STUDY AND IMPROVEMENT OF BRAKE PADS COOLING PROCESS

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Abstract *– This work deals with the improvement of brake pads cooling system, which reduces its temperature after passing through an oven to cure the ink. The cooling is needed in order to make brake pads handling easier and allow them to be packed in the shortest time possible. The equipment was evaluated using mass and energy balances and the results showed that convection is the main heat exchange mechanism and need to be intensified to reduce as much as possible parts temperature. Several changes in the equipment cooling system were tested and measured, culminating in the configuration that showed the best performance. As a result a 53% increase in the brake pads energy withdrawal was obtained and a reduction of 38.9% in its outlet temperature. Due to differences on parts types and environmental conditions in each test, cooling effectiveness was also calculated. The effectiveness increased from 0.62 to 0.86, ensuring that the improvements in equipment performance are valid for any situation.*

Keywords: cooling, heat conduction, convection, radiation, fans, brake pads, heat exchangers.

1. INTRODUCTION

According to Brezolin (2007), brake pads are essentially elements intended to cause friction on the surface of the brake disc to convert kinetic energy into thermal energy. The basic composition of these pieces is: friction material, which makes the contact with the surface of the brake disc; background material, whose function is to improve the adhesion between the friction material and adhesive; adhesive, which is responsible for the adhesion between the friction material and metal support or metal plate, which in turn serves to receive and distribute the force received from the brake piston to the brake friction material, and, hence, to the brake disc. In some cases it is used an anti-noise, that are plates of several composites, and serves to reduce or eliminate potential noise from the friction between the pad and disc brake.

Figure 1 shows a brake pad example.

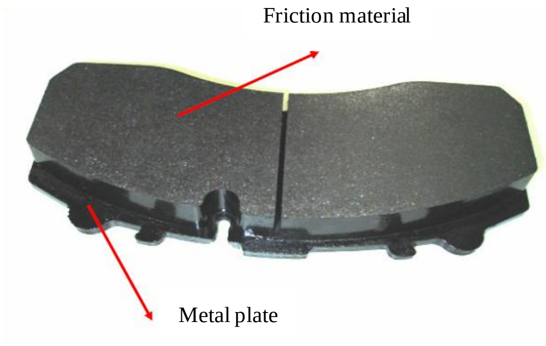

Figure 1 – A brake pad example (www.fras-le.com)

In accordance with Canali (2002), the friction material compositions are, in most cases, industrial secrets. Therefore each manufacturer tends to have their own products with specific characteristics, or yet, each manufacturer tends to produce families of materials with very similar characteristics.

A basic composition of a friction material includes six kinds of main components: binders (thermosetting resins), which serves to keep together the raw materials used; fibers (acrylic fibers, carbon, brass, copper, steel, vegetables, etc.), whose function is to provide mechanical properties to the product and improve its processing ability within different manufacturing stages; abrasives (aluminum oxide, iron, manganese, etc.), which increase the friction coefficient and prevent its reduction with increasing temperature; organics (rubber, coke, graphite, cellulose, polymers, etc.); loads (barite), which are responsible for completing the matrix formulation that already contains the required amount of other components; lubricant (graphite and sulfides of molybdenum, antimony, copper-iron, zinc, titanium, etc.), whose function is to reduce the friction coefficient and wear or abrasion of the brake disc (Canali, 2002).

Powder coatings produce an attractive finish that is characterized by excellent resistance to corrosion, heat, impact, abrasion, weathering and extreme heat or cold. The possibilities of color and finishes are virtually unlimited and may have finish of high, medium or low brightness, metallic effects, pearlescent, transparent and even colorless. Powder coatings can be formulated for application in low or high layer, according to the desired purpose (Durac, 2004 apud Teixeira, 2004).

In this paper is studied the final stage of the production process of brake pads manufacturing for car usage. This final stage refers to the cooling process of brake pads, post-curing process that occurs at moderately high temperature. After cooling, the pieces are stamped and packaged, which might occur in a suitable temperature for the operators. To study this process it is necessary to review the main concepts of Thermodynamics, Fluid Mechanics and Heat Transfer that will be used in sequence. The current literature on these subjects (Borgnakke and Sonntag, 2009, Moran and Shapiro, 2002; Granet, 1995; Fox and McDonald, 2006; Kreith and Bohn, 2003; Incropera and DeWitt, 2003; Braga Filho, 2004; Thomas, 1985) discourses widely about the basic concepts necessary to develop the work presented here.

In order to estimate the cost, feasibility and size of equipment needed to transmit a given amount of heat, at given time, there should be a detailed analysis of the process (Kreith and Bohn, 2003), identifying precisely the fluids/substances flowing through the control surface under study, their characteristics and the ways in which heat transfer takes place.

On great part of heat exchange equipments, convection is the predominant mechanism, specially forced convection, obtained through external agents, such as fans, pumps, etc. The heat exchanged by convection in the binomial surface-fluid can be determined by Eq. (1) representative of the Newton's Law of Cooling:

$$q_{conv} = h \cdot A \cdot (T_S - T_\infty) \tag{1}$$

where, q_{conv} is the rate of convection heat transfer [W]; h is the convection heat transfer coefficient [$W \cdot m^{-2} \cdot K^{-1}$]; T_S is the surface temperature [°C]; e T_∞ is the fluid temperature [°C].

