SOME FLUID DYNAMICS ASPECTS OF VEHICLE AIR CONDITIONING VENTILATED SYSTEM

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Abstract. *This work presents an experimental study of the airflow in an automotive air conditioning system, especially ventilated air conditioning (HVAC) and its ducts. The objective was to compare the performance of the HVAC for different geometries of the air ducts, evaluating the flow rate and pressures. It was used a test bench constituted by an automotive HVAC, their air ducts, and an instrumental panel. The test bench was constituted by an electrical source responsible for the energy supply to the system. The experimental results showed a great potential improvement of the system, either by the pressure drop, or the geometrical characteristics of the air ducts. Significant gains were achieved for the flow rate and homogeneous distribution at the branch ducts with the modifications implemented.*

Keywords: *Automotive HVAC system. Centrifugal blower. Pressure drop. Experimental analysis*

1. INTRODUCTION

Thermal comfort is a state of mind that expresses satisfaction with the surrounding environment, as defined by The American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE, 2004). It is defined as a set of specific combinations of thermal conditions of the environment and personal factors that produce conditions acceptable to the majority of those present at a particular location. Cars have extremely unfavorable conditions for comfort, due to inhomogeneity of their thermal environment, high thermal gradients and air speed, asymmetric solar flow by the glazy area, besides thermal insulation provided by the tissues of the seats (GIMENEZ, 2006).

One of the most difficult tasks in the design of an automotive air conditioning system is to ensure thermal comfort in all possible situations. The evaluation of the thermal comfort inside vehicle cabins has been studied in depth for a long time. Alahmer et. al. (2011) presented a comprehensive review of the different models developed to predict vehicular cabins thermal comfort, in addition to the different experimental techniques used. The review classified the published work into the two modeling efforts; the human physiological and the psychological models. Additionally the authors discuss the different experimental approaches used to capture the different parameters affecting the in-cabin conditions.

Kaynakli and Kilic (2005) describe a study of thermal comfort during the heating period inside a vehicle. During unsteady conditions, a computational model simulated heat and mass transfer between the human body and the interior environment of an automobile, and the results were compared with experimental data. Temperature, humidity and air velocity were measured inside the automobile. The change of temperature was determined experimentally and theoretically in 16 sedentary segments of the human body, in order to investigate the thermal conditions on the human physiology and thermal comfort. Zhang et. al. (2009a and 2009b) evaluated the air flow and temperature fields inside the passenger compartment, with and without passengers. The numerical model was developed using the FLUENT software and was validated with experimental data.

Basically, an automotive air-conditioning system consists of subsystems of ventilation and direction, heating, cooling and air dehumidification. Through these subsystems, the temperature, the speed and the direction of airflow may be controlled, ensuring appropriate conditions for the comfort of vehicle occupants. Lee and Yoo (2000) developed a performance simulation program for integrated automobile air conditioning system combining performance analysis programs of the separate components. The air conditioning system consists of a laminated type evaporator, a swash plate type compressor, a parallel flow type condenser, a receiver drier and an externally equalized thermostatic expansion valve. Hosoz and Ertunc (2006) used artificial neural networks to predict the performance of automotive air conditioning systems using HFC134a as the refrigerant. With experimental data, a propagation algorithm was used to predict various performance parameters of the system.

It is known that the quantity and quality of air blown into the vehicle should be considered in the design of the automotive air conditioning system, since they play an important role in ensuring the comfort of the vehicle occupants. The entire cabin must be cooled at a rate higher than the air heated by the external environment. The airflow is generated in the air-box, located inside the panel and directed to the passenger compartment through the air ducts. This paper presents an experimental study of the influence of the geometry of the air ducts on the performance of an automotive air conditioning system. The performance was evaluated by the measurement of the pressure and the flow rate.

