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Design and Development of Thai Railway Brake Disc under Temperature Analysis

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Abstract*— The railway transportation pervasively used around the world which also has been using in Thailand. Wherewith, the railway is low cost public transportation and low rate of accident. However, this railway is a mass transportation which can cause enormous damage to life and property. Several causes of accident such as dilapidated rail, the railway drive high speed, and the wheel tire takes off from the wheel set under the brake application etc. One cause of such a problem might come from using shoe brake, which pushes against the wheel tire. This situation creates high temperature and wearing on the wheel tire resulting in splitting off the wheel set. For these reasons, the basic development of railway brake system is considered to retrofit the original brake system with the disc brake system wherewith design of brake disc dimensions are based on the torque requirements from Thai railway locomotive at the initial speed 90 km/hr before brake application. Using mathematical method the brake disc temperature is investigated based on the conditions in which the brake application and cooling period are considered between the stations. The shortest distance between stations in Thailand is 300 meters long. In addition, the distance between stations, which are 900 and 1500 meters long are used to investigate the temperature responses of brake disc for the railway locomotive in Thailand. As a result, the outside and inside diameters of the brake disc are 698 and 280 mm with the 25 mm thickness with the brake pad area with 40200 mm^2

Index Terms — **Railway Brake System, Brake Disc Temperature, Railway Locomotive, Temperature Analysis.**

I. INTRODUCTION

Globally, the railway performance has been continually developed in various components such as power train and suspension including the safety system since 16th century B.C. In Thailand, State Railway of Thailand (SRT) has been established in 1890 , But Thai railway development is gradually increased comparing to the railway transportation from the developed countries. In addition, the maintenance of the railway system is the first majority of State Railway of Thailand to keep the working condition up to the present time. However, the current technology has been developed especially for the brake system in order to improve the safety of railway and prevent detrainment of railway. With such technology, the cause of such detrainment can be eliminated through the retrofit of original brake shoes applying on the wheel tires with brake disc system as shown in Figure 1 [1]. Such brake disc system, the wear of wheel tire can be largely reduced comparing to the original brake system which can cause the splitting off the wheel tire from the wheel. With this technology, the railway performance can be improved in term of safety. For these reasons, the objective of this research is to develop and retrofit the basic development of railway brake system with brake disc system using mathematical method. In such method, the brake disc temperature can be investigated based on the conditions in which the brake application and cooling period are considered between various stations in Thailand.

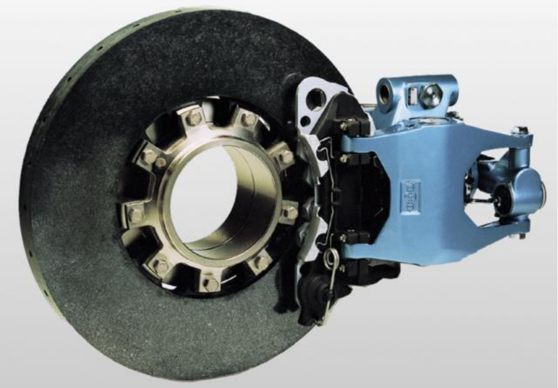

Fig. 1. Retrofit of the Brake Shoe with the Brake Disc System [1]

At first, the diameter of brake disc is designed from the torque requirements of Mechanical Engineering Centre of SRT [2] under current parameters such as braking force of shoe brake, brake pressure, etc. With such requirement, the railway locomotive should run at the initial velocity 90 km/hr before the brake application in order to obtain the brake torque and diameter as shown in Equations 1 and 2. As a result of optimization from the train wheel axle, the outside diameter of brake disc (r_o) is 698 mm and this outside diameter requires the inside diameter (r_i) 280 mm based on the standard and common brake-design parameters for the railway in Thailand [2].

$$T_{brake} = F_N \times \mu_{shoe} \times r_{wheel} \qquad \text{-----------------------------------(1)}$$

$$r_o = \sqrt[3]{\frac{270^\circ \times T_{brake}}{P_{brake} \times \pi \times \theta_A \times \mu_{disc}} - r_i^3} \qquad \text{-----------------------------------(2)}$$

