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RESEARCH

Effect of Frictional Material on Thermal Behavior of Brake System

Aliyaa Q. Mohammed[a], Ihsan Y. Hussain[a], Oday I. Abdullah[b,c,*]

[a]*Mechanical Engineering Department, College of Engineering, University of Baghdad, Baghdad, Iraq,*
[b]*Energy Engineering Department, College of Engineering University of Baghdad, Baghdad, Iraq,*
[c]*System Technology and Mechanical Design Methodology, Hamburg University of Technology, Germany.*

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* Corresponding author:

Oday I. Abdullah
E-mail: oday.abdullah@tuhh.de

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A B S T R A C T

The brake system is one of the most important systems in all vehicles. The main objective of this system is to control the vehicle mechanical movement and stopping. Where, it can't drive safely any vehicle without the brake system to control its acceleration rate and its stability. The effective factors that affected the brake performance are the properties of frictional material of pads and disc. The objective of the present work is to study the thermal behavior of the brake system using different types of frictional material for pads and find the optimal one of them. High amount of heat will be generated between the disc and pads during the braking process; may cause quick failure in future. In the present paper, a 3-dimensional thermal transient model of the brake system has been developed using finite element technique (COMSOL Multiphysics® 5.5). Five types of friction materials (St37-2, HCC, G95, LUK, and Tiger) are used in the numerical analyses. The results presented the variation and distribution of contact temperatures of the proposed frictional materials during braking period. The main conclusion based on the obtained results is that the optimal frictional material is HCC material and the poorest one is LUK material.

1. INTRODUCTION

The Brake system is the most significant part of a vehicle to control the motion during a seriously dangerous state by converting kinetic energy into thermal energy between pads and disc in very short time. The braking performance depends on mating of its main parts: disc and friction pads. Brake system used in many applications such as light vehicles, cars, minibus, agriculture trucks, tractors, building tools, airplanes, trains, and manufacturing tools. Also, it is used in the building tools in static architectures such as cranes, hydraulic brakes, and in agriculture trucks. There are two types of brake systems: dynamic and friction that used in airplane brakes [1].

The brake pad is an important element in the brake system that used in the vehicle. It ensures that the braking system works properly and keeping the stability of the automobile under different working conditions. So, it is necessary to keep the contact pressure distribution uniform as much as possible over the surfaces of pads and disc to avoid appearing the hot spot regions. In order to decrease the negative effect on human health, and the amount of emission of pollution to the environment from industry sector, traffic (exhaust smoke, brakes and tires) or other sources, an alternative friction materials are used.

Belhocine and Omar [2] performed a numerical analysis to find the solution of contact and thermal problems of brakes system. They used finite element method (ANSYS software) to determine the deformation of the contacting parts (pads and disc). The results presented the Von Misses stress, displacement, and contact pressure of the pad at different time of braking process. Their brake system worked under the dry condition. The high surface temperature and stresses caused failures such as cracks, wear, and probably breakage. The most effective parameters on the brake behavior are the rotational speed and double pressure bracket. Finally, the grooves in the plates have negative effects on the mechanical behavior of the brake pad.

Kareem et al. [3] achieved the experimental work to study the lifetime and the behavior of brake sliding-elements (car SAIPA 131 model). Also, three-dimensional finite element model was built to compute the temperature distribution of brake pads during the braking process. Different values of applied pressures [0.3, 0.5, and 0.9 bar] and speeds [60, 80, 100, 120 km/h] were applied. All results showed that the thermal stresses of brake-pads were proportionally distributed between the pad leading and trailing sides.

Kendrick [4] found that the radiations of atoms from three different brake pad materials that has minimum radiations during analyzing the electrification effect of two braking cycles. The experimental tests were done for two types of pads; non-asbestos organic (NAO) and composition of non-asbestos organic (NAO). The results showed that the measured elements were lower than 1μm in dimension of particulate matter (PM1) and the composition type of pads had less emission to environment. Where, it has less negative effect on human health and environment.

Belhocine and Abdullah [5] developed a new mathematical simulation of the thermo-mechanical problem of the braking system. The effect of design parameters on the distribution and variation of the heat generated due to friction between the brake disc and the pads was investigated. The results presented the surface temperature distribution, stress distribution, and contact pressure distribution under different boundary conditions.

