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Analytical solution for the simultaneous heat and mass transfer problem in air washers

J.C. Santos [a], J.M. Medeiros [b], J.C. dos Santos [c], J.M. Gurgel [c], F. Marcondes [d,*]

[a] Department of Mechanical Production Engineering, Regional University of Cariri, Av. Leão Sampaio, S/N – Juazeiro do Norte, Ceará 63040-000, Brazil

[b] Federal Institute of Education, Science and Technology of Pernambuco, Rodovia PE-60 km 14, Ipojuca, Pernambuco 55590-000, Brazil

[c] Department of Mechanical Engineering, Federal University of Paraiba, LES/UFPB – Cidade Universitária, João Pessoa, Paraîba 58090-900, Brazil

[d] Department of Metallurgical Engineering and Material Science, Federal University of Ceará, Campus do Pici, Bloco 714 Fortaleza, Ceará 60455-760, Brazil

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ABSTRACT

An analytical solution approach for the simultaneous heat and mass transfer problem in air washers operating as evaporative coolers is presented. A one-dimensional model using the coupled mass and energy balance equations in the air washer is presented. Then, starting from a linear approach for the experimental curve of the air saturation, an analytical solution for the model was derived. The solution showed an excellent agreement with the available results found in the literature. The influence of several important parameters for the cooling process such as temperature and ambient air humidity, air flow rate and feeding water temperature, in the air cooling rate was investigated. The efficacy of the process can be greatly increased by reducing the cooling water temperature and the applied air flow rate. The analytical solution can be easily included into the models used for simulating desiccant air-conditioning systems operating in conjunction with air washers.

Solution analytique du problème de transfert simultané de chaleur et de masse dans un laveur d'air

Mots clés : Conditionnement d'air ; Laveur d'air ; Système évaporatif ; Calcul ; Transfert de chaleur ; Transfert de masse

* *Corresponding author*. Tel.: +55 85 3366 9355; fax: +55 85 3366 9969.
E-mail address: marcondes@ufc.br (F. Marcondes).

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Nomenclature

a	surface area per unit of chamber volume [$m^2\,m^{-3}$]
A_{cs}	cross-sectional area of the air washer [m^2]
c_L	specific heat of the water [$J\,kg^{-1}\,K^{-1}$]
c_{pm}	specific heat of the moist air at constant pressure [$J\,kg^{-1}\,K^{-1}$]
G	flow rate per unit cross-sectional area of the air washer [$kg\,s^{-1}\,m^{-2}$]
h	heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
H	specific enthalpy [$J\,kg^{-1}$]
H_i	specific enthalpy value in the air–water interface [$J\,kg^{-1}$]
h_a	air heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
h_L	water heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
H_{fg}	enthalpy of vaporization [$J\,kg^{-1}\,K^{-1}$]
K_M	mass transfer coefficient [$kg\,s^{-1}\,m^{-2}$]
x	axial position [m]
L	length of the air washer [m]
$h_a/K_M c_{pm}$	Lewis relation
T	temperature [°C]
W	air humidity ratio [$kg\,kg^{-1}$]
W_i	air humidity ratio in the air–water interface [$kg\,kg^{-1}$]

Subscripts

a	relative to the dry air
H	relative to the heat transfer
M	relative to the mass transfer
i	relative to the air–water interface
L	relative to the water
0	property evaluated at the 0 °C.

1. Introduction

A great majority of air-conditioning machines traditionally used in the applications of the thermal comfort operate with vapour–compression refrigeration cycles. These machines use non-renewable energy and are responsible for increasing the demand for electric power in the summer season. Replacing these machines represents an important problem to be considered, since in many countries, the electric power generation is obtained with the combustion of fossil fuels, and therefore contributing to the global warming. Vapour–compression refrigeration systems still present the disadvantage of the utilization of chlorofluorocarbons and hydrochlorofluorocarbons (CFC/HCFC), which are responsible for the destruction of the ozone layer. As a result, a lot of effort has been devoted for developing air-conditioning systems that employ renewable energy resources and working fluids that cause no damage to the environment. In this context, evaporative cooling systems appear as a correctly and ecologically alternative to replace the traditional air-conditioning systems.

The evaporative cooling is determined by the simultaneous heat and mass transfer processes which occur when the air to be cooled enters in direct contact with a humid surface in such a way that water and air are the working fluid used in the system. In some systems, the humid surface is replaced by jets of water droplets which are sprinkled into the air (air washers). In general, a warm and dry air stream put into contact with a humid surface, where the air supplies the necessary latent heat that leads to the evaporation of a certain amount of water. The air is then cooled as a result of the sensible heat transferred to the water. The heat and mass transfer processes cease when the air reaches its saturation temperature. In this case, the minimum temperature reached by the air in evaporative coolers will be equal to the air wet-bulb temperature at the inlet of the cooler. One of the main advantages of evaporative cooling is that the efficacy of the process increases when the climate is very hot and dry; in other words, when the demand for thermal comfort increases. Another important advantage of the evaporative cooling is the total renewal of the air, eliminating in this way, the proliferation of fungi and bacteria into the conditioned space. Other benefits from this system are listed as follows: low energy consumption, easy maintenance, easy installation and operation, and no utilization of chlorofluorocarbons and hydrochlorofluorocarbons. Even in conditions where the air presents higher humidity content, evaporative coolers can be used together with adsorptive dehumidifiers, propitiating the appropriate thermal comfort conditions with low energy consumption.

