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Transient thermal behavior of automotive dry clutch discs by using Ansys software

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1. Introduction

Current clutches are the result of a long technical evolution which, since 1960, resulted in the single disc configuration with dry diaphragm, spring washer Belleville. The configuration with only one friction disc has a high compactness, very important for applications where cross motor the motor, clutch, gearbox and differential must be contained within the width of the hood of the car. The clutch system is the set of mechanical components that can be coupled and uncoupled gradually the engine to the drive train of the vehicle transmission. The clutch plate receives the rotational movement of the flywheel and transmits it to the transmission input shaft. It basically consists of six elements, the flywheel, the diaphragm spring, the clutch disc, the pressure plate, the clutch housing and the control necessary for operation of the clutch (Fig. 1) [1].

When setting the vehicle movement phase starting or off, the clutch ensures smooth transition between zero speed and the minimum speed of said vehicle idling speed. To ensure movement of the vehicle layout and the limit of retaking regime of the engine, it is sufficient to accelerate the engine gradually while releasing the clutch. During gear ratio changes, the clutch temporarily disengages the motor from the rest of the transmission. This results in a reduction in engine torque and a speed difference between the engine and the input shaft box. After engagement of the gear ratio, the gradual engagement of the clutch ensures synchronization of the rotational speeds of the two shafts. During normal operation, the clutch is fully forward and the engine torque without sliding of the rest of the transmission.

The clutch operation is controlled directly by the driver through the gradual release of the pedal. It consists of four phases: a recovered very fast pedal until reaching the contact point where the pressure plate comes into contact with the friction disc, a slower closing to increase the vehicle acceleration to the desired level a holding position until the synchronization and eventually complete closure. The frictional torque generated by the clutch during the slip or engagement of the clutch phase can be very important and generate a significant amount of heat may cause thermal cracks and deformation of the clutch disc, especially for the high motor vehicles. In this context, the object of this study is to analyze the thermal behavior of a single plate dry clutch during the starting phase and gear changes by modifying a number of parameters influencing tried to minimize heating power of the clutch plate and thus optimize the design of the clutch from the thermal point of view.

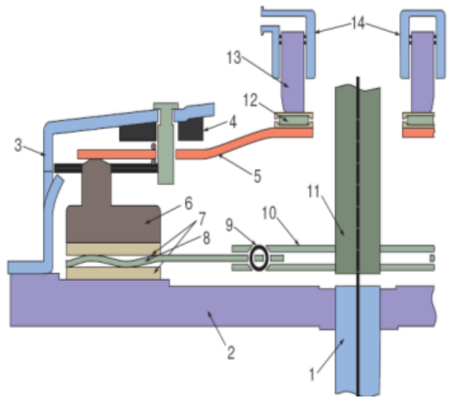


Fig. 1 The structure of a clutch, (axial section): 1 - crankshaft; 2 - flywheel; 3 - clutch casing; 4 - system catchup wear; 5 - Belleville washer; 6 - pressure plate; 7 - disc pad; 8 - spring; 9 - spring-damper; 10 - clutch disc; 11 - primary shaft of box; 12 - needle; 13 - the slave cylinder piston bearing; 14 - cylinder slave

In recent years, many studies have been performed to study the thermal problems associated with dry friction sliding, including disc brakes and clutches. M. Bouchetara et al. applied the finite element method for brake disk contact modeling to analyze the thermoelastic behavior [2]. They determined the heat transfer coefficient by the finite volume method and evaluated the effect of ventilation on the cooling ventilated discs using different models [3]. B. Czél et al [4] used the three-dimensional finite element method to evaluate the temperature distribution within a ceramic disc model, taking into consideration the effect of convection along with cyclic repetitions, achieving an alignment with their experimental results. Seo et al [5] have proposed a thermal model to estimate the temperature distribution for a lubricated clutch, came up with a good approach between their theoretical and experimental results. Zhang et al [6] have developed a new model for oiled clutches, considering the hydrodynamic aspect, using Matlab and Simulink software to calculate the temperature (range) during the course of engagement. A two dimensional conducting model was used to evaluate the temperature distribution along its radial and axial directions, the obtained results have validated the experimental ones. O.I. Abdullah et al [7] used the two dimensional finite element methods to solve the problems related to the clutches, for that purpose they have chosen two types of loads with constant pressure and wears; As a result, they have concluded that the max temperature for the first load is higher than the second loading. O.I. Abdullah and J. Schlattmann [8] developed a two dimensional finite element model for evaluating energy correction factor for a uniform pressure change with a repetitive frequency of 10 engagements. The heat flow commitments are lowest at the inner radius and highest at the outer radius. Choon et al [9] used the finite element method to study the effect of thermomechanical loads applied on pressure plate as well as the friction clutch system, three types of loads are considered, the heat load due to sliding, pressure and the centrifugal force due to the rotation; subsequently, it is recommended to increase the thickness of the pressure plate in order to increase the heat capacity. El-Sherbiny and Newcomb [10] have used the finite difference method for modeling the thermal balance within each part of the clutch, they determined the temperatures at various elements when the contact band occurs between the friction surfaces of an automobile clutch, and they also evaluated the simple and repetitive engagement.

In this work, we study the temperature distribution for three models of clutches during the sliding time and for a single engagement, changing the friction material, the angular velocity and the pressure exerted by the pressure plate. So, the main purpose of this study is to analyze and modify the model of clutch disc to minimize the friction energy converted to heat due to conduction and convection; to find solutions for a better clutch model.

