

Numerical analysis of isothermal membrane-based dehumidification for M-cycle evaporative cooler

陳, 昊楠
九州大学大学院総合理工学府環境エネルギー工学専攻

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Numerical Analysis of Isothermal Membrane-based Dehumidification for M-cycle Evaporative Cooler

Chen Haonan

Supervisor Kyaw Thu

Takahiko Miyazaki

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NUMERICAL ANALYSIS OF ISOTHERMAL MEMBRANE-BASED DEHUMIDIFICATION FOR M-CYCLE EVAPORATIVE COOLER

Chen Haonan

Energy and Environmental Engineering
Graduate School of Engineering Sciences
Kyushu University

Abstract

In this work, M-cycle evaporative cooler (MEC) integrated with isothermal membrane-based dehumidification (IMBD) has been proposed and investigated. The whole system only uses air and water as working flow, instead of any organic refrigerants, and is expected to operate in high humidity area. A thermodynamic model for the system has been established and numerical analysis for different inlet air conditions and operational system parameters has been carried out. Relative humidity significantly affects the supply air temperature and COP. And compare the process of dehumidification to evaporative cooling, removing latent heat account for the most of cooling capacity. Low inlet air velocity and appropriate long IMBD chamber length make supply air temperature cooler. However, for inlet air velocity, the cooling capacity declines because of the small air flux, and a large-scale membrane is limited to manufacturing currently. Finally, the system was evaluated under several summer climate conditions of representative cities around the world. Drawing from the result, the system had a promising application prospect.

1 Introduction

1.1 Energy consumption in the cooling system

Electricity consumed by heating, ventilation, and air conditioning (HVAC) systems accounts for approximately 25% of the total building load in OEDC countries. [1] Especially in summer, the usage of air conditioners for cooling caused a large burden to local electricity power load. The electric energy consumption of room air conditioning accounts for 40.8% of the electricity consumption for the whole residence. And the overall energy consumption for space cooling reached 897.6 TWh in 2019, which accounted for 18% of the total electricity consumption in buildings.[2] Most of air conditioners are based on vapor compression cycles which do not only rely on primary energy but also affect global warming [3].

Reducing the electricity consumption of the cooling system is significant to alleviate the electricity shortage. On the one hand, improving the features of the building envelope can considerably affect air conditioning system energy consumption.[4] for example, reducing heat leaks from indoors to the outside means the building requires less cooling capacity. On the other hand, how to raise the performance of the cooling system has been studied worldwide.

Conventional vapor-compression air conditioning system is mainly composed of 4 parts: compressor, condenser, throttle valve, and evaporator. In one cycle, refrigerant is compressed by the compressor first, becoming gas with high temperature and high pressure. Then, it flows into the condenser, does heat transfer with outdoor air, releases a part of the heat, and becomes a high-pressure liquid. Next,

the refrigerant flow through the throttle valve becomes low temperature and low-pressure gas. Finally, in the evaporator, the refrigerant absorbs heat from the ambient indoor air, cools down the indoor room, and evaporated into vapor.

The property of refrigerants is one of the main factors that affect compressor's efficiency [5]. However, most refrigerants are Hydrofluorocarbons (HFCs) and during the manufacturing process, a large number of greenhouse gases (GHG) were emitted to the earth. HFCs have accounted for 20% of the total GHG emissions in the past 20 years. China's HFC emissions were approximately 200 Mt CO₂, in 2014. [6] Because of the air pollution, many researchers focus on another environmentally friendly technology: evaporative cooling.

1.2 Evaporative cooling

Evaporative cooling is an alternative technology to conventional air conditioning systems. Evaporative cooling is based on water evaporation, which utilizes latent heat. Water absorbs ambient heat and evaporates into vapor. During the process, the heat is removed from the ambient air, making the air temperature cool down.

There are two main types of the evaporative cooling system, direct evaporative cooling (DEC) and indirect evaporative cooling (IEC).

DEC is the process of evaporating liquid water into the ambient air and cooling the air temperature. A typical direct evaporative cooler, in Fig.1.1 uses a fan to draw in outside air through a pad wetting media and circulates the cool air through the building.[7] The latent heat of the air is utilized to evaporate the water. Evaporation cools down the air while increasing its relative humidity. There is no heat added or removed from the air. Hence, it is an adiabatic process of constant enthalpy.[8]

The theoretically ultimate temperature for direct evaporative cooling is the wet-bulb temperature. However, it is hard to reach in a real system. Meanwhile, because of the evaporation, the air becomes humid when supplied to the living space.

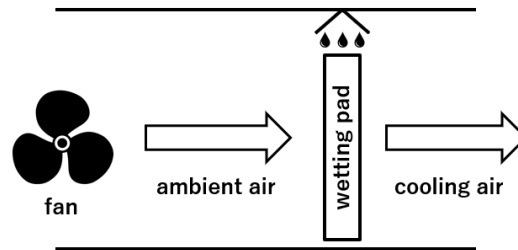


Fig. 1. 1 Schematic of direct evaporative cooling

IEC ordinarily contains two kinds of the channel, working channel and product channel. And air flux is called working stream and supply steam, respectively. Evaporation happens in the working channel so that the supply air is not in contact with water directly. As Fig. 1.2 shows, there is a heat exchange layer between two channels, so that the supply stream can be cooled down without increasing humidity ratio, which makes IEC more attractive than DEC. However, conventional IEC has a wet bulb efficiency of only between 55-75%.[9] Because the heat and mass transfer processes that take place in the working channel are limited to the wet-bulb temperature of the inlet air.

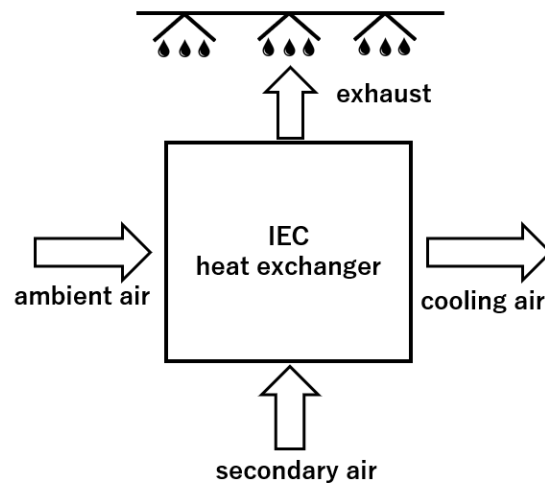


Fig. 1. 2 Schematic of indirect evaporative cooling

1.3 Maisotsenko cycle

Maisotsenko cycle (M-cycle) is an advanced IEC system, which combines thermodynamic processes of heat transfer and evaporative cooling and can theoretically cool down air to dew point temperature of the ambient air beyond its wet-bulb temperature. [10]

As shown in Fig. 1.3., an M-cycle evaporative cooler consists of dry channels and wet channels, namely 2-channel M-cycle. In IEC the channels are called product channel and working channel. Ambient air flows into the dry channel and a fraction of which flows into the wet channel where water evaporates thereby cooling the air in the adjacent dry channel. The cooled fraction of the air flows into indoor space as supply air.

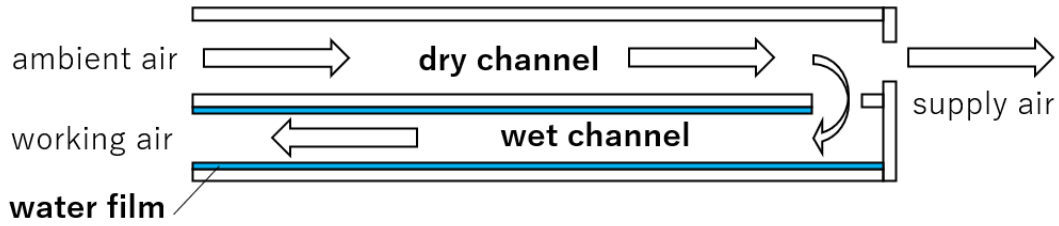


Fig. 1. 3 Schematic of 2-channel M-cycle

modified M-Cycle, named 3-channel M-cycle showed in Fig. 1.4., gives the freedom to recover the heat from any fluid/gas by using an additional dry channel.[11] The addition channel is called the product channel. In cooling and air conditioning applications, the ambient air flows into the dry channel as working air, then all flows back to the wet channel, where heat transfer and evaporation happened. While in the product channel, the ambient air flows through the channel, cool down by the adjacent wet channel, then as supply air flows into indoor space directly.

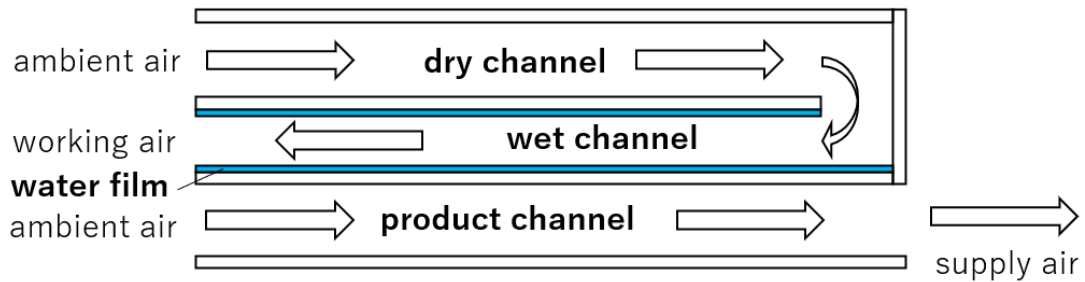


Fig. 1. 4 Schematic of 3-channel M-cycle

In comparison to conventional air conditioning units, M-cycle uses water as its refrigerant and consumes less energy. Compared to the typical IEC, M-cycle has better cooling performance, because the temperature difference between streams in M-cycle is higher than the typical IEC.[12] Thus, M-cycle is considered to have great environmental and economic benefits.

1.4 Dehumidification

In IEC systems, relative humidity is the driving force for evaporation hence it is necessary to remove moisture upstream.

Dehumidification can comprise a significant fraction of cooling loads in hot and humid environments. For example, 70–80% of the total energy consumed by a vapor-compression chiller comes from direct cooling at a relative humidity RH of 70–95%. [13]

There are several technologies for dehumidification, such as vapor condensation, solid adsorption, liquid absorption, and membrane dehumidification.[14] Dehumidification based on vapor compression involves reducing the temperature of the air to below the dew point temperature, resulting in both sensible and latent cooling. Solid adsorption and liquid absorption are collectively called sorbent dehumidification, which involves contacting moist air with a solid or liquid sorbent without sensible cooling to remove the moisture and minimizes the utilization of environmentally harmful refrigerants in cooling systems. This dehumidification is an adiabatic process but requires heat at temperatures typically between 50 °C and 260 °C to regenerate the sorbent or desiccant, with regeneration rates increasing at higher temperatures. Liquid desiccants typically utilize a lower regeneration temperature of 60 °C - 90 °C [15]

Membrane dehumidification is often used with liquid desiccant together and the membrane-based air dehumidifiers can effectively eliminate or overcome the disadvantages of liquid desiccant droplets being entrained into the air stream and the possibility of uneven flow distribution. However, it creates additional mass transfer resistance that reduces the performance of the dehumidifier.[16]

1.5 Isothermal membrane-based dehumidification

Isothermal membrane-based dehumidification (IMBD) is a kind of vacuum dehumidifier that can decrease the humidity of air without sensible cooling. Refer to Fig. 1.5., it contains 2 kinds of the chamber called feed chamber and permeate chamber. Two chambers are separated by a dense-type hydrophilic membrane with selective permeability, which allows water vapor to permeate easily while impeding dry air. In membrane-based separation, the partial pressure difference is the primary driving force for mass transfer which is induced by the ambient pressure in the feed chamber and vacuum pressure in the permeate chamber, thus allowing water vapor to permeate through the membrane continuously.