Different approaches concerning the modelling of the heat and mass transfer processes in evaporative coolers have been developed with the objective of exploring the potentialities of this technology. Maclaine-Cross and Banks (1981) have developed a study to model evaporative coolers. From the results obtained, an approach to correlate the transfer coefficients in humid and dry surface heater exchangers was proposed. Stoecker and Jones (1982) developed a numerical procedure for obtaining the number of transfer units, NTU ($h_c A/c_{pm}$), in cooling towers. In that approach, the cooling tower is divided in elementary control volumes, and then the NTU is obtained through a numerical procedure. According to the authors, the NTU is an important parameter of the cooling tower. They showed that increasing the NTU parameter will increase the thermal performance of the cooling tower, and the outlet water temperature will be more close to the wet-bulb temperature of the air at the inlet of the cooling tower. The authors also described a numerical procedure for obtaining the air conditions into the cooling tower. Although important, the methodology presented by Stoecker and Jones (1982) requires the previous knowledge of the NTU number. More importantly, an interactive procedure is necessary in order to obtain the water conditions at the outlet of the cooling tower. Jaber and Webb (1989) applied the effectiveness-NTU approach used by conventional heat exchanges to evaporative systems such as cooling towers and air washers. They linearized the air saturation curve in order to obtain the equations for counter flow cooling towers in conjunction with the effectiveness-NTU approach. Ismail and Mahmoud (1994) developed a mathematical model for the analysis of air washers. The model was used to determine the feasibility of the utilization of air washers and chemical dehumidifiers in conjunction with liquid coolers in air-conditioning applications. Halasz (1998) developed a mathematical model to describe the physical behaviour of different devices which uses evaporative cooling processes in its operation, such as cooling towers, evaporative condensers, air washers and cooling and dehumidifying coils. In order to simplify the solution process, the author replaces the real saturation curve of the air by a straight saturation line. Dai and Sumathy (2002) studied a direct evaporative cooler made up of an evaporative cell. A mathematical model made up of equations for the liquid film, the gaseous phase and the liquid–gas interface was developed. The results indicated that parameters such as air channel lengths within the evaporative cell, flow rate of the feeding water, and the flow rate of the processed air could be optimized in order to improve the performance of the cooler. Camargo et al. (2003) carried out a thermodynamic analysis of an evaporative cooling system coupled to an adsorptive dehumidifier. They assumed a constant effectiveness in order to perform the analysis. The results indicated that a minimization of the reactivation temperature reduced the operational costs of the system. Camargo et al. (2005) accomplished a theoretical and experimental study of a direct evaporative cooler that uses an evaporative cell. The model presented does not take into account the effect of the heat transfer resistance between the liquid film and the liquid–gas interface. Also, it does not take into account the space variations of temperature and humidity that occur during the air flow along the evaporative cell. The experimental results obtained were used together with the mathematical model in order to calculate the heat transfer coefficient between the water and the air. Beshkani and Hosseini (2006) studied the effects of air velocity and the length of the evaporative panels on saturation efficiency and pressure drop in direct evaporative coolers. An analysis of the results showed that the cooler efficiency increases with the reduction in the velocity of the process air stream, and the increase in the length of the panel. Jin et al. (2007) presented a literature review of the mathematical models employed for cooling towers (Merkel model (Merkel, 1925), effectiveness-NTU method, and Stoecker method (Stoecker, 1976)). According to the authors, the three above models are inadequate for control and optimization of cooling towers operating in real time. In order to avoid such limitation, they proposed an empirical model based on the mass and energy balances to optimize the performance of cooling towers. However, this approach is based only on the inlet and outlet conditions of the cooling tower. This model also cannot be used to evaluate the water and air conditions into the cooling tower. Niksiar and Rahimi (2009) developed a numerical study concerning the energy and exergy efficiencies in an evaporative cooler. A mathematical model of the problem was developed through the application of the mass, energy and exergy balance equations to the cooler. The results showed that the system presents low exergy efficiency, although it had demonstrated high energy efficiency. An analysis of the influence of some operational and design parameters in the exergy destruction was also discussed.