Yevtushenko and Grzes [6] developed a new approach to select the material for brake pad's based on temperature mode records. Numerical calculations for thermal problem of the disc brake through a single braking were achieved. The heat dynamics of friction and wear (HDFW) equations with attention to friction coefficient, thermal sensitivity and thermos-mechanical mass wear amount was applied. The optimal tribological appearance was MCV-50 and FMC-11 ceramic metal. The selected pads depended on the environmental condition, machining cost of material, and strength of material.

Belhocine and Omar [7] investigated the structural and mechanical behavior of 3-dimensional pad models during the braking process under dry contacts slipping conditions without thermal effects. When a vehicle stops, a part of the temperature generated by friction emissions to the air through convection and radiation.

The heat exchange coefficient (h) was calculated based the transient thermal analysis of cast iron brake disc using the finite element (FE) method. A comparison was made between the temperature of a full and ventilated brake disc to show the effect of cooling on the brake system.

Belhocine and Ghazaly [8] discussed FE models of three-dimensional disc brake system that combined the wheel hub and steering knuckle. Experimental analysis of a disc brake system was used firstly to improve the accuracy of FE simulation results. Then, stability of the disc brake assembly with frequencies ranging from 1 kHz to 10 kHz was examined. Initial step of analysis was carried out to predict unstable frequencies by applying complex eigenvalue analysis to the FE model. Finally, the parametric study as a controller was conducted to evaluate the effect of the Young's modulus on the disc brake system.

In the present research, it will be developed a new numerical model to investigate the effect of friction materials of pad on the thermal behavior and performance of the brake system during the braking process (braking time=3s). Five different frictional materials are selected as alternative materials, which are available in the markets that manufactured based on standards criteria. Finite element method was used to simulate the braking process and find the thermal behavior during the braking process using COMSOL Multiphysics® 5.5 software.

2. MATHEMATICAL MODEL

Figure 1 shows the main components (Disc brake and pad) of the brake system for a typical vehicle. When the vehicle moved with specified speed, the pad will be pressed against the disc. This will convert the kinetic energy into thermal energy. This heat will be dissipated through the parts of brake by conduction and convection to the surrounding environment. It can be written the kinetic energy as follows [9];

$$EC = 1/2\ m\, v_\circ^2 \tag{1}$$

Where EC is the kinetic energy (J), m is mass of the car and $v_\circ$ is initial speed of the car. The vehicle slows down with constant deceleration as [9];

$$v = v_\circ\,(1 - t/t_b) \tag{2}$$

Where t is time and t_b is the braking time. Part of heat will be absorbed by the disc and the rest absorbed by the pads. With perfect thermal contact (neglecting thermal contact resistance), the temperatures of the contacting surfaces for both pad and disc are equal. The heat generation due to friction that enter to the pad and disc are [10],

$$q_d(r,\theta,t)\big|_{z=\delta_d} = (1-\gamma)\mu p_o \omega(t) r \tag{3}$$

For

$$r_p \le r \le R_p, 0 \le \theta \le 2\pi, 0 \le t \le t_b$$

$$q_p(r,\theta,t)\big|_{z=\delta p} = \gamma \mu p_o \omega(t) r \tag{4}$$

For

$$r_p \le r \le R_p, |\theta| < 0.5\theta_0, 0 \le t \le t_b$$

Where γ is the heat partition ratio, μ is the coefficient of friction, p_o is the contact pressure (N/m^2), r is the inner radius (m), R is the outer radius (m), δ is the thickness (m), and θ_o is the cover angle (deg). The subscripts p and d represent the pad and the disc, respectively. The heat partition ratio which exemplified the heat generation fraction through the brake cycle is [11];

$$\gamma = \frac{\sqrt{K_p \rho_p C_p}}{\sqrt{K_d \rho_d C_d} + \sqrt{K_p \rho_p C_p}} \tag{5}$$

Where K is thermal conductivity (W/m.K), C is the specific heat (J/kg.K) and ρ is the density (kg/m^3). The total heat generated, heat generated enters to pad and heat generated enters to disc are,

$$q_t = q_d + q_p \tag{6}$$

Were,

$$q_p = \gamma\ q_t \tag{7}$$

and

$$q_d = (1-\gamma)\ q_t \tag{8}$$

There are two approaches which used to simulate the thermal models of the automotive brakes to obtain the temperature field:

(a) Heat partitioning approach: modeling the brake system parts individually based on the heat partition factor to determine the amount of heat which enters into each part of the brake system.

(b) Total heat generated approach: modeling the brake system parts together (whole model of the brake system) and apply the total heat generated at the interface between the contacting parts. Where, the effect of convection is considered for both approaches.

