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THERMAL ANALYSIS OF BRAKE DISCS OF A BAJA SAE PROTOTYPE

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Abstract. *During the design of the brake system for an off-road prototype, the temperature reached by the disc during braking must be taken into account, as a result of friction between the disc and the pad, generating the transformation of kinetic energy into thermal. This increase can cause catastrophic and silent failures, such as: loss of friction with a reduction in the coefficient of friction and formation of bubbles in the fluid. The aim of this study is to perform a thermal analysis of the front and rear axle brake discs of a Baja prototype, which will influence their material selection. Therefore, the analytical method was used, varying the ways to obtain the convective coefficient for parallel flow in flat plates. Considering the solid disk and adopting the method of concentrated systems through the analysis with transient condition, with the aid of Matlab software, the behavior of the surface temperature of the disk was analyzed as a function of 100 consecutive braking, every 30 seconds of cooling. Gray cast iron, stainless steel and SAE 1045 carbon steel were the materials analyzed. Comparing the results, it was found that stainless steel had less temperature variation, and in relation to the shafts, the front disc had greater variations.*

Keywords: *Baja, brake, discs, temperature, materials.*

1. INTRODUCTION

The Baja SAE (Society of Automotive Engineers) program is a challenge to undergraduate students, focusing on engineering projects with the aim of designing and manufacturing a prototype off-road vehicle for competitions. The first one took place in 1976 at the University of South Carolina, and in Brazil it started to be held in 1994. The SamaBAJA team was founded in 2013 as a pioneer in Ifes (Federal Institute of Espírito Santo) representing the São Mateus campus in regional and national competitions, and has as its primary goal the development of a safe prototype, from conception, detailed design (sizing and design of components), manufacturing, to testing, with the lowest possible cost and/or mass and that can overcome obstacles (SamaBaja, 2021). One of the most important parts of the vehicle is the braking system that must slow the speed of prototype or reduce it to zero. This is possible after applying a torque contrary to the rotation movement of the wheels, capable of promoting deceleration regardless of the conditions of use: loaded or unloaded, dry or wet floor, low or high speed, uphill or downhill track, etc. (Leal et al., 2012).

In passenger and competition vehicles it is common to use the drum or disc brake system. The SamaBAJA team uses the disc brake because it provides lower mechanical losses and deformations in the components, has better braking efficiency, higher resistance to high temperatures, and a simple maintenance, when compared to the drum brake (Ciolfi, 2010). On this system, the main element is the disc, coupled to the wheels that are fixed to the axle of the car, and in each of them there is a caliper connected. The system is actuated when the brake pedal is pressed by the driver, and the force applied is multiplied by the lever principle. Then, the fluid is pressurized in the brake connections by the master cylinder, until it meets the caliper piston, which is responsible for promoting the meeting of the brake pads with the brake disc. Thus, in the period of deceleration occurs the heating of the brake components, especially the disc due to friction (Limpert, 1999). For this reason, a detailed analysis of the maximum temperature reached in the discs is of crucial importance in order to avoid friction loss, since this temperature increase causes a reduction in the friction coefficient of the pad and disc; it can cause the formation of bubbles in the fluid, which reduces line pressure; irregular wear, geometric deformation, and thermal cracks in the friction elements, according to Ciolfi (2010) and Iombriller (2002).

This paper aims to verify the temperature variation in the solid disc of the front and rear axle of the SB07 prototype of SamaBAJA team by varying its material (stainless steel, gray cast iron and SAE 1045 steel) and the method of obtaining the convective coefficient, through the correlations proposed by Çengel (2009) and Mitschke (1972).

2. METHODOLOGY

In order to determine the temperature variation in the front and rear axle disc of the prototype after 100 braking cycles, an analytical modeling was performed using the Matlab software, since a code was developed through the relationship between the vehicle dynamics and heat transfer mechanisms, thus using the method of concentrated systems to obtain the surface temperature of the brake discs.

