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Definition of a Subsonic Wind Tunnel Exhauster for Automotive Tests

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***Abstract:** It is a synthesis of the aspects related to the determination of the propulsive element of a subsonic wind tunnel, from the project developed by Salvador, M. W.; Guimarães, N. C. F.; Saldanha, W.A.R. (2018). The designed wind tunnel has a section of tests of 2.25 m² of cross section and 5 m of length, where any model that does not exceed 20% of block of that cross-sectional area can be tested. The maximum velocity reached in the test section will be 50 m / s, with the speed of 37.333 m / s determined for the proposed test so that the rules of similarity between model and prototype are respected. The detailed calculations of the physical and geometric design of the tunnel, and the pressure drops, by components and total, were presented in Salvador, M. W. et al. (2019). The scale of the model determined for the test is 3/8. This definition of scale and speed was made taking into account the feasibility of the dimensions of the tunnel and its propellant. From the energy requirements, according to the pressure drops in each component of the circuit, the choice of the propulsion element for the tunnel is established, so as to guarantee the adequate flow conditions for the proposed test, with an efficiency admissible.*
Keywords: Subsonic Wind Tunnel; Vehicle Aerodynamics; Exhauster; Prototype; Model.

I. INTRODUCTION

One of the major challenges of engineering is to obtain, with precision, a modeling that describes the behavior of physical systems WHITE (2007). In some cases, good modeling can lead to admissible solutions without the need for experimental studies. For this purpose, there are several platforms and methods for modeling and computational analysis, such as:

Finite Element Method – FEM Shi, T. M. (1984), Computational Fluid Dynamics – CFD Thannehill et al. (1997), and various software and programming languages. However, most of the time the purely analytical and computational approach is not as accurate and does not always generate enough information for decision making in complex systems projects, although it has the advantage of being fast and inexpensive. In these cases, the results obtained are conditioned to the accuracy of the hypotheses, approximations and idealizations made in the analysis.

Despite the progress of mathematical and computational methods, it is still necessary to carry out experimental tests to obtain conclusive information about physical systems (ÇENGEL; CIMBALA, 2006). However, according to the same authors, such an approach is expensive, time-consuming and often impractical.

It is in this context that at the end of the 19th century wind tunnels, carefully designed equipment, began to be used for the purpose of carrying out tests on objects subjected to the action of a flow of air generated by a propellant element.

Tests of aerodynamics in wind tunnels is of extreme importance in the development of new and more efficient vehicle models. However, they are very expensive tests when applied in real-scale models, since they require expenditures with many resources, such as: functional prototype, running time, fuel, pilot, component wear, etc. and are at risk of inefficiency, which would result in more modifications and would require more time in the execution of the project. In addition, the computational resources do not yet present a great reliability in their results, which can also hinder the development process of the vehicle.

A feasible alternative in many physical situations is to replicate the real phenomena in small-scale simulations. In this way it is possible to complement the analysis with experimental results.

In fact, through the use of a wind tunnel developed for vehicular aerodynamic tests, it is possible to integrate these two universes, physical and computational, and to present reliable results, with less expenditure and in the time allowed (HUCHO and SOVRAN, 1993).

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II. OBJECTIVES

Because it is an extremely complex equipment, the present work addresses only the constructive characteristics of the components of a wind tunnel (Figure 01).

The main objective is to determine the curve of the system and to present the parameters of power and flow required by the propulsion system in order to carry out tests with considerable precision of a small-scale automotive model submitted to real pre-defined.

III. METHODOLOGY

For each wind tunnel model rules are adopted that define geometric characteristics according to their purpose Barlow, Rae and Pope (1999). However, for the type of tunnel being addressed, we can generically divide the components into 5 main parts: The contraction nozzle, the test section, the diffuser (or diffusers), the stabilization chamber and the propulsion system.