2. Heat flow in sliding phase

The clutch friction mechanism consists mainly of the pressure plate, flywheel and clutch disc, it also, consists of two packing as well as an axial disc, which is in the middle (Fig. 2).

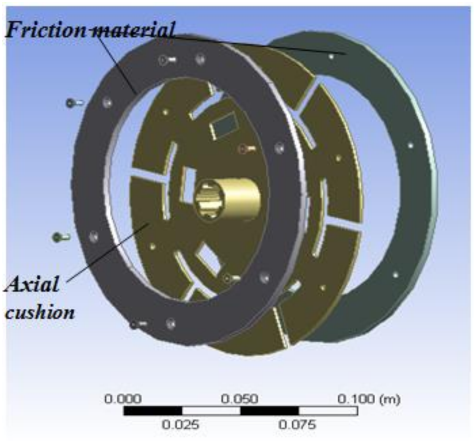


Fig. 2 Parts of clutch disc

During sliding phase, we may notice heat production due to the friction between the two linings of the disc and the different parts. During the sliding phase of the clutch, the speed difference between the friction surfaces may be equated with Coulomb friction model. The friction torque generates a heat flow which is expressed by the following equation [11] (Fig. 3):

$$T = n\,\mu\, r_m F_n\, \omega_s, \tag{1}$$

where T - friction torque of the clutch; μ - dynamic coefficient of friction; r_m - average radius of the gasket crown; F_n - normal Effort applied to friction surfaces; ω_s - angular velocity slip; n - number of friction surfaces in the clutch disc ($n = 2$, double surface disk motor-friction wheel and pressure-plate disc).

Heat flux q expressed the quantity of heat generated by friction and based on the total friction surface:

$$q = \mu\, p r_m \omega_s, \tag{2}$$

p - pressure exerted on the clutch disc.

At time t, the ω_s slip angular velocity equal to:

$$\omega_s = \omega_0\left(1-\frac{t}{t_s}\right), \tag{3}$$

ω_0 - initial angular velocity of slip of the clutch disc; t_s - sliding time.

The heat flow equation becomes:

$$q_{(r,t)} = \mu\, p r \omega_0\left(1-\frac{t}{t_s}\right). \tag{4}$$

In this study, we opted for the case of a uniform lining wear because that choice is decisive for calculating the friction torque and the heat flow. Under these conditions, the friction torque and the heat flow are expressed as follows respectively (Fig. 3) [8]:

$$T = n p_{max} \mu r_i\left(r_e^2 - r_i^2\right); \tag{5}$$

$$q = \frac{T\,\omega_0}{A_f}, \quad 0 \prec t \prec t_s, \tag{6}$$

p_{max} - maximum pressure which determines the maximum transmittable torque; A_f - total friction surface; r_i, r_e, ω_0 represents respectively the inner radius and the outer radius the angular velocity of sliding of the clutch disc.

$$q = \mu\, p_{max} r_i\, \omega_0\left(1-\frac{t}{t_s}\right). \tag{7}$$

The thermal flow generated during the sliding phase is partitioned between the friction portions [12]:

$$q_{(r,t)} = \xi\, \mu\, p\, r\, \omega_0\left(1-\frac{t}{t_s}\right). \tag{8}$$

The coefficient of thermal ξ sharing depends on the physical properties of materials in contact:

$$\xi = \frac{\sqrt{k_c\rho_c c_c}}{\sqrt{k_c\rho_c c_c}\sqrt{k_f\rho_f c_f}} = \frac{\sqrt{k_c\rho_c c_c}}{\sqrt{k_c\rho_c c_c}\sqrt{k_p\rho_p c_p}}, \tag{9}$$

ξ, k, ρ and c are respectively the thermal partition coefficient, thermal conductivity, density and specific heat of materials with their indices: c for clutch disc, f for the flywheel and p for pressure plate.

In this study, we apply total heat for parts of friction, we use contact model [13]:

$$q_1 = q_2 \, . \tag{10}$$

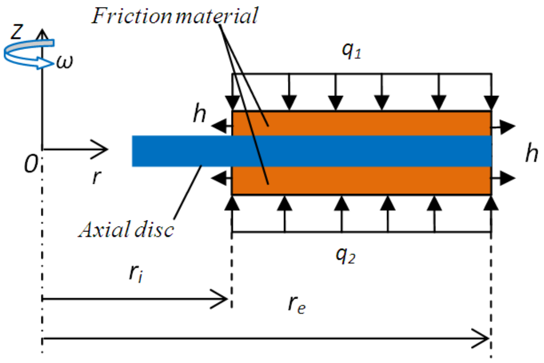


Fig. 3 Thermal loading model with uniform wear

3. Transient eat transfer by conduction and convection

Transient heat conduction in three dimensional heat transfer problem is Governed by The Following differential equation [2]:

$$-\frac{\partial q_x}{\partial x} - \frac{\partial q_y}{\partial y} - \frac{\partial q_z}{\partial z} = \rho c \frac{\partial T}{\partial t} \, , \tag{11}$$

q_x, q_y, and q_z are conduction heat fluxes in x, y and z directions, respectively, c is the specific heat, ρ is the specific mass, and T is the temperature that varies with the coordinates as well as the time t The heat conduction Eq. (11) is given for material with no internal heat production.