Table 1. Heat partition ratio values.

Material	Heat Partition Ratio γ
St	0.089
HCC	0.341
G95	0.306
LUK	0.056
Tiger	0.088

In the present research paper, it was used Total heat generated approach, where no need to calculate the heat partition ratio (the amount of heat that enters to each part). Table 1 shows the values of the heat partition ratio of frictional materials that proposed in this paper.

3. FINITE ELEMENT SIMULATION

The first step in the simulation process is to build the 3-dimensional model of the dry friction brake. The disc brake has four grooves, and two symmetrical pads as shown in Figure 1. After the measurement process, the model of brake system using COMSOL Multiphysics® 5.5 software was build.

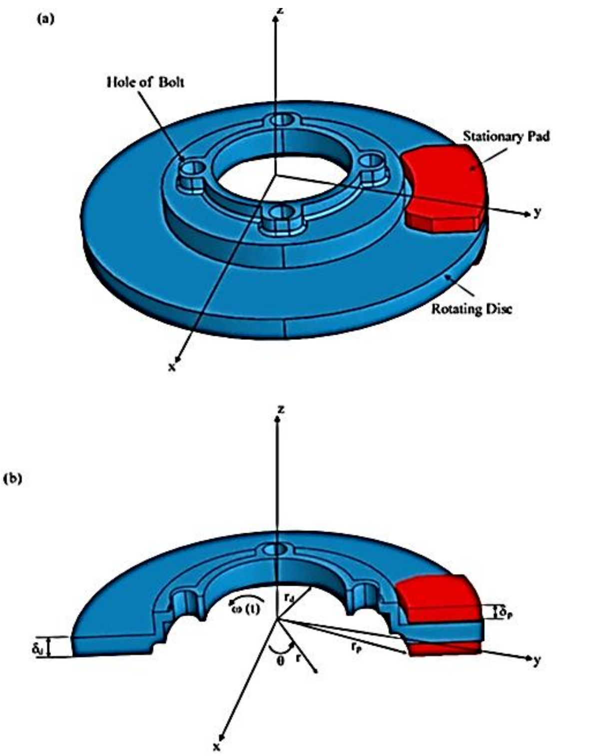

Fig.1. (a) Disc-brake system, (b) Section of disc-brake.

The assumptions of thermal simulation of brake system are;

- The heat-generating of the contact surfaces is transferred through the disc and pad according to the difference in material properties.
- Neglecting the radiation effects because the braking period is very short.
- The transferred heat is due to conduction and convection.
- The model materials are homogeneous-isotropic and independent of temperature (thermo-physical property).
- The coefficient of friction is constant during the braking process.
- The visible pad lining area is the contact area.
- Neglecting wear of pads and disc.
- The applied pressure is uniform within the contact surface.
- Neglecting contact resistance, i.e. perfect contact T_d=T_p.

The temperature distribution of disc and pads is varying with time. According to the cylindrical coordinates. 3-D temperature field's $T_{p,d}(r, \theta, z)$ are considered based on the heat conduction equations. It can be written the heat conduction equation for brake-disc, as (Figure 2) [12];

$$\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}=\frac{1}{\alpha_d}\left(\frac{\partial T}{\partial t}+\omega(t)\frac{\partial T}{\partial \theta}\right) \quad (9)$$

Where; $\omega = v_\circ/R_{tire}$, and R_{tire}=0.276 m

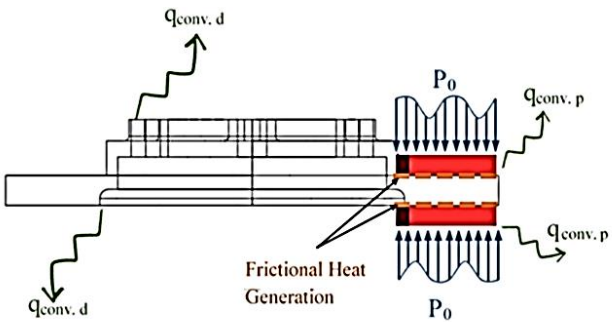

Fig. 2. Thermal boundary conditions of the disc and pad.

