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AIR BLOWER EFFICIENCY AND SPECIFIC FUEL CONSUMPTION OPERATING WITH GASOLINE/ETHANOL BLENDS

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Abstract. *The paper presents results for experimental tests in a small air blower, with a two-stroke internal combustion engine operating on non-commercial fuel for different proportions in gasoline and ethanol blends (A10 up to A30). Appropriate instrumentation was used to carry out the measurements of the main parameters, as fuel consumption (g/min), angular velocity (rpm)and blower air speed (m/s). The methodology was based on regulations from INMETRO (motor vehicles energy conversion efficiency) and ABNT (testing of internal combustion engines and ethanol determination in fuel blends). Results obtained are analyzed and discussed when varying angular velocity (~3000-7000 rpm), for each combination fuel blend, versus fuel consumption (g/min) and specific fuel consumption (g.W/min), air flow power (~10-400 W) and, efficiency (%). Main findings are: air flow power reaches minimum values for A27 (20W@3000 rpm) and A30 (70W@4000 rpm), and also* $\Delta m / \Delta n$ *(g/rpm.min) is minimum for A27.*

Keywords: *Two-stroke engine, Combustion, Biofuels (ethanol), Energy conversion efficiency*

1. INTRODUCTION

We are in constant contact with internal combustion engines, some with a high mechanical complexity, like four-stroke motors, or lower complexity, as two-stroke motors. Because of its simplicity, the lower power to weight ratio and the fact of them being more compact than four-stroke engines, two-stroke engines are ideal for small equipment (Leal da Silva, *et al*. 2014), and is used in hedge trimmers, chainsaws and air blowers, among others.

Internal combustion engines are thermal machines in which fuel combusts, which transforms its chemical energy into thermal energy, which can be used to obtain work. The engines can be classified by ignition (Otto ignition, Diesel,…) or by number of stages of the operation cycle (four-stroke and two-stroke). Two-stroke engines are engines where with just one turn of the motor axis, which corresponds to two strokes of the piston, the cycle is completed. As cited before, the commercial air blower selected as the object of study has an internal two-stroke combustion engine. In this study, it will be operated with different blends of gasoline/ethanol.

The object of study used in the tests is an air blower model BG 86, has been manufactured by STIHL, and operates with commercial fuel mixed with lubricant oil recommended by the manufacturer. The air blower is designed for sweeping work and assisting in cleaning tasks of various environments. The engine generates torque, which moves the axis of the rotor. When an energy transformer (for example, thermal machines) transfers energy from a rotor to a fluid, or vice-versa, we can define it as a turbomachine or a fluid-flow machine (Henn, 2012). This is the case of the air blower, which can be considered as a union between an internal combustion engine and a fluid machine, because the mechanical energy is used to move a fluid (In this case, air). These considerations have led the authors to make adaptations in the mathematical formulation of some studied parameters.

Since the 70´s, the use of alternative fuels (other than fossil fuels) has been increasing, to ensure an independence of oil. Today, the most used alternative fuel is ethanol, which is currently being used blended with gasoline. This promotes a higher evaporation heat, octane number, flammability temperature, and reduction of carbon monoxide emissions, but with the cost of an increase of fuel consumption (Koç, *et al*., 2009). Throughout the years, the Brazilian government has raised the percentage of ethanol in gasoline: In 2006 it was around 23%, and today it is around 27%. Other countries use a smaller percentage, as United States, that uses values between 10% and 15% or Europe, with 10% as reference and some countries as Sweden with 85% (E85).

This work has the intention of studying how this addition interferes with efficiency and specific consumption of two-stroke combustion engines, in function of the angular velocity (or rotational speed). Also, we will establish a correlation between these parameters, fuel consumptions and air flow power.

2. METHODOLOGY

2.1 Instrumentation

The internal combustion engine studied in this work is of a commercial air blower, and is a single cylinder, two-stroke engine manufactured by STHIL. The Table (1) displays technical data about the engine of study and Table (2) shows the measuring instruments and their technical specifications.