2.1 Modeling for a single brake

During braking, the potential and kinetic energy generated due to the motion of the prototype are converted into thermal energy as a result of the friction of the pad on the disc, which can be seen in Eq. (1).

$$E_b = \frac{mV_1^2}{2} + \frac{I\omega_1^2}{2}, \tag{1}$$

A ride without elevation was considered, which indicates that only kinetic energy is predominant during the movement. However, Eq. (1) can be written by adapting it with respect to the rotating parts of the prototype as a function of wheel rotation, resulting in Eq. (2) (Limpert, 1999).

$$E_b = \frac{KmV_1^2}{2}, \tag{2}$$

where K is the correction factor, obtained by the equation: $K = 1 + \frac{I}{R^2 m}$, R the radius of the wheel, m the mass of the prototype and V_1 the maximum speed of the prototype.

Deriving Eq. (2) as a function of time has the average power, in W, which can be seen in Eq. (3) to (5).

$$P_{bav} = \frac{d(E_b)}{dt}, \tag{3}$$

$$P_b = Km(V_1 - at), \tag{4}$$

$$P_{bav} = Km\frac{V_1}{2}, \tag{5}$$

Thus, the power generated during the movement is converted into thermal power. Considering the vehicle as a control volume, each caliper-disc assembly (Figure 1) stores 1/3 of the thermal energy developed during braking, which can be seen in Figure. 2.

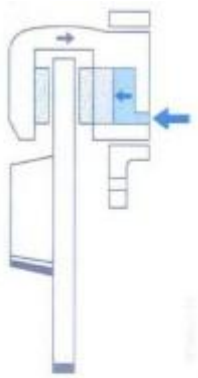


Figure 1. Caliper-Disc Set. Adapted from Brezolin (2007).

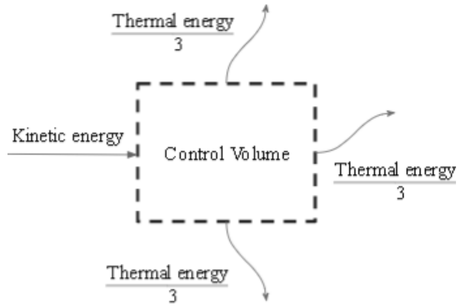


Figure 2. Control volume of the system.

It is worth mentioning that each caliper stores a portion of energy, which can be obtained through Eq. (6) proposed by Limpert (1999).

$$\gamma = \frac{1}{1+\sqrt{\frac{\rho_R c_R k_R}{\rho_P c_P k_P}}}, \quad (6)$$

where ρ_R is the disk density, c_R the disc specific heat, k_R the disc thermal conductivity, ρ_P the pad density, c_P the pad specific heat, and K_P the pad thermal conductivity.

Regarding the material properties of the disc and pad, these are listed in Tables 1 and 2, respectively.

Table 1. Disc material properties. Callister (2002).

Disc material properties	**Gray Cast Iron**	**Stainless Steel**	**SAE 1045 Carbon Steel**
Density, kg/m³	7250	7900	7870
Emissivity	0.21	0.14	0.66
Specific heat, J/kgK	460.24	500	473
Thermal conductivity, W/mK	48	18.75	51.9
Thermal dilatation, m/m°C	10.95 x 10^{-6}	11.90 x 10^{-6}	17.60 x 10^{-6}

Table 2. Pad Properties. Majcherczak et al. (2005).

Pad Properties	**Value**
Density, kg/m³	2500
Specific heat, J/kgK	900
Thermal conductivity, W/mK	12

The prototype has two discs on the front axle and only one disc on the rear axle. Therefore, at the moment of braking, the dynamic load transfer does not occur uniformly due to the dynamic mass distribution of the prototype. This transfer is estimated by knowing the total weight of the vehicle, center of gravity position, wheelbase, and maximum deceleration (Limpert, 1999). Currently, the SB7 prototype has a dynamic load transfer on the front axle equal to 76.2% and on the rear axle equal to 23.8%, with a deceleration equivalent to 1g (9.81 m/s²). A caliper with two pistons is mounted on all discs, and a connection (which may be rigid or flexible) is attached to each one. All these connections have the same matrix: the master cylinder, responsible for pressurizing the fluid in the brake lines, as can be seen in Figure 3.

