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Thermal analysis of both ventilated and full disc brake rotors with frictional heat generation

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Abstract

In automotive engineering, the safety aspect has been considered as a number one priority in development of a new vehicle. Each single system has been studied and developed in order to meet safety requirements. Instead of having air bags, good suspension systems, good handling and safe cornering, one of the most critical systems in a vehicle is the brake system. The objective of this work is to investigate and analyze the temperature distribution of rotor disc during braking operation using ANSYS Multiphysics. The work uses the finite element analysis techniques to predict the temperature distribution on the full and ventilated brake discs and to identify the critical temperature of the rotor. The analysis also gives us the heat flux distribution for the two discs.

Keywords: brake disc, convection, heat-transfer coefficient, thermal analysis, stress

1. Introduction

Braking system is one of the important safety components of a vehicle. It is mainly used to decelerate vehicles from an initial speed to a given speed. A friction based braking system is a common device to convert kinetic energy into thermal energy through a friction between the brake pads and the rotor faces. Because high temperatures can lead to overheating of the brake fluid, seals and other components, the stopping capability of a brake increases with the rate at which heat is dissipated due to forced convection and thermal capacity of the system [24].

Brake disc convective cooling has been historically studied by means of experimental and theoretical methods [17, 18] and the optimization was only boosted with the advent of modern computational resources in the late 1980s [6]. Currently, although of not simple usage and requiring previous understanding of the basics of fluid mechanics and heat transfer coupled with the knowledge of numerical flow modeling, computational fluid dynamics (CFD) has significantly gained preference in the automotive industry design process as a tool for the prediction of complex flow and heat transfer behavior in regions, where otherwise very laborious and time consuming experimental set up work would be needed. As a result, brake disc convective cooling analysis and optimization is nowadays mostly carried out using CFD commercial codes, see, e.g., [23].

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Many investigations of heat flow through ventilated disc brakes are reported in the literature. Michael and Roland [8] discussed the airflow patterns in the disc rotors. Wallis et al. [28] carried out a numerical study using the software Fluent on disc rotor blades to examine the effects of local heat and mass transfer of the axial gap distances for a single co-rotating disc. The study of the single rotating disc showed that heat and mass transfer coefficients are enhanced considerably by decreasing the hub height.

The friction heat generated between two sliding bodies causes thermoelastic deformation, which alters the contact pressure distribution. This coupled thermo-mechanical process is referred to as the frictionally-excited thermoelastic instability (TEI) [14]. Other works have studied the transient brake analysis [15, 26, 30, 32]. Zagrodzki [30] analyzed sliding systems with frictional heating, which exhibit thermoelastic instability (TEI) in friction clutches and brakes when the sliding speed exceeds a critical value. Zhu et al. [32] established the theoretical model of a three-dimensional (3D) transient temperature field to predict the change of brake shoe's temperature field during hoist's emergency braking. Voldřich [26] postulated that the exceeding of the critical sliding velocity in brake discs causes formation of hot spots, non-uniform contact pressure distribution, vibration, and permanent damage of the disc. The analytical model of TEI development was published by Lee and Barber [15].

Many researchers investigated the heat generation phenomenon between contact surfaces in automotive clutches and brakes to predict the temperature distribution and especially the maximum temperature during the clutch engagement and braking to avoid failure before an estimated lifecycle. This process is very complex because of the following characteristics pressure, coefficient of friction and sliding speed. The researchers used different numerical techniques such as the finite element and finite difference methods to compute the sliding surface temperature [13, 27].

Abu Bakar and Ouyang [2] adjusted the surface profiles using measured data of the surface height and produced a more realistic model for brake pads. Direct contact interaction between the disc brake components is represented by a combination of node-to-surface and surface-to-surface contact elements [1].

Gao and Lin [7] stated that there was considerable evidence to show that the contact temperature is an integral factor reflecting the specific power friction influence of combined effect of load, speed, friction coefficient, and the thermo physical and durability properties of the materials of a frictional couple. Experiments showed that the friction coefficient in general decreased with increasing sliding speed and applied load, but increased with increasing disc temperature up to $300\,^{\circ}$C and then decreased above this temperature. The specific wear rate was found to increase with increasing sliding speed and disc temperature [22].

Nouby et al. [20] introduced a nontraditional evaluation tool to examine the effects of different materials that are used in fabricating disc brake components commonly used or special by manufactured for heavy-duty performance and racing cars. An extension of the FE models discussed earlier is a three-dimensional FE model of the disc brake corner that incorporates a wheel hub and steering knuckle developed and validated at both components and assembly levels to predict disc brake squeal. In addition, the real pad surface topography, negative fiction-velocity slope, and friction damping were considered to increase the prediction accuracy of the squeal. Brake squeal is a high frequency noise produced when driver decelerates and/or stops the vehicle moving at low speed. It is the noise caused by a self-excited vibration generated by the friction force variation between the friction material and the rotor. It is a phenomenon of dynamic instability that occurs at one or more natural frequencies of the brake system.

In work of Mosleh et al. [19], a brake dust generated in vehicle brakes causes discoloration of wheels and, more importantly, the emission of particles suspected of health hazard in the environment. Laboratory testing of brake pad materials against cast iron discs revealed that the majority of wear particles are submicrometer in size. Wear particles with a size of 350 nm had the highest percentage in the particle size distribution plots, regardless of the magnitude of the nominal contact pressure and the sliding speed. Due to their predominantly submicrometer size, a significant amount of brake dust particles may be inhalable in environmental and occupational exposure situations.

According to Altuzarra et al. [3], if the sliding speed is high, the resulting thermo-mechanical feedback is unstable, leading to the development of non-uniform contact pressure and local high temperature with important gradients called 'hot spots'. The formation of such localized hot spots is accompanied by high local stresses that can lead to material degradation and eventual failure [10]. Also, the hot spots can be a source of undesirable frictional vibrations, known in the automotive disc brake community as 'hot roughness' or 'hot judder' [29].

A ventilated disc is lighter than a solid one, and with additional convective heat transfer occurring on the surface of the vent hall. Thus, the ventilated disc can control its temperature rise and minimize the effects of thermal problems such as the variation of the pad friction coefficient, brake fade and vapor lock [5, 9]. The ventilated disc, however, may increase judder problems by inducing an uneven temperature field around the disc. Also, the thermal capacity of the ventilated disc is less than that of the solid disc, and the temperature of the ventilated disc can rise relatively faster than that of the solid disc during repetitive braking [11]. Therefore, thermal capacity and thermal deformation should be carefully considered when modifying the shape of the ventilated disc.

In this study, we will model of the thermal behaviour of a dry contact between the discs of brake pads during the braking phase; the strategy of calculation is based on the software ANSYS 11. This last is comprehensive mainly for the resolution of the complex physical problems. As a current study of the problem, ANSYS simulations with less assumption and less program restrictions have been performed for the thermo-mechanical case. A temperature distribution obtained by the transient thermal analysis is used in the calculations of stresses on the disc surface.

2. Numerical procedure

During the braking process, the temperature field changes input heat flux and heat exchange conditions. The input heat flux is mainly dependent on the friction coefficient and angular velocity of the brake disc, while the heat exchange is connected with the friction pair materials and external environmental factors.

Based on the first law of thermodynamics, the kinetic energy during braking is converted to thermal energy. The conversion of energy takes place because of the friction between the main components of the brake disc (pads and disc). Initially, the generated thermal energy is transferred by conduction to the components in contact and further by convection to surrounding. The initial heat flux q_0 into the rotor face is directly calculated using the following formula [25]:

$$q_0 = \frac{1-\phi}{2}\frac{mgv_0 z}{2A_d\varepsilon_p}, \tag{1}$$

where $z = a/g$ is the braking effectiveness, a is the deceleration of the vehicle [m/s^2], ϕ is the rate distribution of the braking forces between the front and rear axle, A_d is the disc surface swept by a brake pad [m^2], v_0 is the initial speed of the vehicle [m/s], ε_p is the factor load distributed on the disc surface, m is the mass of the vehicle [kg] and $g = 9.81$ is the acceleration of gravity [m/s^2].