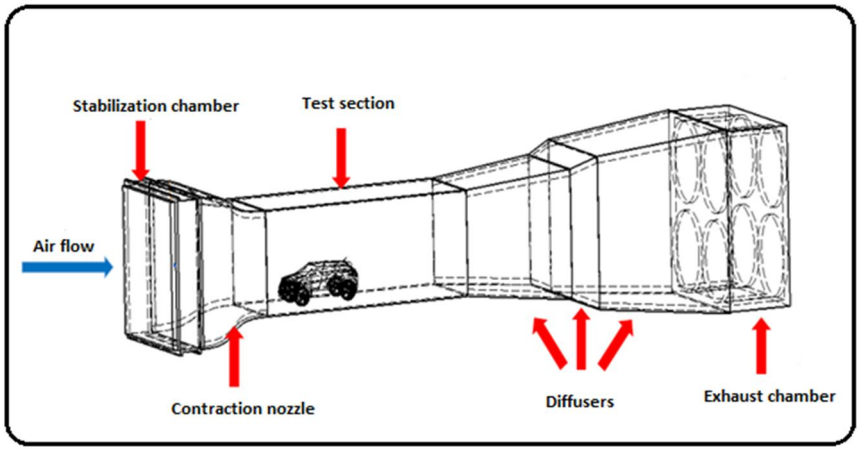


Figure 01: Illustration of the proposed wind tunnel and its components
Source: Salvador, M. W. et al. (2019)

In addition to the above-defined components, there is a variety of apparatuses which can be incremented to the wind tunnel according to the intended use. For example, there are tunnels equipped with air conditioning system, turbulence modelers, vortex generators, roughness simulators, platforms on sloping surfaces, flow brokers, etc.

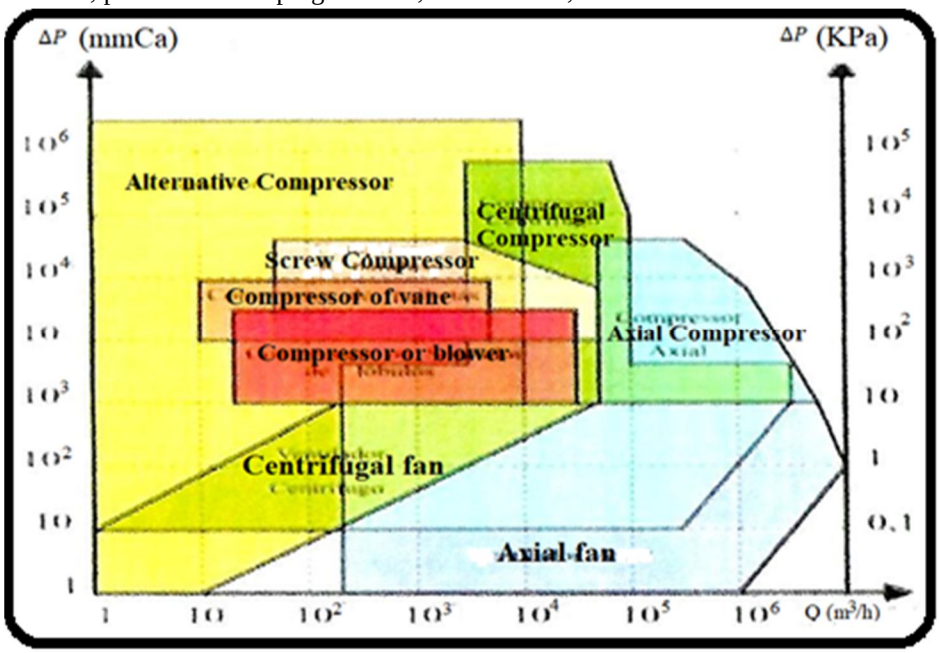


Figure 02: Field of application of fans and compressors
Source: HENN (2006)

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According to Barlow, Rae and Pope (1999), each part of a wind tunnel has specific construction criteria, however the most particular element is the test chamber which must have the proper shape, suitable material, good visibility and space sufficient for a good allocation of the model, so that there is no interference in the flow.

The propulsion system, the main object of this work, is the element responsible for generating the airflow inside the tunnel. It must be dimensioned with the power required to overcome the pressure drop caused by the tunnel components, and maintain desired flow conditions in the test section.

According to Pereira (2011), it is recommended that the ratio between the cross-sectional area of the fan Af and that of the test section AS is between 2 and 3:

$$2 < \frac{A_f}{A_s} > 3 \qquad (01)$$

It is important to identify which flow machine is suitable for the design mass flow rate specified for the tunnel. According to Henn (2006), we can use the figure below to identify the type of flow machine that meets the project parameters.

The choice of the appropriate propulsion system, except for special cases in which the project requires very specific characteristics, and there are no commercial options, is made by crossing the characteristics of the circuit (pressure drop in function of the flow) with the data of the catalog offered by manufacturer.