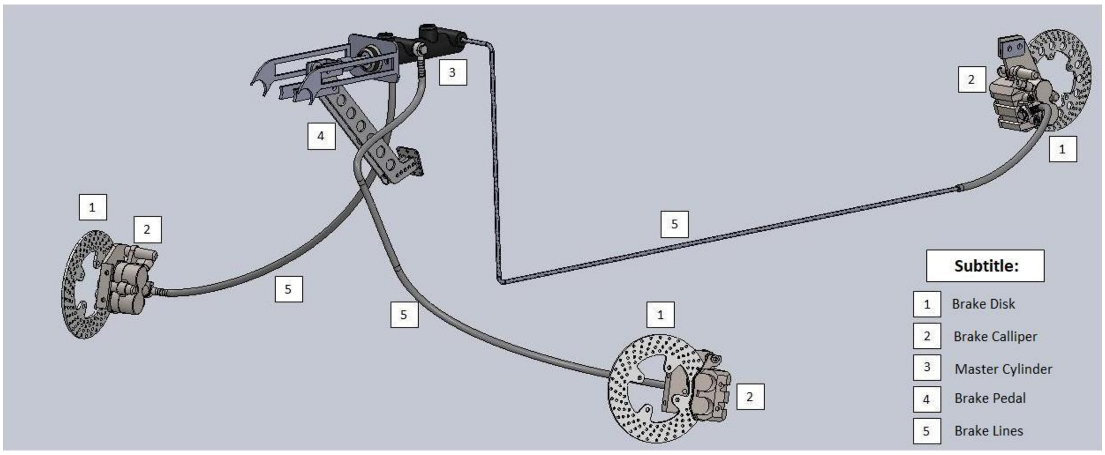


Figure 3. The prototype brake system.

Thus, the thermal power stored in each disc adapted to the prototype conditions becomes that shown in Eq. (7).

$$P_{bav} = \frac{1}{6}\, \gamma . K . m . V_1 . \psi, \tag{7}$$

where ψ is the dynamic load transfer.

It is necessary to determine the temperature of the disc after the first brake, that is, at the moment when the kinetic power is transformed into thermal power. This can be obtained from Eq. (8) and (9).

$$P_{bav} = \dot{Q}, \tag{8}$$

$$\dot{Q} = \frac{\rho_R . \forall . C_R . \Delta T}{\Delta t}, \tag{9}$$

where $\forall$ is the volume of the disc, Δt the total stopping time and ΔT the variation of surface temperature of the disc in the first stop.

2.2 Modeling for successive braking

When it comes to thermal design, brake disc temperature analysis requires knowledge of the total energy absorbed during braking and how it is distributed in the disc. For this it is necessary to determine the heat transfer rates, in W, which can arise through conduction, convection and radiation. However, the most relevant portion is convection, which, according to Ciolfi (2010), increases linearly as the speed increases, and only at higher temperatures radiation becomes considerable.

In convection, two mechanisms are involved: conduction and movement of a fluid, as it is a mode of heat transfer between a solid surface and an adjacent gas or a moving liquid, and can be forced (when an external agent forces the fluid to move) and/or natural (when heat exchanges occur due to gravitational thrust) (Çengel, 2009). In order to verify the relationship between the convection modes, the Grashof number (Gr) given by Eq. (10) is considered.

$$Gr_L = \frac{g\beta(T_S - T_\infty)D^3}{v^2}, \tag{10}$$

where g is the gravity acceleration, β is the air volumetric expansion coefficient, T_S the surface temperature of the disc, T_∞ is the ambient temperature, D is the diameter of the disc and v the cinematic viscosity.

If, according to Çengel (2009), the ratio of Gr by Reynolds (Re) squared is much smaller than 1, the effects of natural convection are negligible; if greater than 1, natural convection dominates; and if equal to 1, the effects of both convection modes are relevant.