The measure of the relative importance of the thermal resistance inside a solid body is given by the ratio of internal resistance (conduction) and external resistance (convection). This ratio is represented by the dimensionless Biot number (Kreith and Bohn, 2003) and given by Eq. (2)

$$Bi = h \cdot L / k_s \tag{2}$$

where, Bi is the Biot number, h is the convection heat transfer coefficient [$W \cdot m^{-2} \cdot K^{-1}$]; L is a characteristic length [m], defined as the ratio of the solid volume and its surface area; and k_s is the thermal conductivity of the solid [$W \cdot m^{-1} \cdot K^{-1}$]. Usually, when the internal resistance is less than ten percent of the external resistance (Biot <0.1), the internal temperature gradients can be neglected, i.e., the temperature of the solid can be considered uniform and the capacity concentrated method is accepted (Incropera and DeWitt, 2002).

In general, the determination of the convection heat transfer coefficient can be made via Nusselt number. This dimensionless relates the convection heat transfer with the conduction heat transfer in the fluid, being a function of Reynolds and Prandtl numbers. Nusselt number can be determined by Eq. (3):

$$Nu = h \cdot L / k_f \tag{3}$$

where, Nu is the Nusselt number (dimensionless); L is the characteristic length [m]; and k_f is the thermal conductivity of the fluid [$W \cdot m^{-1} \cdot K^{-1}$].

According to Braga Filho (2004), the Prandtl number is defined as the ratio between the momentum diffusion and the thermal diffusion for the fluid at rest or on laminar flow. In turbulent conditions, other aspects such as momentum transport and thermal energy must be considered. The Prandtl number is dimensionless and can be obtained by Eq. (4):

$$Pr = c_p \cdot \mu / k = \nu / \alpha \tag{4}$$

where, Pr is the Prandtl number (dimensionless); c_p is the specific heat of the fluid [$J \cdot kg^{-1} \cdot K^{-1}$]; μ is the dynamic viscosity of the fluid [$kg \cdot m^{-1} \cdot s^{-1}$]; k is the thermal conductivity of the fluid [$W \cdot m^{-1} \cdot K^{-1}$]; ν is the kinematic viscosity of the fluid [$m^2 \cdot s^{-1}$];and α is the thermal diffusivity of the fluid [$m^2 \cdot s^{-1}$].

Some heat transfer correlations, which can be applied to the situation studied, are suggested on literature (Braga Filho, 2004; Kreith and Bohn, 2003; Incropera and DeWitt, 2002). For laminar external flows along of plates and Prandtl numbers greater than 0.6, the average Nusselt number can be defined by Eq. (5):

$$Nu = 0.664 \cdot Re^{0.5} \cdot Pr^{0.333} \tag{5}$$

For turbulent external flows along of plates, Prandtl numbers greater than 0.6 and Reynolds number between $5 \cdot 10^5$ and $1 \cdot 10^8$, the average Nusselt number can be defined by Eq. (6):

$$Nu = (0.037 \cdot Re^{0.8} - 871) \cdot Pr^{0.333} \tag{6}$$

2. MATERIALS AND METHODS

Before entering into the equipment where the cooling process of the brake pads occurs, they pass through an electrostatic painting booth, in which a layer of black powder paint is applied. After painting, the pads go through a curing oven, reaching temperatures around 200 °C. Powder particles are fused in the pieces, forming a film, with thickness ranging from 40 to 50 µm. After passing through the curing system, the pads enter the cooling process with the metal plate upwards. The brake pads studied are basically 151 mm long and 43 mm wide, with a mass of 0.294 kg. The cooling system consists of a carbon steel conveyor belt, a hood for air distribution and five fans, of which, only one blows air into the cooling system, while the others remove air. The procedures adopted and the instrumentation used in measurements were the following: a) the speed measures of fans air streams were taken with an Instrutherm digital anemometer, model TAVR-650 with accuracy of ± 3% and resolution of 0.1 $m \cdot s^{-1}$ and in appropriate points at admission (supplying fan) or at discharge (exhaust fans); b) the temperature measures of air flows were performed with an Impac digital thermo-hygrometer, model tastotherm-hum RP2 with a resolution of 0.1 °C and accuracy of ± 1 °C in the range from 0 °C to 35 °C and ± 2 °C outside this range; c) the temperature measures of the brake pads before and after cooling process were performed with a laser thermometer mark Minipa, model MT-350 with accuracy of ± 2% on reading range from -30 °C to 550 °C and resolution of 1 °C. The readings were also checked with a Minipa contact thermometer, model MT-521 with accuracy of ± 0.3% and resolution of 0.1 °C. The air flow rates were obtained multiplying the cross-sectional area of the fan duct by the respective flow velocity.

In order to determine the specific heat of the brake pad an 8 liters calorimeter with temperature measurement using thermocouples type J (limits of error of ± 0.4%) was used. The calorimeter is filled with water at room temperature and brake pad is heated above to the water initial temperature. The brake pad is then immersed in water and the calorimeter closed, slightly stirring the water to maintain its temperature uniformity. Once the water and the brake pad temperatures are in equilibrium, the temperature values are recorded and the brake pad specific heat can be calculated by Eq. (7):

$$m_{BP} \cdot c_{p,BP} \cdot \Delta T_{BP} = -(m_{H2O} \cdot c_{p,H2O} \cdot \Delta T_{H2O}) \tag{7}$$

where, m_{BP} is the mass of the set brake pad-metal plate [kg]; $c_{p,BP}$ is the specific heat of the brake pad [$kJ \cdot kg^{-1} \cdot °C^{-1}$]; ΔT_{BP} is the difference between the initial brake pad temperature and that one at the end of the equilibrium temperatures [°C]; m_{H2O} is the mass of water into the calorimeter [kg]; $c_{p,H2O}$ is the specific heat of water [$kJ \cdot kg^{-1} \cdot °C^{-1}$]; ΔT_{H2O} is the difference between the initial temperature of water and that one at the end of the equilibrium temperatures [°C]. On Equation (7), using the known values of m_{BP}, ΔT_{BP}, m_{H2O}, $c_{p,H2O}$ and ΔT_{H2O}, a $c_{p,BP}$ value equal to 0.925 $kJ \cdot kg^{-1} \cdot °C^{-1}$ is obtained.