For

$$r_d \le r \le R_d, 0 \le \theta \le 2\pi, 0 \le z \le \delta_d, 0 \le t \le t_b$$

Where, α is thermal-diffusivity (α=k/ρc). Boundary conditions of disc surfaces are [12];

$$K_d \left.\frac{\partial T}{\partial r}\right|_{r=R_d} = h\left[T_\infty - T_{(\theta,z,t)}\right] \quad (10)$$

$$K_d \left.\frac{\partial T}{\partial r}\right|_{r=r_d} = h\left[T_{(\theta,z,t)} - T_\infty\right] \quad (11)$$

For

$$0 \le \theta \le 2\pi, 0 \le z \le \delta_d, 0 \le t \le t_b$$

$$K_d \left.\frac{\partial T}{\partial z}\right|_{z=\delta_d} = h\left[T_\infty - T_{(r,\theta,t)}\right] \quad (12)$$

$$K_d \left.\frac{\partial T}{\partial z}\right|_{z=0} = h\left[T_{(r,\theta,t)} - T_\infty\right] \quad (13)$$

For

$$r_d \le r \le R_d, 0 \le \theta \le 2\pi, 0 \le t \le t_b$$

Where h is the heat transfer coefficient (W/m^2.K). Pad surfaces expose to the same average value of convectional coefficient [12] as;

$$\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}=\frac{1}{\alpha_P}\frac{\partial T}{\partial t} \quad (14)$$

For $r_p \le r \le R_p, |\theta| < \theta_0, 0 \le z \le \delta_p, 0 \le t \le t_s$

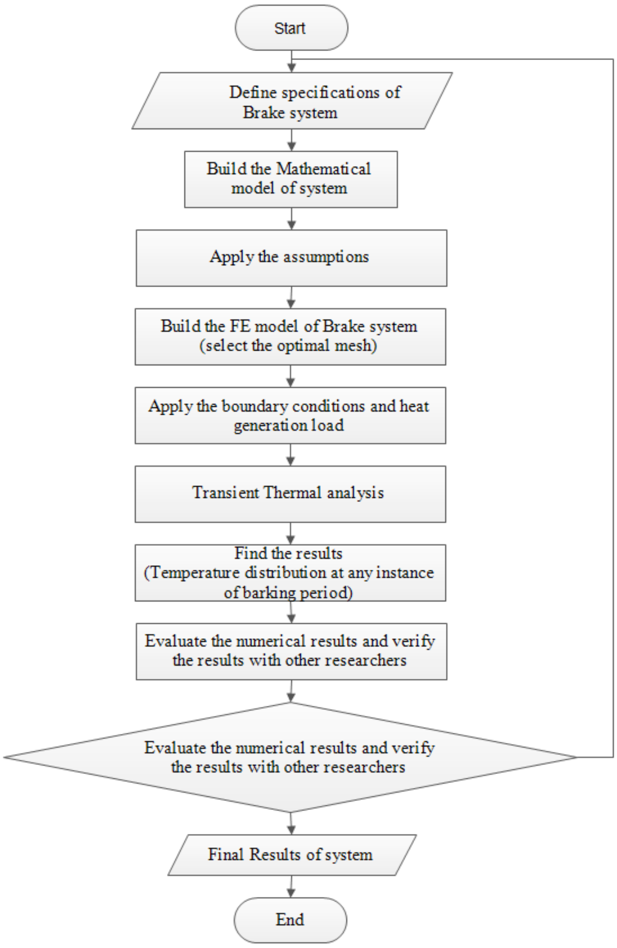

Fig. 3. Flowchart of thermal analysis using finite element method.

Boundary conditions of pad surfaces are [12];

$$K_p \frac{\partial T}{\partial r}\Big|_{r=R_p} = h\left[T_\infty - T_{(\theta,zt)}\right] \quad (15)$$

$$K_p \frac{\partial T}{\partial r}\Big|_{r=r_p} = h\left[T_{(\theta,z,t)} - T_\infty\right] \quad (16)$$

For $|\theta| < 0.5\theta_0, 0 \le z \le \delta_p, 0 \le t \le t_b$

$$K_p \frac{\partial T}{\partial \theta}\Big|_{\theta=0.5\theta_0} = h\left[T_\infty - T_{(r,0.5\theta_0,z,t)}\right] \quad (17)$$

$$K_p \frac{\partial T}{\partial \theta}\Big|_{\theta=-0.5\theta_0} = h\left[T_{(r,-0.5\theta_0 z,t)} - T_\infty\right] \quad (18)$$

For $r_p \le r \le R_p, 0 \le z \le \delta_p, 0 \le t \le t_b$

$$K_p \frac{\partial T}{\partial z}\Big|_{z=\delta_p} = h\left[T_\infty - T_{(r,\theta,t)}\right] \quad (19)$$

