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CHARACTERISTIC CURVES FOR A Ø500 MM FREE BLOW FAN AND ITS ENERGY EFFICIENCY FOR INMETRO STANDARDS AND OVERALL

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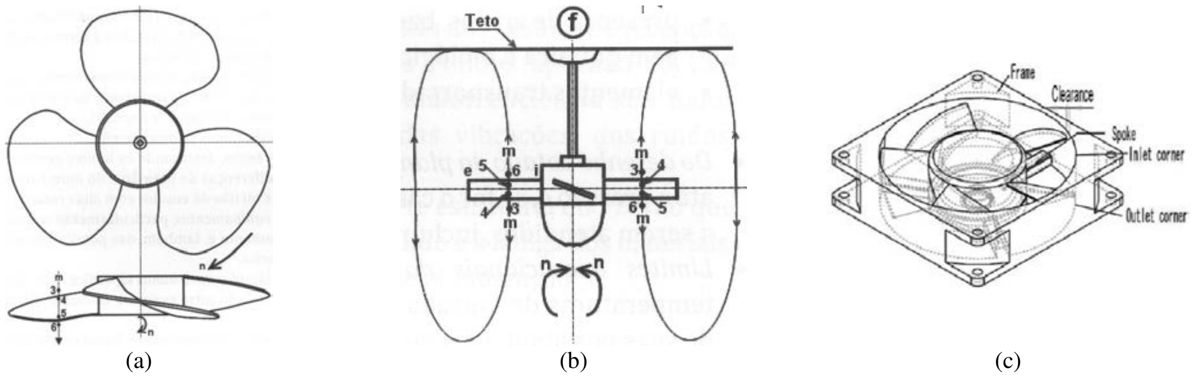
Abstract. *This work consists of experimental evaluation and results analysis of the flow (mass and power) provided by a residential fan and its energy efficiency conversion, according to Brazil´s INMETRO standards. The equipment is a free blow axial fan with 500 mm blades diameter and its angular velocity ranging from 500 up to 1500 rpm. Methodology was adapted from testing for determination of the energy efficiency coefficient for desktop fans for residential use, according to standards of PBE – Brazilian Labeling Program and also an overall eficiency evaluation based on wind turbines analogy. Among the references for bench tests, it was considered INMETRO regulations and ordinances (Requirements for conformity assessment of desktop fans, air circulators and column) and ABNT technical standards (NBR 11829:2008). The results obtained refer to characteristic curves given by flow power (W) and overall efficiency (%) for energy conversion from eletric power into airflow power. Results for normalized energy efficiency coefficient* {[(m^3/s).W].m} *is also included, as defined by INMETRO, for three reference flow rates or rotation velocity (minimum, medium and maximum). Results analysis indicate angular velocities (and flow rates) for the test range values of the characteristics parameters and quality classification (A, B, C or D) according to PBE´s Energy Efficiency regulations.*

Keywords: *Turbomachinery, axial fan, energy conversion, flow measurements, experimental engineering.*

1. INTRODUCTION

This work consists of experimental evaluation and results analysis of the flow (mass and power) provided by a residential fan and its energy efficiency conversion, according to Brazil´s INMETRO standards. The equipment submitted to the tests is a free blow axial fan with 500 mm blades diameter and its angular velocity ranging from 500 up to 1500 rpm. Authors also discuss experimental results for flow velocities visualization just after the fan blades (Z = 0 m) and at the end of the experimental apparatus (Z = 3 m).

Low pressure fans (LPF) corresponds to small size fans and air circulators. These equipments usually have their use in home applications (human thermal comfort), typically for 300 mm ≤ Diameter ≤ 1200 mm; and computers applications (cooling), Diameters < 100 mm (very small sizes). Figure 1 shows a general schematic view and blade geometry for LPF.



(a) (b) (c)

Figure 1. LPF: (a) Desktop fan; (b) Ceiling fan; (c) Computer cooling fan. Souza (2012); Ito, Minorikawa &Fan (2009).

Souza (2012) indicates that for FBF - Free Blow Fans, i.e., ceiling and desktop fans for residential use that do not have scroll, suction occurs in nozzle shape and the air flow is trapped in a recirculation vortex (See Fig. 1).

Fluid machines characteristic curves indicates, for general axial rotors, that higher mass flow (and flow power) is obtained when increasing angular speeds (fan rotational speed). Nevertheless, these conditions imply in global efficiency decreases and noise increases. There are plenty of papers in the engineering literature looking for axial fan efficiency and noise analysis. In literature there are some papers focusing on small size fan analysis.

Zarea and Marval (2007) have evaluated the velocity profiles around an small centrifugal fan, were the maximum radial velocity registered in the tests was 5,0 m/s. Three velocities components were measured, what allowed these

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authors to evaluate velocity fields around the centrifugal fan and its behavior. Left side recirculation vortex was observed as consequence of the recirculation air trapped between fan rotor discharge and suction.

Recent works in the literature look to improve personal computer cooling. A very small axial fans (D = 85 mm) was evaluated by Ito, Minorikawa and Fan (2009). Results investigated by these authors were for aerodynamic and noise performance and some design parameters influence. Fan efficiency improvements were obtained when widening the fan outlet into nozzle shape (corner roundness, i.e., diffuser effect implies in static pressure increase) and also by reducing the blade tip clearance (leakage flows as low as possible).

For noise reduction, tilting the spokes reduces discrete frequency components and the blade tip clearance it plays a significant role (its reduction results in wideband components decreases but also increases discrete frequency components) (Minorikawa and Fan, 2009).

It is worth to mention that, in the literature research performed by the authors, there is no mention to velocity results (quantities or behavior) for desktop or ceiling fans normally used for home applications. Therefore, results presented herein are discussed mainly taking into account theoretical basis for different methodologies (engineering modeling and equations). Only global energy efficiency conversion could be compared to some results obtained in 1978 at UNIFEI laboratories (Souza, 2012) and energy efficiency coefficient established by INMETRO standards (INMETRO, 2012).