The propulsion system is usually the last element to be dimensioned, since it is necessary to first obtain the pressure drop values of each element of the circuit to estimate the power required by the machine. The power Pe in Watt can be estimated from the flow Q in m3 / s and the energy required per unit of mass H in J / Kg, per:

$$H = \frac{\Delta P}{\rho} \qquad (02)$$

$$P_e = \rho Q H \qquad (03)$$

A. Pressure Drop and Energy Efficiency

In each of the elements, except for the propeller or fan, there is a loss of energy to be considered in determining the exhaust fan. In fact, there is a transformation of energy mechanically to heat which results in raising the temperature of the fluid and the solids with which it is in contact. The energy transformation occurs due to the viscous action between the fluid and the solid limits. They refer to this transformation of mechanical energy as "loss":

According to Pereira (2011), the loss related to decreasing pressure in a component i can be obtained by the following expression:

$$\Delta P_i = 0{,}5\rho_i C_i^2 K_i \qquad (04)$$

Where Ci is the mean velocity in the input section of component i and Ki is the dimensionless coefficient of pressure drop of component i.

However, due to the particular geometry of each component, the loss coefficient K_i is calculated with distinct expressions for each of them. Similarly, physical and geometric terms such as the hydraulic diameter DH, Reynolds number Re, friction factor f, assume particular values for each of the components.

According to Barlow, Rae and Pope (1999), input variables, such as speed and dynamic pressure, also vary from one component to another, with the output velocity of the previous component being adopted as the input velocity of the component in question. In the stabilizing elements, the term output speed does not apply, the input speed of the latter being the same as the output of the previous component.

Although each stabilizing element has its own load loss quantification, it is also necessary to consider the load loss of the length contemplated by the stabilization chamber, treating it as a constant area section similar to the test chamber.

The relationship between the power of the flow through the test section and the dissipated power, associated with the loss of load in the circuit, defines the energy efficiency (Er) of a wind tunnel, although in no way does this define the functionality of the tunnel to Research and Development. Other quantities can be used to quantify a coefficient of energy efficiency, such as: electric input power in the motor or mechanical power applied to the exhaust axis (HENN, 2006).

The following relationship by Barlow, Rae and Pope (1999) focuses on aerodynamic aspects of energy consumption and clearly separates the flow properties of the exhaust efficiency circuit and other elements.

Denoting the pressure drop as P, we have the energy efficiency measured as:

$$Er = \frac{Pt}{Pc} \qquad (05)$$

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1) *The Pressure Drop In The Stabilization Chamber:* The pressure drop in the stabilization chamber is given by the sum of the losses of loads of the beehive, the screens and the constant section that is between the two. Therefore, to find the loss one must first determine the coefficient of loss K of each of them.

For the beehive: According to Messina (2011), there are three important parameters for the determination of the pressure drop in the hive: Hydraulic diameter of cell Dh, porosity βh (equation 06), and Reynolds number relative to roughness of the hive material (Re_Δ). For the calculation of the coefficient and pressure drop of the hive Kh we must take into account some variables:

$$K_h = \left(\lambda_h \frac{L_h}{D_h} + 3\right)\left(\frac{1}{\beta_h}\right)^2 + \left(\frac{1}{\beta_h} - 1\right)^2 \quad (06)$$

$$\lambda_{h=} \begin{cases} 0{,}375\left(\frac{\Delta}{D_h}\right)^{0,4} Re_\Delta{}^{-0,1} & Re_\Delta \leq 275 \\ 0{,}214\left(\frac{\Delta}{D_h}\right)^{0,4} & Re_\Delta > 275 \end{cases} \quad (07)$$

Where L_h is the cell length of the beehive, and can be found in tables according to the vendor specification.

For the mesh: Eckert, Mort and Jope (1976) proposed an empirical relation for the screen loss coefficient based on three main parameters: porosity or solidity, the Reynolds number Re_w calculated with wire diameter d_w, and the K_{mesh} factor.

Chik (1966) attributes some values for the mesh factor, being 1.0 for new metallic wires, 1.3 for circular metallic wires and 2.1 for silk fibers. An average value of 1.3 for K_{mesh} is a good choice in most cases. Eckert's empirical equation for calculating the coefficient of pressure drop of the mesh K_m is:

$$K_m = K_{mesh} K_{Rn} \sigma_s + \frac{\sigma_s{}^2}{\beta_s{}^2} \quad (08)$$

Onde:

$$K_{Rn} = \begin{cases} 0{,}785\left(1 - \frac{Re_w}{354}\right) + 1{,}01 & se \;\; 0 \leq Re_w < 400 \\ 1.0 & se \;\; Re_w \geq 400 \end{cases} \quad (09)$$

The porosity value of the screen β_s is found by:

$$\beta_s = \left(1 - \frac{n_w d_w}{l}\right) \quad (10)$$

and the solidity of the screen σ_s per:

$$\sigma_s = 1 - \beta_s \quad (11)$$

Where l is the length of the wire or wire of the wire, n_w is the generic number of wire or wire of the screen and d_w is the diameter of the wire or wire of the screen.

