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► **To cite this version:**

Adam Adamowicz, Piotr Grzes. Influence of Convective Cooling on a Disc Brake Temperature Distribution During Repetitive Braking. Applied Thermal Engineering, 2011, 31 (14-15), pp.2177. 10.1016/j.applthermaleng.2011.05.016 . hal-00781352

HAL Id: hal-00781352

<https://hal.science/hal-00781352>

Submitted on 26 Jan 2013

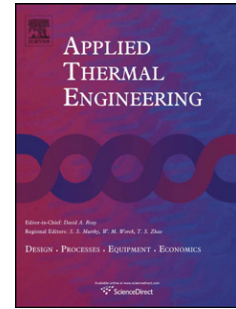
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Accepted Manuscript

Title: Influence of Convective Cooling on a Disc Brake Temperature Distribution During Repetitive Braking

Authors: Adam Adamowicz, Piotr Grzes



PII: S1359-4311(11)00268-7

DOI: [10.1016/j.applthermaleng.2011.05.016](https://doi.org/10.1016/j.applthermaleng.2011.05.016)

Reference: ATE 3563

To appear in: *Applied Thermal Engineering*

Received Date: 21 March 2011

Revised Date: 4 May 2011

Accepted Date: 10 May 2011

Please cite this article as: A. Adamowicz, P. Grzes. Influence of Convective Cooling on a Disc Brake Temperature Distribution During Repetitive Braking, *Applied Thermal Engineering* (2011), doi: 10.1016/j.applthermaleng.2011.05.016

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INFLUENCE OF CONVECTIVE COOLING ON A DISC BRAKE TEMPERATURE DISTRIBUTION DURING REPETITIVE BRAKING

Adam Adamowicz, Piotr Grzes

*Faculty of Mechanical Engineering, Bialystok University of Technology (BUT),
45C Wiejska Street, Bialystok, 15-351, Poland*

Correspondence to: Piotr Grzes

e-mail: p.grzes@doktoranci.pb.edu.pl

phone number: (48) 85 746 93 12,

fax number: (48) 85 746 92 10.

ABSTRACT

The purpose of this study is to evaluate an impact of convective mode of heat transfer on the thermal behaviour of a disc brake system during repetitive braking process with the constant velocity using fully three-dimensional finite element model. The transient thermal analysis to determine the temperature distributions on the contact surface of a disc brake is performed. The issue of non-uniform frictional heating effects of mutual slipping of a disc over fixed pads is tested using FE models with the several possible to occur in automotive application heat transfer coefficients. To have a possibility of comparison of the temperature distributions of a disc during cyclic brake application, the energy transformed during time of every analyzed case of braking process and the subsequent release periods was equal. The time-stepping procedure is employed to develop moving heat source as the boundary heat flux acting interchangeably with the convective cooling terms. The difficulties accounted for the accurate simulation of heating during spin of the rotor is omitted by the use of the code, which enable shaping curves responsible for the thermal flux entering the disc at subsequent moments of time. The resulting evolution of temperature on the friction surface reveals a wide range of variations, distinguishing periods of heating and cooling states. It has been established, that during single braking the convective cooling has insignificant influence on the temperature distributions of a disc brake, consequently is not able to prevent overheat problem. However the brake release period after the braking operation, when the velocity of the vehicle remains on the same level, results in considerable decrease of temperature.

Key words: Braking, Pad/disc system, Finite element method, Frictional heating, Convective cooling.

NOMENCLATURE

- A_j – area of the finite element on the friction surface, m^2 ;
 A_q – area of the finite element on the friction surface covered by the pad, m^2 ;
 c – specific heat, $J/(kgK)$;
 $[C]$ – heat capacity matrix;
 f – coefficient of friction;
 h – heat transfer coefficient, $W/(m^2K)$;
 k – thermal diffusivity, m^2/s ;
 K – thermal conductivity, $W/(mK)$;
 $[K]$ – conductivity matrix;
 p_0 – contact pressure, MPa ;
 q – intensity of heat flux, W/m^2 ;
 $q_j(t)$ – heating function, W/m^2 ;
 r – inner radius, m ;
 R – outer radius, m ;
 $\{R\}$ – heat source vector;
 t – time, s ;
 t_{sn}^i – moment of time which initiates the succeeding brake engagements for n -step of computations, $i = 0, \dots, n-1$, s ;
 t_s – braking time for the first step of the analysis, s ;
 t_{sn} – time of a single brake operation for n -step of computations, s ;
 t_{cn} – time of a single brake release for n -step of computations, s ;
 $t_T = n(t_{sn} + t_{cn})$ – total time of computations for every n -step, s ;
 Δt – time step, s ;
 T – temperature, $^{\circ}C$;
 T_{∞} – ambient temperature, $^{\circ}C$;
 T_0 – initial temperature, $^{\circ}C$;
 $\{T\}$ – temperature vector;

V_0 – velocity of the vehicle, km/h ;

Δx – the mesh size (smallest element dimension), m ;

z – axial coordinate, m ;

Greek symbols

γ – heat partition ratio;

δ – thickness, mm ;

θ – circumferential coordinate, deg ;

ρ – density, kg/m³ ;

θ_0 – cover angle of pad, deg ;

ω_0 – angular velocity of a disc, 1/s ;

Subscripts

d – indicates disc;

p – indicates pad;

w – indicates wheel;

1. INTRODUCTION

Physical interaction between the mating parts of a given brake system during application results in transformation of work done into frictional heat. While the heat is generated, the dissipation of energy proceeds by conduction to adjoining components through the brake assembly and hub, convection from the exposed surfaces according to Newton's law of cooling to surrounding air and radiation to nearby parts. While the influence of radiation heat transfer on the total amount of dissipated energy is frequently considered as insignificant during normal braking conditions either in automotive [1-3] or hoist mine application [4-6], conduction and convection play a comparably central role in contribution of heat exchange of the brake system [7,8]. Furthermore the rotor and neighbouring components of the brake act as the temporary thermal storage devices thus their further cooling participates in proper brake operating. Nonetheless convective magnitude may vary significantly from the conditions of forced, mixed until natural convection due to the velocity of the airflow in the area of a friction pair (laminar or turbulent airflow), time of operation after the brake application as well as geometrical type of the brake either drum or disc. The evident emphasis placed on continually being developed type of disc brakes over drum brakes, stems from the several advantages over the latter. Typically disc brakes have vanes which enlarge the surface of cooling and thereby reduce overheat problem. However in actual the convective heat transfer can not enable sufficient cooling after the brake application [9].

According to the Newton's law the accuracy of calculation of the intensity of the heat flux on the boundary surface of body dependent, except the difference of temperatures on this surface, from precision determining of the proportionality factor known as the heat transfer coefficient. Originally had been developed formulas for the heat transfer coefficient based on experimental correlations with air as the cooling medium, flowing parallel to smooth plain surfaces were frequently employed in the investigation of the thermal behaviour of clutches or brake systems [10,11]. As the turbulent flow took place and the value of Prandtl number was close to unity, the heat transfer coefficient did not depend on speed. The formula for the heat transfer coefficient, which depends on the speed of the vehicle and the drum diameter, was proposed for the evaluation of temperature during both single stop and repetitive brake applications in article [12].

