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Energy and exergy analysis applied to the evaporative cooling process in air washers

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ABSTRACT

This paper presents a study on the performance of the evaporative cooling process in air washers with the use of mass balances, energy and exergy equations. Usually, the performance of these devices is described in terms of their saturation effectiveness based on energy analysis. This method, however, is inappropriate to identify the main sources of irreversibility within the process. It is also unsuitable to give the optimum conditions for operating the system with a proper use of energy resources. Thus, an exergy analysis was developed to complement the energy analysis of the process. A mathematical model based on mass and energy balances was used to determine the water and air properties in the washer. The influence of the humidity and temperature of the air and water at the inlet, the washer length in the saturation effectiveness, and the second law efficiency was investigated. The results showed that the best condition of the intake air for obtaining thermal comfort does not coincide with the best condition for the thermodynamic performance of the system. The energy and exergy analyses were carried out simultaneously to generate the conditions required for optimum performance with minimal exergy depletion.

Analyse énergétique et exergétique appliquée au procédé de refroidissement évaporatif dans les laveurs d’air

Mots clés : refroidissement évaporatif ; laveur d’air ; efficacité de la saturation ; deuxième principe

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Nomenclature

c_P	Specific heat at constant pressure [J kg^{-1} K^{-1}]
$\bar{e}_t$	Total molar flow exergy [J mol^{-1}]
$\bar{e}_{tm}$	Thermomechanical flow exergy [J mol^{-1}]
$\bar{e}_{ch}$	Chemical flow exergy [J mol^{-1}]
E_D	Exergy destruction [W]
h_a	Air Heat transfer per cubic metre of chamber volume [W m^{-3} K^{-1}]
h_L	Water Heat transfer per cubic meter of chamber volume [W m^{-3} K^{-1}]
$\bar{h}$	Molar enthalpy [J mol^{-1}]
x_i	Mole fraction of the i-th constituent in the mixture
K_M	Mass transfer coefficient [kg s^{-1}m^{-2}]
L	Length of the air washer [m]
Le	Lewis number
P	Pressure [Pa]
T	Temperature [°C]
$\bar{R}$	Universal gas constant (8.314 J mol^{-1} K^{-1})
R_a	Ideal gas constant of the dry air (287 J kg^{-1} K^{-1})
$\bar{s}$	Molar Entropy [J mol^{-1} K^{-1}]
W	Air humidity ratio [kg kg^{-1}]
$( )^*$	Properties evaluated at the restricted dead state

Greek Symbols

ε	Saturation effectiveness
μ	Chemical potential [J mol^{-1}]
ϕ	Relative humidity
$\eta\Pi$	Second-law efficiency

Subscripts

a	Relative to the dry air
f	Relative to the saturated liquid
L	Relative to the water
v	Relative to the water vapor
0	Relative to the restricted dead state
00	Relative to the dead state

1. Introduction

Today, we face the challenge of fostering the development of new efficient machines in order to alleviate the environmental impacts associated with industrial activities. In this context, exergy analysis is seen as an indispensable tool for the design of thermal systems, whose most important objective is the efficient use of energy by identifying and minimizing the irreversibilities found in thermodynamic processes.

It is estimated that about 20% of energy consumption worldwide is attributed to heating systems, ventilation and air conditioning (HVAC). This highly correlates with the demand for thermal comfort, especially in the summer months. Most HVAC systems used today work with refrigeration cycles by vapor compression. Such systems have the disadvantage of utilizing chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) that deplete the ozone layer, besides the excessive consumption of nonrenewable energy. A promising alternative for replacing vapor compression machines, in thermal comfort applications, is evaporative cooling.

The evaporative cooling process is based on the exchange of sensible heat for latent heat. In general, a stream of hot, dry air (process air) is placed in close contact with a moist surface which can be a panel saturated with water or water sprayed by sprinklers into the system, as found in air washers. The hot air transfers the latent heat required for the evaporation of a certain amount of water and, thus, the air is cooled and humidified. The efficiency of this process increases as the weather becomes warmer and drier, or as the demand for comfort increases. The main advantages of this process compared to the vapor-compression refrigeration cycles are the following: easy installation and maintenance, low energy consumption, low cost, and a possibility of total air renewal. Even in situations where the processing air exhibits a high humidity content, evaporative coolers can be combined with desiccant rotors to offer the desired thermal comfort with very low energy consumption.