Adopting the perfect gas behavior for air, it is possible to determine its density, considering the local atmospheric pressure (93 kPa) and the absolute environment temperature. Thus, the air mass flow rates through the fans are found.

The shaft powers of the fans are determined from Eq. (8):

$$P = 3^{0,5} \cdot V \cdot I \cdot \cos\phi \cdot \eta_{MOTOR} \tag{8}$$

where, P is the fan shaft power [kW], V is the motor electrical voltage [Volts]; I is the electric current demanded by the motor [Amps], measured with a clamp ammeter; $\cos\phi$ is the power factor of the motor, which is indicated in it identification plate; η_{MOTOR} is the electrical motor efficiency.

Figure 2 shows a basic scheme of the cooling equipment, whose main components are the following: 1 - centrifugal fan (OTAM brand), model RLS 630, transmission by pulleys and belts; 2 - axial fan (OTAM brand), model AFR 630, transmission by pulleys and belts; 3 - centrifugal fan (OTAM brand), model 280 RLS, with electrical motor directly coupled to the fan shaft; 4 and 5 - axial fans (OTAM brand), model AFR 355, with electrical motors directly coupled to the fan shafts; 6 - carbon steel hood, whose function is to distribute the air flow from the fan 1 over the extent of the conveyor belt through a carbon steel perforated plate with holes of 9 mm in diameter distributed in 5.34 m², being the perforated area of about 1.82 m^2, or 33.5% of the total area of the perforated plate. Just below the fan 2, the upper part of the hood, there is a 600 mm round opening of 600 mm (indicated by red line); and 7 - Carbon steel conveyor belt

5.97 m long and 1 m wide. The brake pads passing over the conveyor belt stay 43 mm away from the perforated plate.

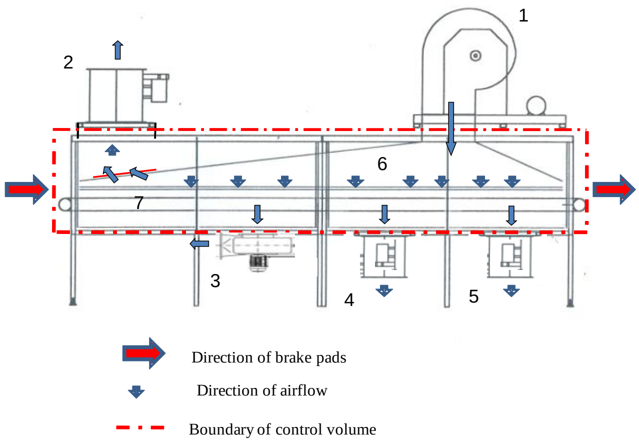

Figure 2 – Cooling equipment scheme

3. THERMAL EXCHANGER VALUES FOR INITIAL CONFIGURATION

At first, the working parameters of the fans were measured, following the configuration shown in Fig. 2, whose operation data of the fans are as follows: Fan 1 – efficiency, η_{F1}=66% (performance curves) and blowing 16,061 m³·h^{-1} of air; Fan 2 - η_{F2}=70% and removing 15,163 m³·h^{-1} of air; Fan 3 - η_{F3}=60% and removing air at a rate of 3,534 m³·h^{-1}; Fan 4 - η_{F4}=70% and removing air of the equipment at a rate of 3,290 m³·h^{-1}; and Fan 5 - η_{F5}=67% with an air removal rate of 3,463 m³·h^{-1}.

In order to calculate the inlet energy flow with the brake pads in the cooling process, the Eq. (9) can be used:

$$q_{inlet,BP} = m_{f,BP} \cdot c_{p,BP} \cdot \Delta T_{BP} \qquad (9)$$

where, $q_{inlet,BP}$ is the inlet energy flow with the brake pads in the cooling process [kW]; $m_{f,BP}$ is the pads mass flow rate [kg·s^{-1}]; $c_{p,BP}$ is the specific heat of the pads [kJ·kg^{-1}·°C^{-1}]; ΔT_{BP} is the difference between the inlet brake pads temperature ($T_{inlet,BP}$) in the cooling process [°C] and a reference temperature (T_R) , here considered equal to 0 °C. For the data known of $m_{f,BP}$ (0.238 kg·s^{-1}), $c_{p,BP}$ e $T_{inlet,BP}$ (204°C) and through Eq. (9), a $q_{inlet,BP}$ value of 44,91 kW is obtained.

The outlet brake pads energy flow rate ($q_{outlet,BP}$) is obtained by a similar way of the Eq. (9), and assuming the mean value measured of the outlet brake pads temperature ($T_{outlet,BP}$) of 88°C, a value of 19.37 kW for this parameter is found.