2. METODOLOGY

Free blow fans for residential use were tested according to INMETRO standards (INMETRO, 2012). The airflow must be in steady state, and the flow velocities must not vary more than 5 % ($\Delta v \leq 5$ %). In the current work, airflow velocities were obtained by using a hot wire anemometer. Then, volumetric flow rate (m³/s) at flowing conditions is calculated taking into account mean airflow velocities (m/s) and flow cross section area (m²) at the wind tunnel inlet (just after the fan rotor) and outlet (end of the experimental apparatus).

Velocity profile is probably the most important (and least understood) influence quantity, i.e., differences from reference conditions (homogeneous, single-phase Newtonian fluid with an approach velocity profile), according to Miller (1996). From an engineering view point, the many variables that affect velocity profile cannot be evaluated for all possible flowmeters and for all pipeline conditions. For this reason, steady flow and fully developed flow profile, as defined by a Newtonian, homogeneous fluid, are initially assumed. Coefficient variation can then be predicted with dimensionless Reynolds number Miller (1996).

2.1 Instrumentation and technical specification

Instruments used for measurements during tests are indicated in Tab. 1, together with its technical specifications.

Table 1 – Instrumentation for tests

Instrument (Manufacturer, Model)	Measurement	Range	Resolution	Precision
Photo-tacometer (Instrutherm, TDR-100)	Angular velocity	0.5 – 19,999 RPM	0.1 RPM < 1,000 RPM	± 0.1 % +1 digit
Wattmeter (Instrutherm, WD-950)	Active Power	0.05 W – 9999 kW	0.001 W ~ 1 kW	± 2% VA ± 5
Thermo-anemometer (Instrutherm, TAFR-180)	Air velocity	0.2 – 20.0 m/s	0.1 m/s	± 3% + 1
Thermo-hygrometer (Instrutherm, HT-200)	Temperature and Humidity	-20 – 70 ºC / 20 % - 99 % RU	0,1 ºC / 1 % RU	± 1 ºC / ± 5 % RU
Termo-Hygrometer-Anemometer-Barometer (Instrutherm, THAB-500)	Barometric pressure	10.0 – 999.9 hPa	0.1 hPa	1.5 hPa
Cronometer (Instrutherm, CD-2800)	Time	23h59'59"	1/100 sec. < 30 minutes	
Measuring Tape (Starrett, T34-5)	Length	0.001 - 5.000 m	0.001 m	± 0.0005 m
Data acquisition software and cables (Instrutherm, SW-U801 and SW10/SW20)				
Wind Tunnel (Built at the lab, D=0,60m;L=3m)				
Residential Fan, D=50cm (Lorensid, Turbo M2)	Nominal power: 160 W			
Desktop computer				

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2.2 Laboratory reference conditions

Tests were conducted in the Energy Engineering laboratory, RAVA – Refrigeration, Air Conditioning, Ventilation and Heating (from the Portuguese, "*Aquecimento*"). Ambient conditions were measured at the beginning and at the end of tests and mean values are presented in Tab. 2. Temperature and relative humidity were registered by using a thermo-hygrometer (Instrutherm, model HT-200) while barometric pressure by using a thermo-hygrometer-anemometer-barometer (Instrutherm, model THAB-500). Uncertainties are from equipments (Table 1).

Table 2 – Laboratory mean ambient conditions registered during tests.

RH (%)(1)	Temperature (K)	Pressure (Pa)
51 ± 5	298.4 ± 1.0	96360 ± 150

(1) Air Relative Humidity

Recommended values (INMETRO, 2012) for laboratory environmental parameters are: 293 K ≤ T ≤ 297 K and 60% ≤ Φ ≤ 90%. There is no range established for ambient pressure (barometric). As indicated in Tab. 1, temperature is slightly higher than recommended values, once ambient conditions could not be controlled as recommended.

Air density can be estimated by Eq. (1) and your uncertain is from Eq. (2), were parameters must be in T_{air} (K) and p_{local} (Pa). Then, air density mean value and its uncertain is 1.125 ± 0.004 kg/m³, a reference value that must be taken into account in fan characteristic curves.

The numbers 150 and 287.1 on Eq. (2) are from Barometer's uncertain and the ideal gases constant respectively.

$$\rho_{air} = \frac{p_{local}}{R.T_{air}} \tag{1}$$

p_{local}: Pressure at the laboratory (barometric), in absolute values [Pa];
ρ_{air}: Air density [kg/m³];
R: Ideal gases constant, air [287.1 J/kmol.K];
T_{air}: Temperature [K].

$$u(\rho) = \sqrt{\left(\frac{150}{287.1T_{air}}\right)^2 + \left(\frac{p_{local}}{287.1T_{air}^2}\right)^2} \tag{2}$$

$u(\rho)$: Uncertain of the air density.

Authors would like to point out that uncertainty analysis in this paper was performed for all parameters were it was feasible, according to Balbinot and Brusamarello (2010). In future works, deeper analysis of these characteristics is intended to be considered.

2.3 Experimental aparattus and measurements procedure

First of all, fan must be in line with wind tunnel central axis, keeping a minimum distance of 150 mm from the inlet, as illustrated by Fig. 3(c). Then, fan must be turned on in maximum speed for at least 30 minutes, to obtain permanent conditions in the equipment, as recommended by INMETRO Standards (INMETRO, 2012).

The velocities of the airflow, in both wind tunnel inlet and outlet, are measured with hotwire anemometer. At the wind tunnel outlet, the anemometer sensor is positioned in 8 different coordinates (X,Y), as indicated in Fig 3(a); at the inlet, it is positioned at the fan blades average radius, which is defined as the arithmetic average of the inner and outer radii, as indicated by Eq. (12).