2. HVAC System

The purpose of HVAC of modern cars is to provide thermal comfort, regardless of weather conditions outside the cabin. The main components of an automotive air conditioning system are the condenser, evaporator, compressor, expansion valve, and the air-box. The main function of the air-box is to transmit to the occupants the desired condition of ventilation, allowing control of airflow, direction, temperature and origin air capture.

To ensure these features, the geometry of the air-box allows the user to select the airflow distribution. The airflow can be directed to the feet, to the front diffuser, to the windshield diffusers and to the side windows. The user can also choose aspiration of the outside air or recirculation of the treated air. Figure 1 shows a schematic of an air-box, with the basic internal components and the possible directions of the airflow.

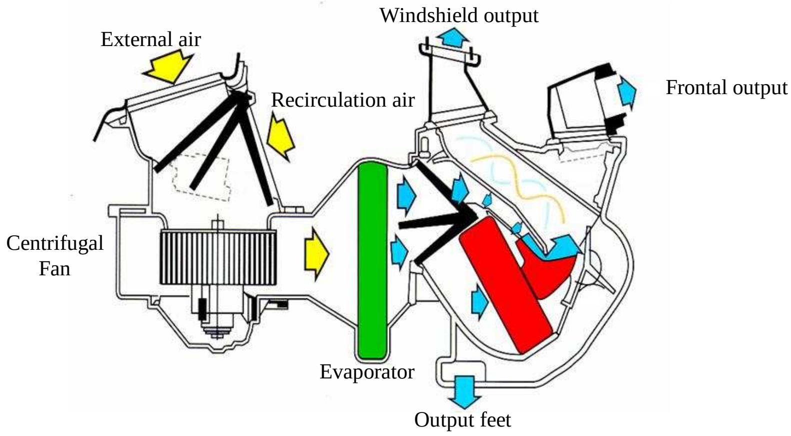

Figure 1. Schematic section of an air-box.

The air temperature can also be controlled by the user, directing some or even all the treated air through a heat exchanger heated within the air-box. Furthermore, the user has the option to select the speed of the centrifugal blower. The control is made through a resistance of three stages, where the fourth speed - the highest rate of flow, is not achieved by electrical resistance, the electric motor is powered directly by the voltage of the alternator.

3. EXPERIMENTAL METHODOLOGY

The centrifugal fan, or blower, that generates the airflow has 35 equally spaced blades, outer diameter of 135 mm; inner diameter of 95 mm; blades length in the radial direction of 20 mm; height of the fan 92,4 mm.

The air ducts connect the air-box to the diffusers of the panel. Two different ducts geometries were analyzed: D1 Configuration (Fig. 2A), with more pronounced curves, and D2 Configuration (Fig. 2B), with smoothed curves.

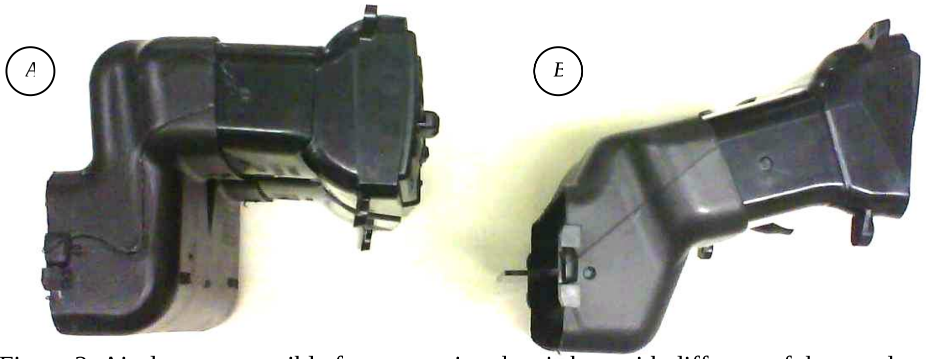

Figure 2. Air ducts responsible for connecting the air-box with diffusers of the panel.
D1 Configuration (A) and D2 Configuration (B).