Where

T_{brake} = Braking torque (N.m)
F_N = Braking force of shoe brake (35,144 N)*
r_{wheel} = Wheel tire radius (0.428 m)*
μ_{shoe} = Friction coefficient between wheel tire and brake shoe (0.168)*
P_{brake} = Brake pressure (380,000 Pa)*
μ_{disc} = Friction coefficient between brake disc and brake pad (0.28)**
θ_A = Degree of brake pad area (45 degree)***

* Mechanical Engineering Centre of SRT [2]
** Asia Compact Company [3]
*** Initial Assumption

II. ANALYTICAL THEORY

A. *Brake Application*

During the brake application, heat flux is generated on the sliding surface between brake disc and pad which are calculated from the braking torque. Generally, energy from brake application is converted into the heat flux and transferred to the disc and pad approximately 95% and 5% respectively [4]. This ratio normally is called the proportion of heat transferred to disc. Based on analytical method, some of heat flux generated from the pad and rotor during the brake application can be determined from the physical parameters as shown in Equation 3. This heat flux is also equated to the conduction heat to the rotor in which the temperature profile can be determined as shown in Equation 4.

$$H = \frac{K_D \times T_{brake} \times \omega_{wheel}}{2 \times A_{disc}} \qquad \text{-----------------------------------(3)}$$

$$H = -k \left.\frac{\partial \mathrm{T}}{\partial x}\right|_{\mathrm{x=0}}^{\mathrm{x=L}} \qquad \text{-----------------------------------(4)}$$

Where

H = Heat flux (W/m^2)
A_{disc} = Sliding surface of brake disc (0.321 m^2 under r_i = 0.28 m, r_o = 0.698 m)
K_D = the proportion of heat transferred to disc (0.95)
L = Brake disc thickness (m)

In theory, the temperature of the brake disc can be simplified and derived from the one dimensional heat equation as shown in Equation 5

$$\frac{\partial}{\partial x}\left(K \frac{\partial T}{\partial x}\right) + G = \rho c_p \frac{\partial T(x,t)}{\partial t} \qquad \text{-----------------------------------(5)}$$

Where

T = Temperature (°C)
x = Thickness of material (m)
ρ = Density (kg/m^3)
c_p = Specific heat (J/kg.K)
K = Thermal conductivity (W/m.K)
t = Conduction period (sec)
G = Internal heat generation (W/m^3)

B. Cooling Period

Under the analysis condition for the cooling period, the railway is accelerated up to the maximum speed of 90 km/hr and then the railway speed is maintained for the certain period. This period in which the heat energy can be transferred to the ambient through convection can be calculated from the time and distance between stations. After the cooling period, the next brake application is taken before the destination station. To determine heat convection coefficient during the cooling period, the dimensionless parameters are used such Nesselt (Nu) and Raynold (Re) numbers together with the air speed factor (k_v = 0.12) [5].

$$h = \frac{Nu \times k_{air}}{r_o} \qquad \text{-----------------------------------(6)}$$

$$Nu = 0.037 \times \mathrm{Re}^{0.8} \qquad \text{-----------------------------------(7)}$$

$$\mathrm{Re} = \frac{V_{air} \times r_o}{\upsilon_{air}} \qquad \text{-----------------------------------(8)}$$

$$V_{air} = k_v \times V_{rail} \qquad \text{-----------------------------------(9)}$$

Where

h = Heat convection coefficient ($\frac{w}{m^2 \cdot k}$)
Nu = Nusselt number
k_{air} = Conductivity of air (0.0241 $\frac{w}{m \cdot k}$) [6]
V_{air} = Air speed (m/s)
V_{rail} = Railway speed (m/s)
υ_{air} = Kinematic viscosity of air (12.34×10^{-6} m^2/s) [7]

To investigate and simplify the heat convection transferred from the brake disc during the cool period, the one dimensional heat equation together with uniform heat convection is used. This uniform heat convection is applied to both sides of brake disc as shown in Figure 2

$$h_1[T_1 - T(x,t)\big|_{x=0}] = K\frac{\partial T(x,t)}{\partial x} \qquad \text{--------------------------------(10)}$$

$$h_2[T_2 - T(x,t)\big|_{x=L}] = K\frac{\partial T(x,t)}{\partial x} \qquad \text{--------------------------------(11)}$$

The equations of uniform heat convection transferred to the ambient are shown in Equation 10 and 11 that are implied for both sides of the brake disc surface. To obtain the temperature responses from these equations during the cooling period and from those equations during the brake application, the freeware solver based on one dimensional heat transfer program is used throughout analysis of brake disc for the railway. [8]

