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Study of the thermal behaviour of dry contacts in the brake discs «application of software Ansys v11.0»

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1. Introduction

As the number of vehicles determining accident rate is increasing and higher and higher requirements are raised to their impact estimation and passengers' security, investigations on the above-ground vehicles safety elements and energy absorbing structures are very actual [1]. The thermal analysis is a primordial stage in the study of the brake systems, because the temperature determines thermomechanical behavior of the structure. In the braking phase, temperatures and thermal gradients are very high. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks [2, 3]. It is then important to determine with precision the temperature field of the brake disc.

In this study, we will present a numerical modeling in three dimensions to analyze the thermal behavior of the full and ventilated disc brake. The modeling will be carried out in transient state, simulating a stop braking of which the rotational speed of the disc and the flux generated by friction are functions of time. The finite element method (FEM) has become the prevalent technique used as an effective tool for analyzing all kinds of physical phenomena in structural, solid and fluid mechanics [4].

The thermal calculation based on the finite element method will be carried out using code ANSYS 11.This last is elaborated out for the resolution of complex physical systems.

2. Heat flux entering the disc

In a braking system, the mechanical energy is transformed into a calorific energy. This energy is characterized by a total heating of the disc and pads during the braking phase. The energy dissipated in the form of heat can generate rises in temperature ranging from 300 to 800°C. Generally, the thermal conductivity of material of the brake pads is smaller than of the disc $\left(k_p < k_d\right)$. We consider that the heat quantity produced will be completely absorbed by the brake disc. The heat flux evacuated of this surface is equal to the power friction. The initial heat flux q_0 entering the disc is calculated by the following formula [5]

$$q_0 = \frac{1-\varphi}{2}\frac{m\,g\,v_0 z}{2A_d\,\varepsilon_p} \tag{1}$$

where $z = a/g$ is braking effectiveness, a is deceleration of the vehicle, ms^{-2}; φ is rate distribution of the braking forces between the front and rear axle; A_d is disc surface swept by a brake pad, m^2; v_0 is initial speed of the vehicle, ms^{-1}; ε_p is factor load distribution of the on the surface of the disc; m is mass of the vehicle, kg.

Fig. 1 shows the ventilated disc – pads and the applied forces.

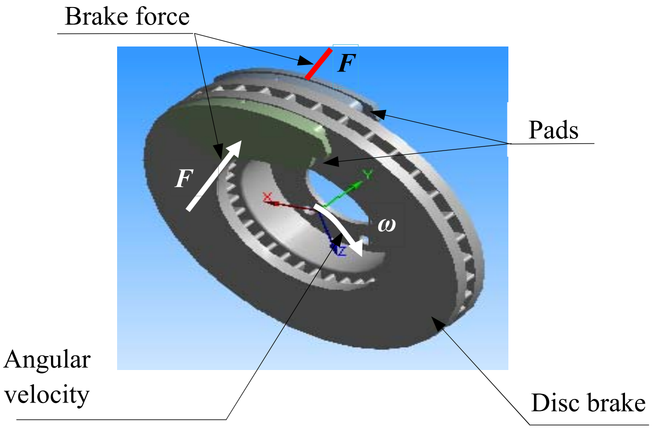

Fig. 1 Disc-pads assembly with forces applied to the disc

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

Table 1
Geometrical dimensions and application parameters of automotive braking

Inner disc diameter, mm	66
Outer disc diameter, mm	262
Disc thickness, mm	29
Disc height, mm	51
Vehicle mass m, kg	1385
Initial speed v_0, km/h	28
Deceleration a, m/s^2	8
Effective rotor radius R_{rotor}, mm	100.5
Rate distribution of the braking forces Φ, %	20
Factor of charge distribution on the disc ε_p	0.5
Surface disc swept by the pad A_d, mm^2	35993

The disc material is gray cast iron with high carbon content FG, with good thermophysical characteristics those are given in Table 2. The thermal conductivity and specific heat are a function of temperature [6], Figs. 2 and 3.

Table 2

Thermophysical properties of the disc

Young modulus E, MPa	138000
Poisson coefficient ν	0.28
Density ρ, kg/m^3	7250
Thermal expansion α, 1/°C	$1.085.10^{-5}$
Tensile strength, MPa	300
Compressive strength, MPa	820