The changes made in the HVAC system intended to increase the volumetric flow rate inside the vehicle cabin. Thus, parameters that cause reduction of head loss affect directly the airflow leaving the diffusers.

Initially, due to the large number of possible configurations of the HVAC system, it was determined the most relevant for the purpose of the study. Therefore, the shape of the duct that connects the air-box and the exit of control panel was studied.

It was adopted only the condition of “maximum cold”, in other words, the air flow doesn’t pass through the heat exchanger. This configuration is usually the standard configuration tested according Fiat Auto 7-C4050 (2004).

As the test bench does not include the vehicle body, only the air-box, ducts and panel, it was adopted the condition of air recirculation.

Finally, all tests were conducted for the four blower speeds, controlled by the selector panel. Measurements were made directly in the diffusers and performed individually.

The tests were performed with controlled and constant ambient temperature. Variables like atmospheric pressure, current and voltage of the centrifugal fan motor were measured, as well as the pressure inside the air-box and the volumetric flow that leaves the diffusers.

For measuring the volumetric flow of air leaving the diffuser it was used an anemometer propeller with a diameter of 100 mm. Each exit diffuser was connected into a strength length (driver), made with PVC and with cylindrical form. The length of 350 mm was used, according to the orientation of the standard Fiat Auto 7-C4050 (2004).

The outlet pressures of the air diffusers were measured using three different pressure taps. This procedure was used to obtain an average value of pressure at the output section. The taps were made in the same section of the drivers, from 200 mm of the initial section of coupling with the diffusers, and lagged 120 ° among them. Flexible plastic hoses were connected to plugs. The pressures values were acquired using a multifunction instrument TESTO 454.

Similarly the output pressure, the measurement of the internal pressure of the air-box (plenum) was performed using three measuring points. They were made in the intermediate region of the air-box, between the centrifugal fan and the outlet of the air ducts. The measurements were repeated three times for each fan speed. Figure 3 shows the test bench fully configured for the tests. The analysis was done considering steady state condition.

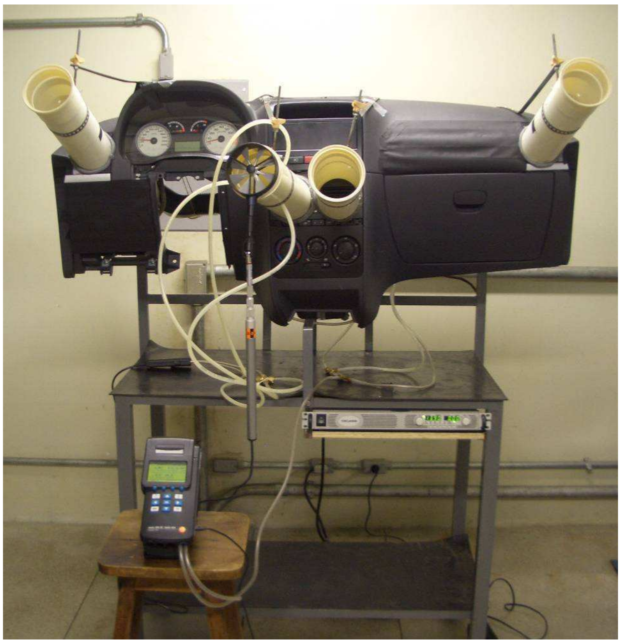

Figure 3. Test bench ready for the tests.

4. RESULTS AND DISCUSSION

The first part presents data pertaining to the flows measured in the total output of the diffusers. The influences of the changes of each component rated - centrifugal fan, air duct and diffusers grills - are presented separately.

Initially it is made a comparison between the modified components and the original system (D1). Made this comparison , it is measured the influence of the component in question over all other possible configurations, but always changing just one element in the comparison.

At the end, the results of all settings are grouped into a single table and a measurement of the influence of all components combined together is performed. It is important to clarify that the comparatives presented here were performed considering the measurements with all diffusers opened.

