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INVESTIGATION OF HEAT CONVECTION DURING VEHICLE BRAKING

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Abstract: The brake system is the most significant active safety system that converts the vehicle's mechanical energy into heat energy based on the braking pair's friction. The transfer of heat in the brake pair during braking occurs by conduction, convection, and radiation. In disc brakes, which are widely used today, the effect of the heat transfer coefficient, which continually changes with the disc's angular velocity on the cooling of the braking pairs is important. The change in heat convection coefficient can be calculated with a numerical approach. In this study, on disc temperature effect of heat convection coefficient and angular velocity has been examined with the finite element method using the experimental data in the SAE J2522 Comprehensive Brake Efficiency Test Standard, fading test procedure. Disc brake design has been designed in the SOLIDWORKS program, and analysis has been carried out using ANSYS 2020 R2 in steady-state thermal analysis. It has been determined that the cooling amounts at the initial of braking are higher than the cooling amounts at the end of braking, and the effect of cooling by convection in the reduction of the temperature difference at the end of braking compared to the initial state is between 91.22% and 91.74%.
Keywords: Disc brake, heat transfer, convection, fading

TAŞIT FRENLEMESİ ESNASINDA OLUŞAN ISI TAŞINIMININ İNCELENMESİ

Özet: Fren sistemi, frenleme çiftleri arasındaki sürtünme esasına göre taşıtın mekanik enerjisini ısı enerjisine dönüştüren, en önemli aktif emniyet sistemidir. Frenleme esnasında fren çiftindeki ısının transfer edilmesi iletim, taşınım ve ışınım yoluyla gerçekleşmektedir. Günümüzde yaygın olarak kullanılan disk frenlerde, diskin açısal hızının etkisiyle sürekli olarak değişen ısı taşınım katsayısının frenleme çiftlerindeki soğutma üzerine etkisi önemlidir. Isı taşınım katsayısındaki değişim ise sayısal yaklaşımla hesaplanabilmektedir. Bu çalışmada SAE J2522 Kapsamlı Fren Etkinlik Test Standardı sıcaklıkla fren zayıflaması test prosedüründeki deneysel veriler kullanılarak, açısal hız ile ısı taşınım katsayısının disk sıcaklıkları üzerine etkisi, sonlu elemanlar metoduyla incelenmiştir. Disk fren tasarımı SOLIDWORKS programında, analiz ise ANSYS 2020 R2'de kararlı durum termik analizde gerçekleştirilmiştir. Frenleme başlangıcındaki soğutma miktarlarının frenleme sonundaki soğutma miktarlarından daha yüksek olduğu ve frenleme sonundaki sıcaklık farkının ilk duruma göre azalmasında taşınım ile soğutma etkisinin %91,22 ile %91,74 arasında olduğu tespit edilmiştir.
Anahtar kelimeler: Disk fren, ısı transferi, taşınım, sıcaklıkla fren zayıflaması

NOMENCLATURE

D	: Characteristic length (mm)
d_o	: Disc's outer diameter (mm)
h	: Heat transfer coefficient (Wm^2/K)
k_a	: Thermal conductivity (W/mK)
Re	: Reynolds number
R	: Tire diameter (mm)
T	: Temperature (K)
t	: Time (s)
w	: Angular velocity rad/s

Greek symbols

ρ	: Density of the air (kg/m^3)
μ	: Dynamic viscosity (kg/ms)
T_{ri}	: Initial temperature of disc
T_{rf}	: Final temperature of disc

INTRODUCTION

In the disc-pad couple of a vehicle, most of the kinetic energy is converted into thermal energy due to the friction effect, and generated heat is dissipated in the surrounding environment. Therefore, one of the fundamental problems of a braking system is how to handle the thermal energy generated throughout its movement. A large of the produced heat flows out to the air, even if the heat dissipation mechanisms are different (conduction, convection, and radiation). As a result, heat is dissipated by convection (Saiz *et al*, 2015).

A large amount of heat produced in the disc-pad couple contact area during braking creates unequal temperature distributions on the disc, and the disc-pad pair temperature rises as a result of the heating of the brake pair during mutual slip. (Belhocine and Bouchetara, 2012). A vehicle's braking performance is importantly influenced by the temperature increase in the disc-pad couple. Increased temperatures during braking may reason brake fade, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally-excited vibration. That is why it is significant to predict a given brake system's temperature rise and assess its thermal performance in the early design stage (Lee, 1999).