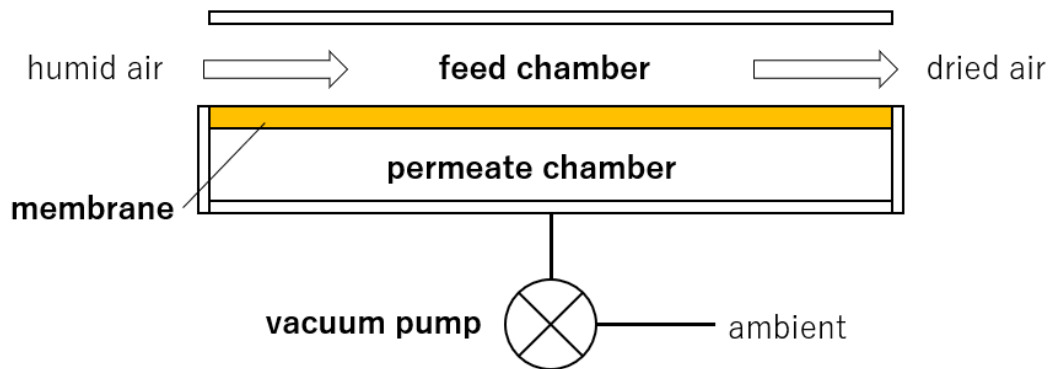


Fig. 1. 5 Schematic of isothermal membrane-based dehumidification

1.6 Literature review of M-cycle evaporative cooler

Zhan et al. [17] developed an experimentally validated model to investigate the performance of cross-flow and counter-flow M-cycle heat exchangers based on different flow patterns of secondary air, under the same conditions. The experimental and numerical results showed that compare to the

crossflow, the cooling capacity of the counter-flow M-cycle heat exchanger was increased less than 20%. The dew point efficiency and wet bulb efficiency were increased by 15-23%.

Moshari [18] combined his simulation data with experiment results also obtained that the wet-bulb effectiveness of counter-flow Regenerative Evaporative Coolers (REC) is 30% higher than crossflow REC. Meanwhile, Increasing the ratio of working air to total inlet air accounts for higher wet bulb effectiveness to counter-flow REC.

B. Riangvilaikul and S. Kumar [19] experimented with M-cycle and indicated that when the intake air temperature exceeds 30 °C, the intake air velocity should be kept below 2.5 m/s to obtain a wet-bulb efficiency greater than 100%. For hot and dry climate applications, this system could provide comfortable living conditions when the inlet temperature and humidity ratio are lower than 45 °C and 11.2 g/kg, respectively.

Zhao et al. [20] studied five main materials suitable for indirect evaporative coolers through experiments, namely metal, fiber, zeolite, ceramics, or carbon, and found that the thermal properties of the materials have little effect on the device, but the structure, shape and corrosion resistance of the materials had a greater impact. A wick attained metal, such as copper or aluminum was recommended.

Cui et al. [21] designed a numerical model of modified M-cycle, where working air and product air is completely separated. The cooler model was tested under different inlet air temperatures and humidity ratios. The simulation results showed that the wet-bulb efficiency could be up to 132% and dew point efficiency was obtained to 93%. To improve the cooling efficiency, the channel length should be at least 200 times the height of the working channel.

Lin et al. [22] integrated the regression model into a multi-objective optimization algorithm, to facilitate robust cooler design and control strategies for M-cycle. The result depicted that the

minimum value of channel length is 0.50 m, and the operating ratio is relatively stable at 0.40. The height of the channel and the air supply speed vary within a small range of 3.0-3.9 mm and 1.2-1.7 m/s, respectively. The COP and dew point efficiency of the cooler can be increased by 60% and 11%, at the maximum loss of cooling capacity is 18%, accordingly. In his experiment [23], He determined the air mixing process through a modified model with exergy analysis and proved that the exergy damage at the entrance of the dry channel is very serious due to air mixing.

Chu [24] made an M-cycle cooler that adopts polymer plant fiber composite material, which has high-efficiency hydrophilicity. The experimental results showed that under standard dry conditions, the wet-bulb efficiency reaches 105.6% and the dew point efficiency reaches 76%.

Liu et al [25] made the heat exchange surface into a corrugated plate to increase the heat exchange area between the dry and wet channels, which enabled the dew point evaporative cooler more efficient. Through this improvement, compared with the existing commercial dew point evaporative cooler of the same capacity, the wet-bulb effectiveness was increased by 29.3%, and the COP was increased by 34.6%. Also, reducing the channel width and channel gap within a reasonable range could improve the cooling efficiency.

Dizaji HS et al. [26] proposed a new model of a multi-stage M-cycle cooler based on a novel wet surface theory, in which the wet plate is continuously wet without water stream on the surface and by a quiet small sprayed water flow, the water evaporation does not make the wet surface dry. The change of the water film surface temperature and channel temperature and humidity distribution could be well predicted by the model.

Gao et al. [27] described a liquid desiccant cooling system through the M-cycle, which consisted of two air-handling processes: moisture removal in the dehumidifier and sensible heat removal in the M-cycle indirect evaporative cooler. He indicated that increasing the inlet air and the humidity ratio

improved the dehumidification performance, but the cooling capacity was declined. Increasing the inlet air temperature had a negative impact on the heat and mass transfer both in the dehumidifier and M-cycle evaporative cooler.

Lin et al. [28] adopted a hybrid membrane liquid desiccant dehumidification into M-cycle and concluded that under hot and humid ambient where the temperature was 30.0 °C and humidity was 20.0 g/kg, the product air at 18.3 °C temperature and 10.9 g/kg humidity was able to be delivered by the hybrid system.

1.7 Literature review of isothermal membrane-based dehumidification

IMBD is also known as Vacuum-membrane Based Dehumidifier (VBD). Bui et al. [29] proposed that the vacuum pump was a vital part of the VMD system. The continuous driving force for water vapor penetration was input by the vacuum pump. At the same time, it also pumps permeate stream from the membrane module to ambient pressure. It is essentially the only source of energy consumption in VMD. He also [30] conducted a series of experiments to study the proper hygroscopicity and performance of polymer composite membranes in vacuum membrane dehumidification. The result was that membrane layers comprising inorganic lithium chloride and polyvinyl alcohol (PVA/LiCl) had a stronger hygroscopic property, while trim ethylene glycol and polyvinyl alcohol (PVA/TEG) was more durable and suitable for dehumidification.

El-Dessouky et al. [31] recommended that the compression ratio is at least 80 (101 kPa/1.25 kPa). To minimize the size of the membrane module, the membrane required a limited driving force to cross through it. Therefore, the pressure was more likely to be close to 1 kPa, and the compression ratio is more likely to be 100.

Xing et al. [32] developed a thin zeolite /metal sheet membrane where water vapor permeance was obtained to 6.8×10^{-6} (mol·m⁻² Pa⁻¹ s⁻¹) and water vapor to air selectivity to 300. It was estimated that when the membrane selectivity is greater than 200, energy efficiency gains 50% or higher over conventional vapor compression systems.

Kudasheva et al. [33] made an ionic liquid membrane of 10 μm thickness with 1-ethyl-3-methylimidazolium ethyl sulfate, 1-ethyl-3-methylimidazolium dicyanamide, and 1-ethyl-3-methylimidazolium tetrafluoroborate. Compared to trim ethylene glycol, the ionic liquid membrane obtained Water recovery rates as high as 10 g/h from an 80%RH air with a flat membrane module. The permeabilities of water vapor were 26,000–46,000 Barrer through the ionic liquids.

Chun et al. [34] suggested a novel curve vacuum membrane-based dehumidification device by the analysis of phase equilibrium effect and created an inversion method to predict the apparent water vapor permeability. Due to the phase equilibrium, the percentage of water vapor removed increased from 4.65% to 8.14%. The apparent water vapor permeability of the inversion method was higher than the water vapor permeability measured by the cup method, and the growth rate exceeds 60%. In addition, the curved vacuum membrane dehumidification device could effectively improve the dehumidification effect. When the radius of curvature was reduced from 30mm to 15mm, the dehumidification efficiency of the convex and concave films increased by 1.96% and 1.74%, respectively.

Claridge et al. [35] proposed Claridge-Culp-Liu dehumidification, which combines a vacuum membrane and a dehumidification system based on sub-condensation. In this system, a condenser was installed behind the vacuum pump to reduce the suction load. Their results showed that the proposed system only consumes 26-56% of the energy required by the Carnot vapor compression system.

Lim et al. [36] studied several types of IMBD systems. A membrane separator with a single vacuum pump was the basic type. Additionally, the other three types, which were the sweep air mixed type, water vapor discharge type, and condenser combined type, were analyzed comparatively with an idealized membrane with infinite selectivity on water vapor. Then concluded that the water vapor discharge type, required two-membrane mass exchangers exhibited the highest dehumidification performance.

1.8 About this study

As mentioned in previous sections, the IEC system has a high cooling capacity when working in dry ambient conditions. Therefore, dehumidification upstream is necessary. Conventional dehumidification processes have some weaknesses. The vapor condensation cycle uses HFCs as a refrigerant and extra heat would be occupied during operation. Absorption or adsorption by liquid and solid desiccant needs a regeneration process and the sorbent creates additional mass transfer resistance to reduce the efficiency. It occupies extra heat release as well, which is not expected in the cooling system.

In this study, a thermodynamic model of isothermal membrane-based dehumidification and M-cycle evaporative cooler was developed. This system was an environmentally friendly device, which utilizes air and water only without any HFCs, to achieve indoor cooling. Because the dehumidification is under isothermal conditions, inlet air temperature maintains constant during the dehumidification process. System performance was evaluated by dehumidification effectiveness, Coefficient of Performance (COP), and supply air temperature.

MEC was mainly based on a 2-channel M-cycle model because, in the two-channel model, the

working air ratio can be analyzed. The 3-channel M-cycle model was also developed as a reference. 3-channel M-cycle has a better prospect in engineering, because of the individual product channel, where supply air could be handled freely. Working air ratio for 2-channel M-cycle was set as 0.5 and, in that way, 2-channel and 3-channel M-cycle models showed almost the same performance. The influence of inlet air temperature, relative humidity, velocity, channel length, and membrane properties were analyzed individually. Moreover, the system was investigated under several summer climate conditions of representative cities around the world. Drawing from the result, the system had a promising application prospect.

2 Methodology

2.1 Modeling for M-cycle evaporative cooler

The model was developed to describe a unit of counterflow M-cycle evaporative cooler, which is composed of 3 channels, dry channel, wet channel, and product channel.

Inlet air was divided into two parts, flow into the dry channel and product channel. The air in the dry channel is cooled down by convective heat transfer. There is no mass transfer in the dry channel, so the absolute humidity ratio remains unchanged.

Then, the air flowed back to the wet channel. Water vapor is evaporated from the water film, taking the heat from the dry channel away. Heat transfer and mass transfer happen simultaneously in the wet channel. Air is heated and becomes humid when flows along the wet channel. Finally, it becomes warm and saturated as exhaust, releasing to the outdoor ambient.

The air in the product channel is similar to air in the dry channel. It is cooled down without absolute humidity ratio change. The difference is that air in the product channel is supplied air, flowing to indoor ambient, rather than flowing back to dry channel.

The model is considered as one dimension calculation. Assumptions are as follows:

1. Steady state.
2. Pressure in each channel is constant.
3. Convection items are dominant; diffusion items can be neglected.
4. Air adjacent to the water film surface is saturated.
5. Heat transfer only happens between channels and the boundary is consists of adiabatic.

Basic air properties are T temperature, p pressure, and x absolute humidity ratio. Other humid air properties referred to in the thermodynamic calculation are imported by CoolProp 6.4.1. listed in Table.1.

