Numerical analysis of temperature behavior in solid and ventilated brake discs during repetitive braking

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ABSTRACT

The braking system is one of the crucial systems in a vehicle, responsible for decelerating it to a lower speed or bringing it to a complete stop. During braking, the friction between the brake pads and the brake disc generates heat, most of which is absorbed by the brake disc rather than dissipated. Consequently, the brake disc can rapidly accumulate significant heat, especially during repeated cyclical braking, leading to high-temperature regions. Under these conditions, several negative effects, such as brake disc wear, cracking, and reduced braking torque, may occur. This paper aims to compare the thermal behavior of solid and ventilated brake discs under repeated cyclical braking. Numerical simulations were conducted using ANSYS software, revealing that the maximum temperatures on the solid and ventilated discs during the final braking cycle were 252.61 °C and 221.12 °C, respectively. Therefore, the maximum temperature on the ventilated disc is approximately 12.5% lower than on the solid disc. These findings demonstrate that ventilated discs are effective in demanding conditions, consistently ensuring a high level of safety.

Keywords: Disc brake; Solid disc brake; Ventilated disc brake; Temperature analysis; ANSYS software.

1. INTRODUCTION

The braking system in automobiles is responsible for reducing speed or bringing the vehicle to an emergency stop. The operating principle of the braking system involves converting the vehicle's kinetic energy into thermal energy during braking. Currently, disc brake systems are widely used in various types of automobiles due to several advantages, such as stable braking performance and efficient heat dissipation. According to several studies, during braking, up to 90% of the generated heat can be absorbed by the brake disc [1, 2]. High brake disc temperatures lead to negative effects, such as increased wear, brake fade, and disc cracking. Research [3] has shown that during braking when the coefficient of friction decreases from 0.4 to 0.3, the loss of braking torque can reach up to 25%.

Many recent studies have focused on enhancing heat dissipation for brake discs to mitigate the harmful effects of braking-generated heat. Nejat et al. [4] proposed a new design for the vanes using an airfoil profile to improve air pumping efficiency, thereby increasing the flow velocity between the vanes. Their results showed that the heat transfer coefficient of the brake disc's ventilation increased by between 17% and 29%. Jafari et al. [5] conducted numerical research on improving the design of ventilated disc brakes to enhance cooling performance, identifying the width of the ventilation gap as the most significant factor. Chengfeng Li et al. [6] investigated the optimization of ventilated disc geometry to enhance cooling performance, finding that increasing the ventilation gap and bending angle of the cooling fins can improve the cooling effect. All these studies have demonstrated the effectiveness of ventilated brake discs in dissipating heat. However, research on the thermal state of solid and ventilated brake discs under extreme conditions remains limited.

The objective of this paper is to conduct a comparative study of the thermal states of solid and ventilated brake discs under extreme braking conditions—specifically, high-speed, cyclic braking. Both types of brake discs in this study are made of gray cast iron. The brake disc models were simulated using ANSYS Workbench 2022 R1 software, providing data on temperature distribution and maximum brake disc temperatures, which enables comparison and evaluation of the thermal states of the two brake disc types.

2. PROBLEM

The purpose of this study is to evaluate the heat dissipation capabilities of solid and ventilated brake discs. To achieve this objective, both brake discs were analyzed under braking conditions—specifically, high-speed, cyclic braking. The test involved five repeated braking events, starting from an initial speed of ω_o = 152.44 rad/s until the vehicle came to a complete stop with a constant deceleration rate. Between each braking event, there was a brief recovery period, Δt, for the brake disc to enhance its heat dissipation capability. Both brake discs were constructed from gray cast iron, commonly used for automotive brake discs due to its favorable thermophysical properties. These properties, including high thermal conductivity and substantial heat storage capacity [7], help prevent the discs from overheating. The key physical, thermal, and mechanical properties of gray cast iron are presented in table 1.

Table 1. *The property values of gray cast iron.*

Material properties	Values	Unit
Density	7850	$Kg.m^{-3}$
Young's modulus	98	Gpa
Poisson's ratio	0.27	-
Thermal expansion coefficient	11	$\mu.K^{-1}$
Thermal conductivity	50	$W.m^{-1}.K^{-1}$
Specific heat capacity	500	$J/K^{-1}.kg^{-1}$

To perform the numerical simulation, the specific heat flux at the pads/disc interface was first calculated. In this study, the heat generation during the braking process is calculated using a microscopic model [8]. The assumptions for calculating heat flux are as follows:

1. The material properties of the brake disc are isotropic and independent of temperature; the brake pad pressure remains constant throughout the braking process;

2. The friction coefficient between the brake disc and the pad is constant; The convective heat transfer coefficient of the brake disc surface is constant, and radiation heat transfer is neglected due to the short braking time and low temperature;

3. All kinetic energy at the disc brake surface is converted into frictional heat, which is uniformly distributed across the brake disc surface through heat flux.