There are several articles in the literature that investigated air-conditioning systems in conjunction with evaporative panels. However, only few have been published on evaporative cooling using air washers. As mentioned above, Jaber and Webb (1989) developed an analytical solution based on the Effectiveness-NTU method. However, this solution cannot be used to predict the temperature and humidity fields inside each point of the system. To the best of our knowledge there is no analytical solution to the heat and mass transfer processes that occurs in air washers available in the literature that can be used to predict the air parameters in each point of the system. These devices have been quite used in conjunction with desiccant rotors for air-conditioning in areas of hot and humid climate (Ando et al., 2005). The main contributions of the present work are the modelling of the heat and mass transfer processes that occurs in these devices, and obtaining an analytical solution that will help in the developing of accurate numerical models to the desiccant air-conditioning systems employing air washers. Heat transfer resistance in liquid film and variation of both humid air and water properties into the air washer will be taken into account in the presented modelling.

2. Mathematical modelling

Fig. 1 shows an air washer with all physical and geometric parameters. For the proposed mathematical model, we assume parallel air flow along with the air washer. We define G_a and G_L the dry air and water flow rates per unit of cross section area (A_{cs}) of the air washer. Because the water evaporates as the air flows along the washer, G_L changes to $G_L + dG_L$ in the differential element dx. In a similar way, variations do occur in other properties of air and water. The equations that describe the physical processes are obtained applying the mass and energy balances in the differential element dx as described in the following sections. Further details can be found in ASHRAE Handbook Fundamentals (1997).

2.1. *Mass transfer to air*

From the mass balance in the differential element dx, one deduces that the reduction in water flow rate due to the evaporation equals that of the gains in air humidity due to the mass transfer between air and water,

$$-dG_L = G_a\,dW = K_M a_M (W_i - W)\,dx, \tag{1}$$

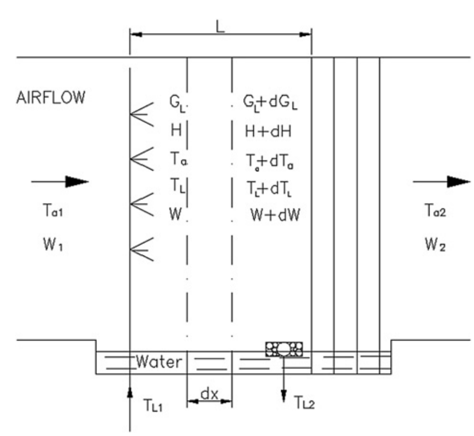

Fig. 1 – Air washer.

where K_M is the mass transfer coefficient, W_i is the air humidity ratio at the air–water interface, W is the air humidity ratio, and a_M is the available area to mass transfer per unit of volume of the air washer.

2.2. Heat transfer to air

The reduction of the air temperature is the result of sensible heat transfer between the air and the air–water interface,

$$G_a c_{pm}\, dT_a = h_a a_H (T_i - T_a)\, dx, \tag{2}$$

where c_{pm} is the specific heat of the moist air at the constant pressure which is considered constant, h_a is the air heat transfer coefficient, a_H is the surface area available to heat transfer per unit of volume of the air washer, T_i is the air temperature at the air–water interface, and T_a is the air temperature.

2.3. Total energy transfer to air

The variation of the air enthalpy is the result of total heat transfer (sensible heat + latent heat) from the air to the air–water interface,

$$G_a\left(c_{pm}\, dT_a + H_{fgo}\, dW\right) = \left[K_M a_M (W_i - W) H_{fg} + h_a a_H (T_i - T_a)\right] dx, \tag{3}$$

where H_{fg} is enthalpy of vaporization. The first two terms on the left-hand side of Eq. (3) represent respectively, the sensible and air enthalpy variation while the other terms on the right-hand side, denote respectively, the latent and the sensible heat transferred from the air to the interface. In direct contact devices, the superficial areas of heat and mass transfer may be regarded as identical ($a_M = a_H$). Neglecting the variations in the water vaporization latent heat (H_{fg}) with the temperature, and considering that the Lewis relation ($h_a/K_M c_{pm}$) is equal to the unit, Eq. (3) can be expressed as

$$G_a\, dH = K_M a_M (H_i - H) dx, \tag{4}$$

where H is the humid air enthalpy which is given by

$$H = c_{pm} T + W H_{fg}. \tag{5}$$

In Equation (4), the driving force for total energy transfer between the air and the interface is the enthalpy potential defined by ($H_i - H$).

2.4. Energy balance

Applying the energy conservation principle to the differential element dx, as shown in Fig. 1, one concludes that the air energy reduction results in an increase of the water energy,

$$G_a\, dH = -G_L c_L\, dT_L. \tag{6}$$

For air-conditioning applications, the percentage changes in G_L are assumed small, and consequently, in the deduction of Eq. (6), G_L is considered constant.

