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A study on an analysis model for the thermo-mechanical behavior of a solid disc brake for rapid transit railway vehicles †

Heerok Hong[1], Minsoo Kim[2], Hoyong Lee[2,*], Insub Jo[1], Naktak Jeong[1], Hyunguk Moon[1], Myungwon Suh[1] and Junghwan Lee[3]

[1]*School of Mechanical Engineering, Sungkyunkwan University, Gyeonggi 440-746, Korea*
[2]*Metropolitan Transportation Research Center, Korea Railroad Research Institute, Gyeonggi 437-757, Korea*
[3]*Department of Automotive Engineering, Osan University, Gyeonggi 447-749, Korea*

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Abstract

One of the primary interests in the railway field is to improve speed as well as to secure the safety of railway vehicles at a high speed. In light of this, the demand for braking systems that offer improved performance and safety has also been growing consistently. The braking system is a critical element directly connected with safety. In order to secure reliability and establish the maintenance standard for the braking system, it is necessary to secure an accurate evaluation technology. As the braking disc of the braking system is very sensitive to safety and maintenance, a method to evaluate thermal cracking must be developed. Because the brake disc goes through a repeated process of heating and cooling due to the friction energy and the heat convection, respectively, in every braking action, it is subject to thermal fatigue. As well, the partial contact area caused by friction between disc and pad during the braking process results in thermal distortion, causing a hot band and a hot spot. This thermal distortion may also cause thermal cracking on the friction surface. The aim of this research was to find the cause of thermal cracking through a thermo-mechanical friction analysis of disc and pad, and to verify the analysis model through the dynamo machine.

Keywords: Solid disc; Brake pad; Finite element analysis; Brake dynamometer; Thermo-mechanical behavior; Temperature field

1. Introduction

The railway, an eco-friendly means of transportation for the 21st century, serves as a bridge between cities. Recently, interest in improving the speed of railway vehicles has been growing due to the economic benefits this would offer. At the same time, there has been an increased demand for improvements in the performance and safety of the braking system, to guarantee the safety of railway vehicles at high speeds. In this climate of high power and high speed railway vehicles, the braking system must meet the required braking distance in a more demanding environment. As braking is a critical element that is directly connected with safety, it is very important to secure the reliability of the braking system, and to establish the related maintenance evaluation technology [1].

Because the brake disc goes through a repeated cycle of heating and cooling due to the friction energy and the heat convection, respectively, in every braking action, it is subject to thermal fatigue. Also, the partial contact area caused due to the heat that is generated results in thermal distortion, causing a hot spot. The term ‘hot spot’ indicates a high thermal gradient at the friction surface. Hot spot and thermal fatigue may cause thermal cracking on the friction surface of the disc [2, 3]. Thermal cracking on the brake disc reduces the life of the disc and increases maintenance expenses, and may result in an accident caused by damage to the disc. Therefore, preventing thermal cracking is a critical technology in terms of operation safety and reduction of maintenance expenses [4].

Many investigations into brake disc thermo-mechanical coupling analysis have been carried out. Dufrenoy and Weichert analytically implemented irregular contact between a brake disc and a pad pin by applying load to the pad under the system where the rear side of the pad is dovetail-fixed. They analyzed the thermal behavior of the disc by applying heat flux on the friction surface rather than conducting a contact analysis between the disc and the pad [5]. Gao and Huang et al. estimated the friction heat flux between the active area of the disc and the pad by using a 3D thermal-structure coupling model, and carried out an analysis in consideration of the change in RPM during the braking action. Through analysis and interpretation, they found that a hot band was created due

*Corresponding author. Tel.: +82 31 460 5715, Fax.: +82 31 460 5028
E-mail address: hylee@krri.re.kr
† Recommended by Associate Editor Jaewook Lee

to the interaction between friction heat, thermal strain and elastic contact [6]. Ali Belhocine and Mostefa Bouchetara found the heat flux, and by using ANSYS CFX, analyzed the thermal behavior of various types of cast iron in the braking mode. Through a numeric simulation, they found that radial ventilation had a considerable influence on the cooling of the disc in the braking stage [7]. Ghadimi and Kowsary et al. found, through experiments, that in a complex architecture of a ventilation-type disc, the sectional heat transfer coefficient distribution changed. They also calculated the heat transfer coefficient value using FLUENT CFD. Following this, they calculated the sectional heat transfer coefficient value according to the speed and temperature distribution, applied the value as the boundary condition to the heat analysis of the brake disc, and calculated the heat flux for analysis [8]. Li and Han et al. carried out a braking test with a full-scale dynamo testing machine in order to test the emergency braking process. They found that it is possible to reflect the influence of failure behavior of the disc through temperature and stress distribution with the FE simulation only based on the change in the microstructure of the brake disc material and the LCF behavior data [9]. Grzes and Oliferuk et al. suggested that the thermal diffusivity of the disc material should be measured with an infrared thermograph and introduced an analysis in order to develop an FE model that conforms to the result of their experiment [10].