At the second part of results, a similar analysis is performed, but involves the measured pressures inside the air-box, after the centrifugal fan (blower).

The analysis is performed taking into account the expanded uncertainty with 95% of confidence, according ABNT (1998) and Albertazzi (2004) orientations, that applies to the calculated flow of 31.0 m^3/h, and the pressure of 28.9 Pa.

4.1. Flow – Influence of Central Duct

Based on the original configuration (D1) the duct in D2 configuration provides an increase in maximum flow of 79.8 m^3/h, equivalent to 23.5% on the measured flow at maximum speed, 339.1 m^3/h (Tab. 1). Looking at Fig. 4 it is concluded that in this case there are insufficient evidences to assert that the duct D2 presents results related to flow better than duct D1.

Table 1. Airflow comparison in all configuration.

		Airflow [m^3/h]			
Blower Speed Comand		1	2	3	4
Configuration	**D1**	131.8	190.6	251.6	339.1
	D2	153.8	228.6	306.2	418.9
Percentage Difference		**16.7%**	**19.9%**	**21.7%**	**23.5%**

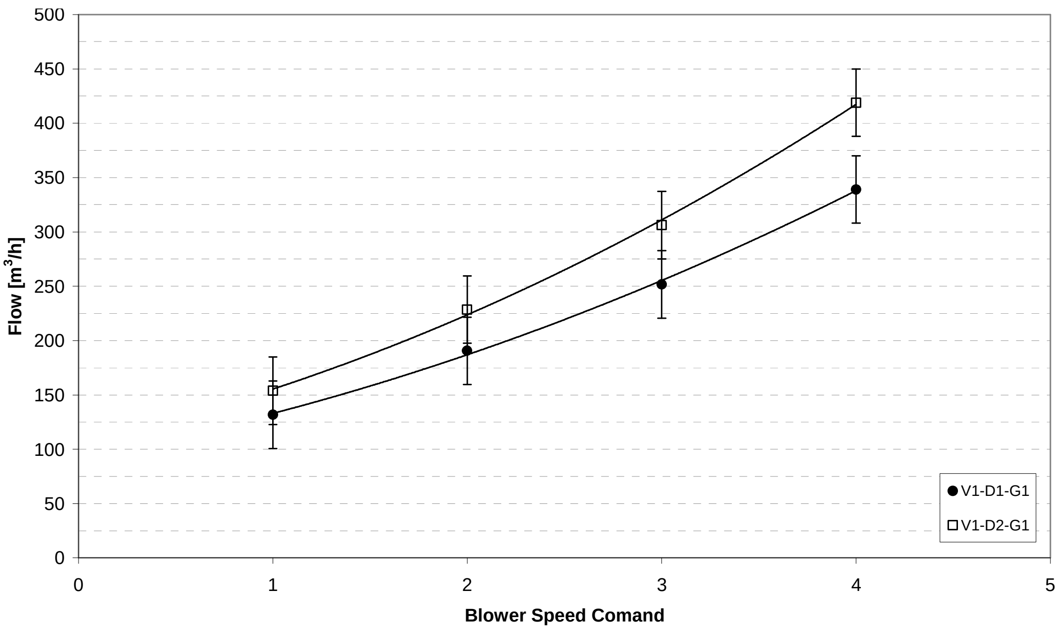

Figure 4. Comparison of flow settings D1 and D2.

4.2. Airflow distribution

Another relevant analysis is the distribution of air flow among diffusers. The flow results presented were always referred to the sum of individual flow measurements in diffusers. By way of facilitating the analysis it will be presented graphics relating to the D1 and D2 configuration. Table 2 contains data obtained in the test for configuration D1.

The nomenclature used to identify the grills is left lateral diffuser (LE), left central diffuser (CE), right central diffuser (CD) and right lateral diffusion (LD)

The results of Tab. 2 show that the central ducts have only 33% of total flow rate, while the lateral ducts represent 67% of the total flow. It is observed that, in this configuration, the central duct offer a restriction on air flow.