The disc rotor vane passage and the sector chosen for the numerical analysis are shown in Fig. 1. The dimensions of the ventilated disc for the basic study are outer diameter of 262 mm, inner diameter of 66 mm and 36 vanes. The geometric model was created using the ANSYS Workbench 11.

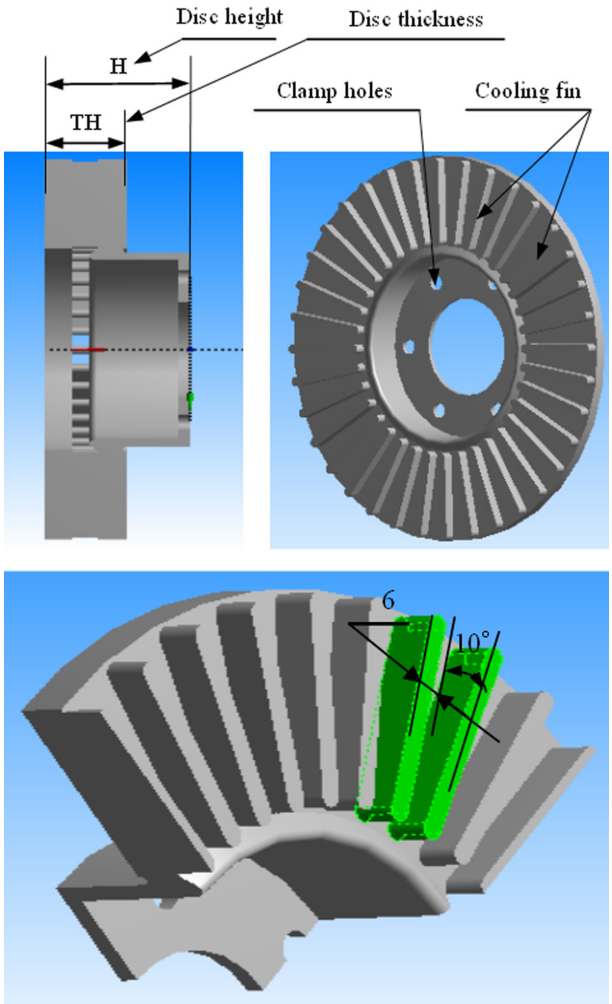


Fig. 1. Geometrical characteristics of the ventilated disc

The loading corresponds to the heat flux on the disc surface. The dimensions and the parameters used in the thermal calculation are summarised in Table 1.

The material of the brake disc is gray cast iron (FG 15) with high carbon content and good thermophysical characteristics. The thermoelastic characteristics of the material adopted in the rotor simulation are listed in Table 2. There are three reasons why rotors are made of the cast iron [4]:

- It is relatively hard and resists wear.
- It is cheaper than steel or aluminum.
- It absorbs and dissipates heat well and helps in cooling of the brakes.

Table 1. Input parameters

Input parameter	Values
Inner disc diameter [mm]	66
Outer disc diameter [mm]	262
Disc thickness (TH) [mm]	29
Disc height (H) [mm]	51
Vehicle mass m [kg]	1 385
Initial speed v_0 [m/s]	28
Deceleration a [m/s^2]	8
Effective rotor radius R_{rotor} [mm]	100.5
Rate distribution of the braking forces ϕ [%]	20
Tire radius R_{tire} [mm]	380
Factor of charge distribution of the disc ε_p	0.5
Surface disc swept by the pad A_d [mm^2]	35 993

Table 2. Material properties of the brake disc

Material properties	Disc
Thermal conductivity k [W/m °C]	57
Density ϱ [kg/m^3]	7 250
Specific heat C [J/Kg °C]	460
Poisson's ratio v	0.28
Thermal expansion $\alpha \cdot 10^{-6}$ [1/K]	10.85
Elastic modulus E [GPa]	138

It is very difficult to exactly model the brake disc, as there are still investigations going on to find out transient thermal behaviour of disc brakes during braking. In this case, to model a complex geometry, some simplifications are always necessary. These simplifications are made, keeping in mind the difficulties involved in the theoretical calculation and the importance of the parameters that are taken and those that are ignored. In modelling, we usually ignore the things of less importance and with little impact on the analysis. The assumptions are always made depending upon the details and accuracy required in the modelling.

By applying brakes on the car disc brake rotor, heat is generated by friction and the thermal flux has to be conducted and dispersed across the disc rotor cross section. The condition of braking is very severe and thus the thermal analysis has to be carried out. The thermal loading as well as structure is axisymmetric. Although, axisymmetric analysis can be performed, in this study we performed a 3D analysis, which is an exact representation of the thermal analysis. With the above load structural analysis, the thermal analysis is also carried out for analyzing the stability of the structure.

To simplify the analysis, several assumptions were also been made as follows [12]:

- All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux.
- The heat transfer involved in this analysis takes place only by conduction and convection. Heat transfer by radiation can be neglected as it amounts only to 5 % to 10 % [16].

- The disc material is considered as homogeneous and isotropic.
- The domain is considered as axisymmetric.
- Inertia and body force effects are negligible during the analysis.
- The disc is stress free before the brake application.
- In this analysis, the ambient temperature and initial temperature is set to $20\ ^{\circ}$C.
- All other possible disc brake loads are neglected.
- Only certain parts of disc brake rotor experience convection heat transfer such as the cooling vanes area, the outer ring diameter area and the disc brake surface.
- Uniform pressure distribution generated by the brake pad onto the disc brake surface is considered.

The thermal conductivity and specific heat are a function of temperature, as illustrated in Figs. 2 and 3.

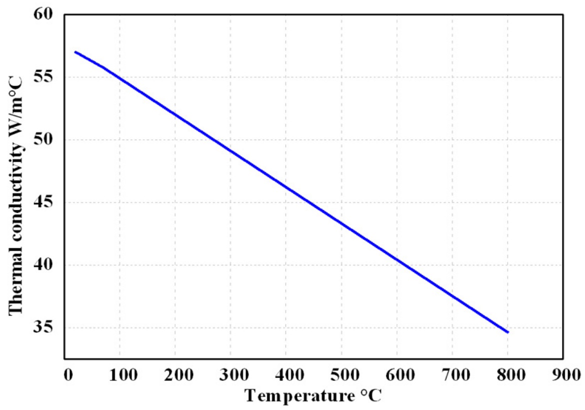


Fig. 2. Thermal conductivity versus temperature

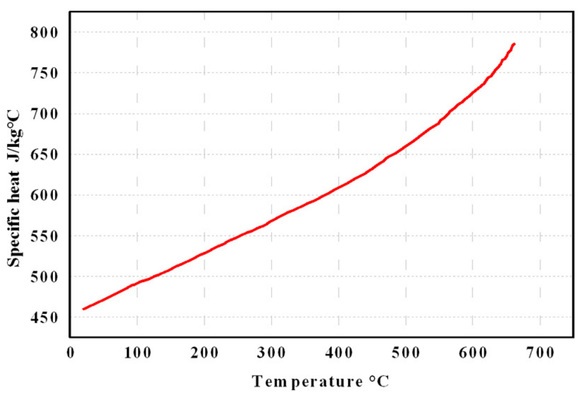


Fig. 3. Specific heat versus temperature

3. Finite element formulation for heat conduction

The unsteady heat conduction equation of each body for an axisymmetric problem described in the cylindrical coordinate system is given as follows:

$$\varrho c \frac{\partial T}{\partial t} = \frac{1}{r} \partial \partial r \left(r k_r \frac{\partial T}{\partial r} \right) + \frac{\partial}{\partial z} \left(k_z \frac{\partial T}{\partial z} \right) \tag{2}$$

with the boundary conditions and initial condition

$$T = T^* \quad \text{on} \quad \Gamma_0, \tag{3}$$

$$q_n = h(T - T_\infty) \quad \text{on} \quad \Gamma_1, \tag{4}$$

$$q_n = q_n^* \quad \text{on} \quad \Gamma_2, \tag{5}$$

$$T = T_0 \quad \text{at time } = 0, \tag{6}$$

where ϱ, c, k_r, and k_z are the density, specific heat and thermal conductivities in the r and z directions of the material, respectively. Also, T^* is the prescribed temperature, h is the heat transfer coefficient, q_n^* is the heat flux at each contact interface caused by friction, T_∞ is the ambient temperature, T_0 is the initial temperature and Γ_0, Γ_1 and Γ_2 are the boundaries on which temperature, convection and heat flux are imposed, respectively.

