

Numerical study and design of a dew point evaporative cooler for buildings

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Abstract. Refreshing air remains a crucial problem in warm climates where electricity consumption for air conditioning has become excessive and irrational for several years, notably in Algeria. Research in this field is increasingly oriented towards new techniques that can reduce costs and environmental impacts. Among these techniques, the evaporative dew point cooling technology is the most promising as it can cool outdoor air to temperatures below its wet bulb temperature. The aim of this work is to model and design a dew point cooler for french and algerian climates. This model is used to study the effect of the cooler parameters such as its length, water temperature and working air ratio on its cooling effectiveness and supply temperature.

1 Introduction

Building sector is the largest energy consuming sector in the Mediterranean region due to lacking awareness about the importance of thermal insulation and housing stock aging. The global warming as well as the depletion of fossil fuels are forcing the construction sector to reduce its energy demand.

In France, 40% of energy consumption is due to buildings and, in Algeria, it raises up to 42% [1]. In particular, the air-conditioning sector is growing fast. In 2010, about 5% of French and 16% of Algerian households were equipped with air conditioning systems which respectively represented an electricity consumption of 450 kWh and 700 kWh per equipped household [2]. It is therefore urgent to find viable alternatives to traditional air conditioning systems. If the optimization of the architectural design improves indoor thermal comfort conditions, it may be incompatible with the architects needs or it can reach its limits during the heat wave periods.

In this context, the concept of evaporative cooling is a proven alternative that contributes to environment preservation. It has low CO₂ emissions and is an economically feasible method for cooling air [3]. The consumed electricity is only used to circulate air and water (fan and pump), which makes it less energy-demanding than classical cooling systems with an economy up to 90% [4].

In regions with a dry and hot climate, this system can be energy efficient and of a positive contribution to the environment [5]. Among the most widely used

evaporative systems are the direct and indirect systems that also have some disadvantages like high humidity supply for the direct system or reduced efficiency for the indirect system [6-8]. For both cases cooling is limited to ambient wet bulb temperatures. Recently new technologies have broken these barriers by using novel heat exchangers and flow path arrangements so that temperatures below wet bulb temperature are reached. These technologies are known as dew point evaporative cooling systems.

Maisotsenko introduced a new design of the heat exchanger of the indirect system [9]. It is a combination of a cross-flow, multi-perforated flat plate and an evaporative cooler, in which the secondary air is pre-cooled in the dry channel before being deviated and passed through the wet channel to obtain, in addition, the heat transfer with the dry channel. The primary air temperature is thus lower than air wet bulb temperature and approaches the incoming air dew point. Hasan [10] presented a theoretical model of several configurations of dew point coolers. He concluded that the ultimate temperature at which the supply air can be cooled is its dew point temperature.

Riangvilaikul et al. [11] have experimentally studied an evaporative dew point cooler with different air inlet conditions (velocity, humidity, and temperature). The results show that the wet bulb effectiveness is high and varies between 92 to 114%. Boukhanouf et al. [13] studied numerically and experimentally a configuration of a counter-current dew point cooler using a saturated porous media instead of the water film. They managed to achieve a system efficiency of the order of 1.024 for a cooling capacity of the order of 225 W.m⁻².

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Lee et al. [13] used counter-current with a wet return surface cooler and showed that temperatures below wet bulb can be achieved. The authors built and tested a prototype with a finned aluminum plate to optimize heat transfer. The results show that for an air inlet temperature of 32 °C with a relative humidity of 50%, the outlet temperature of the dry channel is 22 °C, which is below the corresponding wet bulb temperature of 23.7 °C.

The purpose of this paper is to model and validate a dew point evaporative cooler able to be used for both climates in France and Algeria. First, the system is presented, then its mathematical model, and implementation in SPARK [14,15], a program-oriented object, are shown. The model is validated using data from literature [10-11]. Finally, simulations are run to study the system parameters variation on its cooling effectiveness and supply air temperature.

2 System description

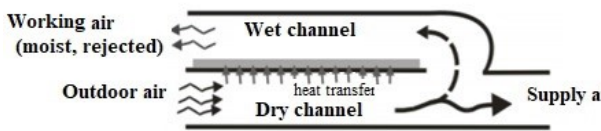


Fig. 1. Schematic representation of the studied evaporative dew point cooler.