The blades angular velocities are measured by using a tachometer. The device was placed at a distance of approximately 500 mm, Fig 3(c), which is within the recommended in the manufacturer's manual (50 - 1500 mm away). Three reflective tapes were attached on the fan blades, intending to increase tachometer resolution in comparison to results preliminarly obtained for only one reflective tape. In this case, angular velocities registered must be divided by 3.

The active power was measured with a Wattmeter, and recorded manually (each 15 seconds), once that instrument has no data acquisition option. The voltage on the wiring should also be measured and registered at the beginning and end of the tests, looking for occasional instabilities during the tests.

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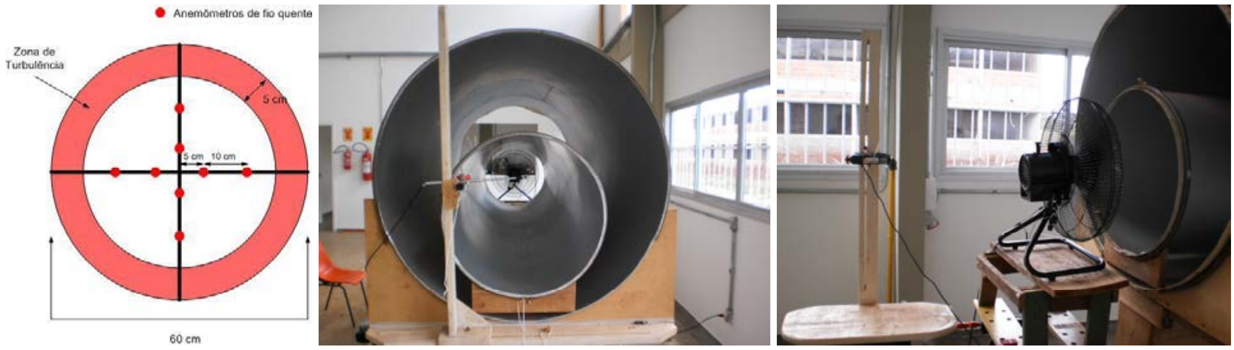
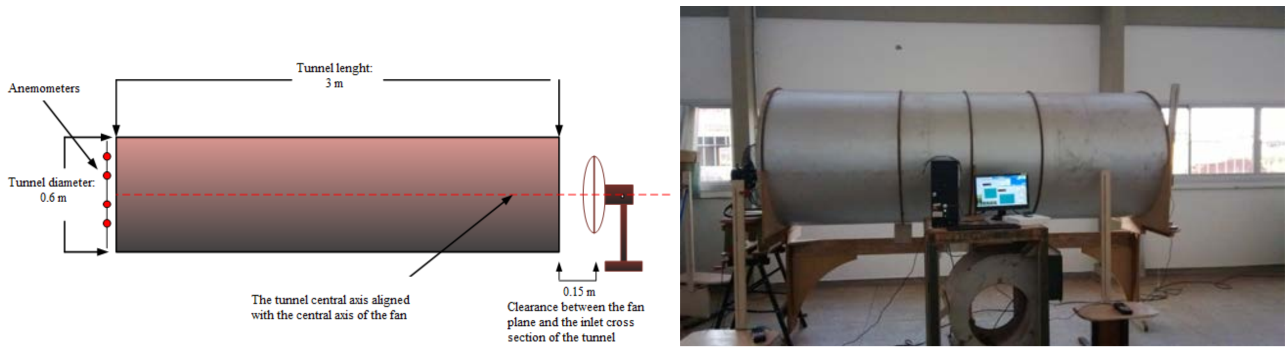


(a) (b) (c)

Figure 2. Experimental apparatus: (a) measurements reference points (INMETRO, 2012); (b) Outlet view; (c) Inlet view

Fan blades angular velocities and flow velocity were recorded by using data acquisition software (model SW-U801), one value for each 1 second, during 1 minute (60 values). Dissimilar from that established by INMETRO standards (INMETRO, 2012) which determines 10 minutes rounds for each of the 8 points, these measurements were recorded simultaneously at 1 minute rounds for each one of the 8 points, to obtain a broader range of results for different angular velocities.

The process was repeated for 8 different positions of the fan potentiometer, which controls the fan angular velocity, but for INMETRO purposes, only 3 positions are considered (low, medium and high) in the results, defined as follows: minimum airflow velocity occurs at the beginning of the movement of the blades with a minimum flow rate of 0.35 m³/s; medium airflow velocity corresponds to the geometric mean of the cursor with minimum airflow of 0.57 m³/s; maximum airflow velocity corresponds to a minimum flow of 0.80 m³/s. See Fig. 3 for details.



(a) (b)

Figure 3. Experimental apparatus: (a) sketch (INMETRO, 2012); (b) Lateral view at RAVA laboratory

2.4 Equations for performance determination

This coefficient is intended to frame the desktop fans in one of the categories defined by ENCE (National Labeling Energy Conservation, from Portuguese "*Etiqueta Nacional de Conservação de Energia*"). It takes into account volumetric flow values at the wind tunnel outlet, i.e., 3 m from the inlet position. To obtain volume flow, first it is necessary to calculate airflow average velocity at the outlet cross section, according to Eq. (3). The same equation is necessary to calculate the airflow average velocity at inlet cross section and the uncertainties for these velocities are from the anemometer shown on Tab. 1.

$$V_{av} = \frac{\sum_{i=1}^{8}\left(\sum \frac{V_{i,n}}{n}\right)}{8} \tag{3}$$

V_{av}: Average velocity at the wind tunnel [m/s];
$V_{i,n}$: Velocity at point "i", coordinates (Xi, Yi) [m/s];
n: Measurements registered (1 per second, during 1 minutes).