Using the Galerkin's approach, a finite element formulation of unsteady heat equation (2) can be written in the following matrix form as

$$C_T \dot{T} + KH_T T = R, \tag{7}$$

where C_T is the capacity matrix, KH_T is the conductivity matrix, T and R are the nodal temperature and heat source vector, respectively.

The most commonly used method for solving Eq. (7) is the direct integration method based on the assumption that the temperature T_t at time t and the temperature $T_{t+\Delta t}$ at time $t + \Delta T$ have the following relation:

$$T_{t+\Delta t} = T_t + [(1-\beta)\dot{T}_t + \beta \dot{T}_{t+\Delta t}]\Delta t. \tag{8}$$

Eq. (8) can be used to reduce the ordinary differential equation (7) to the following implicit algebraic equation:

$$(C_T + b_1 KH_T)T_{t+\Delta t} = (C_T - b_2 KH_T)T_t + b_2 R_t + b_1 R_{t+\Delta t}, \tag{9}$$

where the variable b_1 and b_2 are given by

$$b_1 = \beta \Delta t, \qquad b_2 = (1-\beta)\Delta t. \tag{10}$$

For different values of β, a well-known numerical integration scheme can be obtained [21]. In this study, $0.5 \leq \beta \leq 1.0$ was used, which results in an unconditionally stable scheme.

4. Model geometry and mesh

4.1. *Fluid field*

Considering symmetry in the disc, we took only a quarter of the geometry of the fluid field (Fig. 4) by using the mesh generation software ANSYS ICEM CFD.

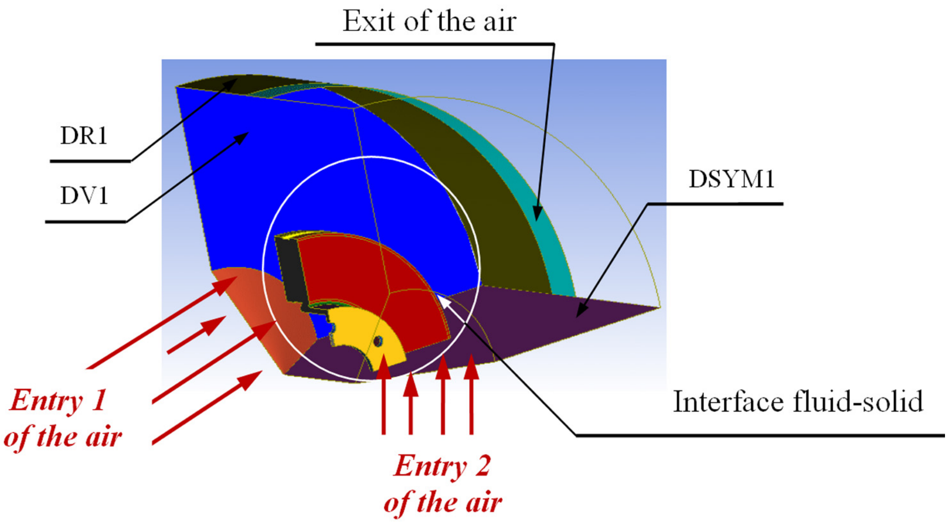


Fig. 4. Definition of surfaces of the fluid field

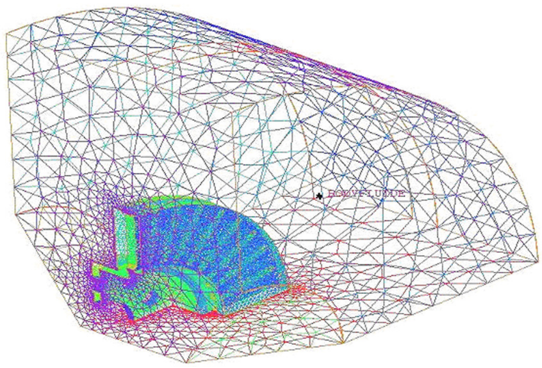


Fig. 5. Mesh of the fluid field

4.2. Mesh

This stage includes the mesh preparation in the fluid field. In our case, we used a linear tetrahedral element with 30 717 nodes and 179 798 elements (Fig. 5).

During the braking process, a part of the frictional heat escapes into the ambient air by convection and radiation. Consequently, the determination of the heat transfer coefficients is essential. Their exact calculation is, however, rather difficult, because these coefficients depend on the location and the construction of the braking system, the speed of the vehicle and consequently, on the air circulation or the air flow conditions (laminar, turbulent, or transition flow). Since the process of heat transfer by radiation is not too significant, we will determine only the convection coefficient h of the disc using the ANSYS CFX 11.0 code. The parameter will be utilized to determine the 3D temperature distribution of the disc.

Because of disc symmetry, we took only a quarter of the geometry in the case of the full and ventilated discs; one kept the tetrahedral form to generate the mesh of the discs (Figs. 6 and 7).

4.3. Modelling in ANSYS CFX

For the mesh generation of the CFD model, it is necessary to define various surfaces of the disc in the integrated computer aided engineering and manufacturing (ICEM) CFD as shown

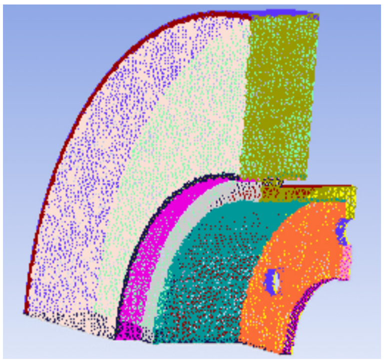


Fig. 6. Mesh of the full disc (Number of elements 272 392)

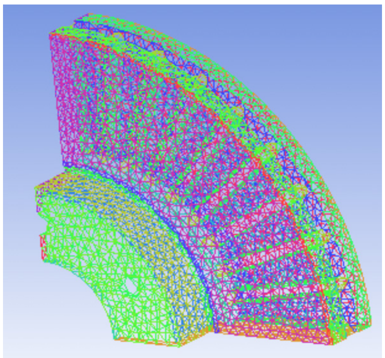


Fig. 7. Mesh of the ventilated disc (Number of elements 27 691)

in Fig. 8. In our case, we used a linear tetrahedral element with 30 717 nodes and 179 798 elements. In order to facilitate the calculation, a non-uniform mesh is used with refinement in locations with gradients.