Belhocine and Bouchetara (2012) have stated that the study aims to analyze the thermal behaviour of full and ventilated brake discs by utilizing parameters such as brake type, disc geometric design, and disc material. They have noticed for a ventilated disc out of cast iron FG15, the temperature rises to 345.44 °C in 1.85 s, behind it drops quickly. They have found out that the variation in temperature between a full and ventilated disc having the same material is about 60 °C at the moment t =1.8839 s. They have concluded that the disc's geometric design is an essential factor in the improvement of the cooling process of the discs. Kishore and Vineesh (2021) have used the axle-mounted disc brake system used in the coaches of Indian railways for braking of the train in their study. In their work, temperature evolution during the braking process has been analyzed with the finite element method. The maximum peak temperature increase of 215 °C is observed on the disc at 290 mm in the radial direction at about 60% of braking time while stop braking from 160 kmph. They indicated that peak temperature at different stop braking speeds states to be linearly rises with stop braking speeds. Abhishek et al (2020) carried out the static structural and thermal analysis with various disc materials (Gray Cast Iron, Titanium Ti-6Al-4V and Chromium-Vanadium Steel) using ANSYS. They indicated that the newly designed ventilated brake disc shows a more appropriate temperature gradient than the drilled contour disc regardless of the three metals utilized in this analysis. Chromium-Vanadium steel of these three materials has established a smaller maximum temperature in the ventilated disc, about 216.22 °C. The drilled contour disc with gray cast iron has showed the best temperature gradient, producing a maximum temperature of 320.01 °C compared to the others.

Jihan *et al* (2020) have embedded heat pipes on the surfaces of the ventilated brake disc. The brake bench test and numerical simulation analysis of heat pipe ventilated brake disc and general ventilated brake disc under the condition of repeated fifteen braking and continuous downhill braking have carried out, respectively. They indicated that the highest temperature has decreased by 7 °C during continuous downhill braking, and better heat transfer effects have achieved through inserting heat pipes on the surfaces of ventilated brake disc under the two braking conditions.

In the literature, the heat transfer coefficient value has been taken constant in most of the thermal studies for the brake disc (Mugilan, 2022; Dubale, 2021; Dhir, 2018; Nathi, 2012). In the thermal analysis carried out in this study, the temperature and total heat flux values have been examined, taking into account the variable heat transfer coefficient depending on the angular velocity values obtained from the tests performed according to the SAE J2522 brake standard (Demir, 2009). The disc design has been designed in SOLIDWORKS, and the steady-state thermal analysis has been carried out in ANSYS 2020 R2.

MATERIAL AND METHOD

Analyzed brake disc-pad pair

There are many different types of friction materials in use today. Whatever kind of friction material is chosen, the functional requirements of friction material for road vehicles include properties such as providing a consistent and reliable friction force, being resistant to wear, being robust against mechanically and thermally applied loads during use (Day, 2014). In this study, gray cast iron disc is chosen that meets the friction material properties.

In this study, the front brake disc parameters of a automobile are used, and the design of the disc is made in the SOLIDWORKS program (see Fig. 1). Geometric dimensions and material properties of the disc-pad couple are given in Table 1 (Demir, 2009).

Table 1. Geometric dimensions and material properties

Properties	Value
Disc’s weight (kg)	5
Disc thickness (mm)	22.05
Pad thickness (mm)	12
Disc diameter (mm)	255
Wheel diameter (mm)	298
Disc’s density (kg/m^3)	7228
Specific heat of the disc (J/kg.K)	419
Conduction coefficient (W/m.K)	48
Pad surface area (mm^2)	4213