Usually, the performance of evaporative coolers is described in terms of their saturation effectiveness as determined by a balance of mass and energy equation (Camargo et al., 2005; Wu et al., 2009a, 2009b; Fouda and Melikyan, 2011; Riangvilaikul and Kumar, 2010). However, this approach fails to identify the main sources of irreversibility in the process; that is, it inadequately the system's optimum operating conditions to efficiently utilize energy resources. However, through exergy analysis it is possible to identify and quantify the system's irreversibility, focusing on the better utilization of the energy applied in the process.

Many studies on the exergy analysis of evaporative systems have been published on this topic. Chengqin et al. (2002) used the principles of exergy analysis to evaluate the performance of different evaporative systems. They studied four different configurations: direct evaporative cooling (DEC), indirect evaporative cooling (IEC), direct-indirect evaporative cooling (DIEC), and regenerative evaporative cooling (REC). Quereshi and Zubair (2003) carried out a thermodynamic study of several psychrometric processes applied to HVAC systems by using the concept of exergy. Important processes such as cooling and dehumidification, adiabatic mixing, heating and humidification, evaporative cooling, and adiabatic saturation were investigated. Muangnoi et al. (2007) performed an exergy analysis in order to determine the exergy depletion of water and air in a cooling tower. A model based on mass balances, energy and exergy was proposed and its numerical results were validated by experimental data found in the literature. Taufiq et al. (2007) made an exergy analysis for the optimization of a direct evaporative cooling process. Through an exergy balance in the system, an expression to determine the exergy efficiency of the process was obtained. The influence of relative humidity and air temperature over the performance of the process was investigated. Qureshi and Zubair (2007) performed a thermodynamic analysis of the counter flow cooling towers and the evaporative heat exchangers by using the first and the second law of thermodynamics. A parametric study was conducted to determine the influence of the wet bulb temperature and the water inlet temperature on the efficiency of the second law and on the exergy depletion. Muangnoi et al. (2008) studied the influence of the environmental temperature and humidity on the performance of a counter flow cooling tower using an exergy analysis. The water and air properties in the tower were obtained by means of a mathematical model which was validated after the experimental results. A parametric analysis of the inlet air effect, keeping the water conditions fixed, was performed. Niksiar and Rahimi (2009) presented a model for the implementation of an energy and exergy analysis over an evaporative cooling device. Mass and energy balances were applied to obtain the water and gas conditions throughout the cooling device. The results of the model were in tune with the available experimental results.

Despite its importance to desiccant air-conditioning systems, little research has been devoted to the study of air washers. Air washers have been used in conjunction with adsorptive dehumidifiers in desiccant air conditioning systems to reduce costs and energy consumption. Another important issue to consider is that many studies that evaluate the performance of evaporative heat exchangers take into account the saturation effectiveness (energy analysis) as a basic parameter, while other studies consider the second law efficiency (exergy analysis) as the most important parameter. Maximizing the second law efficiency does not necessarily imply the maximization of saturation effectiveness. Therefore, this work presents an analysis of the influence of the air temperature and humidity as well as water temperature over the saturation effectiveness and the second law efficiency, with the purpose of considering these efficiencies as parameters to better evaluate the performance of air washers. The influence of the washer length on performance parameters was also investigated. A mathematical model based on mass balances, energy and exergy equations was used to describe the conditions of water and air over the washer. The temperature and humidity data obtained from the solution of the proposed model were used to calculate the saturation effectiveness and the second law efficiency for the cooling device. The proposed model may contribute toward the development of numerical methods for more accurate simulation and evaluation of the performance of desiccant air conditioning systems that use air washers.