The convection transfer rate is expressed by Law of Cooling of Newton, given by the product of the heat transfer coefficient, area, and the temperature difference between of the surface and ambient.

Knowledge of convective heat transfer rate is intertwined with fluid mechanics, which studies the behavior of fluids in motion or at rest and the interaction of solids with fluids. The flow of a fluid can be classified as laminar or turbulent, and to characterize it, there is an admittance parameter known as the Reynolds number (Re), given by Eq. (11), the result of the ratio of the inertial forces to the viscous forces of the fluid. If it is less than 2.4 x 10^5, it is considered laminar flow; if it is greater, it is turbulent (Çengel, 2009).

$$Re = \frac{\rho_{Air} V_{Air} L_c}{\mu_{Air}}, \tag{11}$$

where ρ_{Air} is the air density in kg/m³, V_{Air} the air velocity around the disc in m/s, L_c the characteristic length of the geometry in m, and μ_{Air} the dynamic viscosity in Pa.s.

To determine the type of airflow around the discs, it was necessary to perform an aerodynamic simulation of the prototype, with the aid of SolidWorks, with the input data described in Table 3, resulting in 25.551 km/h and 40.376 km/h for the front and rear axles, respectively. Through the Figure 4, it is possible to note the variations in air speed around the prototype at the front and rear axle disc positions. Given this and considering the air properties at 28.8°C (Table 4) and the disc thickness (4 mm), it was determined that the air flow around the discs is laminar, resulting in a Reynolds number of 60857.449 and 78136.239 for the front and rear axle discs, respectively.

Table 3. Project Considerations.

Parameters	**Value**
Max design speed, km/h	60
Deceleration, g	1
Prototype mass, kg	304

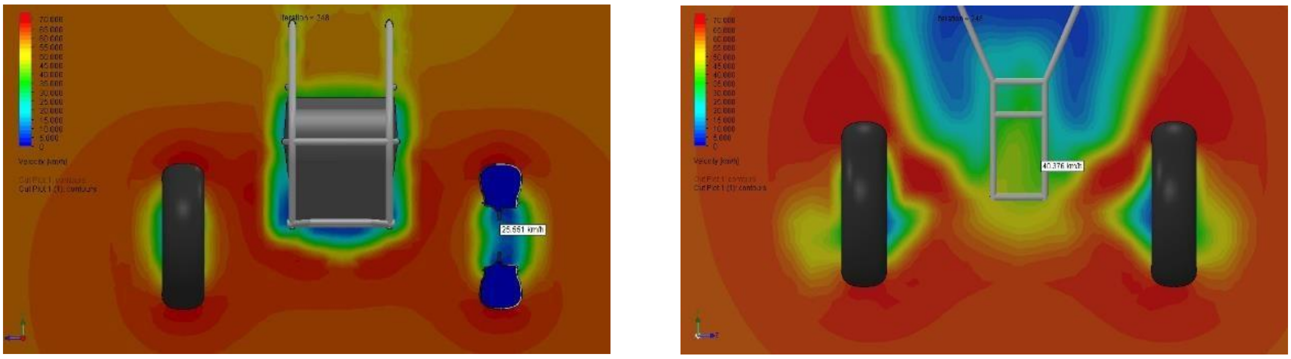


Figure 4. Aerodynamic simulation results for the SB07 prototype.

Table 4. Air properties used in the methodology. Lemmon et al. (2000).

Air Properties	**Value**
Density, kg/m³	1.154
Conductivity, W/m.K	0.02579
Dynamic Viscosity, µPa.s	18.66
Prandtl's Number	0.7271

(1) measured at 28.8°C

It is worth mentioning that there are proposals by different authors to determine the convective coefficient, such as Watel (1997) and Wiesche (2007). In this paper, we will consider the proposal by Çengel (2009), Eq. (12), and Mitschke (1972), Eq. (13).

$$h_1 = 0.664\,\frac{k_{Air}}{D} Re^{0.5} Pr^{1/3}, \qquad 0.6 < \mathrm{Pr} < 60 \tag{12}$$

$$h_2 = 0.7\frac{k_{Air}}{D} Re^{0.55}, \tag{13}$$

where k_{Air} is the air conductivity in W/m.K and Pr the number of Prandtl.