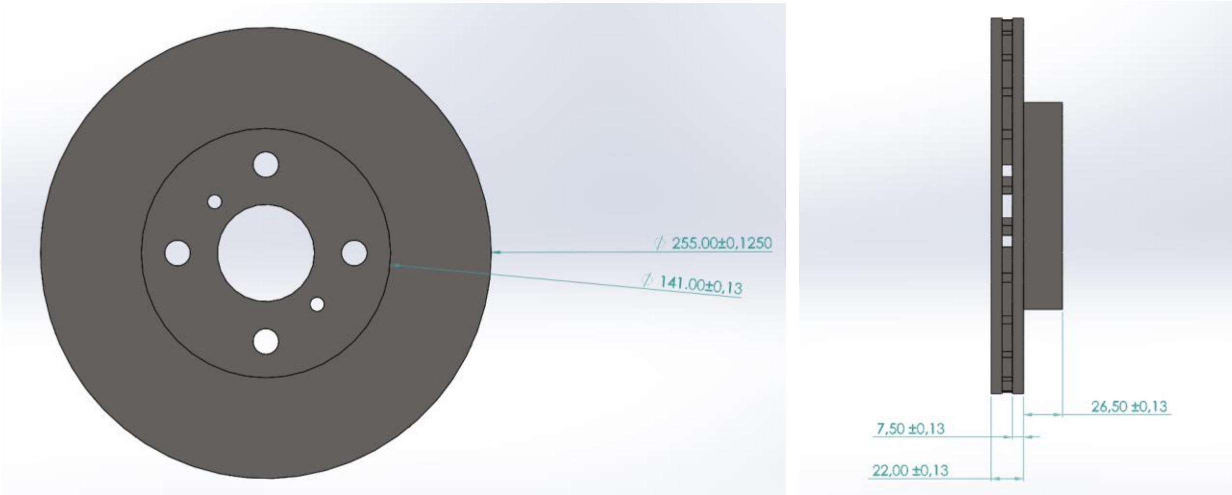

Figure 1. Geometric dimensions of the disc

Braking procedure

SAE Recommended Practice describes an inertia-dynamometer test procedure to evaluate the behavior of friction material related to temperature, pressure, and speed for vehicles appointed with hydraulic brake system. SAE J2522 is a universal effectiveness test handy only when target friction levels for specific sections or a baseline material are available for comparison. Engineers use the SAE J2522 that contains green effectiveness, burnish (or bedding), characteristic check, speed/pressure sensitivity, cold braking check, motorway braking check, fade, recovery, pressure sensitivity, increasing temperature sensitivity, and pressure sensitivity primarily for passenger cars, SUV's, light and medium duty trucks evaluation (Carlos and Ferro, 2005). In this study, the fading procedure of the SAE J2522 brake test standard has been selected for the passenger car front brake disc analysis. The initial condition values received from the experiment made according to the SAE J2522 brake standard are given in Table 2.

Table 2. Braking conditions

Conditions	Value
Braking velocity (km/h)	100
Release velocity (km/h)	5
Friction coefficient	0.412
Brake application control	0.4g
Initial brake temperature (°C)	100-550
Braking time (s)	7.439
Number of decelerations	15

Finite Element Method

The heat transfer radiation is neglected and material properties are assumed to be isotropic and independent of temperature while performing thermal analysis with FEM.

In this article, it requires understanding the equations utilized inside the ANSYS where steady-state thermal analysis is done for the braking. The differential equation in a cylindrical coordinate system (r, z) for steady state is (Lewis *et al*, 2004);

$$k_r \frac{\partial^2 T}{\partial r^2} + \frac{k_r}{r}\frac{\partial T}{\partial r} + k_z \frac{\partial^2 T}{\partial z^2} + G = 0 \tag{1}$$

The boundary conditions are;

$T=T_b$ on T_1

$$k_r \frac{\partial T}{\partial r} \mathrm{l} + k_z \frac{\partial T}{\partial z} \mathrm{n} + \mathrm{h}(\mathrm{T} - \mathrm{T_a}) + q = 0 \quad \text{on } \mathrm{T_2} \tag{2}$$

The temperature distribution is described as follows:

$$T = \frac{1}{2A}(a_i + b_i r + c_i z)T_i + \frac{1}{2A}(a_j + b_j r + c_j z)T_j + \frac{1}{2A}(a_k + b_k r + c_k z)T_k \tag{3}$$

Matrix form of heat transfer problem for steady state;

$$[K][T] = \{f\} \tag{4}$$

$$[K] = \int_{\Omega} [B]^T[D][B]\,\mathrm{d}\Omega + \int h[N]^T[N]\,\mathrm{d}T \tag{5}$$

$$[B] = \begin{Bmatrix} \frac{\partial T}{\partial y} \\ \frac{\partial T}{\partial y} \end{Bmatrix} = \begin{bmatrix} \frac{\partial N_i}{\partial r} & \frac{\partial N_j}{\partial r} & \frac{\partial N_k}{\partial r} \\ \frac{\partial N_i}{\partial z} & \frac{\partial N_j}{\partial z} & \frac{\partial N_k}{\partial z} \end{bmatrix} \tag{6}$$

$$[D] = \begin{bmatrix} k_r & 0 \\ 0 & k_z \end{bmatrix} \tag{7}$$

Thermal stiffness matrix is;

$$[K] = \frac{2\pi\bar{r}k_r}{4A}\begin{bmatrix} b_i^2 & b_ib_j & b_ib_k \\ b_ib_j & b_j^2 & b_jb_k \\ b_ib_k & b_jb_k & b_k^2 \end{bmatrix} + \frac{2\pi\bar{r}k_z}{4A}\begin{bmatrix} c_i^2 & c_ic_j & c_ic_k \\ c_ic_j & c_j^2 & c_jc_k \\ c_ic_k & c_jc_k & c_k^2 \end{bmatrix} + \frac{2\pi hl_{ij}}{12}\begin{bmatrix} 3r_i + r_j & r_i + r_j & 0 \\ r_i + r_j & r_i + 3r_j & 0 \\ 0 & 0 & 0 \end{bmatrix} \quad (8)$$