According to the microscopic model, the quantity of heat generated in the contact zone between the disc and the pad is proportional to the frictional force (figure 1). To account for the influence of abrasive particles, the imperfect contact model is often used in heat calculations [9]. When using this model, the coefficient of heat partition on the brake disc surface is calculated according to formula (1) [8].

$$\gamma = \frac{\sqrt{k_d \rho_d c_d} S_d}{\sqrt{k_d \rho_d c_d} S_d + \sqrt{k_p \rho_p c_p} S_p} \quad (1)$$

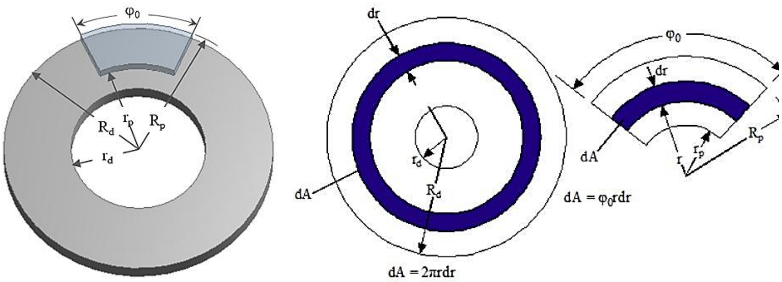
$$\omega(t) = \omega_0 \left(1 - \frac{t}{t_b}\right) \quad (2)$$

where γ denotes the heat partition coefficient, k denotes the heat conduction coefficient, c denotes the specific heat, and ρ denotes the density. S_d and S_p are contact surfaces area of the disc and pad, respectively. During braking, it is assumed that the vehicle velocity decreases uniformly, as described by equation (2). Additionally, the disc and the pads are assumed to be worn uniformly after braking. Consequently, the formula for calculating heat flux depends only on time t (s), as shown below [10].

$$q(t) = \frac{\varphi_0}{2\pi}\gamma\mu p_{max} r_p \omega_0 \left(1 - \frac{t}{t_b}\right) \tag{3}$$

where φ_o is arc angle pad, p_{max} is the brake's maximum pressure, μ is friction coefficient, ω_o is the wheel initial speed, t_b is braking time.

The graph of q(t) over time during a complete braking is shown in figure 2.



(a) (b)

Figure 1. *Models for contacting of the disc-pad (a) and area calculation of the disc-pad (b)*

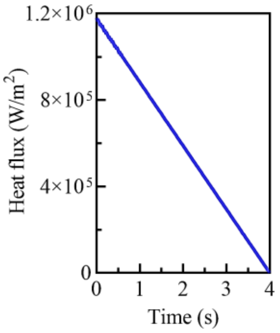


Figure 2. *Specific heat flux over time*

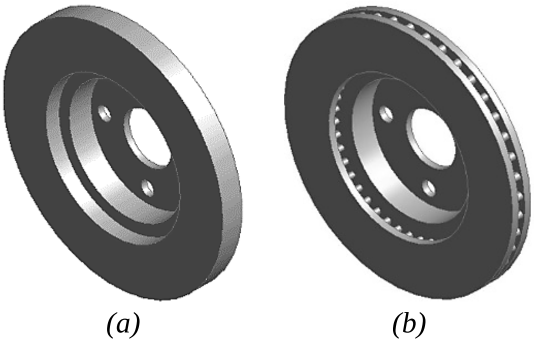
3. MATERIALS AND METHODS

3.1. 3D model

This paper uses the shape and size of the brake disc on a light vehicle to build a brake disc model for simulation. The 3D models are built on the ANSYS Workbench platform (figure 3).

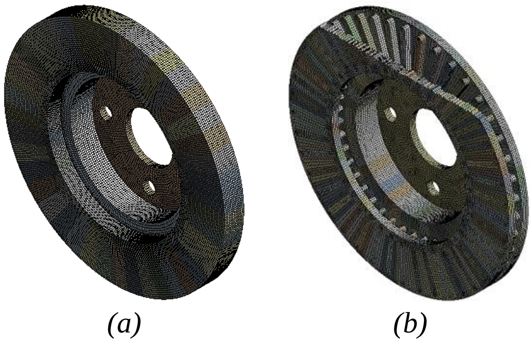
3.2. Mesh

The 3D geometric model is divided using hexahedral mesh elements. Hexahedral elements offer several advantages: they require fewer elements, provide faster solution speeds, and achieve higher precision compared to other types of meshes, such as tetrahedral meshes. Additionally, this simulation employed the skewness method to evaluate mesh quality. The maximum skewness is 0.70, which falls within the 'good' category. The mesh models of the brake disc are shown in figure 4.