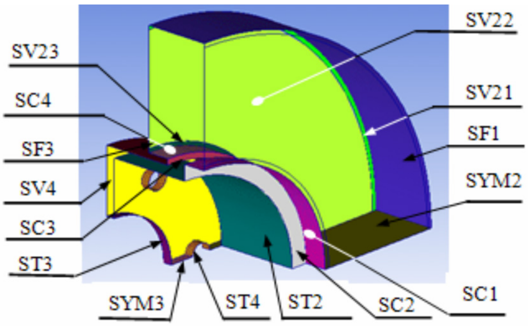


Fig. 8. Definition of surfaces of the full disc

Table 3. Boundary conditions

Boundary	Boundary condition	Parameters
Inlet	Pressure inlet	Atmospheric pressure and temperature
Outlet	Pressure outlet	Atmospheric pressure and temperature
Domain edges	Symmetry	Symmetry
Disc surface	Wall	800 K temperature, thermal properties of grey cast iron

a) *Physical model*

In this step, all physical characteristics of the fluid and the solid are declared. After the meshing, all the parameters of the different models are defined in order to start the analysis.

b) *Definition of the domains*

Initially, the elaborated models are validated and are activated in the option "Thermal Energy — the calculation of heat transfer — Heat Transfer".

Fluid domain — Speed entry: $V_{\text{ent non.st}} = V_{\text{ent}} - Va \cdot t$,
Disc domain — Entering flux: $\text{FLUX}_{\text{non.st}} = (CF)\,(V_{\text{ent non.st}})$,
$CF = 149\,893.838$,
$V_{\text{ent non.st}} = V_{\text{ent}} - Va \cdot t$.

$FLOW_{non.st}$: Non stationary flux entering.
$V_{ent\ non.st}$: Non stationary speed entering of the air.

c) *Definition of materials*

The physical properties of used materials are introduced into the computer code. In this study, we selected the cast iron material (FG 15).

d) *Definition of the boundary conditions*

The first step is to select the Inlet and Outlet faces where the heat flux enters and leaves. These options are found in the insertion menu "Boundary Conditions" in the CFX Pre. Interface.

In the next step, boundary conditions concerning the pads are defined by selecting the options "Wall" and "Symmetry", because there will be the possibility of adjusting a certain number of parameters in the boundary conditions such as the flux entering the disc.

e) *Application of the interfaces domains*

The areas of interfaces are commonly used to create the connection or linkage areas. Surfaces located between the interactions regions (air-disc) are reported as solid-fluid interface.

f) *Initial conditions*

Since the study is focused on the determination of temperature field in a disc brake of average class vehicle during the braking phase, we take the following temporal conditions:

- Braking time = 3.5 [s],
- Increment time = 0.01 [s],
- Initial time = 0 [s].

The airflow through and around the brake disc is analyzed using the ANSYS CFX software package. The ANSYS CFX solver automatically calculates the heat transfer coefficient at the wall boundary. Afterwards, the heat transfer coefficients considering convection are calculated and organized in such a way that they are used as a boundary condition in the thermal analysis. Averaged heat transfer coefficient had to be calculated for both discs using the software ANSYS CFX Post as indicated in Fig. 9.

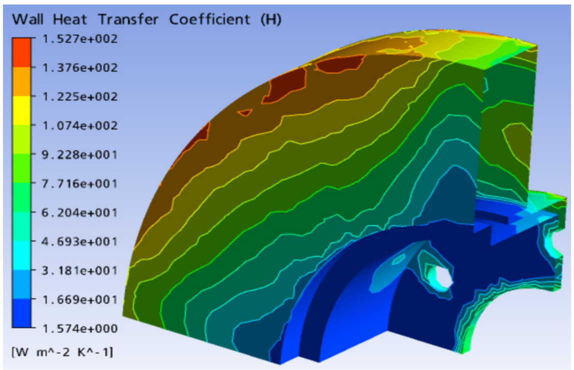


Fig. 9. Distribution of the heat transfer coefficient on a full disc in the steady-state case (FG 15)

g) *Calculation of the heat transfer coefficient h*

The heat transfer coefficient is a parameter related to the air velocity, the brake disc shape, and many other factors. For different air velocity, the heat transfer coefficient in different parts of the brake disc changes with time [31]. It depends on air flow in the region of brake rotor and on the vehicle speed, but it does not depend on the material. In our simulation, this coefficient is determined as an average value of the coefficient h "Wall heat Transfer Coefficient" and variable with time.

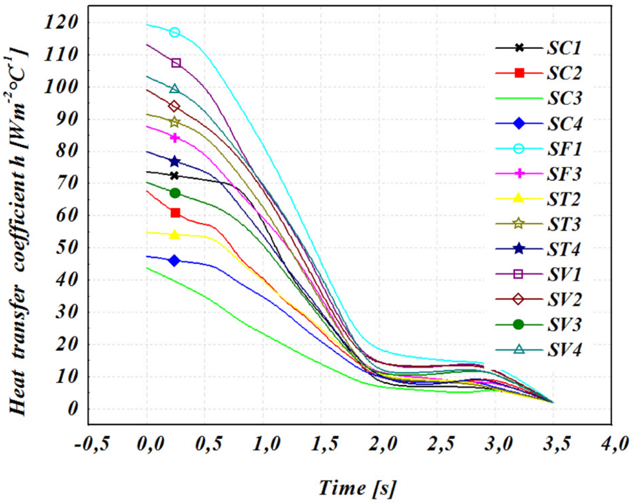


Fig. 10. Variation of the heat transfer coefficient h for various surfaces of the full disc in a transient case (FG 15)

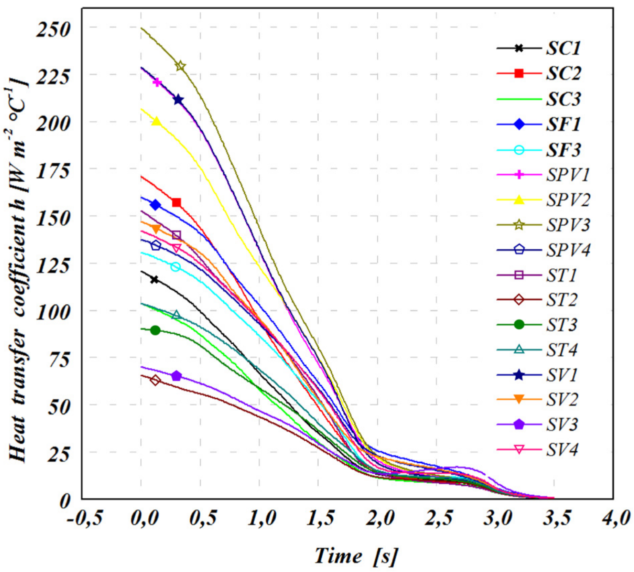


Fig. 11. Variation of the heat transfer coefficient h for various surfaces of the ventilated disc in a transient case (FG 15)

4.4. *Determination of disc temperature*

The modelling of disc temperature is carried out by simulating stop braking of a middle class car (braking of type 0). The characteristics of the vehicle and of the disc brake are listed in Table 1. The vehicle speed decreases linearly with time to the value 0 as shown in Fig. 12. The variation of the heat flux during the simulation time is depicted in Fig. 13.

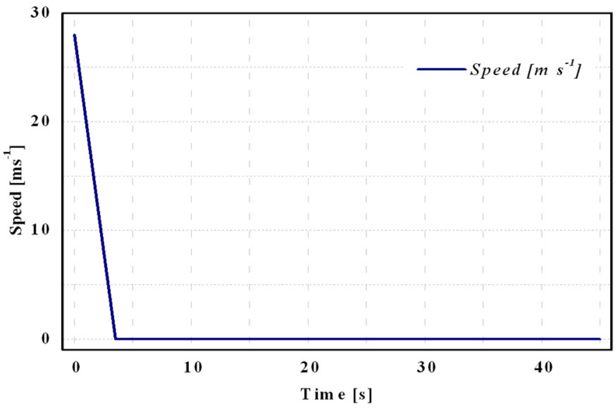


Fig. 12. Speed of the vehicle versus time (braking of type 0)

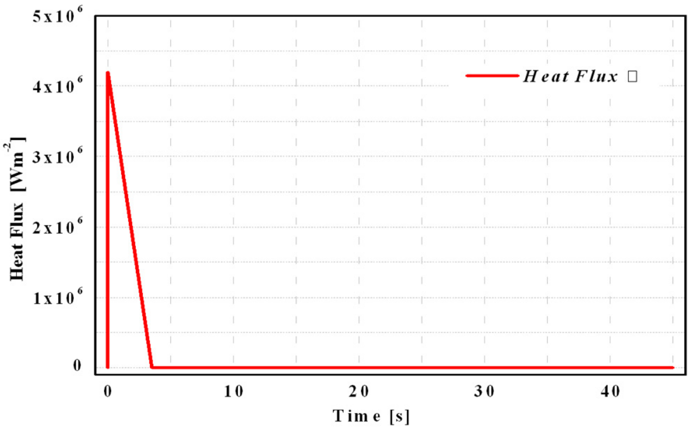


Fig. 13. Heat Flux versus time

4.5. *Finite element mesh for heat transfer analysis*

The mesh generation of the full and ventilated disc is carried out using ANSYS Multiphysics. The basic element used for the meshing is of tetrahedral shape (Figs. 14 and 15).