2. Mathematical model

Fig. 1 shows the air washer investigated with all its physical and geometrical parameters. Details of the model used for the description of the heat and mass transfer processes in air washers can be found in ASHRAE Handbok Fundamentals (1997). To determine the saturation effectiveness and the second law efficiency of the system, the prior determination of air and water conditions at the washer's output is required. To obtain these conditions, the analytical solution developed by Santos et al. (2011) was applied.

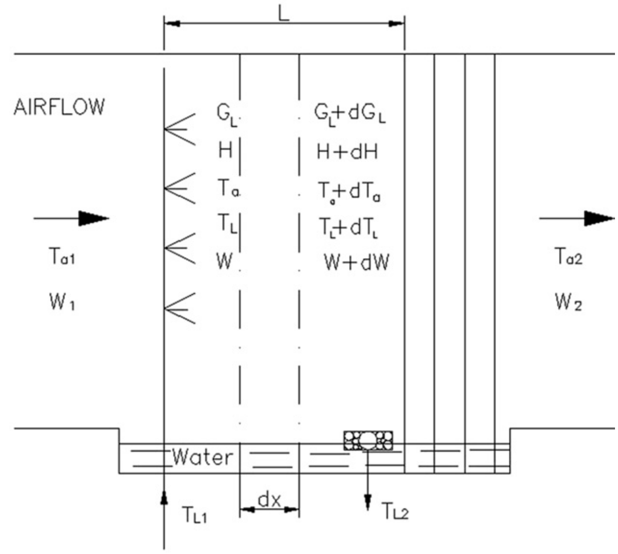

Fig. 1 – An air washer.

2.1. Exergy analysis

From an energy point of view, saturation effectiveness is an important parameter for evaluating the performance of an evaporative cooler. Therefore, for evaporative heat exchangers, the minimum temperature that can be achieved by the air flow is that corresponding to the thermodynamic wet bulb temperature at the inlet air, when the air reaches its saturation state. The saturation effectiveness denotes the ratio between the actual temperature drop recorded for the air current, and the maximum possible temperature drop achieved by the system, when the air is saturated in the outlet of the system. The saturation effectiveness (ε) is given by

$$\varepsilon = 100\left(\frac{T_{a1}-T_{a2}}{T_{a1}-T_{bu}}\right), \tag{13}$$

where T_{a1}, T_{a2} and T_{bu}, denote the dry bulb temperature of inlet air, the dry bulb temperature at the outlet, and the thermodynamic wet bulb temperature of inlet air, respectively. The saturation effectiveness provides an indication of the amount of sensible heat exchanged in the system. However, it does not show what would be the best way to carry out this energy transfer with regards to the preservation of energy resources. The exergy analysis can be applied to the system in order to perform such an analysis.

The second law efficiency is defined by applying an exergy balance to the system. To obtain the exergy balance, one needs to define the exergy associated with the moist air and liquid water currents that flows into the air washer. As described by Bejan (2006), exergy or the maximum work performed by a current consisting of various chemical species that reaches thermal, mechanical and chemical equilibriums with the environment (T_0, P_0, $\mu_{0,1}$, ..., $\mu_{0,n}$) is given by

$$\bar{e}_t = \bar{e}_{tm} + \bar{e}_{ch}, \tag{14}$$

where$\bar{e}_t$, $\bar{e}_{tm}$,and $\bar{e}_{ch}$ denote the total molar flow exergy, the thermo-mechanical exergy, and the chemical exergy of the current, respectively. The thermo-mechanical exergy is achieved only when mechanical and thermal equilibrium with the environment (T_0, P_0) is reached — called the restricted dead state, and the chemical exergy obtained when the stream condition evolves from a restricted dead state to a chemical equilibrium condition with the environment (dead state). The thermo-mechanical exergy and the chemical exergy are given by

$$\bar{e}_{tm} = \bar{h} - \bar{h}^* - T_0(\bar{s} - \bar{s}_0^*), \quad (15)$$

$$\bar{e}_{ch} = \sum_{i=1}^{n} (\mu_i^* - \mu_{0,i})\, x_i. \quad (16)$$

In Eqs. (15) and (16), * indicates that properties are being evaluated in the restricted dead state (T_0, P_0). In order to obtain the total molar exergy of moist air stream, we assumed that humid air is a mixture of ideal gases composed of dry air and water vapor. Keeping that in mind and applying Eq. (14) to each component of the mixture, we obtain