(a) (b)

Figure 3. *Three dimensional solid-type disc brake (a) and ventilated-type disc brake (b).*



(a) (b)

Figure 4. *Mesh models of the solid disc (a) and the ventilated disc (b).*

3.3. Equations

The transient heat conduction equation for the disc brake can be described in the Cartesian coordinate system as detailed below [11].

$$\rho c\left(\frac{\partial T_d}{\partial t}\right) = k\left\{\frac{\partial}{\partial x}\left(\frac{\partial T_d}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\partial T_d}{\partial y}\right) + \frac{\partial}{\partial z}\left(\frac{\partial T_d}{\partial z}\right)\right\} \tag{4}$$

The boundary conditions and initial condition are specified as follows:

$$T_d = T^* \tag{5}$$

$$k_d\left\{\frac{\partial T_d}{\partial x}n_x + \frac{\partial T_d}{\partial y}n_y + \frac{\partial T_d}{\partial z}n_z\right\} = q_2(t) \tag{6}$$

$$k_d\left\{\frac{\partial T_d}{\partial x}n_x + \frac{\partial T_d}{\partial y}n_y + \frac{\partial T_d}{\partial z}n_z\right\} = -h(T_d - T_\infty) \tag{7}$$

$$T_d = T_o = T_\infty \ at \ t = 0 \tag{8}$$

where T* is the specified surface temperature, h is the coefficient of convective heat transfer, T_o is the initial temperature, T∞ is the environmental temperature, and n_x, n_y, n_z are the normal unit vectors respectively.

***Table 2.** The values of the parameters used in numerical simulations.*

Parameters	Values	Unit
Friction coefficient, μ	0.4	-
Inner radius in pad, r_p	84.75	mm
Arc angle pad, φ_o	65	Độ
Initial angular velocity, ω_o	152.44	rad/s
Maximum pressure, p_{max}	1.3	Mpa
heat partition coefficient, γ	0.970589791	-
Braking time, t_b	4	s
Heat flux, q(t)	*1.1773*10^6(1-t/4)*	W/m²

3.4. Boundary conditions

During the five braking and recovery steps, the disc's temperature does not have sufficient time to stabilize. Therefore, the ANSYS transient model has been employed to evaluate the maximum temperature over time and the temperature distribution contours of the discs. Each braking period lasts 4 seconds, while the recovery periods are 30 seconds, which is the time between consecutive braking events. For the first braking simulation, the initial temperature throughout the entire model is 22 °C at time t = 0 seconds. The convective heat transfer coefficient, h, is set at 230 W/m².°C. The model relies solely on the heat flux, calculated using Eq. (3), with its magnitude shown in table 2. It is also important to note that the pad material used in this study is derived from the simulation [10]. During the recovery period simulations, convection is the only applied boundary condition, with convective heat transfer coefficient h = 230 W/m².°C. To simulate the cumulative effect of repeated braking, each thermal transient analysis is linked with the preceding and subsequent ones. Thus, the solution of the i-th step serves as the input data for the following step, and so on. Figure 5 illustrates the schematic diagram of the implemented procedure.

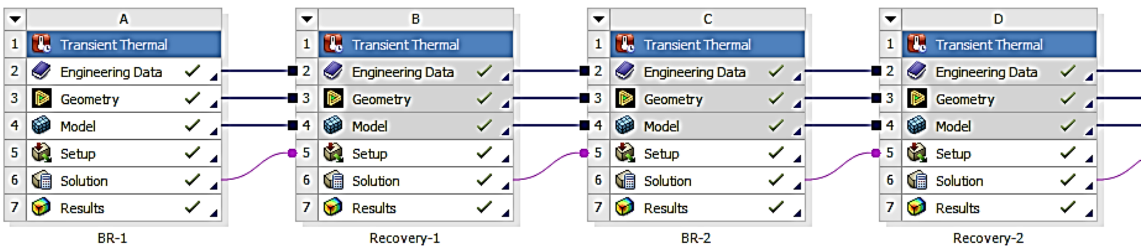


***Figure 5.** Schematic diagram of the braking/recovery simulations.*

3.5. Mesh independent test

To analyze grid independence, three mesh sizes were tested: 50.126, 105.300 and 195.691 elements. Figure 6 illustrates the findings of the grid independence study, depicting the maximum temperature over time for various numbers of elements. The mesh size of 105,300 elements was selected in the simulation.