Table. 1 Humid air properties export by CoolProp

Property	Input	Code example
specific heat of dry air [J/kgK]	T, p, x	<code>cpa = HAP('cp','T',T,'P',p,'W',x)</code>
specific heat of wet air [J/kgK]	T, p, x	<code>cpw = HAP('cp_ha','T',T,'P',p,'W',x)</code>
thermal conductivity of wet air [W/mK]	T, p, x	<code>k = HAP('K','T',T,'P',p,'W',x)</code>
density of wet air [Kg/m3]	T, p, x	<code>rho= 1/HAP('Vha','T',T,'P',p,'W',x)</code>
viscosity of wet air [Pa-s]	T, p, x	<code>vis = HAP('Visc','T',T,'W',x,'P',p)</code>
saturated vapor density [Kg/m3]	T, p, x	<code>rhos = PropsSI('D', 'T', T, 'Q', 1, 'Water')</code>
Absolute humidity ratio [/]	T, p, RH	<code>xda = HAP('W','T',T,'P',p,'RH',RH/100)</code>

2.1.1 Govern equations for 2-channel M-cycle evaporative cooler

The Govern equations for dry channel temperature, wet channel temperature, and product channel temperature are as follows.

$$\rho c_p \left(\frac{\partial T_{da}}{\partial t} + u_{da} \frac{\partial T_{da}}{\partial z} \right) \frac{d}{2} = k_{da} \frac{\partial^2 T_{da}}{\partial z^2} \frac{d}{2} + h_{da} (T_{ds} - T_{da})$$

$$\rho c_p \left(\frac{\partial T_{wa}}{\partial t} + u_{wa} \frac{\partial T_{wa}}{\partial z} \right) \frac{d}{2} = k_{wa} \frac{\partial^2 T_{wa}}{\partial z^2} \frac{d}{2} + h_{wa} (T_{wsd} - T_{wa})$$

Because the mass transfer is only happened in wet channel, the govern equations for mass transfer are as follows:

$$\left(\frac{\partial \rho_{da}}{\partial t} + u_{da} \frac{\partial \rho_{da}}{\partial z} \right) \frac{d}{2} = 0$$

$$\left(\frac{\partial \rho_{wa}}{\partial t} + u_{wa} \frac{\partial \rho_{wa}}{\partial z}\right) \frac{d}{2} = D_{wa} \frac{\partial^2 \rho_{wa}}{\partial z^2} \frac{d}{2} + h_{wa}(\rho_{wsd} - \rho_{wa})$$

where *da* denotes dry channel air, *ds* is dry channel surface, *wa* denotes wet channel air, *and* denotes wet channel surface adjacent to dry channel, ρ ($kg \cdot m^{-3}$) is density, cp ($J \cdot kg^{-1} \cdot K^{-1}$) is specific heat capacity, u ($m \cdot s^{-1}$) is the velocity, d (m) is channel height, D ($m^2 \cdot s^{-1}$) is diffusion coefficient. h ($W \cdot m^{-2} \cdot K^{-1}$) is the convective heat transfer coefficient, and hm ($m \cdot s^{-1}$) is the convective mass transfer coefficient.

While in steady state, and neglect diffusion items, the govern equations can be simplified as:

$$\rho c_p \left(u_{da} \frac{\partial T_{da}}{\partial z}\right) \frac{d}{2} = h_{da}(T_{ds} - T_{da})$$

$$\rho c_p \left(u_{wa} \frac{\partial T_{wa}}{\partial z}\right) \frac{d}{2} = h_{wa}(T_{wsd} - T_{wa})$$

$$\left(u_{da} \frac{\partial \rho_{da}}{\partial z}\right) \frac{d}{2} = 0$$

$$\left(u_{wa} \frac{\partial \rho_{wa}}{\partial z}\right) \frac{d}{2} = h_{wa}(\rho_{wsd} - \rho_{wa})$$

These parameter locations can refer to Fig. 2.1.

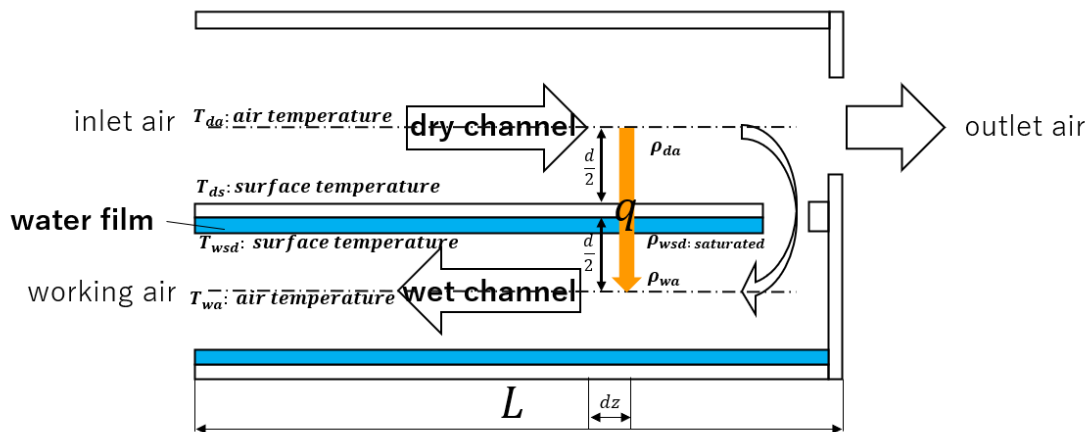


Fig. 2. 1 Calculation of 2-channel M-cycle evaporative cooler

Uniform heat flux q ($W \cdot m^{-2}$) is perpendicular to the channel. Equations are as follows,

Form dry channel to wet channel:

$$\begin{aligned} q &= h_{da}(T_{da} - T_{ds}) \\ q &= \frac{k(T_{ds} - T_{wsd})}{\delta} \\ q &= h_{wa}(T_{wsd} - T_{wa}) + L_w h_m(\rho_{wsd} - \rho_{wa}) \end{aligned}$$

where k ($W \cdot m^{-1} \cdot K^{-1}$) is heat conductivity, δ (m) is channel thickness.

Combine 3 equations, heat flux is expressed as:

$$q = \frac{T_{da} - T_{wa} + \frac{L_w h_m(\rho_{wsd} - \rho_{wa})}{h_{wa}}}{\left(\frac{1}{h_{da}} + \frac{\delta}{k} + \frac{1}{h_{wa}}\right)}$$

By transformation, surface temperature can be derived as

$$\begin{aligned} T_{ds} &= \left(1 - \frac{\frac{1}{h_{da}}}{\frac{1}{h_{da}} + \frac{\delta}{k} + \frac{1}{h_{wa}}}\right) T_{da} + \left(\frac{\frac{1}{h_{da}}}{\frac{1}{h_{da}} + \frac{\delta}{k} + \frac{1}{h_{wa}}}\right) \left\{T_{wa} - \frac{L_w h_m}{h_{wa}}(\rho_{wsd} - \rho_{wa})\right\} \\ T_{wsd} &= \left(\frac{\frac{1}{h_{wa}}}{\frac{1}{h_{da}} + \frac{\delta}{k} + \frac{1}{h_{wa}}}\right) T_{da} + \left(1 - \frac{\frac{1}{h_{wa}}}{\frac{1}{h_{da}} + \frac{\delta}{k} + \frac{1}{h_{wa}}}\right) \left\{T_{wa} - \frac{L_w h_m}{h_{wa}}(\rho_{wsd} - \rho_{wa})\right\} \end{aligned}$$

Define dimensionless number ϕ_d , ϕ_{wd} .

$$\begin{aligned} \phi_d &= \frac{\frac{1}{h_{da}}}{\frac{1}{h_{da}} + \frac{\delta}{k} + \frac{1}{h_{wa}}} \\ \phi_{wd} &= \frac{\frac{1}{h_{wa}}}{\frac{1}{h_{da}} + \frac{\delta}{k} + \frac{1}{h_{wa}}} \end{aligned}$$

Integrate these dimensionless numbers into surface temperature equation:

$$\begin{aligned} T_{ds} &= (1 - \phi_d)T_{da} + \phi_d \left\{T_{wa} - \frac{L_w h_m}{h_{wa}}(\rho_{wsd} - \rho_{wa})\right\} \\ T_{wsd} &= \phi_{wd}T_{da} + (1 - \phi_{wd}) \left\{T_{wa} - \frac{L_w h_m}{h_{wa}}(\rho_{wsd} - \rho_{wa})\right\} \end{aligned}$$

where L_w ($J \cdot kg^{-1}$) is latent heat

2.1.2 Equations for 3-channel M-cycle evaporative cooler

Similar to 2-channel, the governing equations for 3-channel M-cycle evaporative cooler are

$$\begin{aligned}\rho c_p \left(u_{da} \frac{\partial T_{da}}{\partial z} \right) \frac{d}{2} &= h_{da} (T_{ds} - T_{da}) \\ \rho c_p \left(u_{wa} \frac{\partial T_{wa}}{\partial z} \right) \frac{d}{2} &= h_{wa} (T_{wsd} + T_{wsp} - 2T_{wa}) \\ \rho c_p \left(u_{pa} \frac{\partial T_{pa}}{\partial z} \right) \frac{d}{2} &= h_{da} (T_{ps} - T_{pa}) \\ \left(u_{da} \frac{\partial \rho_{da}}{\partial z} \right) \frac{d}{2} &= 0 \\ \left(u_{wa} \frac{\partial \rho_{wa}}{\partial z} \right) \frac{d}{2} &= h_{wa} (\rho_{wsd} + \rho_{wsp} - 2\rho_{wa}) \\ \left(u_{pa} \frac{\partial \rho_{pa}}{\partial z} \right) \frac{d}{2} &= 0\end{aligned}$$

where pa denotes the product channel, wsp denotes wet channel surface adjacent to product channel.

Additional surface temperatures were calculated as

$$\begin{aligned}T_{wsp} &= \phi_{wp} T_{da} + (1 - \phi_{wp}) \left\{ T_{wa} - \frac{L_w h_m}{h_{wa}} (\rho_{wsp} - \rho_{wa}) \right\} \\ T_{ps} &= (1 - \phi_p) T_{da} + \phi_p \left\{ T_{wa} - \frac{L_w h_m}{h_{wa}} (\rho_{wsp} - \rho_{wa}) \right\}\end{aligned}$$

where dimensionless number ϕ_{wp} , ϕ_p are

$$\phi_{wp} = \frac{\frac{1}{h_{wa}}}{\frac{1}{h_{pa}} + \frac{\delta}{k} + \frac{1}{h_{wa}}}$$

$$\phi_p = \frac{\frac{1}{h_{pa}}}{\frac{1}{h_{pa}} + \frac{\delta}{k} + \frac{1}{h_{wa}}}$$

2.1.3 Equations for thermodynamic parameters

Nusselt number for the fully developed thermal region of flat rectangular sectional area is:

$$Nu = 8.235$$

Nusselt number for entrance region is defined as:

$$Nu = \frac{\frac{8.235}{\tanh\left(2.264Gz^{-\frac{1}{3}} + 1.7Gz^{-\frac{2}{3}}\right)} + 0.0499Gz \tanh(Gz^{-1})}{\tanh\left(2.432Pr^{\frac{1}{6}}Gz^{-\frac{1}{6}}\right)}$$

where Gz is Graetz number, related to Reynolds number, Prandtl Number, hydraulic diameter, and thermal entrance length.

$$Gz = \frac{D_h}{l_{en}} Re Pr$$

Hydraulic diameter D_h (m) for rectangular section, is calculated as:

$$D_h = \frac{2(d + w)}{dw}$$

where w is channel *width* (m).

For thermal entrance region, the entrance length (m) is determined as

$$l_{en} = 0.05 Re Pr D_h$$

Reynolds number is

$$Re = \frac{u \rho D_h}{\nu}$$

where ν ($Pa \cdot s$) is viscosity.

Prandtl Number is

$$Pr = \frac{c_p \nu}{k}$$

With Nusselt Number, convective heat transfer coefficient h can be calculated as:

$$h = \frac{Nuk}{D_h}$$

for other parameters, such as h_m , mass transfer coefficient, which is defined as Stanton number divide velocity:

$$h_m = \frac{St_m}{u}$$

where Stanton number is related to 3 dimensionless number: Sherwood number, Reynolds number, and Schmidt number.

$$St_m = \frac{Sh}{ReSc}$$

Schmidt number is

$$Sc = \frac{\nu}{D}$$

Diffusion coefficient D is, calculated as:

$$D = 22.0 \times 10^{-6} \left(\frac{273.15 + T}{273.15} \right)^{1.75} \left(\frac{p_{standard}}{p} \right)$$

here $p_{standard}$ is 103125 Pa. And Sherwood number uses empirical equations,

$$Sh = Nu \left(\frac{Sc}{Pr} \right)^{\frac{1}{3}}$$

2.1.4 Simulation for M-cycle evaporative cooler

The simplified govern equations are ordinary differential equations. ODE solver was utilized for the calculation to get air temperature and absolute humidity ratio in each channel.

Equations of heat flux are a kind of linear equation. Fsolve was adapted to get surface temperatures in each channel.

The shooting method was occupied to find the correct outlet air temperature of the wet channel.

Newton's method can accelerate this process.