$$\bar{e}_a = x_a\left[\bar{h}_a - \bar{h}_a^* - T_0(\bar{s}_a - \bar{s}_a^*) + \mu_a^* - \mu_{0,a}\right] + x_v\left[\bar{h}_v - \bar{h}_v^* - T_0(\bar{s}_v - \bar{s}_v^*) + \mu_v^* - \mu_{0,v}\right]. \quad (17)$$

Using the ideal gas model to evaluate the enthalpy, entropy, and chemical potential of the dry air, we have

$$\bar{h}_a - \bar{h}_a^* = \bar{c}_{P,a}(T - T_0), \quad (18)$$

$$\bar{s}_a - \bar{s}_a^* = \bar{c}_{P,a}\ln\left(\frac{T}{T_0}\right) - \bar{R}\ln\left(\frac{P}{P_0}\right), \quad (19)$$

$$\mu_a^* - \bar{\mu}_{0,a} = \bar{R}T_0\ln\left(\frac{x_a}{x_{0,a}}\right). \quad (20)$$

Using again the ideal gas model, similar results can be obtained for water vapor. Combining Eqs. (18)–(20) together with their expressions for water vapor into Eq. (17), we obtain

$$\bar{e}_a = (x_a\bar{c}_{P,a} + x_v\bar{c}_{P,v})T_0\left[\frac{T}{T_0} - 1 - \ln\left(\frac{T}{T_0}\right)\right] + \bar{R}T_0\ln\left(\frac{P}{P_0}\right) + \bar{R}T_0\left[x_a\ln\left(\frac{x_a}{x_{0,a}}\right) + x_v\ln\left(\frac{x_v}{x_{0,v}}\right)\right]. \quad (21)$$

Discarding the pressure gradient along the washer ($P = P_0$) and rewriting Eq. (21) on a mass basis, we obtain the total exergy of moist air per kg of dry air as

$$e_a = (c_{P,a} + W\, c_{P,v})T_0\left[\frac{T}{T_0} - 1 - \ln\left(\frac{T}{T_0}\right)\right] + R_aT_0\left[(1 + 1.608\, W) \times \ln\left(\frac{1 + 1.608\, W_{00}}{1 + 1.608\, W}\right) + 1.608\, W\ln\left(\frac{W}{W_{00}}\right)\right], \quad (22)$$

where the subscript (00) indicates properties evaluated in a dead state condition.

The total exergy of water (Wark, 1995) can be determined by

$$e_L = h_f(T) - h_f(T_0) - T_0s_f(T) + T_0s_f(T_0) + [P - P_{sat}(T)]\, v_f(T) - R_vT_0\ln(\phi_0), \quad (23)$$

where ϕ_0 denotes the air relative humidity at environment condition.

Using Eqs. (22) and (23), the exergy balance to the washer illustrated in the Fig. 1 is given by

$$(\dot{m}_a e_{a1} + \dot{m}_{L1}e_{L1}) = (\dot{m}_a e_{a2} + \dot{m}_{L2}e_{L2}) + E_D, \quad (24)$$

where the terms in brackets on the left-hand side denote the exergy entering into the washer; whereas, the terms in brackets on the right-hand side denote the exergy leaving the washer, and E_D denotes the exergy destruction.

Following Bejan (2006), the second law efficiency (ηII) is defined by the ratio between the total exergy leaving the system and the total exergy entering in the system:

$$\eta_{II} = \frac{(\dot{m}_a e_{a2} + \dot{m}_{L2}e_{L2})}{(\dot{m}_a e_{a1} + \dot{m}_{L1}e_{L1})} \quad (25)$$

For all results shown in the next section, temperature and relative humidity of the dead state were kept constant and set to $T_0 = 25$ °C and $\phi_0 = 50\%$, respectively.