The conduction heat flows can be written in the form of temperature using Fourier's law. Assuming constant and uniform thermal properties, the conduction heat flowrelations are:

$$q_x = -k_x \frac{\partial T}{\partial x}, \; q_x = -k_y \frac{\partial T}{\partial y}, \; q_z = -k_z \frac{\partial T}{\partial z}, \tag{12}$$

k_x, k_y and k_z are thermal conductivity in x, y and z directions, respectively.

For the case of a disc clutch, the boundary conditions are usually the conduction and convection (Fig. 3). The bondary condition is:

$$T_s = T\left(x, y, z, t\right), \; -q_s = h\left(T_s - T_\infty\right), \tag{13}$$

q_s the specified surface heat flux (positive into a surface); h the convective heat transfer coefficient; T_s the unknown surface temperature, and T_∞ the convective exchange temperature.

4. Determining the heat transfer coefficient by ANSYS CFX

To determine the coefficients of heat transfer by convection, we proceed to the modeling of the clutch disc and the flow of air through and around the disc. Fig. 4 shows the three clutch disc variants selected for this study. Main geometrical characteristics of the clutch plate are given in Table 1. For each variant of the clutch disc, one builds the corresponding CFD model that includes the solid domain (clutch plate) and the air domain (Fig. 5). Using CFX code, we proceeded to the domain mesh solid and air field (Figs. 6 and 7). The element type used is linear tetrahedral four-node.

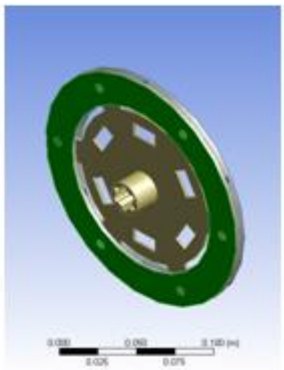
Table 1

Geometric characteristics of the clutch disc [14]

Properties	**Values**
Inner radius of friction lining r_i, m	0.06298
Outer radius of clutch r_e, m	0.08721
Thickness of friction lining, m	0.003
Thickness of axial disc, m	0.0015
Maximum pressure, MPa	1
Maximum angular speed ω_0, rad/s	200
The Groove width, m	0.003
Depth of a Groove, m	0.001

The calculated values of the thermal transfer coefficient $h = h(t)$ on each free surface of the disc will be imported using the ANSYS CFX module. After it gets the configuration by CFX PRE and defines the parameters of the model Physics:

- The fluid domain: ambient air at 22°C and reference pressure of 1atm with a variation of speed, one chooses a turbulent flow of type k-ε. This allows us to observe the turbulence around the disc.
- The solid domain: the clutch disc with a variable speed, we choose the SiC for the friction surfaces and steel for the axial disc, the initial temperature is 40°C .



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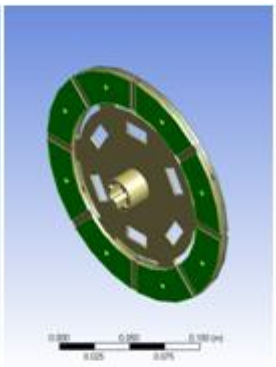

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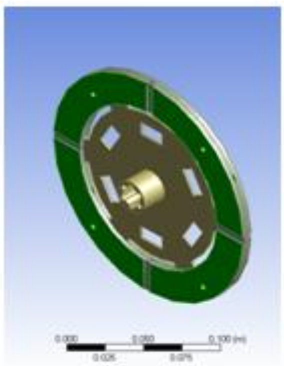


c

Fig. 4 Clutch disc of variants: a - full disc; b - four friction surfaces; c - eight friction surfaces