Table 1. Technical data: Air blower BG 86 - STHIL[1]

Displacement [cm³]	27.2
Diameter of cylinder [mm]	34
Piston Stroke [mm]	30
Fuel tank capacity [cm³]	440
Weight (empty) [kg]	4.4
Power [kW][2]	0.8
Power-weight ratio [kW/kg]	5.500

Table 2. Instrumentation

Equipment – (Manufacturer, model)	Measurement	Range	Resolution	Accuracy
Tachometer – (Instrutherm, TDR100)	Angular Velocity [rpm]	0,5 – 19.999 rpm	0,1<1.000 rpm	± 0,1%
Anemometer – Barometer – (Extech Instruments, HD 350)	Air flow velocity [m/s]	1,0 – 80,00 m/s	0,01	± 0,1%
Balance – (Balmak, ELP-10)	Mass [kg]	Max: 5 kg/ 10kg	1g	1g/ 2g
Chronometer – (Instrutherm, CD-2800)	Time [s]	23h59’59”	1/100”	1/200”
Anemometer, Hygrometer, Thermometer and Barometer – (Instrutherm, THAB-500)	Barometric pressure [hPa]	10,0 – 999,9 hPa	0,1 hPa (10 Pa)	1,5 hPa (150 Pa)
Thermometer and Hygrometer – (Highmed, HM-02)	Temperature and humidity [°C/ %]	-20 – 70 °C/ 20% - 99%	0,1 °C/ 1% UR	± 1°C/ ± 5% UR
Data acquisition Software and cable – (Instrutherm, SW-U801/SW-20)	-	-	-	-

2.2 Preparation of fuel samples

For this work we established six samples of fuel and each sample contains different proportions of ethanol and commercial gasoline in the composition. The blends were prepared according to the standard of ABNT: NBR 6396 (ABNT, 1976); NBR 13992 (ABNT, 2008); NBR 7024(ABNT, 2010); NBR 8689 (ABNT, 2012) and NBR 13993 (ABNT, 2013). And National Agency of Petroleum, Natural gas and Biofuel resolutions: n° 21 (ANP, 2009); n° 23 (ANP, 2010).

According to the definitions and nomenclatures, we understand (in Portuguese) EAR as anhydrous ethanol as reference fuel and EHR as hydrous ethanol reference fuel (or AEAC). The vehicular fuel, named “Gasool” is composed of gasoline and contains a certain percentage of EAR. To refer to a blend with a certain percentage of EAR, we use the AxHx nomenclature, where “x” represents the percentage of EAR, and “y” the percentage of EHR in the sample. For example, Gasohol A15 refers to a compound with 85% of pure gasoline, 15% of EAR and 0% of EHR.

According to the procedure described in the NBR 13992 (ABNT, 2008), the obtaining of the percentage of AEAC is given by a blend of commercial fuel with an aqueous solution. This aqueous solution is composed of pure water and 10% NaCl b/v. This solution, when added to the commercial fuel forms a two-phase mixture, which the upper phase there is pure gasoline and in the lower phase there is ethanol and the aqueous solution of sodium chloride (Figure (1)). A beaker of 100 ml is used for the preparation of the blends, therefore, as shown in Equation (1), the percentage of AEAC (τ) depends on the final volume of the corrected aqueous phase (A).

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[1] Data were obtained from the technical manual supplied by the manufacturer.
[2] Power as ISO 7293.

$$\tau = [\,2.(A-50\,)] \pm 1 \tag{1}$$

To prepare blends with different AEAC concentrations, we use Equations (2) and (3), where the first refers the new blends with content above the reference sample and (3) to new blends with content below the reference sample.

$$E = [C(t-t')/(t'-1)] \tag{2}$$

$$G = [C(t-t')/t'] \tag{3}$$

C parameter is the volume of the commercial sample (ml), t is the content of AEAC in the commercial sample, t' is the desired content, E is the quantity of ethanol to be added and G the quantity of pure gasoline to be added.