3. Analysis of results

In this section, a parametric study will be conducted to investigate the influence of dry bulb temperature and air humidity, water temperature, and the length of the washer over the saturation effectiveness, temperature drop (difference between the air temperature at the inlet and at the outlet), second law efficiency, and exergy destruction. Table 1 shows the physical parameters of the air washer investigated.

3.1. *Effect of dry bulb temperature*

In this analysis, the humidity ratio and the water temperature are equal to 0.002 kg kg^{-1} and 15 °C, respectively. In Fig. 2a, one sees that the increase of the dry bulb temperature of the intake air causes an increase in the saturation effectiveness and an air temperature drop in the washer. The increase in the temperature drop observed in Fig. 2a was due to the fact that by increasing the intake air temperature, the potential for heat exchange between air and water also increases. Thus, a larger amount of sensible heat can be removed from the air to help water evaporation. Also, the temperature drop recorded in the air washer increases linearly with the temperature of intake air. From Fig. 2b, we can verify that an increase in intake air temperature causes an

Table 1 – Data and properties of the air washer investigated.

Spray ratio, G_l/G_a	0.7
Air heat transfer per cubic meter chamber volume, h_a	1.34 kW/(m^3 K)
Water heat transfer per cubic meter of chamber volume, h_L	16.77 kW/(m^3 K)
Air mass flow rate per unit area, G_a	1.628 kg/(s m^2)
Air volumetric flow, Q	3.07 m^3 s^{-1}
Lewis number, $Le = h_a/(K_M\, c_{pm})$	1
Specific heat of moist air at constant pressure, c_{pm}	1.005 kJ/(kg K)

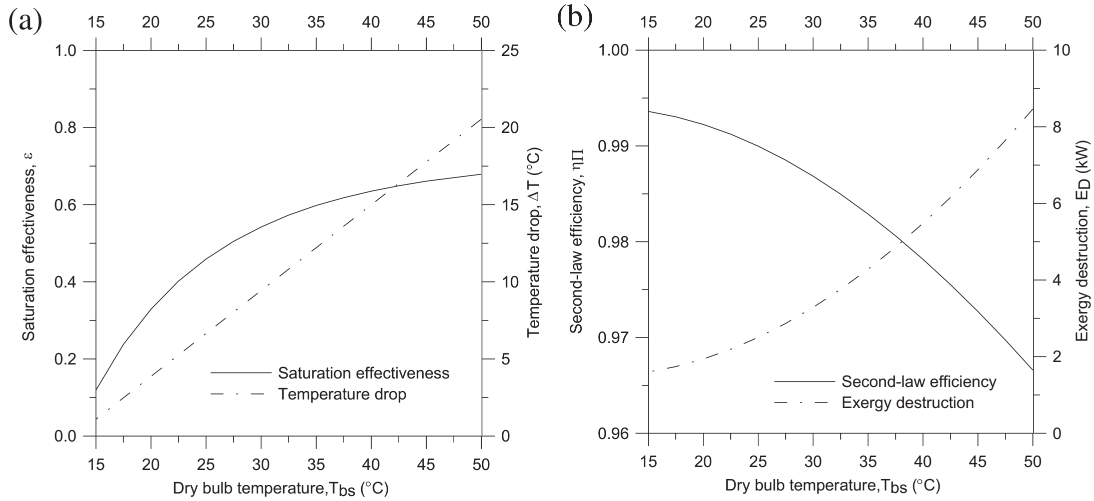

Fig. 2 – Effect of dry bulb temperature on the following parameters: a) saturation effectiveness and temperature drop, and b) second law efficiency and exergy destruction.

increase in exergy destruction and a decrease in the second law efficiency of the system. That is because an increase in the intake air temperature raises the temperature gradient between air and water, increasing the opportunities for heat transfer and the associated irreversibilities. Comparing Fig. 2a and b, one can see that the best thermodynamic performance of the system occurs when air and water present the least difference in temperature at the washer inlet. This is clearly a situation of little practical interest, and in most cases the system operates with large temperature gradients, resulting in higher irreversibility and exergy destruction. It has also been observed that the greatest temperature drop occurs when the second law efficiency decreases. Thus, one may conclude that the optimization of the process should involve both energy and exergy analyses to produce the best conditions of thermal comfort at an admissible thermodynamic cost.