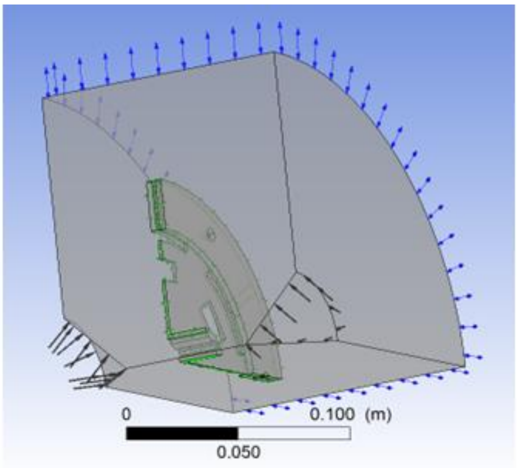


Fig. 5 Model full disk – air

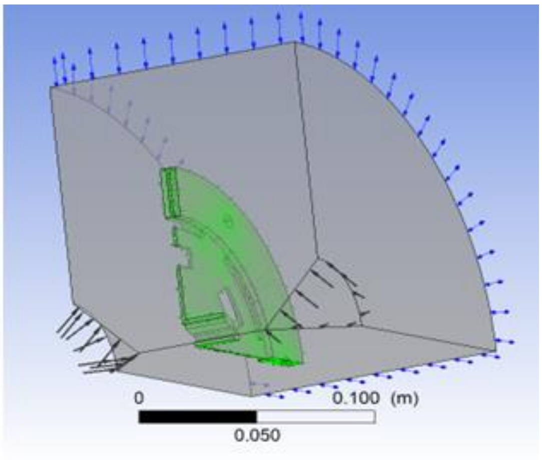


Fig. 6 Meshing of full disk with 26817 elements

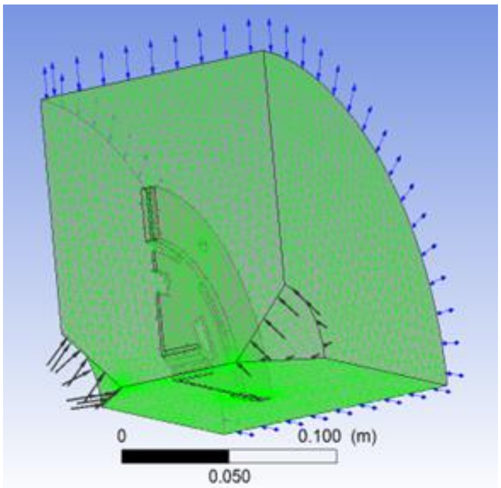
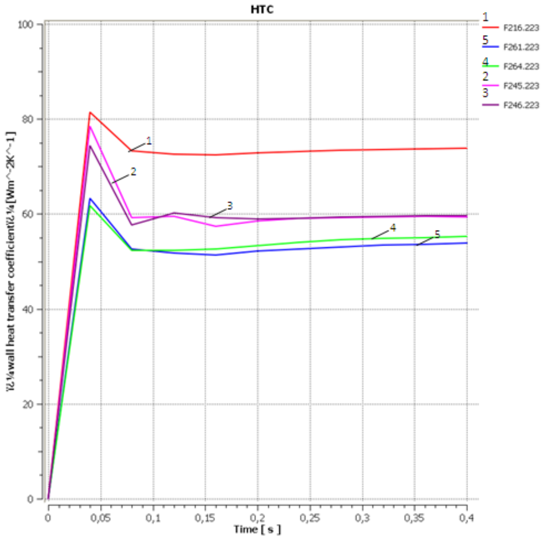


Fig. 7 Meshing for air with 147787element

Then the definition of the boundary conditions is necessary for the fluid domain, the selection of the side inlet of the air "inlet" with a speed of 16.67 m/s that also represents the speed of the vehicle with an intensity of turbulence by 5%. Air enters by "opening" on choosing this type of condition with a relative pressure equal to zero, it also uses the wall condition for fluid and solid domain to indicate adiabatic surfaces, and in the external surface we applied the interfaces of the domains (fluid-solid) and (solid-fluid).



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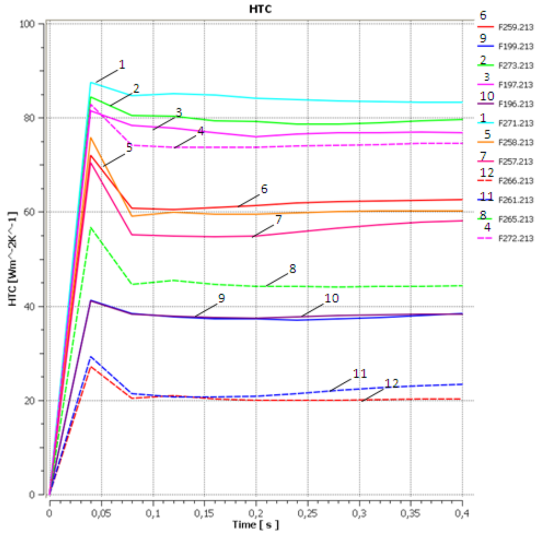

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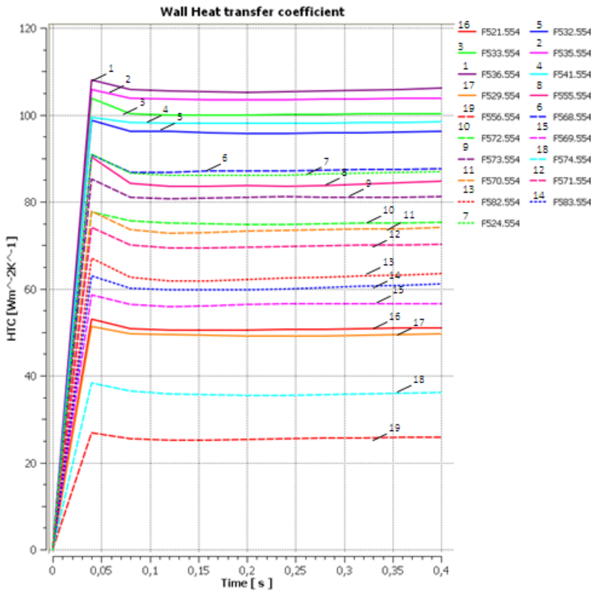


c

Fig. 8 Heat transfer coefficient for interface air disc: a - full disc; b - four surfaces of friction; c - eight surfaces of friction

For modeling, we valid “thermal energy” option, considering that the clutches are axisymmetric and all materials are isotropic, three discs are assumed fixed and heat flux is applied with the relationship (7) on the friction surfaces, and heat flux distribution is uniform on the friction surfaces. But before you start solving the heat problem, we must enable the calculation of the heat transfer coefficient for the surfaces of solid-fluid interfaces. The evolution of the thermal transfer coefficient for the three clutches discs’ variants is given in Fig. 8. We note that ventilation of the disc through a multiple surface of friction has a positive effect on the heat transfer coefficients, using the values of heat transfer coefficient calculated by ANSYS CFX to evaluate the boundary conditions related to the convection of the free surfaces of the clutch disc and then to analyze the transient change in temperature. At the beginning of the sliding phase, part of the frictional heat escapes into the air by convection. The determination of the heat transfer coefficient is necessary [15].

