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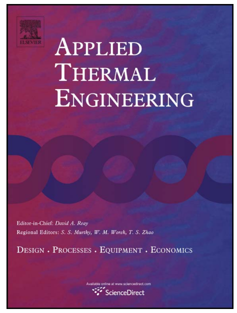

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Indirect Evaporative Cooling of Air to a Sub-Wet Bulb Temperature

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ABSTRACT

Indirect evaporative cooling is a sustainable method for cooling of air. The main constraint that limits the wide use of evaporative coolers is the ultimate temperature of the process, which is the wet bulb temperature of ambient air. In this paper, a method is presented to produce air at a sub-wet bulb temperature by indirect evaporative cooling, without using a vapour compression machine. The main idea consists of manipulating the air flow inside the cooler by branching the working air from the product air, which is indirectly pre-cooled, before it is finally cooled and delivered. A model for the heat and mass transfer process is developed. Four types of coolers are studied: three two-stage coolers (a counter flow, a parallel flow and a combined parallel-regenerative flow) and a single-stage counter flow regenerative cooler.

It is concluded that the proposed method for indirect evaporative cooling is capable of cooling air to temperatures lower than the ambient wet bulb temperature. The ultimate temperature for such a process is the dew point temperature of the ambient air. The wet bulb cooling effectiveness (E_{wb}) for the examples studied is 1.26, 1.09 and 1.31 for the two-stage counter flow, parallel flow and combined parallel-regenerative cooler, respectively, and it is 1.16 for the single-stage counter flow regenerative cooler. Such a

method extends the potential of useful utilisation of evaporative coolers for cooling of buildings as well as other industrial applications.

Keywords: Indirect evaporative cooling; sub-wet bulb temperature; dew point approach

NOMENCLATURE

d	thickness of the thin wall and water film (m)
E_{dp}	dew point effectiveness, $E_{dp} = (T_i - T_o)/(T_i - T_{dp})$
E_{wb}	wet bulb effectiveness, $E_{wb} = (T_i - T_o)/(T_i - T_{wb})$
H	humidity ratio of air (kg water/kg dry air)
h	air enthalpy (J/kg)
L	passage length (m)
M	air mass flow rate in the dry passage (kg/s)
m	air mass flow rate in the wet passage (kg/s)
Q	rate of heat transfer (W)
RH	relative humidity of air (%)
T	air temperature in the dry passage (°C)
t	air temperature in the wet passage (°C)
t_f	temperature of water film (°C)
y	height of the dry or wet passages (m)
Z	passage width (m)
α	convective heat transfer coefficient (W/m^2 K)
β	mass transfer coefficient (kg water/s m^2)/(kg water/kg dry air)

Subscripts

d dry side

dp dew point temperature

i inlet

n node

o outlet

w wet side

wb wet bulb temperature

1 first stage

2 second stage

Superscripts

´ saturated condition at the air-water interface temperature

´´ saturated condition at the water film temperature t_f

1. INTRODUCTION

The building sector accounts for a major part of the world's total end energy consumption. It has the largest single potential for improving the efficiency of energy use. Cooling energy is an important part of this energy and the demand for cooling is continuously increasing due to the growing demand for better indoor comfort conditions in buildings and the effects of global warming. Evaporative cooling is an efficient and economically feasible method. It is a sustainable solution because the working fluids are air and water. Besides, evaporative cooling is not limited to applications in cooling of buildings, but there are also numerous other agricultural and industrial applications [1].

However, conventional evaporative cooling has a serious thermodynamic limitation: the ultimate temperature of the process is the ambient air wet bulb temperature, and in practice the achievable temperature is even higher. For this reason, the coolant fluid is not able to reach a suitable low temperature in many cases and thus the potential of useful utilisation of evaporative cooling is limited. Therefore, new methods and technologies are needed for the production of cooling energy.

Sub-wet bulb temperature evaporative cooling is the solution for this limitation because it enables cooling below the wet-bulb temperature of the ambient air. There are several studies on achieving sub-wet bulb temperatures by evaporative cooling and some innovative ideas exist. However, these methods are not well-known for most HVAC engineers, and the related results are not in common use. Crum et al. [2] indicated that this is achievable by using multistage indirect evaporative cooling and by a cooling tower-heat exchanger combination. They indicated that the cooling tower-heat exchanger combination has the greatest thermal potential for air conditioning applications. It can produce lowest temperatures and highest cooling capacities for any value of fraction of inlet air delivered. They mentioned that the coefficient of performance for this equipment can reach 75 in the range of air states seen in air conditioning practice. Hsu et al. [3] studied, theoretically and experimentally, two configurations of closed-loop wet surface heat exchangers to generate sub-wet bulb temperature cooling by a counter flow and a cross flow. Experimental measurements were indicated for a closed-loop counter flow cooler. They indicated that for the counter-flow closed-loop configuration, the maximum wet-bulb effectiveness is 1.3 and is reached at a dry-passage number of transfer units (NTU) of 10, while for the cross-flow closed-loop configuration, the same maximum effectiveness is reached at NTU of 15. For these two configurations, the effectiveness decreases by < 10% when the ratio of the delivered air to the room increases from zero to 60%. Boxem et al. [4] presented a model for an indirect evaporative cooler: a compact counter flow heat exchanger with louver fins on both sides. The model was used to predict the performance of a 400 m^3/h air flow cooler. The authors indicated that their calculations overestimated the cooler performance by 20% for inlet air temperatures below 24°C and by 10% for higher inlet temperatures. Anisimov et al. [5.6] proposed a combined parallel and regenerative-counter flow indirect evaporative cooler. Based on a mathematical analysis, they indicated that such a cooler would have higher efficiency than other types. Zhao et al. [7] presented a numerical study of a counter-flow indirect evaporative cooler that can achieve a sub-wet bulb temperature. They suggested a range of design conditions to maximise the cooler performance: inlet air velocity 0.3 - 0.5 m/s, height of air passage 6 mm or below, length-to-height ratio of air passage 200 and working-to-intake air ratio around 0.4. They mentioned that the cooler can give wet-bulb effectiveness of up to 1.3 under the UK summer design conditions [8]. Riangvilaikul and Kumar presented [9] experimental results for a sensible evaporative cooling system at different inlet air conditions (temperature, humidity and velocity) covering dry, temperate and humid climates. The results showed that wet bulb effectiveness ranged between 92 and 114%. A continuous operation of the system during a typical day of summer season in a hot and humid climate showed that wet bulb effectiveness was almost constant at about 102%.

The objective of this paper is to study, theoretically, a method to achieve sub-wet bulb temperatures for air produced by indirect evaporate cooling. Four different types of cooler configurations are studied and their performance is compared. A computational model based on mathematical analysis of the heat and mass transfer process inside a cooler is developed for this purpose.

2. INDIRECT EVAPORATIVE COOLING PROCESS

Figure 1 shows an indirect evaporative air cooler. Product air, which is supplied to the room, flows inside the dry passage. A working air stream flows inside the wet passage over a film of water located on a very thin non-permeable wall. This wall separates the dry and wet passages from each other. Therefore, the product air and the water film are not in a direct contact. Direct air-water contact takes place in the wet passage, which lowers the temperature of the water film. Heat transfers from the product air to the water film through the thin wall. This means that the product air is only sensibly cooled. Therefore, indirect evaporative air cooling does not increase the moisture content of the product air. This is a main advantage over direct evaporative air cooling. The enthalpy of the working air increases due, mainly, to evaporation of water to it. Because of its high humidity, the working air is exhausted to the atmosphere. The wet bulb temperature of the entering working air is the lower limit for the direct air-water contact in the wet passage. This means that the temperature of the delivered product air is, nevertheless, higher than that wet bulb temperature. According to the air flow direction in the two passages relative to each other, an indirect evaporative air cooler can be classified as a counter flow or a parallel flow cooler.

Fig. 1.

