Proceedings of COBEM 2007
19th International Congress of Mechanical Engineering
November 5-9, 2007, Brasília, DF

EXPERIMENTAL ANALYSIS OF AN INDUSTRIAL TUMBLER CLOTHING DRYER

Borgert J. A.
Pontifical Catholic University of Parana - PUCPR
borgertnfl@yahoo.com.br

Velásquez J. A.
Pontifical Catholic University of Parana – PUCPR
Paraná Federal University of Technology - UTFPR
jose.velasquez@pucpr.br

Molinetti A. L.
Pontifical Catholic University of Parana - PUCPR
andersonctb@yahoo.com.br

Abstract. *The drying of clothes is a very expensive process that consumes a lot of time and energy, especially in the case of hotels and hospitals. The search for a compromise between the time required for a satisfactory drying process and the energy consumed in this operation is a non trivial task that demands experimental and theoretical analyses. In this work an industrial tumbler clothes dryer was studied in order to assess the influence of several variables on the drying time. The fabric used during the experiments was cotton and the dryer was loaded with approximately 11 kg of dry clothes (17 kg of wet clothes). The results show that adjusting the fan speed for air supplying it is possible to save energy without increasing the drying time. The experiments show also that the drying time can be estimated knowing both the average mass flow rate of evaporated water and the initial water content in clothes. These parameters allow to schedule a proper stopping time, thus preventing unnecessary further operation of the dryer and, consequently, saving energy.*

Keywords: *Tumbler dryer, clothes drying process, energy saving in laundry processes.*

1. INTRODUCTION

Industrial tumbler clothing dryers are intensive consumers of energy and frequently remain in service 24 hours a day. Their use in hospitals, hotels and laundry services is responsible for a fraction of the energy bill that may be as high as 90%, according to Deans (2001). In textile industry the energy expended in the drying process may represent up to 12% of the overall energy costs of the plant (Shelton *et al.*, 1999). In despite of that, manufacturers frequently do not devote especial attention to energy consumption of the dryers and just minor control is included in their construction. This occurs because the more efficient dryers would be more expensive and this could immediately affect the decision of buying it or not. On the other hand, the operational costs associated to energy consumption will appear later, during the use of the dryer and will be paid by the owner who bought the dryer, therefore not affecting the manufacturer. So, nowadays it is common that the clothing dryers do not satisfy normative standards for energy consumption.

Several authors have studied tumbler clothing dryers, suggesting ways to increase their energy efficiency. Lambert *et al.* (1991), Conde (1997) and Deans (2001) studied tumbler dryers, similar in operational principle to the one chosen for the present study, although with considerable differences in load capacity and air mass flow rate, thus turning inappropriate to extent their conclusions to the case of the present work. These differences in load capacity are evident from the data reported in Tab. (1).

In their papers Lambert *et al.* (1991), Conde (1997) and Deans (2001) presented mathematical models based on the use of water activity, although some differences in their formulation can be observed, particularly in the calculation procedure of the leaving drum air temperature and moisture content. All these models focus computational simulation for predicting the behavior of the dryer, however, since this paper is devoted to experimental analysis a more simple approach will be used here. Besides that, these authors also analyzed the reduction in energy consumption by means of the recirculation of part of the humid air that leaves the drum, showing that it is possible to recover partially the energy contained in this flow and so to improve the energy efficiency of the dryer.

Braun *et al.* (2002) and Ameen and Bari (2004) analyzed a tumbler dryer operating in a closed scheme, in which a heat pump was used to condense steam from the humid air exiting the drum. Although this approach allowed reducing the energy consumption in about 15%, the initial and maintenance costs as well as the difficulties associated to the lint removal constitute a serious drawback for the use of such a system, demanding a thorough economical evaluation.

Another device considered in the literature for reducing energy consumption in drying processes is the nozzle dryer, which does not require heat addition. Nevertheless, the difficulties for lint removal and the need of vacuum generation turn this option non attractive for the drying of clothes (Shelton *et al.*, 1999).

Table 1.- Load capacity of tumbler dryers considered in literature.