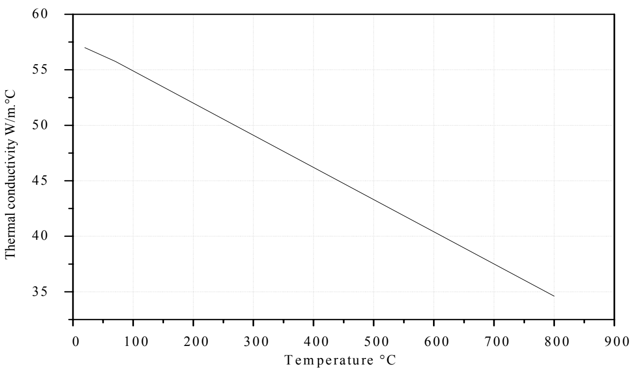

Fig. 2 The thermal conductivity as a function of temperature

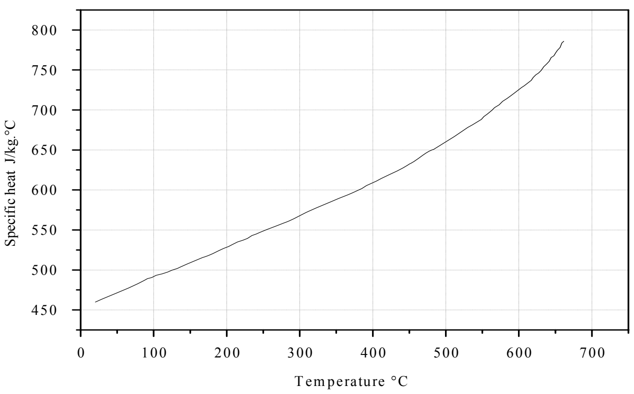

Fig. 3 Specific heat versus temperature

3. Numerical modeling of the thermal problem

3.1. Form differential

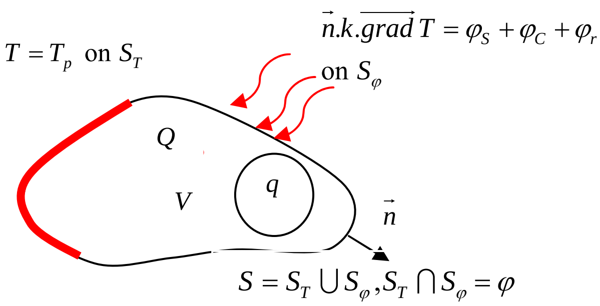

Fig. 4 Thermal loads applied to a continuous medium

The system shown in Fig. 4 is subjected to the following thermal loads [7]:

- a specific heat source Q, W;
- a voluminal heat source q, W/m^3;
- temperature imposed (or prescribed) T_p on a surface S_T;
- flux density φ_c imposed on a S_φ surface, W/m^2;
- heat transfer by convection φ_c on a surface S_φ;
- heat transfer by radiation φ_r on a surface S_φ

The solution of a thermal problem is to find the temperature field $T(x,y,z,t)$ at any point of the solid so that [8]

$$\rho C_p \dot{T} - div\left(-k.\overrightarrow{grad}\,T\right) - q = 0 \tag{2}$$

- with the boundary conditions (Fig. 4)

$$\left\{\begin{array}{l} T = T_p \text{ on } S_p \\ \vec{n}.(-k.\overrightarrow{grad}\;T) = \varphi_S + \underbrace{h(T_f - T)}_{convection} + \underbrace{\varepsilon\,\sigma\left(T_\infty^4 - T^4\right)}_{radiation} \text{on } S_\varphi \\ S = S_T \cup S_\varphi, S_T \cap S_\varphi = \varphi \end{array}\right\} \tag{3}$$

- the initial condition at time $t = t_0$:

$$T(x,y,z,t) = T_0(x,y,z) \tag{4}$$

where ρ is density of material, kg/m^3; C_p is mass heat capacity, J/(kg K), $\vec{n}$ is unit normal with s directed towards the outside of v.

This system of equations is written in weak formulation as follows [9 - 11]

$$\begin{aligned} &\int T^* \rho C_p dV + \int \overrightarrow{grad}\,T^*\left(k.\overrightarrow{grad}\;T\right)dV - \\ &-\int T^*\left(\varphi_S + h\left(T_f - T\right) + \varepsilon\,\sigma\left(T_\infty^4 - T^4\right)\right) - \\ &-\int T^* q dV = 0 \quad \forall T^* \end{aligned} \tag{5}$$

T^* is weight function (or function test).

With the initial and the following boundary conditions

$$T(x,y,z,t) = T_0(x,y,z) \text{ and } T = T_p \text{ on} \tag{6}$$

The temperatures field $T(x,y,z,t)$ has for expression on the whole domain V

$$T(x,y,z,t) = \left[N_1(x,y,z),\ldots,N_i(x,y,z),\ldots,N_n(x,y,z)\right]\left\{\begin{array}{c} T_1(t) \\ \vdots \\ T_i(t) \\ \vdots \\ T_n(t) \end{array}\right\} =$$

$$= [N]\{T\} \tag{7}$$

where $\left[N(x,y,z)\right]$ is the matrix of interpolation; $\left\{T(t)\right\}$ is vector of the nodal temperatures.

By carrying the following relations in the Eq. (5)

$$T = [N]\{T\} \tag{8}$$

$$T^* = [N]\left\{\overset{*}{T}\right\} \tag{9}$$

$$\{gradT\} = [B]\{T\} avec [B] = \left[\{B_1\},\ldots,\{B_i\},\ldots,\{B_n\}\right] \tag{10}$$

$$T^* = [N]\left\{\overset{*}{T}\right\} = \left\{T^*\right\}^T [N]^T, \left\{gradT^*\right\} = [B]\left\{T^*\right\},$$

$$\left\{grad\ T^*\right\}^T = \left\{T^*\right\}^T [B]^T \tag{11}$$

We obtains

$$\left\{\overset{*}{T}\right\}^T \left([C]\left\{\dot{T}\right\} + [K]\{T\} - \{F\} = 0\right) \tag{12}$$

where

$$[C] = \int_V \rho C_p [N]^T [N] dV \tag{13}$$

$$[K] = \int_V [N]^T [\lambda][N] dV + \int_{S_\varphi} h [N]^T [N] dS \tag{14}$$

$$\{F\} = \int_V [N]^T dV + \int_{S_\varphi} [N]^T \left(\varphi_S + hT_f + \varepsilon\sigma\left(T_\infty^4 - T^4\right)\right) dS \tag{15}$$

where $[C]$ is thermal capacity matrix (J/K); $[K]$ is thermal conductivity matrix (W/K), $\{F\}$ is nodal flux vector (W); $\{T\}$ is nodal temperatures vector (K).