5. Evolution of the transient temperature of the clutch disc

5.1. FE model and boundary conditions

The thermal transient simulation is performed using the ANSYS Workbench by introducing the initial conditions (Fig .9) and limits the thermal and loading mode applied to the friction surface (Fig. 10) and also the convection conditions imposed in air-disc interfaces (Fig. 8). The initial and boundary conditions are:

- total time sliding $t = 0.4$ s;
- initial time step = 0.004 s;
- minimum initial time = 0.0004 s;
- initial temperature of disc $T_i = 40$°C at $t = 0$ s;
- disc type analyzed: full disc and disc with four and eight surfaces of friction;
- material: SiC (Table 2);
- the heat flux is applied if a uniform wear;
- the disc initial angular velocity $\omega_0 = 200$ rad / s.

We have used three types of elements: the quadratic Tetrahedron element having10 nodes, APDL name is Mesh 2000 for the volume mesh. Regarding the contact zone, the quadratic triangular target element Targe170, while for the contact between axial disc and friction material the quadratic triangular is used.

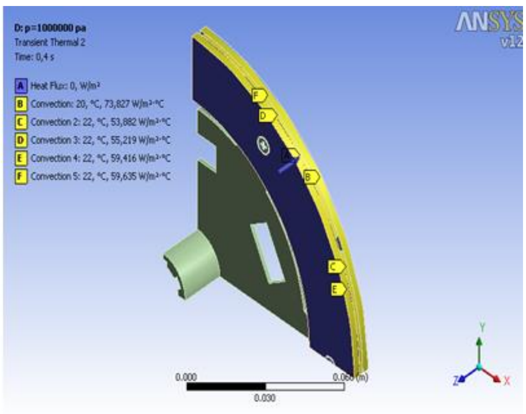


Fig. 9 Boundary conditions and loading

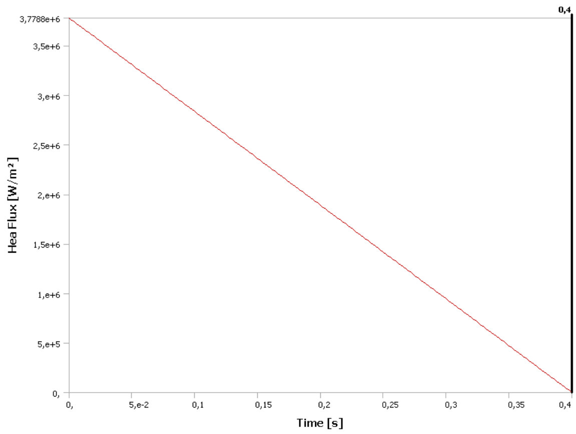


Fig. 10 Evolution of thermal flux in proportion to the sliding time

Table 2

Properties of the clutch plate material [16]

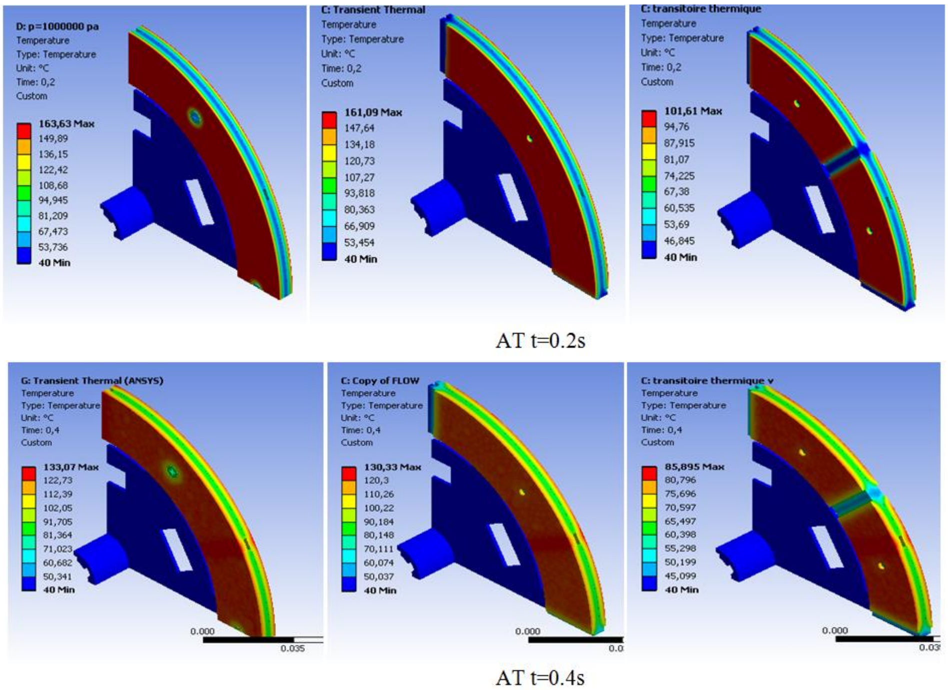
Materials	SIC	SI_3N_4	AL_2O_3	Steel
Friction coefficient /steel side	0.3	0.25	0.18	-
Modulus of elasticity, GPa	420	290	320	25
Heat conductivity, W/m K	35	11	25	42
Specific heat, J/kg K	900	830	950	450
Density, kg/m^3	3150	3100	3980	7800

5.2. Temperature distribution of clutch disc

In this part of the study, comparing different disc variants to assess the temperature levels and the effect of the number of pads on the cooling disc. The Fig. 11 and Fig. 12 show that the disc fully filling reached a temperature maximum $T = 161.1$°C at $t = 0.2$ s and $T = 133.07$°C at the end of the slip time $t = 0.4$ s.