Subtracting the $q_{outlet,BP}$ of $q_{inlet,BP}$, it is verified that the equipment removes 25.54 kW of energy from the brake pads. If would be desired an outlet temperature of 30 °C, for example, an energy removal of 38.31 kW would be needed, i.e., 50% more.

Analyzing the air flow through the control volume, the results reported in Table 1 were found.

Table 1 – Air mass flow rates and temperatures

INLET	MASS FLOW RATES (kg·s^{-1})	TEMPERATURE (°C)
Fan 1	4.96	18.1
Openings through the conveyor	2.80	19.5
OUTLET		
Fan 2	4.64	20.7
Fan 3	1.07	26.0
Fan 4	0.99	24.0
Fan 5	1.06	21.0

The air flow through the openings where pass the conveyor belt was determined by the difference between air flows of the fans, because the anemometer measurement at these sites showed large variations. So it can be calculated the inlet and the outlet energy flows with air through the control volume boundary and, hence, the power supplied by the electrical motors shaft work should be considered. The inlet air energy flow in the cooling process can be calculated by Eq. (10):

$$q_{inlet,AIR} = c_{p,AIR}\cdot(m_{AIR,inlet\text{-}FAN1}\cdot\Delta T_{AIR,inlet\text{-}FAN1} + m_{AIR}\cdot\Delta T_{inlet,AIR}) \tag{10}$$

where, $q_{inlet,AIR}$ is the inlet air energy flow in the cooling process and before fan 1 [kW]; $c_{p,AIR}$ is the specific heat of air [kJ·kg^{-1}·°C^{-1}]; $m_{AIR,inlet\text{-}FAN1}$ is the air mass flow rate aspirate by fan 1 [kg·s^{-1}]; $\Delta T_{AIR,inlet\text{-}FAN1}$ is the difference between the inlet air temperature in the cooling process before fan1 [°C] and the reference temperature (T_R = 0°C); m_{AIR} is the inlet air mass flow through the openings in the conveyor [kg·s^{-1}]; $\cdot\Delta T_{inlet,AIR}$ is the difference between the inlet air temperature through the openings in the conveyor [°C] and the reference temperature. With the values of $c_{p,AIR}$ =1.006 kJ·kg^{-1}·°C^{-1} and the values of the Tab. 1 of the inlet air flows and the respective temperatures, a $q_{inlet,AIR}$ equal to 145,24 kW is obtained.

The outlet air energy flow rates of the cooling process can be determined by Eq. (11) (considering the powers supplied by shaft works):

$$q_{outlet,AIR} = c_{p,AIR}\cdot[(m\cdot\Delta T)_{outlet,AIR\text{-}FAN2} + (m\cdot\Delta T)_{outlet,AIR\text{-}FAN3} + (m\cdot\Delta T)_{outlet,AIR\text{-}FAN4}] - V\cdot 3^{0,5}\cdot 10^{-3}\cdot[(I\cdot\eta\cdot\cos\phi)_{MOTOR\text{-}FAN1}\cdot\eta_{F1} + (I\cdot\eta\cdot\cos\phi)_{MOTOR\text{-}FAN2}\cdot\eta_{F2} + (I\cdot\eta\cdot\cos\phi)_{MOTOR\text{-}FAN3}\cdot\eta_{F3} + I_{MOTOR\text{-}FAN4}\cdot\eta_{F4} + I_{MOTOR\text{-}FAN5}\cdot\eta_{F5}] \tag{11}$$

where, $q_{outlet,AIR}$ is the outlet air energy flow rate of the cooling process [kW]; m is the air mass flow rate of each fan [kg·s^{-1}]; ΔT are the difference between the outlet air temperatures of the respective fans [°C] and the reference temperature; V is the voltage of the feeding network of the electrical motors of the fans [Volts]; and I is the electrical current required by the electrical motor [Amp]. The portions regarding the fans 4 and 5 do not include the efficiency and Power factor because its electrical motors are installed inside the housing. The transmission efficiency of the motors-fans 1 and 2 was considered equal to 1. Assigning values for Eq. (11) a $q_{outlet,AIR}$ of 166.01 kW is determined, being the first term of the right side of this equation equal to 170.91 kW and is the outlet air thermal power. Adding the energy flow removed of the brake pads (25.54 kW) with the inlet air energy flow rate (145.24 kW), a value of 170.78 kW is obtained, value 4.77 kW higher than that one calculated by the outlet air. The difference may be due to heat transfer from the brake pads to pieces of the equipment, such as side plates, and metallic elements of the conveyor.

The equipment studied can be treated as a heat exchanger of opposite streams, therefore, its effectiveness is determined by Eq. (12):

$$\varepsilon = C_h\cdot(T_{h,e} - T_{h,s}) / C_{min}\cdot(T_{h,e} - T_{c,e}) \tag{12}$$

where, ε is the heat exchanger effectiveness, dimensionless; C_h is the thermal capacity of the hot fluid [kW·K^{-1}]; $T_{h,e}$ is the inlet hot fluid temperature [K]; $T_{h,s}$ is the outlet hot fluid temperature [K]; C_{min} is the lower heat capacity, of the hot fluid or cold fluid [kW·K^{-1}]; and $T_{c,e}$ is the inlet cold fluid temperature [K]. In the process studied here, the hot fluid are the brake pads, whose thermal capacity (C_{BP}) is equal to 0.22 kW·K^{-1} (0.238 · 0.925); the thermal capacity of the cold fluid (air) is equal to 7.81 kW·K^{-1} [(4.96 + 2.8)·1.006], hence, C_{min} = C_{BP}. As yet, $T_{h,e}$ = 204°C, $T_{h,s}$ = 88°C e $T_{c,e}$ = 18.1°C, the value of the effectiveness (ε) of the cooling process is 0.62.