Figure 1 shows the studied evaporative dew point cooler. It is made of a counter flow heat exchanger of two channels (dry and wet channel) exchanging heat through a thin wall. The latter is in contact with air flow in the wet channel through a thin water film or a saturated wet media whereas it is made impermeable to moisture on its back surface in contact with the dry channel. Intake outdoor air enters the dry channel and loses heat to the wet channel. At the end of the channel, it splits into two parts: supply dry air which is delivered to the conditioned space and the remaining part called working air is diverted into the wet channel, it absorbs heat from the dry channel as well as moisture evaporating from the wet wall. The ratio of airflow in the wet channel to that of the dry is called working air ratio and it generally varies between 0.3 and 0.7 [16].

3 Mathematical Model

To simplify problem analysis the following assumptions have been made [10, 11, 13, 16]:

- (1) the heat exchanger is adiabatic; there is no heat transfer between system and the surroundings;
- (2) the height of the channels is small compared to their width, thus the flow is unidirectional;
- (3) the water film is evenly distributed over the entire surface of the plate;
- (4) the water and the plate are at the same temperature;
- (5) the

airflow is supposed to be stable and incompressible; (6) the water film is renewed in a constant way ; (7) heat transfer via the channel walls is in the vertical direction.

Considering the computational element shown in Figure 2, the energy conservation balance for the airflow in the dry channel gives:

$$\frac{\dot{m}_d C_{pa}}{D} \frac{dT_d}{dx} = -U_d(T_d - T_{fw}) \quad (1)$$

where (T_d) is air dry bulb temperature in the dry channel, (T_{fw}) water film surface temperature, (U_d) the overall heat transfer coefficient between the dry channel and water film, (D) channel width and (\dot{m}_d) airflow rate in the dry channel.

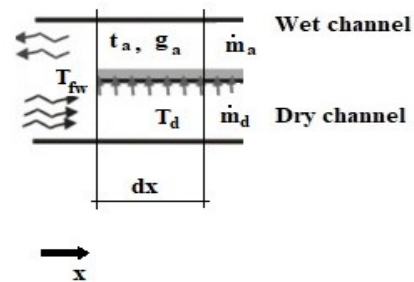


Fig. 2. Cross section of the computational element.

The energy conservation balance for the airflow in the wet channel gives:

$$\frac{\dot{m}_a C_{pa}}{D} \frac{dt_a}{dx} = -h_{aw}(T_{fw} - t_a) - \rho h_m(g_{fw} - g_a)C_{pv}t_a \quad (2)$$

where (t_a) is air dry bulb temperature in the wet channel, (g_a) is the humidity ratio of air (kg/kg_d) and (g_{fw}) is humidity ratio at saturation near water film. The difference between both humidity ratios is the driving force of water evaporation in the wet channel. (\dot{m}_a) is airflow rate in the wet channel. (ρ) is mass density of humid air, (h_{aw}) is the heat transfer coefficient in the wet channel and (h_m) is the convective mass transfer coefficient between the wet airflow and the water film surface. They are related by the Lewis relation:

$$\frac{h_{aw}}{h_m} = \rho C_p Le^{2/3} \quad (3)$$

Applying the mass conservation equation to the air in the wet side of the computational channel gives:

$$\frac{\dot{m}_a}{D} \frac{dg_a}{dx} = -\rho h_m(g_{fw} - g_a) \quad (4)$$

Considering the energy balance of a coupled dry and wet element on the media between the channels gives:

$$\frac{\dot{m}_{fw} C_{pfw}}{D} \frac{dT_{fw}}{dx} = -U_d(T_d - T_{fw}) + \rho h_m(g_{fw} - g_a)h_{fg} + h_{aw}(T_{fw} - t_a) \quad (5)$$

To solve this system of equations, the Simulation Problem Analysis and Research Kernel (SPARK) is used. The latter allows solving efficiently differential equation systems [14]. The equations system was discretized using the finite difference method and the physical system was divided into 20 computational elements.