3.6. Simulation results and discussion

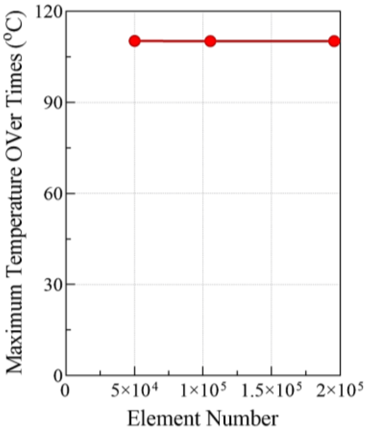
Figure 7 illustrates the maximum temperature of the brake disc over time. It shows that not all heat generated during braking is effectively dissipated into the environment, leading to a significant increase in the brake disc temperature. However, this temperature increase varies between the two types of brake discs. While the maximum temperature increase after each braking event diminishes for the ventilated brake disc, it increases for the solid brake disc. Consequently, if braking continues repeatedly, the maximum temperature of the solid brake disc may exceed the material's safety limits.

During the recovery phase, with no additional heat supplied and due to convective heat transfer, the temperature of the brake disc tends to decrease. However, since the recovery time is insufficient to return to initial temperatures, the temperature remains progressively higher from the start to the end of the simulation. Additionally, the temperature reduction during the recovery phase is faster for the ventilated brake disc compared to the solid brake disc, as shown in the temperature graph.

The analysis further highlights a significant difference between the solid and ventilated brake discs. After the first two braking events, the maximum temperature of the ventilated brake disc is higher than that of the solid brake disc. However, from the third braking event onward, the maximum temperature of the ventilated brake disc is lower. This analysis demonstrates the effectiveness of convective heat transfer for the brake discs. The convective heat transfer equation for the brake disc is detailed in Equation (9).

$$Q = Ah(T_d - T_\infty) \tag{9}$$

From equation 9, it can be observed that increasing the convective heat transfer area (A) will enhance heat dissipation from the brake disc to the environment. It is well known that the convective heat transfer area of a ventilated brake disc exceeds that of a solid brake disc. Therefore, if the heat exchange time is sufficiently long, the temperature of the ventilated brake disc will be lower. Consequently, the results from both the theoretical analysis and the simulation are consistent.



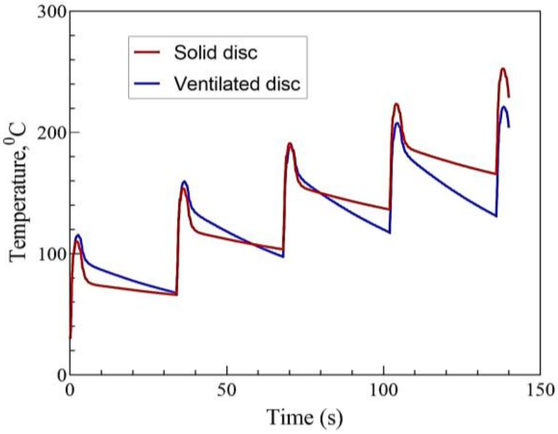


Figure 6. *Mesh independency test.*

Figure 7. *Maximum temperature over time.*

Figure 8 shows the temperature distribution on the surface and through the thickness of a quarter of the disc brake. It can be observed that during the first braking event, the solid disc, owing to its larger heat capacity, retains more heat, resulting in a lower surface temperature compared to the ventilated disc. However, in subsequent braking events, this initial advantage of the solid disc diminishes. Meanwhile, the ventilated disc, with its greater convective heat transfer area, maintains a lower temperature due to its enhanced ability to dissipate heat.