Then, we can calculate airflow at the wind tunnel by using Eq. (4) and the uncertainties are calculated by using Eq. (5). The number 10^{-2} is from measuring tape's uncertain multiplied by 2 from first partial derivate of Eq. (4) and the numbers 0.03 and 0.1 on Eq. (5) are from anemometer's uncertain.

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$$Q = v_{av} A_s \tag{4}$$

Q: Volumetric airflow [m³/s];
A_s: Cross section area [m²].

$$u(Q) = \sqrt{(10^{-2}\pi R_s v_{av})^2 + [\pi R_s^2 (0.03 v_{av} + 0.1)]^2} \tag{5}$$

$u(Q)$: Uncertain of volumetric flow;
R_s: Radius of cross section area.

Normalized energy efficiency coefficient, η_{ENCE} (INMETRO,2012)

With the volumetric flow from Eq. (4) and active power registered by the Wattmeter, we can calculate the energy efficiency coefficient, Eq (6), and the uncertain to that number is from Eq. (7), on this equation the number 5 is from wattmeter's uncertain:

$$\eta_{ENCE} = \frac{Q}{P_A} \tag{6}$$

η_{ENCE}: Energy efficiency coefficient [m³/(s.W)];
P_A: Active power [W].

$$u(\eta_{ENCE}) = \sqrt{\left(\frac{u(Q)}{P_A}\right)^2 + \left(\frac{5.Q}{P_A^2}\right)^2} \tag{7}$$

$u(\eta_{ENCE})$: Uncertain of energy efficiency coefficient.

Finally, multiplying it by fan blade diameter, it is normalized, according to Eq (8). The uncertain that accompanies the coefficient is calculated by Eq. (9), can be seen in equation that the parameter to normalizer the energy efficiency coefficient is the same utilized to make correction on uncertain this occurs because the parameter is a nominal length:

$$\eta_{ENCE,n} = \eta_{ENCE}.d \tag{8}$$

$\eta_{ENCE,n}$: Normalized energy efficiency coefficient [(m^3/s.W).m];
d: Propeller fan nominal diameter [m].

$$u(\eta_{ENCE,n}) = d.u(\eta_{ENCE}) \tag{9}$$

$u(\eta_{ENCE,n})$: Uncertain of normalized energy efficiency coefficient.

Then, it is possible to classify desktop fan performance ("A" for better; "E" for worst) by comparing values obtained in Eq. (9) to reference values in Tab. 3.

Table 3. Desktop fans energy efficiency classification (INMETRO, 2012).

Classification	Low Velocity (m³/s.W).m	Medium Velocity (m³/s.W).m	High Velocity (m³/s.W).m
A	n > 0.0040	n > 0.0040	n > 0.0040
B	0.0040 ≥ n> 0.0035	0.0040 ≥ n> 0.0035	0.0040 ≥ n> 0.0035
C	0.0035 ≥ n > 0.0030	0.0035 ≥ n > 0.0030	0.0035 ≥ n > 0.0030
D	n ≤ 0.0030	n ≤ 0.0030	n ≤ 0.0030

Energy efficiency, η_{global}

Energy efficiency conversion in terms of absolute values, η(%), can be defined by another methodology. Here it is performed by modeling the flow as proposed by Custódio (2009), taking into account fan airflow power as an analogy to the flow in a wind turbine. See Equation (10) for air flow power and the uncertain that accompanies is calculated by Eq. (11). Where cross section is the same as indicated in Eq. (4) and the radius is the same of as indicated in Eq. (5).

The number 0.005 is from measuring tape's uncertain, the absence of divisor 2 is because it is multiplied by 2 from first partial derivate with respect to radius of Eq. (10) and the number 3 is from first partial derivate with respect to velocity of Eq. (10).

$$P_{flow} = \frac{\rho.Q_{inlet}.v_{inlet}^2}{2} = \frac{\rho.(A_s.v_{inlet}).v_{inlet}^2}{2} = \frac{\rho.A_s.v_{inlet}^3}{2} \tag{10}$$

P_{flow}: Airflow power [W];
Q_{inlet}: Inlet airflow [m³/s];
v_{inlet}: Average velocity at the wind tunnel inlet [m/s].

$$u(P_{flow}) = \sqrt{\left(\frac{\pi.R_s{}^2.v_{inlet}^2}{2}.u(\rho)\right)^2 + \left(0.005.R_s.v_{inlet}^3.\rho\right)^2 + \left(\frac{3.\pi.R_s^2.v_{inlet}^2.\rho.u(v_{inlet})}{2}\right)^2} \tag{11}$$

$u(P_{flow})$: Uncertain of power flow.

Inlet velocity is registered at one single point, at the fan blades mean radius. It is defined as pointed out in Eq. (12).

$$R_{ave} = \frac{(R_{inner} + R_{outer})}{2} \tag{12}$$

R_{ave} Average radius of the fan blades [m];
R_{outer}: Outer radius of the fan blades [m];
R_{inner}: Inner radius of the fan blades [m].

Finally, calculation of the overall energy efficiency is performed by Eq. (13), where active power is the same as indicated in Eq. (6).

$$\eta_{overall} = \frac{P_{flow}}{P_A}.100 \tag{13}$$

$\eta_{overall}$: Overall energy efficiency [%].