And finally, radiation, the result of changes in the electron configurations of atoms, is defined as the energy emitted in the form of electromagnetic waves by matter (Çengel, 2009). Its maximum rate is determined by the Stefan-Boltzmann

Law, which is defined, based on the black body, the heat flux emitted by it is proportional to the fourth power of the temperature. The radioactive coefficient is shown in Eq. (14).

$$h_{rad} = \frac{\sigma\varepsilon({T_S}^4 - {T_\infty}^4)}{T_S - T_\infty}, \tag{14}$$

where σ is the Stefan-Boltzmann constant, equal to 5.6697 x 10^8 W/m²K^4.

To arrive at the correlation that estimates the temperature gradient in the disc, it is considered as a flat plate, where for low rotations and airflow, the rotation of the disc slightly affects the flow field and the local temperature distribution, while the heat transfer remains constant, the rotation increases heat convection on the moving side (where the rotation speed and airflow are in the same direction) and decreases on the counter-moving side (where the rotation speed and airflow are in opposite directions) making it virtually unchanged (Latour, 2011). In addition, it was taken into account that the temperature varies with time (analysis with transient condition) according to the concentrated systems method. However, to validate the use of the method, it is necessary to know the Biot number, Eq. (15), in which represents the ratio of convection at the body surface and conduction inside the body. It must be less than or equal to 0.1, representing uniform temperature fields inside the body (Çengel, 2009).

$$Bi = \frac{h_i . e_p}{k_R}, \tag{15}$$

where h_i is the convective coefficient and e_p the disc thickness.

As a result of derivations and integrations, it can be observed the Eq. (16) which results in the temperature after several braking cycles:

$$T_n - T_\infty = \Delta T \, \frac{1 - e^{\frac{-n.(h_i + h_{rad}).A_S.t_c}{\rho_R.\mathcal{V}.C_R}}}{1 - e^{\frac{-(h_i + h_{rad}).A_S.t_c}{\rho_R.\mathcal{V}.C_R}}}, \tag{16}$$

where T_n is the temperature in successive braking, n is the number of brakes, A_s the surface area of the disc and t_c the time of cooling (30 seconds).

The temperature variation can result in deformation distribution of thermal origin and, to estimate it, one can consider the Eq. (17).

$$\delta = \alpha . e_p . \Delta T \tag{17}$$

where α is the thermal dilatation of material in m/m°C.

3. RESULTS

The Biot number was analyzed for the validation of the method of concentrated systems. Thus, Table 5 shows that both the correlation of Mitschke (1972) and that of Cengel (2009), all the materials studied presented values lower than 0.1, as this result was expected. The brake disc has a very small thickness (4 mm), which means that its thermal resistance to conduction is insignificant compared to its resistance to convection. Therefore, the method can be used to study the temperature behavior around a brake disc.

Table 5. Number of Biot through the methodology of Çengel (2009) and Mitschke (1972).

	Axle	Gray cast iron	Stainless steel	SAE 1045 Carbon steel
Çengel (2009)	Front disc	0.002	0.005	0.002
	Rear disc	0.003	0.008	0.003
Mitschke (1972)	Front disc	0.004	0.011	0.004
	Rear disc	0.006	0.016	0.006

It is also necessary to analyze the behavior of thermal radiation, as it corresponds to a small portion of cooling. Thus, Table 6 shows the contribution of radiation to the cooling at the end of 100 brakes for the Çengel (2009) and Mitschke (1972) methods.

Table 6. Radiation portion in the disc cooling process by the Çengel (2009) and Mitschke (1972) methods.