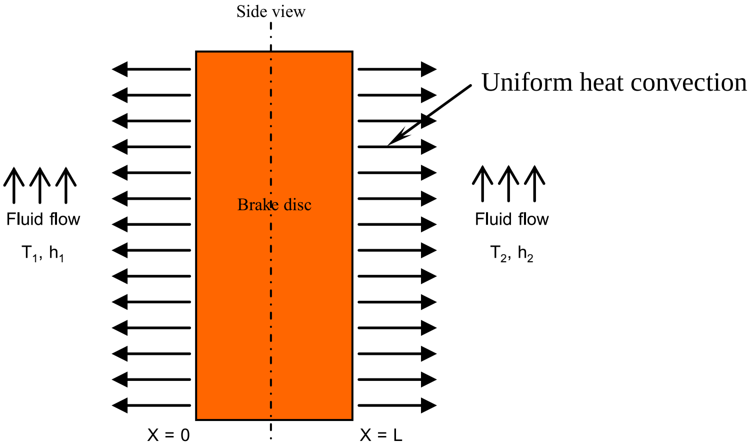

Fig. 2. Convection Heat Transfer

III. METHODOLOGY

A. *Determination of Brake Disc Thickness from SRT Requirement*

To obtain the thickness of brake disc and temperature characteristics, the mathematical method based on the brake application and cooling period are used to investigate the maximum and gradient temperature responses between the stations. Maximum velocity of 90 km/hr is used to determine heat flux applying the brake disc. As a result from Equation 1 and 3, it is found that the maximum torque is 2,527.16 N.m and the uniform heat flux is 359,349 W/m^2 under such SRT conditions. In addition, the brake disc temperature responses under uniform heat flux with various thicknesses of brake disc i.e. 20, 25, 30 mm are investigated using the one dimensional heat transfer program with the cast iron as shown in Table 1.[9]

Table. 1. Properties of Cast Iron Tensile Strength 220 N/Mm^2 [10]

Properties	
Density, ρ (kg/m^3)	7150
Thermal conductivity, K (W/m.K)	48.6
Specific heat, C_p (J/kg.K)	465

As a result, it is found that the optimum 25 mm thickness with 40200 mm^2 of the brake pad area provides the maximum temperature of 332 degree Celsius at brake disc surface. However, the temperature responses under brake application and cooling period between stations are subjected to investigate in the following section. Therefore, braking and cooling travel time between stations are firstly investigated in order to allocate heat flux and cooling periods.

B. *Investigation of Braking and Cooling Times under Operations*

When distance between stations in the capital of Thailand is compared, the shortest distance between stations in Thailand is 300 meters long. In addition, the longer distance between stations which are 900 and 1500 meters long are used to investigate the temperature responses of brake disc for the railway locomotive in Thailand. In addition, Thai railway locomotive can typically accelerate up to 0.7 m/s^2 in which the three different acceleration conditions are determined at 0.3, 0.5, and 0.7 m/s^2 for analysis. Under braking condition, the maximum deceleration from the railway locomotive in Thailand is at 0.739 m/s^2. Therefore, the duration time under various station distances are calculated from equation of motion and summarized as shown in Table. 2

Table. 2. Time Period between Stations under Various Accelerated Conditions

Case	**Travel Conditions between stations**		
	Acceleration Time (t_a)	**Constant speed Time** (t_v)	**Braking Time** (t_d)

Short distance	$\sqrt{\dfrac{s_{all}}{\frac{1}{2}\vec{a}_a - \frac{\vec{a}_a^2}{\vec{a}_d} + \frac{1}{2}\frac{\vec{a}_a^2}{\vec{a}_d}}}$	–	$-\dfrac{\vec{a}_a}{\vec{a}_d}t_a$
Long distance	$\dfrac{v_a}{\vec{a}_a}$	$\dfrac{s_{all} - \frac{v_a t_a}{2} - u_d t_d + \frac{u_d t_d}{2}}{v_a}$	$-\dfrac{u_d}{\vec{a}_d}$

To investigate travel time and velocity under various conditions, the distance (s_{all}), acceleration ($\vec{a}_a$), deceleration ($\vec{a}_d$), and maximum accelerated velocity (v_a) at 90 km/hr, initial decelerated velocity (u_d) at 90 km/hr into equations in Table 2 are initially substituted. As a result, relationship between travel velocity and time under three different acceleration and station-distance conditions are shown in Figure 3, 4, and 5.