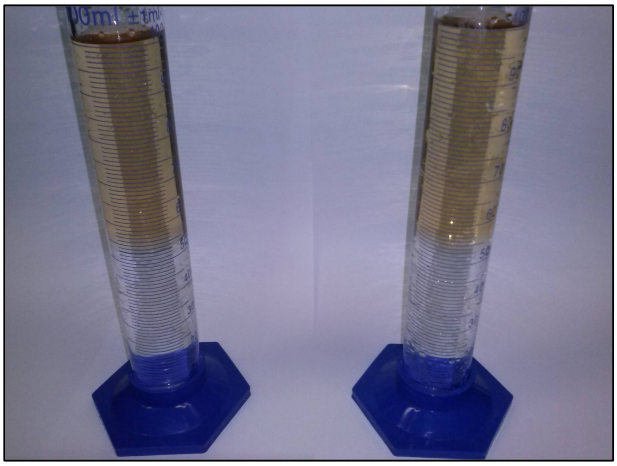

Figure 1 – Blend biphasic pure gasoline/aqueous solution.

2.3 Environmental Conditions

The tests in this work took place in the Energy Engineering laboratory at UFGD campus (Grande Dourados Federal University). The environments parameters of the laboratory conditions are recorded in Table (3), and are the average values obtained during the tests.

Table 3. Environmental conditions

Fuel Blend	**A10**	**A15**	**A20**	**A25**	**A27**	**A30**
Temperature [°C]	27.4	26.7	28.3	29.4	27.4	31.8
Pressure [kPa]	95.8	95.2	95.6	95.6	95.5	95.9
Relative Humidity [%]	63	70	68	54	69	54

All tests, as well as the instrumentation and blends that were used, comply with ABNT normalization and other institutes of standards. Besides, with the sole exception of A30, where the average temperature during the test was higher than 30°C, the data exposed in the Table (3) is within the range of values required by NBR 7024 (ABNT, 2010). However, the environment conditions of the laboratory could not be controlled, due to the lack of installations in the laboratory for this matter.

All samples analyzed follow the lubrication standard as indicated in the manual of the air blower. This is L1:50, where 1:50 symbolizes that for each milliliter of lubricant, fifty milliliters of fuel have to be added.

2.4 Experimental Procedure

The experimental procedures used in this study took into account the NBR 6396 (ABNT, 1976), concerning the performance tests for two-stroke engines.

The methods described by NBR 7024 (ABNT, 2010) were used as the base of the measurements of fuel consumption. However, due to laboratory limitations, some alterations in the methodology became necessary. Using the gravimetric method, the fuel tank mass was obtained with a precision balance, and manually annotated. Thus, the mass variation during the test Δm ($m_{final} - m_{initial}$) could be obtained. Each test lasted two minutes, and the consumption ($\dot{m}$) indicated for Equation (4) is expressed in g/min.

$$\dot{m} = \Delta m / 2 \quad (4)$$

The angular velocity *n* (rpm) was obtained with the use of an optical tachometer positioned 0.5 meters from the axis of rotation and connected to a computer using the commercial software datalogger (Instrutherm). It recorded one measure of rotation by second, for 140 seconds. This interval was stipulated in an intention to improve the precision of the results, when associated to other measurements.

The fuel power is determined by Equation (7), which depends of the heating value (PCI) and the specific mass. According to the literature, the PCI for original gasoline (PCI_g) is around 3,230 GJ/m³ and for ethanol (PCI_e) it ranges around 2,050 GJ/m³. The specific mass for gasoline (ρ_g) and ethanol (ρ_e) are, respectively, approximately 735 kg/m³ and 791,5 kg/m³. The PCI_{blend} and ρ_{blend} can be obtained by Equations (5) and (6), where V_g and V_e are, respectively, the volume in percentage of gasoline and ethanol in the blend.

$$PCI_{blend} = (PCI_g.V_g + PCI_e.V_e)/100\% \quad (5)$$

$$\rho_{blend} = (\rho_g.V_g + \rho_e.V_e)/100\% \quad (6)$$

$$Pg = PCI_{blend}.\dot{m}/\rho_{blend} \quad (7)$$

The air flow power Pe (W), the specific consumption Sc (g/Wh) and efficiency η, were determined by Equations (8), (9) and (10).

$$Pe = \rho_{ar}.A.v^3/2 \quad (8)$$

$$Sc = \dot{m}/Pe \quad (9)$$

$$\eta = Pe/Pg \quad (10)$$

In these equations, ρ_{ar} is the specific mass of the air, A is transverse section area, v is the velocity of the air flow and Pg (W) is the fuel power.