3.2. *Effect of the humidity ratio*

In this analysis, the dry bulb temperature of the air and water at the inlet of the air washer were equal to 45 °C and 15 °C, respectively. Fig. 3a shows that increasing the humidity ratio of the inlet air will increase the saturation effectiveness. However, in this case, an increase in the saturation effectiveness does not mean an improvement in the system performance. As shown in Fig. 3a, when the humidity ratio of the intake air is increased, a reduction in the air temperature drop at the exit of the washer is observed. Consequently, in this case, the saturation effectiveness cannot be taken as a suitable parameter for evaluating the system performance. This is due to the fact the high humidity of the intake air causes the driving force that promotes mass transfer between water and air to decrease. This results in less evaporation and lower system performance. Fig. 3b illustrates the system evaluation

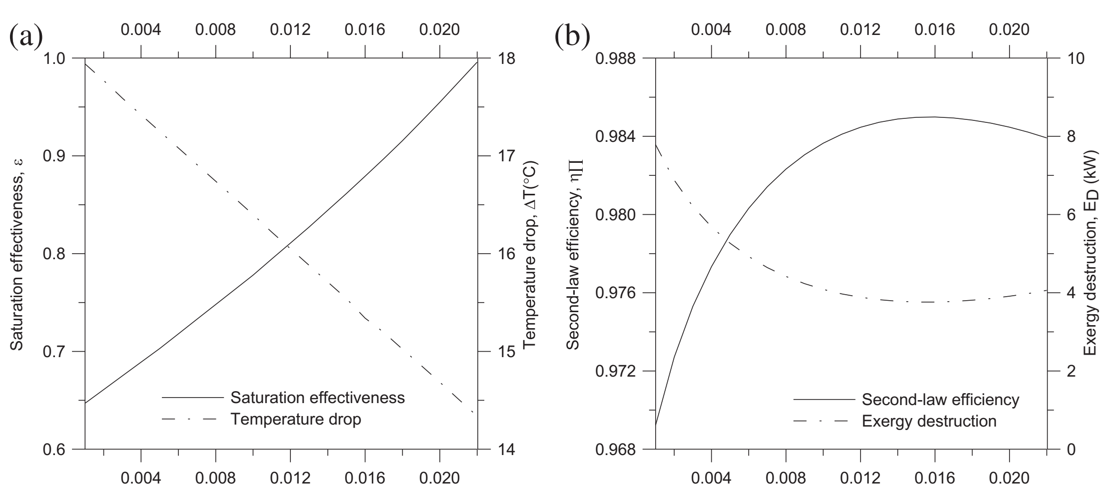

Fig. 3 – Effect of humidity ratio on the following parameters: a) saturation effectiveness and temperature drop, and b) the second law efficiency and exergy depletion.

based on the second law of thermodynamics. It is observed that when the moisture content of the intake air is increased, the second law efficiency increases up to a value around 0.016 kg kg^{-1} (maximum value); after this point, it decreases smoothly. On the other hand, we observe that exergy depletion decreases when the moisture content is increased up to 0.016 kg kg^{-1}, and then increases smoothly. As explained above, such behavior is due the fact that an increase in moisture content in the intake air decreases mass transfer in the washer. This results in the decrease of exergy depletion and an improved efficiency of the second law. When the air humidity ratio is equal to 0.016 kg kg^{-1}, the system offers the least opportunity for mass transfer, and, as a result, exergy depletion reaches its minimum value. The increase of exergy destruction and a reduction in the second law efficiency after a humidity ratio equal to 016 kg kg^{-1} is due to the reversal direction occurring in the mass transfer process, i.e., as humidity ratio increases, the mass transfer that comes from the interface into the air (humidification) decreases down to a point where the reversal takes place; in other words, from air to water (dehumidification). Regardless of the process direction, there is always exergy depletion associated with mass transfer. Considering Fig. 3a and b further, we conclude that a maximum drop in temperature within the system happens when the air is drier and the opportunities for mass transfer are better; in such a way that the exergy depletion is at its maximum value, and the second law efficiency drops to its minimum value. Again, one can easily see here that the most favorable situation for obtaining thermal comfort does correspond to the thermodynamic system's worst performance. This demonstrates that the optimization process must take into account the thermodynamic cost associated with the best thermal comfort condition.