The detailed steps are as follows:

1. Set the initial air conditions, (T, p, RH, u)
2. Set the initial temperature for the interaction point. At the end of the M-cycle evaporative cooler, the outlet temperature of the dry channel and inlet temperature of the wet channel is the same. In addition, for a steady state, the outlet temperature of the product channel is also the same. Use dew point temperature as the initial temperature.
3. Calculate surface temperature by F_{solve} and send these values to ODE solver to calculate govern equations of heat and mass transfer. The direction of calculation is from the outlet to the inlet.
4. Compare the calculated inlet temperature of the dry channel with the initial one. If it is out of error margin, change the temperature value of the interaction point by Newton's method. Calculate temperature profile again.
5. Compare the new calculated inlet temperature of the dry channel with the previously calculated value. If it is still out of error margin, try step 5 until satisfying the set inlet temperature. Fig. 2.2 shows how the shooting method works.
6. Transfer the air density of each channel into the absolute humidity ratio. Output temperature and absolute humidity ratio distribution profile. End the calculation.

All of the steps were summarized in Fig. 2.3.

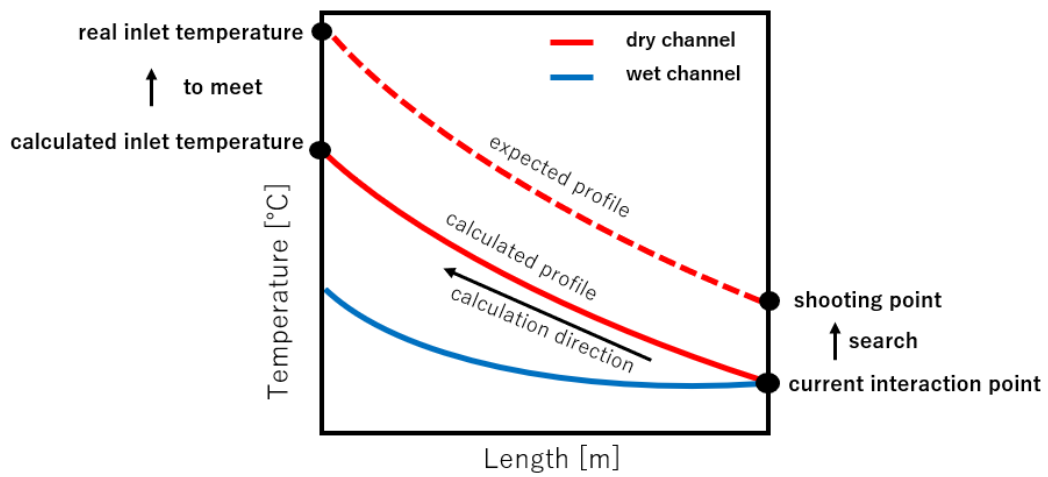


Fig. 2. 2 Description of the shooting method

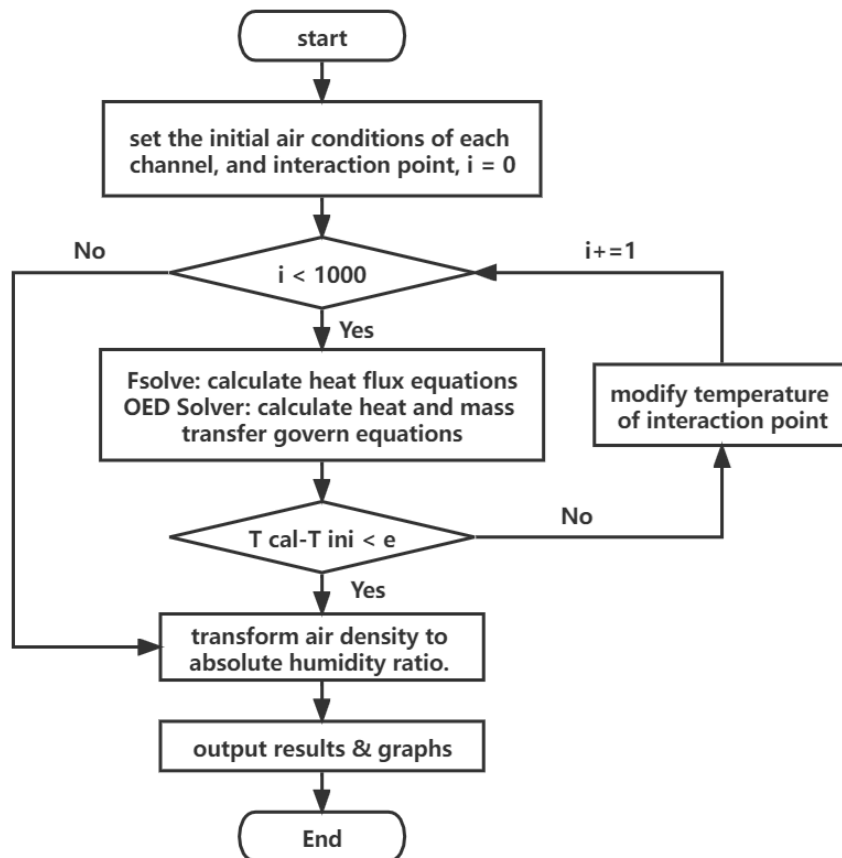


Fig. 2. 3 Flow chart of M-cycle evaporative cooler

2.1.5 Simulation verification

The simulation results were compared to experimental data, which was delivered by B. Riangvilaikul and S. Kumar in 2010 [19]. The parameters and values involved are listed in Table.2.

Table. 2 Parameters of M-cycle evaporative cooler for verification simulation

Parameter	value
Channel width	80 <i>mm</i>
Channel depth	0.5 <i>mm</i>
Channel length	1200 <i>mm</i>
Inlet air velocity	2.4 <i>m·s⁻¹</i>
Working air ratio	0.33

The range of inlet air temperature was from 25 to 45 °C. And humidity ratio was varied by 6.9, 11.2, 20.0, and 26.4 *g/kg*. Fig.2.4 is the simulation result of the M-cycle evaporative cooler compared to Riangvilaikul and S. Kumar's experiment data on the dependence of inlet air temperature and humidity ratio.

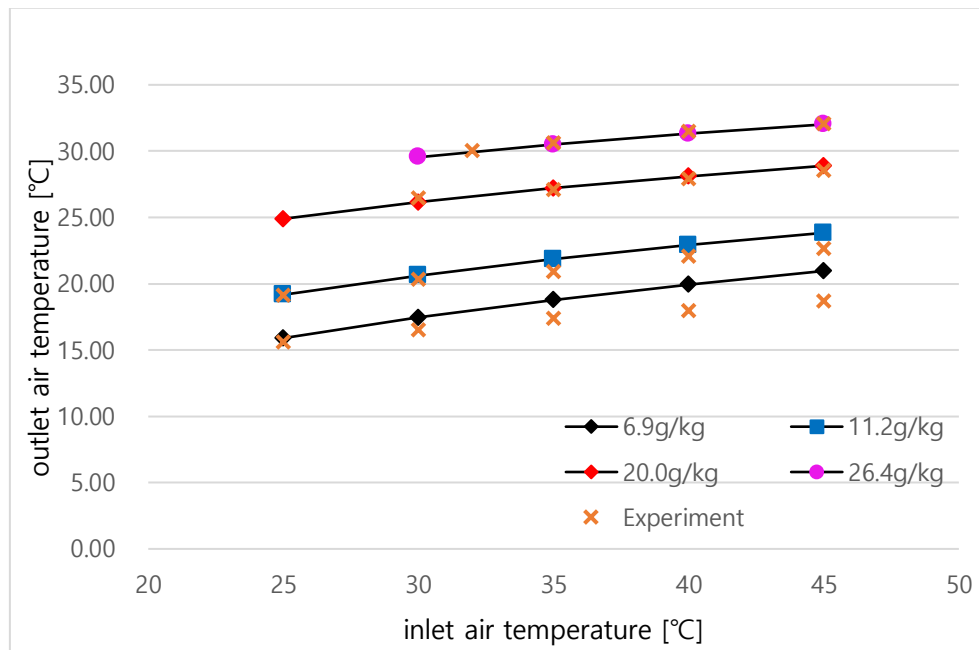


Fig. 2. 4 Comparison between simulation results with experimental data on the dependence of inlet air temperature and humidity ratio.

Fig.2.4 shows that when the humidity ratio was constant, the outlet air temperature was almost linear to the inlet air temperature. The simulation results fit well with experimental data when the humidity ratio was high than 20 g/kg . While at high temperature and low humidity ratio, where the temperature was between 30-45 $^{\circ}C$ and the humidity ratio was 6.9 g/kg , the simulation result had deviation compared with experiment data.

Fig.2.5 shows the simulation result of outlet air temperature varied to inlet air velocities, and it agreed with experimental data very well.

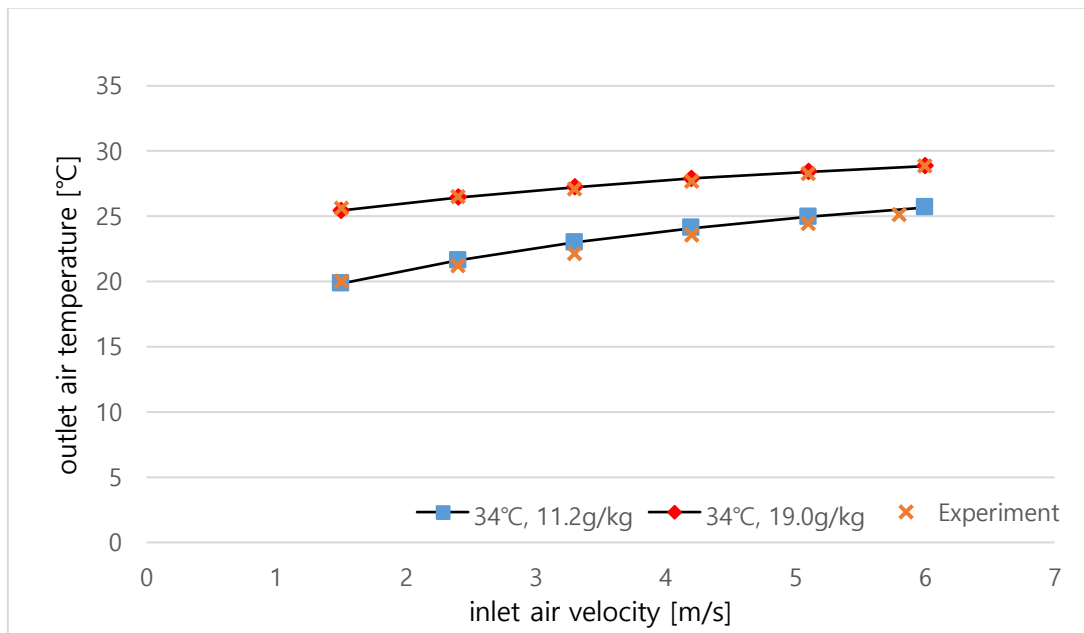


Fig. 2. 5 Comparison between simulation results with experimental data on the dependence of inlet air velocity.

2.2 Modeling of isothermal membrane-based dehumidification

A one-dimension model was developed for the simulation of the isothermal membrane-based separator. The working membrane was considered thin and flat, separate feed chamber and permeate chamber.

Ambient air flows into the feed chamber with ordinary pressure. A vacuum pump is connected to permeate chamber to create a low-pressure environment, which is lower than the water vapor partial pressure of ambient air. Due to the partial pressure difference between two channels, a part of water vapor would permeate through the membrane. Meanwhile, because of the selectivity of the membrane, a small part of dry air would also be permeated. After dehumidification, air flows out from the feed chamber would supply it to M-cycle evaporative cooler.

Assumptions are as follows:

1. Humid air is an ideal binary gas, which only contains dry air and water vapor.
2. Humid air flux is incompressible, laminar, and steady plug flow.
3. The separator is working under isothermal conditions
4. Pressure drop in each chamber is negligible.
5. Air and water vapor permeate through the membrane independently and membrane permeability is constant, which is not dependent on pressure difference or relative humidity.
6. Mass flux in feed channel obeys Fick's Law.

2.2.1 Equations for isothermal membrane-based dehumidification

The permeate flux ($mol \cdot m^{-2} \cdot s^{-1}$) through the membrane for each component can be described as

$$f^i = K^i (P_f y_f^i - P_p y_p^i)$$

where i denotes either air or water vapor, f denotes the feed chamber, p denotes the permeate chamber. K ($mol \cdot m^{-2} \cdot Pa^{-1} \cdot s^{-1}$) is the permeability, P (Pa) is the pressure of the chamber, and y is molar fraction ratio. Partial pressure difference is the driving force for the membrane permeate process.

