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STUDY OF THE AIRFLOW VARIATION AND THE ELECTRICAL POWER CONSUMPTION IN A VENTILATION SYSTEM DUE TO THE INFLUENCE OF AIR FILTERS

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Abstract. *This article presents the results of an experimental work carried out on a test bench where it was verified the difference of the electrical power consumption by the ventilation system when testing three different types of configurations concerning ventilation system without air filters, with clean air filters and dirty air filters. Two methods were employed for the airflow control. In the first method, a DAMPER valve controlled the airflow; in the second method, a frequency inverter controlled the motor speed. This study demonstrated that in the first method the airflow was considerably reduced while in the second method the electrical power had to be increased to produce the same airflow. The rehearsals were carried out under conditions where the local barometric pressure and the temperature of the ambient dry air bulb circulating ventilation were kept constant, besides the relative humidity between 50 and 80 %. The tests were performed on a test bench controlled by a data acquisition supervisory software. All the results are herein graphically presented and commented. In conclusion, the study proved that when the filter is very dirty the consumption of energy increases almost twice than when it is clean.*

Keywords*: energetic efficiency; ventilation; air filters; maintenance.*

1. INTRODUCTION

It is becoming each more frequent the concern of the governmental sectors or the non governmental entities, private enterprises and the society in general to the approach between the future of energy and the environment when facing questions referred to the need of protecting the environment, the reduction of gas emissions that provokes the stove effect, the preservation of the ozone layer and the substitution of non renewable sources of energy. The social-environmental responsibility is one of the top premises of the foreign investors. In Brazil, financial institutions do not finance any enterprise if the proponent does not present a project in the social and environmental areas. There will be no future investment for an energetic solution if there is no social conscience that everyone should act with responsibility and guarantee one's own sustainable development.

Even with the experience of energy rationalization occurred in 2001, the Brazilian government practically ignored the most practicable and accessible ways to guarantee the country growth security with the rationalization of the use of energy by means of actions, policies and campaigns of energy efficiency. The estimates of the 2007-2016 Decennial Energy Expansion Plan underestimated the potential of conserving energy and the increase of the use of renewable sources of the nation matrix for the next decade. (MME, 2007)

The energetic efficiency may be the chief pillar and the way out for most of the world problems, as it can be transferred to the social and industrial segments.

The choice for a system to move air by an axial ventilator is justified by the fact that it is the most employed in moving a mass volume of air with little drop of pressure along the whole process. That is the case of operation provided by mechanical means to control parameters such as temperature, air distribution, humidity, elimination of contaminant agents or pollutants of which gases, vapors, dust, mist, microorganisms and odors are of paramount importance amongst others. (Macintyre, 1988) This operation consumes energy and consequently has a cost that should be minimized. The search for equipments that present a consuming profile determined by efficiency will be the projectors´ preferred choice. (Mesquita et al, 1988)

An analysis of an installation, as demonstrated by this work, lets it evident that the knowledge of each component performance (ventilator, motor, airflow control and pressure loss) is decisive in the choice, maintenance and conservation of its components.

This article is the result of an experimental work carried out on a test bench of the UNESP Energetic Efficiency Laboratory – Guaratinguetá - SP, Brazil, in order to determine the influence of maintaining the air filters in view of the different power demanded by an axial ventilator and in the variation of the airflow. Airflow variation can be controlled by two distinct methods: by a DAMPER control valve or by a frequency inverter, through which the motor speed is directly controlled.

The scope of this work is to demonstrate that by maintaining the air filters always clean, one can profit by the energy savings with greater circulating air purity. The benefits are not only economical but also to the individual health.

2. MATERIALS AND METHODS

The work was carried out on a test bench of the *UNESP Energetic Efficiency Laboratory* - Campus of Guaratinguetá, SP, Brazil – in order to determine the influence of maintaining air filters as refers to the power required by an assembly of motor-axial ventilator airflow variation, the resulted charge loss and the system efficiency. The airflow variation was obtained by two distinct methods: one, by means of a DAMPER valve and the other, by a frequency inverter that directly controls the motor speed.

The rehearsals were carried out with the ambient dry bulb temperature controlled circa of 23 ºC, a mean barometric pressure of 712 mmHg (95 kPa) and the relative air humidity between 50 and 80 %. The test bench was provided with a data acquisition supervisory system capable of collecting 102 variables 4 times per second. The bench axial ventilator was fitted with 8 blades symmetrically distributed; project pressure coefficient of 0,18; cube relation of 0,43; external diameter of 350 mm and the deep point of 10 mm directly operated by the electrical motor. Figure 1 shows the test bench embodying an air conditioner installation, with the main devices pointed out by arrows.