Most of the previous studies have analyzed the thermal behavior of the brake disc by estimating heat flux rather than by using the thermo-mechanical contact analysis of the disc and the pad. This type of analytical approach reduces the cost of analysis and shows uniform temperature distribution and stress, but results in significant differences from actual braking conditions. In this study, a contact analysis between the disc and the pad was carried out to obtain a result that is similar to the actual braking conditions, the hot band caused by thermo-mechanical friction between the disc and the pad, and the analysis model was verified using the dynamic machine.

2. Formulation for brake disc

2.1 Heat flux for thermal load

The kinetic energy of a moving vehicle is converted to thermal energy due to friction at braking. 99 % of the thermal energy is released through the brake disc and the pad [11]. The following equation shows the kinetic energy of a running vehicle [8, 12]:

$$E_b = \frac{M}{2}\left(v_1^2 - v_2^2\right) + \frac{I}{2}\left(\omega_1^2 - \omega_2^2\right) \tag{1}$$

where M indicates the mass of a vehicle, v_1 indicates the initial speed, and v_2 indicates the later speed. I is the mass moment of inertia of the wheel, ω_1 is the initial angular velocity of the wheel, and ω_2 is the later angular velocity of the wheel. When a vehicle is stopped completely, because $v_2 = \omega_2 = 0$, the kinetic energy is expressed as follows:

$$E_b = \frac{M}{2}v_1^2 + \frac{I}{2}\omega_1^2 . \tag{2}$$

If the initial velocity $v_1 / r = \omega_1$ is applied, the overall kinetic energy is expressed as follows:

$$E_b = \frac{M}{2}(1 + \frac{I}{r^2 m})v_1^2 = \frac{kmv_1^2}{2} . \tag{3}$$

If the acceleration/deceleration a is constant, the vehicle velocity $v(t)$ is expressed as follows:

$$v(t) = v_1 - at . \tag{4}$$

The power generated in braking action equals the overall energy differentiated with time t, and is expressed as follows:

$$P_b = \frac{d(E_b)}{dt} = kma(v_1 - at) . \tag{5}$$

Therefore, if the acceleration/deceleration a is constant, the braking power is the highest at the initial stage (t = 0), and decreases linearly to 0 when the vehicle is completely stopped. Heat flux, which is the heating value per unit area on the friction surface of a single disc generated when a vehicle is completely stopped, is expressed as follows:

$$q''_d = kma(v_1 - at)\frac{1}{2AN} \tag{6}$$

where A is the cross-sectional area of the disc friction area, and N is the number of discs.

2.2 Thermal distribution rate and convective heat transfer coefficient

Friction heat is generated due to the friction between the disc and the pad upon braking of the railway vehicles, and is transferred to the disc and the pad, with the exception of the amount of heat cooled down in the air. If the cooling effect of air is ignored, the heat distribution rate is determined by the physical properties of the disc and the pad, and can be estimated with the following equation [8, 12]:

$$\gamma = \frac{1}{1 + \left(\frac{\rho_p c_p k_p}{\rho_d c_d k_d}\right)^{\frac{1}{2}}} \tag{7}$$

where ρ_d and ρ_p are the density of the disc and the pad, respectively, c is the specific heat, and k is the thermal conductivity.

The heat transfer coefficient of the disc surface is calculated

Table 1. Material properties and dimensions of disc and pad.

Thermal properties	Disc	Pad
Density [kg/m^3]	7850	5120
Elastic of modulus [GPa]	202	102
Poisson's ratio	0.3	0.25
Specific heat [J/kgK]	460	500
Thermal conductivity [W/mK]	45	24
Thermal expansion coefficient [1/K]	1.05×10^{-5}	1.67×10^{-5}

Table 2. Specifications of the railway vehicles.

Item	Spec.
Weight of railway vehicles [kg]	49060
Max. velocity [km/h]	300
Deceleration of railway vehicles [m/s^2]	1.25
Average friction coefficient	0.36
Initial temperature [°C]	60
Wheel diameter [mm]	920
Outer disc diameter [mm]	640
Inner disc diameter [mm]	350
Thickness [mm]	44.3
Diameter of pad pin [mm]	40

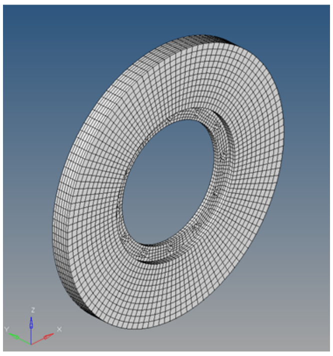


Fig. 1. FE model of brake disc.

