A numerical analysis of temperature behavior in solid and VBDs during repetitive braking proposed to compare the thermal behavior of solid and VBDs under repeated cyclical braking [7].
A numerical investigation of the thermal effect of material variations on the brake disc proposed to to understand the effects of material changes on their thermal properties by subjecting brake discs to Computational Fluid Dynamics (CFD) flow analysis and to offer suggestions [8]. A comparative analysis of heat transfer in the brake mechanism using comsol software proposed to simulate the heat generation and dissipation process during braking process [9]. The simulation of thermal stability and optimization of brake mechanism of a passenger car using NX Siemens software proposed to design brake drum and brake pad models, and then ansys software was applied for detailed analysis using the finite element method [10]. A model of the drum brake mechanism of heavy truck using ansys software proposed to evaluate the heat transfer process of drum-brake mechanism [11]. The numerical simulations using three disc types: Solid, ventilated with 32 fins, and ventilated with 42 fins proposed to investigate the transient thermal responses of both solid and ventilated disc brakes during single-stop braking [12]. The objective of this paper is to investigate the temperature distribution on the surface of a ventilated brake disc with different materials under emergency braking conditions. The brake disc model is simulated using ANSYS Workbench 2022 R1, providing results on temperature distribution and the maximum temperature reached on the disc. From these results, recommendations could be made for selecting suitable disc materials to limit the maximum temperature of the brake disc and maintain effective braking performance.

2. Materials and methods

2.1. Heat flux density transferred into the brake pad and disc

Heat flux at the surfaces is determined from the ratio between thermal energy and the contact area of each component. The contact model of the friction pair is illustrated in Fig.1. The rate of heat generated due to friction between these surfaces could be calculated using the following equations [6]:

$$d\dot{E} = dp = r\omega\mu\varphi_0 r dr \tag{1}$$

$$d\dot{E} = d\dot{E}_p + d\dot{E}_d \tag{2}$$

$$d\dot{E}_p = (1-\gamma)dp = (1-\gamma)\omega\mu\varphi_0 r^2 \tag{3}$$

$$d\dot{E}_d = \gamma dp = \gamma\omega\mu\varphi_0 r^2 dr \tag{4}$$

where, $d\dot{E}$ – the heat generated between the friction surfaces, $d\dot{E}_p$ - the heat absorbed by the brake pads, $d\dot{E}_p$ – the heat absorbed by the brake disc, v = r.ω – the rotational velocity of the disc, φ_0 – the contact angle between the pad and the disc, γ -the heat distribution coefficient on the brake pad and disc.
The heat distribution coefficient [12] couls be determined by the following expression:

$$\gamma = \frac{\xi_d S_d}{\xi_d S_d + \xi_p S_p} \tag{5}$$

where, S_d, S_p – are the contact areas of the disc and the brake pad, respectively; ξ_d *and* ξ_p – are the thermal diffusion coefficients of the disc and the brake pad. The area calculation model is illustrated in Fig. 2.

Fig. 1 presents the structural configuration of the HPS. A nonlinear dynamic model has been formulated based on this structural representation, incorporating the stiffness coefficient k_h and damping coefficient c_h, as illustrated in Fig. 2.

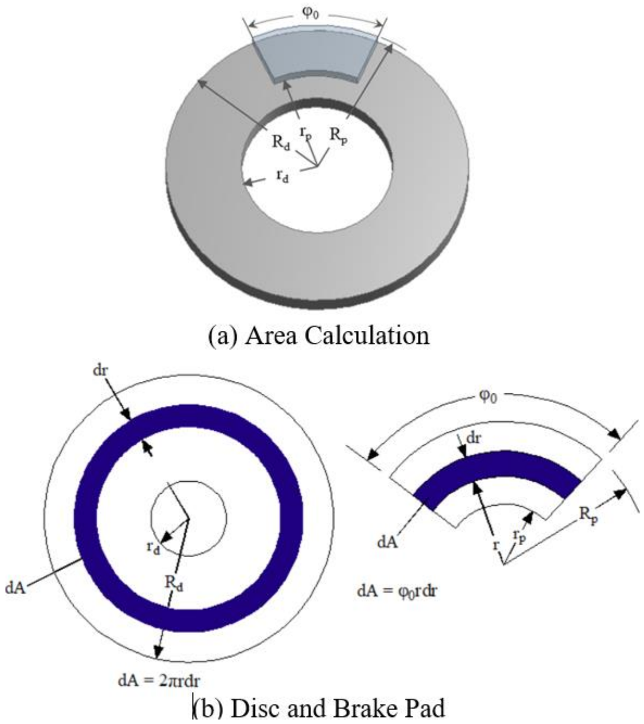


Fig 1: Contact Model

There are two types of pressure distribution: uniform pressure distribution and uniform wear pressure distribution. In the case of uniform pressure, where $p = p_{max}$, the heat flux density is a function dependent on both time and space, with the variable *r*. This is because, during braking, the angular velocity decreases over time, and an increase in the spatial variable *r* leads to an increase in the work done by frictional force. The heat flux density *q* in the contact area is updated according to the pressure distribution, and the heat flux density in the brake pad could be described by the following equations:

$$q_p(r,t) = \frac{d\dot{E}_p}{dS_p} = \frac{(1-\gamma)\omega\mu p\varphi_0 r^2 dr}{\varphi_0 r dr} = (1-\gamma)\mu pr\omega(t) \tag{6}$$

$$q_{0p}(r,0) = \frac{d\dot{E}_p}{dS_p} = \frac{(1-\gamma)\omega\mu p\varphi_0 r^2 dr}{\varphi_0 r dr} = (1-\gamma)\mu pr\omega_0 \tag{7}$$

Furthermore, for the disc, the heat flux density couls be expressed as follows:

$$q_d(r,t) = \frac{d\dot{E}_{pd}}{dS_d} = \frac{\gamma\omega\mu p\varphi_0 r^2 dr}{2\pi r dr} = \frac{\varphi_0}{2\pi}\gamma\mu pr\omega(t) \tag{8}$$

$$q_{0d}(r,0) = \frac{d\dot{E}_{pd}}{dS_d} = \frac{\gamma\omega\mu p\varphi_0 r^2 dr}{2\pi r dr} = \frac{\varphi_0}{2\pi}\gamma\mu pr\omega_0 \tag{9}$$

where, dS_d, dS_p – are the contact surface areas of the disc and the brake pad, respectively.
The uniform wear pressure method is considered more realistic after several braking cycles. In this case, the heat flux

density becomes a function of time and does not depend on the spatial variable. This means that the work done by the frictional force is uniformly distributed along the radial direction.
By substituting $p = p_{max} (r_p / r)$ into equations (6) and (8), the corresponding results are obtained.

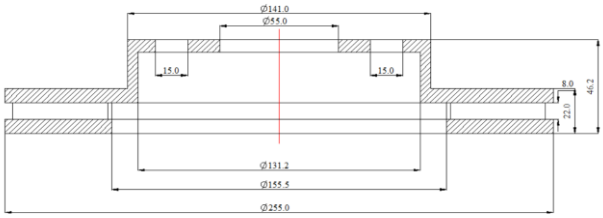
$$q_p(r,t) = (1-\gamma)\mu p_{max} r_p \omega_0 \left(1-\frac{t}{t_b}\right) \quad (10)$$

$$q_d(r,t) = q_{0d}(r) * \left(1-\frac{t}{t_b}\right) \quad (11)$$

$$q_d(r,t) = \frac{\varphi_0}{2\pi}\gamma\mu p_{max} r_d \omega_0 \left(1-\frac{t}{t_b}\right) \quad (12)$$

2.2. Simulation Study Using ANSYS Workbench

The simulation study was conducted as follows: First, the ANSYS transient thermal model was used to simulate the heat distribution of the brake disc.
Geometric model and meshing: This paper uses the shape and size of the brake disc from a Toyota Vios car to construct a brake disc model for simulation. The 3D model was developed with ANSYS Workbench software. The properties of each material used for the brake disc are provided in Tab.1. The 3D geometric model was meshed using hexagonal elements. Geometric model and meshing is illustrated in Fig.2.



(a) Dimensions of VBD

(b) Geometric model

(c) Meshing

Fig 2: Geometric model and meshing

Boundary conditions: The initial temperature of the brake disc at time t = 0s was set to 22°C. The simulation time was 4s. The convective heat transfer coefficient was assumed to be 230 W.m^{-2}. °C^{-1}. The only heat source transferred to the two surfaces of the brake disc is the heat flow density q_d (r, t) calculated according to equation (12) and given in Table 2.