From the initial configuration of the equipment, some changes were made. The perforated plate was removed and a tube installed between the hood and the inlet fan 2. After that the convective coefficient between the brake pads and air was estimated. In this way, the mean air velocity and the mean air temperature across the pads were measured, whose values are: V_{AIR} = 11.3 m·s^{-1} e T_{AIR} = 15.3 °C. Considering the air properties in that temperature (288.45 K) and the velocity measured, the Re and Pr values were 114,825 e 0.71, respectively; so, according to Eq. (5), Nu = 200.7 and by Eq. (4), h = 33.7 W·m^{-2}·K^{-1}. It's important to mention that the air velocity measured is close to Fan 1 and so, it might be reduced throughout the conveyor, as long as the heat transfer coefficient.

However, the changes mentioned did not bring advantages, because the outlet temperature of the brake pads increased from 88°C to 90°C, although the effectiveness of the cooling process has increased from 0.62 to 0.68.

To identify internal temperature gradients in the brake pads, the Biot number (Bi) from Eq. (2) was calculated. Thus, based on the length of each pad and its mean temperature, a Bi value of 0.013 was found, i.e., lesser than 0.1, indicating that the assumption of uniform temperature is valid.

4. TEMPERATURE DROP OF THE BRAKE PADS ALONG OF THE CONVEYOR

In order to measure the temperature drop of the brake pads along the length of the conveyor, a PT-100 temperature sensor was inserted into a pad with a cable of length slightly greater than the length of the conveyor, being this cable connected to a digital temperature controller. In the cable were placed tapes spaced 300 mm from one to another, and when the brake pad comes out of the oven and enters the cooling zone, the PT-100 was inserted into it, so every time a tape was crossing the entrance of cooling, a temperature value was recorded in the controller. The full record of the brake pad temperature was repeated three times. Figure 3 shows details of the arrangement of measurement.

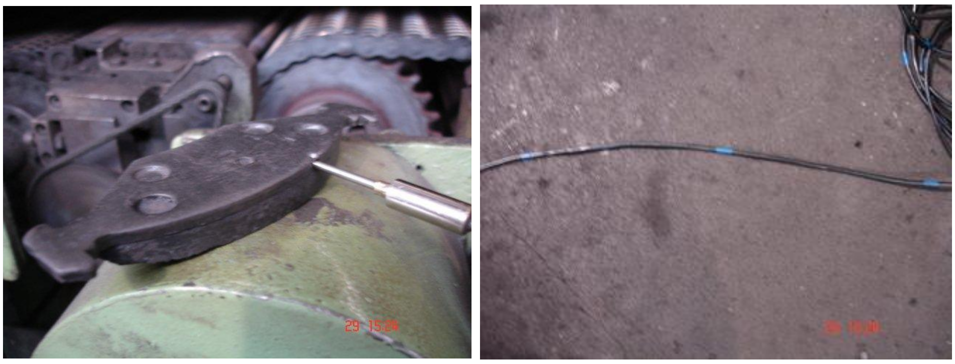

Figure 3 – Drake pad with PT-100 inserted and tapes in the sensor cable

Some other changes were carried out in the equipment and the brake pad temperature measurements along of the cooling process were made for each case. The new settings were six and are as follows: Config. 1 – no perforated plate and the lower fans (3, 4 and 5) on operation; Config. 2 – no perforated plate, but with the lower fans off; Config. 3 - blind plate and the lower fans off; Config. 4 - blind plate and the lower fans on; Config. 5 - perforated plate with the lower fans on (similar at initial configuration); and Config. 6 – perforated plate with the lower fans off. Figure 4 shows for the six configurations specified the chart with all the measurements.