3. COMPUTATIONAL MODEL

An indirect evaporative air cooler is shown in Fig. 2. The heights of the two passages (y_1 and y_2) are very small (a few millimetres). The thickness of the thin wall and water film together is (d). Water, in the wet passage, is kept in place as a stagnant film. There are small masses of air flow inside the dry passage (M) and the wet passage (m). A low air velocity is obtained, which makes the air flow laminar in the two passages. The small height of the passages results in higher heat and mass transfer coefficients in this laminar flow. The surface of the wet passage, where the water film exists, is made of a porous media with a high water-retaining capacity, which maintains the water film in place. Zhao et al. [10] investigated several types of materials, namely metals, fibres, ceramics, zeolite and carbon, which have potential to be used as heat and mass transfer medium in indirect evaporative coolers. Their conclusion is that wick (sintered, meshes, grooves or whiskers) attained metals (copper or aluminium) are the most adequate structure/material over the others.

A one-dimensional model is developed to calculate the local distributions of temperature, enthalpy and humidity inside the evaporative air cooler. To find the solution, the cooler length (L) is divided into small elements (100 elements in this solution). An element of length (dx) containing three nodes (on dry air, wet air and water film) is shown in Fig. 2b. Heat and mass balance is applied to each element. The following assumptions are made to simplify the problem: (1) the cooler is assumed to be well insulated from its surroundings; (2) thermal conduction in the wall and water film in the x-direction is neglected; (3) the heat and mass transfer coefficients inside each passage are constants.

Fig. 2.

For the element shown in Fig.2b, heat transfer from air flowing inside the dry passage to the water film gives

$$M\ C_{\mathrm{p}}(T_{\mathrm{ni}} - T_{\mathrm{no}}) = U\,(T_{\mathrm{n}} - t_{\mathrm{fn}})\mathrm{d}A \qquad (1)$$

where (C_{p}) is the specific heat capacity of air and dA = Z dx. (U) is the overall heat transfer coefficient $U = \left(\frac{1}{\alpha_{\mathrm{d}}} + \frac{d}{k}\right)^{-1}$ where (d) and (k) are, respectively, the thickness and the thermal conductivity of the combined thin wall and the water film and (α_{d}) is the convective heat transfer coefficient on the dry air side.

For air flowing inside the wet passage, heat transfer between the air stream and the air-water interface consists of sensible and latent parts:

$$m(h_{\mathrm{no}} - h_{\mathrm{ni}}) = \alpha_{\mathrm{w}}(t'_{\mathrm{n}} - t_{\mathrm{n}})\mathrm{d}A + \beta(H'_{\mathrm{n}} - H_{\mathrm{n}})\,h_{\mathrm{fg}}\mathrm{d}A \qquad (2)$$

where (α_{w}) is the convective heat transfer coefficient on the air side in the wet passage, (β) is the mass transfer coefficient, (h_{fg}) is the latent heat of evaporation of water and (H'_{n}) is the humidity ratio of saturated moist air at the air-water interface temperature (t'_{n}). The enthalpy of air-water vapour mixtures can be represented by

$$h = C_{\mathrm{H}}\,t + h_{\mathrm{fg}}H \qquad (3)$$

where (C_{H}) is the humid specific heat capacity of air. Thus, the temperature of moist air can be written as

$$t = \frac{h - h_{\mathrm{fg}}H}{C_{\mathrm{H}}} \qquad (4)$$

Substituting for (t_n) and (t'_{n}) from Eq. (4) in Eq. (2) yields

$$m(h_{\mathrm{no}} - h_{\mathrm{ni}}) = \left\{\ \alpha_{\mathrm{w}}\left[\frac{(h'_{\mathrm{n}} - h_{\mathrm{fg}}H'_{\mathrm{n}})}{C_{\mathrm{H}}} - \frac{(h_{\mathrm{n}} - h_{\mathrm{fg}}H_{\mathrm{n}})}{C_{\mathrm{H}}}\right] + \ \beta(H'_{\mathrm{n}} - H_{\mathrm{n}})h_{\mathrm{fg}}\ \right\}\mathrm{d}A \qquad (5)$$

Thus

$$m(h_{no} - h_{ni}) = \left[\frac{\alpha_w}{C_H}(h'_n - h_n) + h_{fg}\beta(1 - \frac{\alpha_w}{\beta C_H})(H'_n - H_n) \right] dA \qquad (6)$$

The term ($\alpha_w / \beta C_H$) appearing on the right hand side of Eq. (6) is the Lewis relation. The magnitude of the Lewis number expresses relative rates of propagation of energy and mass within a system. For air-water vapour mixtures at low diffusion rates, where the heat-mass transfer analogy is valid, the Lewis relation could be taken as unity [11]

$$\alpha_w / \beta C_H \approx 1 \qquad (7)$$

Hence, Eq. (6) is reduced to

$$m(h_{no} - h_{ni}) = \beta(h'_n - h_n)\, dA \qquad (8)$$

Assuming that the liquid side of the interface offers a negligible resistance to heat transfer, so that the interface enthalpy (h'_n) in Eq. (8) is considered equal to (h''_n), the saturated air enthalpy at the water film temperature (t_f). Therefore

$$m(h_{no} - h_{ni}) = \beta(h''_n - h_n)\, dA \qquad (9)$$

Eq. (9) is called the Merkel equation [12]. This equation means that the energy transfer could be represented by an overall process based on enthalpy potential difference, between air-water interface and bulk air, as the driving force.

Now, from the energy balance on the two air streams flowing inside the element

$$M\, C_p\, (T_{ni} - T_{no}) = m(h_{no} - h_{ni}) \qquad (10)$$

The mass balance for the water vapour inside the wet passage of the element gives

$$m(H_o - H_i) = \beta(H''_n - H_n)\, dA \qquad (11)$$

The convective heat transfer coefficient (α) of air flowing in the two passages can be approximated by the following formula for fully developed laminar flow inside parallel plates with constant wall temperature [13]

$Nu = 4.861$ (12)

where (Nu) is the Nusselt Number. The mass transfer coefficient (β) in the wet passage is calculated from this formula using the Lewis relation. Equations (1, 9, 10 and 11) can now be solved, according to the type of air flow in the two passages (counter, parallel or regenerative flow), to find the four unknowns (T_n, h_n, t_{fn}, H_n) for each element when the inlet operating conditions to the cooler are given.

To validate this model, it is used in finding the performance of a counter flow regenerative indirect evaporative air cooler, which was experimentally measured by Hsu et al. [3]. The results are shown in Fig. 3, which indicates that the model can predict very well the cooler performance. The deviation between the model results and the experimental data for the outlet temperature of the cooler is 7.4%.

Fig. 3.

4. METHOD FOR A SUB-WET BULB TEMPERATURE PROCESS

The wet bulb temperature of ambient air is the ultimate temperature for indirect evaporative air cooler. This leads one thinking about a method to produce air at a sub-wet bulb temperature. The main idea to achieve this objective consists of branching the working air from the product air, which is indirectly pre-cooled, before it is finally cooled and delivered. Four types of coolers are studied in this paper: three two-stage coolers (a counter flow, a parallel flow and a combined parallel-regenerative flow) and a single-stage counter flow regenerative cooler.

The dimensions of the cooler in the examples studied are: $y_1 = y_2 = 0.0035$ m, $L = Z = 0.5$ m. The water film thickness is 1 mm and the wall thickness is 0.5 mm, which makes $d = 0.0015$ m. Therefore a volume of 250 cm^3 of water is needed to keep the thickness of the water film. The mass flow rates of air are: total inlet air 0.0014 kg/s, total working air 0.00098 kg/s, and the product air 0.00042 kg/s. This latter is equal to1.3 m^3/h. The air flow rates in these examples are selected to demonstrate the sub-wet bulb concept but are still indicative for a real application. For a cooler with 100 two-passage cells packed together, the face area is about 0.5 m^2, and the total product air flow rate is 130 m^3/h. This is equivalent to 0.5 air changes per hour if this cooler supplies ventilation air to a house with 100 m^2 floor area and 2.6 m internal height. Depending on how big the cooling load of the house is, the rest of the cooling load could be handled by water-based units inside the house (e.g. chilled ceilings, cooling panels or under-floor cooling) connected to a closed-wet cooling tower which directly supplies water to these units [14,15]. This offers a complete sustainable cooling solution to the house. With an outlet temperature of product air of 16 °C and a room temperature of 26 °C, this delivered air flow gives a cooling effect of 420 W, which is 3.2 W per m^3/h. This is approximately in the middle of the range mentioned by Zhao et al. [16] for cooling output from such coolers. However, it is still possible to handle the whole cooling load of the house by evaporative cooling by optimising the air flow rate in the dry and wet passages and using more than one cooler. There are indirect evaporative cooler products available in the market from different manufacturers that are basically able to supply air at sub-wet bulb temperatures and mainly operate according to the regenerative principle. The range of product air flow rate in these coolers is from 400 to 4000 m^3/h.