3.2. Initial conditions

We suppose that the initial temperature of the disc is constant.

$$T(x,y,z,t) = 60\,°\text{C at time } t = 0 \tag{16}$$

3.3. Boundary conditions

This is a transient thermal problem with two boundary conditions:

- a heat flux entering the disc localized in the contact zone disc-pad in both sides;
- a heat transfer by convection on all the free surfaces of the disc of which the exchange coefficient *h* depends on time because rotational speed of the disc varies with time.

4. Presentation of the computing code ANSYS

ANSYS is software program, created in 1970 in the United States; its modules are software programs that implement the finite element method to solve models previously discretized. The modules used for this study are:

- ANSYS Workbench: This platform offers a different approach in the construction of model using the original computer code ANSYS [12]. It is particularly adapted to handling cases with complex geometry and to the unconfirmed users;
- ANSYS ICEM CFD: It is mesh generation software for applications in fluid mechanics and mechanical structures;
- ANSYS CFX: This software is designed to perform simulations in fluid mechanics;
- ANSYS Metaphysics: This product contains all modules of ANSYS simulation code.

Fig. 5 shows the stages of simulation with ANSYS CFX in Workbench.

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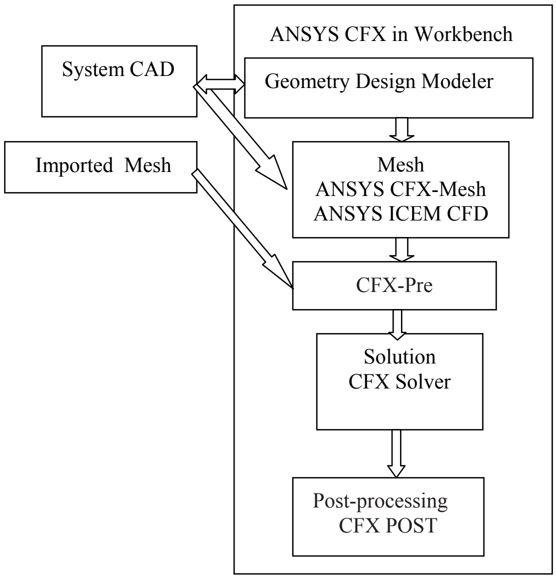


Fig. 5 Simulation steps with CFX [12]

5. Determination of the coefficient of exchange by convection (*h*)

5.1. Introduction

The thermal analysis of the braking system requires a precise determination of the quantity of heat friction produced and as well as the distribution of this energy between the disc and the brake lining. When a vehicle is braked, a part of the frictional heat escapes in the air ambient 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 travel and consequently of the air circulation. Since the process of heat transfer by radiation is not too important, we will determine using code ANSYS CFX only the convection coefficient *h* of the disc**.** This parameter will be exploited to determine the three-dimensional distribution of the temperature of the disc.

5.2. Modeling in ANSYS CFX

The first stage is to create the model CFD which contains the fields to be studied in Ansys Workbench. In our case, we took only one quarter of the disc, then we defined the field of the air surrounding this disc. ANSYS ICEM CFD will prepare various surfaces for the two fields in order to facilitate the mesh on which that one will export the results towards CFX using the command "Output to CFX".After obtaining the model on CFX Pre and specified the boundary conditions, we must define these physical values come into play on CFX to start calculation.

The disc is related to four adiabatic surfaces and two surfaces of symmetry in the fluid domain whose ambient temperature of the air is taken equal at 20°C [13]. An unsteady-state analysis is necessary.

Fig. 6 shows the elaborate model CFD which will be used in ANSYS CFX Pre.

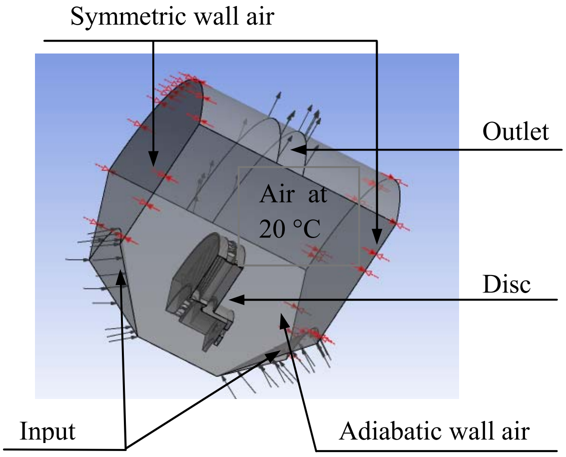


Fig. 6 Brake disc CFD model

a) Physical model.

