DOI: 10.21134/1gvs1b05

CYTEF XII Congreso Ibérico y X Congreso Iberoamericano de Ciencias y Técnicas del Frio
Universidad Miguel Hernández de Elche
26-06 28-06/2024
UNIVERSITAS Miguel Hernández

ANALYTICAL MODELLING OF AN INDIRECT EVAPORATIVE COOLER BASED ON THE MAISOTSENKO CYCLE FOR HVAC APPLICATIONS

Javier Ruiz Ramírez[1]*, Clara Gascó Arranz[1], Michael Opolot[2], Sangkyoung Lee[3] y Chunrong Zhao[4]

1: Engineering Research Institute of Elche, Miguel Hernández University.
Avda. de la Universidad, s/n, 03202 Elche, España.
e-mail: j.ruiz@umh.es
2: Centre for Hydrogen & Renewable Energy, Central Queensland University, Gladstone, Australia.
3: School of Mechanical and Mining Engineering, The University of Queensland, Brisbane, Australia.
4: School of Aerospace, Mechanical and Mechatronic Engineering, The University of Sydney, Australia.

Abstract: *Highly-efficient evaporative cooling systems can significantly reduce the energy use in buildings, thereby contributing to meet Europe's climate commitments, including building decarbonisation by 2050. The Maisotsenko Cycle is an indirect, multi-stage type of evaporative cooling which can be used in different air-conditioning applications. The main objective of this study is to develop an analytical model of an indirect evaporative cooler based on the Maisotsenko cycle for HVAC applications. The conservation equations governing the heat and mass transfer processes occurring in the indirect evaporative cooler were solved by means of numerical methods, and the results provided by the model were validated using experimental data reported in the literature (differences lower than 2%). Finally, a parametric study was conducted where the influence of various operational and environmental variables on the system's performance were numerically analysed.*

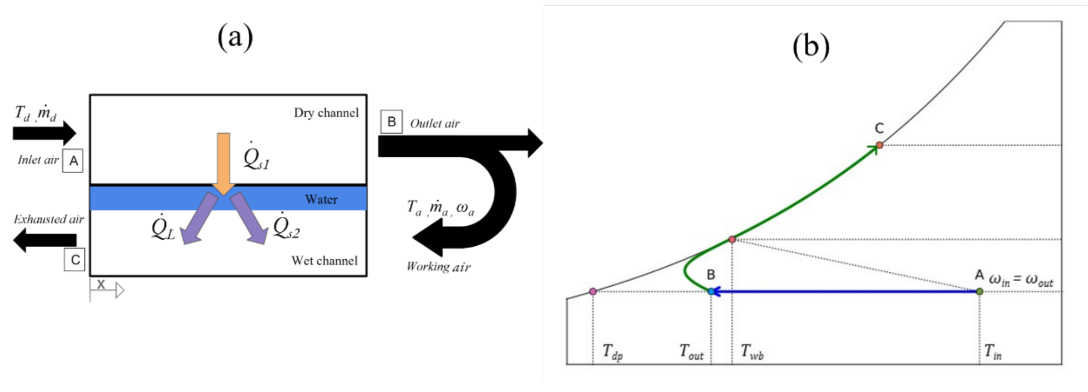


Figure 1. Schematic diagram of indirect evaporative cooler based on M-cycle: (a) configuration; (b) psychrometric evolution.

Keywords: Air conditioning, evaporative cooling, Maisotsenko.

1. INTRODUCTION

According to the International Energy Agency (IEA), emissions from the buildings sector have been steadily increasing, averaging a 1% rise per year since 2015. Alarmingly, this sector alone accounts for over one-third of the total global emissions This increase largely stems from the energy demands of maintaining comfortable indoor conditions, which are expected to grow due to rising global temperatures (due to the effects of climate change) and rapid economic development and urbanisation in the world's hottest regions. Traditional mechanical vapour-compression air conditioners, primarily powered by fossil fuels, are significant contributors to CO_2 emissions. In this context, exploring alternative cooling technologies to mitigate the environmental impact of the space cooling becomes crucial.