4 Model validation

To validate the numerical model, the simulation was carried out based on a similar dew point cooler module which had published with numerical or experimental results (Hasan [10] and Riangvilaikul et al. [11]). Table 2 shows the cooler parameters used in each of these studies as well as inlet air climatic conditions.

Table 2. Cooler parameters and climatic conditions used in the published data.

| Parameters | Hasan [10] | Riangvilaikul [11] |
|--|------------|--------------------|
| Channel length (m) | 0.5 | 1.2 |
| Channel width (m) | 0.5 | 0.08 |
| Channel gap (m) | 0.0035 | 0.005 |
| Velocity of inlet (m.s ⁻¹) | 0.676 | 2.4 |
| Working air ratio | 0.7 | 0.33 |
| Inlet air temperature (°C) | 30 | 25-30-35-40-45 |
| Inlet air humidity ratio(g/kgd) | 9 | 6.9-11.2-20-26.4 |

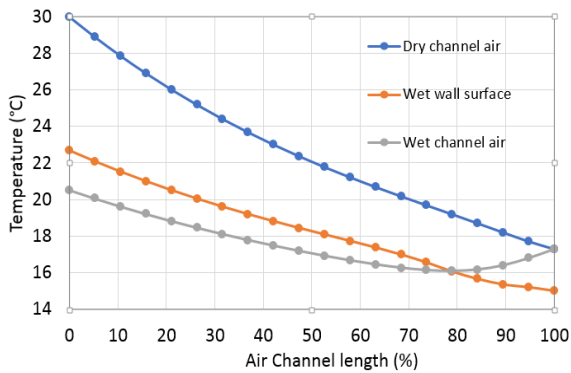


Fig. 2. Numerical temperature profiles along dry and wet channel given by SPARK.

Figure 2 shows simulation dynamic profiles for air temperature in the dry and wet channel as well as the wet wall surface temperature. These results are obtained from the SPARK model applied to Hasan [10] case study. Air in the dry channel flows in the opposite side of the air in the wet channel. One can notice that inlet air temperature drops from 30°C to 17.3°C in the dry channel which is lower than air wet bulb temperature (18.8°C).

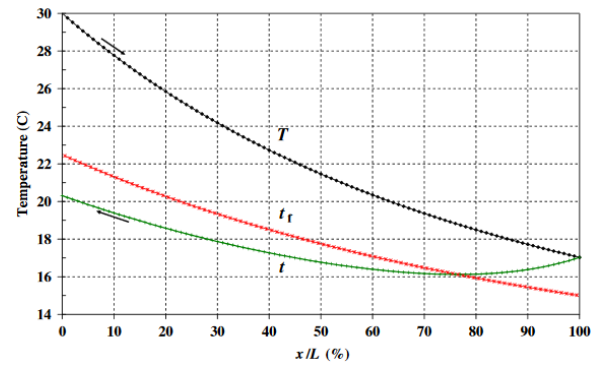


Fig. 3. Temperature profiles given by Hasan [10].

For the wet channel, working air enters at 17.3°C, its temperature keeps dropping for a short initial stage (about 20% of the channel length) because of low temperature of water on the wet wall surface and then it begins increasing because of heat transfer from the dry channel. Concerning the temperature of the wet wall surface it keeps increasing with airflow in the wet channel because of heat from the dry channel.

Figure 3 shows data presented by Hasan et al. [10]. A good agreement is found with those shown in Figure 2 for SPARK results. In this case, air is supplied at 17°C which is a difference of 0.3°C with SPARK or 1.76%.

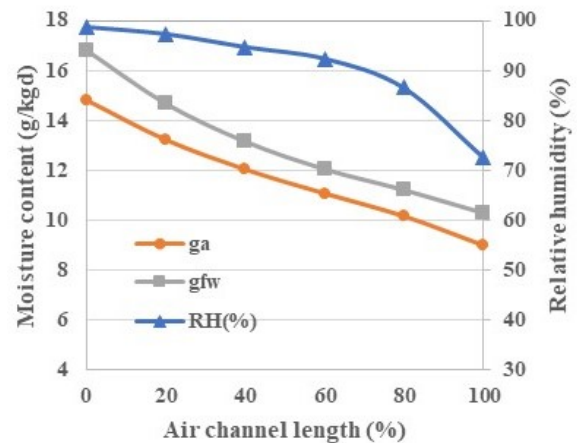


Fig. 4. Air properties in the wet channel as computed by SPARK (moisture content g and relative humidity RH).