One of ways accounting of influence of convective cooling on temperature in disc brake is introduction of the heat transfer coefficient (the Biot's number) to the heat partition ratio. The various forms of this ratio and their influence on contact temperature in a pad/disc

system is presented in article by Yevtushenko and Grzes [13]. Despite the fact, that have been made advances of numerical methods for the calculations of temperature distributions during sliding, based on a motion equations of a fluid, the evaluations of a contact temperature in disc brake are still examined in experimental tests [14-17]. The enough well correlations between analytical and experimental methods of determination of the heat transfer coefficient for both solid and ventilated type of a disc have been obtained in articles [18-24].

Due to the fact of complexity of the phenomenon of frictional heat generation, the problem of concurrent fluid flow within the channels of the ventilated type of disc brakes is often analysed in individual studies based on the CFD method [25-28]. Nonetheless often an average, constant value of the heat transfer coefficient is used at temperature calculations in disc brakes [3,29,30]. The review of the FEM applications of frictional heat generation in disc brakes and clutches systems was developed in an article by Yevtushenko and Grzes [31].

The three-dimensional FE model for calculation of the temperature in the pad/disc system at single braking, based on the theory of a moving heat sources, has been proposed in authors' previous study [32]. In this article the thermal finite element analysis of a solid disc brake during single as well as repetitive braking process with different number of brake application was developed. We investigate the influence of the heat transfer coefficient on the temperature distributions of a brake rotor. In order to determine comprehensive terms of convective cooling during considered time of braking and interchangeable phases free from heating, an individual values of the convective heat transfer possible to occur in automotive disc brakes were implemented to the model [33]. The dimensions as well as the thermophysical material properties of a solid disc brake were adopted from the braking process of a common passenger vehicle.

2. STATEMENT OF THE PROBLEM

The mutual sliding process of the mating parts of a brake system results in the heat generation. The forces that act on friction elements against the disc, opposes motion and the vehicle slows down or the velocity remains on the same level (mountain descent). The accumulated amount of kinetic and potential energy is converted into thermal energy between the frictional elements of a brake. The heat generated on the contact surfaces of a pad and a disc, is dissipated by the forced convection due to enlarged airflow and the natural convection after the full stop, whereas conduction absorbs energy from the interface of the friction pair. Because of the relatively low temperatures attained and negligibly small share in the total

energy that is transformed during the braking action, the thermal radiation in this study is not under consideration.

In the temperature analysis of the disc brake system solely two parts of the assembly are typically taken into account, namely the rotating disc and immovable pad (Fig. 1). Furthermore the considered model of heat transfer of the rotor is limited to the area crucial for the temperature change omitting the hat section by which the disc is connected to the axle for the convenience purposes without losing the general implication. However if the material properties and operation conditions were able to cause the temperature rise in this region, obviously the results would be misleading. In this study the preliminary tests were carried out to assess the range of variable temperature of the rotor.

By virtue of the symmetry of the problem about the mid-plane owing the caliper's mechanism designed to equalize the forces that act on pads, the analyzed region was restricted to the half of the entire disc of thickness δ_d and one pad. Moreover solely the disc volume was included during computations.

It is assumed that the thermophysical properties of materials are isotropic and independent of temperature. Such an assumption make a case for the fact of relatively low temperature attained during the investigated process of frictional heating of brake components. The nominal surface of contact of a disc equals the real surface of contact thereby the contact pressure is uniform at every moment of time t of braking and equals p_0 . The angular velocity ω_0 is constant during the analysis. Assuming that the total friction energy is converted into heat, the intensities of heat fluxes that enters from the contact surface $z = 0$ into the pad and into the disc are calculated from the formulae

$$q_p(r, \theta, t) = \gamma p_0 \omega_0 r, \quad r_p \leq r \leq R_p, \quad 0 \leq \theta \leq \theta_0, \quad 0 \leq t \leq t_s, \quad (1)$$

$$q_d(r, \theta, t) = (1 - \gamma) p_0 \omega_0 r, \quad r_p \leq r \leq R_p, \quad 0 \leq \theta \leq 2\pi, \quad 0 \leq t \leq t_s, \quad (2)$$

where γ is the heat partition ratio [34]

$$\gamma = \frac{K_p \sqrt{k_d}}{K_p \sqrt{k_d} + K_d \sqrt{k_p}}. \quad (3)$$

and (r, θ, z) are variables of the cylindrical coordinate system (Fig. 1). We note that the sum of intensities of heat fluxes (1) and (2) is equal to specific capacity of friction $q = fp_0 \omega_0 r$. On other surfaces of a disc we assume, that:

– the surface placed in the disc's mid-plane being a plane of symmetry is insulated;

- in the region of the contact surface free from frictional heating, on the internal and external cylindrical surfaces of a disc the convective cooling takes place;
- the heat exchange through forced convection was assumed to be independent of the radius within the range of maximal and minimal radius of the pad and consequently all of the exposed surfaces of disc have the same convective cooling conditions for specific computational case.

Under mentioned above assumptions we find the distribution of the three-dimensional transient temperature field $T(r, z, \theta, t)$ in rotating disc from the solution of the following boundary-value heat conduction problem [32]:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} = \frac{1}{k_d} \left(\frac{\partial T}{\partial t} + \omega \frac{\partial T}{\partial \theta} \right), \quad r_d \leq r \leq R_d, 0 \leq \theta \leq 2\pi, 0 < z < \delta_d, t > 0, \quad (4)$$

$$K_d \frac{\partial T}{\partial z} \Big|_{z=0} = q_d(r, \theta, t), \quad (r, \theta) \in \Gamma = \{r_p \leq r \leq R_p, 0 \leq \theta \leq 2\pi\}, 0 \leq t \leq t_s, \quad (5)$$

$$K_d \frac{\partial T}{\partial z} \Big|_{z=0} = h[T_\infty - T(r, \theta, t)], \quad (r, \theta, z) \in \Omega_1 = \{r_d \leq r \leq r_p, 0 \leq \theta \leq 2\pi, z = 0\}, t \geq 0, \quad (6)$$

$$K_d \frac{\partial T}{\partial r} \Big|_{r=R_d} = h[T_\infty - T(\theta, z, t)], \quad (r, \theta, z) \in \Omega_2 = \{r = R_d, 0 \leq \theta \leq 2\pi, 0 \leq z \leq \delta_d\}, t \geq 0, \quad (7)$$

$$K_d \frac{\partial T}{\partial r} \Big|_{r=r_d} = -h[T_\infty - T(\theta, z, t)], \quad (r, \theta, z) \in \Omega_3 = \{r = r_d, 0 \leq \theta \leq 2\pi, 0 \leq z \leq \delta_d\}, t \geq 0, \quad (8)$$

$$\frac{\partial T}{\partial z} \Big|_{z=\delta_d} = 0, \quad (r, \theta, z) \in \Omega_4 = \{r_d \leq r \leq R_d, 0 \leq \theta \leq 2\pi, z = \delta_d\}, t \geq 0, \quad (9)$$

$$T(r, \theta, z, 0) = T_0, \quad r_d \leq r \leq R_d, 0 \leq \theta \leq 2\pi, 0 \leq z \leq \delta_d, \quad (10)$$

where the function $q_d(r, \theta, t)$ has the form (2), (3).