2.5. Heat transfer to water

An increase in water temperature takes place as a result of heat transfer between the liquid–gas interface and the water,

$$-G_L c_L\, dT_L = h_L a_H (T_L - T_i) dx. \tag{7}$$

Equations (1), (2), (4), (6), and (7) comprise the system of ordinary differential equations that furnishes the solution for the simultaneous heat and mass transfer problem for the air washers. The solution of the system of ordinary equations makes possible the determination of the following variables: $W(x)$, $T_a(x)$, $H(x)$, $T_L(x)$ and $T_i(x)$. The ordinary differential equations presented are subjected to the following initial conditions prescribed at the inlet of the air washer:

$$W(0) = W_1, \tag{8}$$

$$T_a(0) = T_{a1}, \tag{9}$$

$$H(0) = H_1, \tag{10}$$

$$T_L(0) = T_{L1}, \tag{11}$$

$$T_i(0) = T_{i1}. \tag{12}$$

3. Solution procedure

In order to facilitate the solution procedure, a linear adjustment of the air saturation curve in an enthalpy–temperature chart has been considered. Therefore, a linear function for the enthalpy–temperature in the liquid–gas interface was modelled by

$$H_i(T_i) = a_1 T_i + a_0, \tag{13}$$

where a_1 and a_0 are the adjusted coefficients. In the present work the coefficients were obtained for the temperature range between 15 and 30 °C, which represents well the working conditions imposed in the washer.

Combining Eqs. (4), (6) and (7) and considering $a_H = a_M$, we obtain

$$\frac{H - H_i}{T_L - T_i} = -\frac{h_L}{K_M}. \tag{14}$$

Integrating Eq. (6), we have,

$$H = H_1 - \frac{G_L c_L}{G_a}(T_L - T_{L1}). \tag{15}$$

Substituting Eqs. (13) and (15) into Eq. (14), we obtain a linear relation between T_L and T_i given as

$$T_L = b_1 T_i + b_0, \tag{16}$$

where b_0 and b_1 are given by

$$b_0 = \frac{-H_1 - \dfrac{G_L c_L T_{L1}}{G_a} + a_0}{\dfrac{h_L}{K_M} - \dfrac{G_L c_L}{G_a}}, \tag{17}$$

$$b_1 = \frac{a_1 + \dfrac{h_L}{K_M}}{\dfrac{h_L}{K_M} - \dfrac{G_L c_L}{G_a}}. \tag{18}$$

From Eq. (16), we obtain

$$dT_L = b_1\, dT_i. \tag{19}$$

Substituting Eqs. (16) and (19) into Eq. (7), we obtain the following equation to determine T_i:

$$-G_L c_L b_1\, dT_i = h_L a_H [b_0 + (b_1 - 1)T_i]dx. \tag{20}$$

Separating the variables in Eq. (20) and performing the integration, we obtain the air–water interface temperature T_i,

$$T_i(x) = C_1\, e^{\frac{-h_L a_H (b_1-1)x}{G_L c_L b_1}} - \frac{b_0}{(b_1 - 1)}. \tag{21}$$

Using Eq. (16) to relate T_{i1} with T_{L1}, and applying the inlet condition given by Eq. (12) into Eq. (21), we obtain the constant C_1, which is given by

$$C_1 = \frac{(T_{L1} - b_0)}{b_1} + \frac{b_0}{(b_1 - 1)}. \tag{22}$$

Substituting Eq. (21) into Eq. (2), we obtain the following differential equation to determinate the air temperature T_a:

$$\frac{dT_a}{dx} + P(x)T_a = Q(x), \tag{23}$$

where $P(x)$ and $Q(x)$ are given by

$$P(x) = \frac{h_a a_H}{G_a c_{pm}}, \tag{24}$$

$$Q(x) = P(x)T_i(x). \tag{25}$$

Equation (23) is a first order linear differential equation and the general solution has been described in the literature (Boyce and DiPrima, 2006).

$$T_a(x) = e^{-\int P(x)dx} \int Q(x)\, e^{\int P(x)dx} dx + C\, e^{-\int P(x)dx}, \tag{26}$$

where C represents an integration constant.

Integrating Eq. (23), and applying the initial condition given by Eq. (9), we will obtain the following expression for the air temperature T_a:

$$T_a(x) = \frac{C_2\, e^{-\frac{h_L a_H}{G_L c_L}\frac{(b_1-1)x}{b_1}}}{\left(\frac{h_a a_H}{G_a c_{pm}} - \frac{h_L a_H}{G_L c_L}\frac{(b_1 - 1)}{b_1}\right)} - \frac{b_0}{(b_1 - 1)} + C_3\, e^{-\frac{h_a a_H x}{G_a c_{pm}}}, \tag{27}$$

where x denotes the axial position in the air washer and the constants C_2 and C_3 are given, respectively, by

$$C_2 = C_1 \frac{h_a a_H}{G_a c_{pm}}, \tag{28}$$

$$C_3 = T_{a1} - \left\{ \frac{C_2}{\left[\frac{h_a a_H}{G_a c_{pm}} - \frac{h_L a_H}{G_L c_L}\frac{(b_1 - 1)}{b_1}\right]} - \frac{b_0}{(b_1 - 1)} \right\}. \tag{29}$$

Using a linear approach for the air saturation curve in a temperature–humidity chart, we obtain the following relation between humidity ratio and temperature in the air–water interface:

$$W_i(T_i) = d_1 T_i + d_0, \tag{30}$$

where d_1 and d_0 are adjusted coefficients.