	Axle	Gray cast iron	Stainless steel	SAE 1045 Carbon steel
Çengel (2009)	Front disc, %	8.341	5.408	21.116
	Rear disc, %	5.240	3.389	14.415
Mitschke (1972)	Front disc, %	4.297	2.721	12.005
	Rear disc, %	2.588	1.652	7.615

It is observed that for the two methods presented, the carbon steel SAE 1045 presented greater influences of cooling by thermal radiation, which is justified due to its properties. Among the three materials presented, it has a higher emissivity (Table 1), where its surface has a higher heat transfer capacity by thermal radiation. Thus, it is essential to consider this portion through a combined heat transfer coefficient, considering that the portion of contribution in cooling exceeds the 5% reported by Limpert (1975) in tests of solid and ventilated discs. Therefore, it is observed that the difference between the two correlations does not exceed 10% for the coefficient of thermal radiation for both axes analyzed.

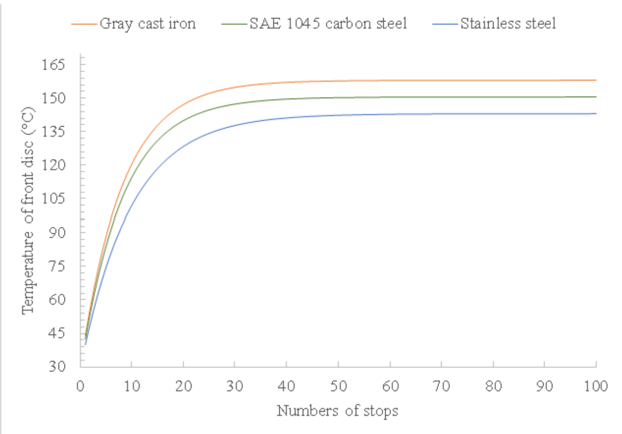
Therefore, it becomes necessary to analyze the influence of natural convection during the cooling process of the disc. Thus, the criterion factor, Eq. (10), was calculated to analyze the influence of natural convection during the 100 brakes.

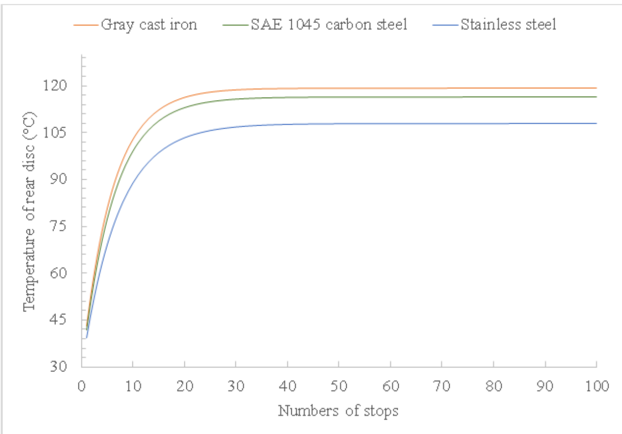
Table 7 shows very small values, as expected, since the air is forced to flow between the discs due to the high speed of the prototype. Therefore, natural convection can be disregarded in the analysis.

Table 7. Summary of results for the natural convection criterion by the correlation proposed by Çengel (2009) and Mitschke (1972).

	Axle	Gray cast iron	Stainless steel	SAE 1045 Carbon steel
Çengel (2009)	Front disc	0.043	0.041	0.041
	Rear disc	0.012	0.012	0.012
Mitschke (1972)	Front disc	0.043	0.041	0.042
	Rear disc	0.012	0.012	0.012