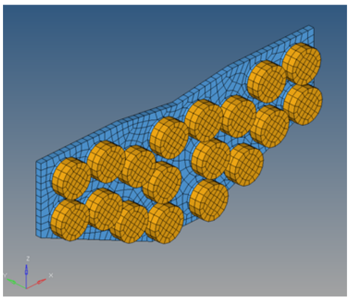


Fig. 2. FE model of brake pad.

with the following empirical equations for the Reynolds number of turbulent flow and laminar flow, respectively [8, 12]:

$$h = 0.04\left(\frac{k_a}{D}\right)\mathrm{Re}^{0.8} \qquad (\mathrm{Re} > 2.4\times10^5) \tag{8}$$

$$h = 0.70\left(\frac{k_a}{D}\right)\mathrm{Re}^{0.55} \qquad (\mathrm{Re} < 2.4\times10^5) \tag{9}$$

where k_a is the heat transfer coefficient of the air, D is the outer diameter of the disc, and Re is the Reynolds number. The Reynolds number is equivalent to $\rho_a v d_o/\mu_a$, where ρ_a is the density of the air, v is the velocity of the railway vehicles, d_o is the outer diameter of the disc, and μ_a is the coefficient of viscosity of the air. Table 1 shows the material properties of the disc and the pad, and Table 2 shows the specifications of the railway vehicles.

3. Comparison of thermal behavior of the brake disc for thermal load and thermo-mechanical load

3.1 Structure of analytical model for thermal load

The braking disc for the high-speed railway vehicles used in this study is a solid type disc made with forged steel. Fig. 1 shows the 3D model used in the analysis. The model used for analysis is the disc brake applied in KTX trains. 4 disc brakes are installed per axle. Fig. 2 shows a studded brake pad. The 3D models of the disc and the pad are the 3D geometry solid models established with pro-engineer. Using HyperMesh preprocessor and the hexahedron solid element (C3D8RT) for thermo-mechanical analysis, the finite-element models were made with 10168 nodes and 7232 elements. Simulation was carried out using the heat transfer analysis function of the ABAQUS/Standard finite-element analysis program. In order to simplify the analysis model, the disc was analyzed without the joint with bogie axle and the pad. KTX was set to the maximum speed of 300 km/h, and the analysis time was set as 67 sec, which was the KTX speed divided by the acceleration/deceleration. The initial temperature was set to 60 °C according to the test conditions. The heat flux was calculated with the conditions in Table 2 applied to Eq. (6), and was applied to the friction surface where the disc meets the pad. The value of heat flux is 820 kW/m^2. Then, the heat flux was assumed to decrease linearly over time. For all surfaces other than the friction surface to which the heat flux was applied, the convective heat transfer coefficient depending on the speed was calculated using Eqs. (8) and (9). Fig. 3 shows the change of the convective heat transfer coefficient according to the braking time. The convective heat transfer coefficient is decreased linearly, assuming that the deceleration rate is constant. At the speed of 300 km/h, the convective heat transfer coefficient is 273 W/m^{2}K.

3.2 Structure of analytical model for thermo-mechanical load

UIC 541-3 E.2: Test programme No. 5 was referred to in order to simulate the braking action of a high-speed railway

Table 3. Vehicle speed and braking force following the UIC standard.

	Step-1 brake (Service / emergency)	Step-2 brake (Service / emergency)
Speed [km/h]	300 ~ 215	215 ~ 0
Initial temperature [°C]	60	-
Braking force [kN]	14 / 19	18 / 25

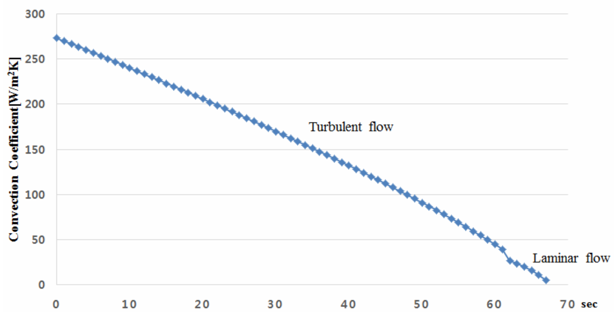


Fig. 3. Change of convective heat transfer coefficient according to braking time.

vehicle's brake disc [13]. In this standard, the braking performance is evaluated for a high-speed railway vehicles with the maximum speed of 320 km/h, and the initial temperature is set to 50~60 °C for every experiment. Therefore, step 1 braking force must be applied to the vehicle from 300 km/h to 215 km/h, and step 2 from 215 km/h to 0 km/h. Braking force is classified into service braking force and emergency braking force. Emergency braking force (19 kN / 25 kN) was adopted in this study for analysis and experiment. Table 3 shows the vehicle speed and the braking force according to the UIC standard.

Simulation was carried out with the coupled temp-displacement analysis function of ABAQUS/Explicit. To reduce the time required for analysis, an axial symmetrical boundary condition was given. On the assumption of constant acceleration/deceleration, the speed was assumed to decrease linearly from 300 km/h. The initial temperature was set to 60 °C according to the test conditions, and in order to consider the rotational inertia of disc, mass of 4000 kg (axle load per disc) and inertia of 792 kgm^2 were applied. Because there is no displacement other than the axial rotation of the disc, all degrees of freedom other than the axial rotation were restricted. For all surfaces of the disc and the pad, the convective heat transfer coefficient according to the speed was calculated using Eqs. (8) and (9). Fig. 4 shows the conditions for thermo-mechanical analysis.