3. FINITE ELEMENT FORMULATION

The value 10^5 of the Peclet number Pe is typical for the disc brakes and clutches [9,35]. It is known that for values $Pe > 2$ the Galerkin-type finite element approximation leads to oscillations of the solutions in space [36]. Consequently such an approach for the solution of the problem of heat conductivity with fast-moving heat source becomes unfeasible. However it does not become apparent solely in the finite element analysis as all other spatial discretization techniques have the same difficulties. For the purpose of eliminating spatial oscillations the procedure such as Petrov-Galerkin [37] and Streamline Upwind Petrov

Galerkin (SUPG) [36] are commonly employed equivalent unwinding schemes [38]. In this paper we omit unstable results and oscillations to occur neglecting the component of a velocity from analytical formulation of the problem (Eq. 4), and considering in actual fixed disc. The problem of a moving heat source is solved by means of specific formulation of a boundary heat flux which acts separately on every of element on the rubbing path so that the simulated transition of a pad is conducted [32].

The discrete FE analogue of the differential equation (4) has the matrix form

$$[\mathbf{C}]\frac{d\{\mathbf{T}\}}{dt} + [\mathbf{K}]\{\mathbf{T}\} = \{\mathbf{R}\}. \quad (11)$$

In order to solve the ordinary differential equation (11) the following relation of the Crank-Nicolson method was used [39]:

$$\frac{1}{\Delta t}[\{\mathbf{T}\}_{t+\Delta t} - \{\mathbf{T}\}_t] = (1 - \beta)\left\{\frac{d\mathbf{T}}{dt}\right\}_t + \beta\left\{\frac{d\mathbf{T}}{dt}\right\}_{t+\Delta t}, \quad (12)$$

where $\{T\}_t$ and $\{T\}_{t+\Delta t}$ are the values of the temperature at time moments t and $t+\Delta t$, respectively. Substituting relations (12) into Eq. (11) we obtain the following system of linear algebraic equation:

$$([\mathbf{C}] + \beta\Delta t[\mathbf{K}])\{\mathbf{T}\}_{t+\Delta t} = ([\mathbf{C}] - (1 - \beta)[\mathbf{K}]\Delta t)\{\mathbf{T}\}_t + (1 - \beta)\Delta t\{\mathbf{R}\}_t + \beta\Delta t\{\mathbf{R}\}_{t+\Delta t}, \quad (13)$$

where the value of the parameter $0.5 \leq \beta \leq 1$ is chosen from the conditions of achievement of the set accuracy and stability of the computational scheme.

In order to calculate temperature distributions of a disc brake, appropriate mesh division regarding respective coordinates of the model is crucial. High temperature gradients require very fine mesh, therefore in the area adjacent to the contact surface, lower dimension of the elements was established. Nonetheless for the purpose of validation of the final mesh, tests of different grids of elements owing every direction with the speed of the vehicle of 25 km/h were carried out. As a result the final mesh consists of linear 43200 eight-node hexahedral elements and 33693 nodes (360 elements in the circumference, 4 in axial direction, and 20 in the radial direction (Fig. 1). As the temperature gradients in region beneath friction surface were relatively low, the mesh division was of “paver” type (irregular). The finite element generation was accomplished by the code corresponding with further boundary conditions of a moving heat source problem formulation.

4. BRAKING SCHEME AND NUMERICAL ALGORITHM

The repetitive braking process to determine crucial factors that affect the interface temperature was studied in article by Qi and Day [30]. The sliding speed, braking load, type of friction materials as well as the number of braking applications was included in the analysis. However, despite the fact of complexity of the investigation and use of a statistical design of experiments incorporating experimental tests with thermocouple technique, no emphasis was placed on the possible to occur in reality range of the heat transfer coefficient that may influence the resulting temperature.

Therefore, we proposed the following scheme of interchangeable braking with the release periods incorporating different number of brake application with the same global time (Fig. 3). It was assumed that during individual braking, the number of brake applications increases twice from single ($n=1$) braking to five ($n=5$) brake applications for the last case. Furthermore, duration of every phase of frictional sliding decreases twice, which gives eventually the same total time. The time of a single braking for the first computational step equals $t_{s1} = 5$ s. After that time, phase of cooling of a disc takes place ($t_{c1} = 50$ s). For the cases of repetitive braking actions, the initial time of succeeding brake applications occurs at the moment of time resulting from the relation $t_{sn}^i = (t_T/n)i$, whereas duration of each single braking phase lasts $t_{sn} = t_s/n$. For example, if we consider three brake applications during the total time $t_T = 55$ s, the moments of succeeding brake applications equal $t_{sn}^i = (t_T/n)i = t_{s3}^0 = (55s/3)0$, $t_{s3}^1 = (55s/3)1$, $t_{s3}^2 = (55s/3)2$, respectively, and each single phase of disc heating will last $t_{sn} = t_s/n = t_{s3} = 5/3$ s. The coefficient which denotes relation between the cooling time to the braking time equals always $t_{cn}/t_{sn} = 10$. The constant value of 100 km/h of the vehicle speed was used in calculations.

During mutual sliding process of rotational rotor and immovable pad, the friction surface of a disc is subjected to non-axisymmetric thermal load. A fully 3D FE model of a thermal behaviour of the disc rotor is used to simulate the repetitive braking with different time of brake applications during the subsequent steps of the analysis. In case of braking with constant speed (such as mountain descent), the time of every rotation of a disc is equal regarding specific spot on the circumference within the contact region. Both heating and cooling phase may be distinguished. The heating time equals $\theta_0/2\pi$ of time of one rotation of the wheel, whereas cooling phase equals $1 - \theta_0/2\pi$ of time of one rotation. In the present

numerical analysis the boundary intensity of heat flux, as well as cooling terms are calculated individually for every element of the mesh (regarding radial location) for the specific time t .

The final mesh with the moving thermal flux entering the disc is shown in Fig. 2. The intensity of heat flux as well as the convective terms are interchangeably calculated in the region of contact surface. The most important in the proposed approach is such a formulation of boundary conditions established on the finite elements of the rubbing path so as to simulate the transition of the heat source. Due to the assumption of uniformity of the intensity of heat flux in the region of heat generation, it is necessary for every element located on the rubbing path to determine a function $q_j(t)$. The shape of that function arises directly from the rate of cover of a specific element by the boundary thermal flux and ranges from 0 (pad beyond the element) to $q_d(t)$ (full cover of a specific element by the pad). The following form of that function is given by

$$q_j(t) = q_d(t) \cdot \begin{cases} 0 & \text{if } (A_j \cap A_q)(t) = 0, \\ \vdots & \\ (0.2) & \text{if } 0 < (A_j \cap A_q)(t) < A_j, \\ \vdots & \\ 1 & \text{if } (A_j \cap A_q)(t) = A_j. \end{cases} \quad (0.2)$$

Calculation of the heating curves of boundary heat flux for every finite element was developed numerically. The curves of the intensity of heat flux, which correspond to the elements from Fig. 4, are shown in Fig. 5. In the element denoted (c), which is located on the verge of the rubbing path, incomplete covering may be observed due to the pad position relating to the disc (1/5 of total area of (c) element is covered by the pad). Shape of the function depends directly on a position in relation to circumferential coordinate θ , and these functions are shifted to each other in time. Notice that for time $t = 0.0178$ s the area of (c) element A_j is covered by $A_q = 0.2A_j$, therefore the heat flux that acts on element equals $q_j = 0.2q_d$.