Ideally each diffuser should have the same flow rate value. If this is not possible in terms of design, the central ducts should have greater flow rate values, since (once) this action improves the thermal comfort of occupants.

Table 2. Distribution of the flow for configuration D1.

		Flow distribution			
Blower Speed Comand		1	2	3	4
Difusor	LE	33.4%	33.6%	33.7%	33.8%
	CE	14.1%	14.0%	14.0%	14.0%
	CD	19.4%	19.1%	18.8%	18.6%
	LD	33.1%	33.3%	33.5%	33.7%

A fact that was observed is that there isn't a significant variation in the distribution of the flow from the grills by changing the flow velocity.

Figure 5 presents the absolute values of flow rate showed at Tab. 2. There is a misdistribution of flow rate, causing a thermal comfort reduction of the vehicle occupants.

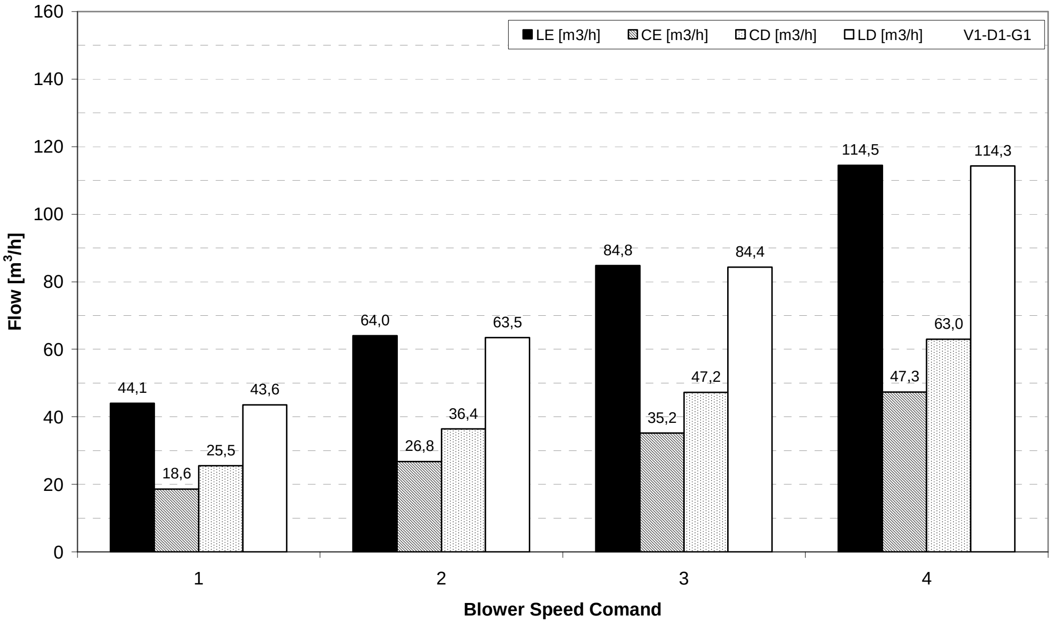

Figure 5. Distribution of the flow for configuration D1.

Table 3 presents data on the distribution of the flow for configuration D2, which is similar to the previous one except for the adoption of the proposed duct. It is interesting to observe the change of the distribution of flow when duct D2 is adopted, becoming higher in central diffusers. From the evaluation of the Fig. 6, this change of behavior can be clearly observed.

It is interesting to observe the changing of flow rate among the branch ducts, and a more homogeneous distribution. Figure 6 shows the absolute value of flow rate. The values are higher than the first configuration studied (D1), for all branch ducts, and reveals that the greater the speed of command, the greater the flow rate is.

Table 3. Distribution of the flow for configuration D2.