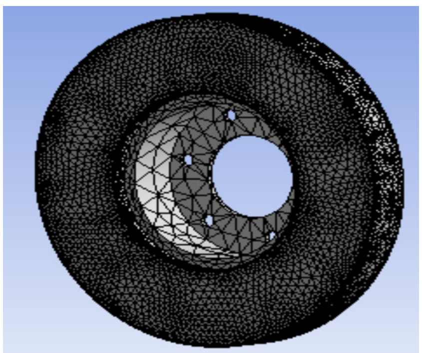


Fig. 14. Full type disc mesh model (total number of nodes 172 103 — total number of elements 114 421)

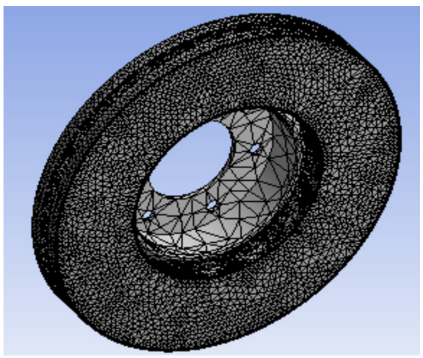


Fig. 15. Ventilated type disc mesh model (total number of nodes 154 679 — total number of elements 94 117)

4.6. Boundary conditions

The boundary conditions are introduced into within the module ANSYS Workbench [Multiphysics] by choosing the mode of the first simulation (permanent or transitory), and by defining the physical properties of the materials. These conditions constitute the initial conditions of our simulation. After having fixed these parameters, we introduce appropriate boundary conditions associated with each surface and specify the following computational parameters:

- Total time of simulation = 45 s.
- Increment of initial time = 0.25 s.
- Increment of minimal initial time = 0.125 s.
- Increment of maximal initial time = 0.5 s.
- Initial temperature of the disc = 20 °C.
- Materials: Grey Cast iron FG 5.
- Convection: We introduce the values of the heat transfer coefficient h obtained for each surface in the shape of a curve (Figs. 10 and 11).
- Heat flux: We apply the values obtained by means of the CFX code.

5. Results and discussions

The temperature in the disc brake is modeled with regard to the variation of a certain number of parameters such as the type of braking, the cooling mode of the disc and the choice of disc material. The brake discs are made of cast iron with high carbon content. The contact surface of the disc receives an entering heat flux calculated by Eq. (1).

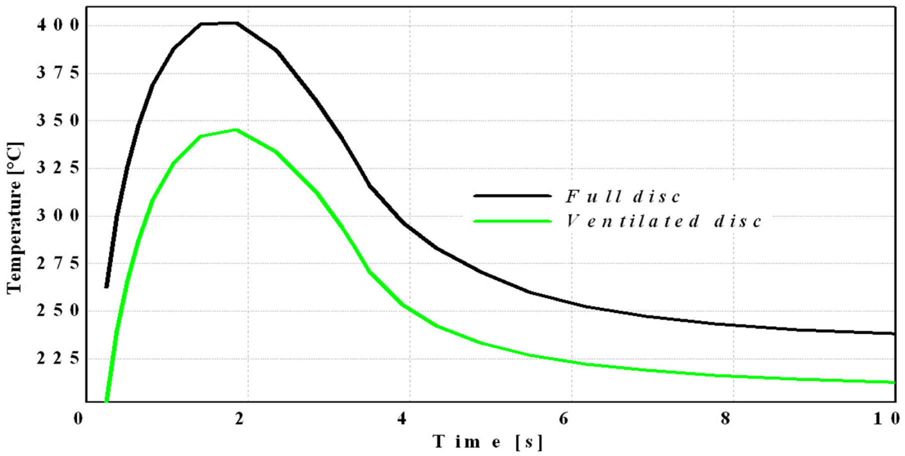
The model presents a 3D solid disc squeezed between two finite-width friction materials. The entire surface S of the disc has three different regions including S_1 and S_2. On S_1, the heat flux is specified with respect to the frictional heating between the pads and the disc, and S_2 is defined as a convection boundary. The rest of the surfaces without S_1 and S_2 is either temperature specified or assumed to be insulated: the inner and outer rim areas of the disc. Since an axisymmetric model is considered, all the nodes on the hub radius are fixed so that the nodal displacements in the hub become zero, i.e., in radial, axial and angular directions.

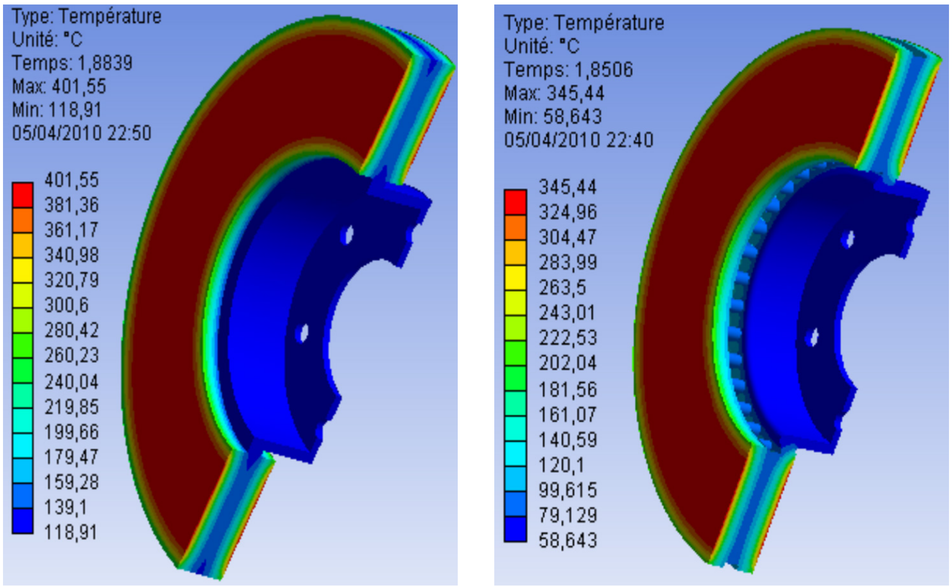
5.1. Influence of construction of the disc

Fig. 16 shows the variation of temperature with time during the total time simulation of braking for both the full and ventilated discs. The highest temperatures are reached on the contact surface of the disc pads. The initial sudden increase in temperature is due to the short duration of the braking phase and to the speed of the physical phenomenon. For the two types of discs, one disc has an immediate, fast temperature increase followed by a decrease in temperature after a certain time of braking.

On the basic of these observations, we can conclude that the geometric design of the disc is an essential factor in the improvement of the cooling process of the discs.

Figs. 17 and 18 show the temperature variation according to the radius and thickness, respectively. It is noted that there is an appreciable variation of temperature between the two types of discs. The influence of ventilation on the temperature field appears clearly at the end of the braking ($t = 3.5$ s).