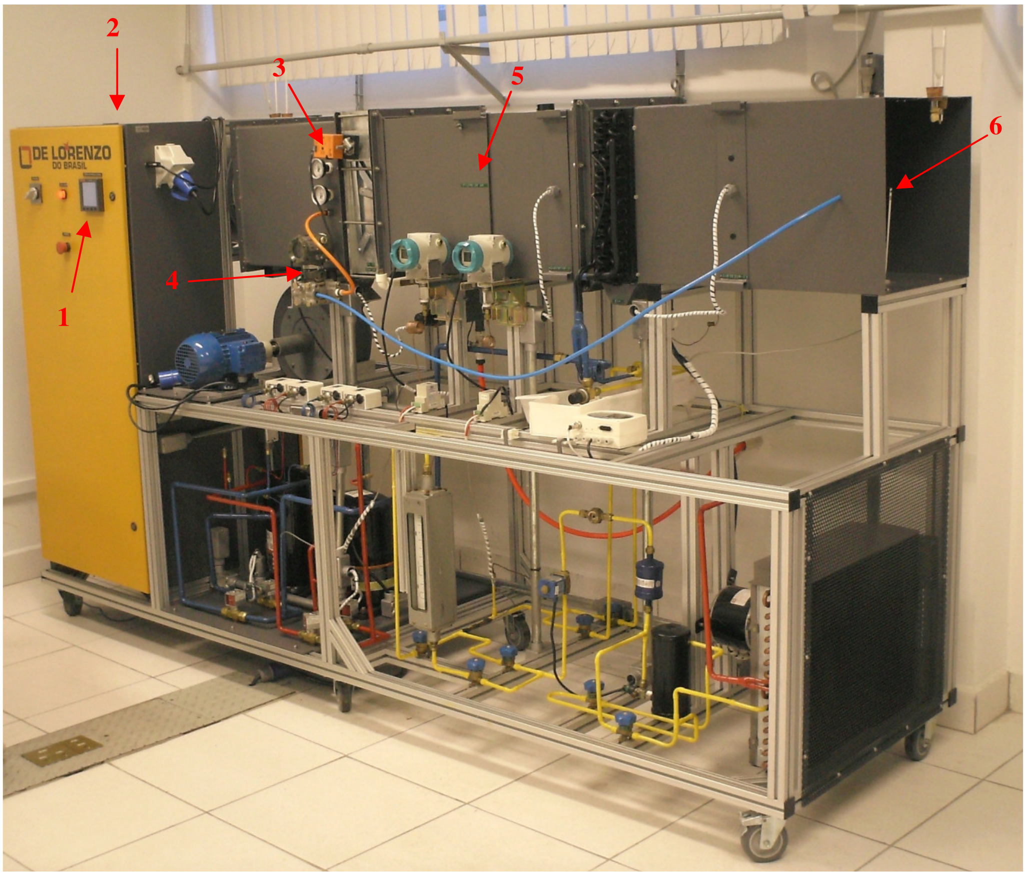

Figure 1. Test bench.

Reference numbers are:

1 => PLC (Programmable Logic Controller) and energy measuring instrument;
2 => Axial fan motor with the frequency inverter;
3 => Electrical damper driver;
4 => Manometer;
5 => Air filter;
6 => Speed transducer and air temperature.

Figure 2 illustrates the supervisory system command screen of the test bench, in which the rehearsals were carried out with the axial motor-ventilator system.

