Performance Investigation of the UTeM Eco-Car Disc Brake System

M. K. Khalid, M. R. Mansor, S. I. Abdul Kudus, M. M. Tahir, and M. Z. Hassan

***Abstract*—The aim of this study is to investigate the braking performance of the UTeM Eco-Car disc brake system. The disc brake system utilized a single cross-drilled rotor with fixed calliper design. The brake system performance in term of its thermal property was determined in transient condition using ABAQUS CAE finite element analysis software. Results from the thermal analysis showed that the maximum temperature generated at the brake disc surface was 119.2^{o}C, which is within the allowable service temperature of the disc material. This indicates that the UTeM Eco-Car disc brake system is able to perform safely as per design requirement.**

***Index Terms*—Perodua Eco-Challenge, UTeM Eco-Car, Disc Brake System, Thermal Analysis.**

I. Introduction

PERODUA Eco-Challenge is a national design competition organized by Perodua Sdn. Bhd. in 2011. Universiti Teknikal Malaysia Melaka has took part in the event where participants compete in the challenge to design and build a working vehicle that can travel the furthest distance using 0.5 liter of RON95 fuel [1]. Among the major system involved in the design of the eco-car was the braking system. The braking system is very crucial in stopping the car on all moving stages including during high speed, sharp cornering and downhill movements [2]. The ability to bring a vehicle safe controlled stop is absolutely essential in preventing accidental vehicle damage and personal injury [3]. Thus, in this project, the braking performance for the UTeM Eco-Car brake system which participated in the Perodua Eco-Challenge 2011 competition was analyzed using ABAQUS finite element analysis software. The performance investigation was focused to the linear thermal analysis in transient mode. The aim was to determine its thermal capacity when operated in real track condition. This is to ensure that the new brake system designed for the UTeM Eco-Car is able to operate in actual condition without failure due to overheating thus ensuring the safety of the driver involved.

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II. Literature review

Generally, the main function of disc brake is to transmit mechanical force and dissipation of heat produced implies to be functioning at both medium and high temperature. The rotor provides braking or friction surface for brake pads to rub against when brake force is applied.

As explained by Tirovic and Ali [4], friction brakes are exposed to high mechanical and thermal loads. Mechanical loads are generated by clamping, friction and centrifugal forces, as well as by brake acceleration in different directions whereby the thermal loads are the result of the frictional heat generation on the brake friction surfaces. They also stated that thermal loads are often much more severe than mechanical loads and also much more difficult to predict accurately. As a result, the design process for most brake concepts is concentrated on thermal loads. In determining the performance of a brake system, the prediction of the brake surface temperature is the main problem in the analysis of thermal dissipation and brake surface degradations as highlighted by Dufrenoy [5]. Formation of hot spots due to excessive increase of surface temperature often resulted in brake failure due to premature wear, surface cracks, thermal distortion, judder etc [6] especially when the temperature is near or exceeds the allowable service temperature of the brake materials. Thus, to gain a safe braking system performance, the brake must be sufficiently designed to be able to dissipate the heat generated from the braking process adequately, so that the brake surface temperature is kept within the brake material in acceptable operating range [7].

Currently, there are two types of disc brake rotor used in passenger car which is solid disc and ventilated disc [8]. A solid rotor is simply a solid piece of metal with friction surface on each side and this type of rotor is light, simple, cheap and easy to manufacture. A ventilated disc meanwhile refers to–a rotor with various opening profiles (holes, grooves etc) which provide better cooling performance (additional cheat transfer function) and weight savings as well as aesthetic appearance [9]. Therefore, it is widely used compared to solid disc.