Table 1: Material properties of brake disc

Material properties	Grey Cast Iron	Inconel 718	Inconel 625	Unit
Density	7850	8220	8400	Kg.m^{-3}
Young's modulus	98	200	207	Gpa
Poisson's ratio	0.27	0.29	0,278	-
Thermal expansion coefficient	11	13	11	µ.K^{-1}
Thermal conductivity	50	11	15	W.m^{-1}.K^{-1}
Specific heat capacity	500	435	402	J/K^{-1}.kg^{-1}

Table 2: Parameter values used in simulation

Parameters	Grey Cast Iron	Inconel 718	Inconel 625	Unit
Friction coefficient, µ	0.4			-
Inner radius in pad, r_p	84.75			mm
Arc angle pad, φ_0	65			Degree
Initial angular velocity, ω_0	152.44			rad/s
Maximum pressure, p_{max}	1.3			Mpa
heat partition coefficient, γ	0.97	0.94	0.95	-
Braking time, t_b	4	4	4	s
Heat flux, q(t)	*1.1773*10^6(1-t/4)*	*1.1361*10^6(1-t/4)*	*1.1486*10^6(1-t/4)*	W/m^2

Evaluation of mesh iindependence: To evaluate the mesh independence, the simulation was performed with the following mesh element quantities: 886,640; 910,421; and 910,421, respectively.

Based on the results, the mesh with 910,421 elements was selected for the entire simulation in this study.

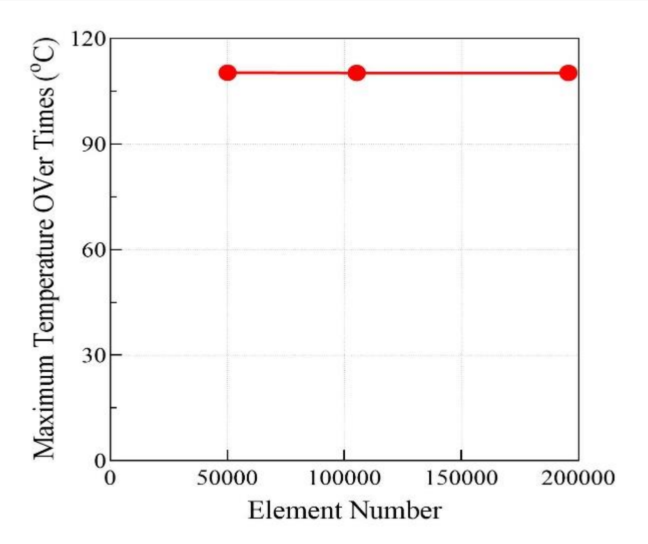


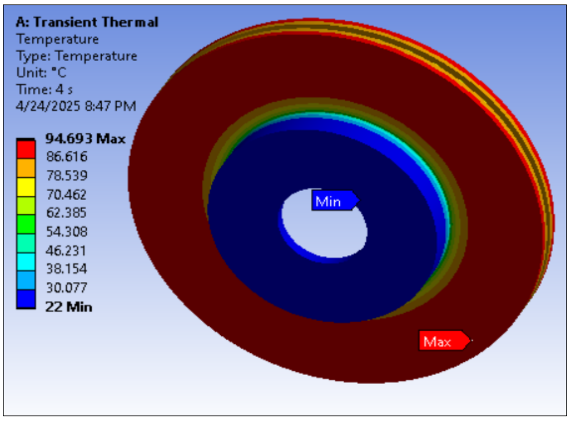
Fig 3: Mesh independency test

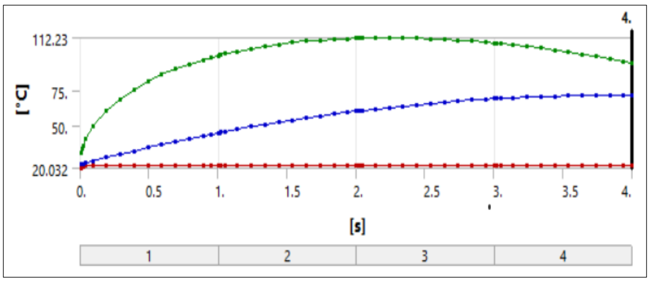
3. Results and discussion

In this study, radiative heat transfer was neglected [4]. Heat from the brake disc would be transferred to various components of the braking system, such as brake cylinders, brake fluid, and wheel bearings. However, these components are limited by the maximum allowable temperature to maintain optimal working conditions. Therefore, measures are needed to quickly dissipate the heat absorbed by the brake disc. The main measure for this is through convective heat exchange. The convective heat transfer equation for the brake disc can be written as Eq. (13).