Load vector is;

$$\{f\} = \int_{\Omega} G[N]^T \, \mathrm{d}\Omega - \int_T q\,[N]^T \, \mathrm{d}T + \int_T h\,T_a[N]^T \, \mathrm{d}T$$

$$= \frac{2\pi GA}{12} \cdot \begin{bmatrix} 2 & 1 & 1 \\ 1 & 2 & 1 \\ 1 & 1 & 2 \end{bmatrix} \cdot \begin{bmatrix} r_i \\ r_j \\ r_k \end{bmatrix} - \frac{2\pi q l_{jk}}{6}\begin{bmatrix} 0 \\ 2r_j + r_k \\ r_j + 2r_k \end{bmatrix} + \frac{2\pi h T_a l_{ij}}{6}\begin{bmatrix} 2r_i + r_j \\ r_i + 2r_j \\ 0 \end{bmatrix} \quad (9)$$

To see the temperature values of brake disc has made steady-state thermal analysis utilizing the finite element method. Here, the temperature and convection surfaces are chosen as in Fig. 2. The disc-pad couple is meshed consisting of 27704 nodes and 13952 elements. In the model, skewness is 0.46371, and orthogonal quality is 0.53403. These values approve the high mesh quality in this study (ANSYS, 2015).

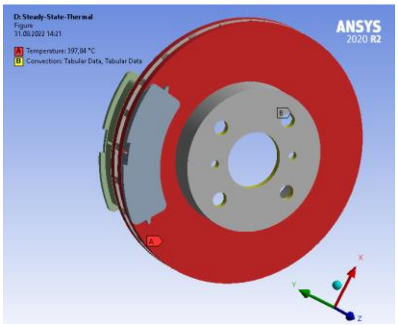

Fig. 2 Boundary conditions of steady-state thermal analysis

Heat transfer coefficient

All properties of air such as temperature, density, viscosity, and thermal conductivity play a significant role in calculating the heat transfer coefficient. Considering that all these properties of air can be determined, it can be thought that the heat transfer coefficient for the brake disc will take a fixed value. However, due to the rotation of the disc around its axis at different speeds during braking, the ambient air and the angular velocity value of the brake disc affect each other. This situation makes it possible to determine the heat transfer coefficient values in return to varying angular velocity values during braking. First of all, the values corresponding to the temperature of air (295.15 K) in the ambient condition should be determined by the interpolation method (Çengel and Ghajar, 2015). With the values (ρ_{air} = 1.2002 kg/m^3, μ_{air} = 1.82.10^{-5} kg/m.s) obtained for 295.15 K, the disc's outer diameter (d_o = 0.256 m) and the tire diameter (R = 0.596 m) is positioned in the formula, and the Re number depending on the angular velocity is obtained as 10061.63*w by using Equation (10).

$$Re = \frac{wR\rho_a d_o}{\mu_a} = 10061.63 * w \quad (10)$$

The heat transfer coefficient formula changing as dependent on airflow characteristic has been given in Equation (11) (Limpert, 2001). (Laminar flow if Re ≤ 2.4*10^5, turbulent flow if Re > 2.4*10^5)

$$h = \begin{cases} 0.7\left(\frac{k_a}{D}\right) \cdot Re^{0.55}, & Re \leq 2.4.10^5 \\ 0.4\left(\frac{k_a}{D}\right) \cdot Re^{0.8}, & Re > 2.4.10^5 \end{cases} \quad (11)$$

The Re number (10061.63*w), the thermal conductivity (k_{air} =0.02585 W/m.K), and the disc's outer diameter (D = 0.256 m) is written in Equation (11). Equation (12) has been obtained by leaving the angular velocity alone in the section on the right side of the equation. If the value of angular velocity is less than or equal to 23.85, the upper part of Equation (12) is used; if it is large, the lower part is used. In this way, different heat transfer coefficient values are obtained different time of the braking procedure alternating angular velocities.

$$h = \begin{cases} 11.24 \cdot w^{0.55}, & w \leq 23.85 \\ 6.43 \cdot w^{0.8}, & w > 23.85 \end{cases} \quad (12)$$

Braking Analysis

One of the most important parameters affecting the rising temperatures during braking is the convection in the regions where the disc comes into contact with air. Because very high temperatures are undesirable for the disc and pad couple, the transfer of heat to air via convection, in other words, the cooling of the disc, is an issue that needs to be examined. In this study, while the heat convection coefficient values were defined in the ANSYS program, a function was created with the angular velocity values obtained from the tests performed according to the SAE J2522 standard. In this function, heat convection coefficient data was obtained for each angular velocity value, and 1000 data sets were created by defining the steps in the program. Graphs consisting of 1000 data of temperature and heat transfer coefficient for braking in the temperature ranges of 284-417 °C, 397-507 °C, 421-526 °C, and 496-584 °C are shown in Fig. 3, respectively.