Figure 4 shows relative humidity and moisture content of air in the wet channel g_a as well as saturation moisture content at wet wall temperature g_{fw} . Channel length of 100% corresponds to wet channel entrance and the 0% is for outlet. The moisture content increases from 9 g.kgd⁻¹ at the inlet of the wet channel to 14.82 g.kgd⁻¹ indicating that air exits the wet channel fully saturated. The difference ($g_{fw} - g_a$) is the driving force for water evaporation in the channel. These results are also validated by Hasan data.

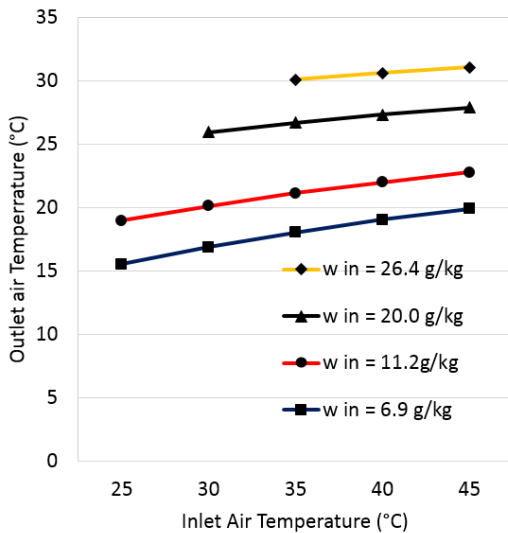


Fig. 5. SPARK numerical results for different inlet air conditions as studied in Rianguilaikul et al [11].

Figure 5 shows SPARK results for the experimental conditions studied in Rianguilaikul et al. [11] for different inlet air temperatures and humidity ratios. The increasing inlet air temperature and humidity leads to the rise of the outlet air temperature. In average, the change of 10 g/kg of inlet humidity can provide 6.3°C variation on the outlet temperature while 10°C change of inlet air temperature gives about 1.59°C difference. Therefore, the major significant factor affecting the outlet air condition is the inlet air humidity. These results are in agreement with those shown in [11] (not shown here). For high humidity ratios error is about 4% because of the lack of knowledge of water conditions in the experiment in this case.

Our results suggest that the model is in agreement with published data and therefore it can be used to evaluate and optimize system efficiency.

5 Parametric analysis

In order to design a system with high efficiency a parametric analysis is run to assess the effect of channel length, wet wall inlet temperature, and working air ratio (airflow mass ratio in the wet channel to that of dry channel).

For this case, we considered air inlet conditions of 35°C and 40% relative humidity which can be found in hot days in south France and north Algeria. The reference case is the same as studied by Rianguilaikul and shown in table 2. Water inlet temperature at the wet wall surface is of 21°C. Cooler efficiency is assessed in terms of its dew point cooling effectiveness and wet bulb cooling effectiveness defined as:

$$\epsilon_{dp} = \frac{T_{in} - T_{out}}{T_{in} - T_{dp}} \quad (6)$$

$$\epsilon_{wb} = \frac{T_{in} - T_{out}}{T_{in} - T_{wb}} \quad (7)$$

Figure 6 shows dew point and wet bulb effectiveness and supply air temperature versus channel length. As channel length increases both effectiveness increase and supply air temperature decreases from 25.72°C to 22.43°C. It is noticed that ϵ_{wb} increases faster than ϵ_{dp} . When channel length varies from 0.6 to 1.2 m dew point effectiveness increases from 59.45 to 80.6% whereas wet bulb effectiveness increases from 82.9 to 112.4% especially for the first 40 cm channel increment. A length higher than 1m can provide the wet bulb effectiveness greater than 100%. However, this increment is accompanied by an increase in fan power consumption to overcome additional resistance due to channel.