	Mass of dry clothes (kg)	Mass of wet clothes (kg)
Lambert	4.00	5.40
Conde	8.32	12.48
	5.72	8.58
Deans	5.00	----
Present work (nominal data)	12.00	up to 18.00

2. THE TUMBLER DRYER AND THE DRYING MODEL

Most tumbler clothing dryers work according to the so called open scheme, in which atmospheric air enters the dryer and warm humid air is discharged to the ambient, carrying the water that was in the clothes. Nowadays, all clothes dryers manufactured in Brazil operate according to this scheme and by this reason it was chosen for the study presented in this paper.

Figure (1) shows the main components of a tumbler dryer operating in an open cycle. A fan is used to blow the atmospheric air into the electric heater, where the air temperature is raised to 80 – 110°C. This hot air is then conducted towards the drum, where the wet clothes are in continuous tumbler movement. Due to the heat transfer as well as to the mass diffusion, the water content of the wet clothes is evaporated and the air moisture increases. Then, finally, this humid air leaves the drum and is discharged to the ambient. The operation proceeds until a moment when no additional water can be removed from the clothes and the air leaving the drum has almost the same moisture content that it had when entered the fan.

The mass flow rate of the water evaporated in the drum can be evaluated knowing the air mass flow rate and the humidity ratio (water content) of the air flow entering and leaving the drum, according to the following equation

$$\dot{m}_{w,evap} = \dot{m}_a(\omega_{out} - \omega_{in}) \tag{1}$$

where ω is the humidity ratio; $\dot{m}$ is the mass flow rate; the subscripts *w,evap* and *a* refer to evaporated water and air, respectively, while the subscripts *in* and *out* stand for the conditions at the entrance and at the exit of the drum, respectively.

The mass of water evaporated from the clothes between two measurements elapsed by a time Δt $(\Delta m_{w,evap})$ can be evaluated as follows

$$\Delta m_{w,evap} = \dot{m}_{w,evap} \cdot \Delta t \tag{2}$$

The humidity ratios of the streams entering and exiting the dryer are calculated from the measured dry and wet bulb temperatures as follows (see Moran, 1999, page 661),

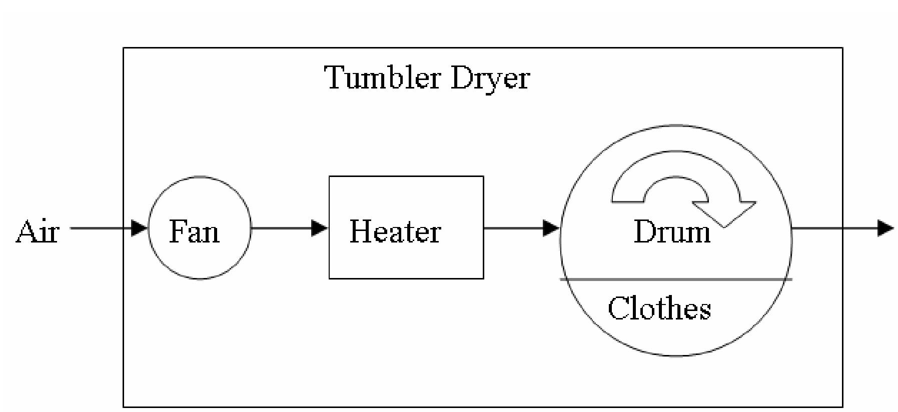

Figure 1. The tumbler dryer operating in an open cycle.

$$\omega = \frac{h_{a,TWB} - h_{a,TDB} + \omega' \cdot h_{w,evap,TWB}}{h_{w,vap,TDB} - h_{w,liq,TWB}} \quad (3)$$

were

$$\omega' = \frac{0.622 \cdot p_{sat,w,TWB}}{p - p_{sat,w,TWB}} \quad (4)$$

and p_{sat} is the saturation pressure of water, p is the ambient pressure, h is the enthalpy and the subscripts *TWB, TDB, evap, vap, liq* stand for wet bulb temperature, dry bulb temperature, evaporation, saturated vapor and saturated liquid, respectively.