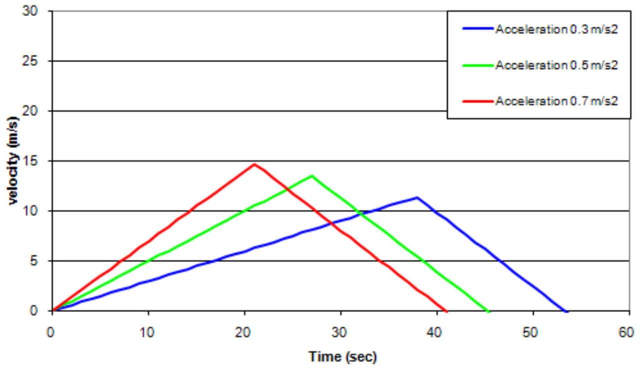

Fig. 3 Time and Velocity Relation Under 300 Meter Distance

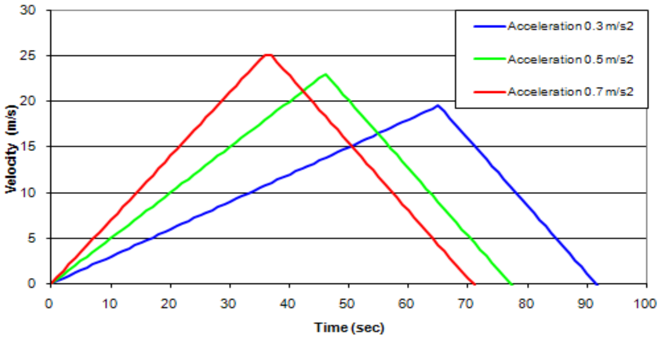

Fig. 4 Time and Velocity Relations Under 900 Meter Distance

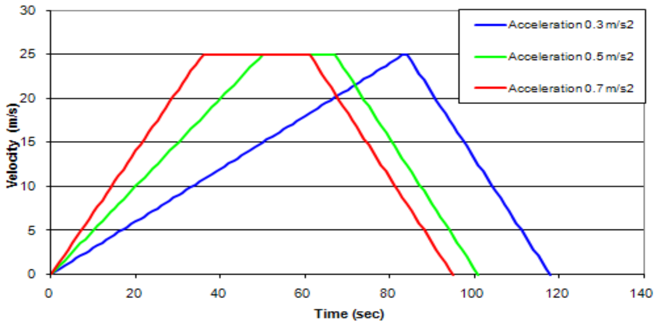

Fig. 5. Time and Velocity Relations Under 1500 Meter Distance

In each acceleration condition, period of time for one unit of speed increment at 1 m/s can be calculated from the equation of motion as shown in Table 3. The period of time for such speed increment is used to determine the heat flux and convection for each speed increment under various acceleration conditions.

Table. 3. Period of Time under One Unit of Speed Increment at 1 M/S

Acceleration/Deceleration (m/s^2)	Period of time (sec)
0.3	3.33
0.5	2.00
0.7	1.43
-0.739	1.35

C. Load Conditions under Braking and Cooling Periods

From previous analysis, the constant heat flux is used to investigate the brake disc response under variation of disc thickness. Such load condition is unable to be used to investigate temperature response of brake disc between stations since the velocity of brake disc under brake application is decreased resulting in the reduction of heat flux. Similarly, during the cooling period under deceleration, the heat convection coefficient is gradually decreased depending on the railway speed. Therefore, the velocity dependent heat flux and convection loads should be determined. To obtain these, the average heat convection under velocity increment by 1 m/s and average heat flux under velocity decrement by 1 m/s between stations are calculated as shown in Equation 12 and 13. Based on such conditions, maximum speed of Thai railway locomotive is 90 km/hr or 25 m/s. The other maximum railway speeds of various accelerations between stations are shown in Figure 3-5. These velocities can be used to determine average heat flux and convection of brake disc. Thus, Figure 6 shows the average heat convection values when the velocity is changed from v_{N-1} to v_N. Similarly, the average heat flux values from v_N to v_{N-1} are shown in Figure 7.