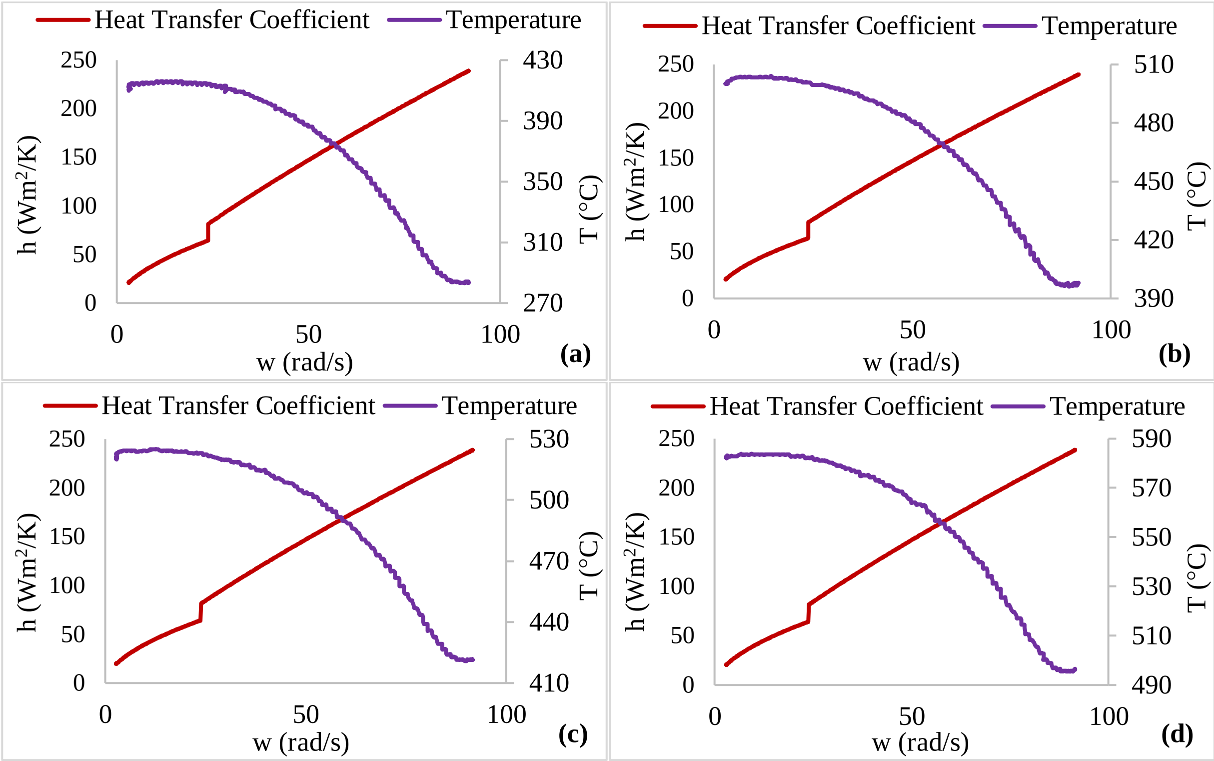

Figure 3. Change of temperature and heat transfer coeficient, 284-417 °C (a), 397-507 °C (b), 421-526 °C (c), and 496-584 °C (d)

During the braking analysis, while the temperature increased over time, the heat transfer coefficient values decreased due to the decrease in the angular velocity of the disc. There was a sudden decrease in the heat transfer coefficient when the angular velocity was 23.85 rad/s for braking in different temperature ranges. The breakage seen in Fig. 3 shows the transition of the flow from turbulent to laminar, after a particular value of the change in Re number (Laminar flow if Re ≤ 2.4 10^5, turbulent flow if Re > 2.4 10^5) with together decrease in the flow velocity. This study, it has been aimed to see the effect of heat transfer coefficient on temperatures depending on the angular velocity with the data used during thermal analysis.