For the disc with 4 surfaces, the maximum temperature reached is $T = 163.63$°C at $t = 0.20$ s and $T = 130.33$°C at the end of the slip time, while the disc with 8 surfaces its maximum temperature is $T = 101.61$°C at $t = 0.20$ s and $T = 85.89$°C at the end of the slip time. These values clearly show that end of the slip time increasing the number of friction surfaces of the disc linings greatly reduces the temperature level and thus reduces thermal stress.

Fig. 13 shows the variation of the temperature along the radius of the disc. It is found that the temperature increases in an almost linear manner in the radial direction of the disc. This means that the temperatures at the inner edge of the disc are minimum and maximum on the periphery of the disc. For full disc, the temperature rises from $T = 71.21$°C in internal radius at $T = 124.15$°C in outer radius, a difference of 53°C. For the disc with 4 surfaces, the temperature difference is 42°C. For the disc with 8surfaces, it ranges from $T = 54.53$°C to $T = 82.27$°C in a linear manner (Fig. 13). The temperature difference is only 28°C. The temperature variation depending on the thickness of the disc is shown in Fig. 14. In the case of full disc, the temperature is maximum at the ends $T = 122$°C and the minimum temperature in the middle of the axial disc with $T = 85$°C. The other two variants have a similar behavior but with lower temperatures. For the disc with 4 surfaces, the temperature is maximum at the ends of the disc with an average $T = 119$°C and in the middle of the axial disc 83.03°C. In the disc with 8 surfaces the average temperature at the ends of the disc is 80°C, and at the axial disc, it is 62.34°C.



a b c

Fig. 11 Temperature distribution at $t = 0.2$ s and $t = 0.4$ s for: a - full disc; b - disc with 4 surfaces of friction; c - disc with 8 surfaces of friction

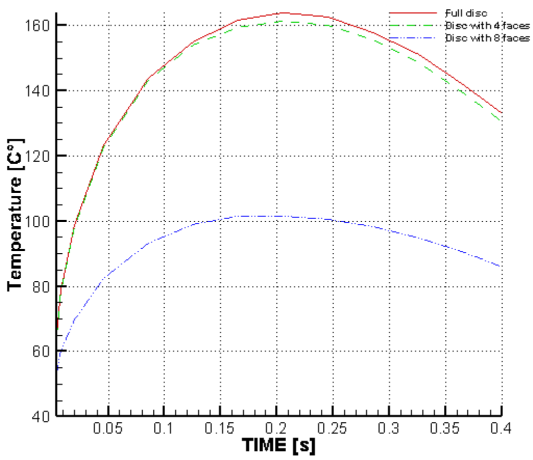


Fig. 12 Comparison between different types of clutches

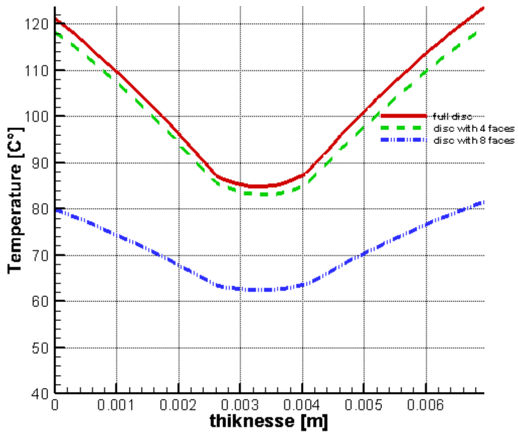


Fig. 14 Temperature variation depending on the disc thickness

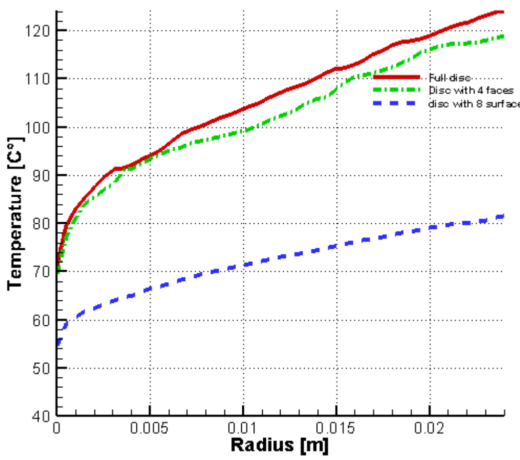
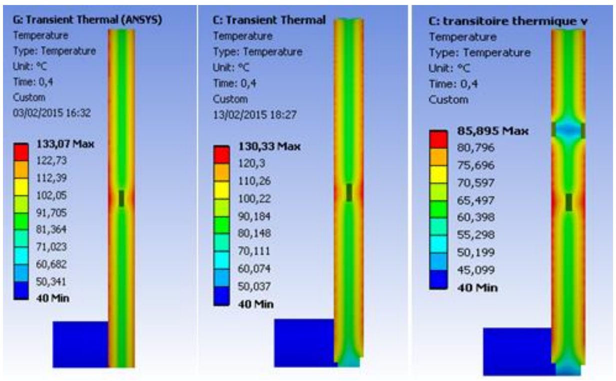


Fig. 13 Temperature variation along the disc radius



a b c

Fig. 15 Temperature variation through a disc's thickness for three variants of discs: a - full disc; b - four surfaces of friction; c - eight surfaces of friction

Fig. 15 shows the temperature distribution of the three following variants clutch disc thickness. It is noted at the level of the groove there is a temperature drop relative to the flat areas of the clutch disc. This is due to the flow of cooling air. The ventilation effect of disc with 4 surfaces and 8 surfaces of friction is visible through the temperature drop.