The temperature distributions calculated in this study by means of the above numerical approach agree well with the previous authors' calculations on frictional heat generation problem [32] and an issue of its separation between the components of the brake system [13], in which comparison with experimental data derived from Zhu et al. [6] was made.

5. RESULTS AND DISCUSSION

The 3D finite element analysis of the temperature of the disc during single braking was developed in article by Adamowicz and Grzes [32]. In given article we expanded this analysis on a case of a repetitive braking process. The dimensions of the brake system, the thermophysical properties of materials, as well as the parameters of operation used in calculations are given in Tab. 1 [3]. The calculations of the temperature distributions on the friction surface of a disc $z = 0$ m were made for the schemes of the braking process from single braking with the subsequent period of cooling to five brake applications.

The first step of this analysis was to study the temperature distributions during $t_{s1} = 5$ s of frictional sliding phase with the constant speed of the vehicle of 100 km/h and the following cooling phase lasting $t_{c1} = 50$ s. Further single brake application will be extended to the case of repetitive braking process according to the algorithm presented in Fig. 3. We note, that the cooling phases after the separation of the brake components allow to use relatively long step of the time due to uniform cooling of the entire disc. However, the numerical simulation of periodic heating during rotational motion of a disc over fixed pad requires considerably shorter time step. Therefore the data obtained after the process of frictional heating (during each of the considered braking scheme) were implemented to the separate calculation of disc temperature distributions concerned the uniform cooling of a disc. Furthermore, each case of braking according to the established order of the brake application incorporates each single value of the heat transfer coefficient from the range of $0 \div 100$ W/(m²K) with the step of 10 W/(m²K).

The evolution of temperature for the chosen circumferential location $\theta = 0^\circ$ on the contact surface of the disc is presented in Fig. 6. In order to study the contact temperatures generated primarily during 5 s of continuous sliding of the brake components, the following period of the brake release incorporating exclusive cooling of 50 s in this figure it is not shown. The convective terms of the most intensive cooling $h = 100$ W/(m²K) as well as fully adiabatic conditions $h = 0$ W/(m²K) for the exposed surfaces of a disc are confronted for corresponding radial locations. Assuming different variation of the velocity of vehicle as well as the friction force, the resulting temperature evolution may vary significantly. However the presence of heating (concave curve) and cooling phases (convex curve) for the selected spot on the circumference, which describes the operation of applied thermal flux over the rotational disc is evident. Such a phenomenon agree well with several experimental and numerical studies on frictional heat generation in disc brakes [9,30,40,41]. During initial,

relatively short period of braking, the temperature increases nonlinearly. Similar effect was also observed by Bialecki and Wawrzonek [14] for a disc coal mining application disc brake using code based on the finite element volume approach and the experimental tests as well as in article by Adamowicz and Grzes [32]. Since the same value of the heat transfer coefficient was established on the surface of a disc free from heating, the observable decrease of discrepancy of temperatures with the decrease of a radius may be accounted directly by the expression for the convective heat flux.

In the stop time moment the differences of temperature on the contact surface is equal 15.68, 7.73, 3.86 and 2.19 °C for radial distance 113.5, 95, 76.5, 66 mm respectively. We note, that the values of the heat transfer coefficient $h = 0$ and $h = 100 \text{ W}/(\text{m}^2\text{K})$ refer to extreme conditions of cooling [33]. The value of this coefficient in actual process of braking is between these limiting values. Consequently the obtained results from the relatively simple geometry of the analyzed disc provide evidence to support the claim that the influence of convective heat transfer on the temperatures of the contact surface of the disc with regard to total generated heat during single braking without the period of cooling is insignificant [9,25].

The evolutions of temperature on the contact surface of a disc at $r = 95 \text{ mm}$, $\theta = 0^\circ$ and two values of the heat transfer coefficient $h = 0$ and $100 \text{ W}/(\text{m}^2\text{K})$ for single and five brake applications are shown in Fig. 7. Due to the constant speed of the vehicle (100 km/h) during entire process of simulation of 55 s, continual increase of temperature for both single and repetitive braking operation during phase of mutual sliding is noticeable. As it has been established earlier, influence of convective cooling on the temperature of disc rubbing surface in the stop time moment is insignificant (Fig. 6). However the following period of cooling for single braking reveals considerable decrease of temperature. The slope of the temperature curve representing the solution for single braking ($n = 1$) at the adiabatic boundary conditions indicates that the temperature within the entire volume of a disc equalizes at the end of the process of heating. The difference of temperatures between the model with applied $h = 0$ and $100 \text{ W}/(\text{m}^2\text{K})$ reached at the end of the process of single braking equals $(389.3 - 284.5)^\circ\text{C} = 104.8^\circ\text{C}$. In case of repetitive braking process ($n = 5$) the maximal temperature on the friction surface is attained just before the end of operation. Therefore the time to cool the disc after it is heated to the highest temperature is shorter. As a result the difference of temperatures at the end of the operation during 5 brake applications relating to single brake application is smaller and equals $(405.0 - 335.6)^\circ\text{C} = 69.4^\circ\text{C}$.

The results obtained single as well as repetitive braking process for the applications $n = 1, 2, 3, 4, 5$ are presented in Figs. 8-10. Dependence of the maximal temperature on the contact surface (the solid lines) as well as the temperature at the time moment $t = 55$ s (the dashed lines) on the heat transfer coefficient for the fixed location of the mean radius $r = 95$ mm are shown in Fig. 8. The maximal temperature in each case occurred in the period prior to the moment of brake release (last brake application). As can be seen the highest temperature $T = 561.34$ °C is attained for the single brake application ($n = 1$) then its value lowers linearly with the increase of the heat transfer coefficient to $T = 553.82$ °C. This curve reveals the lowest slope which stems from the fact that the moment follows after the sliding of the brake components without succeeding cooling periods. When the number of brake application within the same time of 55 s increases the decrease of maximal temperature for that process is noticeable. Every increase of a number of brake applications causes more steep temperature curve. Exactly the opposite situation takes place owing the temperature at the moment of time $t = 55$ s. The biggest slope of the temperature curve is observable for single braking ($n = 1$). The total time allocated for disc continuous cooling equal (45 s) for each number of brake application in that case doesn't include periods of braking.

The corresponding dependences for internal boundary of a contact region ($r = 66$ mm) are shown on Fig. 9. Unlike the data presented on Fig. 8, here we see the non-linear reduction of temperature with an increase of the heat transfer coefficient. One more difference between the results shown on Figs. 8 and 9 consists that the maximal temperatures and temperatures at end of operation on the internal surface of a disc coincide for number of brake application of 2, 3, 4, 5 (Fig. 9), which indicates continuous increase of temperature on the internal surface of a disc. Nonetheless for single brake application ($n = 1$) the maximal temperature and the temperature at the final moment of the process (dashed curve) more rapidly within the range of the heat transfer coefficient from 40 to 100 W/(m²K).

