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Análisis del caudal de aire en un disco de freno automotriz con perfiles de ventilación tipo N-38

Analysis of the flow air on an automotive brake disc with profiles N-38 type

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***Abstract*— The braking system of a car must meet a complex set of requirements, with safety being the most important. In the design of these systems, it directly influences a correct geometry and an adequate selection of the material for its correct operation. The objective of this work is to propose a new geometric arrangement for the optimization of airflow in an automotive brake disc, taking into account the ventilation pillars based on aerodynamic profiles type N-38. In the validation of this design proposal, a 1:1 scale prototype was built by means of additive manufacturing and thus be able to measure the field of generating speeds in the suction and discharge zone using Particle Image Velocimetry (VIP). The geometric arrangement was carried out under two angular velocity conditions: 541 and 941 rpm. The results showed the optimization of the air velocity in the discharge zone of 0.2121 and 0.743 m/s respectively for the speed ranges; which demonstrates the importance of experimental designs with which the geometry of self-ventilated disc brakes can be improved and thus the efficiency and safety of the system are required.**

***Index Terms*—Disc Brakes, Geometry, Profiles, N Series, Speed.**

***Resumen*—El sistema de frenado de un automóvil debe satisfacer un conjunto complejo de requerimientos, siendo la seguridad lo más importante. En el diseño de estos sistemas, influye directamente una correcta geometría y una selección adecuada del material para garantizar su correcto funcionamiento. El objetivo de este trabajo, es proponer un nuevo arreglo geométrico para la optimización del flujo de aire en un disco de freno automotriz, teniendo en cuenta pilares de ventilación fundamentado en perfiles aerodinámicos tipo N-38. Para validar esta propuesta de diseño, fue construido un prototipo a escala 1:1 por medio de manufactura aditiva y de esta manera poder medir el campo de velocidades generado en la zona de succión y descarga utilizando Velocimetría por Imágenes de Partícula (VIP). El arreglo geométrico se llevó a cabo bajo dos condiciones de velocidad angular: 541 y 941 rpm. Los resultados obtenidos muestran la optimización de la velocidad del aire en la zona de descarga de 0.2121 y 0.743 m/s respectivamente para los rangos de velocidades; con lo que se evidencio la importancia de diseños experimentales con los cuales se pueda mejorar la geometría de los frenos de disco autoventilados y de esta manera garantizar la eficiencia y seguridad del sistema.**

***Palabras claves*—Frenos de Disco, Geometría, Perfiles, Serie N, Velocidad.**

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I. INTRODUCTION

The brake disc system it is undoubtedly the most important component for road safety of the car because on it depends on the total or partial detention of the vehicle, and consequently the integrity of its passengers. Generally, 70% of the kinetic energy produced in the movement is absorbed by front disc brake and the remaining 30% by the rear brake, which is usually drum. During repetitions in the braking process, the kinetic energy is transformed in thermal energy, due to the friction generated between the braking track and the pad where temperatures of up to 900°C are reached according to [1], 90% of the heat is distributed and absorbed the disc brake and the remaining 10% by the pad.

The principle of operation these systems is based on friction to stop the movement of the vehicle, with the hydraulic pressure that pushes the brake pads against the nodular gray cast iron disc [2]. A consequence of this phenomena, considerably high heat is created during braking due to the absorption of kinetic energy, increasing the friction temperature; this heat dissipates rapidly with the environment (surrounding air) through the convection phenomena, which is defined as the transfer of heat that occurs between masses at different temperatures [3]. It is important to mention that environmental factors also play an important role in the heat transfer stage to occur. In addition, when the temperature reaches high values, the phenomena of radiation heat transfer appears, which also helps dissipate energy in the form of heat stored in the disk [4][5][6][7]. Disc brake maintenance is cheaper compared to drum brakes. Therefore, the geometric characteristics of the discs depend on the loading and operating capacity, which is an important factor in the initial design phase. In most cases, designs must avoid overheating that arises between the brake and the pad due to the effect of friction, also taking into account the physical, mechanical and chemical properties of the materials used in the braking system [8][9]. The design phase of ventilated disc brakes, it is very important to analyze the behavior of the associated thermo-fluids (surrounding air) where the characteristics and operation of the fluids on the surface of the disc can be observed always guaranteeing the effectiveness of the process for braking and heat dissipation through the surface and ventilation channels [10][11].