For constant sections, we have the loss factor associated with the constant section that is between the beehive and the screens. The coefficient of loss K_l in a constant section can be found by:

$$K_l = f\frac{L}{D_h} \quad (12)$$

Where L is the length and D_h section hydraulic diameter. To find the friction factor, the universal law for smooth pipes in high Reynolds numbers is used, according to Prandtl:

$$\frac{1}{\sqrt{f}} = 2\log\left(Re\sqrt{f}\right) - 0{,}8 \quad (13)$$

2) *The Pressure Drop In The Contraction Nozzle:* The pressure drop in the contraction nozzle is considerable due to the abrupt variation of the cross section. According to Messina, Brusca and Lanzafame (2011), we can calculate the coefficient of loss of charge in the nozzle K_{nt} by the expression:

$$K_{nt} = 0{,}32 f_{av} \frac{L_n}{D_{ts}} \quad (14)$$

L_n is the nozzle length and D_{ts} is the hydraulic diameter of the nozzle seating chamber and the mean friction factor f_{av} can be found by Equation (13).

3) *Coefficient Of Pressure Drop In The Test Section:* The test chamber is a constant section, so it can have its load loss coefficient K_t calculated by:

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$$K_t = f\frac{L}{D_h} \quad (15)$$

$$\frac{1}{\sqrt{f}} = 2\log\left(Re\sqrt{f}\right) - 0{,}8 \quad (16)$$

4) *Coefficient Of Pressure Drop In Diffusers:* According to Barlow, Rae, Pope (1999), fluid behavior in diffusers is quite complex because the flow may depend on the details of the inlet flow profiles, which in turn will vary with the specific configurations of the test model in a wind tunnel.

The main parameters are the conical expansion angle ϑ_e and the relation between the inlet and outlet areas of the cross section $A_r = \frac{A_2}{A_1}$. The coefficient of loss K_{dif} in the diffusers can be determined by the sum of the coefficient of friction $K_f$with the coefficient of expansion K_{exp} (PEREIRA, 2011).

$$K_{dif} = K_f + K_{exp} \quad (17)$$

$$K_f = \left(1 - \frac{1}{A^2{}_{rel\ Dif}}\right)\frac{f}{8\,sen\vartheta_e} \quad (18)$$

$$K_{exp} = k_e(\vartheta_e)\left(\frac{A_{rel\ Dif} - 1}{A_{rel\ Dif}}\right)^2 \quad (19)$$

The determination of diffuser loss is very complex, thus, depending on the shape of the cross section and the equivalent cone angle of the section. There are some numerical approximation functions for calculating k_e that are particular to the geometry of the section used in the diffuser design (ECKERT; MORT; JOPE, 1976). According to Pereira (2011), the lower the conical equivalent expansion angle, the lower the loss.

The following functions refer to the circular and square cross-sections:

$$K_{eCircular}: \begin{cases} A_1 + B_1\vartheta_e & se\ 0 < \vartheta_e < 1{,}5° \\ A_2 + B_2\vartheta_e + C_2{\vartheta_e}^2 + D_2{\vartheta_e}^3 + E_2{\vartheta_e}^4 + F_2{\vartheta_e}^5 + G_2{\vartheta_e}^6 & se\ 1{,}5 \le \vartheta_e \ge 5° \\ A_3 + B_3\vartheta_e & se\ \vartheta_e > 5° \end{cases} \quad (20)$$

$$K_{eQuadrado}: \begin{cases} A_1 + B_1\vartheta_e & se\ 0 < \vartheta_e < 1{,}5° \\ A_2 - B_2\vartheta_e + C_2{\vartheta_e}^2 - D_2{\vartheta_e}^3 + E_2{\vartheta_e}^4 - F_2{\vartheta_e}^5 - G_2{\vartheta_e}^6 & se\ 1{,}5 \le \vartheta_e \ge 5° \\ A_3 + B_3\vartheta_e & se\ \vartheta_e > 5° \end{cases} \quad (21)$$

Where the variables A, B, C, D, E, F and G are proposed in the Eckert table below, according to the geometry of its section:

Table 01: Parameters of Eckert

Parameter	Circular	Square
A_1	0.1033	0.09623
B_1	-0.02389	-0.004152
A_2	0.1709	0.1222
B_2	-0.1170	0.04590
C_2	0.03260	0.02203
D_2	0.001078	0.003269
E_2	-0.0009076	-0.0006145
F_2	-0.00001331	-0.0000280
G_2	0.0001345	0.00002337
A_3	-0.09661	-0.01322
B_3	0.04672	0.05866

Source: PEREIRA (2011)

5) *Total Pressure Drop:* The total pressure drop ΔP_{total} total can be calculated by summing the pressure drop of all tunnel components ΔP_{comp}, according to the equation below:

$$\Delta P_{total} = \sum \Delta P_{comp} \quad (22)$$

In Table 2 we have the total pressure drop and the percentage relative to the pressure drop of each of the components of the tuned wind tunnel, and in table 15 the actual pressure drop of the system.

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The total system pressure drop $\Delta P_{tot;sistema}$ can be found by adding the total pressure loss to the pressure recovery ΔP_{rec}. The pressure recovery ΔP_{rec} is the loss occurring due to the pressure variation at the outlet of the diffuser 2 and can be represented as the dynamic pressure at the outlet of the diffuser 2 Pd_{dif2}:

$$\Delta P_{rec} = Pd_{dif2} \quad (23)$$

The dynamic pressure Pd is found using a variation of equation 7, where the terms not considerable in the current analysis were excluded:

$$Pd = \frac{1}{2}\rho V^2 \quad (24)$$

Where the velocity V is the exit velocity of the diffuser 2, C_{sdif2}:

$$Pd_{dif2} = \frac{1}{2}\rho {C_{sdif2}}^2 \quad (25)$$

$$Pd_{dif2} = \frac{1}{2}.1{,}225.6{,}223^2 \quad (26)$$

$$Pd_{dif2} = \Delta P_{rec} = 23{,}4117\ Pa \quad (27)$$

Table 02: Total pressure drop

	(Pa)	%
INLET NOZZLE	0,3115	0,25%
TEST SECTION	26,3231	20,84%
DIFFUSER 1	11,2984	8,95%
OPEN-ANGLE DIFFUSER	0,4637	0,37%
DIFFUSER 2	0,5759	0,16%
SCREENS	81,2823	64,36%
BEEHIVE	5,9807	4,74%
STABILIZATION CHAMBER	0,0518	0,01%
TOTAL PRESSURE DROP	126,290	Pa

Source: Authors (2018)

Table 03: Synthesis of total losses in the dimensioned wind tunnel

	Pa	mmH2O
Total Pressure Drop	126,2903	12,8780
Pressure Recovery	23,4117	2,3873
Real Pressure Drop	149,70	15,2654

Source: Authors (2018)

The above information is of great importance for the critical analysis of each of the components. The actual head loss of the wind tunnel is associated with the velocity, defined a priori, of 37.33 m / s in the test section and a Q air flow rate of 83.9925 m³ / s.

B. Exhaust Definition

1) *Characteristic Curve of the System:* Designing an equipment that has as initial operating limit, as stated above, is not appropriate and correct. If the propeller element is designed for a flow corresponding to the maximum request, it will spend most of the time working out of its maximum efficiency. This would imply a loss of efficiency, as well as a possible reduction in the life of the equipment due to the operation overloads. For determining the operating point of the flow generator in an installation, in addition to the knowledge of the energy that the machine will be able to supply, it is indispensable to know what energy will be required by the system where the machine is installed to emphasize a certain fluid flow rate considered (HENN, 2006). It is essential, for an adequate exhaust fan design, to calculate the system curve, and for this it was necessary to modify the initial flow to arbitrary values to find the loss of charge in the system for each one of the flows of the fluid, within a range of operation workable. The ideal is to obtain an equipment in which its characteristic curve touches several points of operation of the system curve, according to the example of Figure 03 below:

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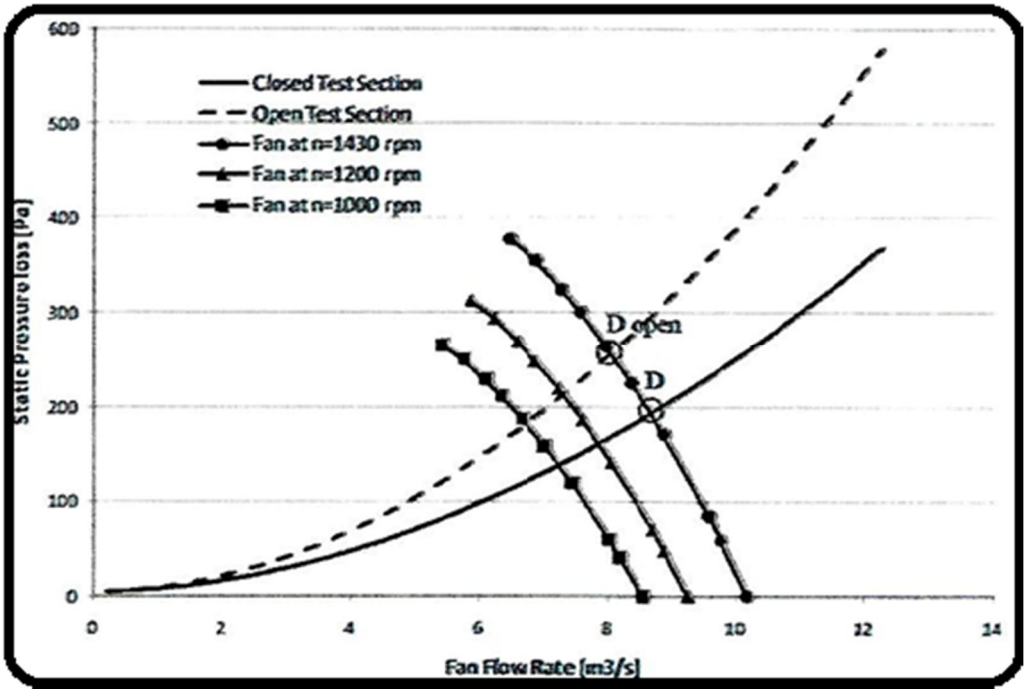


Figure O3: Example of System Curve x Exhaust Curve
Source: PEREIRA (2011)

It is possible to compare the operating curve of the equipment, provided in the manufacturer's catalog, with the characteristic curve of the system. According to Henn (2006), since the generator-flow machine cannot function outside its characteristic curve and that, in order to displace a certain fluid flow rate, it must satisfy the energy requirement indicated by the characteristic curve of the system, the intersection must be at cross of these two curves.

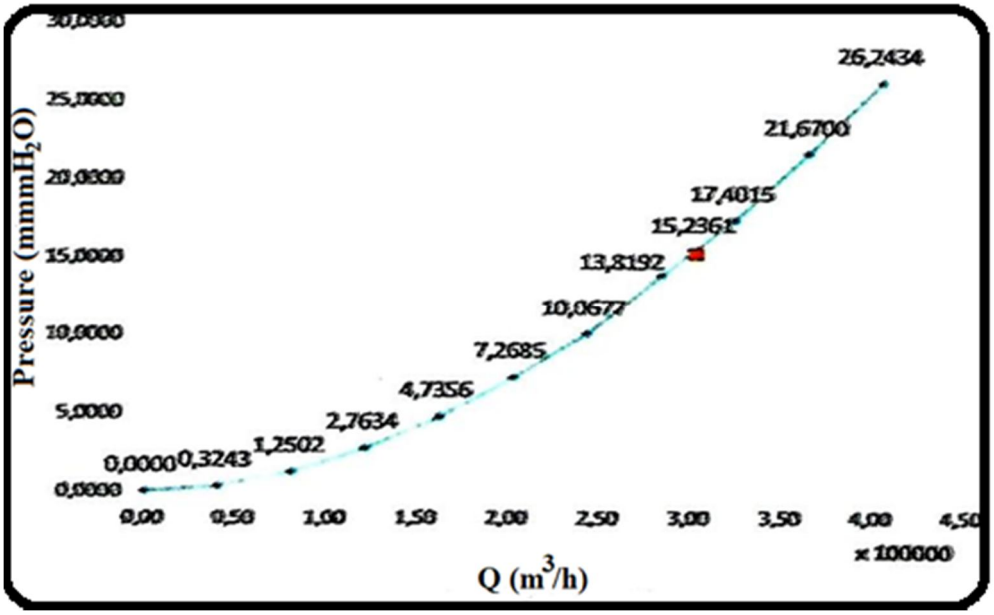


Figure 04: System Curve
Source: Authors (2018)

The design condition was indicated by the point highlighted in Figure 04 above, with a fluid flow rate of 83.9925 m³ / s and $\Delta P_{tot;sistema}$ of 149.70 Pa.