In this step, one declares all of the physical characteristics of the fluid and the solid. After the meshing, are defined all the parameters of the different models to be able to start the analysis.

b) Definition of the domains.

Initially, one valide the elaborated models and one activate in the option "Thermal Energy " the calculation of heat transfer "Heat Transfer".

Fluid domain: Speed entry: $V_{ent\ non.st} = V_{ent} - Va{\cdot}t$

Disc domain: Entering flux: $FLUX_{non.st} = (CF)\ (V_{ent\ non.st})$,

$CF = 149893.838$

$V_{ent\ non.st} = V_{ent} - Va{\cdot}t$

where $FLUX_{non.st}$ is nonstationary flux entering: $V_{ent\ non.st}$ is nonstationary speed entering of the air.

c) Definition of materials.

We introduce into the library the physical properties of used materials. In this study we selected three cast iron materials (FG 25 Al, FG 20 and FG 15).

d) Definition of the boundary conditions.

The first step is to select the Inlet and Outlet faces of the heat flux. These options are found in the insertion menu “Boundary Conditions” in the CFX Pre.

The boundary conditions concerning the pads will be also defined. One selects the options “Wall” and "Symmetry ", because there will be the possibility of adjusting a certain number of parameters in the boundary conditions such as 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-disk) are reported as solid-fluid interface.

f) Temporary condition.

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

- braking time = 3.5 s;
- increment time = 0.01 s;
- Initial time = 0 s.

Before starting the calculation and the analysis with ANSYS CFX PRE, it is ensured that the model does not contain any error.

g) Launch of the calculation.

After verification of the model and boundary conditions, we run the calculation by opening the menu "File" and clicking on "Write solver file".

h) Results of the calculation of the coefficient *h*.

Figs. 7 and 8 show the variation of the heat transfer coefficient *h* of different surfaces respectively for a full and ventilated disc in cast iron (FG 15) in transient state. We found that after a short time all the curves of *h* are decreasing with time.

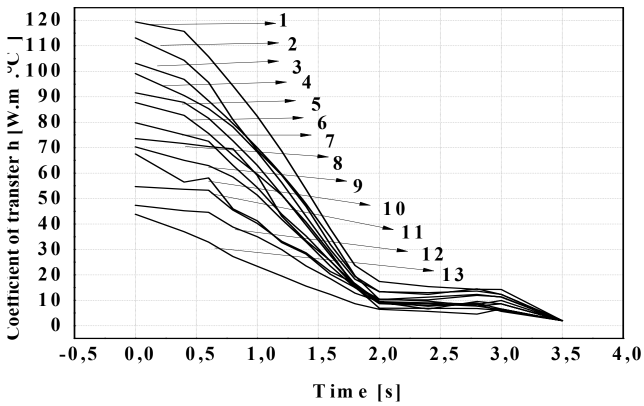


Fig. 7 Variation of heat transfer coefficient *h* of various surfaces for a full disc in the nonstationary case (FG 15)

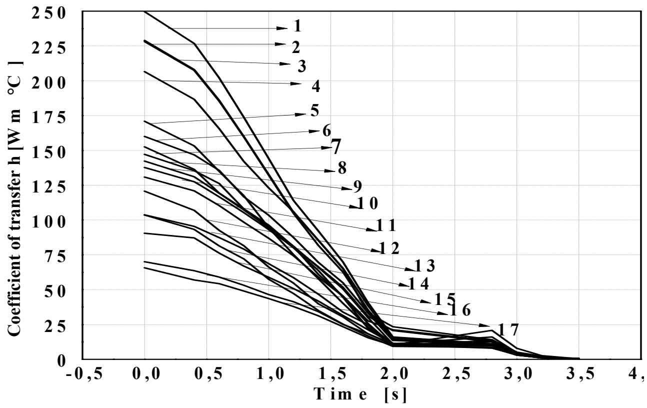


Fig. 8 Variation of heat transfer coefficient *h* of various surfaces for a ventilated disc in transient case (FG 15)

6. Determination of the disc temperature

The modeling of the disc temperature is carried out by simulating a 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 until the value 0 as shown in

Fig. 9. The variation of the heat flux during the simulation time is represented on the Fig. 10.

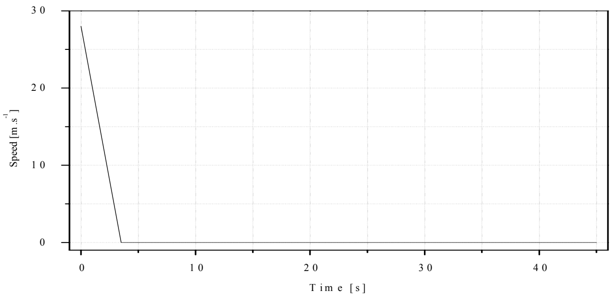


Fig. 9 Speed of braking versus time (braking of type 0)