For $(r,\theta) \in \Omega, 0 \le t \le t_b$

Where, Ω is the contact area (m^2). The friction thermal problem is solved numerically using the finite element method with COMSOL Multiphysics® 5.5 software. Firstly, the parabolic heat conduction equation was evaluated using the Galerkin's method [13, 15] as,

$$[C]\{T\}dt + [K]\{T\} = \{Q\} \quad (20)$$

Where [C] is the capacitance matrix; {T} is the vector of temperature, [K] is the conductivity matrix, and {Q} is the applied heat flux load vector. The heat transfer coefficient of each material was calculated by [16];

$$h = 3.691\left(\frac{v_\circ}{r_{tire}}\right)^{0.8} \quad (21)$$

Figure 3 Shows flowchart of thermal analysis using Finite Element Method, where it can be seen the main steps to compute the distribution of temperature in brake system at any time during the braking process.

4. CASE STUDY

The dimensions of the brake system for a selected car are described in Table 2. The dimensions of pad and disc were taken from the actual car. Five different friction materials were selected for the pads which are St-37-2 [3], HCC, G95, LUK and Tiger materials [14]. Properties and operation conditions of the brake model of the rotating disc and the stationary pad are identified. Also, the input parameters to simulate the thermal problem of the brake system using the software are presented in Tables 3 and 4. The thermal transient analysis was applied to show the variation of the surface temperature of the contacting surfaces.

Table. 2. Dimensions of disc brake model [17].

Dimensions (m)	Disc	Pad
Outer radius	0.11	0.105
Inner radius	0.06	0.07
Thickness	0.013	0.009
Number of holes	4	----
Holes Diameter	0.013	----
Slot Length	----	0.06

After building the 3D geometry of the brake system by COMSOL Multiphysics® 5.5; the first step is selecting the time of starting and step (0, 0.001 s), and end of braking time (3s). The Grid Independency Test (GIT) was made for the average temperature of pad and disc surfaces as shown in Figures 4 and 5. This step is considered essential to find the most optimal mesh and reduce the run time of the numerical analyses. The final No. of elements of disc is 139359 and 58544 elements for pad. The linear tetrahedral element was selected to build the FE model, where the minimum and maximum element sizes are 0.9 mm and 12.4, respectively.

Table. 3. Properties of the brake system materials.

Materials	Disc	Pad				
		St	HCC	G95	LUK	Tiger
Friction Coefficient	----	0.14	0.45	0.55	0.3	0.3
Specific Heat J/Kg.K	490	1200	900	935	1010	1780
Thermal Conductivity W/m.K	60.2	0.6	33	24	0.437	0.726
Density Kg/m^3	7273	2864	1950	1870	1734	1570

Table. 4. Model operation condition.

Operation Condition	Value
Initial Velocity m/s	24
Initial Temperature K	300
Contact Pressure N/m^2	160000
Braking Start Time s	0
Braking End Time s	3
Deceleration m/s^2	8

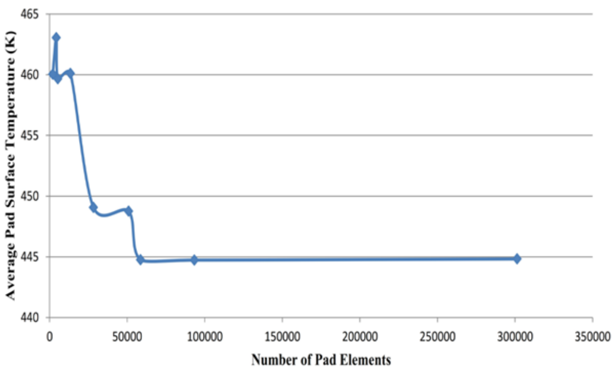

Fig. 4. Pad grid independency test (PGIT).

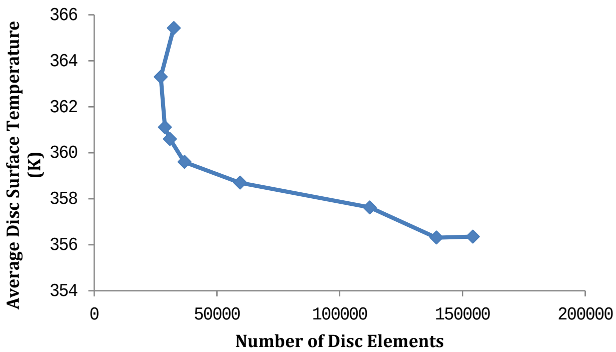

Fig. 5. Disc grid independency test (DGIT).