Technologies based on evaporative air-cooling offer promising solutions. Within this category, the indirect evaporative cooler based on the Maisotsenko cycle (M-cycle onwards) distinguishes itself by its capability to cool air to the dew-point temperature rather than the wet-bulb temperature achieved by direct evaporative cooling, without adding moisture to the air stream. Additionally, the M-cycle has low energy requirements for its operation. This cycle is characterised as an indirect, multi-stage type of evaporative cooling system, leveraging the psychrometric renewable energy available from the latent heat of water evaporating into the air. In the M-cycle, ambient air is drawn into a dry channel, where it releases sensible heat to a wet channel. It proves advantageous to redirect a portion of this air to serve as the working air in the wet channel. This working air is then humidified and absorbs heat from the dry channel before being exhausted into the atmosphere. Consequently, the outlet air temperature can be lowered below the ambient wet bulb temperature without altering humidity levels. The process described is illustrated in Figure 1 (a) and represented on a psychrometric chart in Figure 1 (b).

Numerous studies have investigated the potential of indirect dew-point evaporative coolers for space cooling applications. Beginning with the experimental studies, Duan et al. [1] investigated the operational performance and impact factors of a counter-flow regenerative evaporative cooler achieving a 31% increase in wet-bulb effectiveness and 40% growth in EER compared to conventional indirect evaporative cooler. B. Riangvilaikul and Kumar [2] built a novel counter flow configuration of a dew point evaporative cooling system to investigate the outlet air conditions and the system effectiveness at different inlet air conditions. Bruno [3] tested a counter-flow flat-plate dew point cooler for commercial and residential applications under a wide range of ambient conditions concluding the potential electrical energy savings of this technology. Jradi and Riffat [4] carried out an experimental and numerical investigation of a dew-point cooler with cross-flow heat and mass exchanger with the aim of studying the effect of different operational parameters and optimising the cooling system performance to achieve the indicated thermal comfort levels in buildings. Numerical studies have been conducted as well. For instance, Pandelidis et al. [5] developed a two-dimensional M-Cycle cross-flow heat exchanger model, providing insights into their performance under various operational conditions. Similarly, Boukhanouf et al. [6] presented a computer and experimental model of a sub-wet bulb temperature indirect evaporative cooling system for space cooling in buildings, achieving a cooling capacity of 225 W/m^2. Caliskan et al. [7] conducted an energetic, exergetic and sustainability analysis of the M-cycle compared with three conventional types of air cooling systems for building applications, obtaining better results for the M-cycle. Investigating the design of the evaporative cooler, Kabeel et al. [8] modified the heat exchanger adding internal baffle, finding significant improvements in supply air temperature reduction (21%) and coefficient of performance (71%) compared to previous designs. Examining the feasibility of dew-point evaporative coolers for air conditioning in specific ambient conditions, Zhao et al.[9] found this technology suitable for most of UK and Chinese regions, particularly those with hot, dry climates, while adaptation measures are needed for humid regions. Similarly, Jaber and Ajib [10] studied its feasibility in Mediterranean regions with two different configurations, concluding of its high potential and high energy and economic savings.

The main objective of this study is to analyse the influence of geometric, operational, and environmental variables on the performance of an indirect evaporative cooler based on the M-cycle with the aim of optimising its operation in HVAC applications. For this purpose, a mathematical model of the system was developed and validated against experimental data reported in the literature (differences less than 2%), followed by a parametric study.

2. METHOLODOGY

This section summarises the M-cycle modelling along with the criteria used to evaluate its performance. To carry out, the investigation a detailed mathematical model of the M-cycle has been developed. To simplify the analysis, the following assumptions were made:

- There are no heat losses to the surroundings.
- The thermal resistance of the wall and temperature difference of wall surfaces between the dry and wet side are neglected because of the thickness of the plate.
- Air is considered an incompressible fluid. Hence, all its properties are uniform within the incremental control volume.
- The secondary air stream is assumed to be equally saturated with the water film.
- The Lewis number is equal to 1 (Le=1).