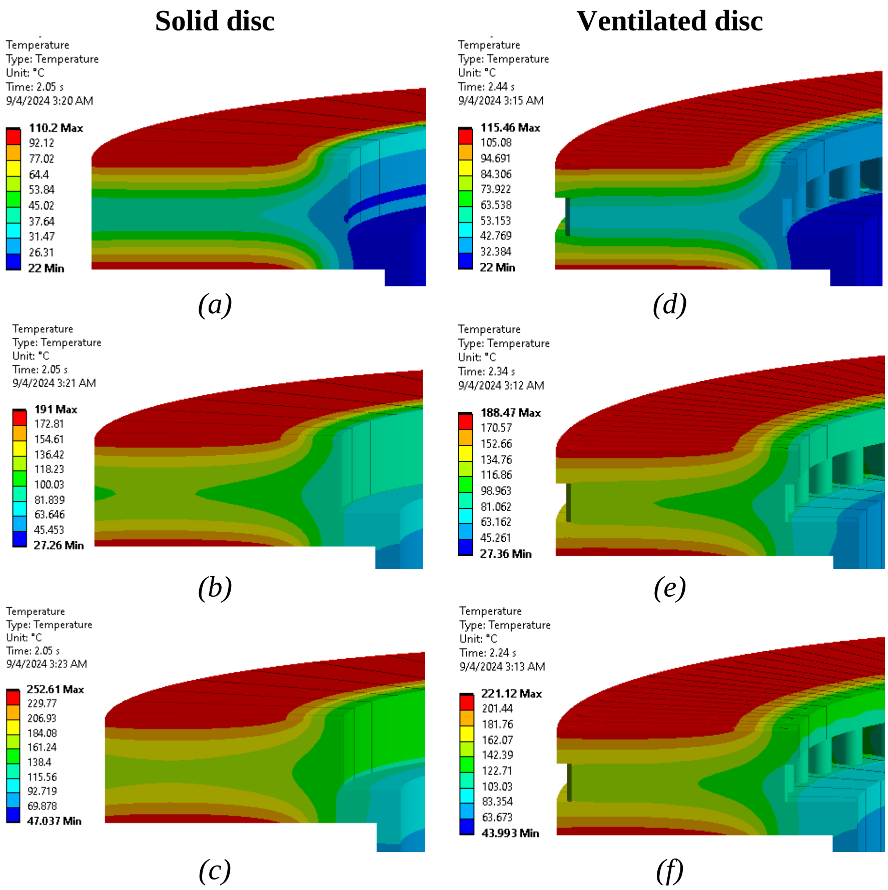


Figure 8. *Temperature variation on the surface and through the disc thickness at the first braking (a and d), at the third braking (b and e), and at the final braking (c and f).*

4. CONCLUSIONS

This paper conducted a simulation study using ANSYS Workbench to compare the thermal performance of two types of brake discs: solid and ventilated. Both disc types are fabricated from gray cast iron and tested under extremely harsh braking conditions, including high speeds and cyclic braking. The results reveal that the maximum temperatures for the solid and ventilated brake discs at the final braking event are 252.61 °C and 221.12 °C, respectively. Consequently, the maximum temperature of the ventilated brake disc is approximately 12.5% lower than that of the solid brake disc. These findings demonstrate that ventilated brake discs offer significantly improved cooling efficiency due to their larger convective heat transfer area.

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TÓM TẮT

Nghiên cứu trạng thái nhiệt của đĩa phanh đặc và đĩa phanh có thông gió trong quá trình phanh lặp lại theo chu kỳ bằng mô phỏng số

Hệ thống phanh là một trong những hệ thống quan trọng của ô tô, với nhiệm vụ giảm tốc độ hoặc dừng xe khẩn cấp. Trong quá trình phanh, nhiệt lượng sinh ra do ma sát giữa má phanh và đĩa phanh; phần lớn nhiệt này được đĩa phanh hấp thụ thay vì được tản ra ngoài môi trường. Vì vậy, đĩa phanh có thể nhanh chóng tích tụ một lượng nhiệt đáng kể, đặc biệt là trong quá trình phanh lặp lại theo chu kỳ, dẫn đến việc xuất hiện các vùng có nhiệt độ rất cao trên đĩa phanh. Trong những điều kiện này, có thể xảy ra một số ảnh hưởng tiêu cực như mòn đĩa phanh, nứt và giảm mô-men phanh. Bài báo này nhằm mục đích so sánh trạng thái nhiệt của đĩa phanh đặc và đĩa phanh thông gió trong quá trình phanh lặp lại theo chu kỳ. Các mô phỏng số được thực hiện bằng phần mềm ANSYS cho thấy nhiệt độ tối đa trên đĩa phanh đặc và đĩa phanh thông gió trong chu kỳ phanh cuối cùng lần lượt là 252,61 °C và 221,12 °C. Như vậy, nhiệt độ cực đại trên đĩa thông gió thấp hơn khoảng 12,5% so với đĩa phanh đặc. Kết quả nghiên cứu này chứng minh rằng đĩa phanh thông gió có hiệu quả hơn trong những điều kiện khắc nghiệt, đảm bảo mức độ an toàn cao trong quá trình phanh.

Từ khoá: Đĩa phanh; Đĩa phanh đặc; Đĩa phanh có thông gió; Phân tích nhiệt; Phần mềm ANSYS.