Substituting Eq. (30) into Eq. (1), the mass balance for the humid air can be rewritten in the following form:

$$\frac{dW}{dx} + P(x)W = Q(x), \tag{31}$$

where $P(x)$ and $Q(x)$ are given by

$$P(x) = \frac{K_M a_M}{G_a}, \tag{32}$$

$$Q(x) = \frac{K_M a_M}{G_a}[d_0 + d_1 T_i(x)]. \tag{33}$$

Equation (31) has the same equivalent form of Eq. (23), and therefore its general solution is similar to Eq. (26). Making the required integrations, the general solution of Eq. (31) is given by

$$W(x) = d_0 - \frac{d_1 b_0}{(b_1 - 1)} + \frac{K_M a_M d_1 C_1\, e^{-\frac{h_L a_H (b_1-1)x}{G_L c_L b_1}}}{\left[K_M a_M - \frac{G_a h_L a_H (b_1 - 1)}{G_L c_L b_1}\right]} + C_4\, e^{-\frac{K_M a_M x}{G_a}}, \tag{34}$$

where C_4 denotes a integration constant. Applying the initial condition given by Eq. (8) to Eq. (34), the constant C_4 is given by

$$C_4 = W_1 - d_0 + \frac{d_1 b_0}{(b_1 - 1)} - \frac{K_M a_M d_1 C_1}{\left[K_M a_M - \frac{G_a h_L a_H (b_1 - 1)}{G_L c_L b_1}\right]}. \tag{35}$$

4. Results and discussions

4.1. *Comparison with published results*

In Chapter 5 of the ASHRAE Handbook Fundamentals (1997), a case study focusing on the heating and humidifying process in an air washer was discussed. In order to obtain the performance of the air washer, the graphical procedure described by Kusuda (1957) was used. The data and properties of the air washer used in this case study are described in Table 1. In order to test the analytical solution provided by Eq. (27), Fig. 2 shows a comparison between the solution obtained in the present work and the data extracted from the graphic solution presented in the ASHRAE Handbook Fundamentals (1997). The solid line denotes the analytical solution whereas the points 1–a–b–c–d–2 were extracted from the graphic solution obtained for the heating and humidification process of the air. It is also shown in Fig. 2, a numerical solution using a polynomial adjustment of the air saturation curve taken from the psychrometric chart. This numerical solution was solved using the subroutine LSODE described in Radhakrishnan and Hindmarsh (1993). As can be seen in Fig. 2 there is a good match between the graphical solution, numerical, and the analytical solution, except in the areas close to the air washer exit. These minor differences take place as a result of the linearization procedure employed for the air saturation curve adopted in the present methodology. Since the saturation curve is perfectly represented by a quadratic function, small errors may occur when a linear adjustment is performed. However, as can be observed in Fig. 2, the maximum difference between the two solutions was less than 1%. These errors tend to decrease with the reducing of the temperature range employed. The main advantage in the use of the analytical solution given by Eq. (27), lies in its simplicity in obtaining results under different

Table 1 – Data set and properties of the air washer used for case study 1.

Length of the air washer, L	1.2 m
Air temperature at inlet, T_{a1}	18.3 °C
Air enthalpy at inlet, H_1	41.1 kJ kg^{-1}
Water temperature at inlet, T_{L1}	35 °C
Spray ratio, G_l/G_a	0.7
Air heat transfer per cubic metre of chamber volume, $h_a a_H$	1.34 kW m^{-3} K^{-1}
Water heat transfer per cubic metre of chamber volume, $h_L a_H$	16.77 kW m^{-3} K^{-1}
Air mass flow rate per unit area, G_a	1.628 kg m^{-2} s^{-1}
Air volumetric flow rate, m	3.07 m^3 s^{-1}
Lewis relation, $h_a/(K_M c_{pm})$	1
Specific heat of moist air at constant pressure, c_{pm}	1.005 kJ kg^{-1} K^{-1}

air inlet conditions. The analytical solution developed in the present work can be useful in the simulation air-conditioning systems that employ desiccant rotors together with air washers, where process air conditions vary continuously.