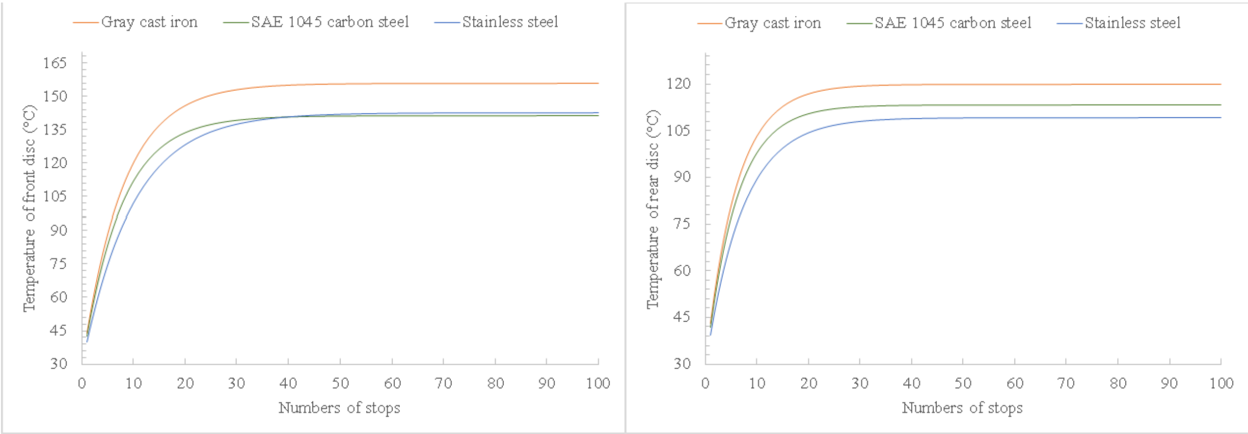
Therefore, using Eq. (16) the maximum brake disc temperature for the 3 materials was calculated and plotted in Figures 5 and 6 for the front and rear discs, respectively.





(a) (b)

Figure 5. Disc temperature variation by Mitschke method (1972) for gray cast iron, SAE 1045 carbon steel and stainless steel, (a) in the front disc (b) and rear disc.



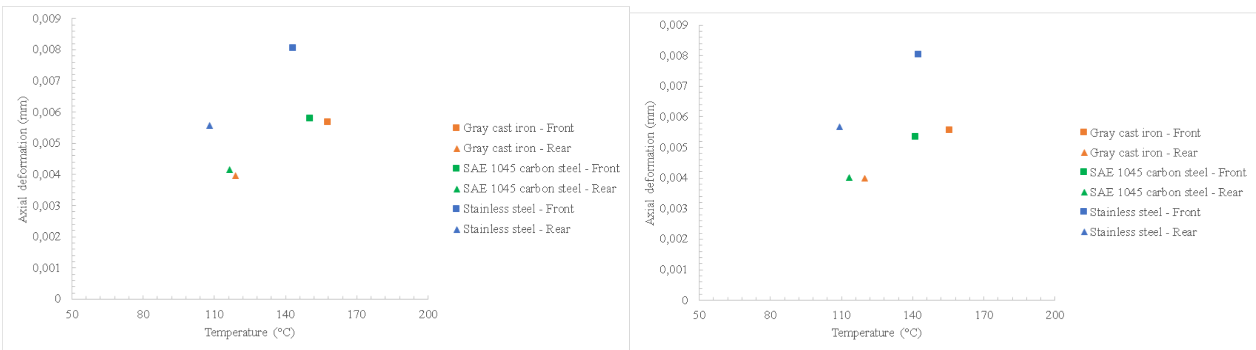
(a) (b)

Figure 6. Disc temperature variation by the method of Çengel (2009) for gray cast iron, SAE 1045 carbon steel and stainless steel, (a) in the front disc (b) and rear disc.

The temperature has an exponential behavior, considering that it varies according to the number of breakings. However, after a specific number of breakings the surface temperature of the disc of the three materials analyzed tends to a maximum value. Thus, from a stabilization point until the 100th brake, the surface temperature approaches a constant value, which is independent of the correlations for the convective coefficient. Thus, given a fixed time for cooling, when $n \to \infty$, the temperature T tends to be constant.