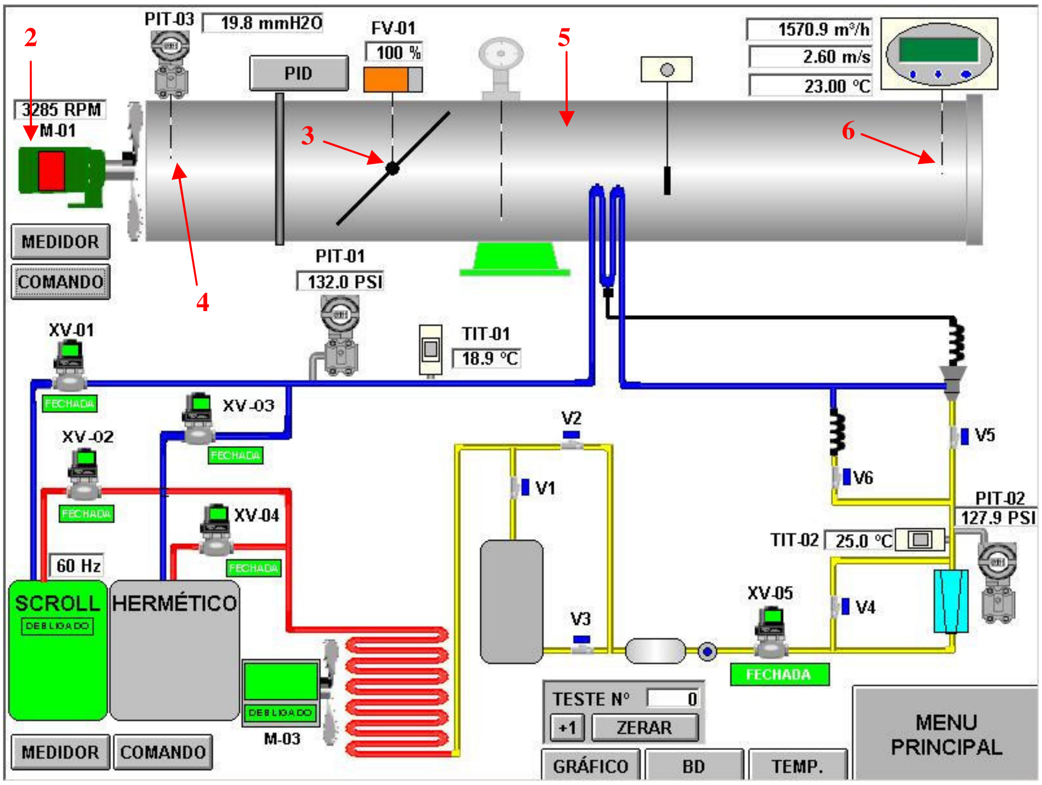

Figure 2. Command window of the bench tests.

2.1. Used materials

- A frequency inverter manufactured by WEG - model CFW-09, with a MODBUS port;
- A PLC made by Schneider Electric provided with an Ethernet card, serial port, power supply, 16 inputs and 8 outputs with MODBUS port;
- A multifunction Energy gauge made by ABB, with MODBUS protocol;
- An asynchronous three phase motor made by WEG, 2 poles, 1 ½ CV, 220/380 V - high efficiency type;
- An axial ventilator made by POLUTEC, model Axial ATD 350/8 with a maximums airflow of 5.000 m³/h, pressure of 24 mmCA, diameter 350 mm 1,5 cv motor compatible with 3600 rpm;
- An electrical damper, made by TROX DO BRASIL, model Varicontrol, code TVR-Easy, compatible with the ventilator pressure flow, variable between 0 to 100 %, furnishing indication of current position on the supervisory software screen;
- An air speed/temperature transducer, manufactured by Tecnovip, code TECCVT-200-HO300, 0 - 20 m/s, temperature range 0 – 50 ºC, with an analogical output 4-20 mA, Precision 0,1 m/s + 2 %, 24 VDC power supply, Protection IP65, screw connections of 1,5 mm^2
- A differential pressure transducer, made by YOKOGAWA, code EJA210 DM, Piezometer with stainless steel body, pressure range of 0 - 300 mmH_2O, 24 VDC power supply with 0,5% precision, protection IP 65 pipe compatible, pressure and power, output signal of 4 to 20 mA, TAG PIT-03;
- Air filters, made by Trox do Brasil, model Trox Technik, compatible with the installation.

2.2. Methods used for the acquisition of data and calculus of the variables

The electrical power required to operate the fan-motor assembly can be expressed by Eq. (1) (VIANA, 2004). That magnitude was determined and compared with the one directly measured by the test bench instruments for an estimate of the total gain by the assembly.

$$P_{el} = \frac{\gamma \cdot Q \cdot \Delta p_{t(\,fan\,)}}{\eta total} \tag{1}$$

where:
P_{el} is the electrical operating power of the set fan-motor (W).
γ is the specific air weight at the temperature of the rehearsal (N/m³).
Q is the volumetric air mean flow, corrected to normal condition (Nm³/s).
$\Delta p_{t(\,fan\,)}$ is the air total pressure difference between the admission and exhaust of the fan-motor assembly at the installation (mCar).
$\eta total$ is the total gain of the fan-motor assembly in the installation.