Figure 1 (a) shows a schematic description of the M-cycle model. The equations that describe the thermophysical processes occurring are defined for a differential control volume through the following equations. Along the dry channel, only sensible heat transfer occurs from the primary air stream to the water film on the wet channel. Along the wet channel, both sensible and latent heat transfers occur between the water film and the air stream, along with mass transfer due to water evaporating into the air stream.

$$\delta\dot{Q}_{s1} = h_{cd} a (T_d - T_w) dx$$

$$\delta\dot{Q}_{s2} = h_c a (T_w - T_a) dx$$

$$\delta\dot{Q}_L = h_{Lv} \cdot d\dot{m}_w = -h_{Lv} d\dot{m}_v$$

$$d\dot{m}_w = -h_m \rho_w a (\omega_s - \omega_a) dx \quad (1)$$

Where the mass transfer coefficient between the secondary air stream and the water film is represented by a function of the Lewis number and the convective heat transfer coefficient, as follows:

$$h_m \rho_v = \frac{h_c}{c_{ph} \cdot \mathrm{Le}} \quad (2)$$

Overall, by applying energy and mass balances to the different control volumes (dry channel, water film, and secondary air stream), a system of five differential nonlinear equations is derived. Equation (3) is the energy balance on the dry channel, where the difference on the air's enthalpy is due to the loss of sensible heat. Equation (4) represents the energy balance on the water film, detailing how water enthalpy changes due to sensible heat gain from the primary air stream and heat exchange (sensible and latent) with the secondary channel's air stream. Equation (5) states the energy balance for the secondary air stream, driven by sensible heat transfer with the water film. Equation (6) corresponds to the mass balance for the secondary air stream, illustrating changes in humidity ratio due to water evaporation. Equation (7) is the mass balance of the water film, equating water loss with water vapor gain in the air stream.

$$\dot{m}_d \cdot c_{ph} \cdot dT_d = -\delta\dot{Q}_{s1} \quad (3)$$

$$\dot{m}_w c_{pw} dT_w = -\delta\dot{Q}_{s2} + \delta\dot{Q}_{s1} - \delta\dot{Q}_L \quad (4)$$

$$\dot{m}_a c_{ph} \frac{\partial T_a}{\partial x} = \delta\dot{Q}_{s2} \quad (5)$$

$$d\omega_a = \frac{d\dot{m}_v}{\dot{m}_a} = \frac{h_m \rho_v a}{\dot{m}_a} (\omega_s - \omega_a) dx \quad (6)$$

$$d\dot{m}_w = -\dot{m}_a d\omega_a \quad (7)$$

The system is solved by means of numerical methods. Consequently, the developed model provides predictions for the temperature evolution of the primary and secondary air streams and the water, along with the humidity variation of the secondary air stream and the water mass flow along the secondary channel. Additionally, to drive the system there is a need to use a fan to overcome the total pressure drop of the heat exchanger. The pressure drop of the airflow is due to the frictional losses, which can be estimated with the Darcy-Weisbach equation. The total power consumption of the fan is calculated through equation (8). The cooling capacity represents the sensible energy delivered by the cooler, which is estimated through equation (9).

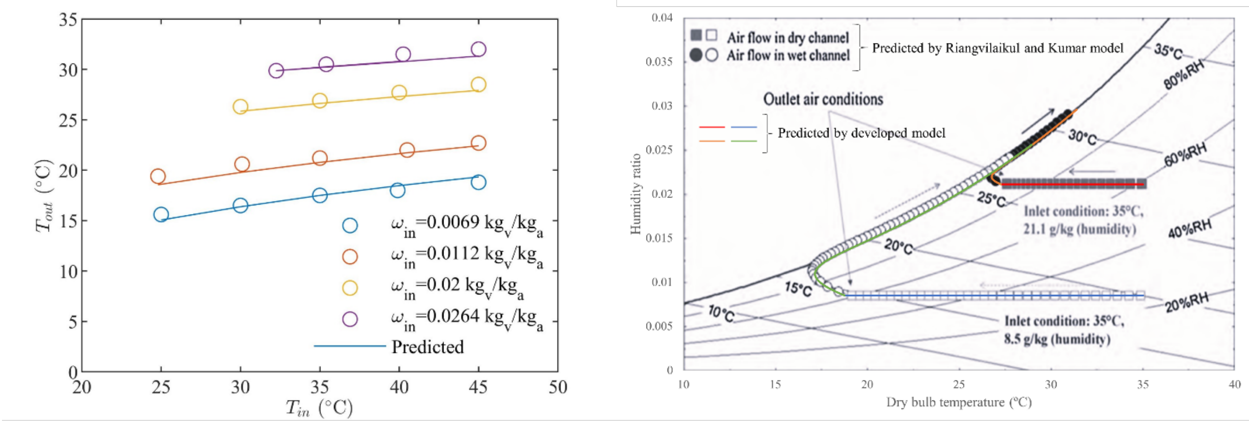
$$\dot{W}_f = \frac{\Delta P \cdot Q}{\eta} \quad (8)$$

$$\dot{Q}_{cool} = c_{pa} \cdot \dot{m}_d \cdot (1 - r) \cdot (T_{in} - T_{out}) \quad (9)$$