The relative humidity φ can be evaluated by means of the following expression,

$$\varphi = \frac{p_w}{p_{sat,TDB}} \quad (5)$$

where p_w is the partial pressure of steam in the humid air, given by

$$p_w = \frac{p \cdot \omega}{\omega + 0.622} \quad (6)$$

The total mass of water evaporated from the clothes during a drying process started at a time 0 and proceeding until a time t is obtained as the sum of the mass increments calculated through Eq. (2), as follows

$$m_{w,evap,total} = \sum_{j=0}^{t} \Delta m_{w,evap,j} = \sum_{j=0}^{t} \dot{m}_a (\omega_{out,j} - \omega_{in,j}) \cdot \Delta t \quad (7)$$

The air mass flow rate was evaluated taking into account that the total mass of water evaporated from the clothes during the drying process is equal to the clothes mass difference ($\Delta m_{clothes}$), which is determined experimentally by weighing the clothes before and after the drying operation

$$m_{w,evap,total} = \Delta m_{clothes} \quad (8)$$

$$\dot{m}_a = \frac{\Delta m_{clothes}}{\Delta t \cdot \sum_{j=0}^{t} (\omega_{out,j} - \omega_{in,j})} \quad (9)$$

3. EXPERIMENTAL PROCEDURE

The experiments were conducted using an average load of 17 kg of wet clothes, which contained approximately 11 kg of dry clothes made of a 100% cotton fabric. Figure (2) shows the measuring points where the instruments were allocated to obtain experimental data. Type T thermocouples were used to measure the temperature at points 1 through 4. These thermocouples were calibrated prior to the tests and when possible they were installed each attached to a copper mass, as shown in Fig. (3). The copper masses were used in order to damp spurious oscillations of the temperature, this way smoothing the measured curves. Figure (4) shows the schematic of a simple device, which was built for measuring the wet bulb temperature at points 1 and 4 of Fig. (2). In this device a continuous supply of water for embedding the cotton cover that involves the thermocouple was obtained by using a physiologic serum dropper. It is worth mentioning that a special care was taken for placing the thermocouples on the center line of the ducts and that all temperature data were collected by means of an electronic acquisition system, which was set to make one reading per minute.

In order to calculate the air mass flow rate, the average flow speed at section 5 (Fig. 2) was evaluated. This was made by measuring local air velocities with a hot wire anemometer and then determining the area averaged value. The air mass flow rate was determined by means of the following expression,

$$\dot{m}_a = \frac{\rho \cdot \vec{V}_{avg} \cdot A}{1+\omega} \tag{10}$$

where ρ is the density of the moist air, $\vec{V}_{avg}$ is the average flow velocity at section 5 and A is the cross section area of the duct. This mass flow rate can be, therefore, compared to that calculated through Eq. (9).

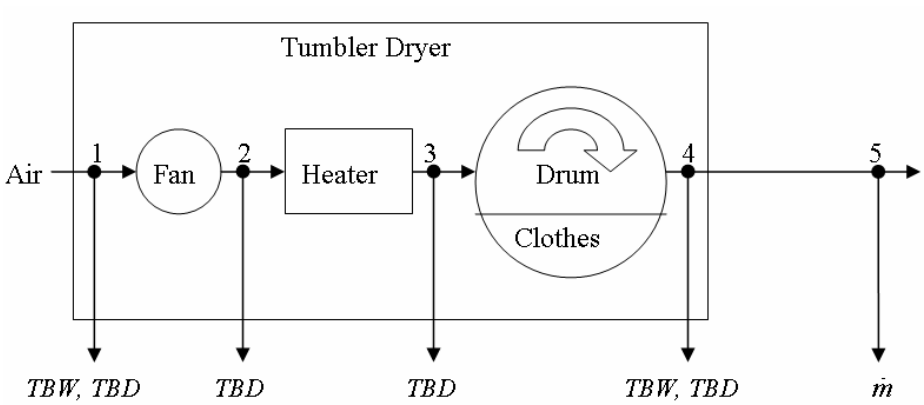

Figure 2. Schematic of the instrumentation.

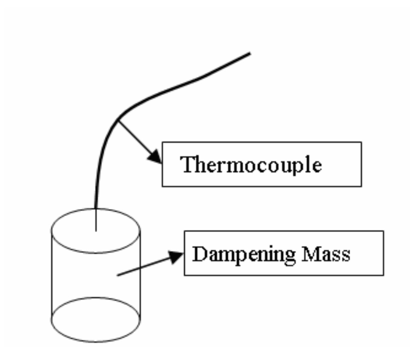

Figure 3. Thermocouple with damping mass.