The assumed inlet operating conditions for the cooler in the examples studied are: inlet air is the ambient air at a dry bulb temperature of 30 °C and a humidity ratio of 0.009 kg water/kg dry air (relative humidity = 34%). Therefore, the ambient air wet bulb temperature T_{wb}= 18.8 °C and the dew point T_{dp} = 12.5 °C.

5. RESULTS AND DISCUSSION

5.1. Two-stage counter flow cooler

In the multistage arrangement, the working air for a next stage is branched from the product air of a previous stage. This can be seen in Fig. 4, which shows an example of a two-stage counter flow arrangement. The total length in this example is equally divided into two stages. The total flow rate of inlet air is 0.0014 kg/s. The product air flow rates are: M_1 = 0.000767 kg/s and M_2 = 0.00042 kg/s. The total flow rate of working air (m = $m_1 + m_2$) is 0.00098 kg/s, where m_1 = 0.000633 kg/s and m_2 = 0.000347 kg/s. These flow rates are selected so that $m_1/M_1 = m_2/M_2$ = 0.825. The flow ratio of total working air to total inlet air $m/ (M_1 + m_1)$ = 0.7.

The model results for the product air temperatures are indicated on Fig. 4. The temperature of product air from the first stage (T_{out1}) is 20.6 °C. The temperature of final product air (T_{out2}) is 15.9 °C. This is lower than the wet bulb temperature of ambient air (18.8 °C). The whole process is shown on a pscychrometric chart in Fig. 5. The main advantage comes from the fact that, the wet bulb temperature of the working air entering the second stage, which is branched from the product air leaving the first stage, is lower than the ambient air wet bulb temperature (15.6 °C compared with 18.8 °C) and as shown in Fig. 5. This means that the air in the second stage is trying to approach 15.6 °C. This is the main concept of the method presented in this paper: pre-cooling of the working air by branching it from the product air, which has already been cooled, before it is finally used. The cooling effectiveness for this two-stage process is: wet bulb effectiveness E_{wb} = 1.26 and dew point effectiveness E_{dp} = 0.81. The temperature distributions for the air flowing in the dry and wet passages and that for the water film along the two stages are shown in Fig. 6. The arrows in the figure refer to the direction of the air flow in the air passages for each stage. However, there is a big difficulty in constructing such a counter flow cooler because there are many complications in diverting the product air from the end of a previous stage to the end of a next stage to make it act as working air.

Fig. 4.

Fig. 5.

Fig. 6.

5.2. Two-stage parallel flow cooler

Making the direction of the working air flow in Fig. 4 in the same direction of the product air, a two-stage parallel flow is obtained (Fig. 7). This will affect the temperature of the delivered product air due to different characteristics of the parallel flow compared with the counter flow. In this parallel flow example, the total inlet air, working air and outlet air flow rates and the internal flows (M_1, m_1, M_2 and m_2) are similar to those for the two-stage counter flow. The temperature of the product air from the first-stage (T_{out1}) is 22.1 °C, and the final product air temperature from the second-stage (T_{out2}) is 17.8 °C. This latter is also a sub-wet bulb temperature, but it is higher than that produced by the two-stage counter flow arrangement. On the other hand, construction of the parallel flow cooler would be possible because the branching point of the working air is on the same side with its entrance to the wet passage. The wet bulb effectiveness E_{wb} =1.09 and the dew point effectiveness E_{dp} = 0.70. The process is indicated on the psychrometric chart in Fig. 8. The wet bulb temperature of the working air at its entrance to the second stage is 16.1 °C. The temperature distribution inside the cooler is presented in Fig. 9.

Fig. 7.

Fig. 8.

Fig. 9.

5.3. Single-stage counter flow regenerative cooler

Another way for delivering product air at a temperature lower than the ambient wet bulb temperature is to use a counter flow regenerative evaporative cooler. Fig. 10 shows the arrangement for this type of coolers. The working air is branched from the product air, which is indirectly pre-cooled. This allows the wet bulb temperature of the inlet working air to be lower than that for the ambient air. This is in accordance with the principle indicated for the two-stage counter and parallel flow examples studied before. This type is explained in detail in the counter flow regenerative example shown in Fig. 11. It has the same dimensions (y_1, y_2, L, Z and d) and total air mass flow rates as those for the two-stage counter flow and parallel flow examples described before (total inlet air = 0.0014 kg/s, product air = 0.00042 kg/s, working air = 0.00098 kg/s and m/M = 0.7). The process is indicated on the psychrometric chart in Fig. 12. It can be noted that the temperature of the product air (T_{out}) is 17.0 °C, which is lower than the wet bulb temperature of ambient air. The wet bulb temperature of the working air at its entrance is 14.3 °C. The achieved temperature of the product air is lower than that for the two-stage parallel flow, but is higher than that for the two-stage counter flow. The wet bulb effectiveness E_{wb} = 1.16 and the dew point effectiveness E_{dp} = 0.74. This is a major advantage achieved by this single-stage cooler compared with typical single-stage parallel or counter flow coolers, which are limited by the wet bulb temperature of ambient air. For these two latter types, a two-stage unit is needed to achieve a sub-wet temperature, where the first stage acts as a pre-cooler of the working air, as noticed in the two examples described before. The diversion of the working air from the dry passage to the wet passage in the regenerative cooler is easy because it happens at one end of the cooler, which makes the construction possible.

Fig. 10.

Fig. 11.

Fig. 12.

More results are shown in Figs. 13-15. From Fig. 13, we can note that the product air loses its heat to the water film because its temperature (T) is higher than that for the water film (t_f). This latter is higher than the working air temperature (t) for most of the length, but not for the last 23% of (L) close to the branching point. In the two other figures (Figs. 14-15), the local distribution of air humidity and enthalpy and rate of heat transfer in this cooler are presented. Figure 14 shows the properties of the working air along the cooler length. The wet air enthalpy, humidity ratio and relative humidity increase along its direction of flow starting from $x = L$ and exiting at a saturation state. The local distribution of heat transfer is presented in Fig. 15. This is dependent on the temperature and humidity gradients. Heat lost by the product air (Q_{dry}) is equal to that gained by the working air (Q_{wet}), and they fall on one line in Fig. 15. Due to higher temperature gradients at the beginning of the heat transfer area, the total rate of heat transfer on the left hand side of this figure is higher than that for the right hand side. The amount of heat gained by the working air in the wet passage is a result of summation of the sensible heat (Q_{sen}) and latent heat (Q_{lat}). The sensible heat is negative for the final 23% of (L) according to the definition of Eq. 2, which is due to higher temperatures of the working air with respect to the water film and as indicated in Fig. 13. One can notice that the latent heat transfer on both ends of Fig. 15 is higher than that for the middle part. This is related to the gradient of the humidity ratio ($H'' - H$), which is higher on the ends as indicated in Fig. 14.

Fig. 13.

Fig. 14.

Fig. 15.

5.4. Combined parallel-regenerative cooler

Parallel flow of air in the wet and dry passages makes better thermal performance in the beginning of the transfer area compared with counter flow or regenerative flow. This is due to lower water film temperature for the parallel flow at the beginning of the cooler. The regenerative flow is better on the other end of the cooler. Anisimov et al. [5,6] referred to this behaviour and suggested a cooler with a combined air flow that consists of two stages: a parallel flow for the first part and a regenerative flow for the remaining part of the cooler.