6. Parametric investigation

6.1. Effect of the friction material on the thermal behavior of the clutch

As for the case of disc brakes, the choice of friction material has a positive effect on the thermal and tribological behavior of the clutch elements. In this section, we try to compare the thermal behavior of three different materials of a single disc clutch linings. The chosen materials are ceramics SiC carbide, alumina Al2O3 and Si3N4 silicon nitride. They are generally used in the tribological and thermomechanical domain because of their good wear behavior and friction and resistance to thermal stresses.

From Fig. 16, the temperature of the liner Si_3N_4 during the sliding phase is much larger than the other two types of SiC and alumina linings. The liner Si_3N_4 reaches a maximum temperature $T = 234.78$°C at $t = 0.2$ s and at the end of the period of sliding $T = 183$°C, while for the material of alumina Al_2O_3 is $T = 115.48$°C at $t = 0.21$ s and $T = 95.10$°C to $t = 0.4$ s. The maximum variation in temperature at $t = 0.20$ s is 120°C and at $t = 0.40$ s it is 88°C. In case of the lining of SiC, the maximum temperature at the instant $t = 0.20$ s is $T = 163.91$°C and passes at $T = 133.07$°C at $t = 0.40$ s. From the thermal point of view, the best alternative is the lining of alumina Al_2O_3.

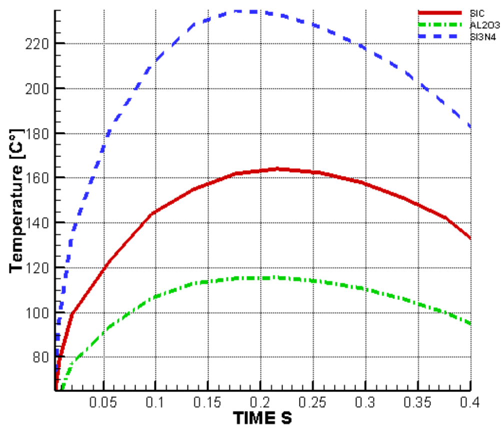


Fig. 16 Effect of friction material

6.2. Influence of the speed of rotation

The initial rotational speed has an influence on the friction torque and the thermal flows into and thus on the thermal behavior of the clutch disc. Fig. 17 shows the evolution of the maximum temperature of the full clutch disc according to the slip time for different initial speeds. For an initial rotational velocity of $\omega = 200$ rad / s, the maximum temperature of the disc at $t = 0.2$ s is for $T = 164$°C, at the end of the sliding period $T = 133.10$°C. For $\omega = 150$ rad / s, then $T = 133$°C at $t = 0.20$ s and $T = 109.50$°C at the end of sliding. For $\omega = 100$ rad / s, T_{max} reached 120°C at $t = 0.2$ s and at the end of sliding time $T = 86$°C.

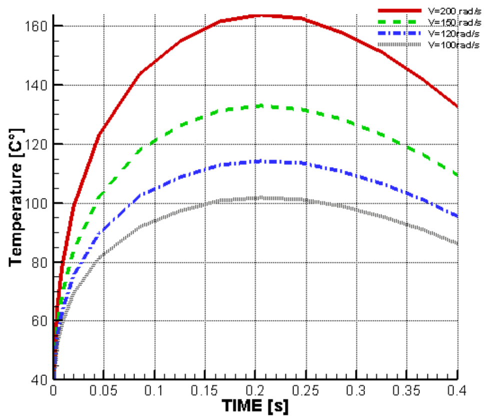


Fig. 17 Influence of rotational velocity on the maximum temperature of the full clutch disc

6.3. Effect of slidding time on the disc temperature

In this part, we chose the full disc variant SiC using the same initial and boundary conditions used in paragraph 5.The sliding total time which corresponds to total closure was only varied,the Fig. 18 shows the temperature field for different sliding time.