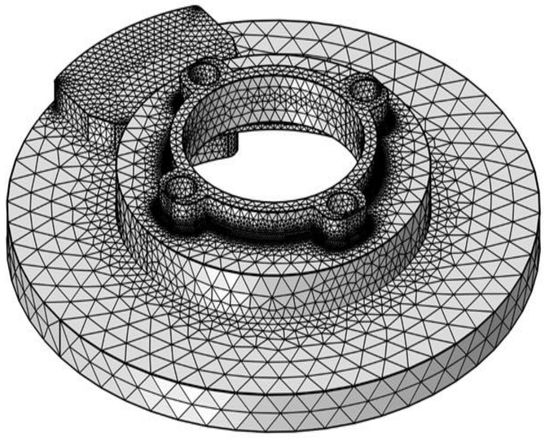

Fig. 6. Finite element model of brake system.

Figure 6 shows the finite element model of the brake system. The numerical model was verified by comparing the results of maximum temperature using the developed model with reference [18] and the percentage different between them is not exceeding 1.9%.

5. RESULTS AND DISCUSSIONS

The objective of the current analysis is to develop a numerical solution for studying the effect of using different frictional materials of pad on the distribution of surface temperature of contacting elements of brake system.

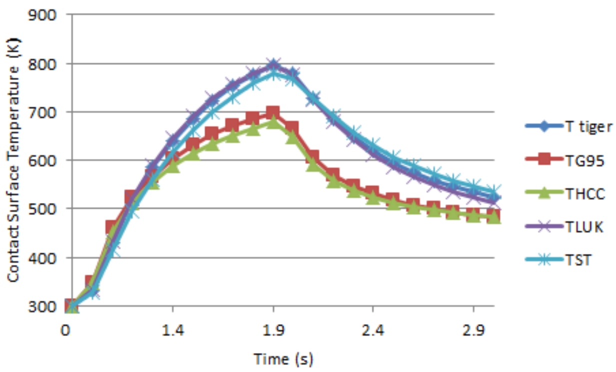

Fig. 7. The maximum surface temperature with time using different frictional materials.

Figure 7 shows the variation of the maximum contact surface temperature using five different frictional materials (St37, HCC, G95, LUK and Tiger) during the braking process at a location of 77 mm in r-direction. It can be seen that the temperature increased dramatically and reached the peak value after 1.9 s from the starting time for all cases. Later on, the values of temperature decreased gradually to the minimum value at the end of the braking process 3s. The highest contact surface temperature occurred when using LUK material and the lowest one occurred when using HCC frictional material. The difference between the peak values of the highest and lowest surface temperatures is found to be 118.26 K (14.84%). The main reason to obtain these results is the different values of the thermal properties, especially the conductivity and the friction coefficient values that play important role on the thermal behavior of the contacting surfaces.

Figures 8-12 illustrate the temperature distribution on contact surfaces (disc and pads) at selected intervals time using different frictional materials. It can be seen that the maximum temperatures occurred during the braking process corresponding to St, HCC, G95, LUK and Tiger are 779.63, 678.43, 697.47, 796.69 and 795.35 respectively. The results of the maximum contact temperatures gave another point of view to the relationship between the friction material (the type of the frictional material of pad) and the highest local surface temperature (flash temperature). It can be seen very clearly that the high local surface temperature occurred when using the frictional material (St, LUK and Tiger) which have the lowest value of thermal conductivity. While, when using the frictional material (HCC and G95) which have the higher thermal conductivity, the high local surface temperature approximately disappeared.

The final temperature of the brake system is a function of many factors such as the cooling of the system, initial velocity, contact pressure design of the brake system, and temperature of the environment. The other effective factors on the amount of the heat generated are the coefficient of friction and thermal conductivity. Although the values of coefficient of friction for the frictional materials (HCC and G95) are very high compared with other three materials, but the results of temperatures are lower than them. The reason for these results is the good thermal properties of the (HCC and G95) materials compared with other materials (St, LUK and Tiger). Considering the values of thermal conductivity and coefficient of friction, the conductivity value of HCC material (33 W/m.K) is higher than the conductivity value of LUK material (0.4378 W/m.K) with approximately 75 times. Of course, the other properties have influence on the thermal behavior of the system but the dominant factor is the conductivity.