The thermal stability of the disc shape is influenced by the quantity of the material and the heat treatment before machining as well as the basic design for the disc rotor. Some of the thermally most important properties of disc brake rotor are as follows [9]:

i) Thermal capacitance (density and specific heat) is the ability to store the heat. Initially on braking process, a significant amount of frictional heat is stored and during short braking, this thermal capacitance is dominates.
ii) Heat dissipation becomes important consideration at long braking times (between 2 to 3 minutes). The convection is occurring for more than 90 percent of the total heat dissipation in most of the braking condition, whereby radiation is almost negligible.
iii) Thermal conductivity is the ability to re-distribute the thermal energy. During long and low intensity braking, the peak temperature is depends largely on the disc material's conductivity. However, the thermal conductivity has a little effect during short braking.
iv) Thermal expansion coefficient (related to location of friction contact due to the thermal deformation) affects the tendency towards hot spotting and thermal disc thickness variation (DTV) generation. The temperature gradients of the disc brake can cause to temporary DTV owing to the uneven thermal expansion of the material.

Heat transfer is energy in transit, which occurs as result of a temperature gradient or difference. This temperature difference is thought of as a driving force that causes heat to flow. Heat transfer for a ventilated disc brake rotor occurs by three mechanism or modes: conduction, convection and radiation. Ventilated disc brake generally exhibit convective heat transfer coefficient that is approximately twice as large as those associated with solid discs. During a continued braking, a ventilated disc usually tends to reach a temperature which is approximately 60% of the temperature of a solid disc. The effect of radiation however are usually neglected in most disc brake analysis since the only contributes around 5% to 10% of the total heat transfer from the disc (in cases where the brake is subjected to normal operating temperature) [10].

III. RESEARCH METHODOLOGY

There are two major stages involved in this project, which are disc brake design and performance evaluation. In the disc brake design stage, suitable type of disc brake was selected to be installed in UTeM Eco-Car. Selection criteria were focused on price, system compatibility, and lightweight. Thus, in the end of the selection process, a commercial Yamaha LC135 motorcycle disc brake system was selected for the vehicle. The reasons are due to compatibility to the 13 inch rim used for the vehicle, as well as lightweight property and lower cost compared to a passenger car disc brake system. The disc brake system utilized a cross-drilled rotor with fixed calliper design for better heat dissipation.

The disc brake rotor is made from gray cast iron material which provides good wear resistance with high thermal conductivity and the production cost is low compared to other high performance disc brake rotor materials such as Al-MMC, carbon composites and ceramic based composites [11]. Grieve et al. also highlighted that although advanced brake materials such as aluminium metal matrix composite offer significant weight advantages compared with the traditional cast iron rotor, the aluminium metal matrix composite material has a much lower maximum operating temperature which limit its application [12]. The UTeM Eco-Car was installed with all-wheel disc brake system to maximize its braking capability and provide higher safety to the driver throughout the race. The vehicle is estimated to weigh approximately 460kg including the driver.

During the performance evaluation stage, 3D model of the disc brake selected was developed based on its exact dimensions measured using the Coordinate Measurement Machine (CMM) equipment. The disc brake 3D model was later imported in ABAQUS CAE finite element analysis software as the geometry model to perform the performance analysis. Load analysis was also performed to determine the heat flux and convection heat transfer coefficient for the brake system during its operation. Thermal analysis results obtained at the end of the analysis were validated and compared to the allowable material specification for the rotor. If the selected disc brake failed the thermal analysis process, then a new disc brake design will be selected as the replacement. The performance analysis is repeated until the disc brake met the design requirement and the disc brake will be installed on the actual UTeM Eco-Car later on. The research flow chart for this project is summarized in Fig. 1 below. Fig. 2 shows the 3D model of the UTeM Eco-Car disc brake system generated using CATIA V5R16 software while Table I summarized the overall disc brake design specifications.