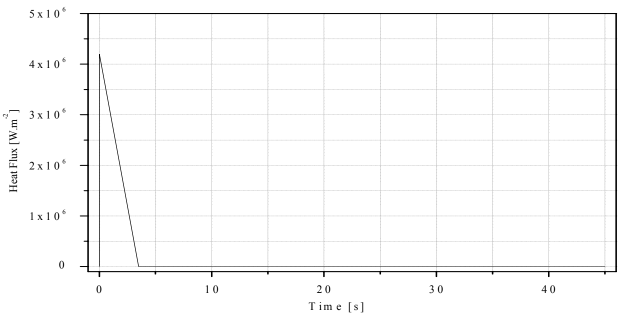


Fig. 10 Heat flux versus time

6.1. Meshing of the disc

The elements used for the meshing of the full and ventilated disc are tetrahedral three-dimensional elements with 10 nodes (isoparametric) (Figs. 11 and 12). In this simulation, the meshing was refined in the contact zone (disc-pad). This is important because in this zone the temperature varies significantly.

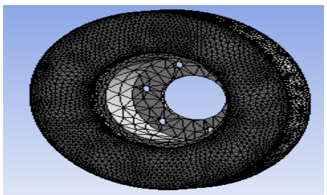


Fig. 11 Meshing of a full disc in ANSYS Multiphysics (172103 nodes – 114421 elements)

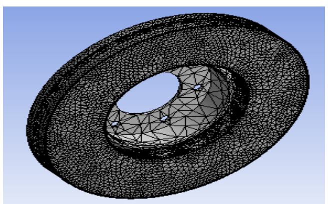


Fig. 12 Meshing of a ventilated disc in ANSYS Multiphysics (154679 nodes- 94117 elements)

6.2. Loading and boundary conditions

The thermal loading is characterized by the heat flux entering the disc through the real contact area (two sides of the disc).The initial and boundary conditions are introduced into module ANSYS Workbench. The thermal calculation will be carried out by choosing the transient state and by introducing physical properties of the materials. The selected data for the numerical application are summarized as follows:

- 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 = 60°C;
- materials: three types of cast iron (FG 25 AL, FG 20, FG 15).

7. Results and discussions

The modeling of temperature in the disc brake will be carried out by taking account of 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 the relation (1).

7.1. Influence of construction of the disc

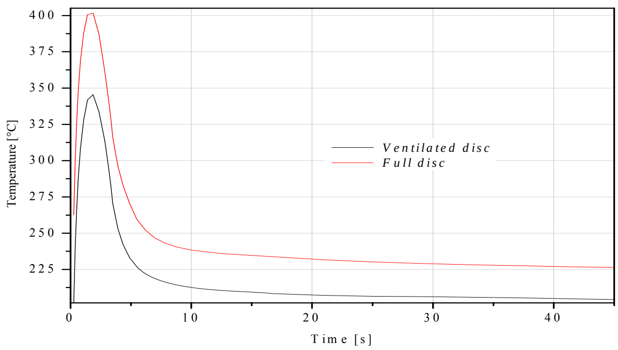
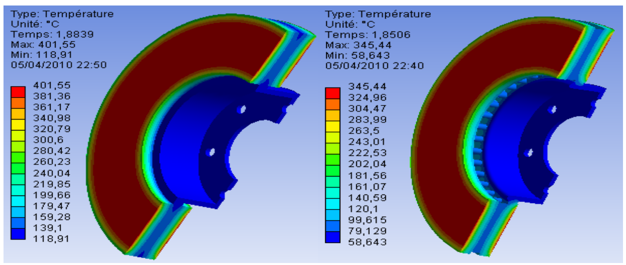


Fig. 13 Temperature variation of a full and ventilated disc (FG 15) versus time



a b

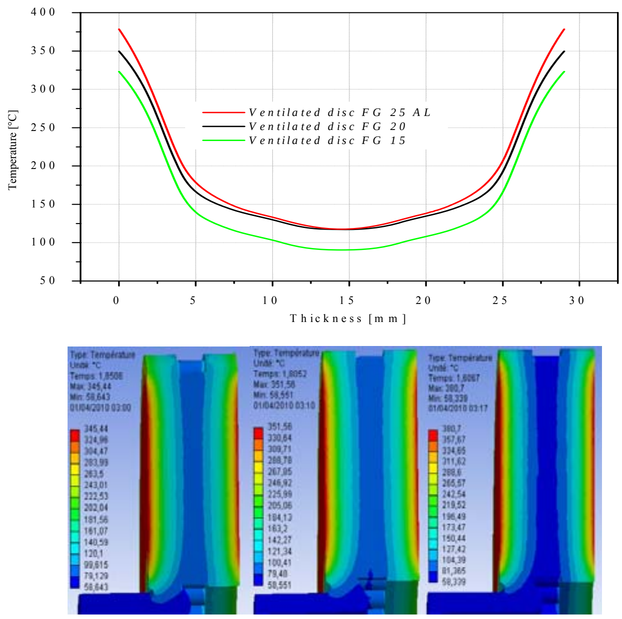
Fig. 14 Temperature distribution of a full (a) and ventilated disc (b) of cast iron (FG 15)

Fig. 13 shows the variation of the temperature versus time during the total time simulation of braking for a full disc and a ventilated disc. The highest temperatures are reached at the contact surface disc-pads. The strong rise 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 notices that starting from the first step of time one has a fast rise of the temperature of the disc followed by a fall of temperature after a certain time of braking.