(a) (b)

Fig. 16. Temperature distribution of a full (a) and ventilated disc (b) of cast iron (FG 15)

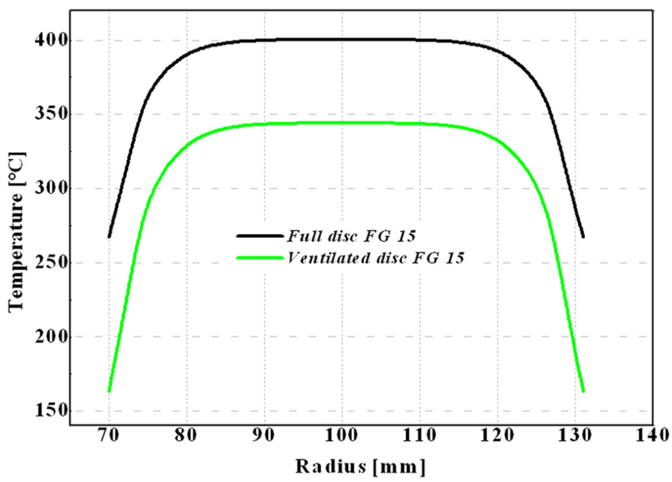
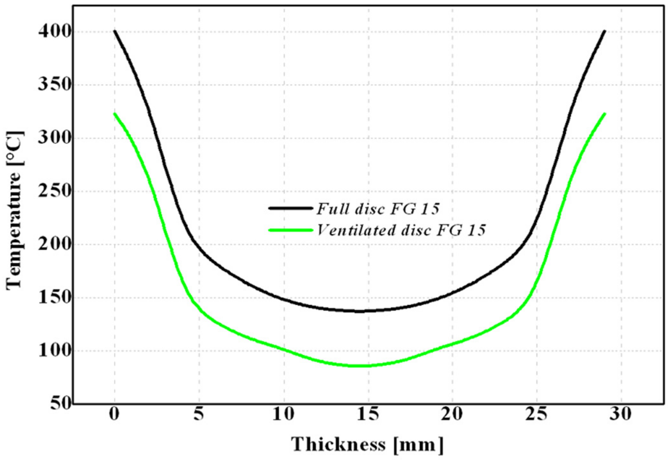


Fig. 17. Temperature variation through a radius for both designs of the same material (FG 15)



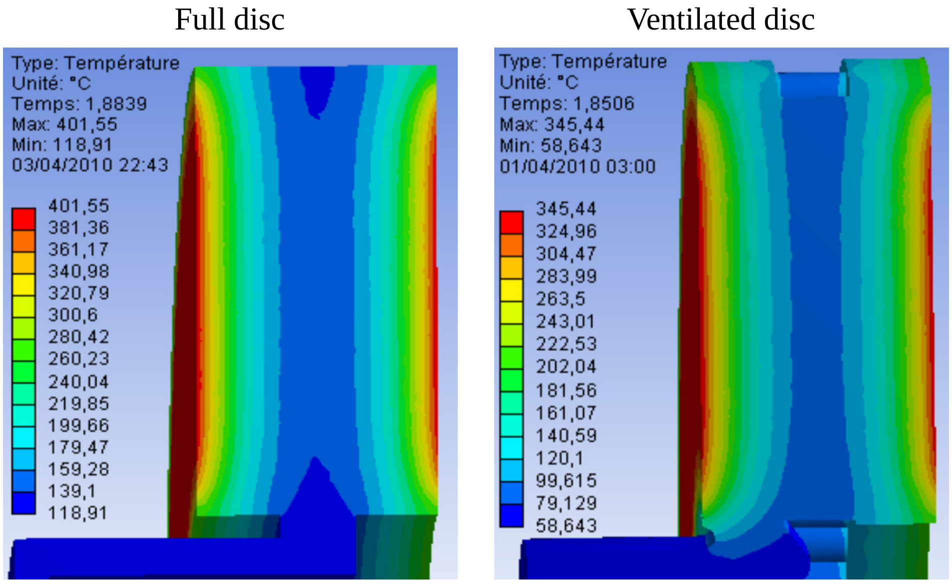
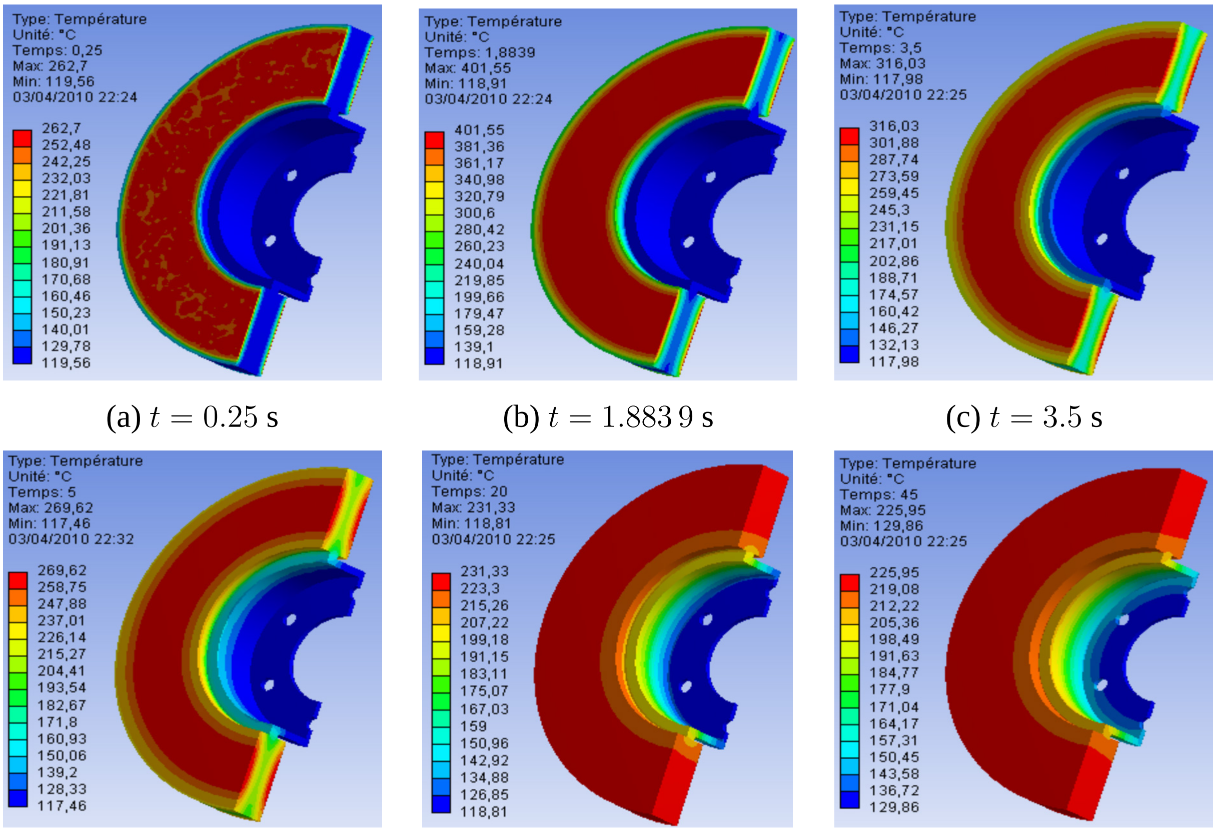


Fig. 18. Temperature variation through the thickness for both designs of the same material (FG 15)

5.2. *Comparison between the full and ventilated discs*

In this part, the maps of total and directional heat flux as well as the temperature distribution in the full and ventilated discs of cast iron FG 15 for each braking phase are presented. The temperature distribution of the disc at the beginning of the braking ($t = 0.25$ s) is inhomogenous (see Fig. 19). According to experimental tests, the braking often begins with the formation from hot circles relatively on the uniform surfaces of the disc in the circumferential direction, moving radially and then transforming into hot points (hot spot). The appearance of the phenomenon of the hot points is due to the non-uniform dissipation of the heat flux.