TABLE I
OVERALL SPECIFICATIONS OF THE UTEM DISC BRAKES

Disc Geometry	
Disc outer diameter	0.15 m
Disc inner diameter	0.22 m
Disc thickness	0.036 m
Cross drilled hole diameter	0.007 m
Total disc surface area	0.038 m^2
Disc rotor material properties [13]	
Material type	Grey cast iron (BS220)
Density	7.1 to 7.2x10^3 kg/m^3
Thermal conductivity	46 to 55 W/m.K
Specific heat	460 to 500 J/kg.K
Service temperature	-150^{o}C to 550^{o}C

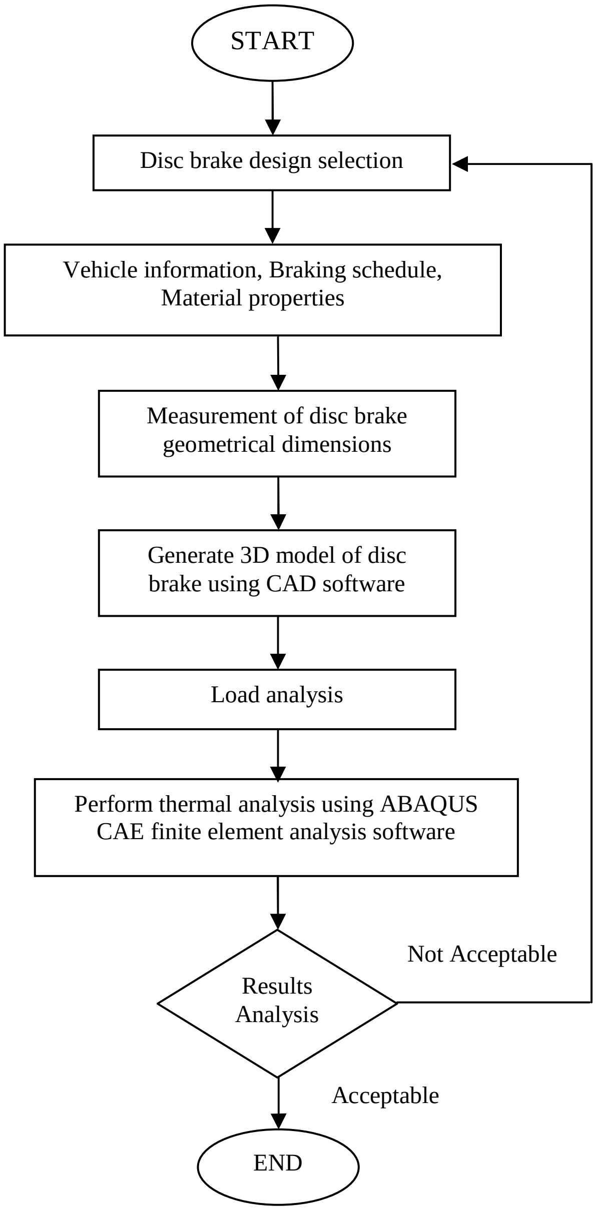

Fig. 1. Overall project flow chart

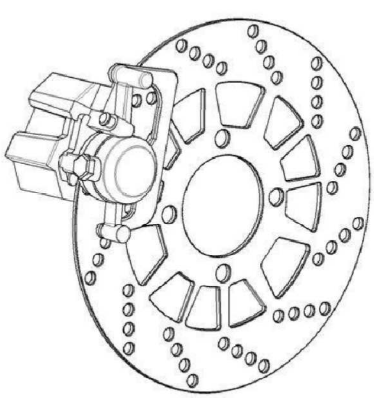

Fig. 2. UTeM Eco-Car 3D disc brake model

IV. LOAD ANALYSIS

The load analysis was performed to determine the braking load and disc brake rotor cooling characteristics when the brake system is in operation. As stated in the Perodua Eco-Challenge 2011 rules and regulations, all participating car must be able to stop without failure at given initial speed of 50km/h [14]. Thus, this value was used to simulate the worst case condition for the car in order to determine the maximum braking load subjected to the brake rotor.

A part from that, based on the track layout for the event as shown in Fig. 3, it is estimated that a total of six (6) repeated numbers of braking are needed for the UTeM Eco-Car to safely manoeuvre through the entire 1.6km course. It is also assumed that for each corner, the driver requires 3 seconds of braking time. Fig. 4 shows the summarized schematic of the braking characteristic need to be completed by UTeM Eco-Car for the whole track.