The change of temperature on the contact surface of a disc for the fixed location $r = 95$ mm with the increase of number of brake applications are presented in Fig. 10. The maximal values of temperature reached during repetitive braking as well as the temperatures at the moment $t = 55$ s for fixed values of the heat transfer coefficient are confronted with the number of brake applications. In can be seen that the maximal temperatures decrease with the increase of number of brake application from one to five, whereas the temperature at the end of the process is lower for the lower number of brake application. At the further increase of number of braking these dependences are kept, but in the smoother form.

Since the location of a mean radius is the least affected by the finite geometry of the disc, the outcomes of temperature at this characteristic point may provide more general information of a nature of the convective cooling problem of the rotating disc. Therefore the isotherms on the friction surface at $r = 95$ mm, $\theta = 0^\circ$ after the braking operation when the velocity of the vehicle remains on the same level are shown in Fig. 11. The temperature decreases with an increase of the heat transfer coefficient, furthermore it may be clearly observed that at the fixed value of the heat transfer coefficient the rapidity of the descent of temperature diminishes with the time. This phenomenon is more evident for the lowest values of the heat transfer coefficient. Moreover for the chosen range of the heat transfer coefficient the discrepancy of the resulting temperatures is greater for its lower values.

The corresponding isotherms for internal boundary of a contact region ($r = 66$ mm, $\theta = 0^\circ$) are shown in Fig. 12. The presented nodal temperatures on the contact surface are in the longest radial distant from the rubbing path, thus the increase of temperature despite the end of the braking phase (frictional heating) is evident and is more rapid at the beginning of the cooling period. However after the time of about 35 s intense cooling conditions (100 W/(m²K)) cause the decrease of the temperature.

6. CONCLUSIONS

The aim of this paper was to investigate the temperature distributions caused by the frictional sliding process of the disc brake components with a special emphasis placed on the convective heat exchange from the exposed surfaces of a disc. The obtained results allow us to formulate the following conclusions:

- the highest difference of temperatures on the contact surface of a disc (mean radius of the rubbing path) subjected to continuous frictional sliding process for the applied extreme boundary convective conditions (range from 0 to 100 W/(m²K)) is lower than 3%. Thus in terms of single braking process the convective heat transfer mode doesn't allow to significantly lower the temperatures of the rotor for typical operation conditions of passenger vehicle;
- when the repetitive brake application with the constant velocity occurs, on the understanding, that the total transformed energy is equal (the same total braking time during each case of operation) the maximal temperature (fixed spot on the friction surface) attained during the operation decreases with the number of brake applications, whereas the opposite situation takes place for the temperature at the end of the process;

– the maximal temperature as well as the temperature reached at the end of the process on the contact surface occurred during repetitive braking with different convective cooling conditions (but constant velocity of the vehicle) for the relation of cooling time $t_{cn} = 50$ s to the braking time $t_{sn} = 5$ s equal 10, decreases with the increase of the heat transfer coefficient, and the relationship was found to be approximately linear with time. The slope of the temperature curves depends upon the number of brake applications.

Finally, we note that during the real braking process, an increase of a number of brake applications affects the conditions of a local contact, which may results in the increase of a real contact area [30]. Hence assuming that the total load, relative sliding velocity and the coefficient of friction is constant, the decrease of average normal pressure is apparent. As a result the average temperature on the rubbing path will be lower. However in this study we analyze solely the question of a moving heat source over a rigid body therefore physical interactions do not share in the simulation process.

7. ACKNOWLEDGEMENTS

The authors are pleased to acknowledge support from the project "Increasing the capacity of universities as a factor of development of knowledge-based economy" financed by the ESF Human Capital Operational Programme and State Budget of Republic of Poland.

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Figure Captions

1. Heat dissipation from a pad/disc tribosystem.
2. A schematic diagram of a disc brake with the boundary conditions and section of the FE mesh.
3. Scheme of brake applications for thermal analysis.
4. Numerical algorithm of a boundary heat flux.
5. Example of a heating function for the chosen finite elements on the rubbing path.
6. Evolution of temperature on the contact surface of a disc at $\theta = 0^\circ$ for two values of the heat transfer coefficient $h = 100$ (solid curves) and $h = 0$ W/(m²K) (dashed curves) at single braking.
7. Evolution of temperature on the contact surface of a disc at $\theta = 0^\circ$ for the heat transfer coefficient $h = 0 \div 100$ W/(m²K) with the step of 10 W/(m²K) at single ($n = 1$) and repetitive ($n = 5$) braking.
8. Maximal temperature (solid curves) on the contact surface and temperature at the end of the process (dashed curves) at the mean radius $r = 95$ mm versus the heat transfer coefficient for different number of brake applications n .
9. Maximal temperature (solid curves) on the contact surface and temperature at the end of the process (dashed curves) at the radius $r = 66$ mm versus the heat transfer coefficient for different number of brake applications n .
10. Maximal (solid curves) temperature on the contact surface and temperature at the end of the process (dashed curves) at the mean radius $r = 95$ mm versus number of brake applications n for different heat transfer coefficients.
11. Nodal temperature on the contact surface of a disc ($r = 95$ mm mean radius, $\theta = 0^\circ$) during phase of cooling after single braking against the heat transfer coefficient.
12. Nodal temperature on the contact surface of a disc ($r = 66$ mm minimal radius, $\theta = 0^\circ$) during phase of cooling after single braking against the heat transfer coefficient.

- Convection does not allow to lower temperature of disc during single braking process
- Maximal temperature of disc decreases with number of brake applications
- Temperature at the end of braking increases with number of brake applications

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Table 1: Operation conditions and thermophysical properties of materials from Talati and Jalalifar [3]

Items	Disc	Pad
thermal conductivity, K [W/(mK)]	43	12
specific heat capacity, c [J/(kgK)]	445	900
density, ρ [kg/m ³]	7850	2500
inner radius, r [mm]	66	76.5
outer radius, R [mm]	113.5	
pad arc length, θ_0 [deg]		64.5
thickness, δ [mm]	5.5	10
radius of the wheel, R_w [mm]	314	
velocity of the vehicle, V_0 [km/h]	100	
total braking time, t_T [s]	55	
heat transfer coefficient, h [W/(m ² K)]	0÷100	
contact pressure, p_0 [MPa]	3.17	
coefficient of friction, f	0.5	
initial temperature, T_0 [°C]	20	
ambient temperature, T_∞ [°C]	20	