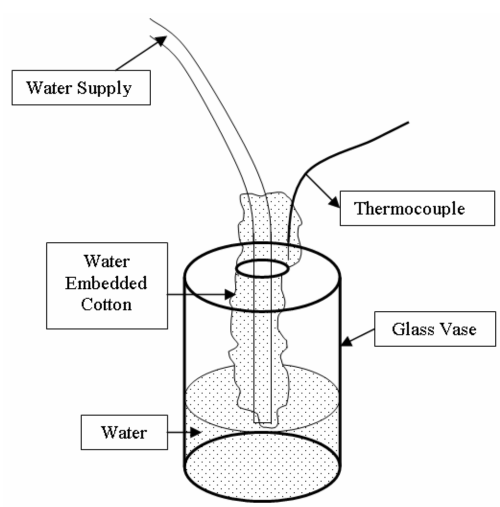

Figure 4. Schematic of the wet bulb temperature measuring device.

The fan speed was controlled by using a frequency inversor, this way adjusting the inlet air mass flow rate. During the experiments three rotating speeds were investigated: 3400 rpm, 2900 rpm and 2500 rpm, the first one being the nominal value. In addition, it was investigated the operation of the dryer in two conditions: when the dryer was started after a long time stop (cold condition) and when the dryer was started after a rapid stop for reloading (hot condition). In this latter condition a 10 minutes period was necessary to unload the dryer, wet the clothes, centrifugate until the desired water content and reinsert the clothes into the dryer. Due to the limited capacity of the centrifugation machine, it was required to centrifugate the entire charge of wet clothes in three parts, destining 2 minutes for each one.

The criterion for stopping the tests was based on the dry bulb temperature of the air leaving the drum (point 4 of Fig. 2). When this temperature reached 5ºC above the set point value the dryer was turned off. The set point value corresponds to the temperature of the humid air at the end of the initial stage, in which this temperature experiences a fast growing as can be seen in Fig. (5). It is important to observe that other authors (Lambert et. al. (1991) and Conde (1997)) used the equality between the humidity ratios at the entrance and at the exit of the drum as stopping criterion, however, from early tests it was found that this condition does not occur in the tested dryer, even though the clothes inside the drum were already dry.

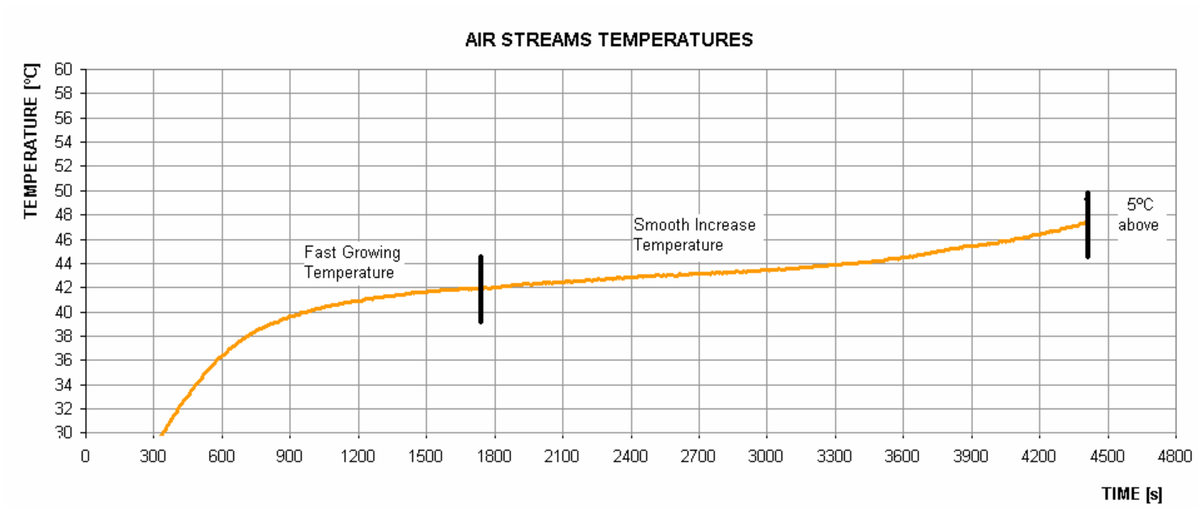

Figure 5. Temperature criterion for stopping the experiment.