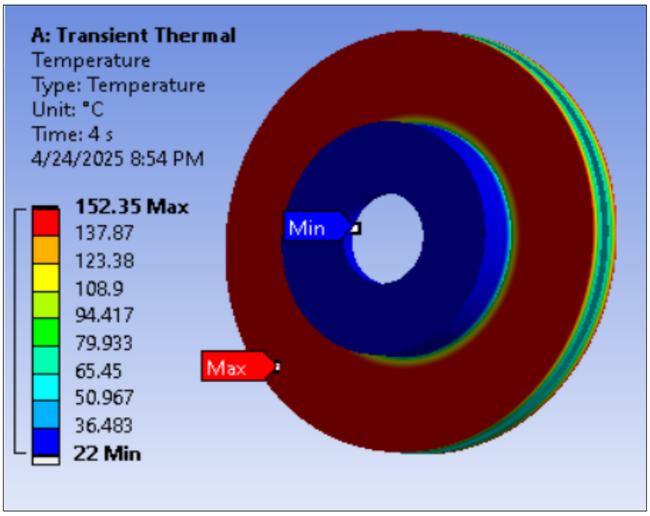
$$Q=Ah(T_d-T_\infty) \qquad (13)$$

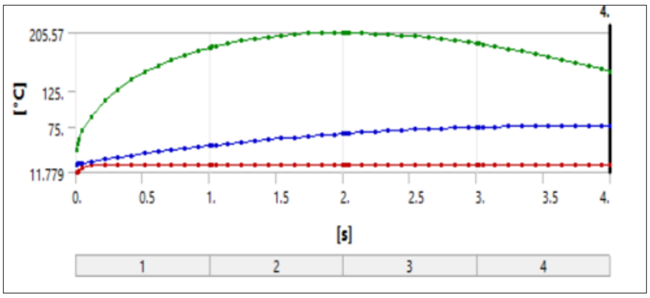
From Eq. (13), it could be seen that increasing the convective heat transfer area (A) will enhance the amount of heat transferred from the disc to the surrounding environment. To clarify this issue, the author conducted a simulation study on the temperature distribution of ventilated brake discs under emergency braking conditions. After the simulation, the results showed that the temperature distribution on each type of ventilated brake disc from various materials is different over a 4-second braking period.



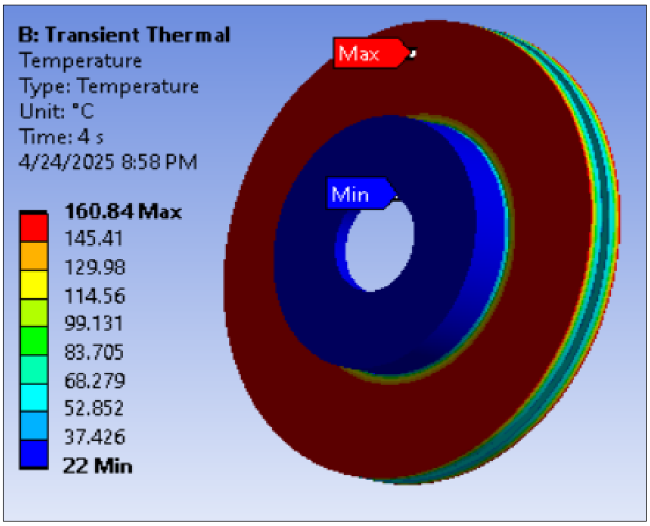


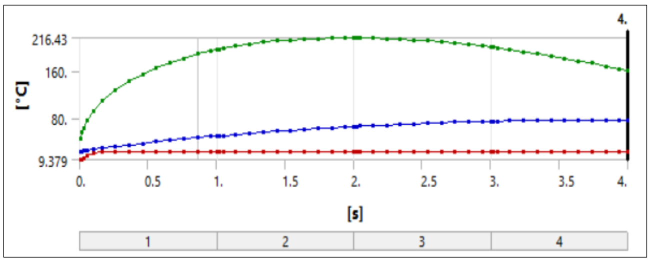
(a) Grey Cast Iron





(b) Inconel 718





(c) Inconel 625

Fig 4: Temperature distribution of VBD

The simulation results in Fig. 4 show that the temperatures of all three types of brake discs at the beginning of the braking process increased and reached their peak at the 2-second mark. After that, the temperatures gradually decreased and ended at the 4-second mark. At that point, the temperatures of the brake discs made from Grey Cast Iron, Inconel 718, and Inconel 625 are 94.693°C, 152.35°C, and 160.84°C, respectively.

4. Conclusions

In this study, a simulation study using ANSYS Workbench software was established to investigate the temperature distribution field on ventilated brake disc with different disc materials under emergency braking conditions. The simulation results have shown that the ventilated brake disc made from Grey Cast Iron exhibited the lowest maximum temperature distribution compared to the other two materials under emergency braking conditions. In addition, the research results are the basis for optimizing materials to improve the thermal efficiency of disc brake mechanisms.

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