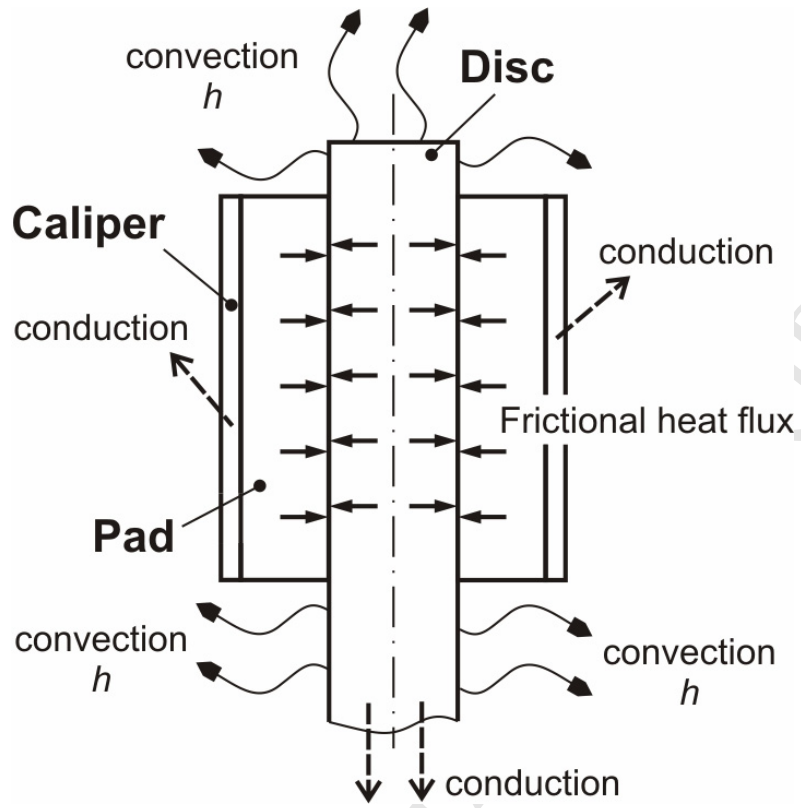


Fig. 1.

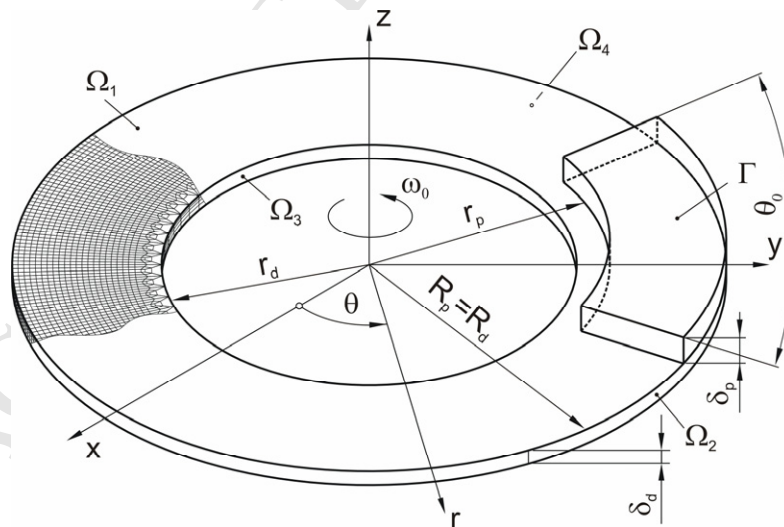


Fig. 2.

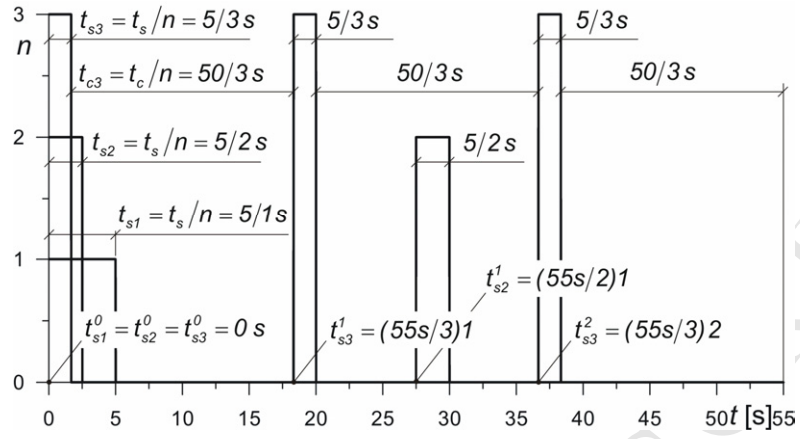


Fig. 3.

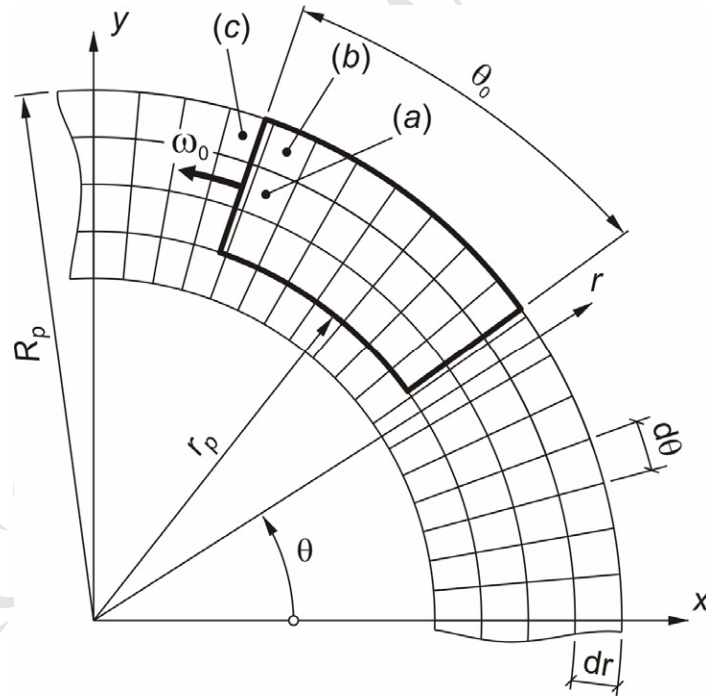


Fig.4.

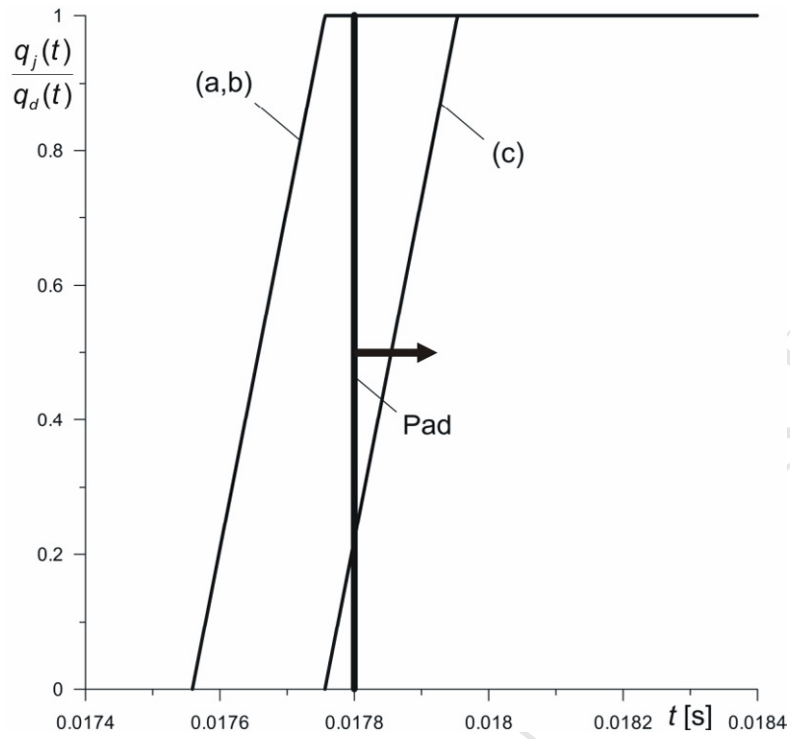


Fig. 5.