Fig. 16 shows an example of this type of coolers. The total air flow rates (total inlet flow, total working air and final product air) are similar to those for the three examples studied before. The total inlet air flow rate = 0.0014 kg/s. The flow rates for the parallel flow are: M = 0.000767 kg/s, m_1 = 0.000633 kg/s, thus m_1/M = 0.825, which are similar to those for the first stage of the two-stage counter flow and parallel flow coolers studied before. The working air flow in the regenerative part m_2 = 0.000347 kg/s and m_2/M = 0.452. The parallel flow part and the counter flow regenerative part cover 20% and 80% of the total length (L), respectively. The model results indicate that the product air temperature from the first part (T_{out1}) is 24.1 °C and from the second part (T_{out2}) 15.3 °C. The effectiveness E_{wb} = 1.31 and E_{dp} = 0.84. These are better than those obtained by the previous three examples. The process is indicated on the psychrometric chart in Fig. 17. In Fig. 18, the temperature distribution along the cooler is shown. It is clear that the advantage of cooling by parallel flow in the first stage is well utilised. It makes the product air temperature to drop from 30 to 24.1 °C in 20% of the total length. The cooler then makes use of the good features of the regenerative flow in the second stage, achieving the final temperature for (T).

Fig. 16.

Fig. 17.

Fig. 18.

To evaluate the performance of the combined parallel-regenerative cooler with respect to the two-stage counter flow cooler, which has the most complex structure, Fig. 19 shows the final outlet temperature from the coolers (T_{out2}) with different length ratios (L_1/L). It is apparent from this figure that the optimal length ratio for the two-stage counter flow still gives higher final outlet temperature compared with the selected length (L_1/L= 0.2) for the combined parallel-regenerative type. This confirms the advantage of the combined processes in the latter cooler.

Fig. 19.

6. CONCLUSIONS

A computational model for an indirect evaporative cooler is developed based on mathematical analysis of the heat and mass transfer process inside the cooler. The model results showed very good agreement when validated against available experimental data from literature. From the analysis presented in this paper, it is concluded that indirect evaporative cooling is able to supply air at temperatures lower than the ambient wet bulb temperature when implementing the proposed method. The idea is to manipulate the air flow by branching the working air from the product air, which is indirectly pre-cooled, before it is finally cooled and delivered.

The wet bulb cooling effectiveness (E_{wb}) for the examples studied is 1.26, 1.09 and 1.31 for the two-stage counter flow, parallel flow and combined parallel-regenerative cooler, respectively, and it is 1.16 for the single-stage counter flow regenerative cooler.

Referring to the different processes for sub-wet bulb temperature cooling indicated on the psychrometirc charts in Figs. 5, 8, 12 and 17, it is concluded that with higher number of staged coolers, which work according to the concept indicated in this paper, the ultimate temperature to be reached is the dew point of ambient air. Therefore, we now can talk about “approach to the dew point of ambient air” when using these indirect evaporative coolers instead of the commonly used “approach to the wet bulb temperature”.

The cooling effect obtained by any indirect evaporative coolers is dependent on both the temperature and flow rate of the delivered product air to the room. For a specified total inlet air flow rate for a cooler, increasing the working air flow rate results in a lower temperature, but also a lower flow rate, for the delivered product air, and vice versa. This is then an optimisation problem where the objective is maximising the cooling power to the delivered product air.

The method presented in this paper extends the potential of useful utilisation of evaporative cooling for the purpose of cooling of buildings in terms of lower product air temperature. The same principle could also be applied to water-based cooling systems, which utilise evaporative cooling for the rejection of heat to the atmosphere (e. g. cooling towers).

This method is not limited to applications in cooling of buildings, but can also be applied to other industrial applications where indirect evaporative cooling is used.

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Fig. 19. Final outlet temperature (T_{out2}) from the two-stage counter flow cooler and the combined parallel-regenerative cooler with different length ratios for the two stages.

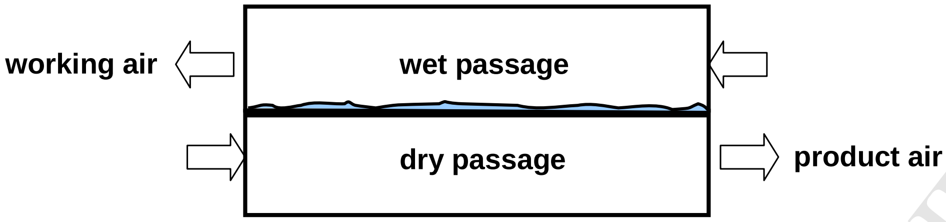

Fig. 1. A counter flow indirect evaporative air cooler.

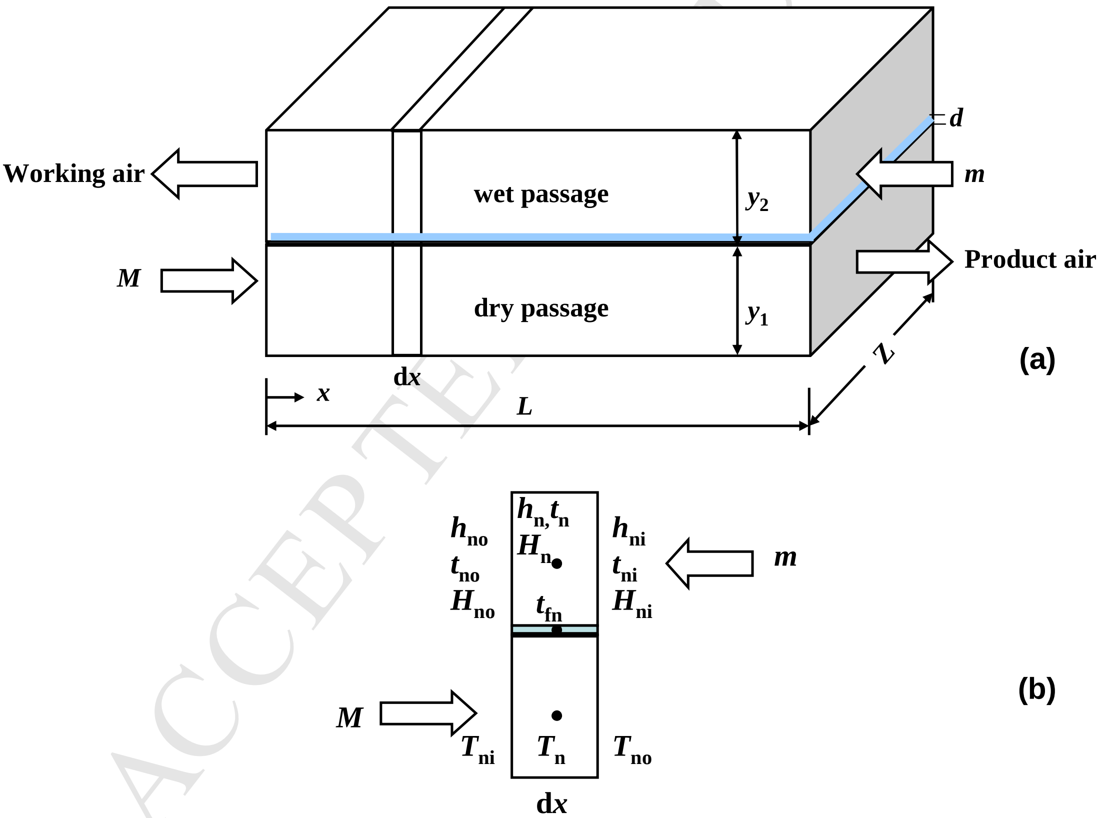

Fig. 2. (a) Indirect evaporative air cooler, (b) Dividing a cooler into elements.