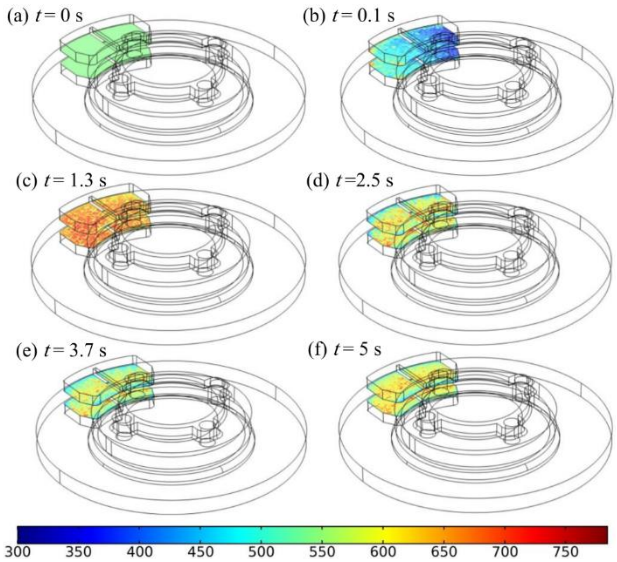

Fig 8. Temperature Distribution of brake system (St37-2).

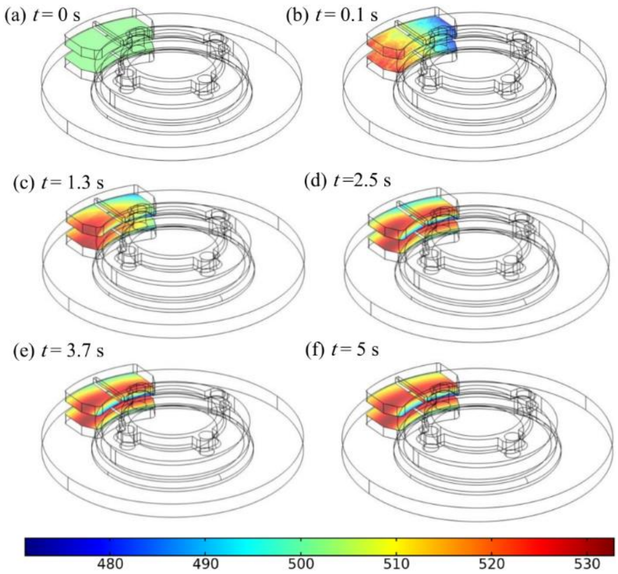

Fig 9. Temperature distribution of brake system (HCC).

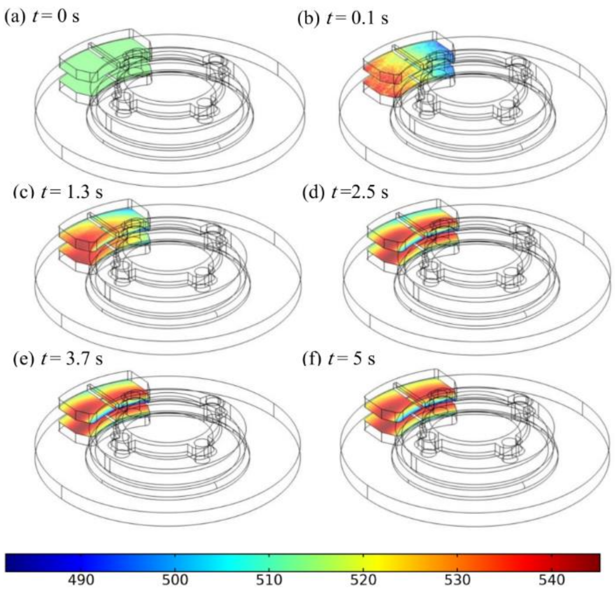

Fig 10. Temperature distribution of brake system (G95).