It is observed that for the correlation proposed by Mitschke (1972) in Fig. 5 (a), stainless steel and cast iron presented the lowest and highest temperature gradient, respectively, both in the front and rear axle disc. However, for the front axle disc by the correlation presented by Çengel (2009), the temperature curve of SAE 1045 carbon steel showed a small reduction (9.387 °C), demonstrating that it has a lower temperature gradient than stainless steel, unlike the rear axle, which obtained similar results to Fig.6 (b). Regarding the gray cast iron, it presents a higher temperature in the front and rear axle, corresponding to 141.189°C and 113.143 °C, respectively, by the correlation of Çengel (2009) and 150.576 °C and 116.313 °C, respectively, by the correlation of Mitschke (1972), in which it is justified due to its low density and specific heat when compared to SAE 1045 carbon steel and stainless steel.

According to Eq. (17), one can obtain the values of deformation in the materials analyzed for the different correlations, in which it can observed in Figure 7 through the variation of deformation due to the surface temperature of the disc at the end of 100 breaking cycles.



(a) (b)

Figure 7. Variation of disc deformation for gray cast iron, SAE 1045 carbon steel, and stainless steel, (a) by the correlation of Mitschke (1972) (b) and by the correlation of Çengel (2009).

It can be seen that even though stainless steel has a lower temperature gradient by the correlation of Mitschke (1972) in the front and rear axle (143.124 °C and 107.889 °C, respectively) and by the correlation of Çengel (2009) in the rear axle (109.266 °C), it has a higher thermal deformation for the front axle, equivalent to 0.008 mm. This is justified because stainless steel has a high coefficient of thermal expansion, as can be seen in Table 1. Regarding the two remaining materials, it is noted that both presented values close to the axial strain, but the 1045 carbon steel deformed more than the cast iron by the correlation of Mitschke (1972), Figure 7 (a), and by that of Çengel (2009), the opposite occurred, Figure 7 (b).

Furthermore, the percentage of thermal energy stored by the disc in relation to the brake pad was analyzed. It can be observed in Figure 8 that in relation to the pad, the SAE 1045 carbon steel has a greater capacity to store heat (73%), while stainless steel has a lower capacity to store heat (62%).

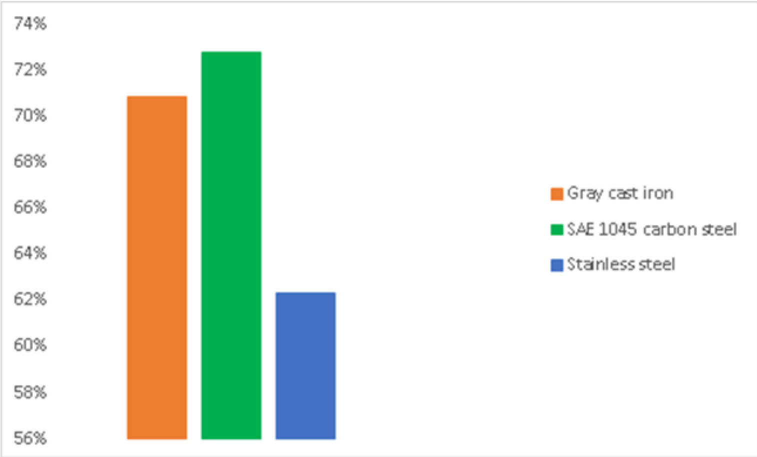


Figure 8. Capacity of the heat storage disc materials.

This is justified because, according to Eq. (6), the product between the density (ρ_R), specific heat (C_R) and thermal conductivity (k_R) in the stainless steel disc is lower when compared to gray cast iron and SAE 1045 carbon steel, thus promoting less heat storage.

4. CONCLUSION

Through the thermal analysis of the disc of the front and rear axles of the SamaBAJA 07 prototype, it is possible to conclude that among the three materials analyzed: stainless steel, gray cast iron, and SAE 1045 steel, the one that presented the best thermal performance with lower temperatures after several braking cycles and 30 seconds of cooling was the 1045 carbon steel. As for the power transmission shafts, the front discs because receive higher dynamic transfer rates, heat up more during deceleration.

5. ACKNOWLEDGEMENTS

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7. RESPONSIBILITY NOTICE

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