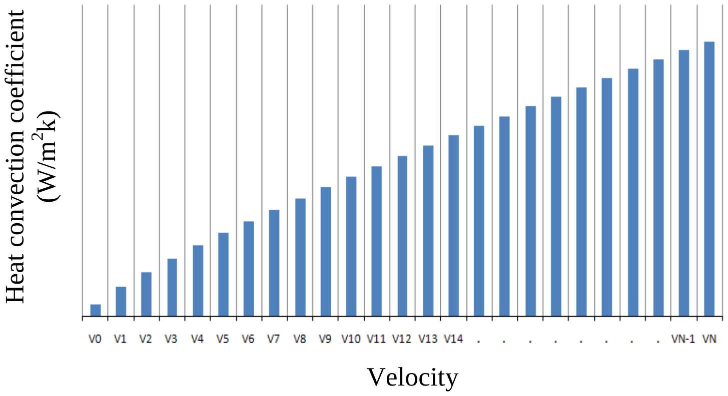

Fig. 6. Average Heat Convection Coefficient under Velocity Increment by 1 M/S

The average heat convection coefficient under cooling period takes the following equation.

$$h_{ave} = \frac{h_{V_N} + h_{V_{N-1}}}{2} \qquad \text{---------------------------------(12)}$$

Where

h_{ave} = Average heat convection coefficient)W/m^2)
h_{V_N} = Heat convection coefficient at V_N (W/m^2)
h_{V_N-1} = Heat convection coefficient at V_{N-1} (W/m^2)

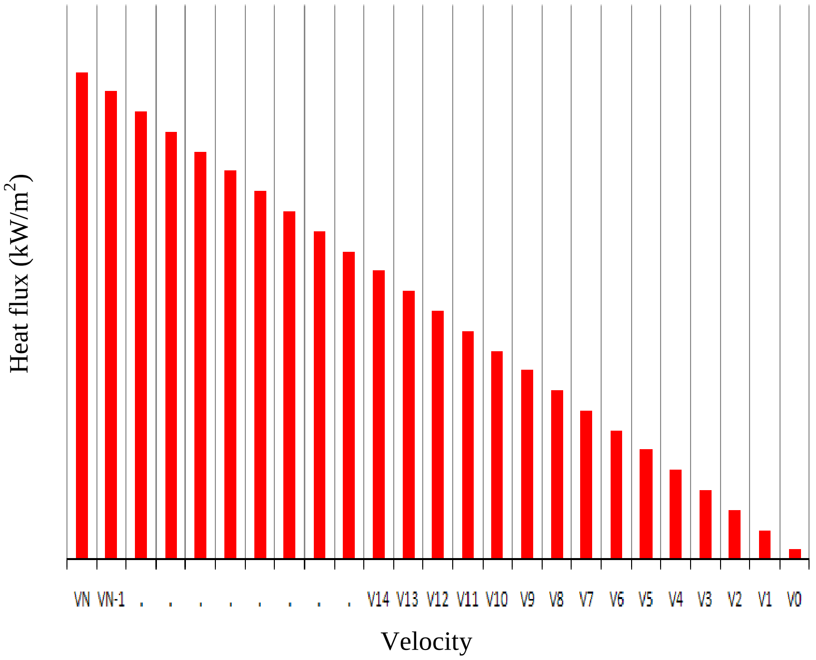

Fig. 7. Velocity dependent Heat flux

The average heat flux of brake application takes the following equation.

$$H_{ave} = \frac{H_{V_N} + H_{V_{N-1}}}{2} \qquad \text{--------------------------------(13)}$$

Where

H_{ave} = Average Heat flux (W/m^2)
H_{V_N} = Heat flux ที่ V_N (W/m^2)
$H_{V_N - 1}$ = Heat flux ที่ V_{N-1} (W/m^2)

D. Temperature Responses of Brake Disc between Stations

To determine temperature response of brake disc between stations, the initial brake disc temperature at 332°c obtained from previous analysis is used as starting value for the start station at zero velocity. Then the locomotive is accelerated from 0 m/s to 1 m/s. Such time increments for various accelerations are 3.33, 2.00 and 1.43 second as shown in Table 3. These values are used to calculate the average heat convection coefficient. For example, the average heat convection coefficient becomes 0.98 W/m^2.K when velocity increases from 0 m/s to 1 m/s at 0.3 m/s^2 acceleration which takes 3.33 second. Therefore, the calculation of brake disc temperature response on sliding surface under heat convection coefficient of 0.98 W/m^2.K and initial temperature at 332°c is performed using 1D heat transfer program.