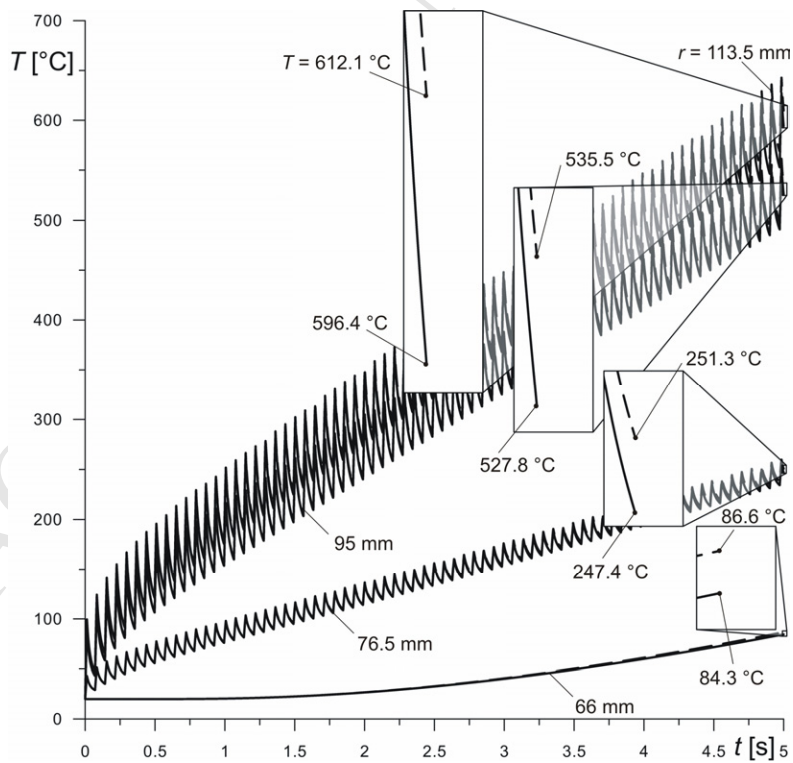


Fig. 6.

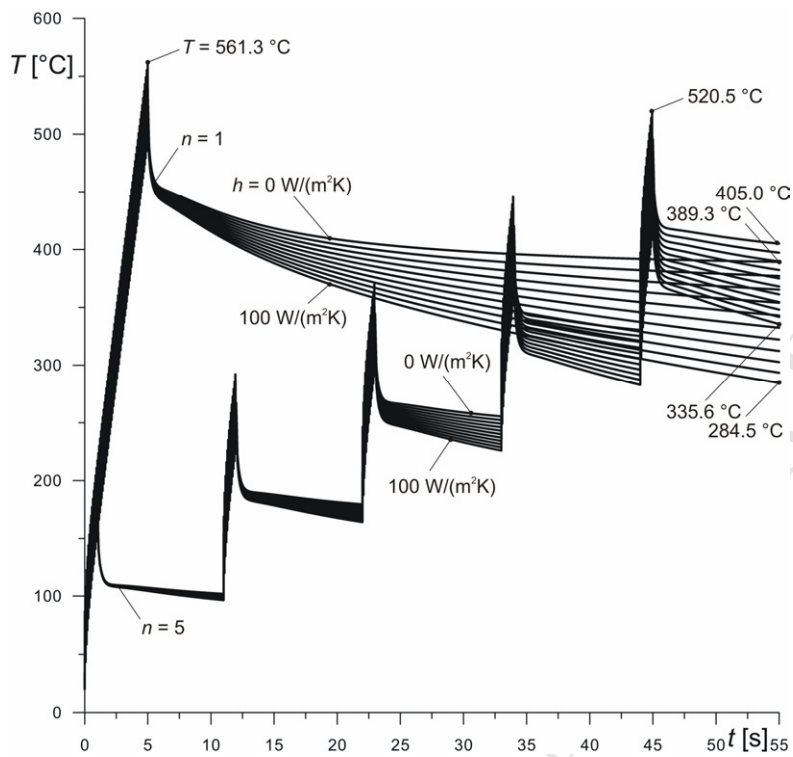


Fig. 7.

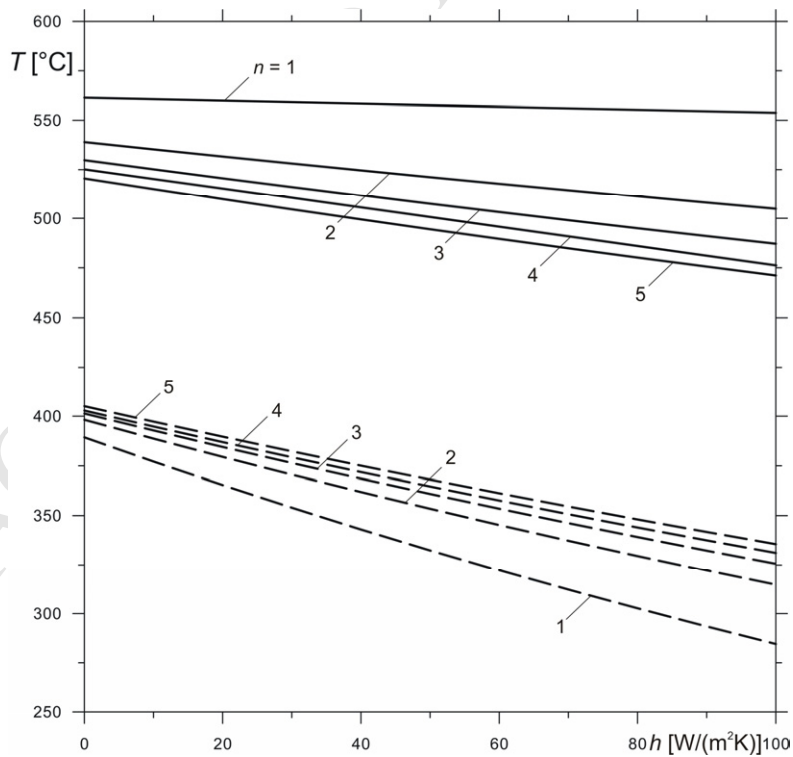


Fig. 8.