The govern equations of mass transfer for water vapor and air in the feed channel is:

$$S u_a \nabla c^i + S D \nabla^2 c^i - f^i dA = 0$$

where c ($mol \cdot m^{-3}$) is the molar fraction, S (m^2) is the cross section area of chamber, u_a ($m \cdot s^{-1}$) is the velocity of inlet air, dA (m^2) is the membrane area of the control volume, D ($m^2 \cdot s^{-1}$) is the diffusion coefficient of air and water.

The diffusion coefficient is based on an empirical equation: [37]

$$D_{wa} = \frac{3.203 \cdot 10^4 \cdot T^{1.75}}{P \left(v_w^{\frac{1}{3}} + v_a^{\frac{1}{3}} \right)} \sqrt{\frac{1}{M^w} + \frac{1}{M^a}}$$

where P is the pressure of feed chamber, M ($kg \cdot mol^{-1}$) donates the molar mass, T (K) is the temperature, and v is the molecular diffusion volumes. $v_w = 12.7$ and $v_a = 20.1$.

For permeate chamber, due to steady state, there is such relation for molar fraction ratio:

$$\frac{y_p^w}{y_p^a} = \frac{f^w dA}{f^a dA}$$

At a steady state, the flux permeating into permeate chamber should be equal to flux discharged by a vacuum pump. Therefore, the ratio of the molar fraction is equal to the ratio of permeate flux.

Fig.11. illustrates the control volume of the separator.

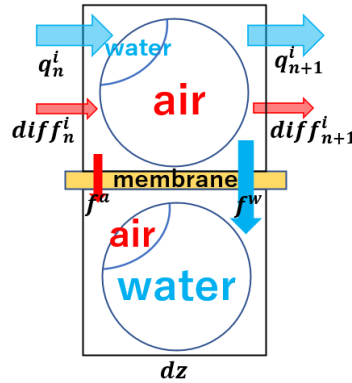


Fig. 2. 6 Control volume of the isothermal membrane-based separator

2.2.2 Simulation for isothermal membrane-based separator

The govern equation in the feed chamber was calculated by Finite-difference Method.

Successive Approximation was adapted to find out the steady state conditions.

The detailed steps are as follows:

Set the initial air conditions, (T, p, RH, u)

1. Divide the feed chamber into 100 control volumes. The molar flux of both components is uniform in each control volume initially.
2. Based on the initial molar flux condition, calculate molar flux distributions of both components along with the feed chamber, save the data.
3. Use the calculated data as the current condition. Then calculate molar flux distributions again.
Replace previous data with newly calculated one.
4. Repeat step5 1000 times. Save the final data as molar flux distributions at a steady state.
5. Transform molar flux distributions into partial pressure and molar flux distribution. Output graphs. End calculation.

Fig.2.7 has summarized all of the steps in a flow chart

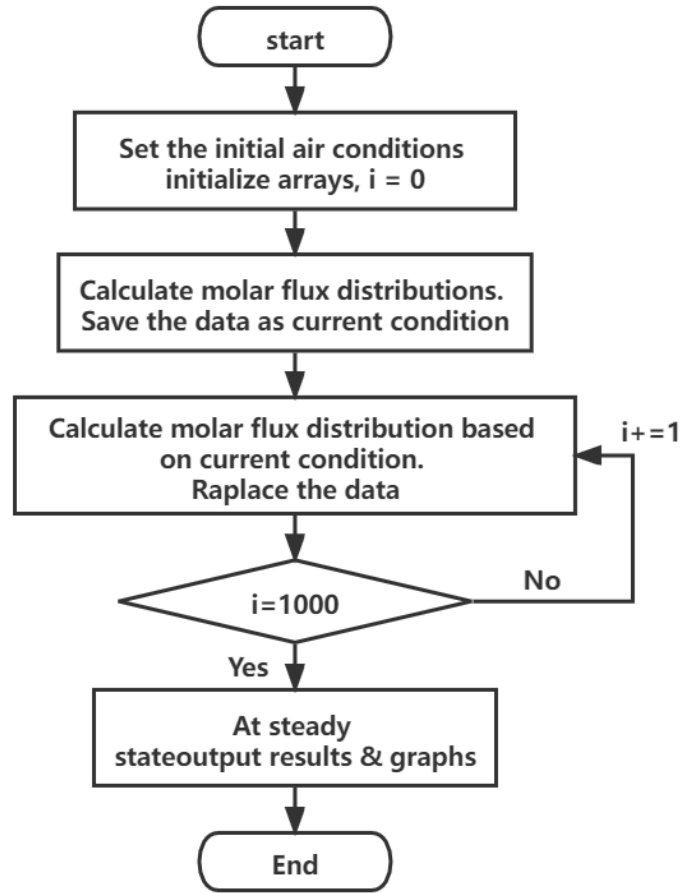


Fig. 2. 7 Flow chart of isothermal membrane-based separator

2.2.3 Verification of simulation

The simulation result was compared to experimental data which was established by T. D. Bui, F. Chen, etc. (2015).[29] The air condition and chamber parameters were shown in Table.3.

Table. 3 parameters of IMBD for verification simulation

parameter	value
inlet air temperature	24 °C
inlet air relative humidity	90%
inlet air velocity	0.3 $m \cdot s^{-1}$
chamber length	0.1 m
permeate pressure	0.01 Pa
water vapor permanence	$4.5 \times 10^{-7} mol \cdot m^{-2} \cdot Pa^{-1} \cdot s^{-1}$
membrane selectivity	450

Bui's experiment data showed that under vacuum conditions, the effectiveness of dehumidification was in ranged from 18.30% to 21.79%. However, in this work, the simulation effectiveness was only 13.82%. Chun et al. (2020) [34] used phase equilibrium to explain the deviation. He suggested that the permeability of the membrane was measured by the cup method, which was under atmospheric pressure. While in a vacuum, due to phase equilibrium, water vapor is more easily desorbed from the membrane, so the real permeability was larger than that measured by the cup method. According to this hypothesis, the permeability was modified from $4.5 \times 10^{-7} \text{ (mol}\cdot\text{m}^{-2}\cdot\text{Pa}^{-1}\cdot\text{s}^{-1}\text{)}$ to $7.25 \times 10^{-7} \text{ (mol}\cdot\text{m}^{-2}\cdot\text{Pa}^{-1}\cdot\text{s}^{-1}\text{)}$, and the result of the effectiveness became 21.34%, in the range of experiment data, which revealed that the modified simulation has matched the experimental data.

3 Analysis

3.1 Analysis of M-cycle evaporative cooler

Simulations were carried out to investigate the effect of different operations on the overall performance of the counter-flow M-cycle evaporative cooler. These parameters are inlet air temperature, inlet air relative humidity, inlet air velocity, channel length, permeate pressure, and membrane selectivity. The initial conditions of these parameters are listed in Table.3.

Table. 4 Initial conditions of M-cycle evaporative cooler

parameter	value
inlet air temperature	35 °C
inlet air relative humidity	45%
inlet air velocity	1 m·s ⁻¹
channel length	0.6 m
working air ratio	0.5

The effects of those parameters on outlet air temperature of product channel, COP, dew point effectiveness, and wet bulb effectiveness were investigated and analyzed. The effectiveness calculation was based on product channel air conditions.

Dew point effectiveness is expressed as:

$$\varepsilon_{dp} = \frac{T_{in} - T_{out}}{T_{in} - T_{dp}} \times 100\%$$

wherein denotes the inlet, out denotes the outlet, and dp denotes the dew point temperature of the inlet.

Wet bulb effectiveness is:

$$\varepsilon_{wb} = \frac{T_{in} - T_{out}}{T_{in} - T_{wb}} \times 100\%$$

where wb denotes the wet bulb temperature of the inlet.

The COP in M-cycle evaporative cooler measures the efficiency of sensible heat removing, which is described as:

$$COP_M = \frac{Q_{sensible}}{W_{blower}}$$

Sensible heat Q is calculated as:

$$Q_{sensible} = \dot{m}c_p(T_{in} - T_{out})$$

where $\dot{m}$ ($kg \cdot s^{-1}$) is the mass flux

The work of blower is [38]:

$$W_{blower} = W_{blower,base} \left(\frac{\rho}{\rho_{base}} \right) \left(\frac{\dot{V}}{\dot{V}_{base}} \right) \frac{SH}{SH_{base}}$$

where base denotes the base condition, ρ ($kg \cdot m^{-3}$) is the air density, $\dot{V}$ ($m^3 \cdot h^{-1}$) is the air volume flux, SH is the fan's static head, which can be determined as:

$$SH = f \times \Delta p$$

where f is a safety factor, and Δp (Pa) is the pressure drop.

Because the ratio of air density was almost one, and according to assumptions, the pressure drop is neglectable, the equation can be simplified as:

$$W_{blower} = W_{blower,base} \left(\frac{\dot{V}}{\dot{V}_{base}} \right)$$

here, $W_{blower,base}$ is 17 (W), and $\dot{V}_{base}$ is 100 ($m^3 \cdot h^{-1}$).

3.1.1 Verify of the performance of 2-channel and 3-channel

Fig. 3.1 reflects the impact of inlet air temperature on outlet air temperature for 2-channel and 3-channel MEC. The basic initial conditions were the same as Table.3. It is apparent that the two models have the almost same performance when the working air ratio is equal to 0.5. The following

analysis was all based on 2-channel MEC with a working ratio of 0.5. So that the result can be referred to 3-channel MEC to some degree.

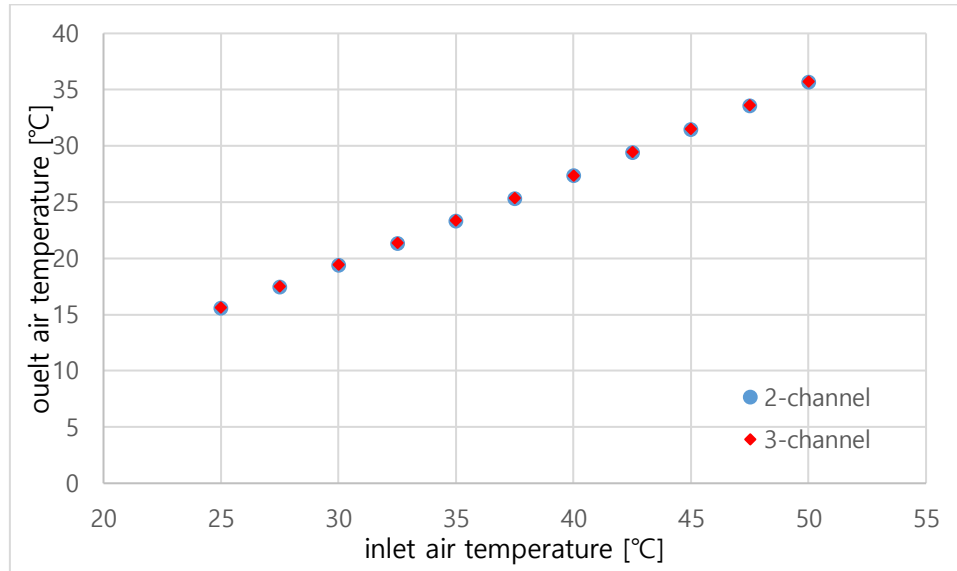


Fig. 3. 1 Comparison between 2-channel and 3-channel M-cycle evaporative cooler

3.1.2 Influence of inlet air temperature

Fig. 3.2 shows the influence of inlet air temperature on the outlet air temperature and the COP. The range of inlet air temperature is from 25 °C to 50 °C. The outlet air temperature and the COP increase steadily when the inlet air temperature goes up.

The COP increased from 9.08 to 12.80, which indicates that the M-cycle evaporative cooler has a larger cooling capacity in high temperature ambient. In 25 °C, the temperature drop of scale is nearly 10 °C, while in 50 °C, the scale reaches to 15 °C. Meanwhile, in Fig. 3.3 the dew point effectiveness is more than 90% when the inlet air temperature is higher than 43 °C.