The mean airflow was calculated by the product of the mean speed and the outflow transversal section area. The maximum speed of the air outflow was obtained by the transducer positioned at the center of the transversal section, located at the exhaust of the test bench. The profile of the outflow duct section is of a square type with an area of 0,1681 m². Starting from this point, the mean speed was calculated where, for the Reynolds Number range, was 1,4 x 10^4 and 7,8 x 10^4 and the mean correction factor calculated as 0,81 (Fox et al, 2006). For the universalization of the results, the rehearsals were processed with an ambient dry air bulb temperature circa of 23 ºC and at the mean barometric pressure of 712 mmHg (95 kPa), the mean volumetric airflow was reduced to the normal condition (0 ºC e 760 mmHg).

Eq. (2) (VIANA, 2004) defines the difference of the total pressure.

$$\Delta p_{t(\,fan\,)} = \rho \cdot g \cdot Hp + \rho \cdot \frac{V^2}{2} \tag{2}$$

where:
$\Delta p_{t(\,fan\,)}$ is de difference of the air total pressure between admission and exhaustion of the fan-motor assembly at the installation (Pa).
ρ is the air specific mass at the temperature of the rehearsal (kg/m³).
g is the gravity acceleration (m/s²).
Hp is the charge loss in the suction line and the pressure at the installation (mCar).
V is the mean corrected speed in the outflow section at the installation (m/s).

The specific mass of the fluid, considered as a perfect gas, the temperature and pressure functions were determined by Eq. (3), according to Sonntag et al (2004).

$$\rho = \frac{p}{R \cdot T} \tag{3}$$

where:
p is the local atmospheric pressure (kPa).
ρ is the air specific mass at the temperature of the rehearsal (kg/m³).
R is the air constant (kJ/kg.K).
T is the air absolute temperature (K).

The hydraulic power of the ventilator was calculated by Eq. (4) (VIANA, 2004).

$$P_h = \gamma \, Q \, \Delta p_{t(\,fan\,)} \tag{4}$$

where:
Ph is the hydraulic power (W).
$\Delta p_{t(\,fan\,)}$ is the difference of the air total pressure between admission and exhaustion of the motor-ventilator assembly at the installation (mCar).

The efficiency of the fan-motor assembly at the installation was obtained by Eq. (5) (VIANA, 2004).

$$\eta_{total} = \frac{P_h}{P_{el}} \tag{5}$$

The charge loss between the exhaust of the fan-motor assembly and the test bench outflow was considered as a charge loss at the installation originated by the tubing itself and the devices implanted in the assembly, i.e., two electrical resistors, the evaporator and the air filter, was measured directly by a manometer. The charge loss versus airflow shown in the graphics is the difference presented by the symbol Δp measured in mmH_2O.

3. RESULTS AND DISCUSSION

The results are herein presented and commented by means of graphics. A table of the experimental uncertainty was also included to the results in order to determine their precision. Based in Holman (2001) and Cruz et al. (1998) the uncertainties (μ) were determined for each primary measure, that is, obtained without calculus but read directly from the instrument and, thereon, the uncertainty results were calculated. Tab. 1 shows the results.

Table 1. Experimental uncertainties.

Measure	μ
Atmospheric pressure	± 0,5 mmHg
Area	± 0,0005 m^2
Electrical Power	± 3,5 W
Pressure difference	±0,09 mm H_2O
Speed	± 0,03 m/s
Airflow	± 0,03 Nm^3/s
Temperature	± 0,3 ºC
Air specific mass	± 0,58 kg/m^3

3.1 Airflow controlled by damper

The graphics of the operating electrical power, charge loss and the system efficiency for the first method, obtained by the fan-motor assembly, are presented in 3.1.1 and 3.1.3

Taken into account just the filter influence when the valve provides the flux control, the fan motor is actuated by direct start and remains at the frequency of 60 Hz, as supplied by the electrical supply, until the completion of the experiment. One can see in Figure 3.1.1 that by this manner of control, considering the maximum reached flow, the relations of flow by required power is 1550 Nm^3/h for 1250 W, which corresponds to 1,24 $Nm^3/h.W$ without the air filter. With a clean filter, that relation became 1280 Nm^3/h for 1320 W, that is: 0,70 $Nm^3/h.W$. In this case, while the considered flow was reduced with the airflow control by the DAMPER, that airflow/power relation decreased linearly, reaching approximately 500 Nm^3/h for 1550, that is: 0,32 $Nm^3/h.W$, either for the clean or the dirt filter.

It is also possible to observe in Figure 3.1.2 that the greatest airflow is reached by this method of control for the system same charge loss value without the filter. Otherwise, with a dirt filter the system presented the smallest airflow.

As refers to the system efficiency, one can notice in Figure 3.1.3, as expected, that it diminished when the filter was installed and decreased even more when the filter was dirty.