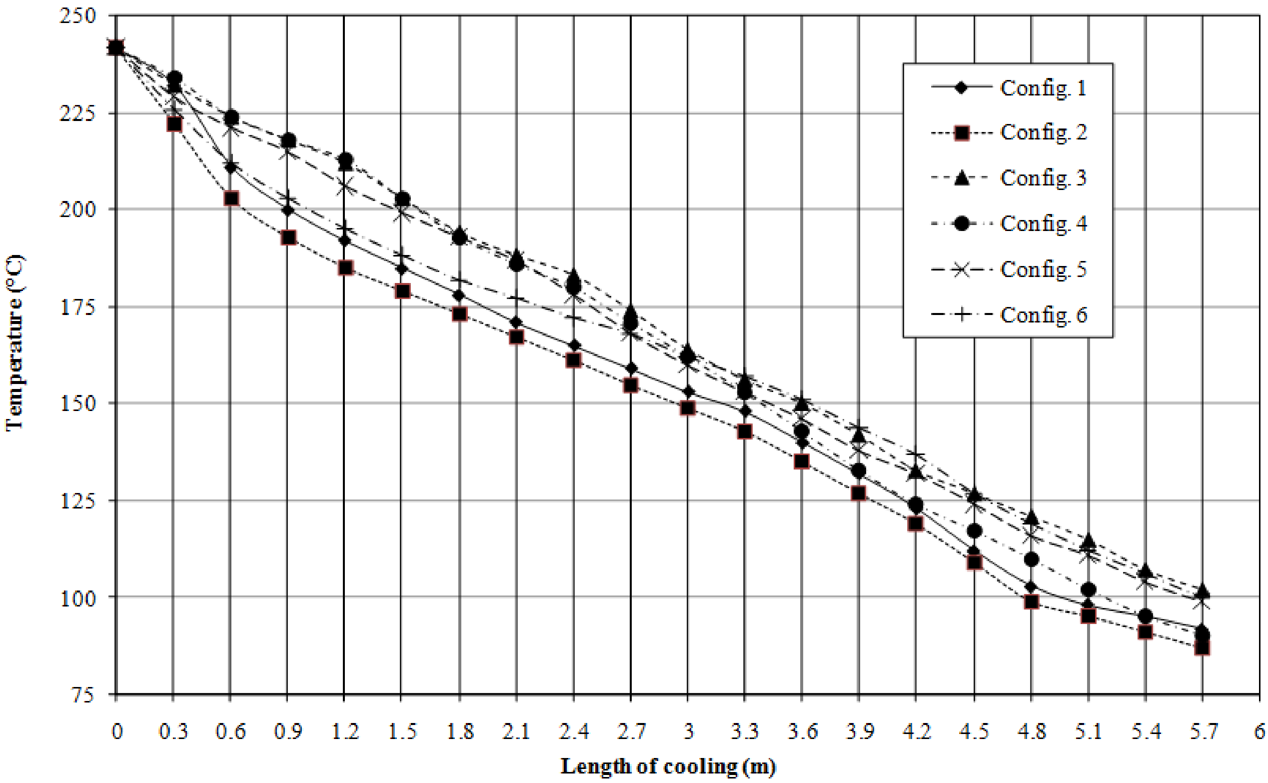

Figure 4 – Brake pad temperature drop along of the cooling process

Table 2 shows the values that produced the graphic of the Fig. 4. From the changes made and the results obtained and shown in Fig. 3 and Tab. 2, it is noted that, without plate, the lower fans were detrimental to the performance, although on the blind plate configuration the lower fans help to the yield, removing more than 12 °C when they were connected (configuration 4). Nevertheless, the better result of this configuration is worse than the result with no plate and fan off. On the configuration with perforated plate (5 and 6), the lower fans almost didn't change the yield of the equipment, and these configurations presented the worst yield.

Table 2 – Brake pad temperatures along of the cooling process

Length (m)	Temperature (°C)					
	No plate/ Fans On	No plate/ Fans Off	Blind plate/ Fans Off	Blind plate/ Fans On	Perf. plate/ Fans On	Perf. plate/ Fans Off
0.0	242	242	242	242	242	242
0.3	232	222	232	234	229	226
0.6	211	203	224	224	221	212
0.9	200	193	218	218	215	203
1.2	192	185	212	213	206	195
1.5	185	179	203	203	199	188
1.8	178	173	194	193	193	182
2.1	171	167	188	186	187	177
2.4	165	161	183	180	178	172
2.7	159	155	174	171	168	168
3.0	153	149	164	162	160	162
3.3	148	143	156	153	153	157
3.6	140	135	150	143	146	151
3.9	132	127	142	133	138	144
4.2	123	119	133	124	132	137
4.5	112	109	127	117	124	127
4.8	103	99	121	110	116	119
5.1	98	95	115	102	111	112
5.4	95	91	107	95	104	106
5.7	92	87	102	90	99	100

The configurations with perforated plate (5 and 6) probably presented the worst results due to the pressure drop that the perforated plate carries out to the upper streams of the fans 1 and 2, and also because it hasn't the same efficiency, regarding to radiation, than the condition with black painted blind plate. It's also observed, that the configuration without the perforated plate and with the lower fans off (2) and that one with the blind plate and the lower fans on operation (4) presented the best results, showing the importance of convection on pads heat removal.

Analyzing only the results of the configuration without the perforated plate and the lower fans off (2), it is noted that there is an increase in the rate of the brake pad temperature drop in different regions of the equipment. Dividing the difference between the temperature of a given position and the position immediately before by the length traversed by the pad gives the rate of temperature drop. Tab. 3 shows the values of the rate of temperature drop according to the length of cooling.

Table 3 – Rate of temperature drop to the configuration without the perforated plate and with the lower fans off

Distance from the entry (m)	T (°C)	$\Delta T / \Delta L$ (°C·m^{-1})	Distance from the entry (m)	T (°C)	$\Delta T / \Delta L$ (°C·m^{-1})
0.0	242	–	3.0	149	20.0
0.3	222	66.7	3.3	143	20.0
0.6	203	63.3	3.6	135	26.7
0.9	193	35.0	3.9	127	26.7
1.2	185	25.0	4.2	119	28.3
1.5	179	21.7	4.5	109	33.3
1.8	173	18.3	4.8	99	33.3
2.1	167	21.7	5.1	95	13.3
2.4	161	18.3	5.4	91	13.3
2.7	155	20.0	5.7	87	13.3
				Average	**21.7**

Overlapping the temperature profile of the configuration without perforated plate and with lower fans off (presented in Fig. 4) over the figure of the cooling process equipment (Fig. 2) it is verified that the highest rates of temperature drop occur in regions close to the higher fans 1 and 2, with values around 64 °C·m^{-1} below the fan 2 and up to 33 °C·m^{-1} below the fan 1. Figure 5 shows this overlapping with a length scale below and without representing fans 3, 4 and 5, because they were disconnected.