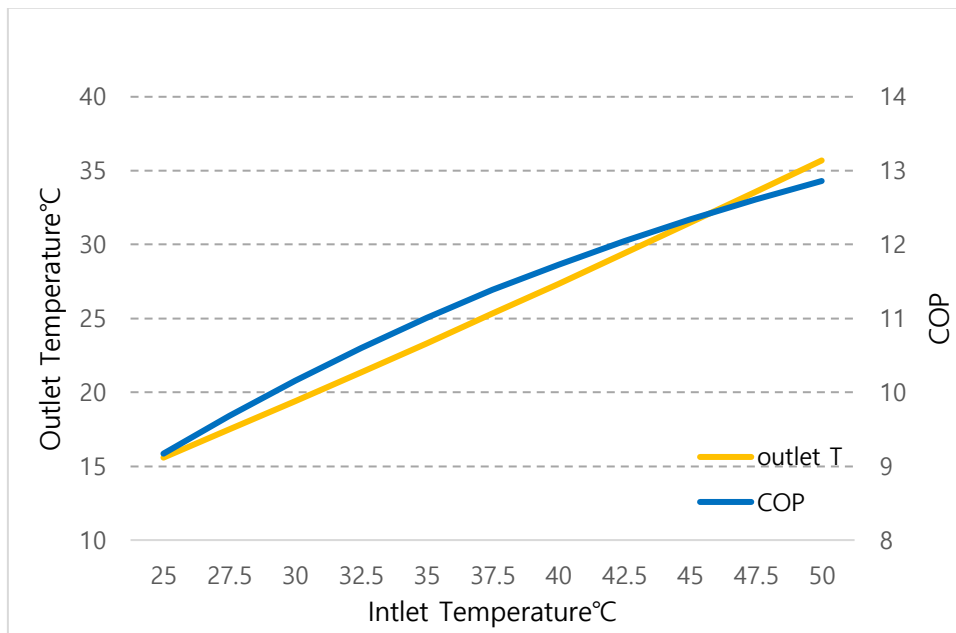


Fig. 3. 2 Impact of inlet air temperature on outlet temperature and COP for MEC

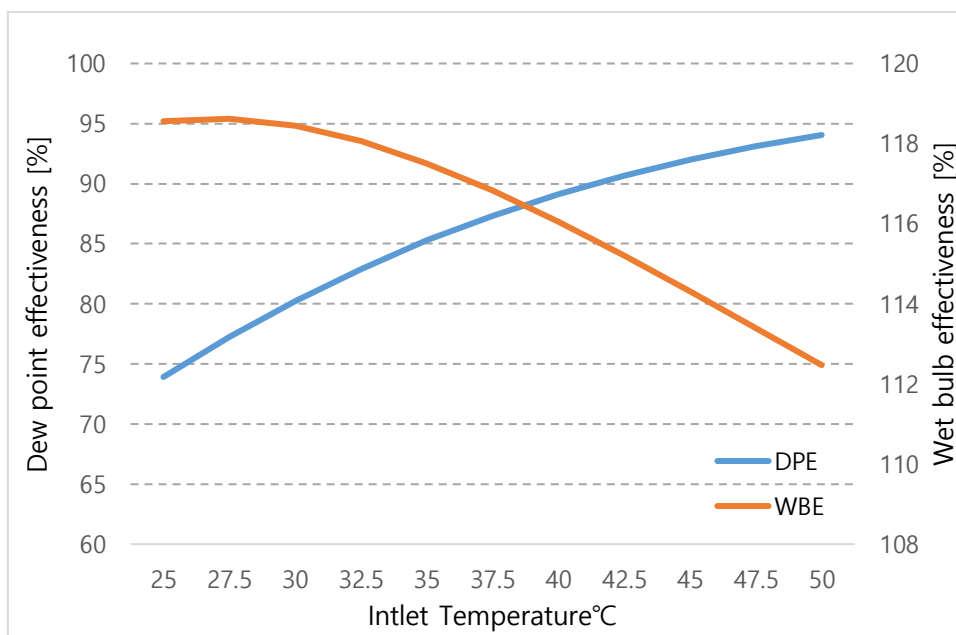


Fig. 3. 3 Impact of inlet air temperature on dew point and wet bulb effectiveness for MEC

3.1.3 Influence of inlet air relative humidity

Fig. 3.4 depicts the influence of inlet relative humidity, which is strongly affected the performance

of the M-cycle cooler. The range of relative humidity is from 15% to 95%. It is obvious that low inlet air relative humidity contains an extremely large cooling capacity, and the COP even reaches 19.85 when the relative humidity is 15%.

On the other hand, a low inlet air humidity ratio requires high effectiveness in the dehumidification part. Usually, it demands more electricity for dehumidification. Therefore, it is a kind of trade-off.

Detailed analysis for the whole system is described in section 3.3.

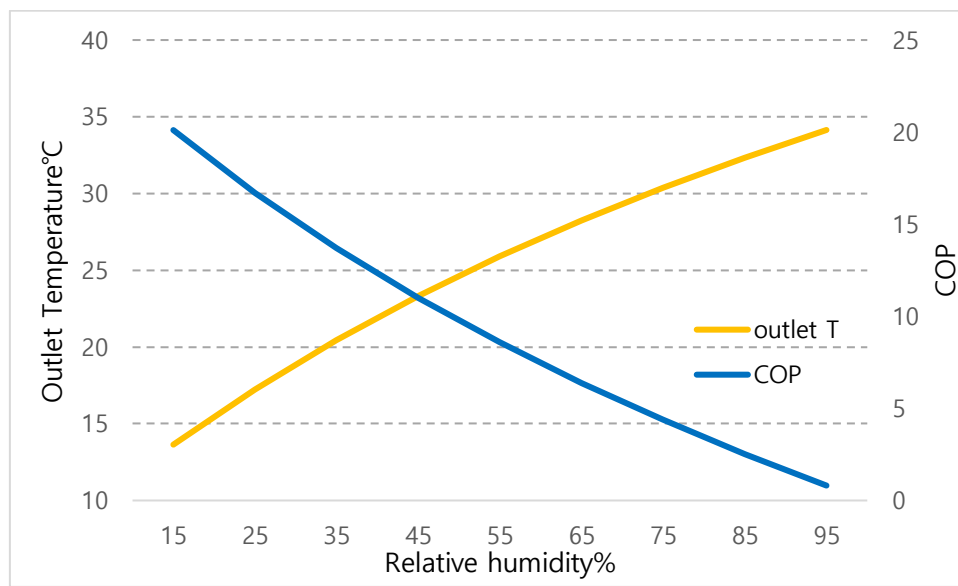


Fig. 3. 4 Impact of relative humidity on outlet temperature and COP for MEC

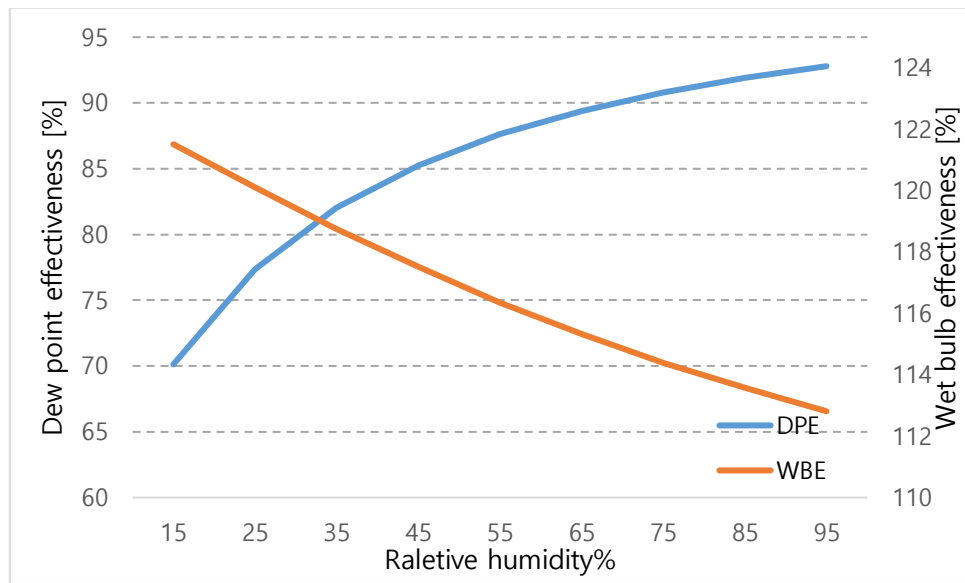


Fig. 3. 5 Impact of relative humidity on dew point and wet bulb effectiveness for MEC

3.1.4 Influence of inlet air velocity

Fig. 3.6 shows how to channel length affects the M-cycle performance. The range of inlet air velocity varied from $1.0 \text{ m}\cdot\text{s}^{-1}$ to $3.0 \text{ m}\cdot\text{s}^{-1}$. And as the inlet air velocity increased, the outlet air temperature also increased. Increasing the velocity account for less heat and mess transfer time, the cooling capacity is reduced as well. Moreover, higher air velocity means larger air flux. So that the blower demands more power, which makes the COP decline.

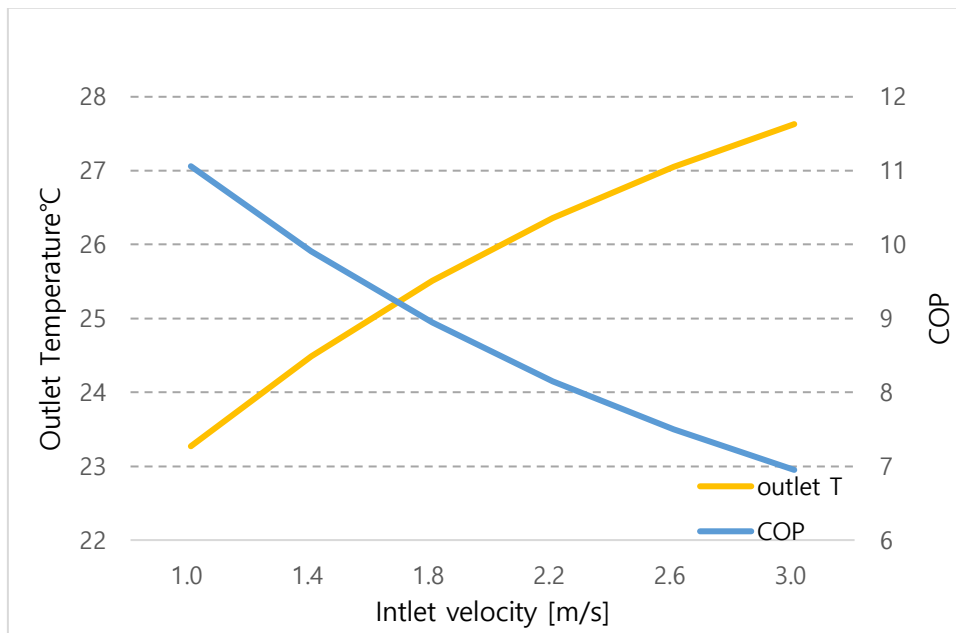


Fig. 3. 6 Impact of inlet air velocity on outlet temperature and COP for MEC

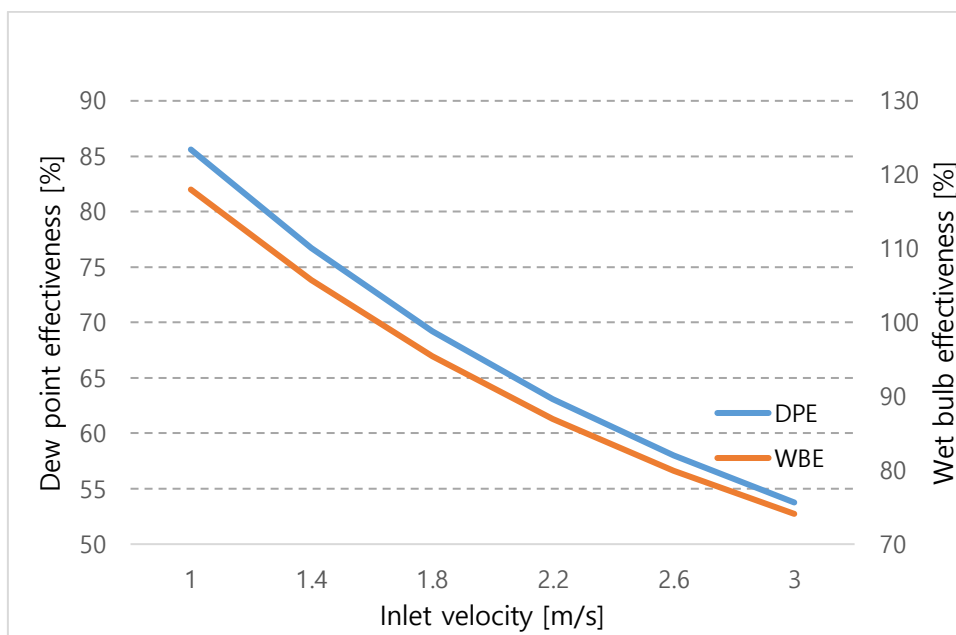


Fig. 3. 7 Impact of inlet air velocity on dew point and wet bulb effectiveness for MEC

3.1.5 Influence of channel length

Fig. 3.8 reflects the influence of channel length during evaporative cooling. The range of chamber

length was from 0.2 m to 1.4 m. The outlet temperature sharply decreased from 27.63 to 22.64 °C, when the channel length was between 0.2 m to 0.8 m. Meanwhile, the COP increase was significant, too. It was from 6.95 to 11.83.