All of those observations are clearly visible when the airflow control is carried out by the second method.

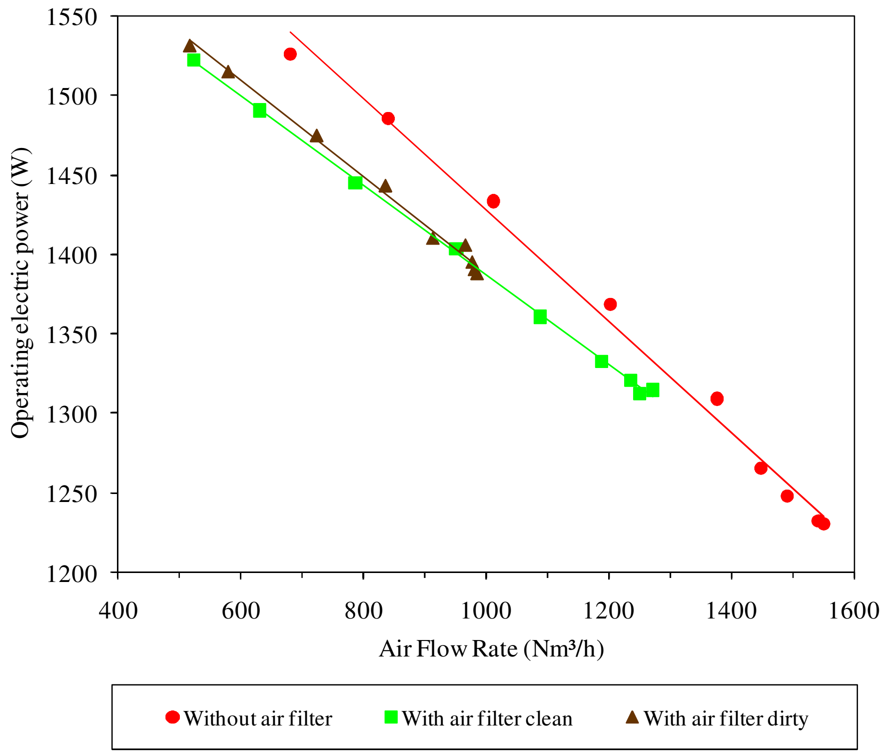

Figure 3.1.1. Electrical power in function of airflow by the first method control.

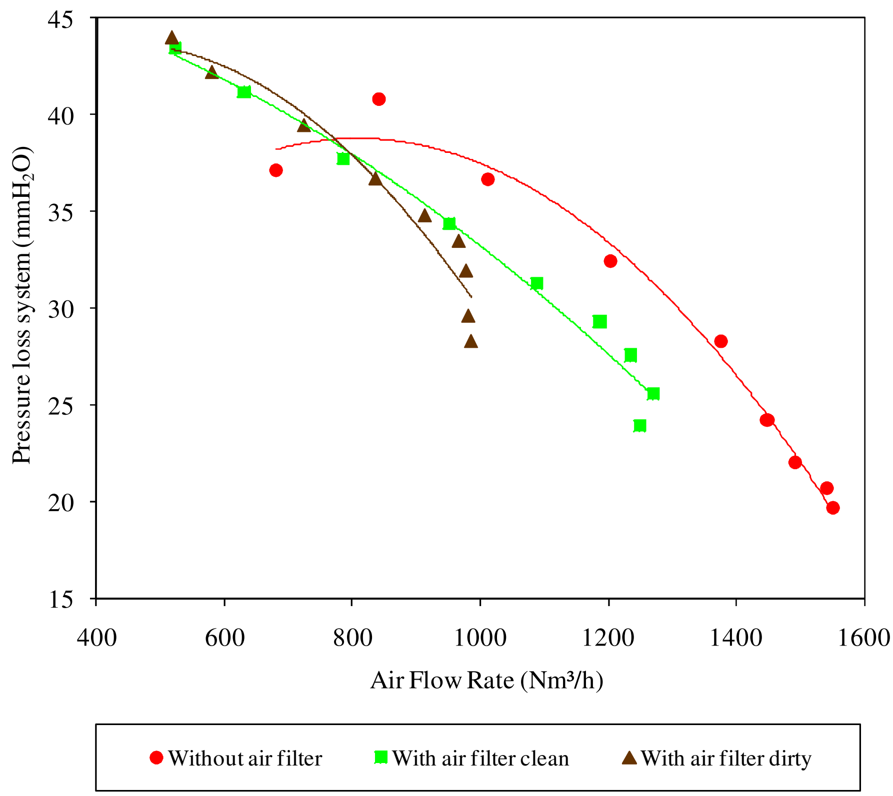

Figure 3.1.2. System pressure drop in function of airflow by the first method control.