4. RESULTS

Table (2) shows the conditions at the beginning of the drying operation. Each run was identified with a code started with a letter C or H. A run name started with the letter C refers to a drying process started with the dryer cold, while the H letter refers to a run subsequent to another one, so that the dryer was considered hot. The results for all these tests are shown in Tab. (3).

Table 2 – Operational conditions in tests.

Run	Fan Speed (rpm)	Wet Mass of Clothes (kg)	Ambient Temperature (ºC)	Ambient Relative Humidity (%)
C13	2500	17.134	22.9	65.9
H13	2500	17.100	24.2	66.7
C12	2900	17.152	25.4	65.9
H12	2900	17.106	24.8	66.8
C20	3400	17.176	25.6	63.0
H120	3400	17.052	25.5	65.5
H220	3400	17.062	25.8	66.0
C23	3400	17.094	20.2	63.5
H123	3400	17.104	21.3	61.9
H223	3400	17.182	21.7	61.9

Table 3 – Results of the drying tests.

Run	Initial Mass of Dry Clothes (kg)	Final Mass of Dried Clothes (kg)	Evaporated Water Mass (kg)	Drying time (s)
C13	10.740	10.852	6.282	4415
H13	11.085	10.845	6.255	4605
C12	10.796	10.883	6.269	4465
H12	11.259	10.777	6.329	4490
C20	10.856	10.816	6.360	4580
H120	10.816	10.852	6.200	4405
H220	10.852	11.126	5.936	4260
C23	11.286	11.186	5.908	4265
H123	11.186	10.952	6.062	4360
H223	10.952	11.004	6.178	4370

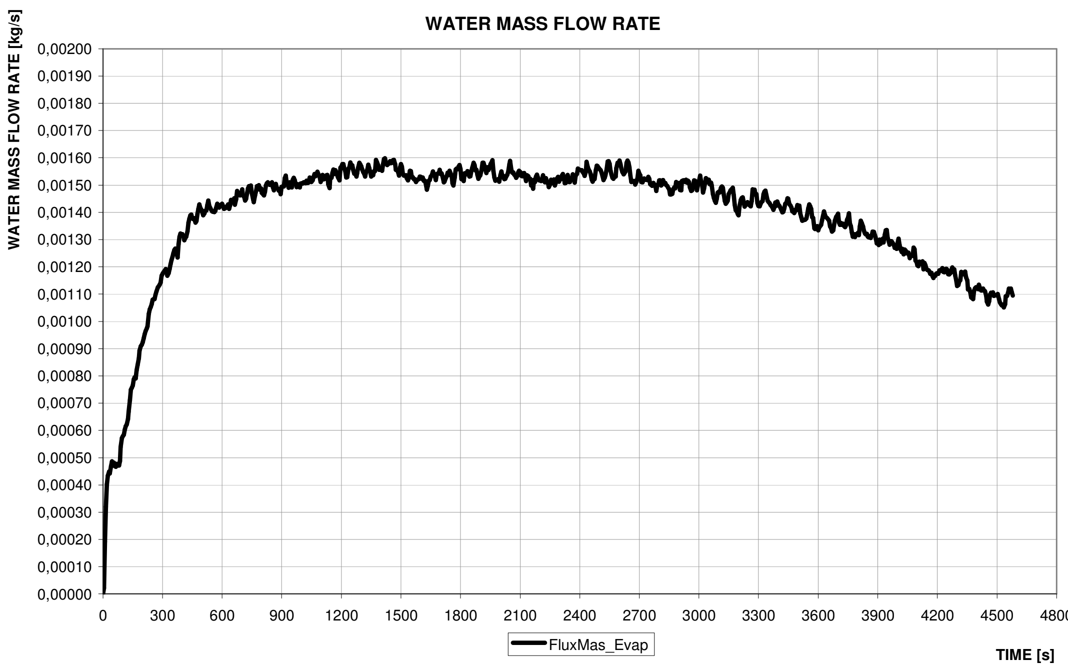

Figure 6. Typical curve of evaporated water mass flow rate.