The braking force required to stop the disc was converted into pressure and applied to the studs of the pad. By using Eq. (7), the thermal distribution rate caused by the contact between the disc and the pad is 0.61 for the disc and 0.39 for the pad. The friction coefficient between the disc and the pad was taken from the disc test standard UIC 541-3 E.2: Test programme No. 5. The friction coefficient changes depending on the speed. For efficiency of analysis, a constant nominal value

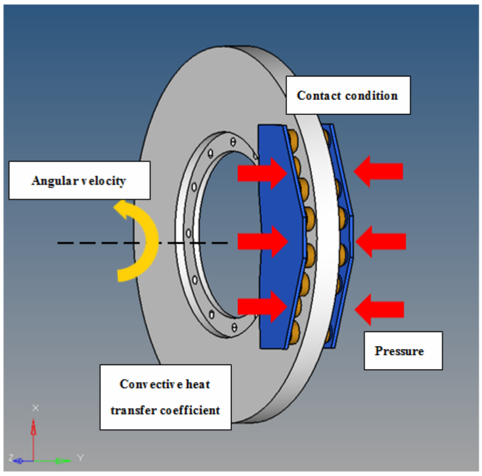
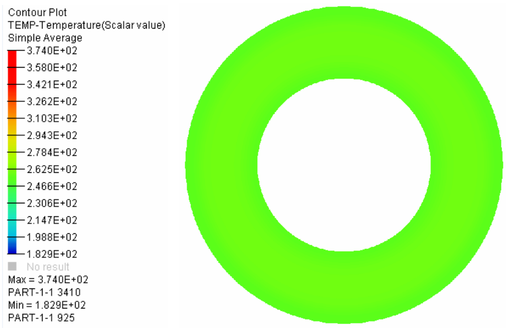
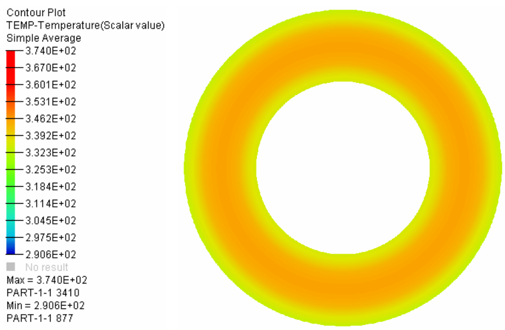


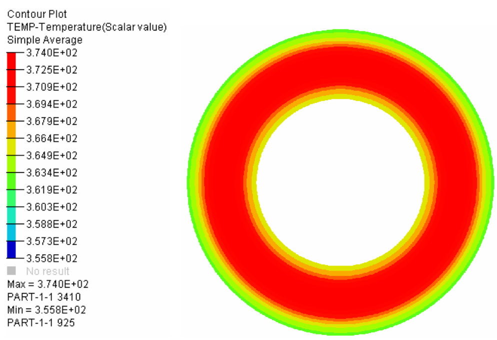
Fig. 4. Boundary condition of thermo-mechanical analysis.