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2) Exhaust Selection

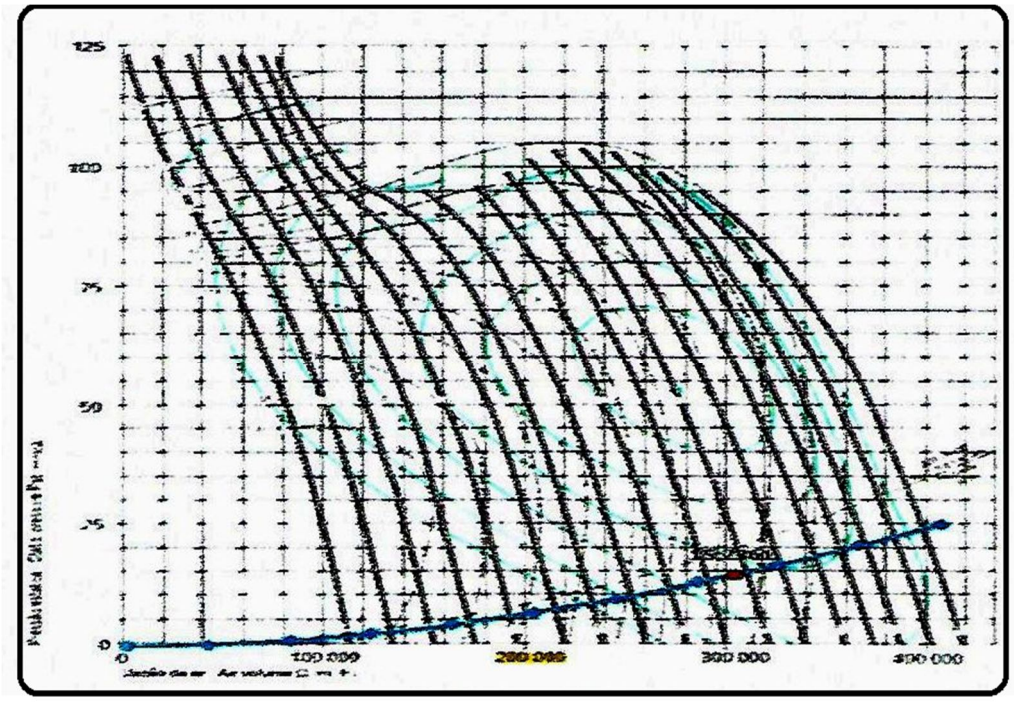


Figura 05: System Curve x Exhaust Curve
Source: Authors (2018)

With the characteristic curve of the system, it was possible to search for equipment that meets the operating point by consulting manufacturers' catalogs.
By consulting the specifications and comparing the exhaust curve to the characteristic curve of the system, the equipment that would best meet the proposed conditions would be the model hood AVR-AL 2000/12 Arr.4 Cl. I HM-H - 850 rpm (Figure 05).
This choice will result in an efficiency of approximately 70% for the given operating point.
The required power to the rotor axis of the exhaust fan that will drive the propellers can be obtained by Equation 28.

$$P_e = Q.\Delta P_{tot,sistema} = \frac{83{,}9925 m^3}{s} 149{,}70\, Pa = 12{,}550\, KW \quad (28)$$

The motor must also supply the required power, which is the actual power a fan requires to move a given volume of air at a given pressure.

$$P_{req} = \frac{P_e}{\eta_f} = \frac{12550}{0{,}70} 17{,}928\, KW\ ou\ 24\, HP \quad (29)$$

The engine used in the propulsion element shall have a power of approximately 24 HP.

IV. DISCUSSION

The synthesis of the dimensioning of a subsonic wind tunnel, of the air suction type, with the closed test section, was carried out, with the purpose of performing tests on small scale models of cars and generating important data for the study of its aerodynamics.
The work presents a viable solution for the demand of aerodynamic tests in automotive models. Field research has shown the great difficulty in meeting real-scale test parameters in wind tunnels, due to the high cost involved in the construction and operation of such equipment, which led to the choice of method of a model wind-tunnel model. Thus, the designed wind tunnel assumes the generic dimensions and functional characteristics presented through Table 04.
The work was limited to the synthesis of the geometric dimensioning of the components, and in the determination of the energy requirements of the propulsion system in order that the design conditions were met.

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Table 04: Tunnel Features

Total length	**15,83 m**
Overall width	3,7 m
Total Width (without stand)	3,7 m
Size of the test section	2,25 m²
Recommended test speed	37,33 m/s
Maximum allowed test speed	50 m/s
Required Power	~24 HP

Source: Authors (2018)

The power of the tunnel propellant system, exhaust fan, the main object of the present work, is approximately 24 HP.