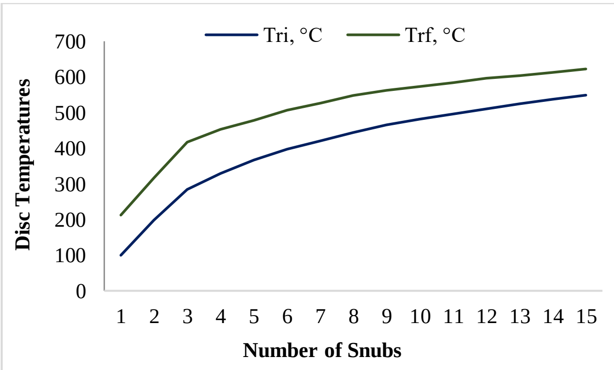

Figure 4. Disc's initial and final temperatures in repeated braking

RESULTS AND DISCUSSIONS

Experimental analysis of repeated braking

The graph of the initial and final temperatures of the disc measured as a result of repeated braking in the experimental study made according to the SAE J2522 brake standard is presented in Fig. 4. The test is done at Link Engineering Company. The temperature measurements are carried out with a thermocouple. In the experimental study, the results of repetitive braking have revealed that the temperatures increased at the finish of each braking repetition and 15 braking repetitions.

Table 3 has showed that the temperature obtained at the end of a cycle decreased until the next cycle, that is, cooling occurred. However, the cooling that occurs between cycles is insufficient for the disc brake to reach its initial temperature. In the next cycle, the disc has started to heat up again before it can reach the initial temperature. This situation increased the energy accumulated on the disc, and the temperatures increased with the increase in the number of repetitive braking.

Table 3. Initial and final temperature values in repeated braking

Stop Number	T_{ri} (°C)	T_{rf} (°C)
1	100	212
2	198	316
3	284	417
4	329	452
5	367	478
6	397	507
7	421	526
8	444	548
9	465	562
10	482	574
11	496	584
12	511	595
13	525	604
14	537	612
15	549	622

Steady State Thermal Analysis

Steady-state thermal analysis was carried out for the temperature ranges of 284-417 °C, 397-507 °C, 421-526 °C, and 496-584 °C. Close to each other visual results were obtained for these temperature ranges. It is given Table 4 the maximum and minimum temperature values obtained from thermal analysis for temperature ranges of 284-417 °C, 397-507 °C, 421-526 °C, and 496-584 °C. In the thermal analysis carried out in the temperature range of 284-417 °C, the temperature visuals at 0.008, 5.406, and 7.515th seconds, and the total heat flow visual at 7.515th second were shown in Fig. 5. Throughout braking analysis, the difference between the maximum and minimum temperature values were 56.46 °C at initial of braking, 33.18 °C in the middle, and 8.86 °C at end. When temperature drops were observed in cold region of the disc where the convection is more effective than in the outer radius of the disc, the maximum temperature value was obtained because friction occurs in the outer radius of the disc. The total heat flux was determined to be a minimum of 8.6727e-16 W/mm^2 and a maximum of 0.017641 W/mm^2.