After each sample was prepared, it was placed in the fuel tank, and the engine was switched on. The time of stabilization is determined by the authors and is equal to 10 minutes. Thus the tests are started. Each test lasted 120 seconds and was made for six different rotations, two of them previously established: slow (~2500-3000 rpm) and maximum velocity (~7000 rpm). Figures (2) and (3) show the test configuration.

Figure 2. Experimental apparatus at UFGD facilities.

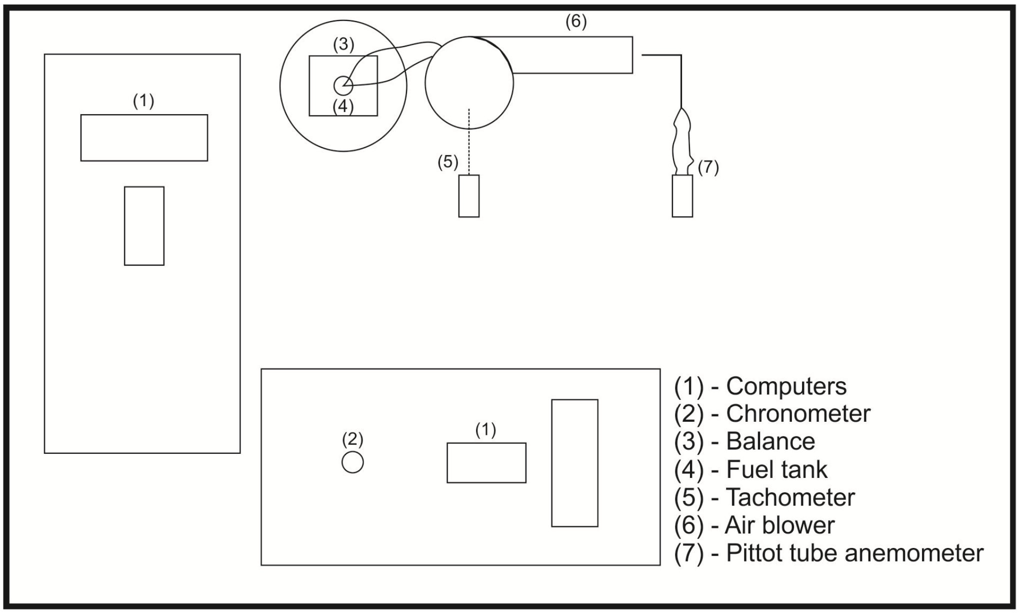

Figure 3. Experimental setup sketch.

All the tests and measurements, as well as the determination of AEAC content in the samples were done in duplicate, with discrepancies in the results lower than 5%. Besides, all the tests were executed according to the procedures cited previously. The calculation of discrepancy (D) is expressed in the Equation (11), where T_1 is the test 1 and T_2 refers to duplicate or test 2.

$$D = [T_1 - T_2) / T_1].100\% \tag{11}$$

3. RESULTS AND DISCUSSIONS

Results for three different fuel blends (A10, A27 and A30) are presented in this work. The results presented in this work are important for understanding the behavior of the engine operated with different blends, and also of easing the understanding of correlation of the analyzed parameters. For the first analysis, three blends that present a high difference are compared between themself, in a proportion of gasoline/ethanol in order to qualify the blends by a better performance. All the results are for the indicated recommended lubrication by the manufacturer, which is L1:50.

Fuel consumption ($\dot{m}$, [g/min]) and specific fuel consumption (Sc, [g/Wh]) versus angular velocity (*n*, rpm), are available in Figure (4). The curves presented in this figure represent polynomial third degree functions, with the exception of the curve for $\dot{m}$ in sample A27 and Sc in sample A30, that represent second degree functions. There are maximum (6000 rpm and 6800 rpm) and minimum (3600 rpm and 4000 rpm) $\dot{m}$ values clearly indicated for A10 and A30, respectively. It is not clearly defined for A27, where maximum and minimum $\dot{m}$ values occur at maximum/minimum *n* values.

Also in Figure (4), at the right y-axis, results for Sc are available. General behavior indicate that Sc decreases as *n* increases, but for A10 there is an exception, as noticed for the 4600-6600 rpm range.