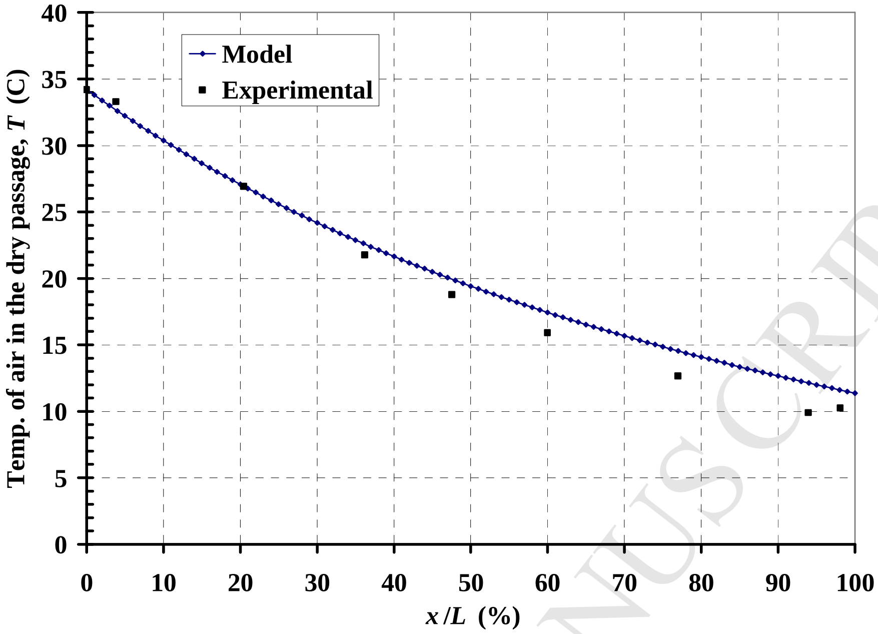

Fig. 3. Model results for the product air temperature (T) along the cooler in comparison with experimental data from [3].

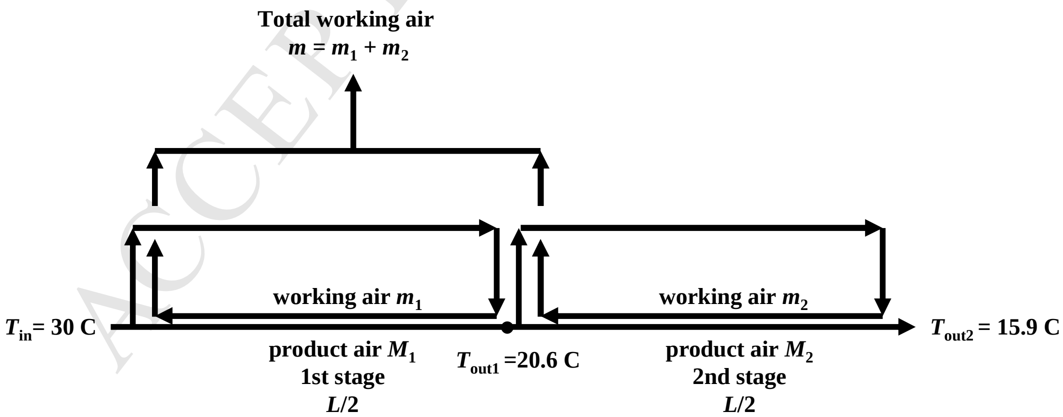

Fig. 4. A two-stage counter flow example.

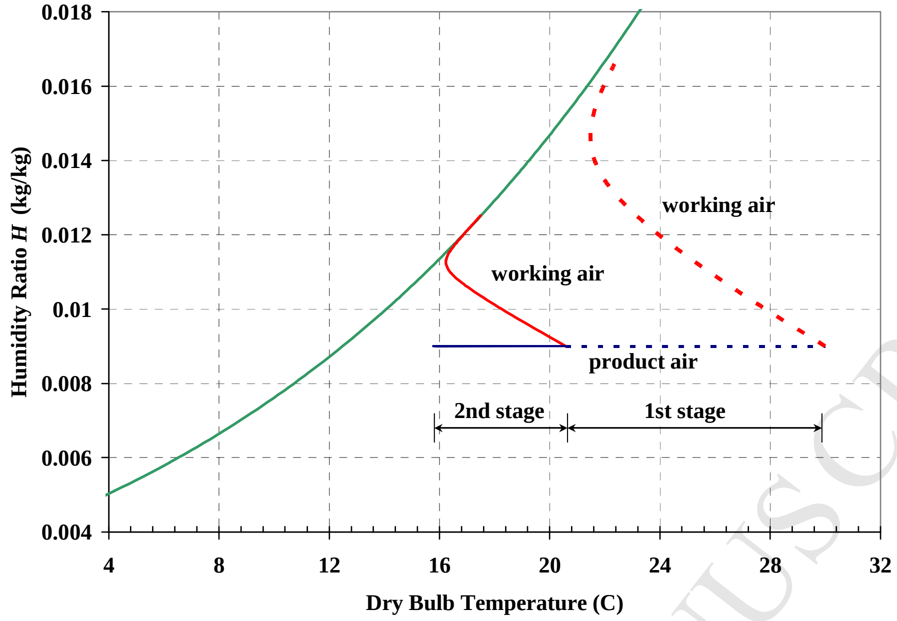

Fig. 5. Product air and working air conditions on the psychrometric chart for the two-stage counter flow example.

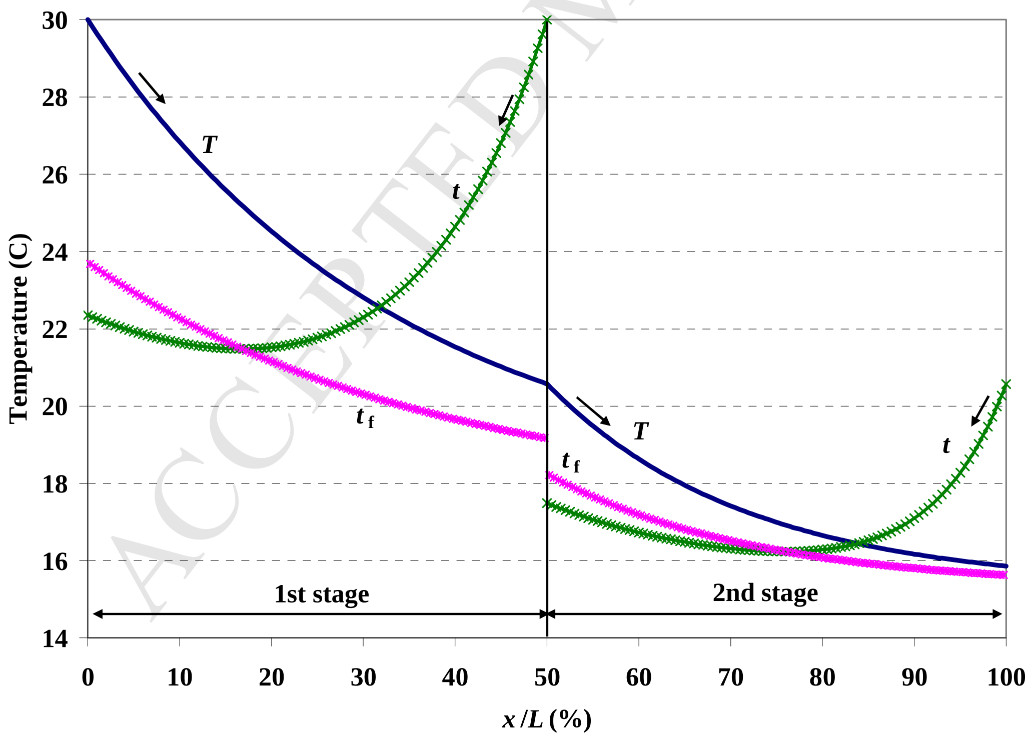

Fig. 6. Temperature distribution of product air (T), working air (t) and water film (t_f) in the two-stage counter flow example.