We quickly notice that for a ventilated disc out of cast iron FG15, the temperature increases until T_{max} = = 345.44°C at the moment t = 1.85 s, then it decreases rapidly in the course of time. The variation in temperature between a full and ventilated disc having same material is about 60°C at the moment t = 1.8839 s. We can conclude that the geometric design of the disc is an essential factor in the improvement of the cooling process of the discs (Fig. 14).

7.2. Infuence of material of the disc

Fig. 15, a shows for each type of the selected cast iron the temperature variation as a function of thickness at the moment when the temperature is maximum. The allure of the three curves is similar. We also note that the temperature decreases in the direction of median plane of the disc to reach its minimal value. In Fig. 15, b we see that there is inside the disc symmetry of colors. The part far away from the surface of blue contact color is not requested too much thermically. More the thermal conductivity of the material is low, more its temperature is high. The FG 15 is differentiated from the two other cast iron by



FG 15 FG 20 FG25 AL

Fig. 15 Temperature variation through a disc thickness for three types of cast irons (FG 25 AL, FG 20 and FG 15)

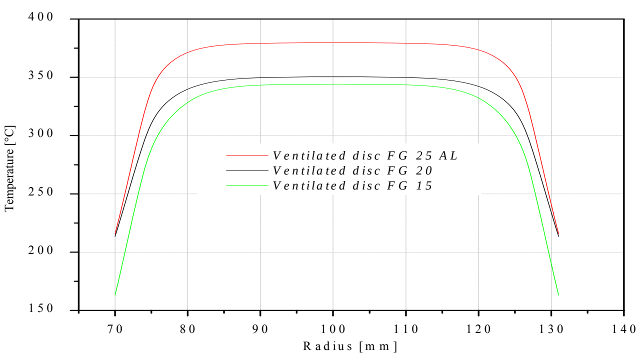


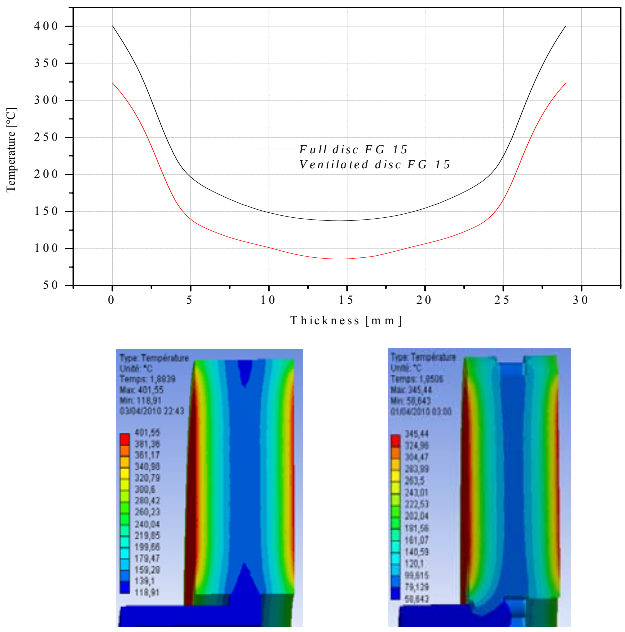
Fig. 16 Temperature variation through a radius for three types of cast irons (FG 25 AL, FG 20 and FG 15)

smaller temperatures. On Fig. 16, the temperature variation versus radius for three materials (FG 25 Al, FG 20, FG 15) is presented. The shape of the temperature curves are the same one. The maximal temperature is in area of the mean disc radius. According to Figs. 15 and 16 the cast iron FG 15 has the best thermal behavior.

Figs. 17 and 18 respectively show the temperature variation according to the thickness and radius. It is noted that there is an appreciable variation of temperature between the two types of full and ventilated disc.

The influence of ventilation on the temperature field appears clearly at the end of the braking (t = 3.5 s).

Among the parameters having an influence on the thermal behavior of the discs brake there is the braking mode which depends on the driver and the circulation conditions. Certain modes of braking can involve the destruction of the disc and consequently to cause serious accidents of circulation. A braking mode is represented in the form of braking cycles, which describe the variation of vehicle speed versus time $v = f(t)$.



Full disc Ventilated disc

Fig. 17 Temperature variation through the thickness for both designs with same material (FG15)

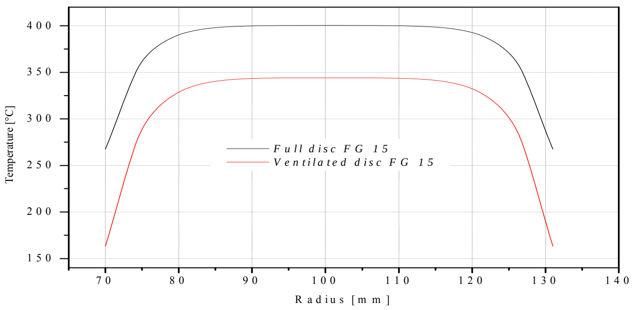


Fig. 18 Temperature variation through a radius or both designs with the same material (FG15)

These cycles may consist of a series of emergency brakings or cycles comprising of the braking phases followed by a downtime.