2.5 Reynolds number (Re), velocity coefficient (φ) and fan specific speed (n_{qA})

Reynolds number is pointed out in Eq. (14) and the uncertain is calculated by Eq. (15), the number 0.005 is from measuring tape's uncertain. Notice that calculation is performed for inlet or outlet cross sections. Thus, reference velocity at the inlet is $v_{av,inlet}$, while at the outlet, reference velocity is $v_{av,outlet}$. Its results are going to be considered for three INMETRO reference conditions (low, medium and high airflow, i.e., fan angular velocity), at inlet and outlet conditions for velocity field visualization and discussion.

$$\mathrm{Re} = \frac{\rho v_{reference} D}{\mu} \tag{14}$$

Re: Reynolds number [dimensionless];
$v_{reference}$: Average velocity at the cross section [m/s];
D: Wind tunnel diameter [m];
μ: fluid viscosity, for air 1.8x10^5 (Wickert, J., 2007) [kg/(m.s)].

$$u(\mathrm{Re}) = \sqrt{\left(\frac{v_{reference}.D.u(\rho)}{\mu}\right)^2 + \left(\frac{\rho.D.u(v_{reference})}{\mu}\right)^2 + \left(\frac{0.005.\rho.v_{reference}}{\mu}\right)^2} \tag{15}$$

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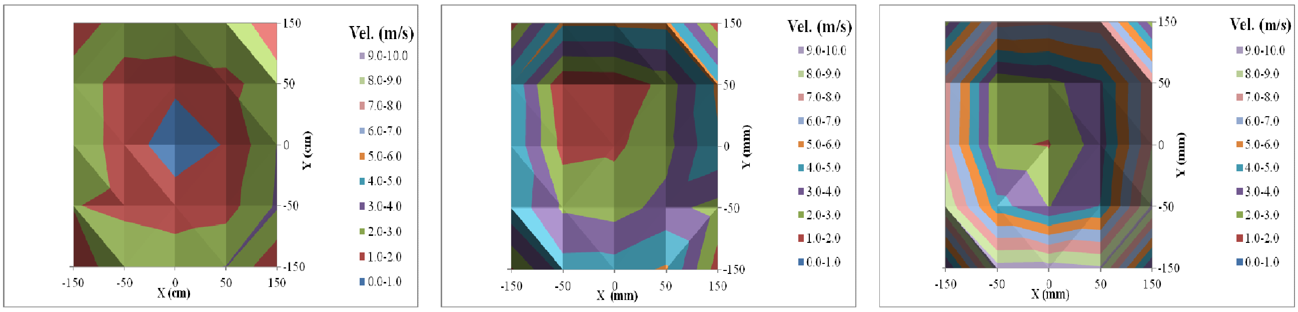
3. EXPERIMENTAL RESULTS AND DISCUSSIONS

Figures 4 and 5 show the velocity fields, respectively at the wind tunnel inlet and outlet, for three reference angular velocities (low, medium and high). These results corresponds to 25 different (X,Y) positions, beginning at the wind tunnel central axis (X = 0, Y = 0): -150mm ≤ X ≤ +150 mm and -150 mm ≤ Y ≤ +150 mm, at the wind tunnel cross sections. As the flow has inherent rotational effects due to fan blades operation, using velocities at only 8 points as in Fig. 3(a), does not provide a good view of the real flow.

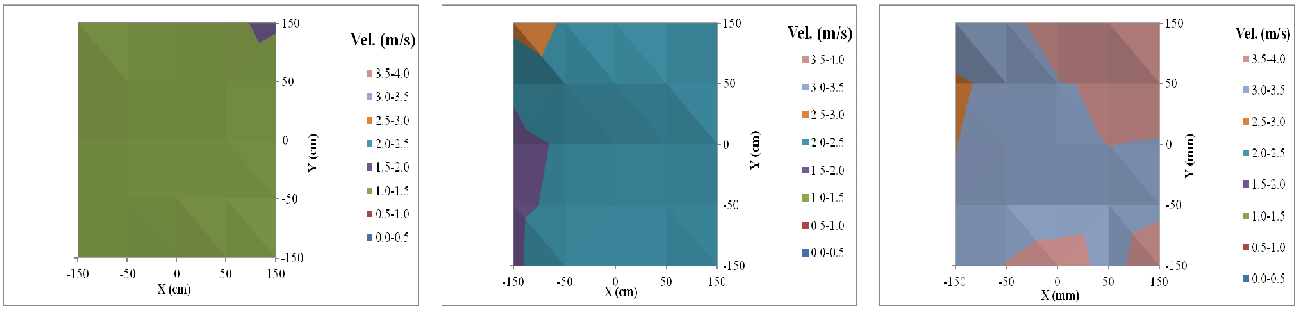


(a) (b) (c)

Figure 4. Inlet velocity fields for angular velocities: a) Low; b) Medium; c) High.

Figure 4, velocities at (a), (b) and (c) conditions are, respectively: 0.5 m/s up to 3.1 m/s (average 2.0 ± 0.2 m/s); 0.5 m/s up to 3.1 m/s (average 3.3 ± 0.2 m/s); 1.9 m/s up to 9.5 m/s (average 5.5 ± 0.3 m/s), and these uncertainties are from the Table 1.

At inlet conditions, velocities at the wind tunnel central axis are affected by the rotor geometry fan front grill, once there are no blades area in this region (D= 10.8 ± 0.5 mm). Moving from the central axis into the outer direction, there are axial velocities increases. Regular rings are observed mainly in Fig. 5(c), showing a more uniform behavior in (X;Y) areas. However, there are decreases in velocity values in the corners (X= ±150 mm; Y= ±150 mm). Then, we have a minimum velocity at the center, increasing values in the outer direction up to a limit when this tendency inverts and goes to a minimum at the walls. Wind tunnel diameter is 600 mm, then, if velocities were taken at the geometric “corners” (X= ±300 mm; Y= ±300 mm), we would obtain zero values at the walls, i.e., no slip condition.



(a) (b) (c)

Figure 5. Outlet velocity fields for angular velocities: a) Low; b) Medium; c) High.

Figure 5, velocities at (a), (b) and (c) conditions are, respectively: from 1.3 m/s up to 1.5 m/s (average 1.4 ± 0.1 m/s); from 1.9 m/s up to 2.6 m/s (average 2.22 ± 0.2 m/s) and from 2.9 m/s up to 3.9 m/s (average 3.42 ± 0.2 m/s), and these uncertainties are from the Table 1. Furthermore, a more uniform set of velocities is observed at the outlet cross section, what corresponds to development in the flow vectors after flowing L = 3 m. Only few peak values at the corners is observed, due to rotational effect from the blades in the flow.