4.2. *Effect of the air dry bulb temperature*

In order to test the analytical solution, we have studied a case in which the air washer shown in Fig. 1 was used for heating and humidification of air. However, the main purpose of the present work is to study the performance of air washers that work as cooling and humidifying devices. For such investigation one should only modify both the water and the process air conditions in the inlet of the device. In order to assess the effect of dry bulb temperature in air cooling rate, three distinct values have been considered: 25, 30 and 35 °C. In the present case study, the inlet air humidity ratio was considered equal to 0.006 kg kg^{-1} while the inlet water temperature was assumed equal to 20 °C. Fig. 3 shows the air temperature profiles along the axial position in the air washer. The investigations showed that for a certain humidity ratio value, when increasing the air dry bulb temperature at the inlet, greater

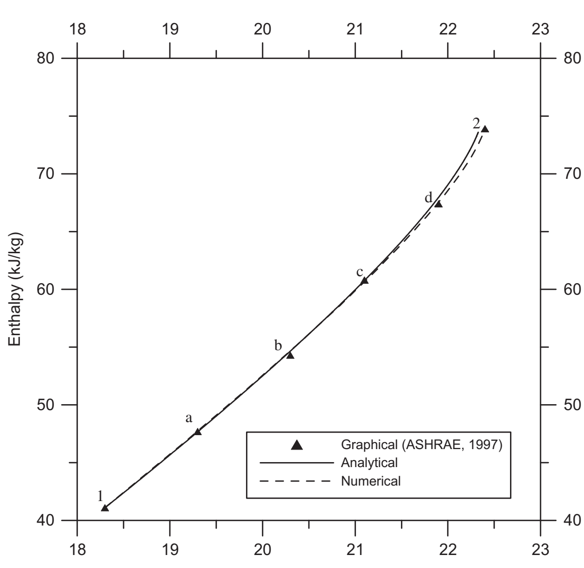

Fig. 2 – Comparison of the analytical, numerical, and graphic solutions.

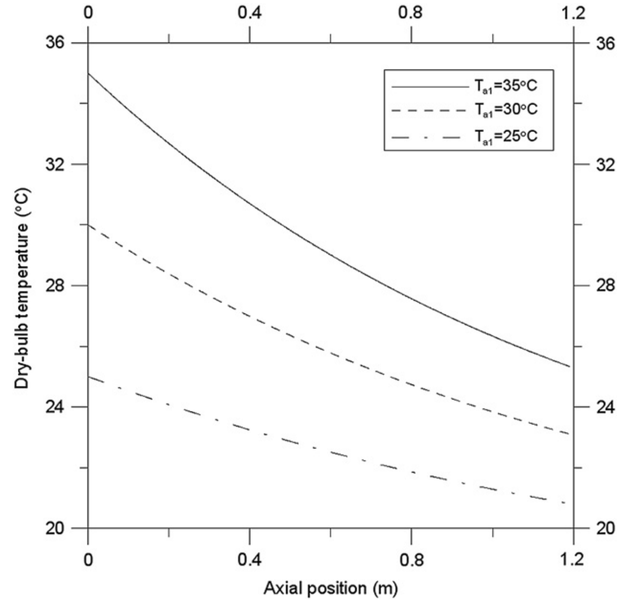

Fig. 3 – Influence of the air dry bulb temperature in the temperature drop along the washer.

temperature drop is obtained. This happens because higher dry bulb temperatures with constant humidity ratio mean smaller relative humidity. When the air relative humidity is small, there emerges a high difference in the partial pressure of water vapour, favouring in this way, the mass transfer between the water and the air. Consequently, more sensible heat is extracted from the air for water evaporation, resulting in larger temperature drop in the air. In the present case study, when inlet air temperature was set to 35 °C, the temperature drop was about 10 °C. However, when the inlet temperature was 25 °C, there occurred a temperature drop of only 4.2 °C.

4.3. *Effect of the inlet humidity ratio*

In the present case study, the air dry bulb temperature at the inlet was set to 35 °C, while three different values for the air humidity ratio were considered: 0.002, 0.006 and 0.010 kg kg^{-1}. The inlet water temperature was considered equal to 20 °C. Fig. 4 shows that an increase in air humidity ratio reduces the cooling rate into the air washer, in such way that the air exit temperature increases. Due to the increasing in the ambient air humidity, the driving force to mass transfer between the air and the water decreases. Therefore, a smaller amount of sensible heat is extracted from the air in order to evaporate the water and so the air temperature increases. From Figs. 3 and 4, it can be verified that the efficacy of the evaporative cooling process increases when the weather is more hot and dry.

4.4. *Effect of the inlet water temperature*

For this investigation the air dry bulb temperature at inlet was set to 35 °C; while the inlet humidity ratio is maintained at 0.006 kg kg^{-1}. Fig. 5 shows the results for three different values for the inlet water temperature: 15, 20 and 25 °C. From Fig. 5, one can see that the reduction in the feed water temperature decreases the air temperature at the exit of the air washer.