3. RESULTS AND DISCUSSION

3.1. Model validation

Figure 2 displays the validation of the M-cycle model developed. In particular, Figure 2 (a) contains the experimental data reported by Riangvilaikul and Kumar [2]. The authors conducted 16 experiments, where the inlet air temperature (T_{in}) was varied in 5 levels (ranging from 25 to 45 in 5ºC intervals) and the inlet air absolute humidity (ω_{in}) was modified in 4 levels (0.0069, 0.0112, 0.02 and 0.0264 kg_v/kg_a). The results provided by the model show differences of less than 1.5% compared to those published in the literature in terms of the outlet air temperature for the 16 cases compared. These two authors also developed an analytical model of the system which was validated experimentally [11]. Figure 2 (b) depicts the psychrometric evolution in the dry and wet channels for two different inlet air conditions predicted by Riangvilaikul and Kumar model. It can be observed that the results from the developed model closely align with those reported in the literature.



(a) (b)

Figure 2: Comparison between experimental and predicted results M-cycle model

3.2. Parametric study

Once the model was validated, a parametric study was conducted to assess the performance of the dew-point evaporative cooler under various conditions. Simulations were carried out by varying the inlet air conditions (temperature and humidity), geometric parameters (channel length and width), and operational parameters (air inlet velocity and working to intake air mass flow ratio). Table 1 summarises the different variables and levels considered for the parametric study, resulting in a total of 2025 simulated cases. The channel gap remained constant at 5 mm, the water feeding temperature was set equal to the intake air temperature, and the inlet water mass flow rate was maintained at 60 g/h. Additionally, the fan efficiency was estimated to be 75%.

The parametric study focused on evaluating the Evaporative Energy Efficiency Ratio ($\mathrm{EER}_{\mathrm{evap}}$), which measures the ratio of cooling capacity ($\dot{Q}_{cool}$) to electrical power ($\dot{W}_f$). Thus, it was considered the key parameter for optimising the dew-point evaporative cooler, and it is expressed as follows:

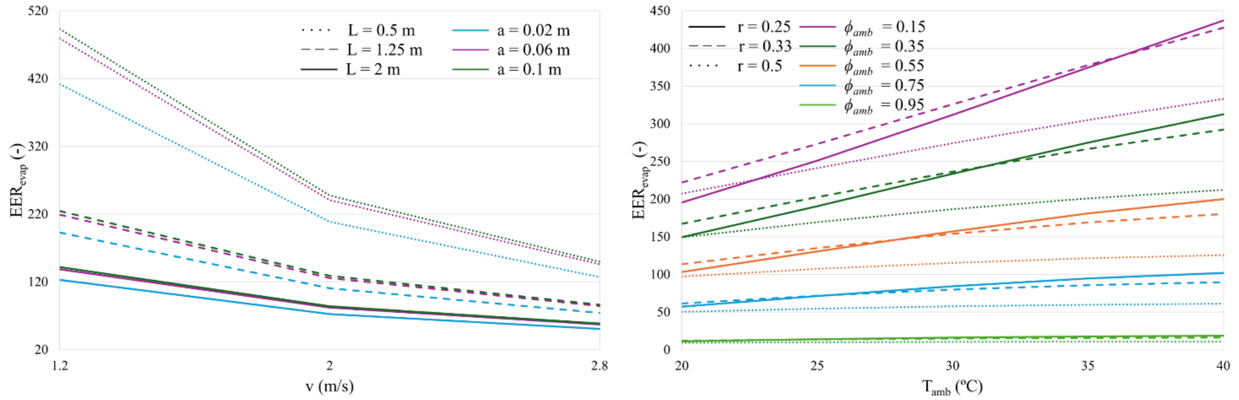
$$EER_{evap} = \frac{\dot{Q}_{cool}}{\dot{W}_f} \tag{10}$$