3.3. *Effect of water temperature*

In this case study, the dry bulb temperature and the humidity ratio of the inlet air were equal to 45 °C and 0.010 kg kg^{-1}, respectively. Fig. 4a shows that increasing the temperature of the water, the saturation effectiveness and the air temperature drop along the washer decrease. This happens because an increase in water temperature reduces the driving force for heat transfer between water and air; consequently, less sensible heat will be extracted from the air and, therefore, the system performance decreases. On the other hand, as shown in Fig. 4b, when the water temperature increases, the second law efficiency increases up to the point where the water temperature reaches around 25 °C at which point efficiency reaches its maximum value. Beyond this point, the efficiency decreases. Conversely, it was observed that exergy depletion decreases when the water temperature increases up to 25 °C and conversely increases after that point. Upon analyzing these results, it is important to note that exergy depletion in the system is mainly due to heat and mass transfer. In conditions where humidity and temperature of the inlet air are fixed, an increase in water temperature reduces the driving force for heat transfer, but increases the potential for mass transfer. From Fig. 4b, it is clear that when the water temperature rises from 15 to 25 °C, the mass transfer mechanism is not important and, therefore, the exergy depletion decreases and the second law efficiency increases. On the other hand, if the water temperature rises beyond 25 °C, the mass transfer starts to be a dominat mechanism, thus increasing the exergy depletion and decreasing the second law efficiency. As already mentioned above, the best performance from the point of view of thermal comfort conditions (elevated temperature drops) does not match with the best thermodynamic performance (high second law efficiency), thus a careful optimization study should be undertaken in order to produce the desired thermal comfort at some permissible level of exergy depletion.

3.4. *Effect of the washer's length*

In this analysis, the dry bulb temperature, humidity ratio, and water temperature were set to 45 °C, 0.010 kg kg^{-1} and 15 °C, respectively, and the washer length varied from 0.25 to 2 m. As one can see in Fig. 5a, when the length of the air washer is

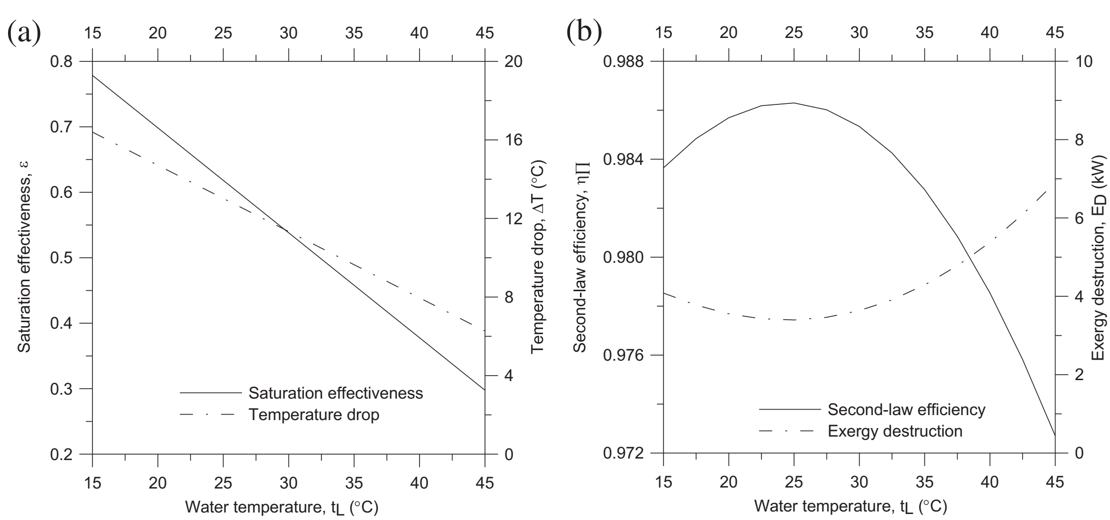

Fig. 4 – Effect of water temperature on the following parameters: a) saturation effectiveness and temperature drop, and b) second law efficiency and exergy depletion.