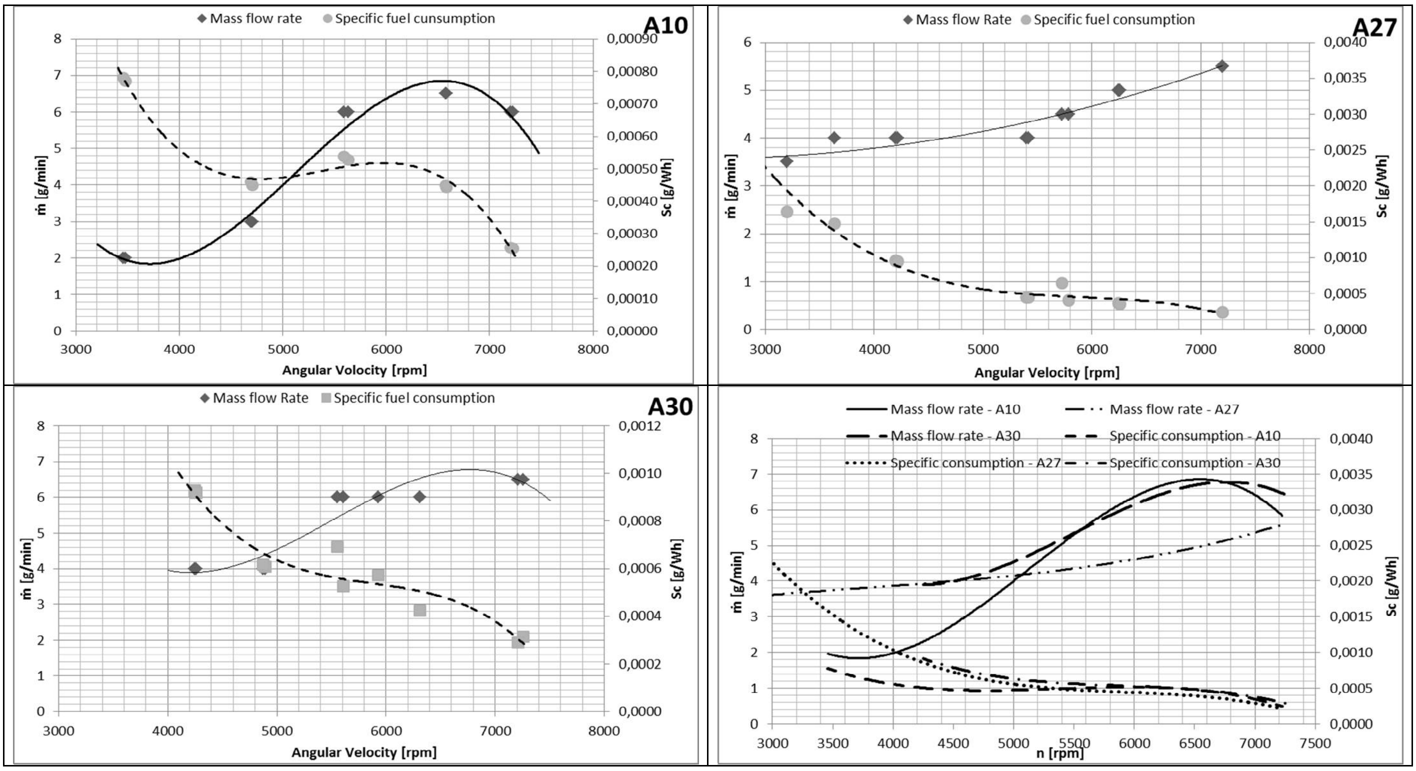

Figure 4. Fuel consumption ($\dot{m}$) and specific fuel consumption (Sc) versus angular velocity (n)

Table (4) presents the variation between maximum and minimum values for: fuel consumption ($\Delta\dot{m}$), specific fuel consumption (ΔSc) and angular velocity related to each parameter (Δn-$\dot{m}$ and Δn-Sc). The ratios between these parameters ($\Delta\dot{m}$/Δn and ΔSc/Δn) are also indicated. Table (4) also shows the effect in ethanol fuel consumption and increase in the specific fuel consumption compared with the speed rate of change. The opposite behavior between fuel consumption and the specific fuel consumption can be observed: from A10 to A27 blends, the increase of ethanol reflects in a decrease in fuel consumption and in an increase of specific fuel consumption. However, this trend is reversed for A27 to A30, which also has $\Delta\dot{m}$/Δn growth but a decrease of ΔSc/Δn. From Equation (9), it is possible to infer that this behavior indicates that the air flow power has a greater influence on specific fuel consumption.