Table 5, next, indicates the airflow velocity measured in the axial direction. They were taken at two different cross sections at the wind tunnel apparatus: inlet and outlet. The axial fan angular velocity varies from near 500 rpm up to 1400 rpm, from a minimum around 2.1±0.2 m/s up to 3,0 ± 0.2 m/s at the outlet and around 3.0 ± 0.2 m/s up to 6.7 ± 0.3 m/s at the inlet. As the airflow pressure is neglected for small axial fans, all differences between velocities at the inlet and outlet are due to pressure drop along the wind tunnel length (L = 3 m).

Table 5. Reynolds numbers at the inlet and outlet wind tunnel cross sections.
Average results for three INMETRO reference angular velocities and its respective volumetric flow.

Laminar flow, Re < 2,10E+04	Angular velocity (RPM) and volumetric flow (m³/s)		
Turbulent flow, Re >2,40E+04	524 RPM and 0.57 m³/s	985 RPM and 0.94 m³/s	1335RPM and 1.56m³/s
Re (Z = 0 m) [1]	7.89E+04 ± 6.03E+03	1.31E+05 ± 7.55E+03	2.18E+05 ± 1.01E+04
Re (Z = 3 m) [2]	5.39E+04 ± 5.30E+03	8.75E+04 ± 6.28E+03	1.35E+05 ± 7.68E+03

[1] for inlet parameters; [2] for outlet parameters

Figure 6 shows the behavior of the volumetric flow-rate (m³/s) dependency on the fan angular velocity (rpm) at the wind tunnel inlet and outlet sections. An increase in angular velocity induces an increase in the flow-rate as expected from Equations (6) and (3). A near linear relationship is obtained from experimental results for velocity and calculated volume flow.

Airflow velocities (and volume flow) at the inlet are higher than those in the outlet, once there is friction occurring at the wind tunnel walls. The proportion between inlet and outlet volumetric flow increases, as fan angular velocity increases. It means that pressure drop (Δp) is higher for lower Re values (ex: 500 RPM) and lower for higher Re values (ex: 1500 RPM). It is coherent to friction factor typical decreases in internal flow at laminar or turbulent conditions (See Reynolds > 2,40E+04, obtained in Tab. 5, and fluid mechanic Moody diagrams).

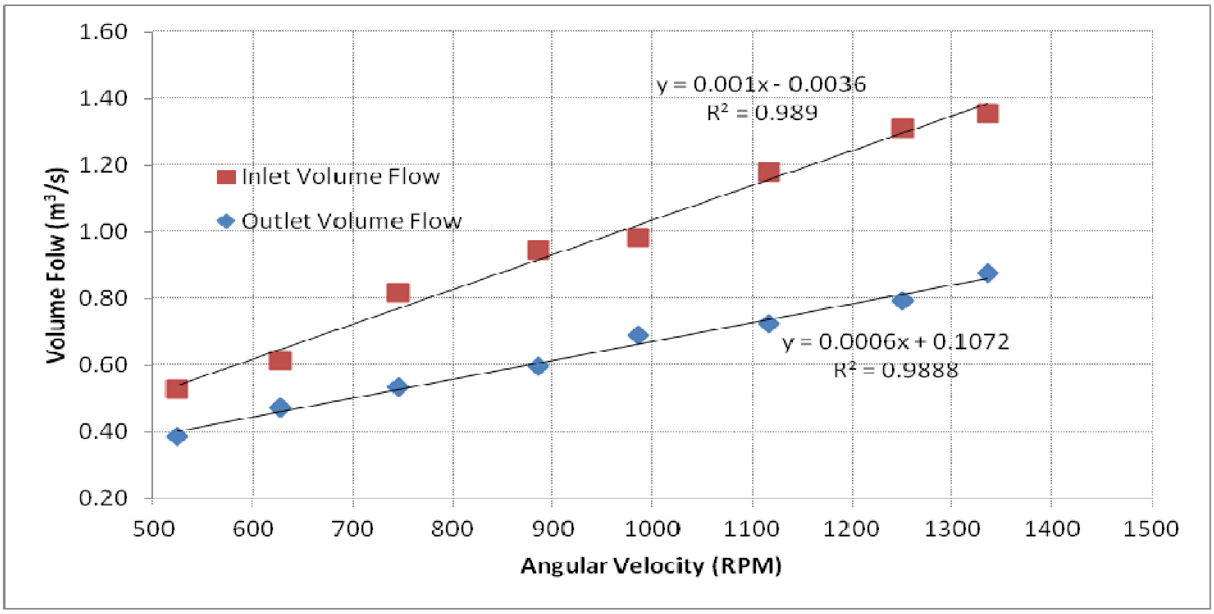


Figure 6. Inlet and outlet volume flow (m³/s) versus angular velocity (rpm).

Figure 7 shows power (W), in the flow (flow power) and electric (active power), versus fan angular velocity (rpm). Parameters, calculated from experimental data for axial velocity and electric power, are directly proportional to flow velocity (m/s) in a third degree polynomial basis, what is coherent with Eq. (10). Also, notice that P_A = U² / R, what results in a nonlinear electric power for constant U and resistance variation on the fan potentiometer, which implies in electric current variation and consequently fan angular velocity variations.