(a) For 20 seconds

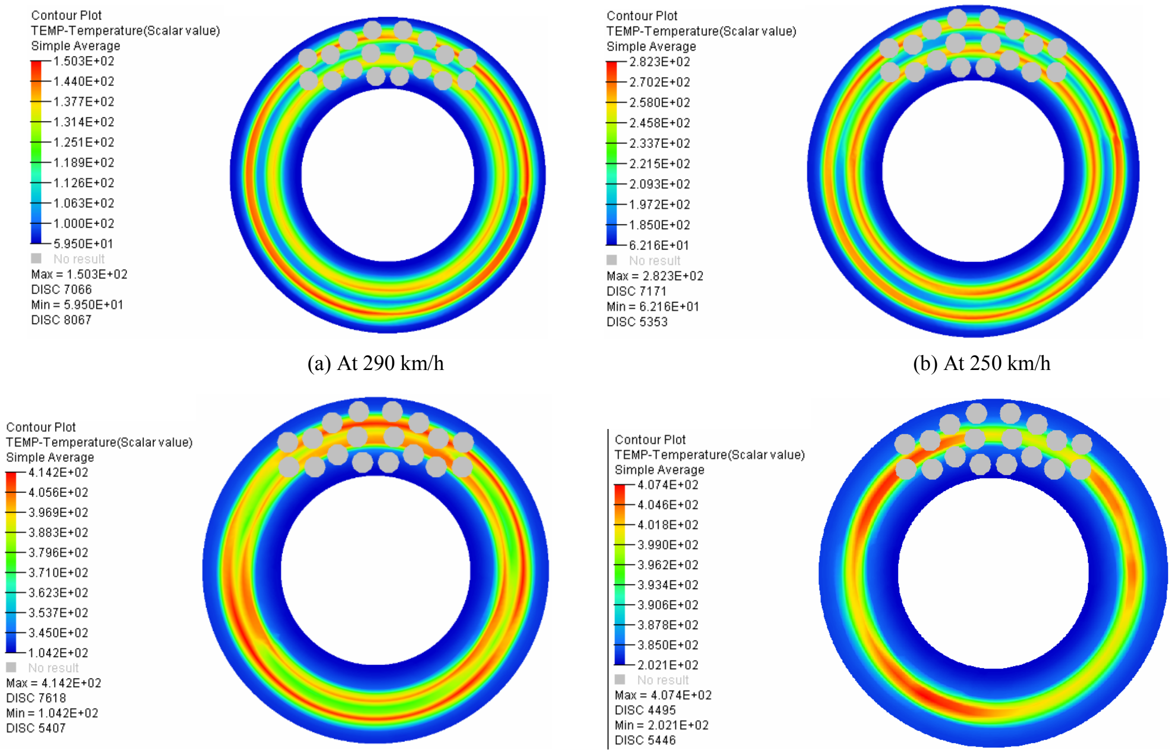


(b) For 40 seconds



(c) At the end of braking action

Fig. 5. Temperature distribution of solid disc due to thermal analysis.



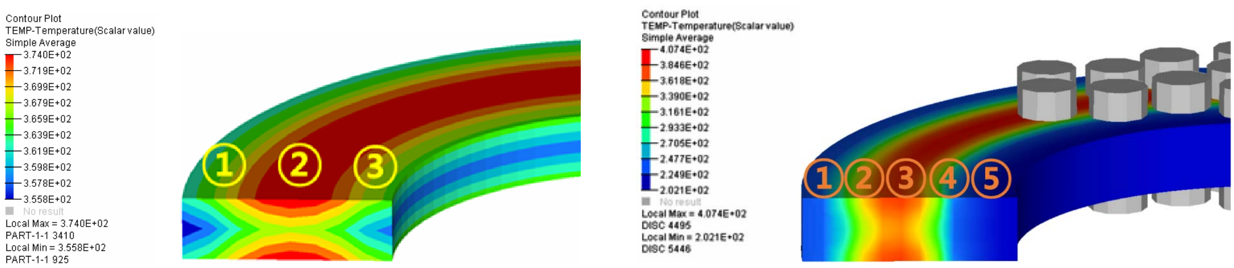
(a) At 290 km/h

(b) At 250 km/h

(c) At 150 km/h

(d) At the end of braking action

Fig. 6. Temperature distribution of solid disc due to thermo-mechanical analysis.



(a) Thermal analysis result

(b) Thermo-mechanical analysis result

Fig. 7. Temperature distribution on the solid disc surface at the end of braking action.

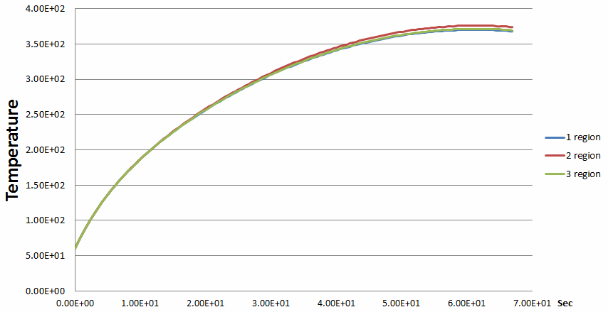
was used as the friction coefficient.

3.3 Comparison of simulation results

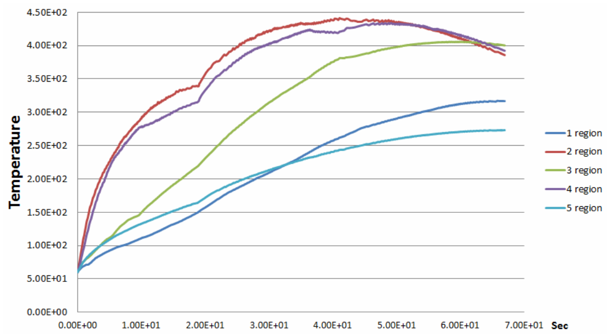
Fig. 5 shows the analysis result when heat flux, rather than friction between disc and pad, was calculated and applied to the disc. Fig. 6 shows the result of the analysis of thermo-mechanical behavior caused by friction between the disc and the pad. Fig. 7 shows the temperature distribution on the cross-section of the disc after the braking action was completed.

The analysis for the application of the heat flux shows that no hot band was formed in the course of braking action. The thermo-mechanical analysis of the friction between the disc and the pad shows that a hot band was formed at the outer rim of the disc at the initial stage of the braking action. The result of the thermo-mechanical analysis shows that another hot band was formed at the inner rim of the disc as time passed, and at 150 km/h, the heat of the two bands was conducted and a large band was formed. The results of the two analyses (thermal analysis and thermo-mechanical analysis) at the end of the braking action show a similar temperature distribution, with the exception of the differences between the maximum temperature and the minimum temperature.