However, in the region between 0.8 m to 1.4 m, both two factors' change tended to be flat. Fig. 3.9 shows that the dew point effectiveness is more than 90% and approaches 100% when the channel length was between 0.8 m to 1.4 m. This result illustrated that the air was fully cooled inner the channel.

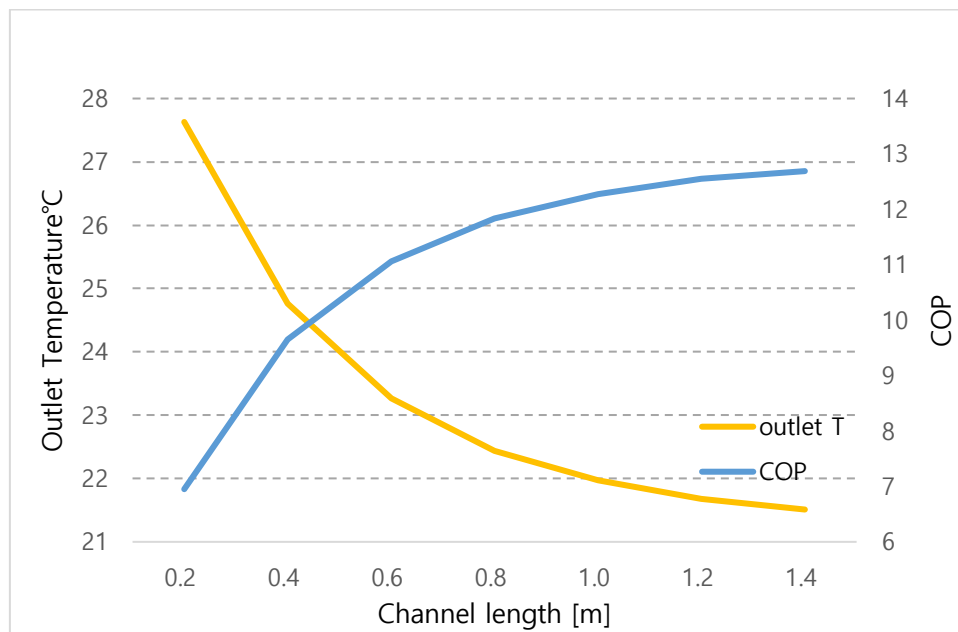


Fig. 3. 8 Impact of channel length on outlet temperature and COP for MEC

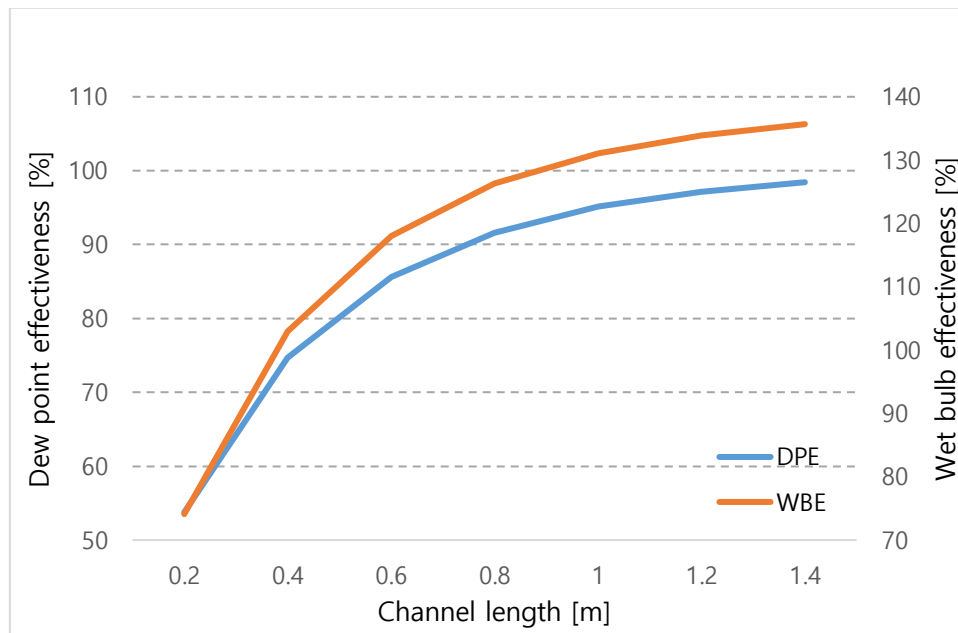


Fig. 3. 9 Impact of channel length on dew point and wet bulb effectiveness for MEC

3.2 Analysis of isothermal membrane-based separator

Simulations were carried out to investigate the effect of different operationalificant on the overall performance of the counter-flow isothermal membrane-based separator. These parameters are inlet air temperature, inlet air relative humidity, inlet air velocity, chamber length, permeate pressure, and membrane selectivity. The initial conditions of these parameters are listed in Table.4.

Table. 5 Initial conditions of isothermal membrane-based dehumidification.

parameter	value
inlet air temperature	35 °C
inlet air relative humidity	80%
inlet air velocity	0.2 m s ⁻¹
chamber length	0.2 m
permeate pressure	800 Pa
water vapor permanence	7.25 × 10 ⁻⁷ mol·m ⁻² ·Pa ⁻¹ ·s ⁻¹
membrane selectivity	450

The effects of those parameters on the dehumidification effectiveness and COP were investigated

and analyzed.

Dehumidification effectiveness is expressed as:

$$\varepsilon_{de} = \frac{x_{in} - x_{out}}{x_{in}} \times 100\%$$

wherein x_{in} denotes inlet, x_{out} denotes outlet, and x (kg/kg dry air) is the humidity ratio.

The COP for dehumidification process is described as the efficiency of removing the latent heat. The vacuum pump was supposed to work in an isothermal way. Ideal COP is calculated as [29]:

$$COP_{iso} = \frac{L_w M^w F^w}{\frac{(F^w + F^a)RT}{0.6} \ln\left(\frac{p_{amb}}{p_{per}}\right)}$$

where w denotes water vapor, a denotes dry air, amb denotes ambient, and per denotes permeate chamber. L_w is latent heat, F ($mol \cdot s^{-1}$) is the molar flux amount permeating through the membrane, R is the ideal gas constant, T is inlet air temperature, and p is pressure. 0.6 is mechanical efficiency.

3.2.1 Influence of inlet air temperature

Fig. 3.10 depicts the influence of inlet air temperature during the dehumidification process. The range of temperature is from 25 to 45 °C. The effectiveness increases significantly when inlet temperature is raised, where the effectiveness is from 37.95% to 49.25%. However, the COP only has a slight change in the temperature range, and the difference is less than 0.1. It is noticed that there is a peak near 35 °C.

When the temperature is getting higher. Under the same relative humidity, the partial pressure of water vapor grows sharply. This phenomenon makes a larger partial difference of water vapor partial pressure, which makes effectiveness increase.

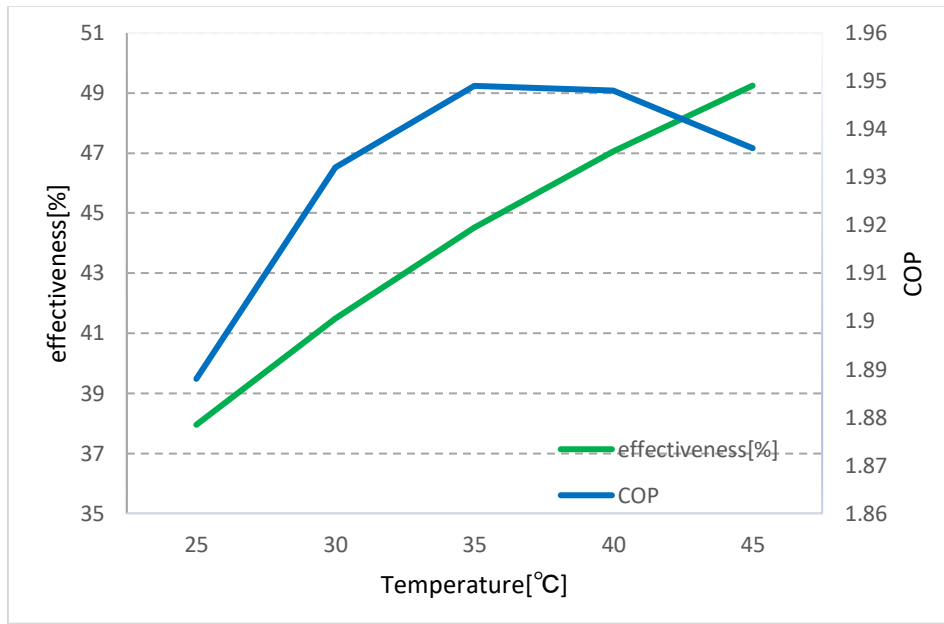


Fig. 3. 10 Impact of inlet air temperature on dehumidification effectiveness and COP for IMBD

3.2.2 Influence of inlet air relative humidity

Fig. 3.11 shows the influence of inlet air relative humidity during the dehumidification process. The range of relative humidity is from 40% to 100%. The effectiveness increases slightly when inlet relative humidity goes up, where the effectiveness is from 37.44% to 46.20%. While, the COP has a significant change, which is from 1.76 to 1.98.

Under the same temperature, when relative humidity increases, the partial pressure of water vapor would increase uniformly. Thus, the increase of effectiveness is steady. On the other hand, there is almost no change in the permeation of dry air. The COP significantly raises due to the increase of water vapor permeation.

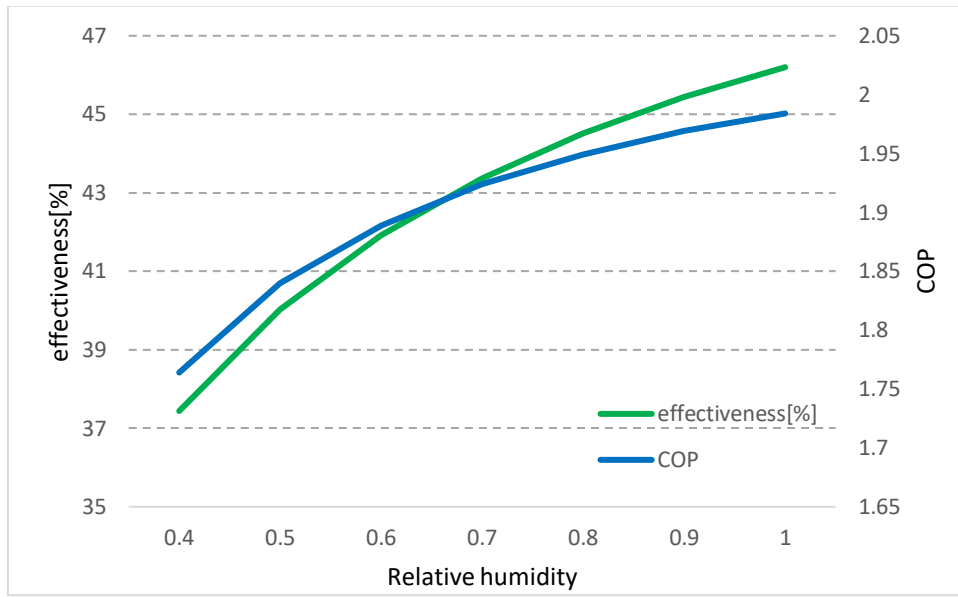


Fig. 3. 11 Impact of relative humidity on dehumidification effectiveness and COP for IMBD

3.2.3 Influence of inlet air velocity

Fig. 3.12 indicates the influence of inlet air velocity for dehumidification. The range of inlet air velocity is from $0.1\text{ m}\cdot\text{s}^{-1}$ to $0.5\text{ m}\cdot\text{s}^{-1}$. The effectiveness has a very sharp decrease as the inlet air velocity is speeded up, where the effectiveness is from 66.54% to 21.62%. Reversely, the COP slightly increases, from 1.90 to 1.98.