		Flow distribution			
Blower Speed Comand		1	2	3	4
Difusor	**LE**	22.0%	21.6%	21.5%	21.4%
	CE	29.8%	30.2%	30.2%	30.3%
	CD	26.5%	26.6%	26.5%	26.5%
	LD	21.7%	21.6%	21.8%	21.8%

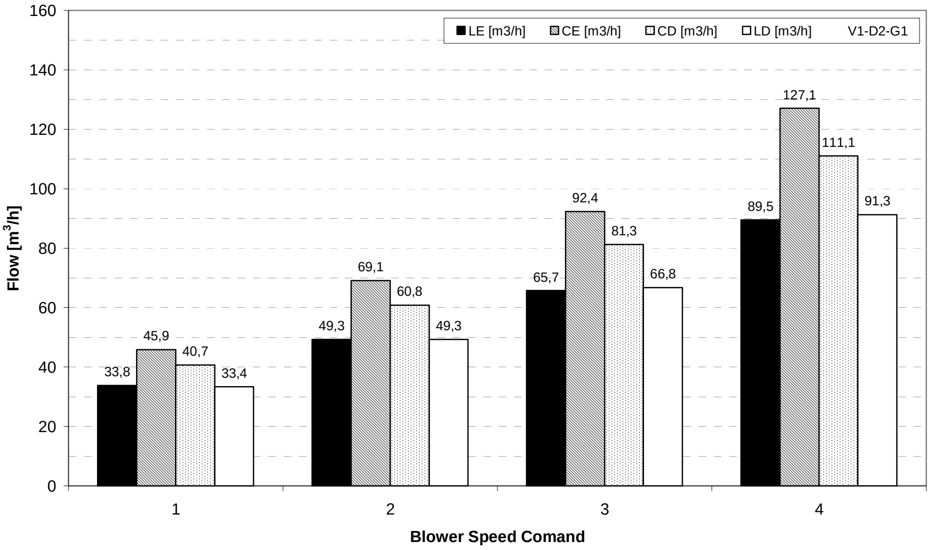

Figure 6. Distribution of the flow for configuration D2.

4.3. Pressure – Influence of Central Duct

The influence of the central duct and its contribution for an airflow was analised. Based on the original configuration (D1), the duct in configuration D2 is responsible for a reduction in pressure up to 118.3 Pa, equivalent to 41.3% on the pressure of 286.8 Pa measured at the maximum speed (Fig. 7).

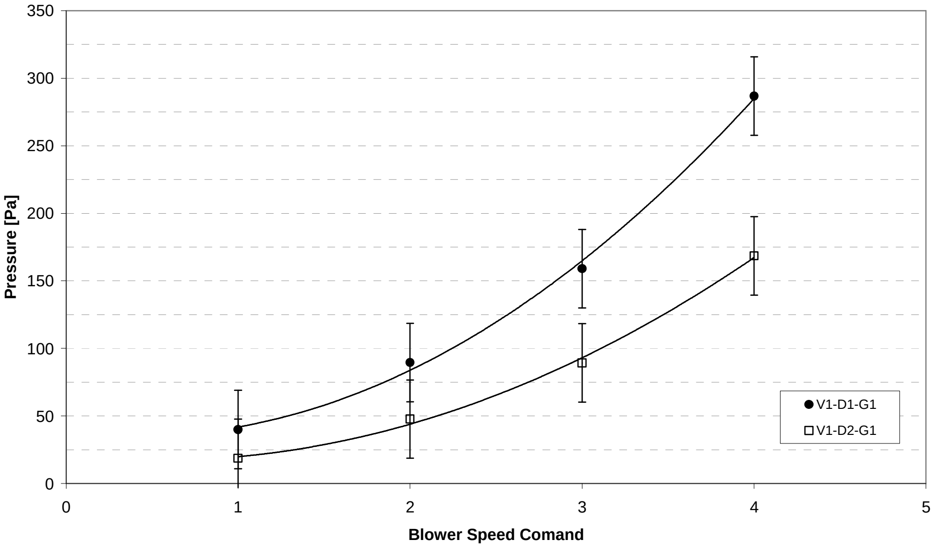

Figure 7. Comparison of pressure for configurations D1 and D2.