7.3. Influence of braking mode

The disc brake and the wheel are dimensioned according to the performance and economic requirements of the vehicle. They must support mechanical and thermal loads increasingly greater at mean velocities in permanent progression.

7.3.1. Repeated braking

During vehicle operating, the braking system is subjected to repeated actions of the driver. In this study, we considered two types of braking of which the total simulation time is estimated to be equal to 135 s.

Fig. 19 shows a driving cycle of fourteen successive brakings, in the form of sawtooth.

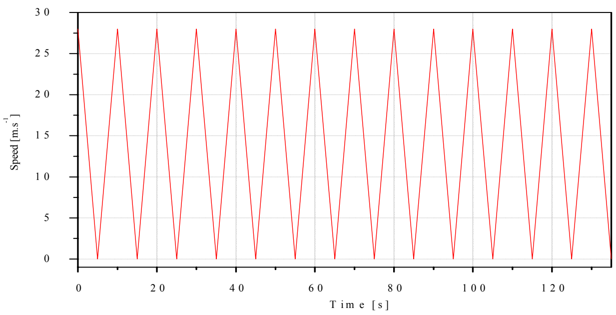


Fig. 19 Driving cycle with fourteen repeated braking (mode 1)

Fig. 20 shows another mode of braking where after each phase of braking one has an idle.

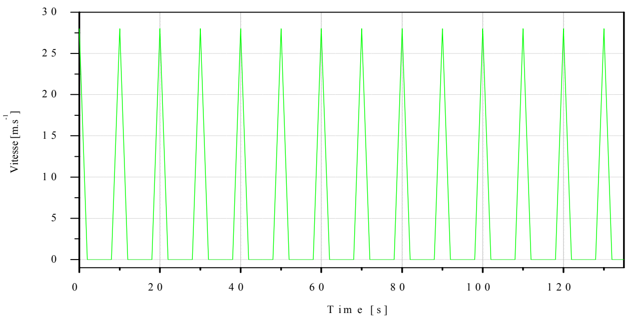


Fig. 20 Cycle braking with phase of idles after each braking (mode 2)

Fig. 21 shows the comparison of the change of temperature of the disc for a cyclic braking process between the first mode and the second mode. For two contours, we note that the temperatures in the disc rise firmly with each application of brake, then begin the exponential decline. The more the number of repetitions of braking increases, the more the maximum temperatures increase. The initial state of the disc changes after each cycle, the downtimes allow only one partial cooling. After each cooling phase, the disc begins to warm again. In fact, during successive brakings the capacity of cooling of the disc is insufficient to lower the surface temperature to near the initial temperature, which causes an accumulation of energy and therefore a higher surface temperature. These results show that the transient thermal behavior of a disc brake depends on the braking cycle imposed and it is dominating because it dictates the cooling time of the disc. According to Fig. 21, we note that in the case of braking cycle mode 2, a reduction of the temperature of approximately 535°C is 45.19% compared to the first cycle. We conclude that the braking mode with a cooling phase influences very positively on the heat transfers in the disc what involves a reduction in the maximum temperature of interface which causes cracking and mechanical wear. In addition this tendency will enable us to ensure safety and fatigue life of the brake system component. Finally it would be interesting to carry out this calculation on brake test benches in order to validate these results of the numerical simulation.

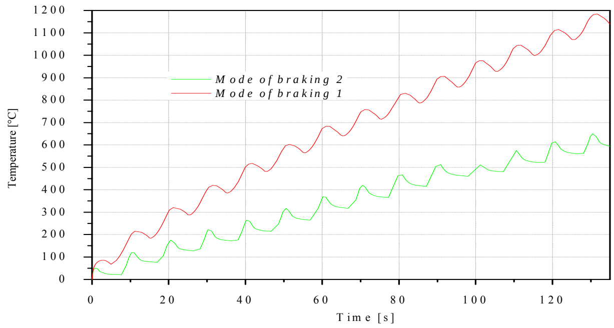


Fig. 21 Temperature variation of the two braking modes versus time

8. Conclusion

In this study, we presented a numerical simulation of the thermal behavior of a full and ventilated disc in transient state. By means the computer code ANSYS 11 we were able to study the thermal behavior of three types of cast iron (AL FG 25, FG 20, FG 15) for a determined braking mode. In addition to the influence of the ventilation of the disc, we also studied the influence of the braking mode on the thermal behavior of the discs brake. The numerical simulation shows that radial ventilation plays a very significant role in cooling of the disc in the braking phase. The obtained results are very useful for the study of the thermomechanical behavior of the disc brake (stress, defomations, efficiency and wear).

Through the numerical simulation, we could note that the quality of the results concerning the temperature field is influenced by several parameters such as:

- technological parameters illustrated by the design;
- numerical parameters represented by the number of element and the step of time;
- physical parameters expressed by the type of materials;
- braking mode implemented.

About the results obtained, in general, on can say that they are satisfactory in comparison with already carried out research tasks. Compared to the prospects, one finds interesting to also make an experimental study of the disc of brake for example on test benches in order to show a good agreement between the model and reality.