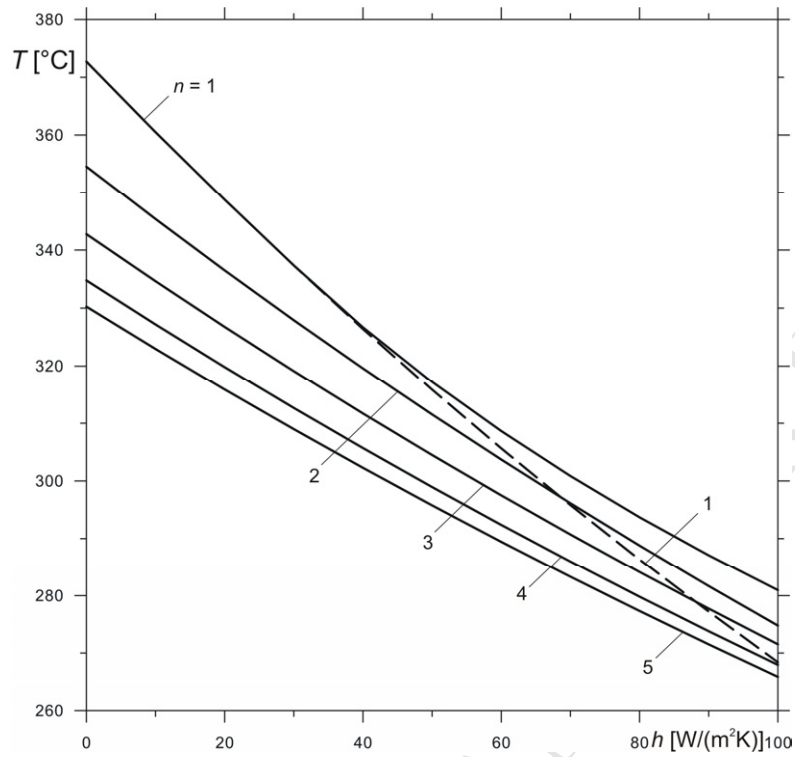


Fig. 9.

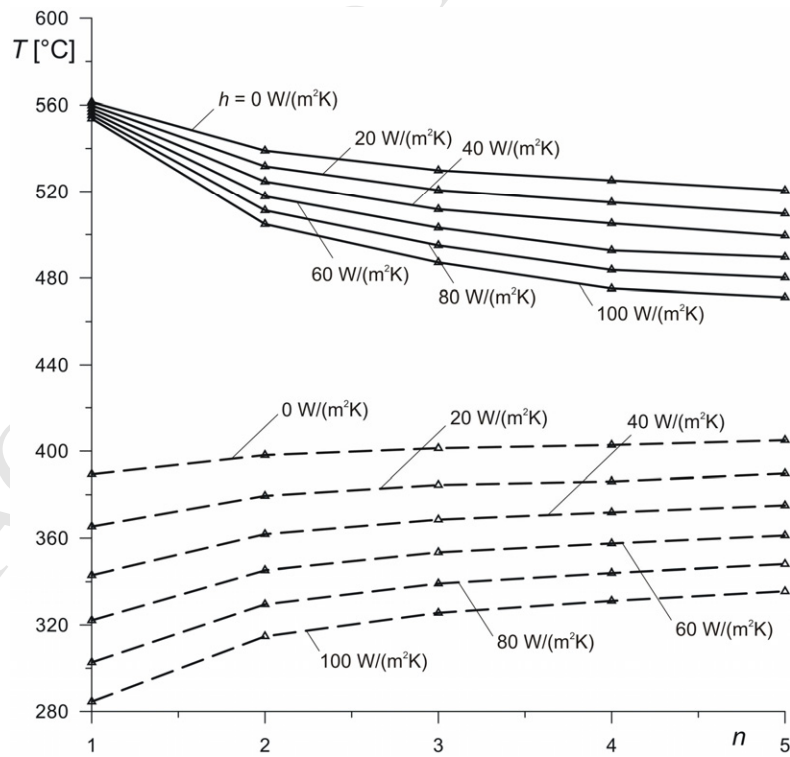


Fig. 10.

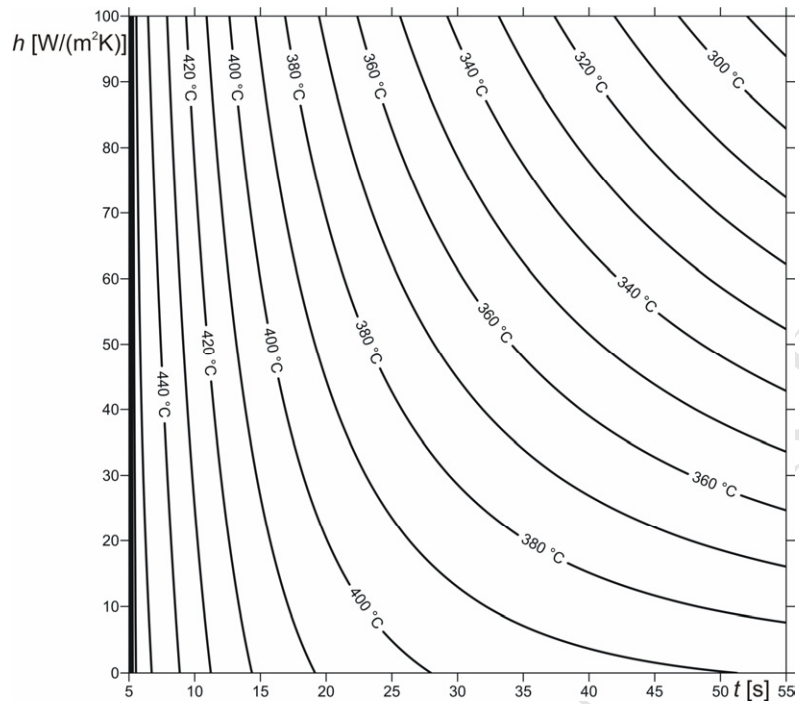


Fig. 11.

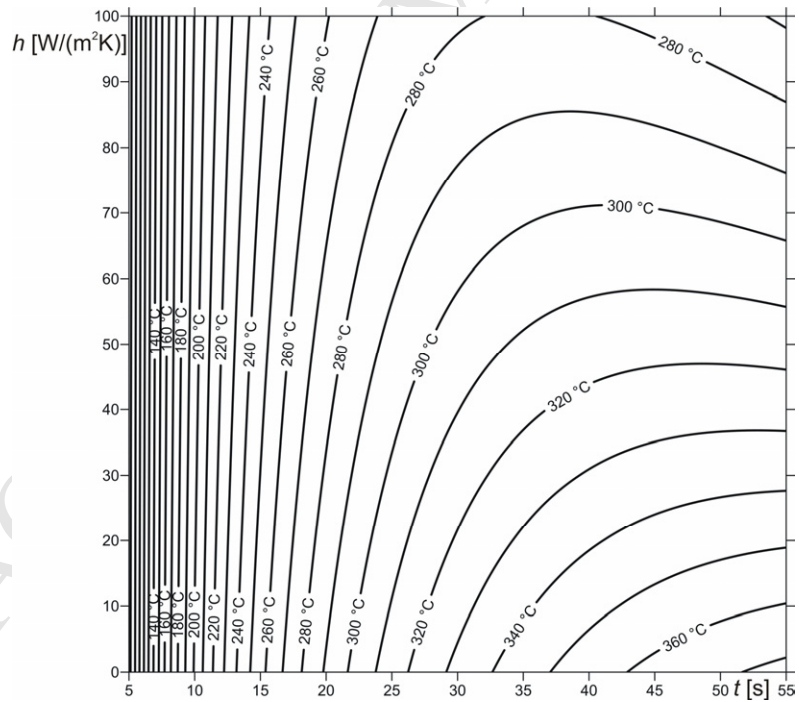


Fig. 12.