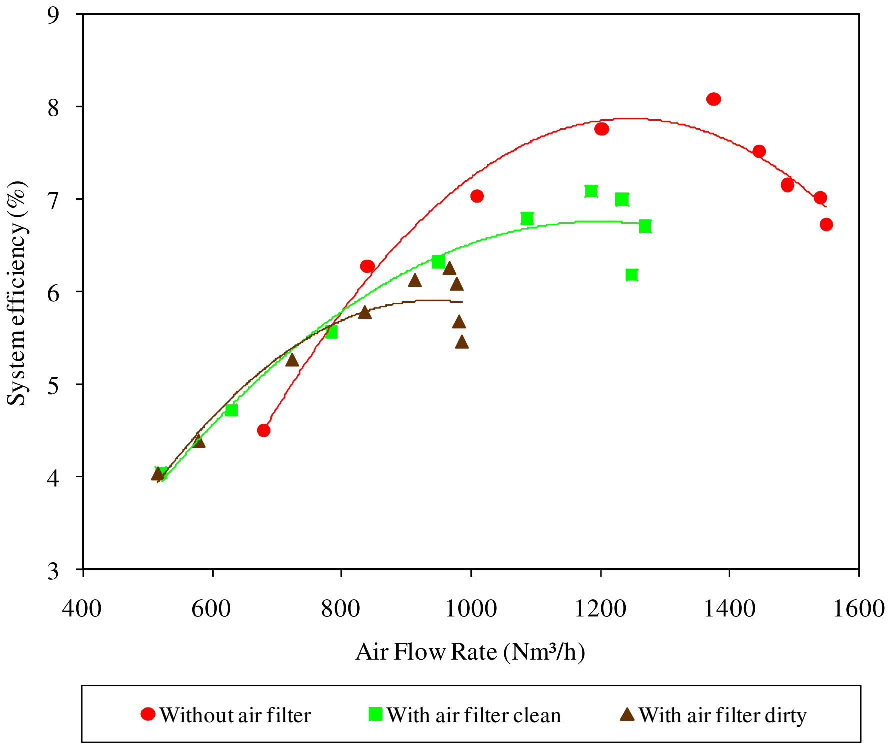

Figure 3.1.3. System efficiency in function of airflow by the first method control.

3.2. Airflow controlled by frequency inverter

The graphics of the operating electrical power, charge loss and the system efficiency for the second method, obtained by the fan-motor assembly, are presented in 3.2.1 and 3.2.3.

Considering only the influence of the filter when the flow control is made by the frequency inverter and also considering the maximum airflow, one can observe that without it the airflow reaches approximately 1500 Nm³/h, requiring a power of approximately 1250 W (the relation airflow/power equals to 1,2 Nm³/h.W). With a clean filter, the relation is 1250 Nm³/h for 1400W, which is equal to 0,89 Nm³/h.W. When the filter is dirty that relation of required airflow by power is quite less and equal to 1000 Nm³/h for 1500 W, which corresponds to 0,67 Nm³/h.W. While the considered airflow was reduced, the frequency inverter for that airflow/power relation growed exponentially, reaching approximately 220 Nm³/h for 20 W, that is: 11 Nm³/h.W, either for a clean or dirty filter.

By this control method one can perfectly notice, observing Figure 3.3.2, that for the same value of charge loss in the system, the maximum value can be obtained without the filter. Once using the filter, the airflow was much reduced; however, when the filter was dirty the flow became even more reduced. It was also observed that for the same airflow the charge loss increased when the filter was used and it augmented even more when it became dirty.

In relation to the system efficiency it can clearly be observed in Figure 3.3.3 what was expected: the efficiency diminished when the filter was installed and decreased even more when it was dirty.