Table 4 shows the pressures measured in all possible configurations. It can also be found a comparison of the percentage of pressure variation face the original configuration (D2 vs. D1, respectively). Thus, there is a reduction of up to 118.3 Pa, or 41.2% when compared with the pressure of 286.8 Pa provided by the original configuration D1 at its maximum speed. It was observed that at lower speeds the percentage difference is higher, up to 53.2% minor when using D2 configuration.

Table 4. Comparison of pressure for all configurations

		Pressure [Pa]			
Blower Speed Comand		1	2	3	4
Configuration	**D1**	40.0	89.6	159.1	286.8
	D2	18.7	47.7	89.3	168.5
Percentage Difference		**-53.2%**	**-46.8%**	**-43.9%**	**-41.2%**

4.4. Power Consumption

The power consumption required by the centrifugal fan was analyzed taking into account the two configurations tested. Both considered the flow rate and electric power uncertainty.

The values of electric power were obtained considering the values of electric current supplied to each condition and read directly from the electrical source. These were multiplied by the constant voltage provided by source and equal to 13.9 V.

Table 5 shows the values of power consumption for the four speeds of fan control. The biggest power consumption difference was obtained at command fan number 4, equal to 6.4%. This result was expected, since bigger is the flow rate, bigger is the energy consumption. However, considering the impact of the thermal comfort inside the vehicle, this result can be considered appropriate to the environment conditions.

Table 5. Comparison of power consumption

		Power Consumption (W)			
Blower Speed Comand		1	2	3	4
Configuration	**D1**	56.4	98.0	156.0	270.1
	D2	57.1	100.3	160.1	287.5
Percentage Difference		**1.2%**	**2.3%**	**2.6%**	**6.4%**

Figure 8 shows the power consumption versus the flow rate for the two configurations and different operation conditions. The flow rate equal to zero was obtained with the diffusers grids closed and the fan operating at different command speeds. As discussed earlier, the increasing at power consumption with D2 configuration is within the range of measurement uncertainty.

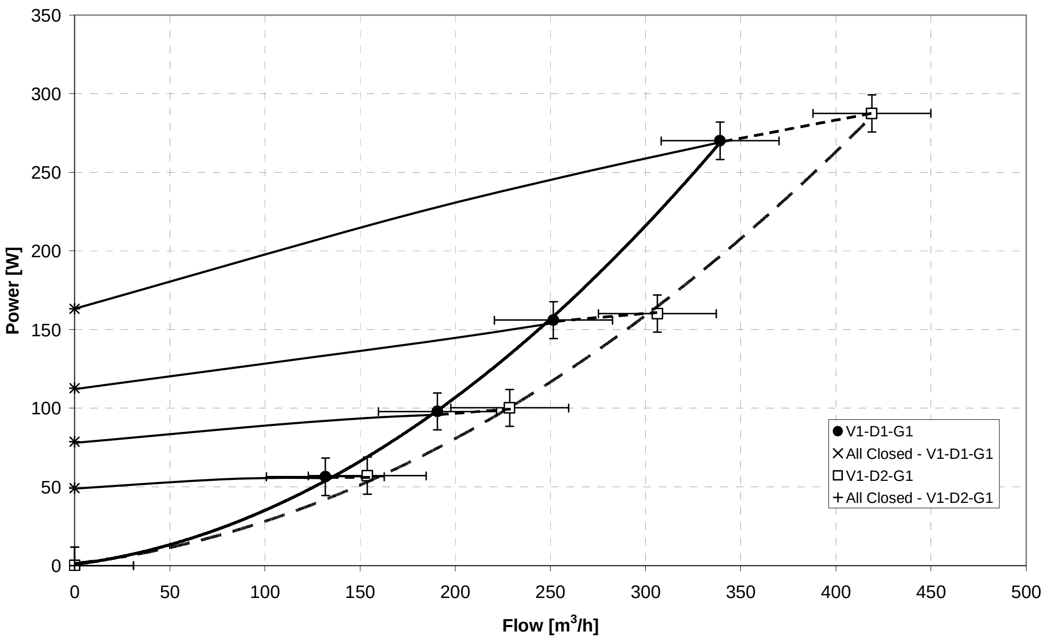

Figure 8. Electrical Power Graph x Flow – comparison between configurations D1 and D2.