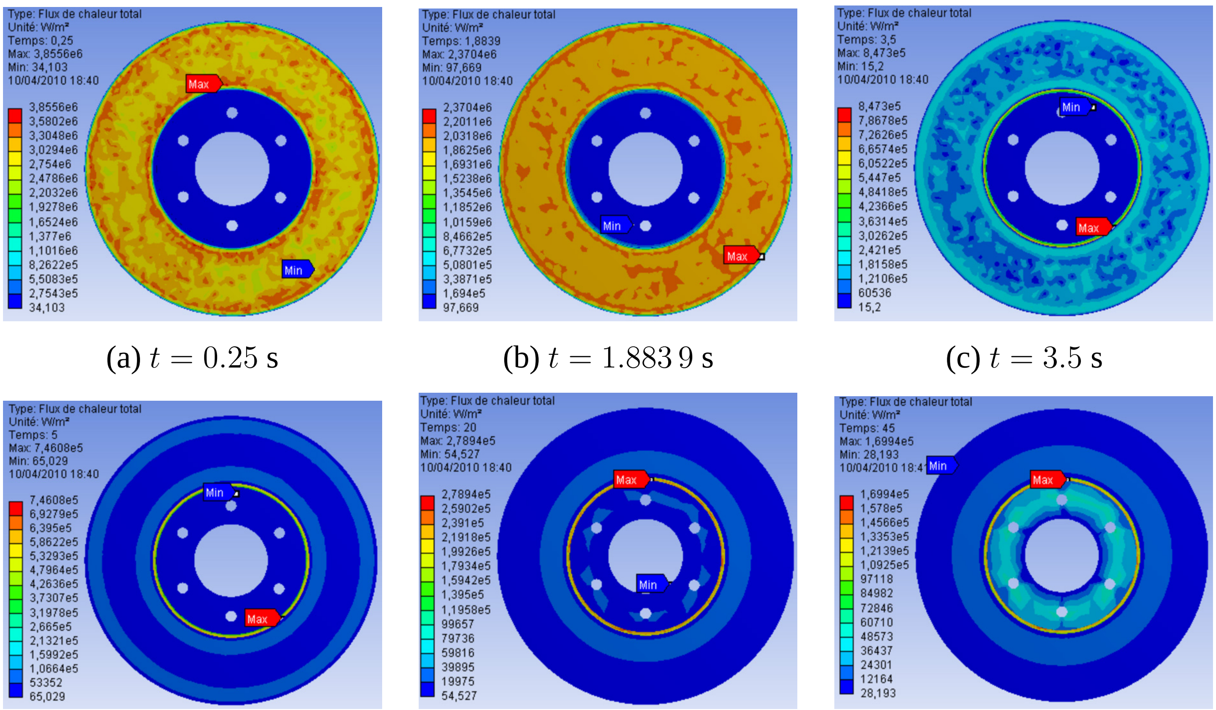
According to Figs. 20 and 23, the maximum value of the total heat flux is located on the level of the calorific throat at the end of braking ($t = 3.5$ s); this is explained by the increase in the gradients and the thermal concentrations in this zone. The calorific throat is manufactured so as to limit the heat flux coming from the friction tracks and moving towards the bowl of the disc brake, avoiding an excessive heating of the rim and the tire. During the heating, the disc is tightened to dilate in the hot zones from where creating of compressive stresses with plasticization. On the other hand, residual stresses of traction appear during the cooling. The rotating disc is, thus, subjected to constraints traction/compression.



(a) $t = 0.25$ s (b) $t = 1.883\,9$ s (c) $t = 3.5$ s

(d) $t = 5$ s (e) $t = 20$ s (f) $t = 45$ s

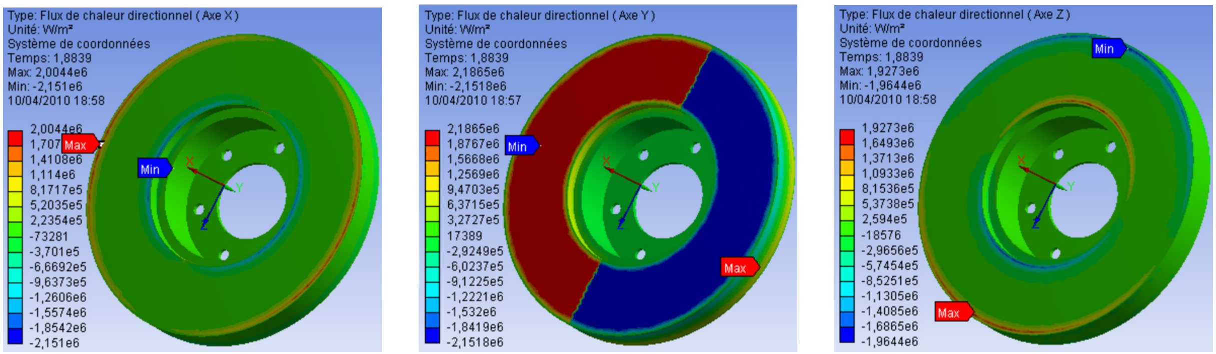
Fig. 19. Temperature distribution for the full disc of material FG 15



(a) $t = 0.25$ s (b) $t = 1.883\,9$ s (c) $t = 3.5$ s

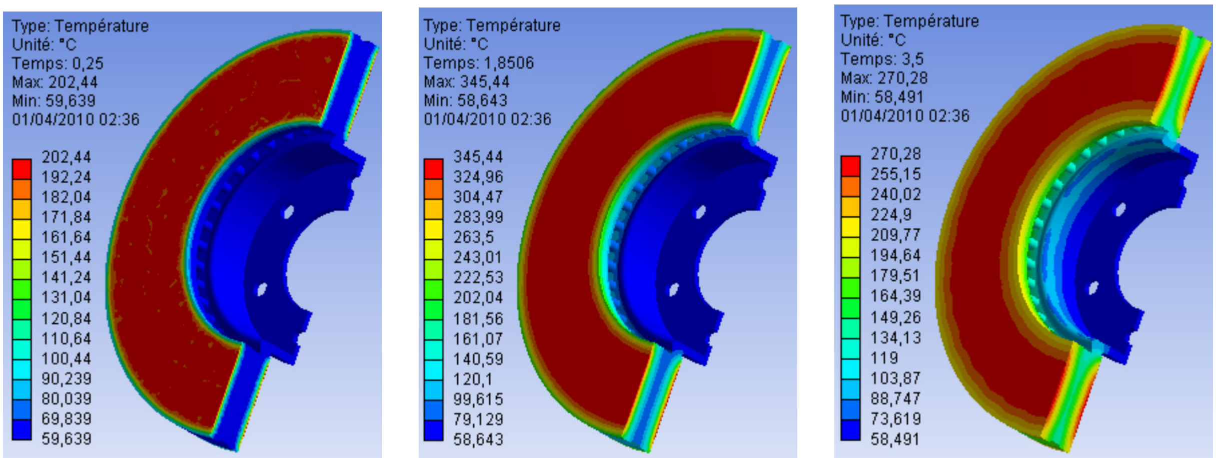
(d) $t = 5$ s (e) $t = 20$ s (f) $t = 45$ s