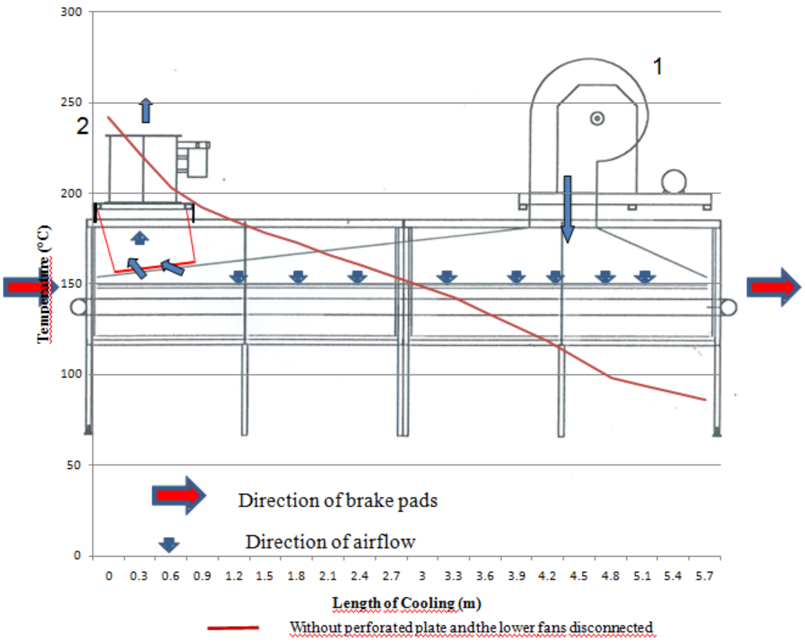

Figure 5 – Overlapping of temperature profile and cooling process equipment

Based on the results it is possible to conclude that the highest heat transfer occurred when the brake pads were crossing the streams of fans 1 and 2, and, therefore, the convection should be intensified to achieve better pads cooling performance.

5. FINAL CONFIGURATION TO IMPROVE THE EQUIPMENT PERFORMANCE

In order to better direct the air flow from fan 1 against the brake pads, a SAE 1010 steel curved sheet 1.9 mm thick was installed. The perforated plate was replaced by a blind plate and were closed the sides of the conveyor belt, thus creating a tunnel 43 mm in height where the air has a constant velocity for the largest possible distance within the equipment and not just close to the fans. The entire surface of the plate facing the brake pads were painted of black, so the air flow besides cooling the brake pads also cools the plate heated by radiation. Figure 6 shows a schematic of the cooling equipment after the changes.

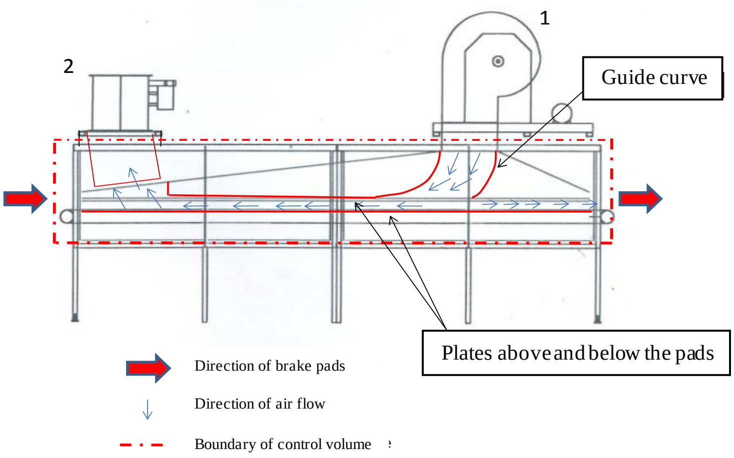

Figure 6 – Scheme of the cooling equipment after the last changes

Table 4 shows the rate of temperature drop according to the length of cooling process for the final configuration.

Table 4 – Rate of temperature drop for the final configuration

Distance from the entry (m)	T (°C)	ΔT / ΔL (°C·m^{-1})	Distance from the entry (m)	T (°C)	ΔT / ΔL (°C·m^{-1})
0.0	242	–	3.0	109	30.0
0.3	214	93.3	3.3	101	26.7
0.6	197	56.7	3.6	92	30.0
0.9	180	56.7	3.9	87	16.7
1.2	166	46.7	4.2	79	26.7
1.5	156	33.3	4.5	70	30.0
1.8	145	26.7	4.8	65	16.7
2.1	135	30.0	5.1	61	13.3
2.4	127	30.0	5.4	58	10.0
2.7	118	26.7	5.7	55	10.0
				Average	**30,0**

Figure 7 presents the curves of the brake pads temperature distribution for the final configuration and for configuration 2. It can be observed from the results that for the final configuration relative to the configuration 2 and the other ones, the outlet temperature is the lowest and the temperature drop is more uniform along the conveyor belt.