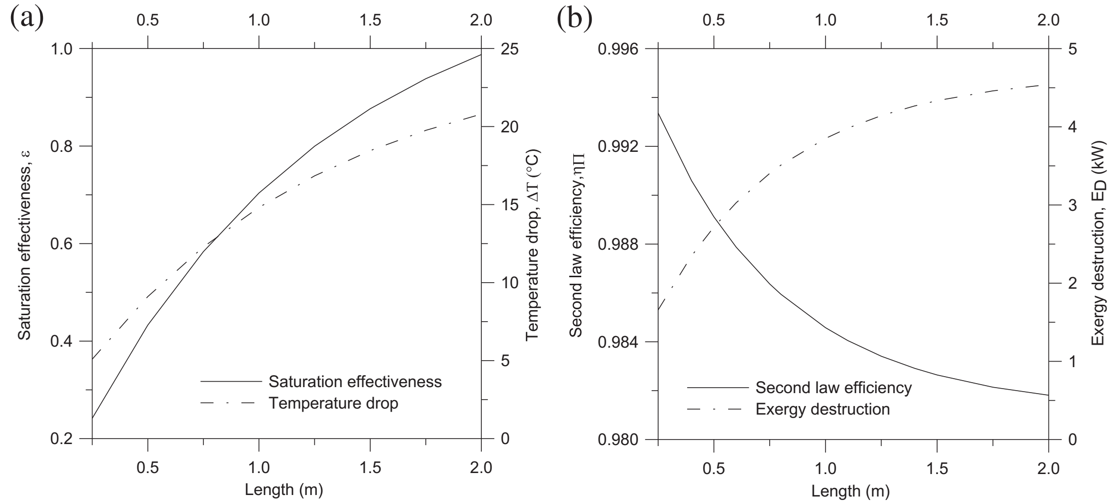

Fig. 5 – Effect of the washer length on the following parameters: a) saturation effectiveness and temperature drop, and b) second law efficiency and exergy depletion.

increased, the saturation effectiveness and air temperature drop increases because of the increase in the residence time of the air flow into washer. For the investigated parameters, the maximum length allowed for the washer is around 2 m, considering that the saturation effectiveness is close to 100% at the exit of the air washer. Fig. 5b, in turn, shows that an increase in the length of the washer causes an increase in exergy depletion and a fall in the second law efficiency. This is mainly due to an increase in air flow residence time in the washer, which flavors the occurrence of heat and mass transfer, giving rise to increased exergy depletion in the system.

4. Conclusions

In this paper, an energy and exergy analysis was carried out for the performance evaluation of the evaporative cooling process in air washers. To determine the air temperature and humidity, an analytical solution for the proposed model, developed in our previous work, was applied. An exergy balance was applied to the system in order to determine the irreversibility associated with heat and mass transfer. The effect of air temperature and humidity, water temperature, and air washer length were investigated in order to see how they affect the system performance in terms of saturation effectiveness, second law efficiency, and exergy depletion. The results obtained from the present study have demonstrated that the best condition for the intake air – where thermal comfort is concerned – does not match the best condition for better thermodynamic performance. Because of this, the analyses of energy and exergy should be implemented simultaneously to promote the optimum performance conditions with minimum exergy depletion. In general, the most favorable air provision to secure thermal comfort in the washer implies higher exergy depletion and a decline in the system's thermodynamic performance. The proposed model can be utilized in the development of more accurate numerical methods for the simulation and performance evaluation of air conditioning systems that uses air washers. Currently, a full modeling of a desiccant air conditioning unit in open cycle based on energy and exergy analysis is being developed.

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