Table 4. Rate variation in function of the angular velocity

Blends	$\Delta\dot{m}$	ΔSc		Δn-$\dot{m}$ (rpm)	Δn-Sc (rpm)	$\Delta\dot{m}$/Δn (g/rpm.min)	ΔSc/Δn (g/W.h.rpm)
A10	4.5	5.2x10-4		3,125	3,768	1.44x10-3	1.39x10-7
A27	2.5	3.2x10-3		4,569	4,576	5.47x10-4	7.08x10-7
A30	2.5	6.3x10-4		3,008	2,964	8.31x10-4	2.15x10-7

For the air flow power curves, Figure (5), we can observe a similar behavior for all fuel blends. For the efficiency curves, the similarity occurs in an increasing tendency, and has a similar behavior in all the tests, but in the variation between the smallest and biggest values of efficiency there is little difference between the blends.

The increasing tendency that we see for the parameters of consumption and air flow power occur in a direct way for some and indirectly for others, due to the increase of angular velocity of the axis that, by consequence, causes the increase of air flow velocity. This, in return, influences directly the air flow power, interfering indirectly in the increase of efficiency and in decrease of specific consumption.

The decrease evidenced by the curve of specific consumption, even with the increase of consumption, shows that the energy variation rate influences more this parameter than its own consumption. When we pay attention to other points obtained, we realize that A27 has the highest variation between the biggest and the smallest values, therefore a flow power variation rate will also be the biggest. This result in the energy consumption for unit of fuel, toward A27 being bigger than in other blends.