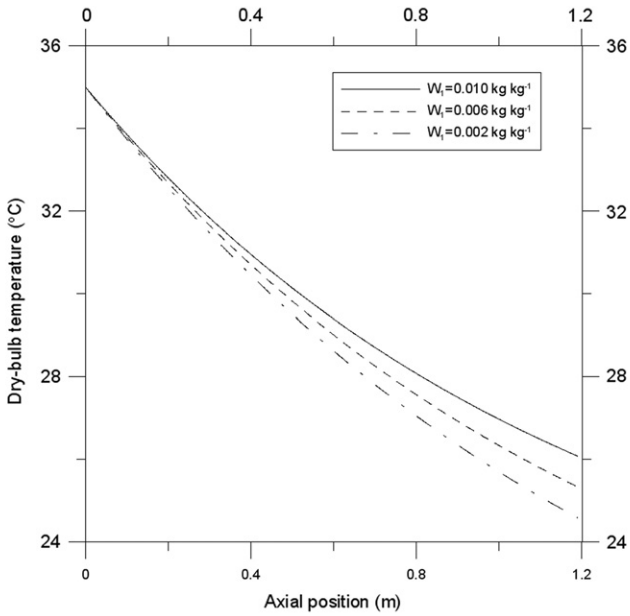

Fig. 4 – Influence of the air humidity ratio in the air cooling rate.

This decrease occurs because a reduction in water temperature stimulates the heat exchange between the air and the water. Under such conditions, a larger amount of sensible heat is taken out from the air, and thus decreases its temperature considerably. Therefore, a reduction in the feed water temperature increases the air cooling rate.

4.5. *Effect of the applied air flow rate*

Fig. 6 shows the effect of the applied air flow rate on the air temperature profiles along the washer. The air dry bulb temperature at the inlet was set to 35 °C, while the inlet

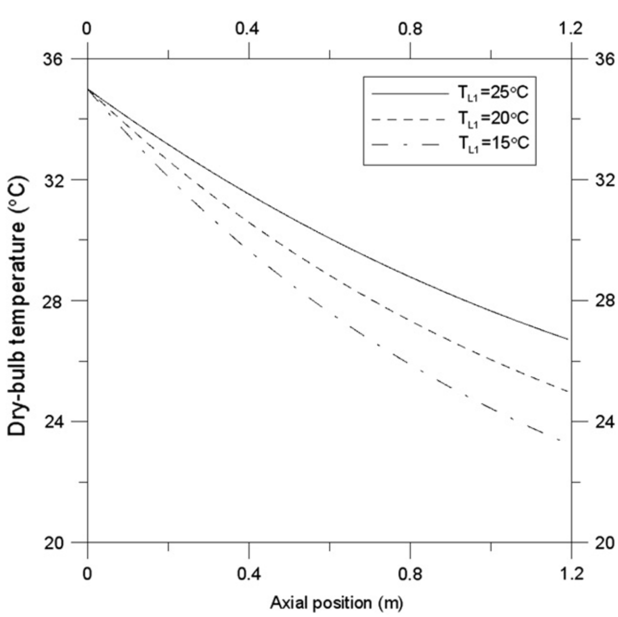

Fig. 5 – Influence of the inlet water temperature in the air cooling rate.

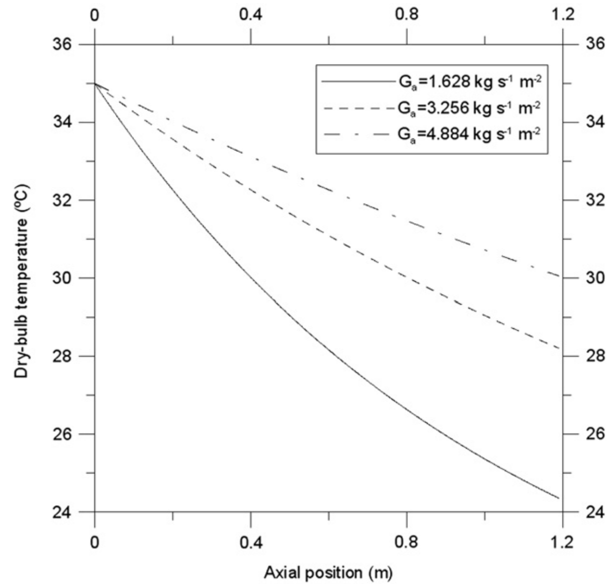

Fig. 6 – Influence of the applied air flow rate in the air cooling rate.

humidity ratio was maintained in 0.010 kg kg^{-1}. Three different values for the mass flow rate per unit cross-sectional area of the washer are considered: 1.628, 3.256, and 4.884 kg m^{-2} s^{-1}. The inlet water temperature was maintained at 15 °C. From Fig. 6, it is possible to observe that an increase in the applied flow rate reduces the air cooling rate. This occurs as a result of the reduction in the contact time between the water and the air as the applied flow rate increases, and as a consequence the temperature of air leaving the air washer increases.