However, the temperature distribution on the cross-section of the disc shows a difference between the thermal analysis and the thermo-mechanical analysis. In the thermal analysis, because the heat flux was applied to the entire friction surface of the disc, the difference between the maximum temperature and the minimum temperature was just 20 °C over the entire



(a) Thermal analysis result



(b) Thermo-mechanical analysis result

Fig. 8. Max. temperature graphs of the solid disc during the braking action.

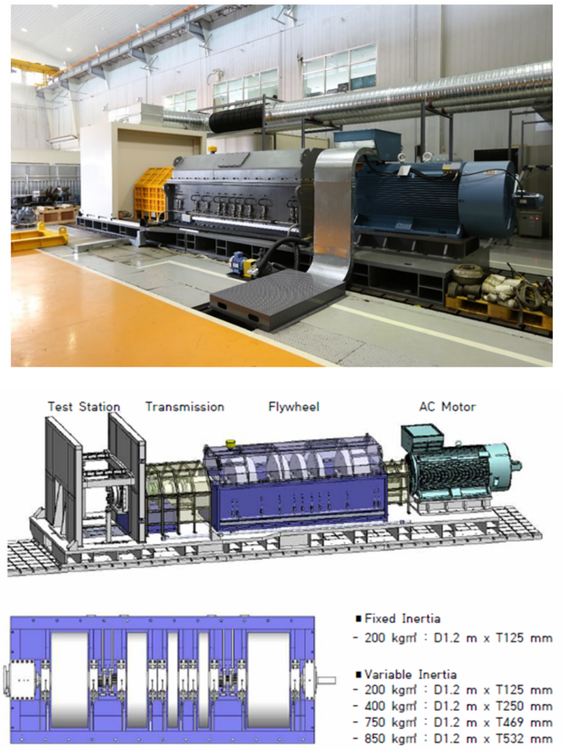


Fig. 9. Brake dynamo test machine.

disc. The maximum temperature was measured at the disc surface only. In the thermo-mechanical analysis, however, the difference between the maximum temperature and the minimum temperature was approximately 200 °C. The maximum temperature was measured over the whole area at the center of the disc. This is because the center of the disc has more friction area, and therefore, receives more friction heat than other parts.

Fig. 8 shows the temperature distribution for every part of the disc during the braking action. The result of the thermal analysis shows a constant increase of temperature until the end of the braking action. Also, the chart shows a similar temperature change in all the areas. This is because the same heat flux was applied to the entire area of the disc. In the thermal analysis, the maximum temperature was 376 °C at the 61st second which is just before the end of the braking action, and 374 °C at the end of the braking action, which indicates that the disc is rarely cooled down. In the thermo-mechanical analysis, the maximum temperature was approximately 441 °C at the 42nd second in zones 2 and 4 where the hot bands occurred. After the maximum temperature section, the disc was cooled down slowly. In zones 1, 3 and 5 where no hot band was found, the temperature kept increasing. The temperature reached the maximum point in zone 3, which is the center of the disc at the end of the braking action. This is because the center of the disc mainly met the pad, and therefore, had more friction area. In braking step 2 when the pressure of the pad grows, the thermal gradient increases.

4. Analysis of thermal behavior of the disc brake using the dynamo test machine

4.1 High-speed brake performance tester

Brake performance evaluation technology has been advanced with the increase of the speed of railway vehicles. Currently, various test procedures and evaluation criteria have been systematically established, from testing and evaluation of single brake parts and testing of control and integrated performance of the brake system to the commissioning and evaluation of actual railway vehicless [14]. The high-speed brake dynamo tester simulates the braking process of a railway vehicles to measure the friction coefficient and the maximum temperature of the disc/pad, providing data that can be utilized in development, conformance test and qualification test of the brake parts, such as brake disc/pad. As illustrated in Fig. 9, a dynamo tester is composed of an AC motor, a flywheel, a transmission and a test station. The flywheel not only simulates the inertia effect of a large railway vehicles, but also allocates inertia depending on the weight of the railway vehicles. Fig. 10 shows the inside of the test station. The test station is equipped with the brake disc and the brake pad in order to implement the actual braking mechanism. Fig. 11 shows the studded brake pad. Table 4 shows the specifications of the high-speed brake performance tester.

4.2 Conditions and result of braking test

UIC 541-3 E.2 Test programme No. 5 was referred to for

Table 4. Specifications of brake dynamo test machine.

Item		Specification
Type of specimen		Brake disc Brake pad
AC	Capacity	630 kW
Motor	Speed	Max. 3000 rpm (503 km/h at 890 ϕ)
	Nominal torque	6014 Nm at 1000 rpm
Max. brake torque		35000 Nm
Mechanical inertia		Min. 200 ~ Max. 2400 kgm^2
Direction of rotation		2
Max. allowed specimen diameter		1350 mm
Laboratory environment		15 ~ 30 °C
Total power supply		Approx. 1000 kW

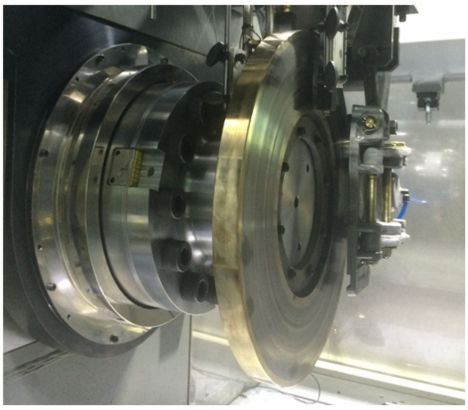


Fig. 10. Test station of brake dynamo test machine.