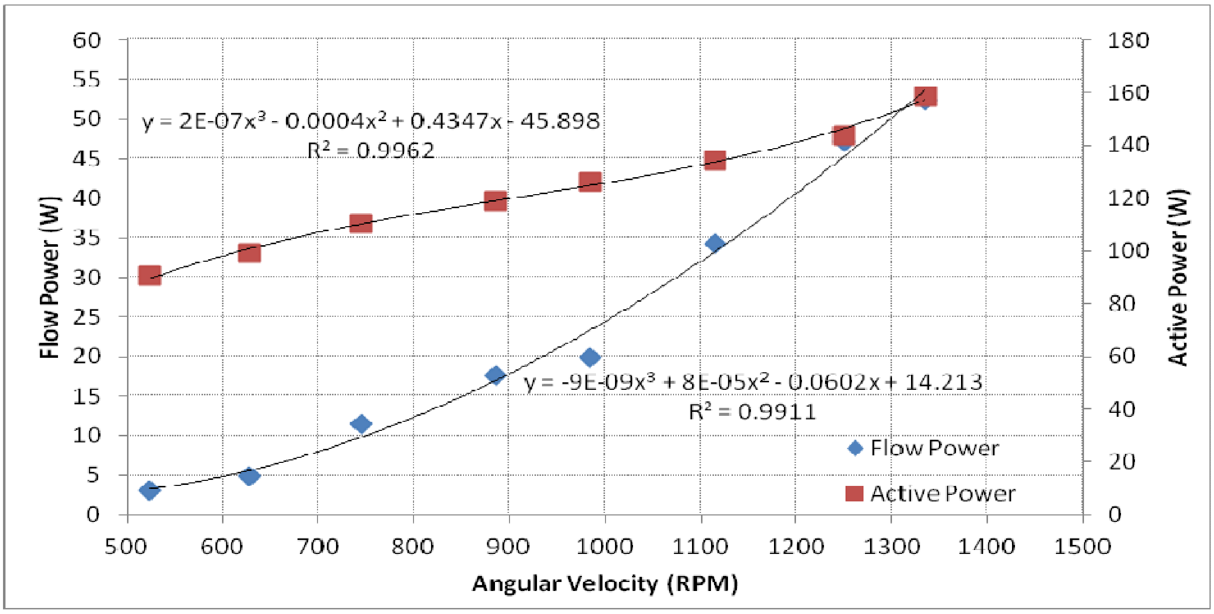


Figure 7. Outlet flow power (W) and active power (W) versus angular velocity (rpm).

Figure 7 shows that the active power goes from a minimum value of 90.5 ± 5.0 W up to a maximum value of 158.8 ± 5.0 W as indicated at right scale, while the flow power goes from a minimum value of 3.13 ± 0.41 W up to a maximum value of 52.40 ± 7.02 W as given in the left scale, these values for uncertain are by from, Table 1 for active power and from Eq. (11) for flow power. Increased slope curves for the flow power are more expressive than the active power slope. It happens due to the fact that fan angular velocity (rpm) is controlled via a resistance potentiometer what implies in P_A = I.R², and it corresponds to ~1.8 (~158.8 / 90.5) times increase from minimum to maximum values. In the other hand, flow power in Eq. (10) corresponds to ~17 (~52.40 / 3.13) times increase from minimum to maximum values in the test range. Then, overall energy efficiency in Eq. (13), will result in ~10 (~17 / 1.8) times increase in the test range.

Energy efficiency conversion at the fan is available in Fig. 8. It was evaluated in overall basis η (%) in Eq. (8), and in a reference basis, i.e., normalized energy efficiency coefficient in Eq. (5). Overall efficiency is continuously higher, as fan angular velocity (rpm) increases, according to the potentiometer range obtained in the experiments (from minimum to maximum). Typically, axial fluid machines must present a maximum efficiency at some angular velocity, but it was not observed for the test range in the equipment under analysis (LorenSid Turbo M2, D=500 mm). It probably means that decrease in overall efficiency occurs for angular velocities higher than 1500 rpm. The same behavior happens in the reference basis, i.e., energy efficiency coefficient increases as angular velocities increases, but mathematical equation that represents tendency behavior indicates that maximum values are in the region for 1300-1400 rpm.

It is possible to infer that the fan manufacturer set the angular velocity range to the limits were energy conversion continuously grows, and begins to decay for angular velocities higher than 1400-1500 rpm. Then, fan operational limits would be in an economical range in terms of energy conversion.

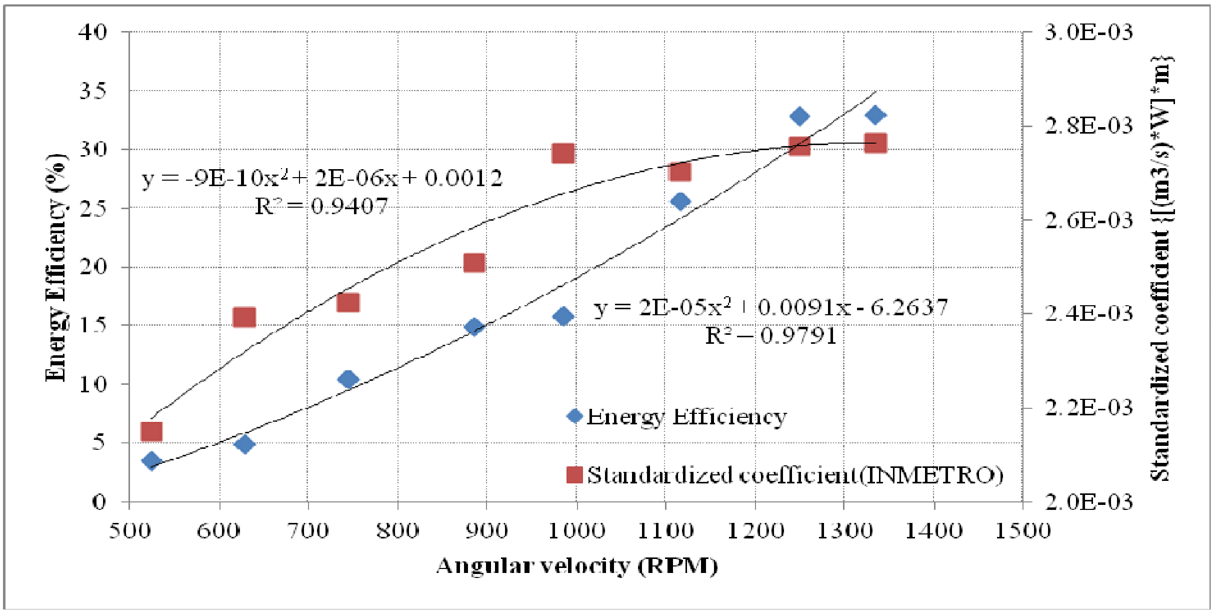


Figure 8. Energy efficiency versus angular velocity (rpm): overall basis, η (%); and reference basis, normalized energy efficiency coefficient {[(m^3/s).W].m}