4.6. *Behaviour of the water and air in the washer*

Fig. 7 shows the water, the liquid–gas interface, and the air temperature profiles along the washer. The air dry bulb

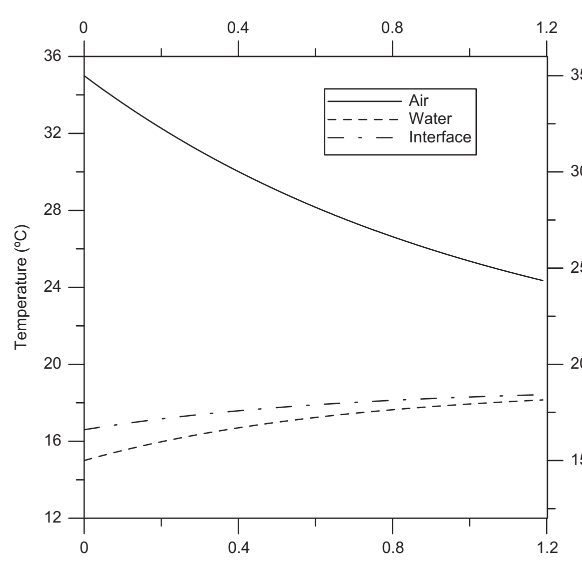

Fig. 7 – Water, air, and interface temperature profiles along the washer.

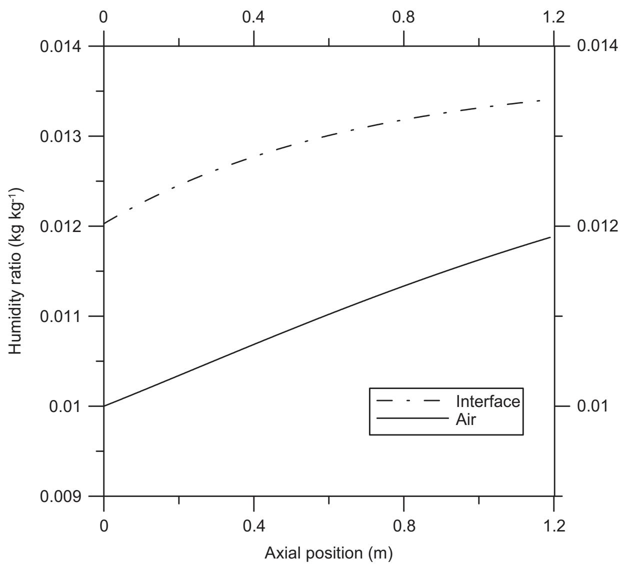

Fig. 8 – Air humidity ratio profiles along the washer.

temperature at the inlet of the washer was maintained at 35 °C, while the inlet humidity ratio was maintained at 0.010 kg kg^{-1}. The inlet water temperature was set to 15 °C. From Fig. 7, it can be observed that the warm and dry air transfers sensible heat for the water causing its heating. The interface liquid–gas, in direct contact with the warm and dry air, presents the same behaviour verified for the water. It has been observed, however, that the temperatures profiles approach each other as the air flows along the air washer. In fact, if the contact area between the air and the water were infinite these temperature profiles would be the same at the exit of the air washer. Fig. 8 shows the profiles of the interface humidity ratio and the air humidity ratio along the washer. It can be observed that the air humidity ratio increases as a consequence of the mass transfer between the air and the water. The air humidifying process happens because the humidity ratio in the interface liquid–gas is higher than the air humidity ratio.

5. Conclusions

In this work, a theoretical study of the simultaneous heat and mass transfer problem in air washers operating as evaporative cooling devices was presented. The equations that describe the heat and mass transfer processes in air washers were presented, and an analytical procedure was used for the solution for the proposed model. In order to simplify the solution procedure, a linear approach for the air saturation curve was proposed. The error verified with the approach used was very small in such way that the analytical solution obtained represents satisfactorily the heat and mass transfer processes in the air washers. The analytical solution developed in this work can be used for the simulating of air-conditioning systems that employ desiccant rotors in conjunction with air washers, where the air conditions vary continuously during the process. The results obtained were described in terms of temperature profiles along the axial position. Results have demonstrated that air washers are far more efficient when they operate in hot and dry climates, producing the lowest exit temperatures. Another parameter that has confirmed the efficacy of air washers concerns with the temperature reduction of the water that is sprinkled into the air. Results have shown that an increase in the applied air flow rate in the washer reduces the time of contact between the water and the air, leading towards a reduction in the cooling rate. As the air cools the water becomes warmer as a result of the heat transfer between them.

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