Figure 3 (a) depicts the $\mathrm{EER}_{\mathrm{evap}}$ relative to channel length, width, and air velocity at an inlet air temperature of 40°C and 55% humidity, with a working air to intake mass flow ratio of 0.5. Meanwhile Figure 3 (b) displays the $\mathrm{EER}_{\mathrm{evap}}$ against inlet air temperature and relative humidity and the working air to intake mass flow ratio for fixed dimensions of 1.25 m channel length, 0.06 m width, and 1.2 m/s velocity. In both cases, the same trends are observed when changing the fixed variables to another values. It is worth mentioning that with favourable conditions high $\mathrm{EER}_{\mathrm{evap}}$ values are obtained (typically in range 400-500, and up to 1000) because of the inherently low energy consumption of the system. This is consistent with previously reported data, [12]. Upon examining the geometric parameters, the $\mathrm{EER}_{\mathrm{evap}}$ increases with the channel width. This can be explained by the increase in cooling capacity while the fan consumption remains nearly unaffected. A larger channel width results in a greater heat transfer surface area, leading to a greater air temperature drop and enhanced cooling capacity. Moreover, for a constant velocity, increasing the channel width reduces pressure loss, albeit with a corresponding increase in air mass flow rate, thereby minimally affecting the energy required to drive the fan. It is worth noting that this effect is more pronounced when transitioning from a channel width of 0.02 m to 0.06 m compared to the transition from 0.06 m to 0.1 m. Conversely, increasing the channel length leads to a decrease in EER_{evap}. Despite increasing the cooling capacity (due to a larger heat transfer surface), the power consumption surpasses this effect because of a significant increase in pressure drop.

Table 1: Variables and levels considered in the parametric study.

T_{amb} (ºC)	ϕ_{amb} (-)	r (-)	L (m)	a (m)	v (m/s)
20	0.15	0.25	0.5	0.02	1.2
25	0.35	0.33	1.25	0.06	2
30	0.55	0.5	2	0.1	2.8
35	0.75	-	-	-	-
40	0.95	-	-	-	-

Regarding the operational parameters, an increase in velocity results in a decrease in EER_{evap}. This trend can be attributed to the lower temperature difference resulting from reduced time for heat transfer. Despite increasing the air mass flow, the previous effect outclasses it, consequently the cooling capacity decreases. Additionally, the power consumption increases due to a rise in pressure drop. For the working air to intake mass flow ratio, it has not been possible to deduce a clear trend because there are opposite effects when varying this parameter. By increasing it, the temperature drop increases, but the outlet mass flow decreases, leading to varying effects on cooling capacity depending on ambient conditions. The energy required to drive the fan is higher when increasing the ratio due to a higher pressure drop.
As for the air inlet conditions, larger wet-bulb depression leads to a higher temperature drop and increased cooling capacity. Thereby, an increase in temperature or a decrease in humidity translate to a better EER_{evap}.



(a) (b)

Figure 3: Influence of studied variables on EER_{evap}: (a) channel length, channel width and inlet air velocity; (b) inlet air dry-bulb temperature, inlet air relative humidity and working air to intake mass flow ratio.

4. CONCLUSIONS

This study has developed and validated a mathematical model of a indirect evaporative cooler based on the M-cycle, aiming to optimise its performance through a detailed parametric study. Our analysis of 2025 simulated cases, considering various inlet air conditions, geometric and operational parameters, yielded these insights:

- Ambient conditions significantly affect the cooler's performance, demonstrating particular effectiveness for low relative humidity and high temperature values, given the greater wet-bulb depression. These findings suggest its potential for HVAC applications in dry and hot regions.
- While increasing the channel length enhances the cooling capacity, it also increases power consumption, decreasing the EER_{evap}. However, the system's low inherent energy use (less than 0.16 W in tested scenarios) means that slight increases in power demand remain manageable and may be beneficial for enhanced cooling.
- Operating the system at lower air velocities substantially enhances performance, with the best results observed at lower speeds (a 70% improvement at 1.2 m/s compared to 2.8 m/s).