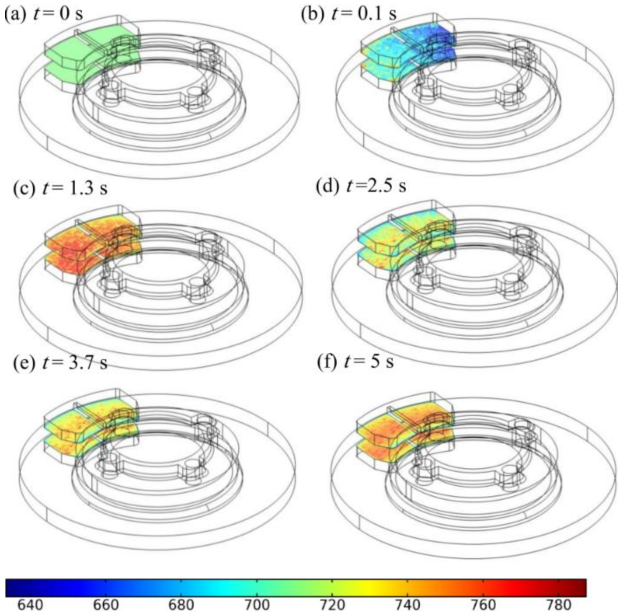

Fig 11. Temperature distribution of brake system (LUK).

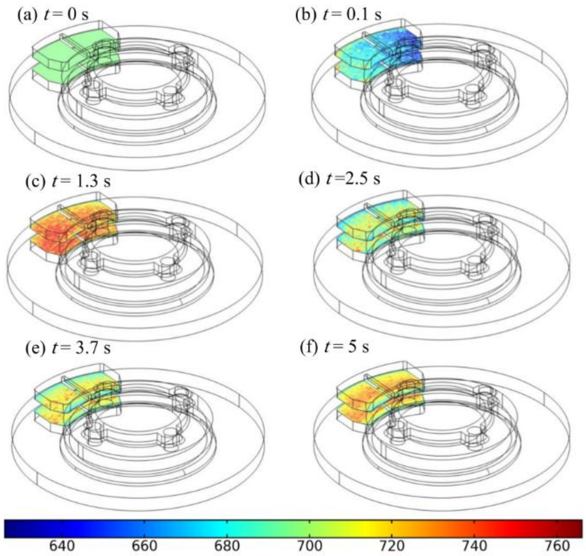

Fig 12. Temperature distribution of brake system (Tiger).

Results indicated that the optimal materials are HCC and G95 because of it have suitable thermal properties that lead to appear the lower contact surface temperature of the pad. Also, it has low prices relatively, and less negative effect on the environment.

In general, based on the values of the Heat Partition for the proposed frictional materials in Table 1, it can be concluded that the friction material that has the lowest Heat Partition value is not considered the optimal one. In other words, when Heat Partition is low, it means that the thermal properties of this material are very poor. When the value of Heat Partition is high, the surface temperature will be decreased, as the properties of this frictional material are relatively good and improve the thermal behavior of the brake system.

6. CONCLUSIONS AND REMARKS

In the present research paper, numerical analysis was developed to investigate the effect of frictional material on the thermal behavior and performance of the brake system during braking time (3 s). Based on the transient thermal analysis of brake system, the main conclusions are:

1. The most optimal frictional material of pad is HCC which has the lower surface temperature (678.43 K), because it has high thermal conductivity and coefficient of friction and heat partition value.
2. The poorest frictional material of pad is LUK that produced the highest surface temperature (796.69 K). Where this friction material has poor thermal properties (low heat partition value).
3. The thermal behaviors when using LUK and Tiger pad materials are very close because of the values of thermal properties and coefficient of friction are close to each other.
4. The thermal behaviors when using HCC and G95 pad materials are very close to each other, because the values of thermal properties and coefficient of friction are close to each other.
5. The peak temperature occurred at the same time for all five (t=1.9 s).
6. The frictional materials (St37-2, LUK, and Tiger) have the highest percentage of reduction in the temperature (between the peak and final times), which is approximately 7.5 %.
7. The frictional materials (HCC and G95) have the lowest percentage of reduction in the temperature (between the peak and final times), which is approximately 4.3 %.

There is another important point which is the dependency of material properties and coefficient of friction on temperature and position under different working conditions that will be studied in future works.

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Nomenclature

EC: Kinetic energy (J)
m: Mass of the vehicle (kg)
q: Generation heat (W/m^2)
c: Specific heat capacity (kJ / kg. K)
h: Convective heat transfer coefficient ($W / m^2.K$)
k: Thermal conductivity (W / m.K)
$p_\circ$: Normal pressure on the friction surface (N/m^2)
t: Time (s)
t_b: Braking time (s)
ω: Angular velocity (rad/s)
$T_\circ$: Initial temperature (K)
$v_\circ$: Initial velocity (m/s)
ρ : Density (kg/m^3)
γ : Heat partition ratio
δ : Thickness (m)
μ: Coefficient of friction
θ : Rotation angle (deg)