Figure (6) shows a typical curve of the evaporated water mass flow rate, calculated according Eq. (1). As can be seen, the curve starts with low mass flow rate values because the heater needs some time to achieve its normal temperature condition. When the heater is operating in a steady-state condition the evaporation rate curve reaches a nearly constant value. At the end of the drying process the water extraction from the clothes becomes more difficult and the evaporation rate diminishes. Nevertheless, the clothes are considered to be dried even before the evaporation rate becomes nil.

In Tab. (4) a comparison is made of the air mass flow rates calculated by means of Eq. (9) and Eq. (10) (the last is called the measured value). The results show that the differences in air mass flow rate are not higher than 4,5%, so that Eq. (10) can be used to estimate the mass flow rate. In addition, it can be noticed that an increase of 100 rpm in the fan speed raises the air mass flow rate in approximately 0.005 kg/s. It is worth noting that the conditions C23, H123 and H223 were tested under ambient temperatures 4 to 5ºC lower than the other 3400 rpm conditions (see Tab. 2) and consequently there was an increase in the air mass flow.

Table (5) shows a comparison between the mean mass flow rate of evaporated water, which was obtained dividing the total mass of evaporated water by the drying time, and the averaged mass flow rate of evaporated water, calculated through a time averaging of a curve similar to that showed in Fig. (6). As can be seen, the mean mass flow rate of evaporated water is very near to the averaged value, and there is a little variation in the averaged value for all experiments (less than 1.3%). Therefore, when ambient conditions are nearly constant, just a minor error should be made if the mean value would be used to predict the drying time. It is interesting to notice that the fan speed seems to cause no difference on the mass flow rate of evaporated water.

Table 4 – Air mass flow rate results.

Run	Measured mass flow rate (kg/s)	Measured average velocity (m/s)	Calculated mass flow rate (Eq. 9) (kg/s)	Error
C13	0.1673kg/s	4.84m/s	0.1684kg/s	0.7%
H13			0.1675kg/s	0.1%
C12	0.1866kg/s	5.40m/s	0.1865kg/s	0.1%
H12			0.1881kg/s	0.8%
C20	0.2105kg/s	6.09m/s	0.2068kg/s	1.8%
H120			0.2033kg/s	3.4%
H220			0.2017kg/s	4.2%
C23			0.2188kg/s	3.9%
H123			0.2157kg/s	2.5%
H223			0.2136kg/s	1.5%
Duct Diameter = 200mm, ρ = 1,1kg/m^3				

Table 5 – Evaporated water mass flow rate results.

Run	Fan speed (rpm)	Mass of evaporated water (kg)	Time for drying (s)	Mean mass flow rate of evaporated water (kg/s)	Averaged mass flow rate of evaporated water (kg/s)	Average value of averaged mass flow rates (kg/s)
C13	2500	6.282	4415	0.001423	0.001421	0.001394 Error ±1.9%
H13	2500	6.255	4605	0.001358	0.001367	
C12	2900	6.269	4465	0.001404	0.001403	0.001406 Error ±0.2%
H12	2900	6.329	4490	0.001410	0.001408	
C20	3400	6.360	4580	0.001389	0.001409	0.001413 Error ±1.8%
H120	3400	6.200	4405	0.001407	0.001432	
H220	3400	5.936	4260	0.001393	0.001420	
C23	3400	5.908	4265	0.001385	0.001387	
H123	3400	6.062	4360	0.001390	0.001401	
H223	3400	6.178	4370	0.001414	0.001427	

Table 6 – Temperature and humidity results

Run	Fan speed (rpm)	Dry bulb temperature at the dryer inlet (°C)	Dry bulb temperature at the dryer outlet (°C)	Temperature difference	Humidity ratio at the drum inlet	Humidity ratio at the drum outlet	Humidity ratio difference
C13	2500	22.9	41.0	18.8ºC ±0.7ºC	0.01270	0.02158	0.00853 ±0.00036
H13	2500	24.2	43.6		0.01413	0.02230	
C12	2900	25.3	41.5	17.2ºC ±1.0ºC	0,01476	0.02228	0.00751 ±0.00002
H12	2900	24.9	43.0		0.01501	0.02250	
C20	3400	25.6	40.9	16.2ºC ±0.9ºC	0.01427	0.02108	0.00674 ±0.00031
H120	3400	25.5	41.8		0.01482	0.02186	
H220	3400	25.8	41.7		0.01516	0.02221	
C23	3400	20.2	36.4		0.01032	0.01666	
H123	3400	21.3	38.1		0.01078	0.01728	
H223	3400	21.7	38.2		0.01110	0.01778	

Table 7 – Drying time results.