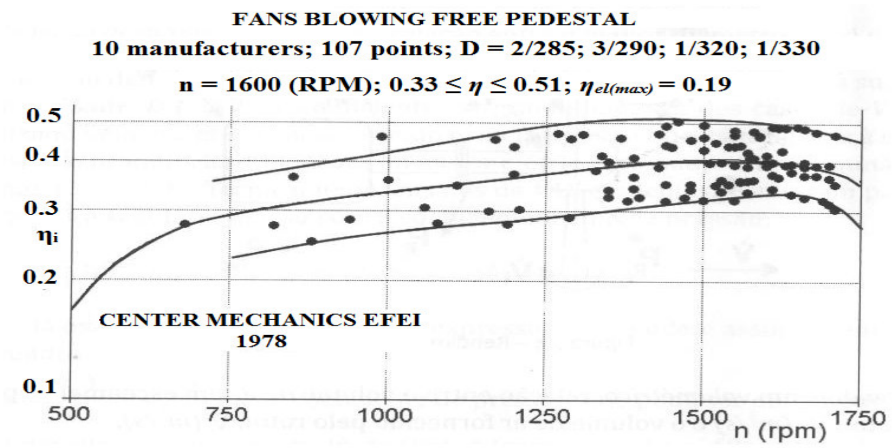


Figure 9. FBL results obtained in 1978 at UNIFEI facilities. Souza (2012)

As pointed out previously, fan energy efficiency (both, overall or reference basis), increases ~7.7 times (~ 30.5 / 4.0) in the test range angular velocities. Analyzing Fig. 9 it can be observed that, in terms of energy efficiency conversion, the recommended operation angular velocity for the desktop fan shall be higher than 1000 rpm (η > 20%).

By comparing results obtained in the actual paper, Fig. 8, and results from the literature (Souza, 2012), Fig 9, it is possible to observe that:

- η < 5% for minimum angular velocity (~500 RPM) agrees with general results in Fig. 9, showing that lower values for η occurs for lower angular velocities;
- η ≈ 15% for medium angular velocity (~1000 RPM), while general results in Fig. 9 show 25% ≤ η ≤ 40% for the few fan models available in 1978;
- η ≈ 30-35% for maximum angular velocity (~1300 RPM), while general results in Fig. 9 show 30% ≤ η ≤ 50% for the a lot of fan models available in 1978;

Table 5 we have the axial fan quality classification, according to INMETRO classification. It is important to inform that tests were not conducted in an accredited laboratory and under environmental conditions slightly outside recommended (INMETRO, 2012) and the standard time cannot be attended for give priority to obtain a broader range of results for different angular velocities. Nevertheless, these results are sufficient to understand that the fan design parameters could be improved (as example, blades aerodynamics, axis torque and angular velocity provided by the electrical engine, among others).

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Table 5 results corroborate to the conclusions pointed out on previous comparison between Figs. 8 and 9, indicating that the axial fan model under analysis is worst than reference values also in the classification proposed by INMETRO standards.

Table 5. Experimental results for energy efficiency coefficient (high, medium and low fan angular velocity). Calculated according to INMETRO standards (1) (INMETRO, 2012).

A	$n > 0.0040$	Classification of adjustments (reference to INMETRO(1) for adjustments 1, 5 and 8)		
B	$0.0040 \geq n > 0.0035$	Low Velocity	Medium Velocity	High Velocity
C	$0.0035 \geq n > 0.0030$	0.0021 ± 0.0003	0.0027 ± 0.0003	0.0028 ± 0.0002
D	$n \leq 0.0030$	D	D	D

(1) The standard time cannot be attended by the time available for the evaluation.

4. CONCLUSIONS

Although the flow is turbulent it is possible to visualize in the velocity field, the eight points of the standard adopted by INMETRO to determine an average speed pertinent to energy efficiency ratings.

With the development in the flow vectors after length the tunnel, we can see a uniform velocity and volumetric flow in different points of the outlet cross section in relation to the inlet, it is also possible to observe a head loss at the wind tunnel walls on the same cross section areas.

With ten times increasing of the flow power under the active power shows a large rating up by the energy conversion in this test ranges, but it is estimated upper this range the conversion will stabilized and down, unfortunately this range cannot be tested because the potentiometer have that limits pre-established.

The overall energy efficiency of the fan is low (< 35%). The only result that agrees with the literature is that for angular velocity near 500 RPM. At the remaining conditions the overall energy efficiency is lower than that expected according the literature.

The energy efficiency coefficient is at the lowest classification (D) defined by INMETRO. Authors would like to emphasize that standard laboratory conditions, as recommended by INMETRO, were not fully obtained.

Then, in general we observe that the axial fan under evaluation in this work, it seems to present worst performance in the whole test range by comparing the axial fans evaluated in literature (SOUZA, 2012). Authors expect to test other axial fan models available in the market, with the same blade diameter (D = 500 mm) and others (D = 300 - 600 mm), and hope to find out better overall energy efficiency and normalized energy efficiency coefficient.

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7. RESPONSIBILITY NOTICE

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