The heat dissipation heat and the performance of the ventilated disc brakes depends largely on the aerodynamic characteristics of the airflow, through ventilation channels and disk brake geometry configurations, which are checked by the implementation of CAD design software that includes the Computational Fluid Dynamics library (CFD) [12][13].

In an investigation conducted by Rivera López, et al. in 2018, proposes a new geometric arrangement for the optimization of airflow in an automotive brake disc, in this proposal, uses NACA type ventilation pillars (4418 and 66-219), with the implementation of performing the speedometer analysis of particles and thus optimized suction conditions [6][14][15].

Taking into account the above, in this investigation, we studied the behavior of air particles in an automotive disc brake with N-38 type ventilation pillars through Particle Image Velocimetry (VIP) evaluating the behaviors and conditions of the elements with the modification of the systems that incur in any area of engineering [16][17].

II. MATERIALS AND METHODS

To make a geometric proposal with classic ventilation channels type N-38, the entry edge must be taken into account; that is, the angle of attack β1 can be modified to allow the flow of the maximum volume of air necessary to prevent the disc from overheating. Figure 1 shows the geometry of the N-38 type ventilation blade. Also, Figure 2 represents the distribution of the profiles on the disk with an angle of β1=60°. For the calculations, the procedure of the geometric outline of the curvature of a radial centrifugal impeller was taken into account, taking into account the method of the Kaplan error triangle, which considers the total development of the ventilation pillar in its entire length, angles and thicknesses in order to make the mathematical considerations of the surrounding flow in the brake disc [18].

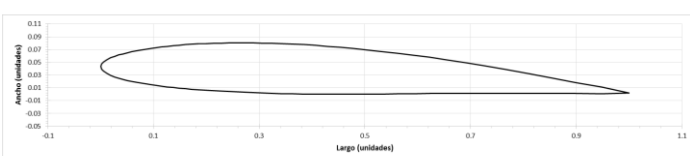


Fig. 1. Classic profile Type N-38.

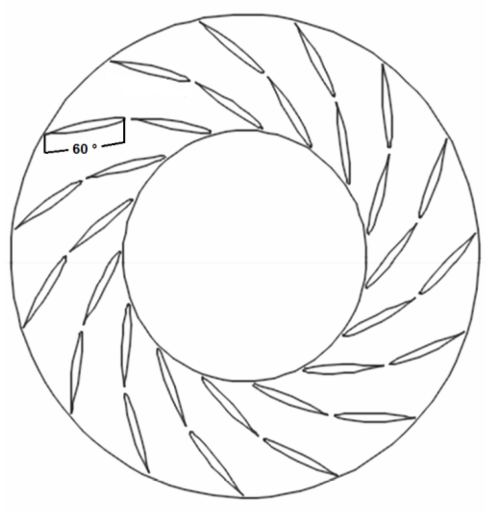


Fig. 2. Configuration of the ventilation profiles.

Taking into account previous work, a configuration of 15 of classic N-38 profile was established. Subsequently, an analysis was carried out in ANSYS where the following Table 1 of results was obtained:

TABLE 1.
RESULTS OF THE FLOW ANALYSIS.

Variable	**541 rpm**	**941 rpm**
Tangencial velocity (m/s)	7.232	12.013
Pressure change (Pa)	120.352	210.526
Mass flow (Kg/s)	3.987	5.294

With the results obtained and the Euler equation for turbomachines, the term could be calculated $\rho\ (U_2V_{t2})$, where U_2 it is the tangential component of the speed at the exit of the disc tracks and V_{t2} is the absolute velocity of the particle at the same point, the density of the air is $\rho = 1.20\ \mathrm{kg/m^3}$ and velocity $U = 2\pi N R_D/60$, where "N" are the revolutions per minute of the disk and "R_D" It is the discharge radius of the brake disc. Similarly, with the results of the pressure change ΔP (Table 1) aerodynamic efficiency was calculated "ψ" for the proposed design, using the following Equation 1:

$$\psi = \frac{\rho\ (U_2V_{t2})}{\Delta P} \qquad (1)$$

Aerodynamic efficiency depends on increases in mass flow, which can be obtained with the following Equation 2:

$$\psi = \psi(\dot{m}) \qquad (2)$$

Finally, it is concluded that the disc with a configuration of 15 profiles is an excellent design option, because it has no blocking effects at low speeds, and in addition, aerodynamics presents and energetically as proposed by [14][19].