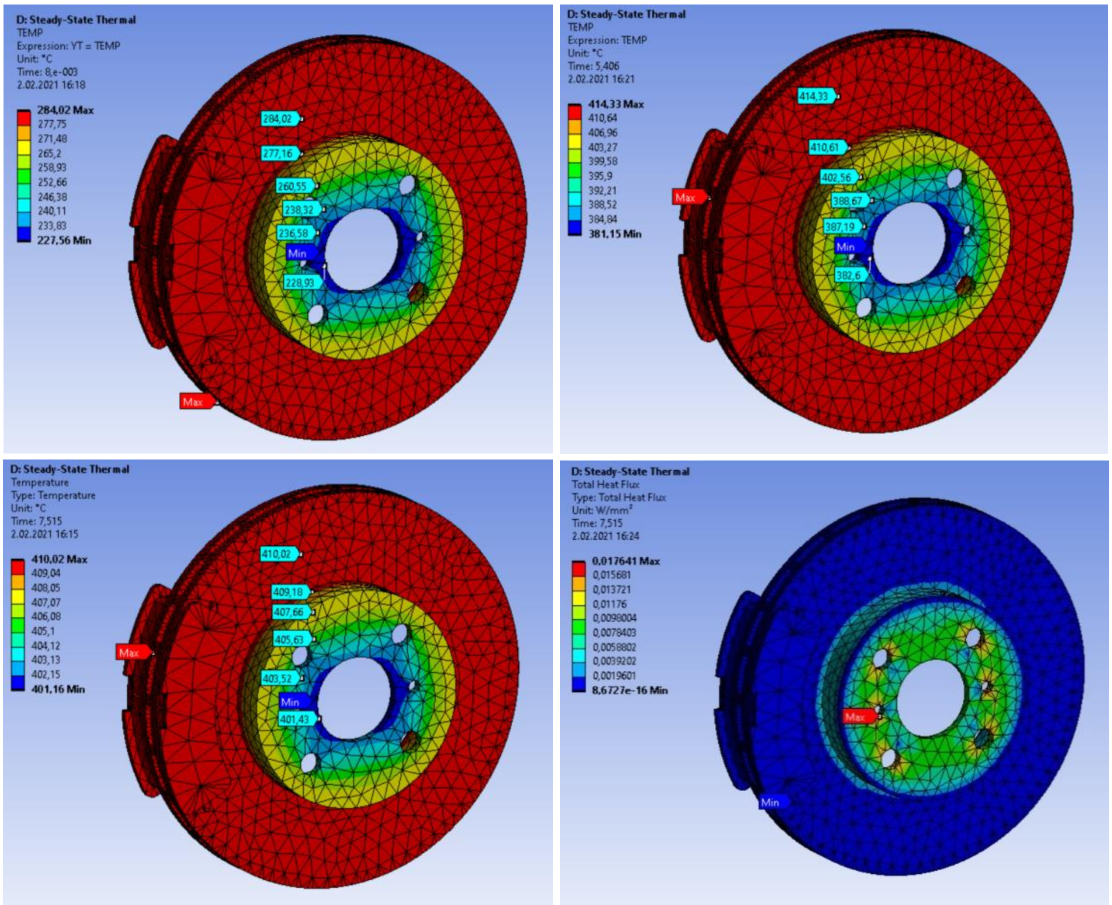

Figure 5. Temperature and total heat flux in thermal analysis in the temperature range of 284-417 °C

Table 4. The maximum and minimum temperature values

	Initial of braking		5.406^{th} second of braking		End of braking	
Temperature range (°C)	T_{max} (°C)	T_{min} (°C)	T_{max} (°C)	T_{min} (°C)	T_{max} (°C)	T_{min} (°C)
284-417	284.02	227.56	414.33	381.15	410.02	401.16
397-507	397.84	316.83	500.74	460.09	500.19	489.44
421-526	421.46	335.47	522.39	479.12	520.08	509.39
496-584	496.35	394.25	581.84	533.24	581.98	569.56

When the total heat flux visual was examined, the heat flux values were higher at the centre of the disc, while the minimum values were obtained in the outer radius of the disc. This situation demonstrated that the heat generated during braking moved away faster than the centre of the disc than the outer radius of the disc.

Experimental data and analysis data

Brake disc cooling is a complex phenomenon governed by conduction, convection, and radiation. Convection, one of these modes, is the most important mechanism of spreading heat from the brake disc while the vehicle is in motion. When the disc temperature rises above air temperature, energy (heat) transfer occurs towards air due to the temperature difference. For the temperature ranges on the brake discs during this process, the number of Prandtl varies very little. Therefore, it can be said that although convection is caused by the temperature difference between the brake disc and air in contact with it, it also depends on the dynamic of the flow near the disc surface. In the dynamic of flow, high velocity flows create large temperature gradients. That is, an upwards amount of heat is transported for high-speed streams than for low-velocity streams. The speed of the flow around the disc is affected by the angular velocity of the disc. This situation is reflected in the heat convection coefficient and affects the cooling amounts in the disc. When the vehicle starts braking from a certain speed, the decrease in angular velocity decreases the heat transfer coefficient. Low heat transfer coefficient values correspond to a lower level of energy transferred, especially for a disc. All of these have reflected in the temperature data obtained as a result of the analysis.

Experimental data and analysis data in the study conducted for the disc-pad pair are given in Fig. 6, temperature ranges of 284-417 °C, 397-507 °C, 421-526 °C, and 496-584 °C. It is determined that the amount of temperature drop at the beginning of braking is higher than the amount of temperature drop at the end of braking. This is explained by the fact that the heat transfer coefficient values are a function of the angular velocity. Because the heat convection coefficient value decreases as the angular velocity value decreases, compared to the beginning of braking decreases at the end of braking, the effect of convection on cooling.

When the experimental data and analysis data were compared for the temperature range of 284-417 °C, it was found that there was a difference of 3.25 °C at the initial of braking and 0.5 °C at the end of braking. Table 5 presents all of the differences between experimental and analysis data at the initial and the end of braking for the four temperature ranges.

At end of braking, it has been determined that in the decrease in temperature difference compared to the initial state, the convective effect is decreased at the rate of 91.22%. At the 5.406^{th} second of braking, in the decrease in temperature difference compared to the initial state, the convection effect is decreased at the rate of 65.67%. At the 5.406^{th} second of the braking, in the decrease in temperature difference compared to end of braking, the convection effect is decreased at the rate of 74.43%. Convective effect decrease rates in temperature difference drop are given in Table 6, comparatively for all temperature ranges.