Fig. 11. Brake pad.

the conditions of experiment of the braking process of a high-speed railway vehicles as in case of the analysis (simulation) work. A solid type brake disc and a studded pad were used for the brake test. The atmospheric temperature was set to room temperature. In order to consider the rotational inertia applied to a single disc, the tester is equipped with a flywheel of 792 kgm^2, and the brake test was carried out with step 2 brake load at 300 km/h. A high-speed camera was used to take photos of the braking action and the hot bands, and a thermo-graphic camera was used to measure the temperature of the hot bands. It is very difficult to compare the results of the brake tests precisely because each test shows different braking time and pressure distribution. Therefore, the result was compared by

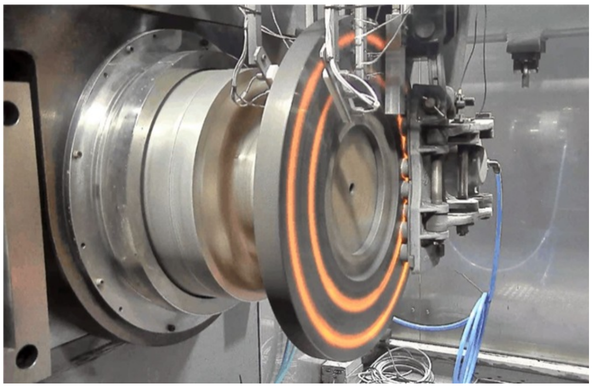


Fig. 12. Image of the disc taken with a high-speed camera during the braking action.

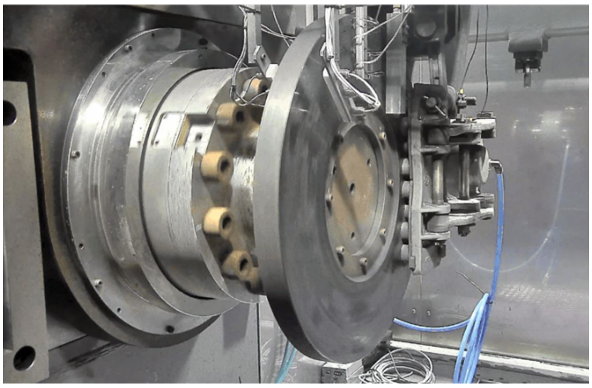


Fig. 13. Image of the disc taken with a high-speed camera after the braking action.

looking at the trend of the analysis results and the test results.

Figs. 12 and 13 show the test images taken with the high-speed camera. Fig. 12 shows that 2 hot bands were formed on the disc. This image shows that the thermo-mechanical behavior of the disc in the actual test and the analysis are similar, proving the feasibility of the analysis model. For the thermo-mechanical analysis, a constant braking force was applied to the pins of the pad. The image taken with a high-speed camera shows similar thermo-mechanical behavior of the disc, indicating that a constant pressure was applied to all of the studs of the pad in the actual test. Fig. 13 shows the image of the disc at the end point of the braking action. In this image, the dark areas on the disc surface show the traces of hot bands.

In Figs. 14 and 15, the images taken with a thermo-graphic camera show that hot bands were formed at similar positions. Fig. 15 shows that the image is more like one of a hot spot than a hot band because of the influence of the radiation factor caused by phase change (phase transformation) at a high temperature and the error which may occur in the image of the disc rotating at a high rpm.

Fig. 16 shows the temperature image taken with a thermo-graphic camera at 3 points on the disc surface. The blue line shows the temperature at point 1, the green line at point 2 and the orange line at point 3. Similar to the result of analysis,

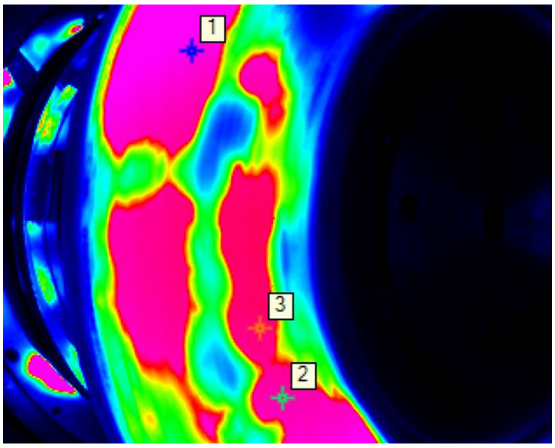


Fig. 14. Image taken with a thermo-graphic camera during the braking action.