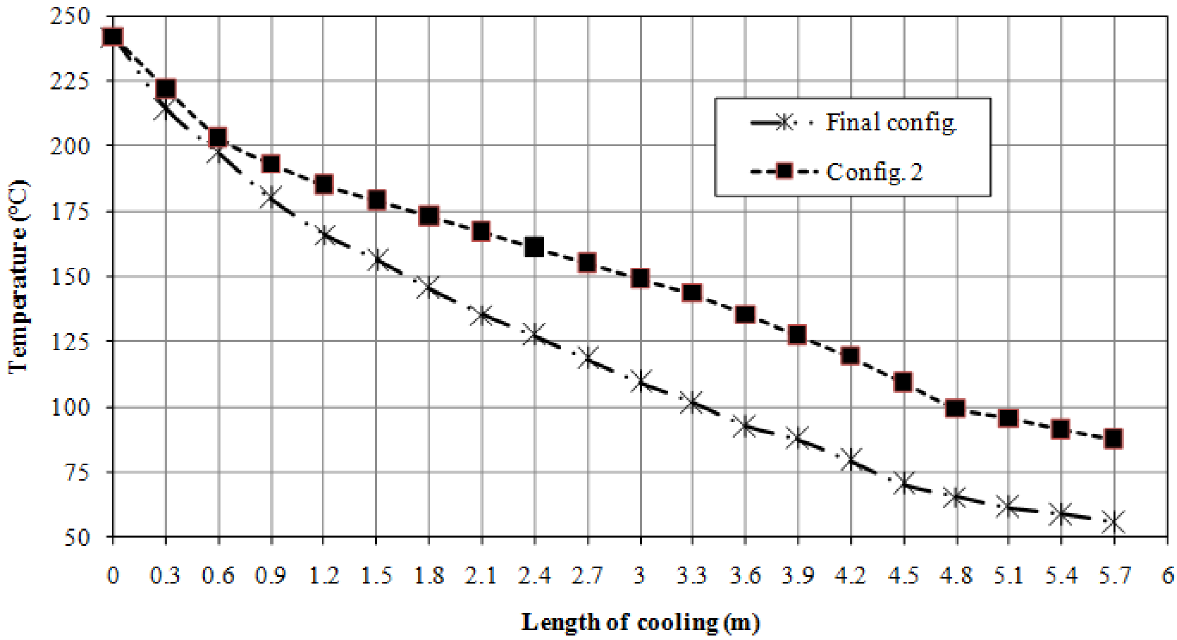

Figure 7 – Brake pad temperature distribution for the final configuration and configuration 2

The rate of temperature drop along the length of the cooling for the last change has a higher average than the configuration without perforated plate (Tab. 3). This is due to the fact that the mean air velocity on the conveyor increased from 11.5 m·s^{-1} to 13.5 m·s^{-1}, being practically constant over the distance between the guide curve of air and the tube of the fan 2. This increase on the velocity provided a h equal to 36 W·m^{-2}·K^{-1} (calculated as already demonstrated in this article, whose value was 33.7 W·m^{-2}·K^{-1}).

Table 5 presents the energy balance for the last change compared with other energy balances held.

Table 5 – Comparison between the energy balances

CONFIGURATION	INLET BRAKE PADS ENERGY FLOW (kW)	INLET AIR ENERGY FLOW (kW)	OUTLET AIR ENERGY FLOW ((kW)	ENERGY REMOVED FROM BRAKE PADS (kW)	DIFFERENCE IN THE ENERGY BALANCE (kW)	COOLING EFFECTIVENESS
Initial	44.91	145.34	165.98	25.54	4.9	0.62
With inlet tube in the fan 2	50.6	196.29	234.51	30.15	-8.07	0.64
No plate/Fans off	50.5	210.72	247.71	31.78	-5.21	0.68
Final	50.59	106.71	140.63	39.09	5.17	0.86

According to reported in Tab.5, the cooling effectiveness on the final configuration calculated by Eq. (12), considering the values of C_h and C_{min} mentioned before and the values of $T_{h,e}$ e $T_{h,s}$ of the Tab. 4, besides $T_{c,e}$ = 24°C, results in a value of 0.86, showing the great improvement obtained with the last modification.

6. CONCLUSIONS

The last modification carried out in the cooling process equipment showed that the convection is the main heat transfer mode that takes place in the equipment. As important as the increase in the convective coefficient was keeping it constant for a longer length of the conveyor, on the contrary of the previous configurations, where the convective coefficient was significant only close to the higher fans. With a final temperature of 55 °C, the brake pads can be stamped and packaged directly without need of waiting, being this the main goal. This will bring financial gains to the company, because the cost with intermediate stocks will disappear. In addition, operators who handle the brake pads on the outlet of the cooling process can pick up the pieces with only one glove, unlike the overlapping gloves that were needed before, causing great satisfaction for the operators, since the previous situation was very uncomfortable, including risk of burns.

The work carried out on this equipment, despite it is based on brake pads for automobiles will be reflected for all references, including pads for commercial vehicles, of greater mass. Also, this work serve as a basis for improvements in other similar devices in the company, considering that there are two other similar equipments whose actions performed in this cooling process can be replicated.

It is also important to note that all changes made in the equipment had a minimum investment, because manpower and materials used were available in the company. Furthermore, by disabling the lower fans, about 2.3 kW of electrical power began to be saved in the cooling operation. Comparing the current situation with the initial situation of the equipment, 0.39 kW of electrical energy was necessary to remove 1 kW of energy flow from the brake pads and now only 0.2 kW of electrical energy to remove 1 kW from the pads is needed. Calculating the COP (Coefficient of Performance) as the ratio between the energy flow removed from the brake pads and electrical power supplied to the cooling process fans, a value of 5.08 for the final configuration is obtained, while the COP for the initial configuration was 2.55, a great improvement on efficiency.

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8. RESPONSIBILITY NOTICE

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