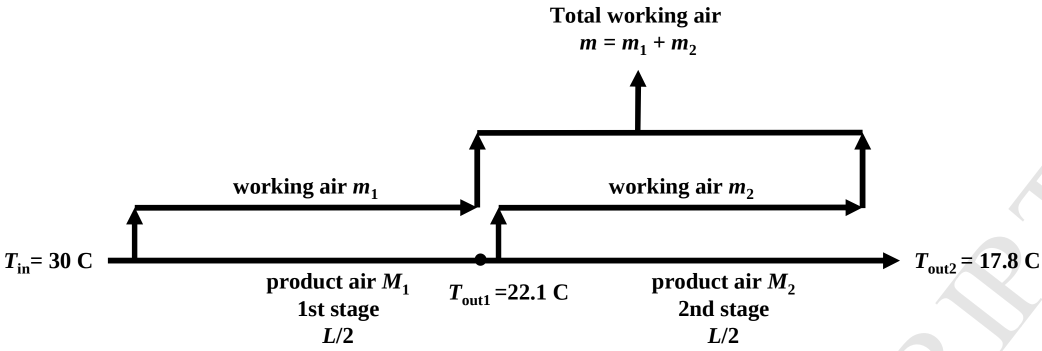

Fig. 7. A two-stage parallel flow example.

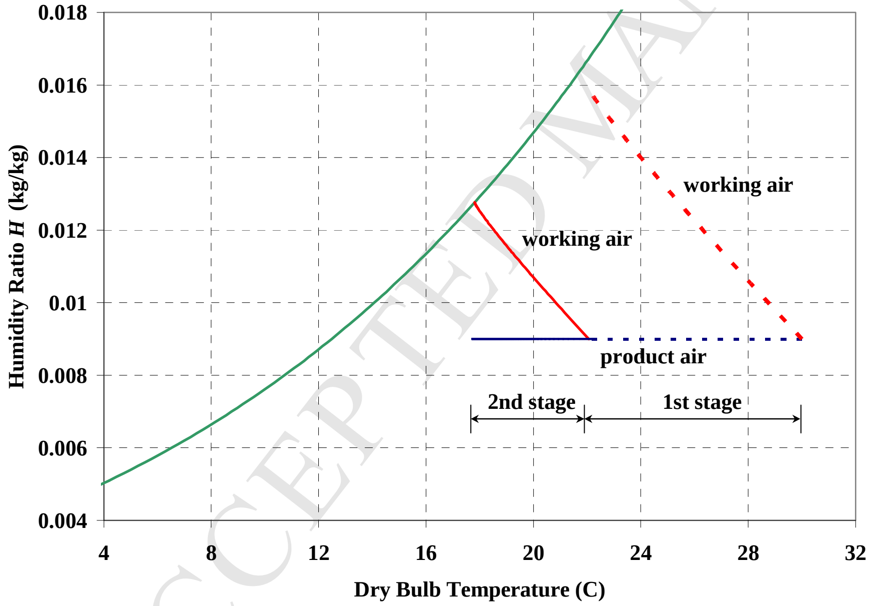

Fig. 8. Product air and working air conditions on the psychrometric chart for the two-stage parallel flow example.

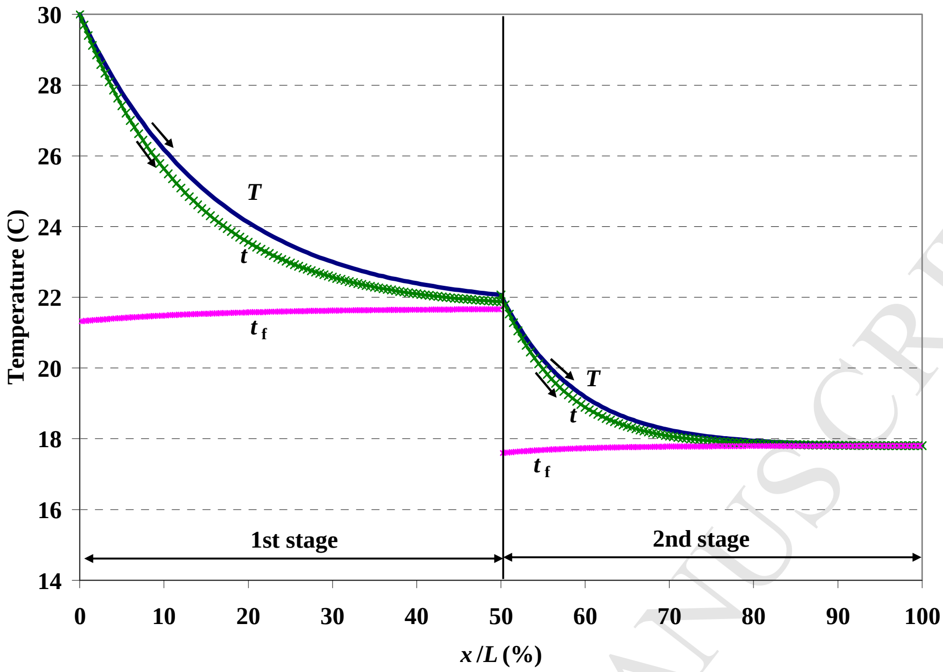

Fig. 9. Temperature distribution of product air (T), working air (t) and water film (t_f) in the two-stage parallel flow example.

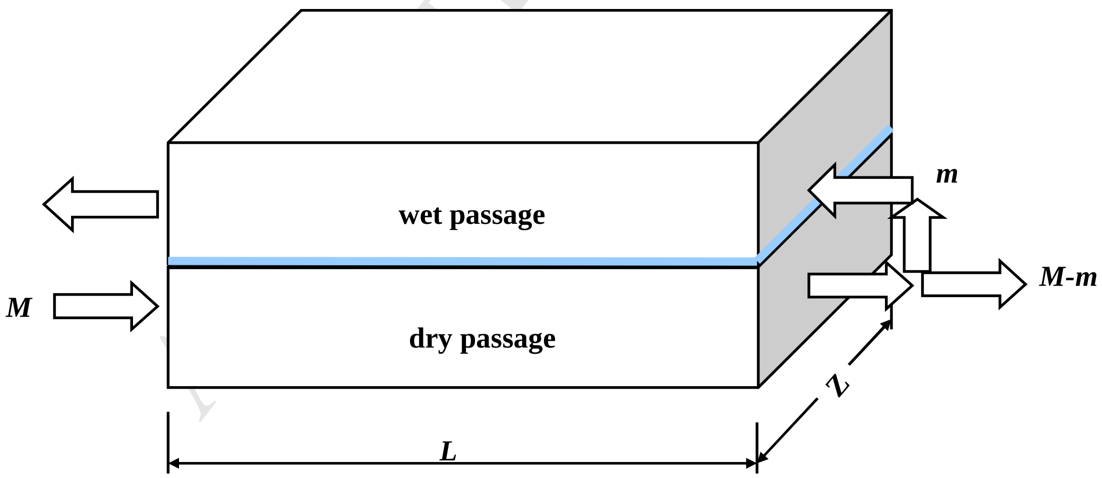

Fig. 10. Arrangement for a single-stage regenerative air cooler.

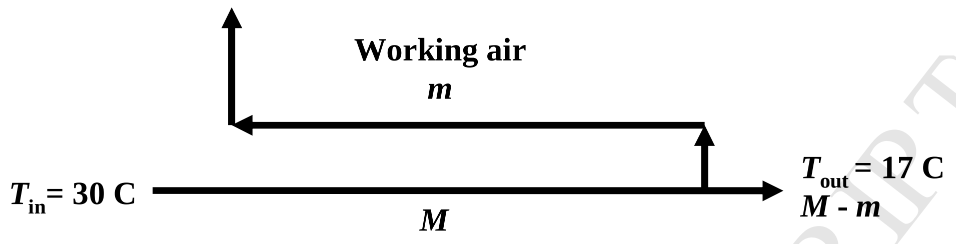

Fig. 11. A single-stage counter flow regenerative example.