Table 5. Difference between experimental and analysis temperatures

Temperature range (°C)	Initial of braking (°C)	End of braking (°C)
284-417	3.25	0.5
397-507	4.66	0.614
421-526	4.96	0.61
496-584	5.88	0.70

Table 6. Convective effect decrease rates in temperature difference drop

Temperature range (°C)	According to the initial of braking at the end of braking (%)	According to the initial of braking at 5.406^{th} second of braking (%)	According to the 5.406^{th} second of braking at the end of braking (%)
284-417	91.22	65.67	74.43
397-507	91.34	64.72	74.46
421-526	91.74	64.75	76.57
496-584	91.46	64.61	75.88

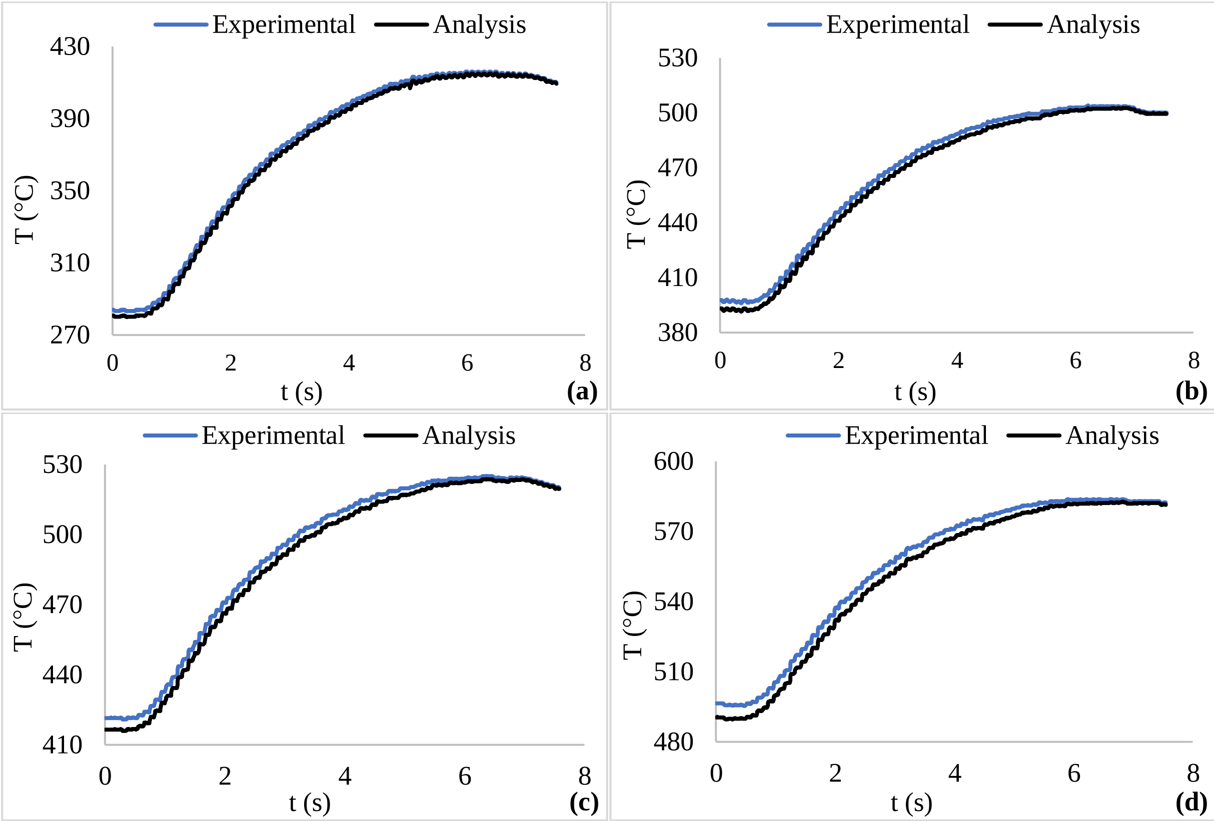

Figure 6. Experimental data and analysis data, 284-417 °C (a), 397-507 °C (b), 421-526 °C (c), and 496-584 °C (d)

CONCLUSION

In the heat convection coefficient-angular velocity graphs, the break that occurred at a value of 23.85 rad/s showed that the flow passed from turbulent to laminar after a certain value of the Re number with the decrease in the velocity of the flow.

Since cold region of the disc is not exposed to friction, minimum temperature values have been observed. In the outer radius of the disc where friction occurred, the temperature has reached the maximum value.

In summary, it has been concluded that the impact of cooling by convection in the decline of the temperature difference at the end of braking compared to the initial state is between 91.22% and 91.74%.

It has been deduced that the cooling amounts at the initial of braking are higher than the cooling amounts at end of braking. This situation is explained by the fact that the heat transfer coefficient values are a function of the angular velocity.

While the total heat flux values reached their maximum in cold region of the disc, minimum values were obtained in the outer radius of the disc.

It has determined that the disc temperature increased as the number of repetitions increased because the disc's temperature could not return to its initial state with the cooling occurring between cycles in repeated braking.

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