References

1. **Černiauskas, E.; Keršys, A.; Lukoševičius, V.; Sapragonas, J.** 2010. Investigation of anti-intrusion

beams in vehicle side doors, Mechanika 6(86): 11-16.
2. **Majcherczak, D.; Dufrénoy, P.; Naït-Abdelaziz, M.** 2001. Thermal simulation of a dry sliding contact using a multiscale model – application to the braking problem, Thermal stresses, Osaka (Japan), 437-440, juin, 2001.
3. **Colin, F.; Floquet, A; Play, D.** 1988. Thermal contact simulation in 2-D and 3-D mechanisms, ASME Journal of Tribology 110: 247-252.
4. **Pileičikienė, G.; Šurna, A.; Skirbutis, G.; Barauskas, R.; Šurna, R.** 2010. Influence of guiding tooth geometry on contact forces distribution in the human masticatory system: a FEM study, Mechanika 3(83): 34-39.
5. **Reimpel, J**. 1998. Technologie de freinage. Vogel Verlag, Würzburg.
6. **Gotowicki, Pier Francesco; Nigrelli, Vinzenco; Mariotti, Gabriele Virzi.** 2005. Numerical and experimental analysis of a pegs-wing ventilated disk brake rotor, with pads and cylinders, 10 th EAEC European Automotive Congress – Paper EAEC05YU-AS04 – P 5, June 2005.
7. **Dhatt, G ; Thouzot, G.** 1984. Une présentation de la méthode des éléments finis. Malouine S.A., Paris: 539p.
8. **Debard, Y.** 2006. Méthode des éléments finis : Modélisation Numérique et Réalité Virtuelle. Master Université du Maine mars 2006.
9. **Bathe, K.-J.** 1996. Finite element procedures in engineering analysis. Englewood cliffs, New Jersey USA: Prentice Hall, 735p.
10. **Huebner, K.H.; Thornton, E.A.; Byrom, T.G.** 1995. The finite element method for engineers, 3 rd ed. John Wiley and sons, New York.
11. **Lewis, R.W.; Morgan, K.; Thomas, H.; Seetharamu, K.N.** 1996. The finite element method in heat transfer analysis. Wiley.
12. www.ansys.com.
13. **Carlos H. Galindo - Lopez.** Evaluating new ways of conducting convective heat dissipation experiments with ventilated brake discs" Cranfield University, Bedfordshire, MK43 OAL, UK.

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SAUSOSIOS TRINTIES STABDŽIŲ DISKŲ ĮŠILIMO TYRIMAS, NAUDOJANTIS PROGRAMA ANSYS v11.0

R e z i u m ė

Stabdant transporto priemonę kinetinė energija yra paverčiama mechanine, o ši išsklaidoma šilumos forma. Stabdymo metu trinties sukuriama šiluma gali įkaitinti stabdžio diskus ir trinkeles iki aukštos temperatūros. Šis reiškinys yra labai svarbus, nes jį sukuria tangentiniai įtempiai ir santykinis slydimas kontakte. Šio darbo tikslas – naudojant ANSYS skaitmeninį kodą yra išnagrinėti neaušinamų ir aušinamų stabdžių diskų šiluminę būklę. Temperatūros pasiskirstymo modeliavimo stabdžių diske tikslas – nustatyti visus stabdymo veiksnius ir parametrus, t. y. stabdymo būdą, disko konstrukciją ir medžiagą. Imitavimo rezultatai patenkinamai sutampa su rezultatais, paskelbtais literatūroje.

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STUDY OF THE THERMAL BEHAVIOUR OF DRY CONTACTS IN THE BRAKE DISCS «APPLICATION OF SOFTWARE ANSYS v11.0»

S u m m a r y

Braking is a process which converts the kinetic energy of the vehicle into mechanical energy which must be dissipated in the form of heat. During the braking phase, the frictional heat generated at the interface disc - pads can lead to high temperatures. This phenomenon is even more important that the tangential stress as well as the relative sliding speeds in contact is important. The objective of this study is to analyze the thermal behavior of the full and ventilated brake discs of the vehicles using computing code ANSYS. The modeling of the temperature distribution in the disc brake is used to identify all the factors and the entering parameters concerned at the time of the braking operation such as the type of braking, the geometric design of the disc and the used material. The results obtained by the simulation are satisfactory compared with those of the specialized literature.

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ИССЛЕДОВАНИЕ НАГРЕВА ТОРМОЗНЫХ ДИСКОВ СУХОГО ТРЕНИЯ, ИСПОЛЬЗУЯ ПРОГРАММУ ANSYS v11.0

Р е з ю м е

При торможении транспортных средств кинематическая энергия превращается в механическую, а последняя рассеивается в форме тепла. Во время торможения тепло, созданное трением, может нагреть тормозные диски и колодки до высокой температуры. Это явление является очень важным так как нагрев создают тангенциальные напряжения и относительное скольжение в контакте. Цель этой работы – исследование теплового состояния невентилируемых и вентилируемых тормозных дисков с использованием числового кода ANSYS. Цель моделирования определение распределения температуры в тормозном диске с учетом всех факторов и параметров, действующих во время торможения, то есть способа торможения, конструкции и материала диска. Результаты, полученные во время имитации удовлетворительно соответствуют опубликованным в литературе.

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