Run	Fan speed (rpm)	Mass of evaporated water (kg)	Measured drying time (s)	Predicted drying Time (s)	Drying time difference (s)
C13	2500	6.282	4415	4463	+48
H13	2500	6.255	4605	4444	-161
C12	2900	6.269	4465	4454	-11
H12	2900	6.329	4490	4497	+7
C20	3400	6.360	4580	4519	-61
H120	3400	6.200	4405	4389	-16
H220	3400	5.936	4260	4217	-43
C23	3400	5.908	4265	4198	-67
H123	3400	6.062	4360	4307	-53
H223	3400	6.178	4370	4389	+11
Mean mass flow rate 0.001408kg/s					

Table (6) allows for an analysis of the temperature and humidity results, showing that lower the fan speed greater the temperature difference between the inlet and outlet of the drum, which is due to a longer residence time of the air inside the drum. Also because of the longer residence time there is an increase in the humidity ratio difference between the entrance and the exit of the drum, which shows that more water was evaporated in the process. These observations explain the little difference among the mass flow rates of evaporated water reported in Tab. (5). It is also noted that the drying time is not affected too much by the dryer operation under hot or cold conditions, although in all hot runs but one the average mass flow rate of evaporated water was higher than in cold runs.

Table (7) shows the predicted drying time based on the mean mass flow rate of evaporated water. Except for one case the difference is no greater than 70 s. This shows that even when the dryer operation parameters vary in a broad range the drying process is equally effective and the mass flow rate of evaporated water may be adequately represented by a mean value.

5. CONCLUSIONS

An industrial tumbler dryer was tested to improve its operation. In the experiments the fan rotating speed was changed as was the initial temperature condition of the dryer. The tumbler dryer was instrumented accordingly and the results of the tests allow for the following conclusions:

- Under continuous operation, when the operation of the dryer is interrupted only for reloading, the drying time is not affected by the duration of the reloading time;
- Under standard fan speed, changes in the ambient temperature does not cause significant impact on the drying time;
- Diminishing the fan speed does not change significantly the drying time. This result represents an opportunity for energy saving since lower fan speeds require less energy.

6. ACKNOWLEDGEMENT

This project was sponsored by the Brazilian National Council for Scientific and Technological Development (CNPq), under the process number 550094/2003-2.

7. NOMENCLATURE

A Cross section area, m^2;
m Mass, kg;
$\dot{m}$ Mass flow rate, kg/s;
p Pressure, kPa;
t Time, s;
T Temperature, °C;
ϕ Relative humidity;
ρ Density, kg/m^3;
ω Humidity ratio, kg stem/kg dry air;

8. REFERENCES

Ameen, A. and Bari, S. Investigation into the efectiveness of heat pump assisted clothes dryer for humid tropics. Energy Conversion and Management, vol. 45, n.9-10, pp. 1397-1405, 2004.

Braun, J. E., Bansal, P. K. and Groll, E. A. Energy efficiency analysis of air cycle heat pump dryers. Internation Journal of Refrigeration, Vol. 25, pp. 954-965, 2002.

Conde, M. R., Energy conservation with tumbler drying in laundries. Applied Thermal Engineering, vol. 17, n.12, pp. 1163-1172, 1997.

Deans, J., The modeling of a domestic tumbler dryer. Applied Thermal Engineering, vol. 21, pp. 977-990, 2001.

Lambert, A. J. D., Spruit, F. P. M. and Claus, J., Modeling as tool for evaluating the effects of energy-saving measures. Case study: A tumbler drier, Applied Energy, vol. 38, pp. 33-47, 1991.

Moran, M. J. and Shapiro, H. N., Fundamentals of Engineering Thermodynamics. John Wiley & Sons Inc., New York, 1999.

Shelton, S. V. and Christiansen, P. J. Analysis of a nozzle condensation drying cycle. Applied Thermal Engineering, vol. 19, pp. 831-845, 1999.

9. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.