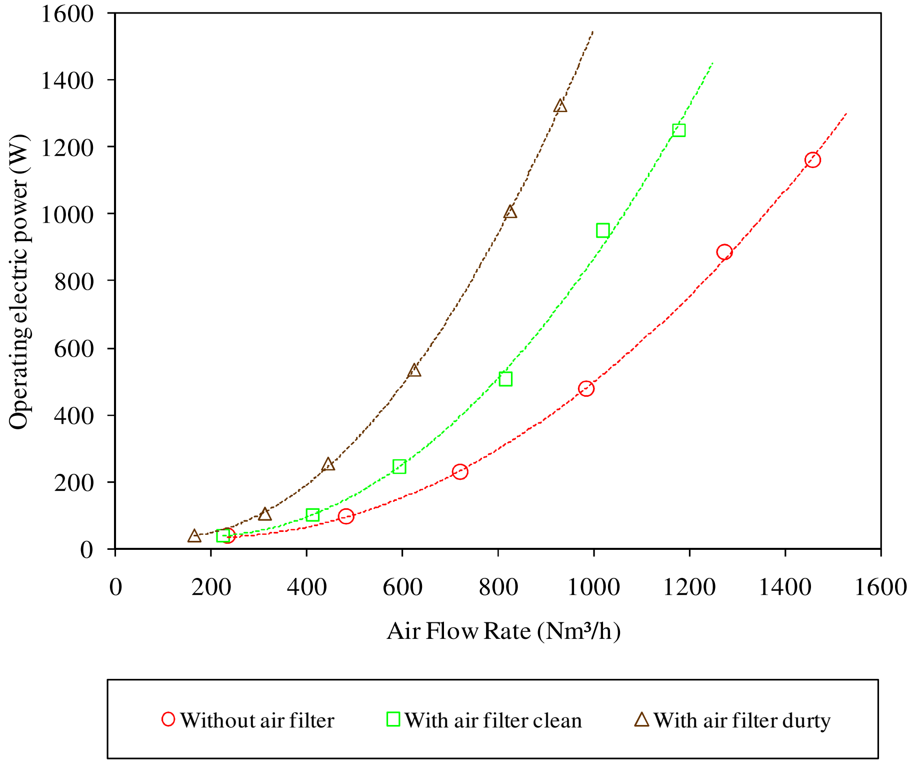

Figure 3.2.1. Operating electric power in function of airflow, according to the second method control.

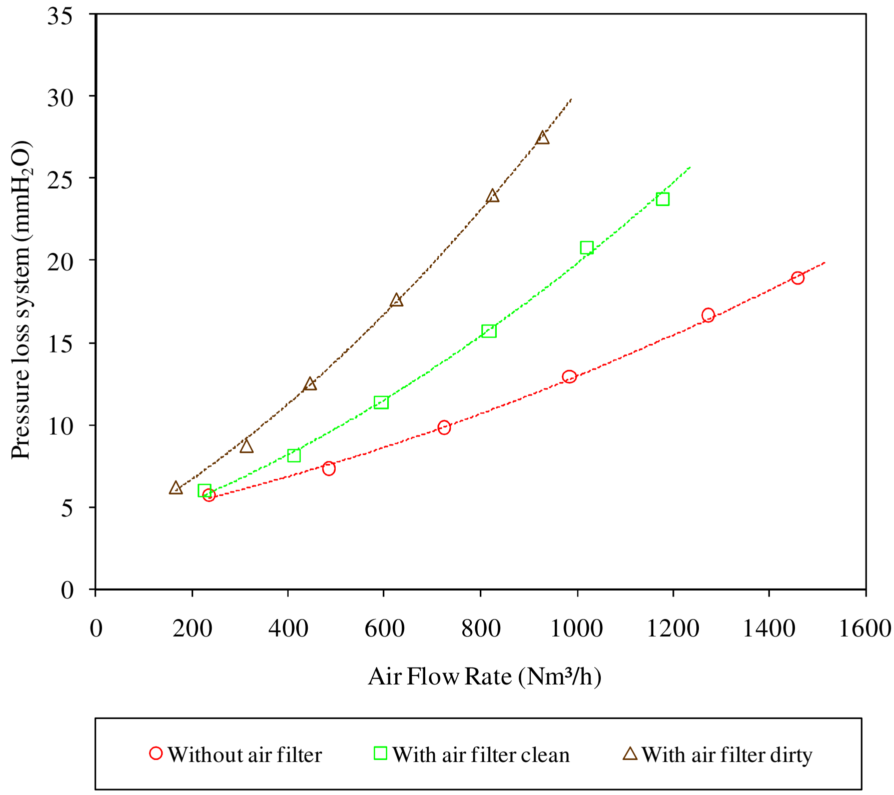

Figure 3.2.2. System pressure drop in function of airflow, according to the second method control.