- The EER_{evap} improves with increased channel width, but this trend stabilizes for widths beyond 0.06 m. As further increases yield minimal gains (less than 2.5% improvement from 0.06 m to 0.1 m), a width of 0.06 m is deemed optimal for balancing performance and spatial efficiency.

REFERENCES

[1] Duan Z. Zhan, Zhao, Dong. Experimental study of a counter-flow regenerative evaporative cooler. In: Building and Environment 104, 2016, 47-58: ISSN 0360-1323.

[2] Riangvilaikul B. and Kumar S. An experimental study of a novel dew point evaporative cooling system. In: Energy and Buildings 42.5, 2010, 637-644: 0378-7788

[3] Bruno F. On-site experimental testing of a novel dew point evaporative cooler. In: Energy and Buildings 43.12, 2011, 3475-3483: 0378-7788

[4] Jradi M and Riffat S. Experimental and numerical investigation of a dew-point cooling system for thermal comfort in buildings. In: Applied Energy 132, 2014, 524-535: 0306-2619

[5] Pandelidis D, Anisimov S, Worek MW. Performance study of the Maisotsenko Cycle heat exchangers in different air-conditioning applications. In: International Journal of Heat and Mass Transfer 81, 2015, 207-221:0017-9310.

[6] Boukhanouf R, Alharbi A, Hatem G et al. Computer modelling and experimental investigation of building integrated sub-wet bulb temperature evaporative cooling system. In: Applied Thermal Engineering 115, 2007, 201-211: 1359-4311.

[7] Caliskan H, Dincer I, Hepbasli A. Exergetic and sustainability performance comparison of novel and conventional air cooling systems for building applications. In: Energy and Buildings 43.6, 2011, 1461-1472: 0378-7788.

[8] Kabeel A, Abdelgaied M, Sathyamurthy R, Arunkumar T. Performance improvement of a hybrid air conditioning system using the indirect evaporative cooler with internal baffles as a pre-cooling unit. In: Alexandria Engineering Journal 56.4, 2017, 395-403: 1110-0168.

[9] Zhao X, Yang S, Duan Z, Riffat SB. Feasibility study of a novel dew point air conditioning system for China building application. In: Building and Environment 44.9, 2009, 1990-1999: 0360-1323.

[10] Jaber S, Ajib S. Evaporative cooling as an efficient system in Mediterranean region. In: Applied Thermal Engineering 31.14, 2011, 2590-2596: 1359-4311.

[11] Riangvilaikul B. and Kumar S. Numerical study of a novel dew point evaporative cooling system. In: Energy and Buildings 42.11, 2010, 2241-2250: 0378-7788.

[12] Zhan C, Duan Z, Zhao X, Smith S, Jin H, Riffat S. Comparative study of the performance of the M-cycle counter-flow and cross-flow heat exchangers for indirect evaporative cooling – Paving the path toward sustainable cooling of buildings. In: Energy 36.12, 2011, 6790-6805: 0360-5442

NOMENCLATURE

a	Channel width, m
c_p	Specific heat, J kg^{-1} K^{-1}
ΔP	Pressure drop, Pa
h_c	Convective heat transfer coefficient, W m^{-2} K^{-1}
h_m	Mass transfer coefficient, m s^{-1}
L	Channel length, m
Le	Lewis number
$\dot{m}$	Mass flow rate, kg s^{-1}
Abbreviations	
EER_{evap}	Evaporative Energy Efficiency Ratio
M-cycle	Maisotsenko cycle
Greek symbols	
ρ	Density, kg m^{-3}
ϕ	Relative humidity
ω	Humidity ratio
ω_s	Saturation humidity ratio
Q	Volumetric flow rate, m^3 s^{-1}
$\dot{Q}_L$	Latent heat transfer rate, W
$\dot{Q}_s$	Sensible heat transfer rate, W
r	Working to intake air mass flow ratio
T	Temperature, ºC
v	Velocity, m s^{-1}
$\dot{W}_f$	Fan power consumption, W
x	Distance of the differential element, m
Superscripts and Subscripts	
a	Wet channel air stream
d	Dry channel air stream
h	Humid air
in	Inlet
out	Outlet
v	Water vapour
w	Water