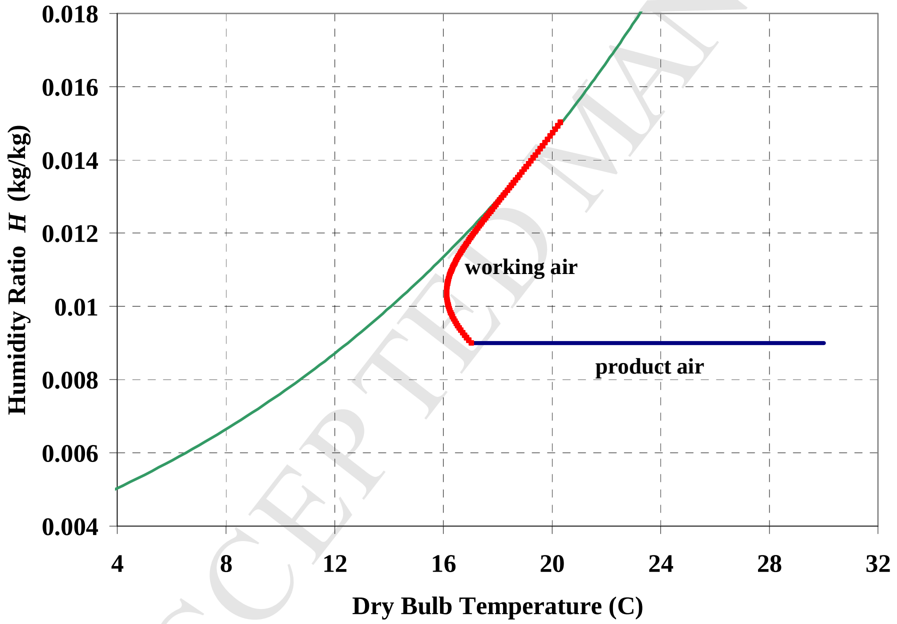

Fig. 12. Product air and working air conditions on the psychrometric chart for the counter flow regenerative example.

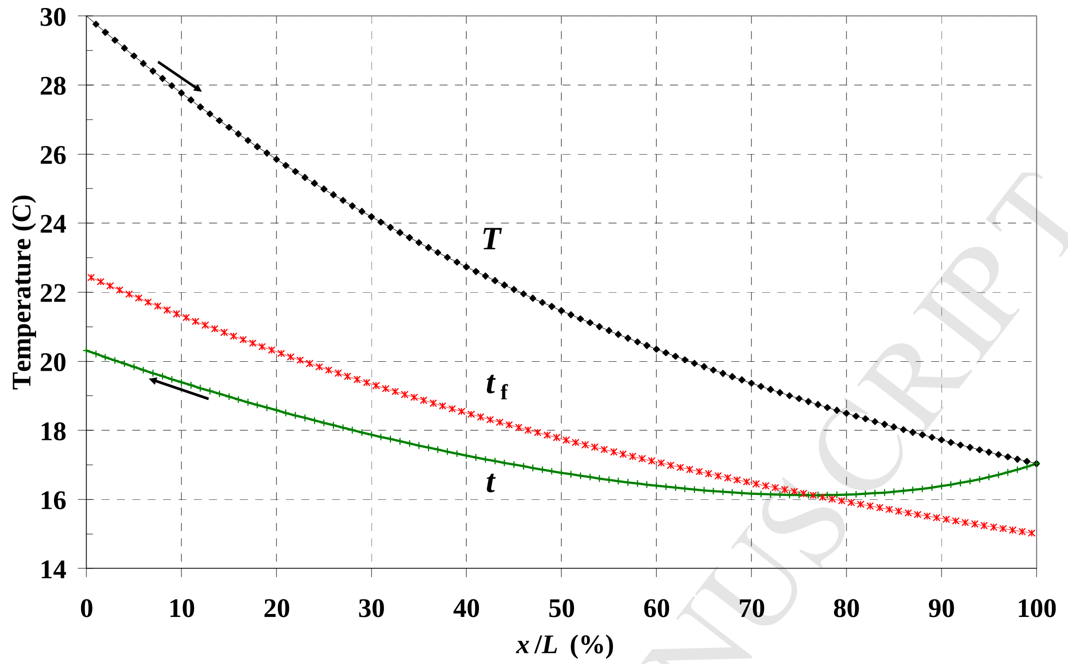

Fig. 13. Temperature distribution of product air (T), working air (t) and water film (t_f) in the counter flow regenerative example.

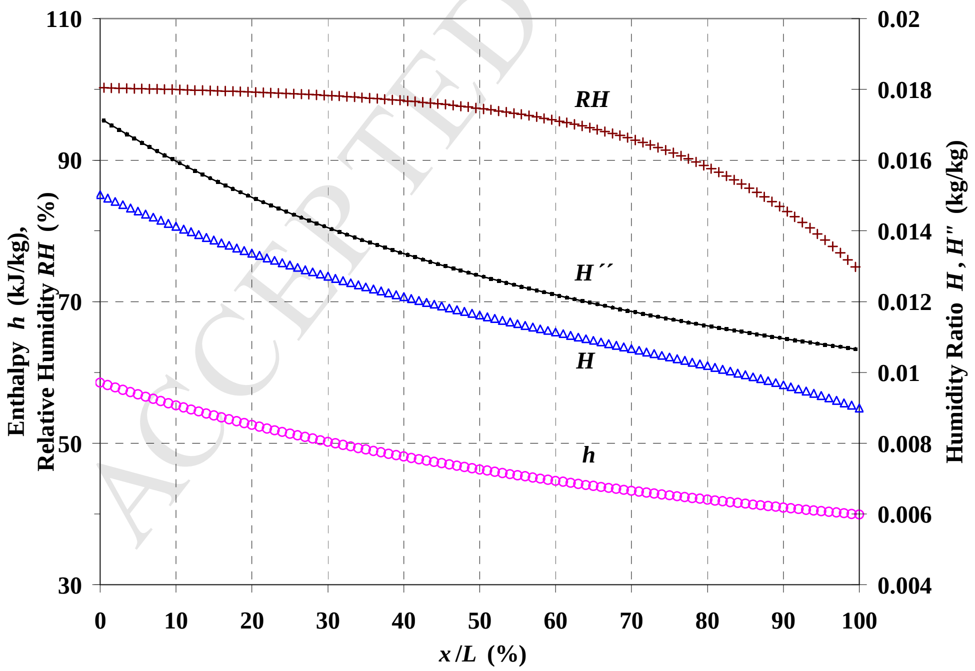

Fig. 14. Properties of the working air in the wet passage of the counter flow regenerative example.

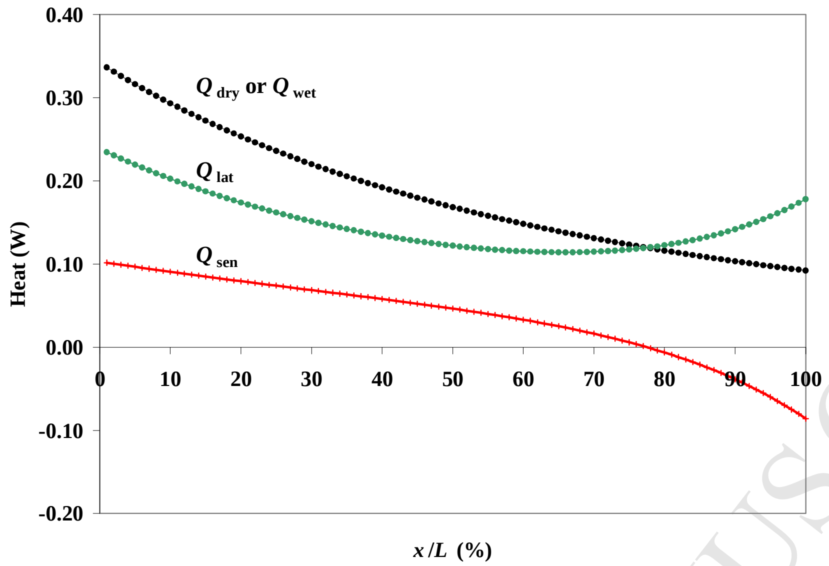

Fig. 15. Local rate of heat transfer for the counter flow regenerative example.