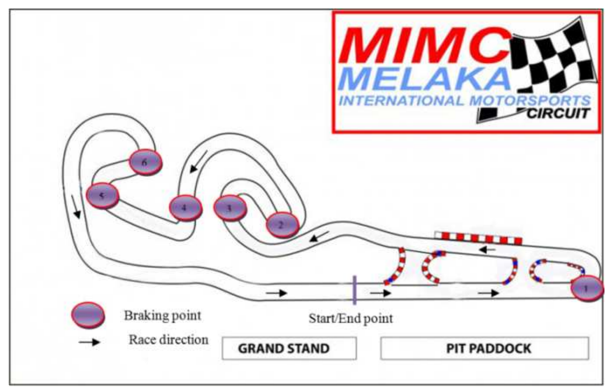

Fig. 3. Braking points based on race track layout

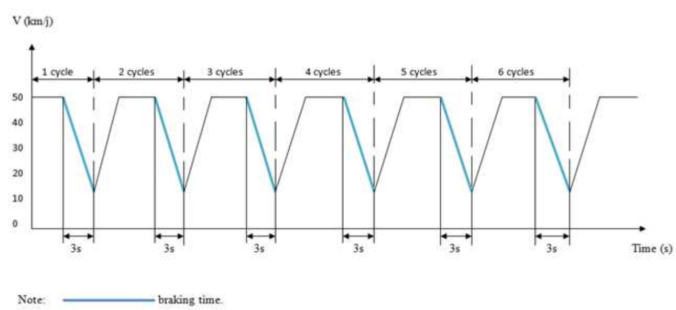

Fig. 4. Braking characteristics for the whole track

Throughout the repeated braking condition, the disc brake rotor is subjected to continuous heating and cooling process. During braking, frictional heat load is subjected to the rotor surface through conduction. After the brake is released, the rotor is then allowed to cool through convection process. The heat transfer process repeats until the end of the 6^{th} cycle. Table II and Fig. 5 below summarized the overall results of the load analysis conducted.

TABLE II
OVERALL RESULTS OF THE LOAD ANALYSIS

Description	Value
Braking Time, Δt	3 s
Kinetic Energy, ΔE	49268.55 J
Braking Energy, L	8129.31 J
Braking Surface, S_{flux}	38 x 10^{-3} m^2
Thermal Flow, q	1625.86 J / s
Heat Flux, $q_{specific}$	42785.8 W / m^2
Heat transfer coefficient by convection for cross-drilled holes, h_R **(A in Fig. 5)**	Reynolds number = 443.7 h_R = 87.65 W / m^2 K
Heat transfer coefficient by convection for side solid part, h_R **(B in Fig. 5)**	Reynolds number =27.3 x 10^3 h_R = 22.98 W / m^2K
Heat transfer coefficient by convection for outer solid part, h_R **(C in Fig. 5)**	Reynolds number = 27.3 x 10^3 h_R = 22.98 W / m^2K
Heat transfer coefficient by convection for esthetical design part, h_R **(D in Fig. 5)**	Reynolds number = 2307.83 h_R = 74.3 W / m^2K

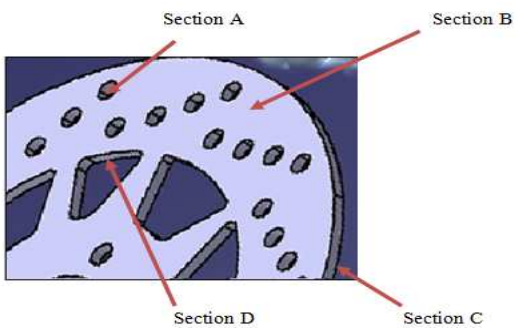

Fig. 5. Location of cooling characteristic on the disc brake rotor

To simplify the analysis, several assumptions have also been made as follows:-

i) All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux.
ii) The heat transfer involved for this analysis only conduction and convection process. This heat transfer radiation can be neglected in this analysis because of small amount which is 5% to 10% [10]
iii) In this analysis, the ambient temperature and initial temperature has been set to 25°C
iv) All other possible disc brake loads are neglected.
v) Only certain parts of disc brake rotor will apply with convection heat transfer such as the cross-drilled area, cooling vanes area, outer ring diameter area and disc brake surface