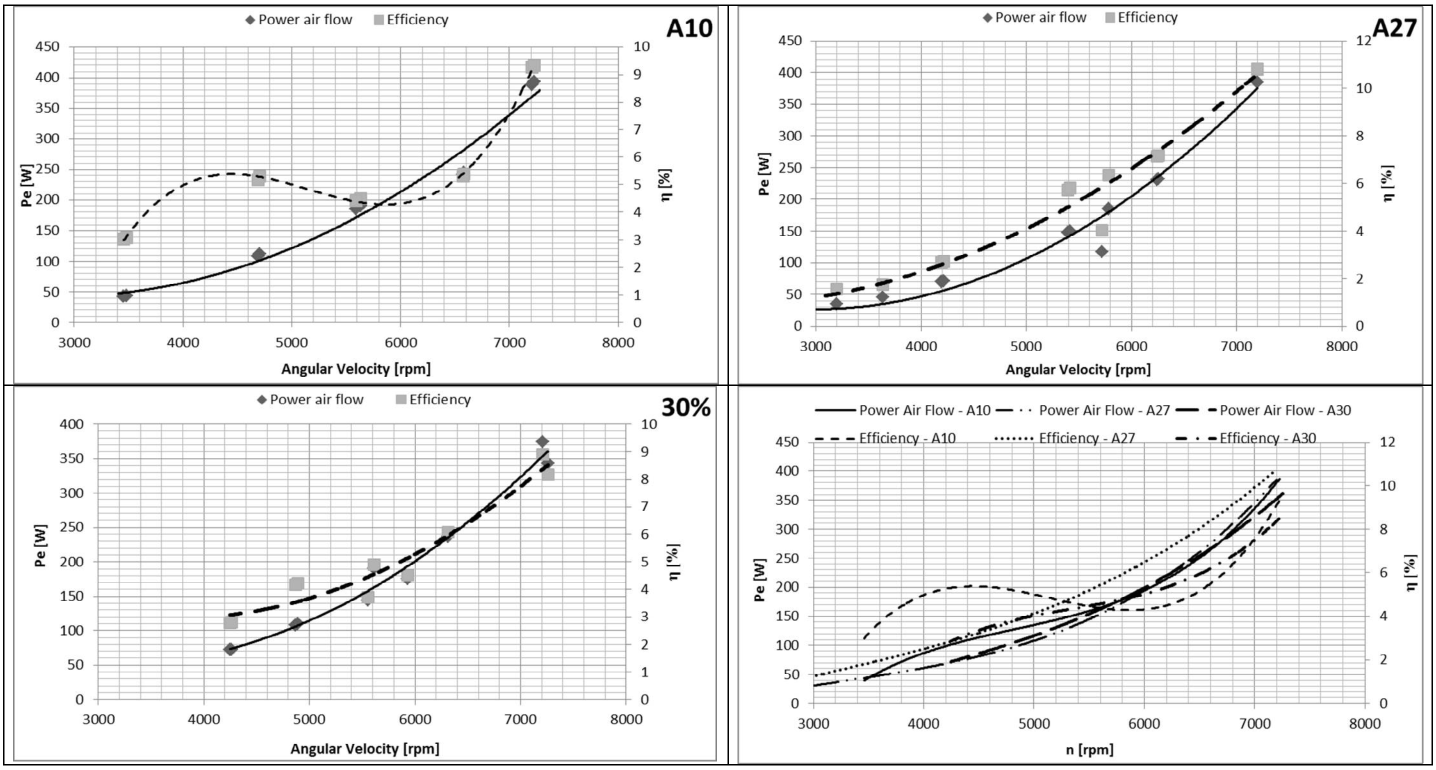

Figure 5. Air flow power (Pe) and efficiency (η) versus angular velocity (n)

According to Brunetti (2012), the engine efficiency is between 10% and 70%, as function of the compression ratio. In the present work, results for two-stroke engine reached a maximum ~10%, Fig. (5), and other effects associated with these low efficiency values can be related to energy losses due to friction (lubrication on two-stroke engines is limited when compared to four-stroke engines) and of the own efficiency of the rotor blades (as a fluid machine). The characteristic curves for fluid machines can take various forms, not only due to angular velocity, but are also dependent of constructive aspects, as the slope angle of the rotor blades, as well as, the number of blades, that have a strong influence in air flow power. According to Henn (2012), for fluid machines, the total efficiency curves pass, gradually, from a plane format to a hook format, when increasing the angular velocity. In other words, with the increase of *n* there is a tendency to obtain a maximum point in the efficiency curve. This behavior was expected. However, it did not occur in the tests, probably due to the fact that angular velocity did not reach values in operational conditions (manufacturer´s limitation) which cause a noticeable effect on global efficiency.

The interpretation of the graph is that the four parameters exposed – consumption, specific consumption, air flow power and efficiency – influence each other. The system starts with the internal combustion, where the heating value of the blends interfere on consumption and in the power and this, in return, interferes on the air flow velocity, that consequently interferes with the air flow power, which alters the parameters of specific consumption and efficiency. Ergo, we can conclude that the heating value of fuel plays a key role in the internal combustion engine performance.

Allied to this discussion and according to what is shown in the tendency curves and results obtained, we conclude that the A10 blend had the best performance, in comparison to the other blends studied in this work. Despite its efficiency at maximum angular velocity not being the highest, A10 presented the smallest average of consumption and specific consumption, to power the air flow as much as others blends. Similar results were obtained by other articles of the literature, where the blends with smaller percentage of ethanol presented smaller consumption (Leal da Silva, *et al.* 2012). For a global analysis, we can conclude that a smaller percentage of ethanol in the blends of gasoline/ethanol present best performance.

4. FINAL CONCLUSIONS

The air blower has a two-stroke internal combustion engine that due to its mechanical simplicity, lower power-weight ratio and compactness, has become very used in daily activities. This frequent contact with the Otto cycle two-stroke engine has led us to have it as an object of study. The air blower engine operates with fuel blends of gasoline/ethanol and lubricant. Formulate characteristic curves for the specific consumptions and efficiency of machines as these one, show us its behavior during operation with different angular velocities, and the fuel blend that optimizes the operation and performance.

Adopting procedures described in ABNT standards and other standard bodies, and defining a methodology to improve the results, reduce possible errors and associated uncertainties with the measurements and calculations, the authors obtained the results shown in Figure (4) and (5), which led us to the conclusion that blend A10 had the best performance, as it can be seen in Table (4), where this sample has the lowest rate variation of specific consumption. In addition, the characteristic curves show that A27 has a good efficiency, although its consumption is not the lowest.

5. ACKNOWLEDGEMENTS

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