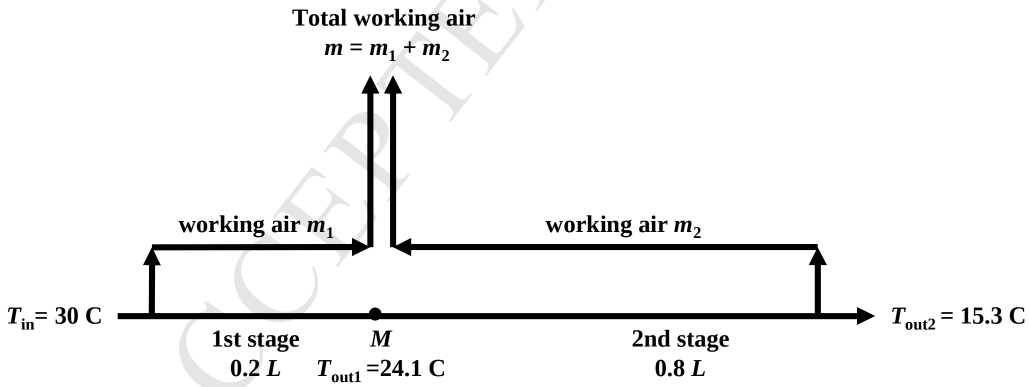

Fig. 16. Combined parallel-regenerative flow example.

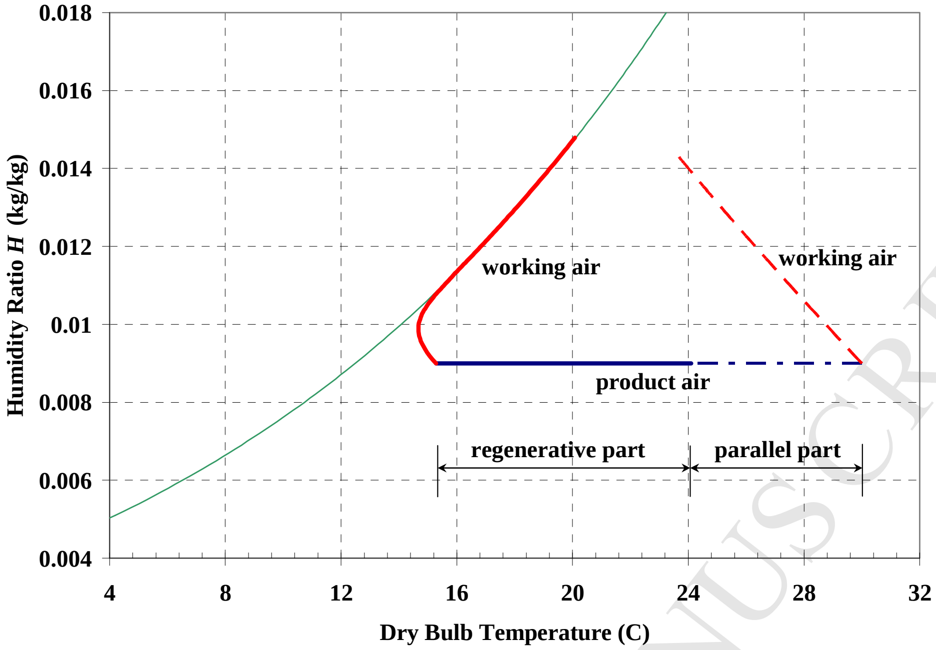

Fig. 17. Product air and working air conditions on the psychrometric chart for the combined parallel-regenerative flow example.

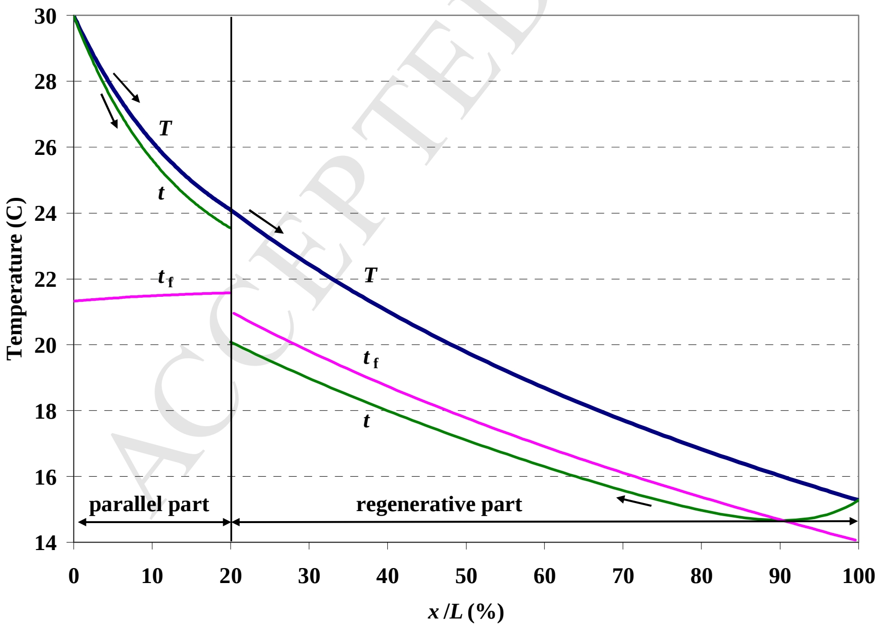

Fig. 18. Temperature distribution of product air (T), working air (t) and water film (t_f) in the combined parallel-regenerative flow example.

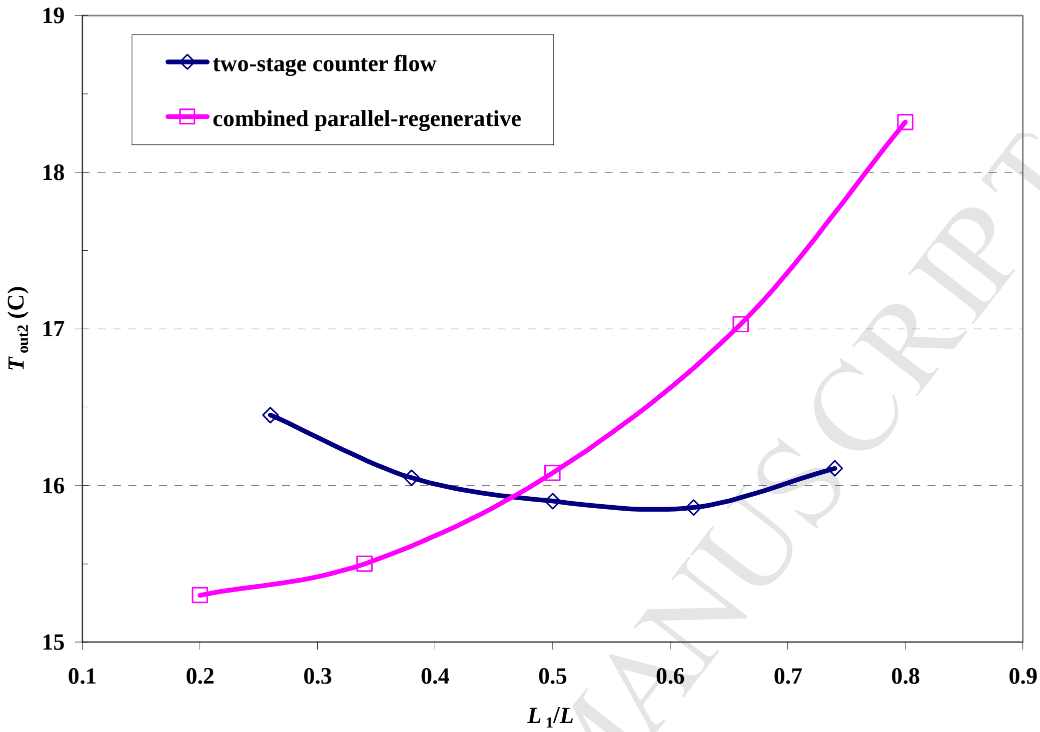

Fig. 19. Final outlet temperature (T_{out2}) from the two-stage counter flow cooler and the combined parallel-regenerative cooler with different length ratios for the two stages.