Figure 7 shows dew point and wet bulb effectiveness and supply air temperature versus water temperature at the inlet of the wet wall surface. When water temperature varies from 15°C to 29°C, air supply temperature increases from 22.75°C to 23.6°C which is an increment of 0.85°C. Dew point effectiveness ϵ_{dp} decreases from 78.5 to 73.2% and wet bulb effectiveness ϵ_{wb} varies from 109.5 to 102%. These results suggest that water temperature effect is small.

Figure 8 shows dew point and wet bulb effectiveness and supply air temperature versus working air ratio. Higher working to intake air ratio enhances water evaporation and improves cooler effectiveness however it decreases supplied airflow rate and thus can reduce the system cooling capacity. When working air ratio increases from 0.2 to 0.8, air supply temperature decreases from 25.16°C to 21.2°C. For a ratio higher then 0.3, ϵ_{wb} is higher than 100%. For values higher than 0.6, both dew point and wet bulb effectiveness grow slowly meaning that the best suiting values for this ratio are between 0.3 and 0.6.

These results show that for the cooler to be effective, its channel length should be higher than 1 m and its working air ratio higher than 0.3.

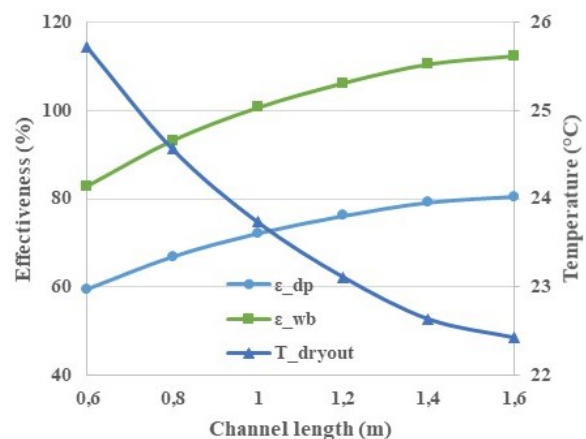


Fig. 6. Cooling effectiveness and supply air temperature versus channel length.

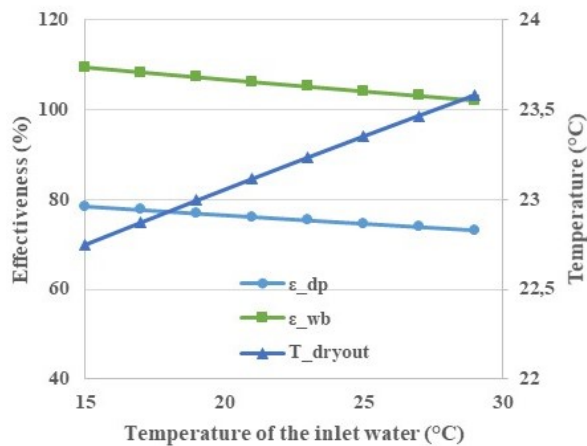


Fig. 7. Cooling effectiveness and supply air temperature as a function of water temperature.

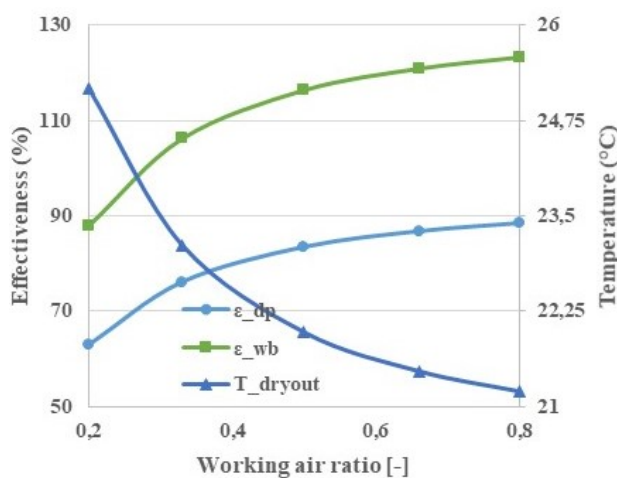


Fig. 8. Cooling effectiveness and supply air temperature as a function of working air ratio.