Fig. 20. Distribution of total heat flux for the full disc of material FG 15

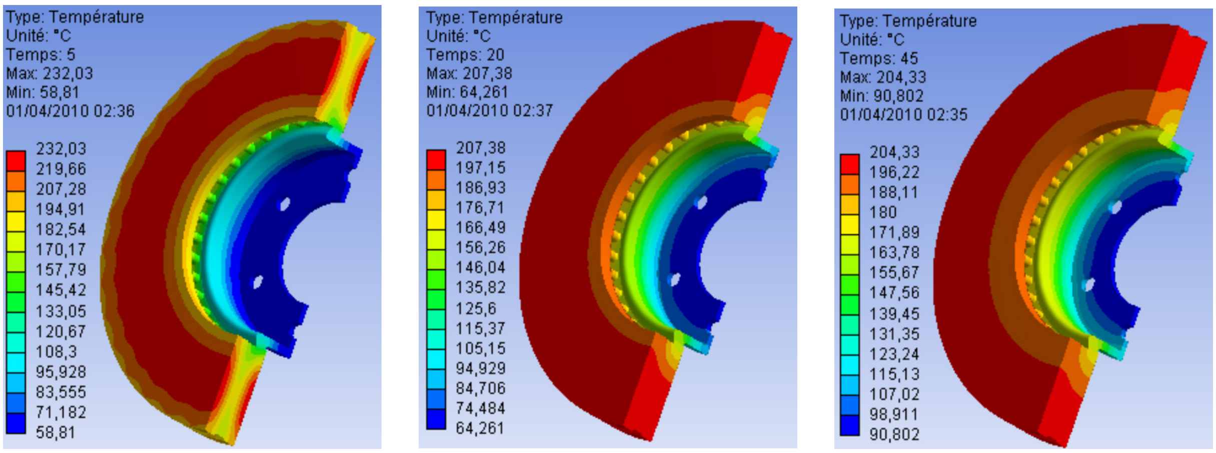


(a) Along the X axis (b) Along the Y axis (c) Along the Z axis

Fig. 21. Distribution of directional heat flux at the time $t = 1.883\,9$ s along the X, Y, Z axes for the full disc of material FG 15

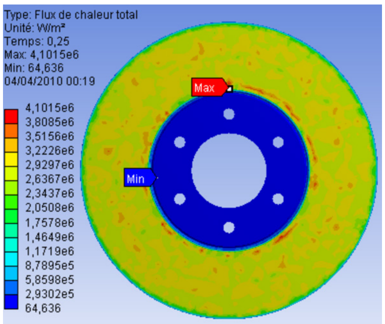


(a) $t = 0.25$ s (b) $t = 1.850\,6$ s (c) $t = 3.5$ s

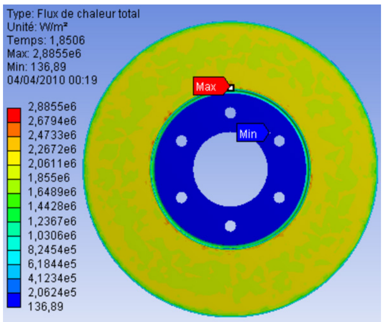


(d) $t = 5$ s (e) $t = 20$ s (f) $t = 45$ s

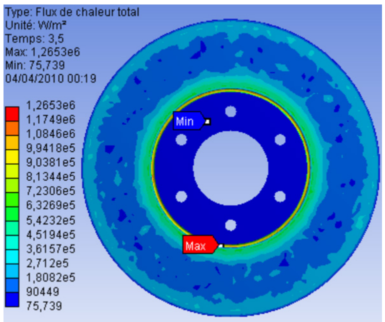
Fig. 22. Temperature distribution for the ventilated disc of material FG 15



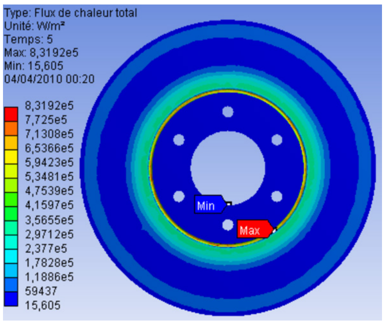
(a) $t = 0.25$ s



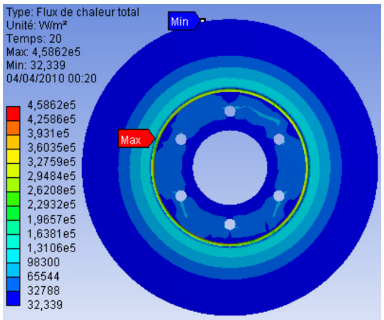
(b) $t = 1.850\,6$ s



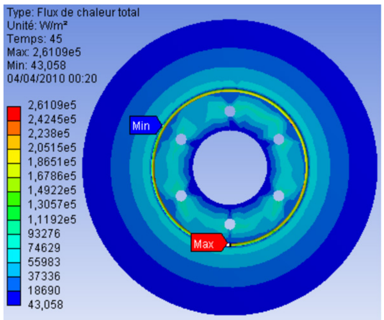
(c) $t = 3.5$ s



(d) $t = 5$ s

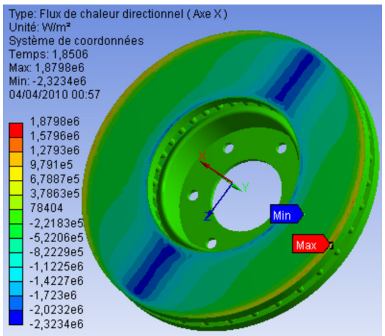


(e) $t = 20$ s

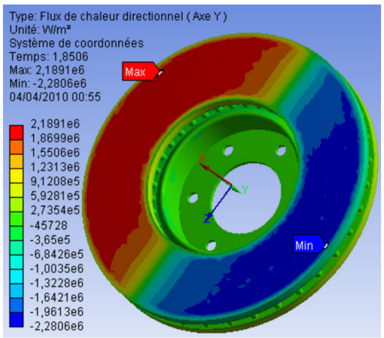


(f) $t = 45$ s

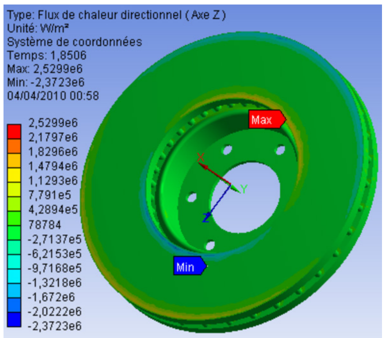
Fig. 23. Distribution of total heat flux for the ventilated disc of material FG 15



(a) Along the X axis



(b) Along the Y axis



(c) Along the Z axis

Fig. 24. Distribution of directional heat flux at the time $t = 1.850\,6$ s along the X, Y, Z axes for the ventilated disc of material FG 15

6. Conclusion

In this study, we presented results of the thermal behaviour of full and ventilated discs in a transient state. By means of the computer code ANSYS 11, we were able to study the thermal behaviour of a gray cast iron (FG 15).

In addition, the influence of disc ventilation on the thermal behaviour of the discs brake was analysed. The numerical simulation shows that radial ventilation plays a very significant role in cooling of the disc during the braking phase. Through the numerical simulation, we could note that the quality of the results concerning the temperature field is influenced by several parameters such as:

(i) Technological parameters illustrated by the design.

(ii) Numerical parameters represented by the number of elements and the step of time.

(iii) Physical parameters expressed by the types of materials.

Regarding the calculation results, we can say that they are satisfactorily in agreement with those commonly found in literature investigations. It would be interesting to solve the problem in thermomechanical disc brakes with an experimental study to validate the numerical results in order to demonstrate a good agreement between the model and reality.

Regarding the outlook, there are three possible improvements related to disc brakes that can be done to further understand the effects of thermomechanical contact between the disc and the pads. They are as follows:

(i) Experimental study to verify the accuracy of the numerical model developed.

(ii) Tribology and vibrations study of the contact disc — pads.

(iii) Study of dry contact sliding under the macroscopic aspect (macroscopic state of the disc and pad surfaces).

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