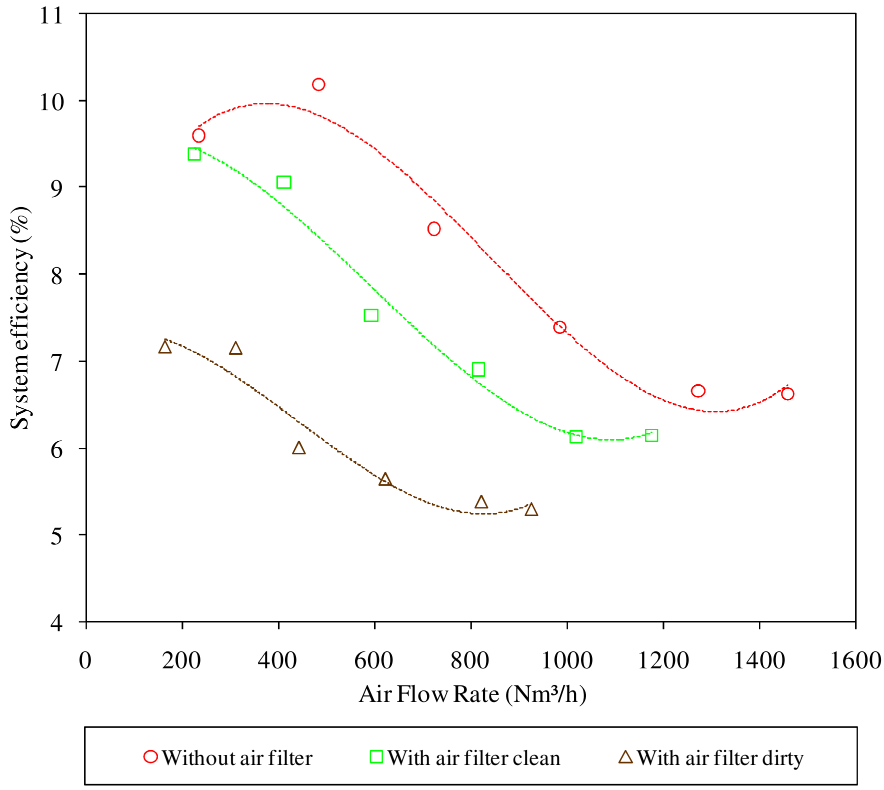

Figure 3.2.3. System efficiency in function of airflow, according to the second method control.

Note: The present results look somewhat low due to the effect of several devices installed after the fan such as the evaporator, electrical resistors, air filter, humidifier and some measuring instruments that could cause loss of charge enough to reduce the global efficiency of the system. Notwithstanding that, those losses do not influence in the final results as all of the rehearsals where carried out under the same conditions.

It was also observed that when the inverter operating frequency was nearly 45 Hz, i.e. 75% of the total operating capacity (60 Hz), that was the point where the efficiency curves crosses with the DAMPER curve. From the total efficiency point of view, increasing that frequency the system with the inverter showed less efficiency than that of the DAMPER system. However, that does not mean that starting from that frequency the inverter system consumes more energy than the DAMPER. That happened because in the calculus of total efficiency the hydraulic power is taken into account, which is the function of the difference of the total pressure; as for the DAMPER, that pressure difference is much higher than that of the inverter and, in consequence, also its total efficiency, even when consuming more energy for the same airflow. Being that the case, one should consider the airflow curve by required power, as the greater that relation the more economic becomes system when only the airflow is needed, notwithstanding the pressure presented in the system.

4. CONCLUSION

It can be seen in the graphics produced by the rehearsals that depending upon the employed airflow control and the state of the air filters, there is in fact a difference by the power required by the fan-motor assembly for the same obtained airflow.

Without the air filter, it is evident that the system presents a higher performance; however, it is well known the necessity of maintaining the circulating air quality. Therefore, its removal is not recommended at all. As such, the system with a clean air filter where the airflow is controlled by a frequency inverter is, considering the analyzes of this study, the most adequate and economical process where the airflow presents a significant variation or the ventilating motor frequency being at least less than 55 Hz. On the contrary, when there is a need to operate the motors at full power, the DAMPER system presents better results as refers to flow/power required relation.

With this work, which resulted from a technical compared analysis, the advantages and disadvantages are demonstrated from the point of view of energetic efficiency for both the methods to control airflow. Based on it one can define which method should be employed according each case so to obtain more economy of electrical energy.

This work also demonstrated the importance, as refers to energetic efficiency, of keeping the equipment in good operating conditions. It is hoped that with this knowledge the practice of preventive maintenance be diffused and the main objective of reaching energy efficiency as well.

5. ACKNOWLEDGEMENTS

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[1] ***Universidade Estadual Paulista*** *- University of São Paulo State;*
[2]***Centrais Elétricas Brasileira S.A.***

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7. RESPONSIBILITY NOTICE

The authors are the only responsible for the work presented in this paper.