V. CONCLUSION

The work presents a feasible solution for the subsonic wind tunnel, which can be used for testing on any models that meet the prerequisite of occupying less than 20% of the cross-sectional area of the test section, which will be 2.25 m². In any essay, the concepts of similarity, geometric and dynamic, between model and prototype must be taken into account.

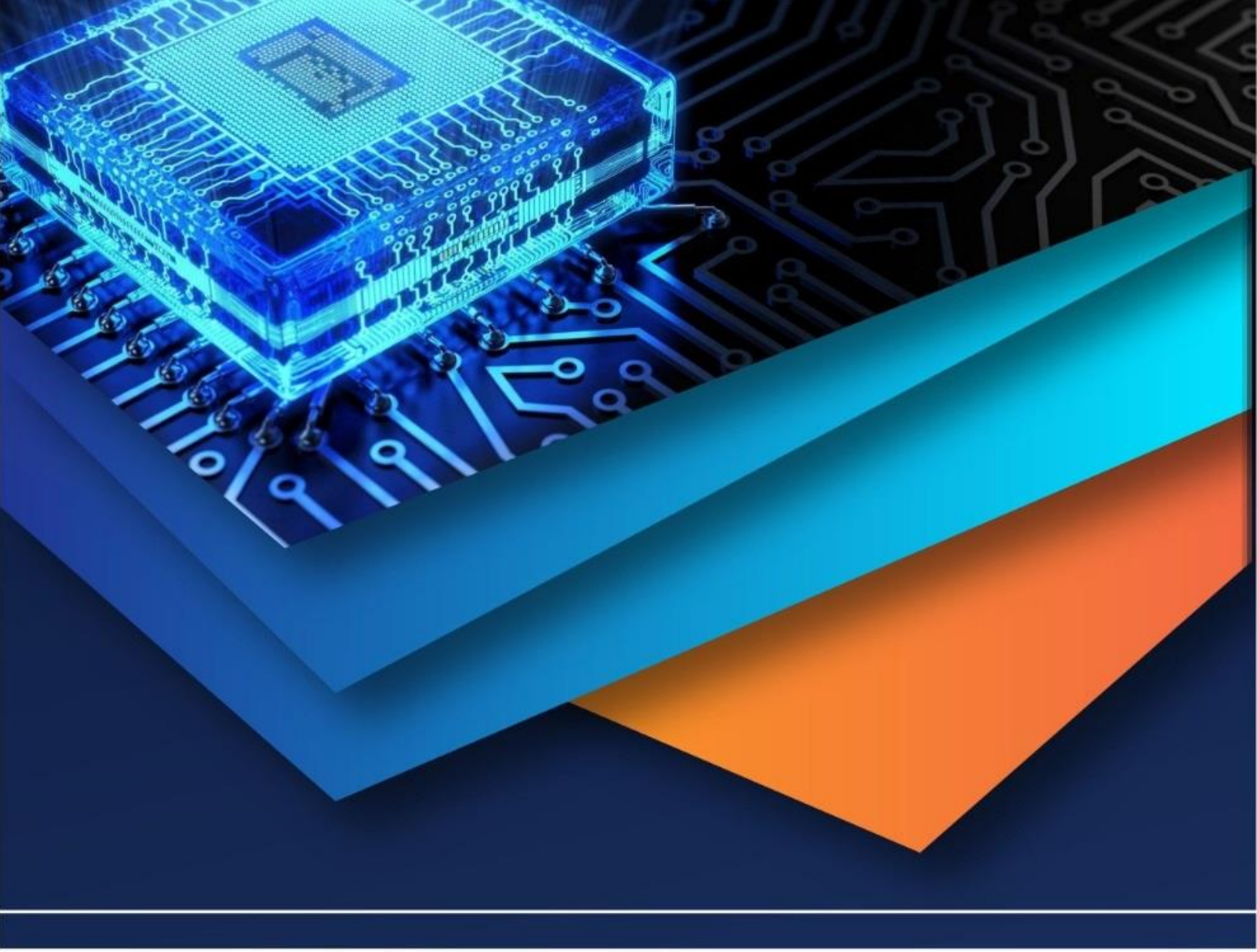
The tunnel reaches the maximum speed of approximately 50 m/s in the test section. However, it is recommended, for reasons of energy efficiency, that the operating range does not exceed 40 m/ s.

The functionality of the designed tunnel should be evaluated after the construction, installation and instrumentation of the equipment. These are some of the subjects that may be part of future work.

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