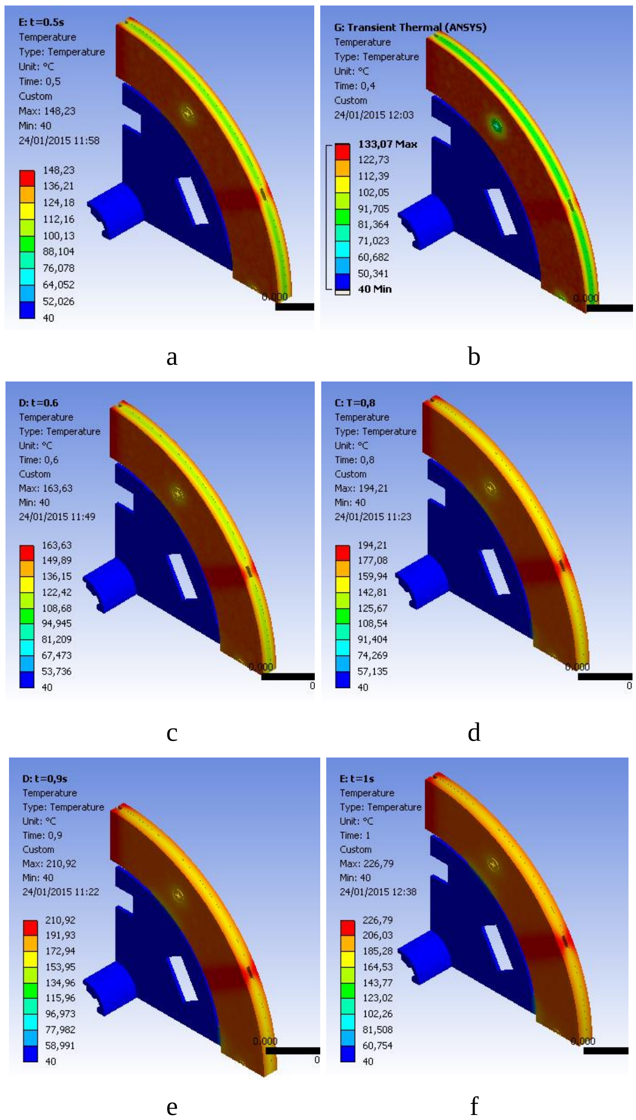


Fig. 18 Temperature field of a full disc for each sliding time: a - $t = 0.5$ s; b - $t = 0.4$ s; c - $t = 0.6$ s; d - $t = 0.8$ s; e - $t = 0.9$ s; f - $t = 1$ s

6.4. Influence of pressure loading exerted on the disc temperature

Fig. 19 shows the influence of the pressure exerted on the two friction faces of the clutch full disc on its thermal behavior as a function of the slip time. The simulation results show that the increase of the pressure on the clutch disc is increased his temperature. This is because of the entering heat flow which is directly dependent on the pressure exerted on the disc.

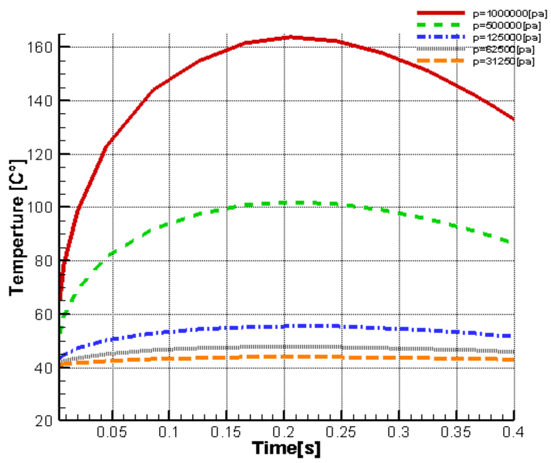


Fig. 19 Effect of loading on the temperature of the clutch disc

7. Conclusion

In this study was analyzed using the calculation code Ansys the thermal behavior of different variants clutch disc during the sliding phase assuming a single engagement of the clutch disc. The models studied are a full disc, four and eight friction surfaces.

The results of the simulation for the case of full disc liner that shows the evolution of the temperature of the disc 3D model is analogous to the 2D model studied by OI. Abdullah [11]. The absolute maximum temperature of the disc is reached half the total shift time (t = 0.2 s), and this regardless of the drive variant chosen. Increasing the number of friction surfaces of the pads, separated by ventilation grooves decreases the friction surfaces and promotes the transfer of heat by convection and therefore the cooling of the clutch disc. At the end of the slip time, the temperature difference between a full disc and a disc with 8 friction surfaces is 48°C, a decrease of 37%. The temperature distribution in the radial and axial direction is not uniform. It is higher on the periphery of the disc relative to the inner edge. In the case of the disc at full filling the radial temperature difference is about 43.69%.

In the parametric study, it was found that the choice of the lining material greatly influences the thermal behavior of the clutch disc. The alumina linings AL_2O_3 have better thermal behavior in the sliding phase in relation to the linings with the two other materials: SiC carbide ceramics and silicon nitride Si_3N_4. Increasing the rotational speed strongly influences the increase in the temperature of disc. We note that reducing of the sliding time also contributes to the improvement of the thermal characteristics of the disc. It is preferable to reduce the sliding period. This also applies to the pressure on the clutch disc.

The thermal behavior of a clutch disc depends on the choice of several factors:

- disc model;
- friction material;
- rotation velocity;
- loading and motorization;
- boundary conditions and loading.

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TRANSIENT THERMAL BEHAVIOR OF AUTOMOTIVE DRY CLUTCH DISCS BY USING ANSYS SOFTWARE

S u m m a r y

The clutch is an important element for comfort during frequent maneuvers so-called start-up and speed gear change for a vehicle equipped with a manual transmission. During the phase of sliding or closing the clutch, the clutch torque is generated by the friction of the pad on the flywheel disc and the plate and then transmitted to the input shaft of the box by the friction disc. This friction generates a quantity of heat which leads to a temperature rise of the clutch. During this sliding phase of the clutch, the speed difference between the friction surfaces may be equated with Coulomb friction model. Currently, the clutches are subjected to high thermal stresses due to the engine. It is therefore useful to estimate the parameters influencing the thermal behavior of the clutch disc when starting or amount gearshift.

In this study, we chose three geometric clutch models designed in 3D using Solid Works software, which are then converted into FEM model. Before beginning the analysis of the transient thermal behavior of each model, we proceed to the evaluation of the heat transfer coefficients using ANSYS CFX code. We also analyzed the effect of parameters such as the time of the slip, the angular speed, the lining material and the pressure exerted by the plate on the thermal behavior of the clutch to arrive at end correlations between the selected parameters and the temperature of the clutch disc.

Keywords: Dry friction, clutch disc, friction materials, heat flow, finite element methods.

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