V. FINITE ELEMENT ANALYSIS

The performance of the UTeM Eco-Car disc brake system was analyzed using ABAQUS/CAE finite element analysis (FEA) software. The 3D modelling of the disc brake rotor which has been modelled using CATIA V5R16 software was imported into the FEA software as the input geometry for the analysis. The input geometry was later meshed using 4-node linear tetrahedron mesh element (ABAQUS: DC3D4) using global seed setting of four (4) as shown in Fig. 6 below. The meshed geometry resulted with a total of 27,953 numbers of elements. The brake heat flux and convection heat transfer coefficient values determined in the earlier load analysis stage were used to define the load and boundary conditions for the analysis. Finally, linear thermal transient analysis was performed in this stage to simulate the repeating braking process of the disc brake system.

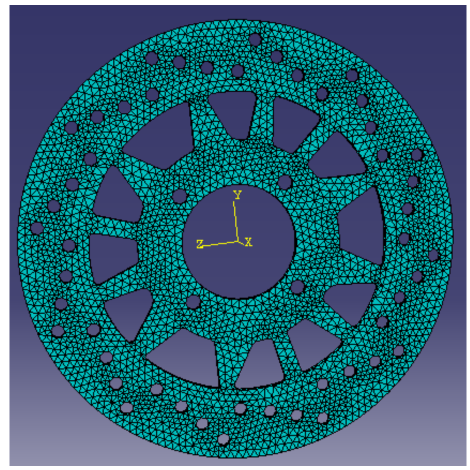

Fig. 6. Meshed model of the disc brake rotor

VI. RESULT AND DISCUSSIONS

In practical, typical service condition for automotive brake discs consist of frequent and quick engagement cycles, therefore making it predominantly in transient scheme. Results from transient analysis are often more desirable in predicting the actual performance of the disc during operation [15]. Result of the finite element thermal analysis for the disc brake rotor in transient condition is shown in Fig. 7 below.

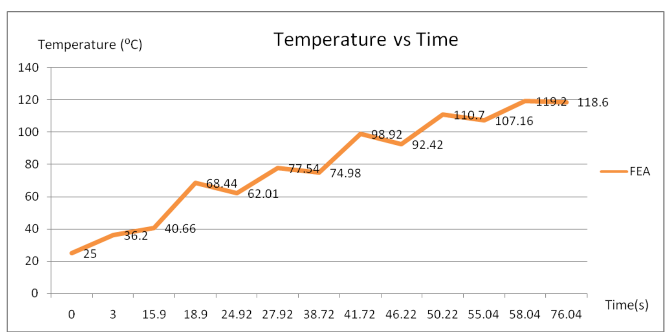

Fig. 7. Temperature increase due to braking on the disc brake rotor surface for the entire track

The above graph shows the behaviour of the disc brake rotor during continuous braking, where the surface temperature on the disc surface increased as the number of subsequent braking is increased. It can also be seen that the maximum temperature generated on the disc surface due to the repeated braking (for the whole race track) is 119.2°C which is within the allowable service temperature of the rotor material which is between -150°C to 550°C. This shows that the disc brake selected for UTeM Eco-Car is able to perform successfully without failure due to overheating.