III. RESULTS AND DISCUSSIONS

After having determined the correct geometry, the experimental analysis by Velocimetry Particle Images (VIP) was performed, where the following activities were necessary to obtain a physical model of the brake disc:

- Purchase of materials.
- Manufacture of the profiles.
- Prototype disc brake assembly.
- Evaluation of the flow behavior.
- Analysis of results.

The analysis was performed using water as an operating fluid to facilitate the monitoring of the particles as they pass through the inside of the disk and thus observe their behavior from the time they enter the suction zone until the exit. For this, it is necessary to make an equivalence between the characteristics of the water and the air, because the latter belongs to the real conditions to which the disc is to be operated. The above was done in order to find an equivalence between the speeds of the disk in air and those used in the test of the disk in water assuming that the Reynolds Number for both cases (water and air) must be the same, for what the following equations 3 and 4 were used:

$$Re_{water} = \frac{V_{water} * \rho_{water}}{\mu_{water}} \qquad (3)$$

$$Re_{air} = \frac{V_{air} * \rho_{air}}{\mu_{air}} \qquad (4)$$

Where:

Re = Reynolds number
V = Disk rotation speed (rpm)
ρ = Density (kg/m^3)
μ = Dynamic viscosity

When matching the equations 3 and 4, the variable is cleared V_{water}, getting the equation 5:

$$\frac{V_{water} * \rho_{water}}{\mu_{water}} = \frac{V_{air} * \rho_{air}}{\mu_{air}}$$

$$V_{water} = \frac{(V_{air})(\rho_{air})(\mu_{water})}{(\rho_{water})(\mu_{air})}$$

$$V_{water} = \left(\frac{\rho_{air}}{\rho_{water}}\right)\left(\frac{\mu_{water}}{\mu_{air}}\right) * V_{air} \qquad (5)$$

Data for air and water conditions are extracted from Cengel [20], to room temperature of 22°C.

$\rho_{air} = 1.196\ kg/m^3$
$\mu_{air} = 1.82 \times 10^{-5}\ Pa/s$
$\rho_{water} = 997.6\ kg/m^3$
$\mu_{water} = 9.684 \times 10^{-4}\ Pa/s$

The values of the disk rotation speed (boundary conditions) are 541 and 941 rpm, equivalent to the vehicle's linear speeds to 60 and 103 km/h. To calculate the rpm of the disc from the speed of the car, an average value of the wheel radius is assumed 0.29 m, so the conversion factor of km/h to rpm is as follows:

$$1\frac{km}{h}\left(\frac{1000\ m}{1\ km}\right)\left(\frac{1\ h}{60\ min}\right)\left(\frac{1}{0.29\ m}\right)\left(\frac{1\ rev}{2\pi\ rad}\right) = 9.147$$

$$1\frac{km}{h} = \frac{rpm}{9.147}$$

The check for the rpm mentioned above is performed as follows:

$$\frac{541\ rpm}{9.147} = 59.2 \approx 60\frac{km}{h}$$

$$\frac{841\ rpm}{9.147} = 102.87 \approx 103\frac{km}{h}$$

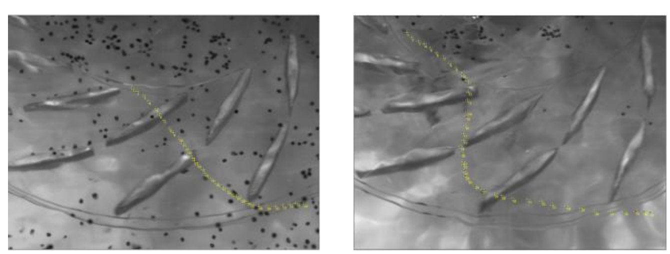
After having the rpm of a disk in an air test, the speeds for the water test are calculated and in this way, the equivalence between data is obtained using the equation 5:

$$V_{water} = 0{,}064 * V_{air}$$

$$V_{water} = 0.064 * 541 = 35\ m/s$$
$$V_{water} = 0.064 * 941 = 60\ m/s$$

The speeds for the water test equivalent to the air test are 35 and 60 rpm, respectively. For greater relevance of the data, the test was carried out in two ways: the particles (ABS) were deposited in the suction zone of the disc and after that, the drill movement began. This movement was described as "submerged". The second way was to first operate the drill to start the rotary movement and after a few seconds, the particles were thrown. This movement was called "in rotation".

Figure 3, the tracking of a particle and its total trajectory while crossing the blade area for each speed described above is detailed.



a) b)

Fig. 3. Path of the particles at different speeds a) Total trajectory 35 rpm in rotation. b) Total trajectory 60 rpm in rotation.

In the previous figure, the general movement of the particle as it passes through the inside of the tracks of the disc is shown. Each point shown in the previous sequence of images is represented in this image so that the general behavior can be appreciated. It should be noted that the more vertical the movement, the greater the speed of the particle.

Then, we processed the images in ImageJ, a table was obtained with the coordinates of each particle along its path through the brake disc. The coordinates were given by the software in pixels and subsequently converted to millimeters. The conversion is obtained from the measurement of a blade in pixels and its actual measurement, and in this way the conversion factor is reached. The measurement of the blade in pixels is 381.21 pixels and 59.24 mm; these values give us a relationship of 0.156mm for each pixel. Having the particle positions in millimeters its speed was determined by calculating the displacement and time elapsed between each image, from which the following Figure was obtained 4:

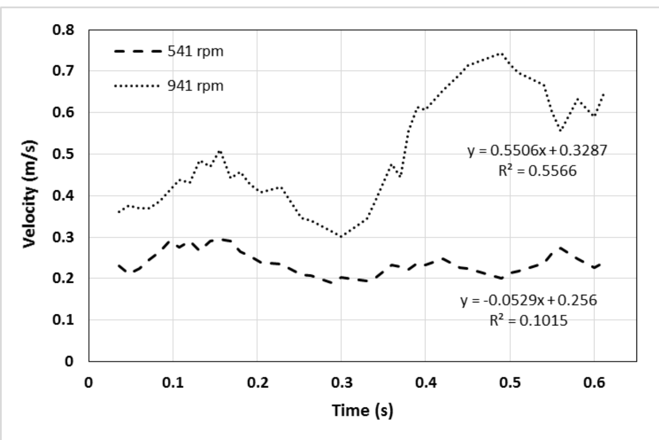


Fig. 4. Experimental radial speed at 541 and 941 rpm in rotation.

The previous figure represents the behavior of the particles along their path through the ventilation channels in the disc brake, so it can be assumed that as the brake speed increases, the particle suction increases of air and, therefore, disc change be much faster in terms of theories of aerodynamics and heat transfer.

IV. Conclusions

The initial speed for each test, submerged and rotating, shows difference between each rpm respectively. In the case of 35 rpm equivalent to a car speed of 60 km/h, in the test starts with a speed of 0.2121 m/s and finish with 0.2958 m/s, speed that remains constant during the course of the test. Subsequently, in the case of 60 rpm equivalent to a car speed of 103 km/h, the test starts with a speed of 0.3758 m/s and finish with 0.743 m/s. From the above, it is obtained that the behavior of the disc brakes for the speed range with this type of ventilation, will be within the behavior obtained experimentally in the test.

As the speed of the car increases the suction is greater, that is why the test time is generally shorter for each test at higher speed. This accelerated flow compensates the amount of heat produced at the time of braking because there is a greater amount of energy to reduce the vehicle.

Finally, for a geometric configuration with a greater number of profiles, the blockage in the air flow can be presented at low rotation speeds, this is largely due to the reduction of the effective surface area of the brake disc; that is to say, the amount of profilespi directly damages the air circulation.

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