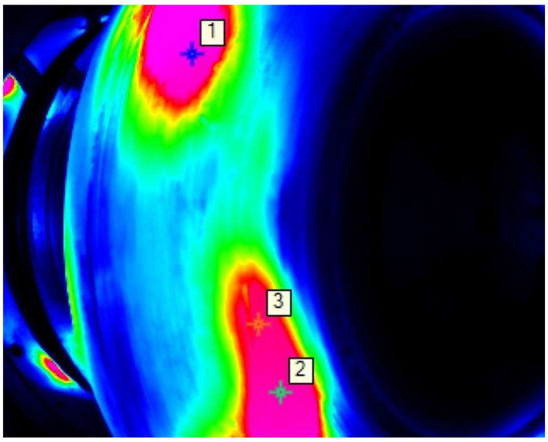


Fig. 15. Image taken with a thermo-graphic camera after the braking action.

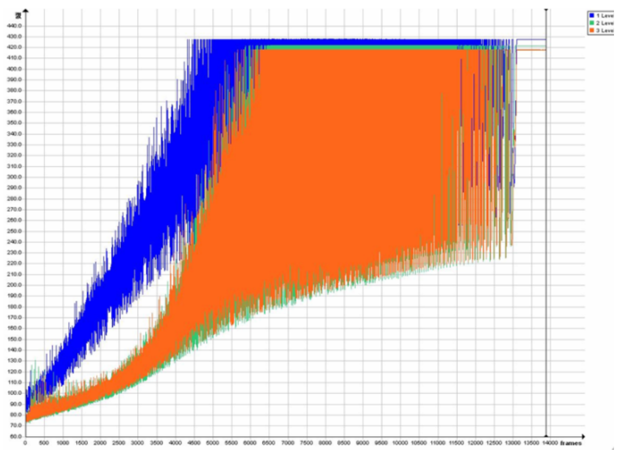


Fig. 16. Result of temperature measurement of a thermo-graphic camera.

there was no substantial change of temperature before/after the maximum temperature. The image shows that the disc was cooled down slowly after the maximum temperature. The maximum temperature after the end of the braking action was 426.9 °C, 421.2 °C and 418.3 °C at each point, respectively, which is similar to the result of the thermo-mechanical analysis.

In the analysis, because the two hot bands were combined to a single band as the braking action came to an end, the internal temperature of the hot bands was similar, and the temperature measured at the test showed a similar trend. Although the result may not be completely identical to the result of the thermo-mechanical analysis because there is a gap with the actual emissivity, the trend is the same. The actual test result is more similar to the thermo-mechanical behavior caused by the disc-pad friction than to the thermal behavior of the disc when the heat flux is applied. Therefore, the analysis with contact conditions between the disc and the pad can be said to be more appropriate than the analysis with heat flux applied to the disc.

5. Conclusions

The aim of this study was to develop a model that simulates the braking action using ABAQUS, a commercial finite element analysis program and to analyze friction between the solid-type disc and the studded pad used in KTX vehicles. The following conclusions were reached through a comparison of the analysis result and the test result.

(1) The result of the heat flux analysis shows that no hot band was formed during the braking action. In the thermo-mechanical analysis of the friction between the disc and the pad, a hot band was formed at the outer part of the disc, and another hot band at the inner part of the disc from the middle stage of the braking action. The two hot bands were combined to a single band due to heat conduction. The results of the two analyses at the end of the braking action show a similar temperature distribution. The difference of the temperature distribution on the cross-section of the disc between the results indicates that there is a difference in thermal behavior between the two analysis models.

(2) The result of the thermo-mechanical analysis and test shows that hot bands were formed at the initial stage of the braking action. If the braking process is repeated, the resulting one-sided wear and concentration of temperature may cause a phase transformation of the material, deteriorate the strength of the disc material, and result in thermal cracks due to contraction/expansion. Thermal crack has a substantial influence on the safety of railway vehicles, as it reduces the life of discs and causes thermal fatigue followed by damage. To prevent the occurrence of hot bands, therefore, it is necessary to apply a constant pressure to the disc-pad brake system, and to develop the relevant pad technology.

(3) To evaluate the feasibility of the analysis model and to verify the reliability of the analysis result, a braking test was carried out with the high-speed braking for performance tester. According to the test result, the temperature distribution measured at the thermo-mechanical analysis showed a similar maximum temperature and temperature increase/decrease

trend with that of the test. It is difficult to closely compare the test result with the analysis result due to inaccurate emissivity, but the trend shows that the thermo-mechanical analysis, rather than the thermal analysis, has higher feasibility of model and reliability of the result.

In further research, analysis and testing of the irregular pressure of the brake pad will be carried out, and the resulting thermo-mechanical behavior of the disc will be compared and verified.

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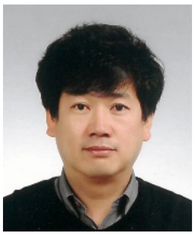

Ho-Yong Lee received the Ph.D. degree in mechanical design engineering from Sungkyunkwan University in 2005. He is currently a Senior Researcher of Korea Railroad Research Institute. His research areas include the structure analysis, fatigue/fracture mechanics, system reliability, and brake system.