Then the resulted temperature of such average heat flux is used at initial brake disc temperature for next average heat convection when velocity changes from 1 m/s to 2 m/s. This process is repeatedly used to determine brake disc temperature response between stations. During stations, the cooling period will change to the braking period when locomotive velocity reaches to the maximum value as shown in Figure 3-5 which vary on the station distances and acceleration values.

As a result, brake disc temperature responses of sliding surface can be determined throughout from the start to final stations under various acceleration conditions as shown in Figure 8-10.

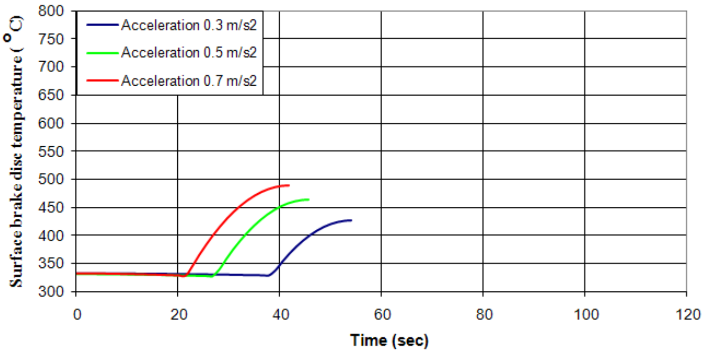

Fig. 8. Brake Disc Temperature Responses under 300 Meter Station Distance

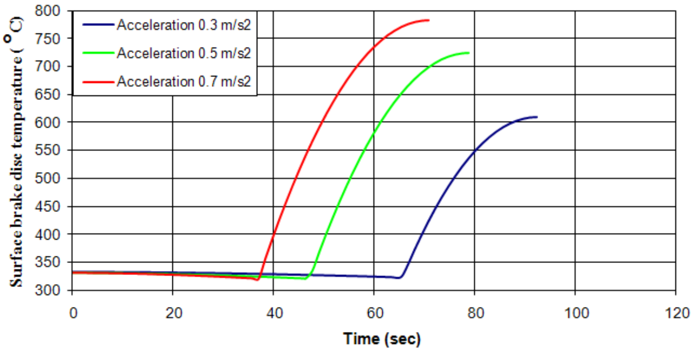

Fig. 9. Brake Disc Temperature Response Under 900 Meter Station Distance

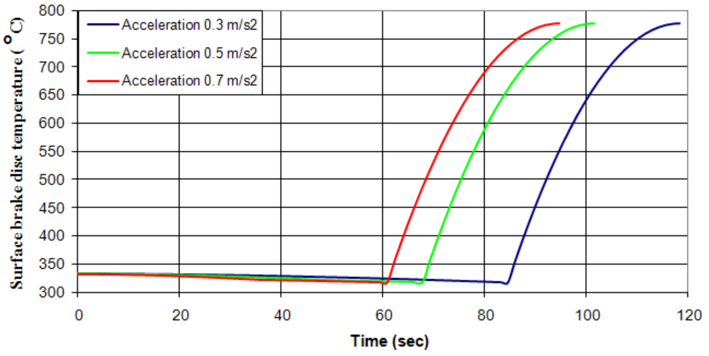

Fig. 10. Brake Disc Temperature Response Under 1500 Meter Station Distance

Consequently, the temperature response under acceleration 0.7 m/s^2 provides the best cooling performance comparing to that for 0.5 and 0.3 m/s^2. Then, rapidly increment of velocity causes better cooling condition. But, when the brake application is started before reaching the final destination station, the temperature response is the highest comparing to the other acceleration conditions. This is due to higher velocity, resulting in more heat flux application to the brake disc as shown in Figure 10 and 11.

IV. CONCLUSION

- One dimension heat transfer can be used to design brake disc geometry and determine temperature responses of brake disc under various load conditions and requirement conditions from State Railway of Thailand.

- The effects of station distance and acceleration conditions dominate the traveling time and cooling period and the braking period.
- Maximum speed of locomotive affects high temperature response of brake disc even though high speed causes high cooling performance of brake disc.
- In brake disc, heat conduction and convection occur at the same time. But, program can predict the temperature response only in one dimension. To increase more accuracy in temperature analysis, two or three dimension temperature analysis is suggested to predict more accurately.

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