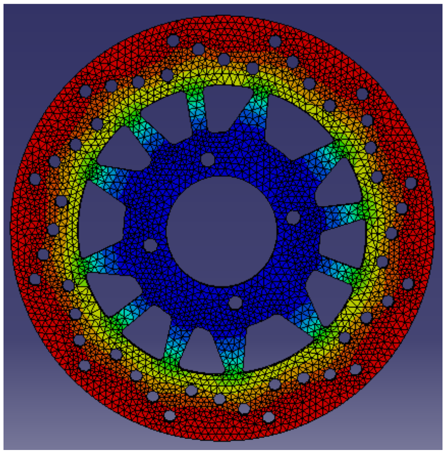

Fig. 8. Temperature distribution on the disc surface at time= 76.04s (6th braking cycle)

A part from the disc critical surface temperature result, thermal analysis conducted also reveals the temperature distribution on the disc surface when braking load is applied as shown in Fig. 8 above. Temperature contour shows the area where the location of the maximum and minimum temperature occurred, which varies throughout the disc surface. The maximum temperature generated is at the exact point of contact between the disc brake calliper and the disc rotor, and the temperature gradually decreases along the surface until it reaches the disc rotor hub.

VII. Result Validation

The results obtained through the finite element analysis were later validated by adapting equation from Budynas and Nisbett [16]. The temperature increase at the end of each braking cycle was determined using (1) while the temperature decrease at each braking cycle due to release of brake (cruising) was determined using (2).

$$\Delta T = \frac{E}{C_p m} \qquad (1)$$

$$\frac{T - T_\infty}{T_i - T_\infty} = \exp\left(\frac{-h_{cr}.A.t}{C_p m}\right) \qquad (2)$$

where

ΔT = temperature rise (^{o}C)
E = braking energy (J)
T = temperature at time (^{o}C)
T_i = initial temperature (^{o}C)
T_∞= environment temperature (^{o}C)
A = surface area (m^2)
m = rotor mass (kg)
C_p = specific heat capacity of the rotor (J/kg.^{o}C)
h_{cr} = overall coefficient of heat transfer (W/m^2.^{o}C)

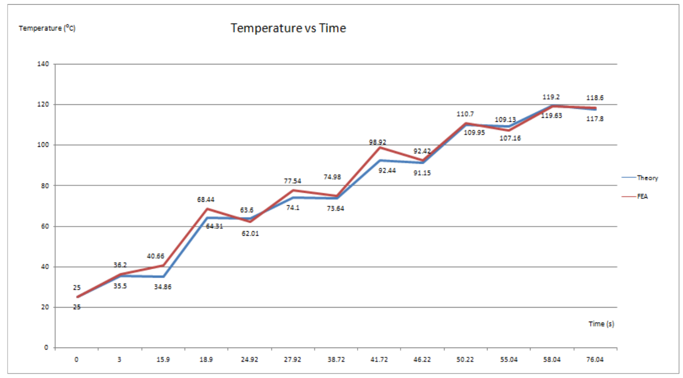

Fig. 9. Validation result for disc brake rotor temperature profile.

As shown in Fig. 9, the thermal analysis results obtained from finite element method showed very good agreement when compared with the results obtained through analytical method performed. At the end of the 6th cycle, the maximum temperature generated at the disc surface found through analytical method was 119.6^{o}C compared to maximum surface temperature of 119.2^{o}C found through finite element method. The results of the simulation are proven to be accurate as the difference between the maximum temperatures analytical is smaller. Based on the validation results, the performance of the brake disc has been successfully predicted and achieved.

VIII. Conclusion

In conclusion, thermal analysis for the braking performance of the UTeM Eco-Car disc brake system shows that the maximum temperature generated on the disc surface due to the repeated braking is 119.2^{o}C which is within the allowable service temperature of the rotor material which is between -150^{o}C to 550^{o}C. The simulation result indicates that the UTeM Eco-Car disc brake system is able to perform safely in real track condition as per design requirement.

IX. Recommendation

In this project, the behaviour of the disc brake system in term of heat properties was determined. Thus, for further work, the behaviour in term of the disc brake strength properties during the braking process can also be analyzed using both analytical and finite element method. The properties of the stress and deflection of the component due to the heat generated can be determined where both results are very useful to predict the disc brake performance in term of the mechanical strength. Therefore, the overall performance of the system can be evaluated more thoroughly both using the maximum temperature and maximum stress failure criteria and able to raise the level of confidence for the system design.

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