5. CONCLUSIONS

From the analysis and tests performed for the two configurations tested at the evaluation of variables that influence the air flow at the vehicle air conditioning ventilated system, it can be concluded:

- There is a great potential for improving the fluid dynamic system studied, since, with the modifications evaluated, significant gains were achieved for the flow rate and homogeneous distribution at the branch ducts;
- The change in the shape of the air ducts was an effective modification, accounting for 23.5% increase in air flow and pressure drop inside the air-box up to 41.2%;
- The pressure drop introduced by the central air ducts also evaluated presents significant influence on the distribution of flow between the branch ducts.
- By altering the shape of the ducts, it was modified the original distribution of flow rate, distributing it almost uniformly, with 67.0% of average flow opened by side diffusers and 33.0% of in flow by central diffusers in the original configuration.

- The power consumption of the tested configurations indicates a higher value for the D2 configuration, but within the range of measurement uncertainty.

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7. REFERENCES

Associação Brasileira de Normas Técnicas; Instituto Nacional de Metrologia, Normalização e Qualidade Industrial, 1998, "Guia para a Expressão da Incerteza da Medição". 3ª Edição, ABNT; INMETRO, Rio de Janeiro.

Albertazzi, A. G. Jr., 2004, "Metrologia – Parte I" – Universidade Federal de Santa Catarina, Florianópolis, SC.

Alahmer, A., Mayyas, A., Mayyas, A., Omar, M.A. and Shan, D., 2011. "Vehicular thermal comfort models; a comprehensive review". Applied Thermal Engineering, Vol. 31, pp. 995-1002.

American Society of Heating Refrigetating and Air-Conditionig Engineers, 1992, "ANSI/ASHRAE Standard 55-2004, Thermal environmental conditions for human occupancy". ASHRAE, Atlanta.

Fiat Auto Normazione, 2004, "7-C4050 – Ripartizione portata ária su veicolo/tronco di scocca". 3ª edição, Satiz Normazione, Itália, 34p.

Gimenez, Rogério Reder; Felix, Victor Barbosa; Tribess, Arlindo, 2006, "Avaliação do nível de conforto térmico global e local em automóveis com ar condicionado utilizando manequim". XV Congresso e Exposições Internacionais da Tecnologia da Mobilidade, SAE, São Paulo.

Hosoz, M., Ertunc, H.M., 2006. "Artificial neural network analysis of an automobile air conditioning system". Energy Conversion and Management, Vol. 47, pp. 1574–1587

Kaynakli, O. and Kilic, M., 2005. "An investigation of thermal comfort inside an automobile during the heating period". Applied Ergonomics, Vol. 36, pp. 301-312.

Lee, G.H. and Yoo, J.Y., 2000, "Performance analysis and simulation of automobile air conditioning system". International Journal of Refrigeration v.23, p.243-254.

Zhang, H., Dai, L., Xu, G., Li, Y., Chen, W. and Tao, W, 2009a "Studies of air-flow and temperature fields inside a passenger compartment for improving thermal comfort and saving energy. Part I: Test/numerical model and validation". Applied Thermal Engineering, Vol. 29, pp. 2022-2027

Zhang, H., Dai, L., Xu, G., Li, Y., Chen, W. and Tao, W, 2009b. "Studies of air-flow and temperature fields inside a passenger compartment for improving thermal comfort and saving energy. Part II: Simulation results and discussion". Applied Thermal Engineering, Vol. 29, pp. 2028-2036

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