6 Conclusions

From the analysis presented in this paper, it can be deduced that dew point evaporative cooling is able to supply air at temperatures lower than the ambient wet bulb temperature. Finite difference model of the system has been developed based on heat and mass transfer processes. The system performance, namely, outlet air condition and cooling effectiveness are predicted from the model with the known inlet parameters. The simulation results are compared with the experimental findings from literature and are in a good agreement. Simulations are also used to design a cooler for Mediterranean climatic conditions. It is found that a length of 1 m and a working intake air ratio of at least 0.3 could achieve high wet bulb effectiveness. More investigations will be done in order to investigate effect of channel width, water flow rate and pressure drop within the channel. These results will be used to set an experiment in order to show and prove system feasibility in Algeria and France.

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References

1. El Habib El Andaloussi, Rafik Missaoui, Adel Mourtada, Stéphane Pouffary et Ariane Rozo, *Energie, changement climatique et bâtiment en Méditerranée : perspectives régionales ; plan bleu*, Sophia Antipolis (2011)
2. Enerdata, Alcor. *Tendances de l'efficacité énergétique dans les pays du bassin méditerranéen*, Rapport régional élaboré par le réseau MEDENER (2013)
3. R. Boukhanouf, H. G. Ibrahim, A. Alharbi, and M. Kanzari, "Investigation of an Evaporative Cooler for Buildings in Hot and Dry Climates", *Journal of Clean Energy Technologies*, Vol. **2**, No. 3, pp. 221-225, (2014)
4. R. Boukhanouf, A. Alharbi, O. Amer, and H. G. Ibrahim, "Experimental and Numerical Study of a Heat Pipe Based Indirect Porous Ceramic Evaporative Cooler", *International Journal of Environmental Science and Development*, Vol. **6**, No. 2, pp.104-110 (2015)
5. N. Lechner, *Heating, Cooling, Lighting: Sustainable Design Methods for Architects*, 3rd ed. New Jersey, U.S.A. : Wiley (2009)
6. J.R. Camargo, C.D. Ebinumca, J.L. Silveira "Thermoeconomic analysis of an evaporative desiccant air conditioning system", *Applied Thermal Engineering*, **23**, p.1537-1549 (2003)
7. R. Belarbi "Développement d'outils méthodologiques d'évaluation et d'intégration des systèmes évaporatifs pour le rafraîchissement passif des bâtiments", PhD, University of La Rochelle, France (1998)
8. C. Maalouf, *Etude du potentiel de rafraîchissement d'un système évaporatif à désorption avec régénération solaire*, PhD, University of La Rochelle, France (2006)
9. V. Maisotsenko, L. E. Gillan, T. L. Heaton, and A. D. Gillan, "Method and plate apparatus for dew point evaporative cooler," U.S. Patent US6581402 B2, June 24 (2003)
10. A. Hasan " Indirect evaporative cooling of air to a sub-wet bulb temperature," *Appl. Therm. Eng.*, **30**, pp. 2460-2468 (2010)
11. B. Riangvilaikul, S. Kumar "An experimental study of a novel dew point evaporative cooling system," *Energy Build.*, **42**, pp. 637-644 (2010)
12. J. Lee, D.Y. Lee "Experimental study of a counter flow regenerative evaporative cooler with finned channels," *Int. J. Heat Mass Transf.*, **65**, pp. 173-179 (2013)
13. R. Boukhanouf, A. Alharbi, H. G. Ibrahim, O. Amer, M. Worall, "Computer modelling and experimental

- investigation of building integrated sub-wet bulb temperature evaporative cooling system” *Appl. Th. Eng.*, **115**, 25 pp. 201-211 (2017)
14. E. F. Sowell and P. Haves, “Efficient solution strategies for building energy system simulation,” *Build. Phys.*, **33**, pp. 309–317,(2001)
 15. C. Maalouf, E. Wurtz, L. Mora, “Effect of Free Cooling on the Operation of a Desiccant Evaporative Cooling System”, *International Journal of Ventilation*, **7**, pp. 125-138, (2008)
 16. P. Xu, X. Ma, X. Zhao; *A Dew Point Air Cooler toward Super Performance From conception, simulation, fabrication to laboratory testing and applied evaluation*, Lambert Academic Publishing, pp. 332 (2018)