Higher velocity leads to less permeation time in each control volume. As Fig shows, the lower velocity the better effectiveness. In realistic, supply air amount should be guaranteed. Under a certain supply air amount, low inlet velocity requires a large sectional area for the separator, which contains more channels. Therefore, researchers need to do a compromise for it.

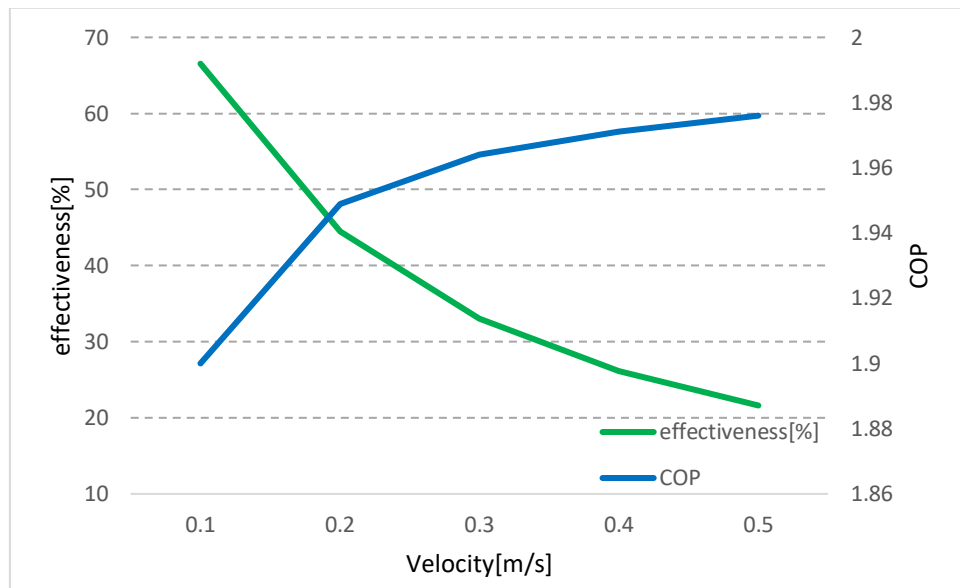


Fig. 3. 12 Impact of inlet air velocity on dehumidification effectiveness and COP for IMBD

3.2.4 Influence of chamber length

Fig. 3.13 shows the influence of chamber length during dehumidification. The range of chamber length is from 0.1m to 0.5m. The effectiveness has a very sharp increase when the chamber length gets longer, where the effectiveness is from 26.14% to 73.07%. The COP slightly decreases, from 1.97 to 1.87.

A long chamber provides a longer permeation period, which means more water vapor could be removed. However, in the latter half of the chamber, the partial pressure of water vapor was declined significantly, so that the pump waste some power to suck air, where the COP decreases. The COP does not change a lot with respect neither to chamber length nor inlet air velocity. Similar to slowing down the inlet air velocity, chamber length is also restricted to room limitation.

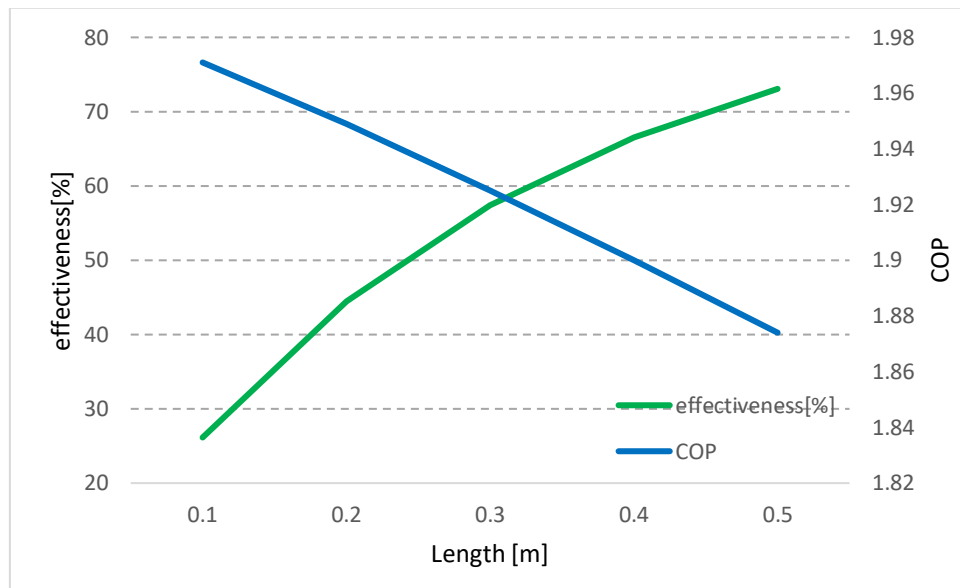


Fig. 3. 13 Impact of chamber length on dehumidification effectiveness and COP for IMBD

3.2.5 Influence of permeate pressure

The permeate pressure is the most important factor affecting electricity consumption during the dehumidification process. The partial pressure of water vapor in ambient air is 4523 Pa, under the condition of Table.4, Fig. 3.14 indicates how the permeate pressure influences the separator performance. The setting of permeate pressure was from 3600 Pa to 1 Pa. The tradeoff is apparently. High permeate pressure leads to low electricity consumption and narrows the partial pressure difference. The COP can be above 2.5, but effectiveness is less than 25%. Reducing the permeate pressure approaching to vacuum makes the COP drop dramatically. However, the effectiveness raise is extremely small. According to Fig. 3.14, for the balance of the effectiveness and the COP, permeate pressure, no less than 400 Pa is an efficiently setting.

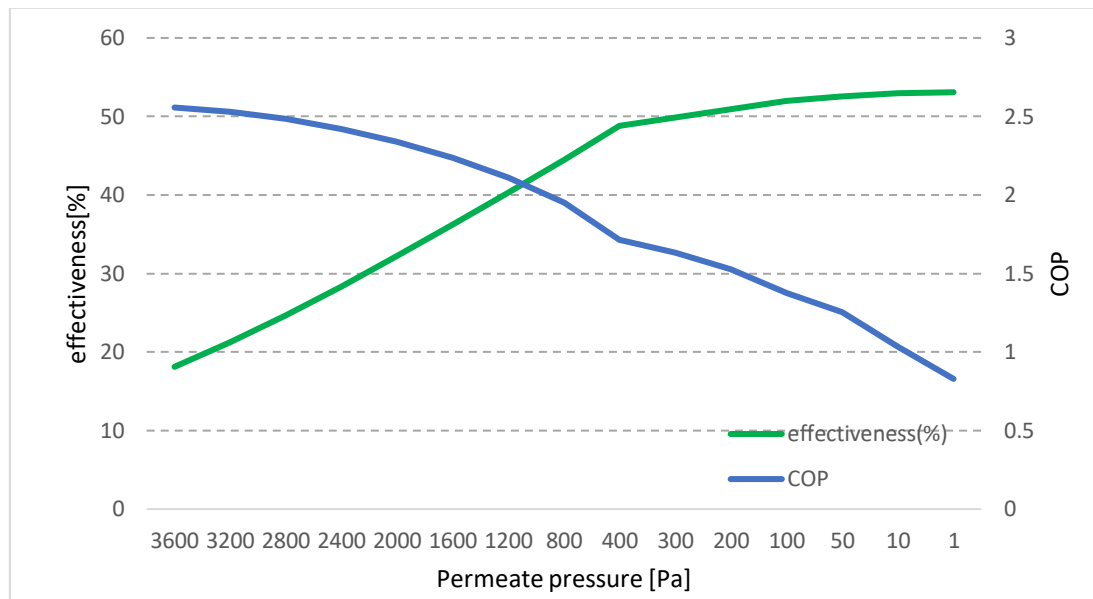


Fig. 3. 14 Impact of permeate pressure on dehumidification effectiveness and COP for IMBD

3.2.6 Influence of selectivity

Selectivity is one of the membrane properties and depends on the manufacturing technique. In practical utilization, the selectivity can hardly change. While in simulation, it can be investigated.

Fig. 3.15 shows how selectivity influences the effectiveness and the COP in dehumidification. In low selectivity, although the effectiveness is higher, the COP is even less than 1. That was because most of the work input is used to permeate dry air, which makes the system inefficiency. On the contrary, if the selectivity is high enough, the work adapted in permeating dry air becomes neglectable, even approaching ideal conditions, where the selectivity is infinitely large. That is why in high selectivity, the effectiveness and the COP changes more mildly than in the low range.

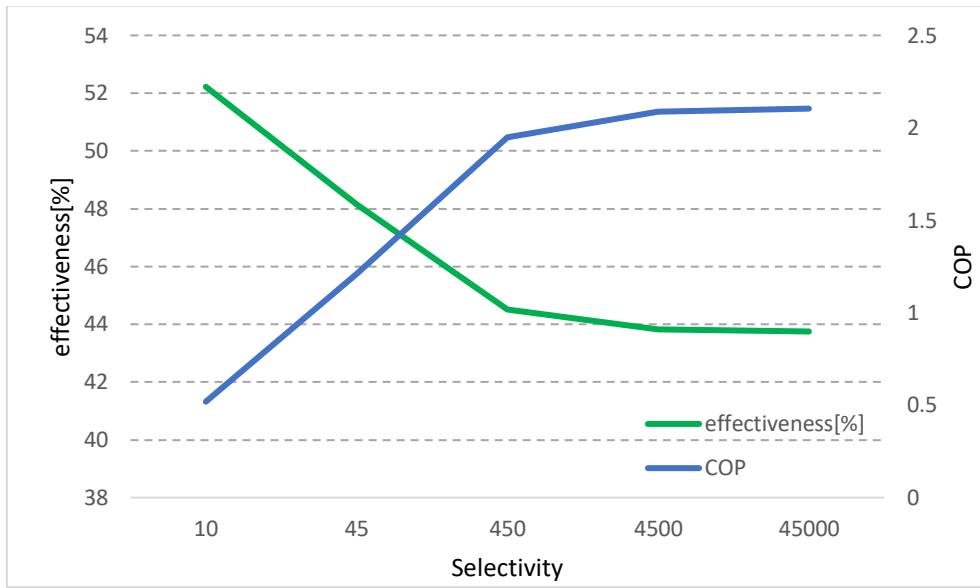


Fig. 3. 15 Impact of membrane selectivity on dehumidification effectiveness and COP for IMBD

3.3 Analysis of the whole cooling system

The whole cooling system was integrated by MEC and IMBD. And the performance was mainly investigated by four factors, inlet air temperature, inlet air relative humidity, inlet air velocity, and channel length. Due to the difference between dehumidification and evaporative cooling, the velocity ratio of MEC to IMBD was set as 4. The total length of the system was 1.2 m. Basic initial air conditions were listed in Table.5.

Table. 6 Initial conditions of the whole cooling system.

parameter	value
Inlet air temperature	35 °C
Inlet air relative humidity	80%
Inlet air velocity of MEC	1 m·s ⁻¹
Velocity ratio	4
Length of IMBD	0.2 m
Length of MEC	0.6 m

Other properties of IMBD and MEC were same in Table.3 And Table.4.

The total COP is calculated by

$$COP_{total} = \frac{L_w M_w F_w + \dot{m} c_p (T_{in} - T_{out})}{\frac{(F_a + F_w) R T \ln \left(\frac{p_{amb}}{p_{vac}} \right)}{0.6} + W_{blower}}$$

The image of the whole system is shown in Fig. 3.16.

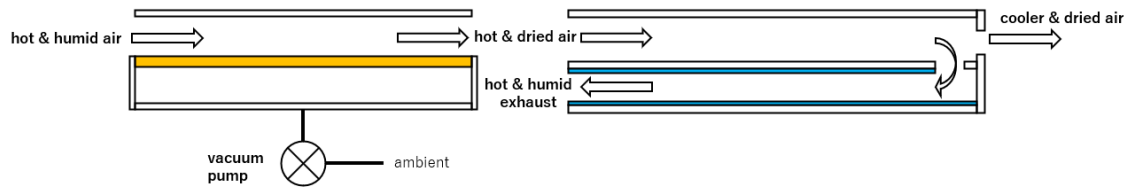


Fig. 3. 16 Schematic of the whole cooling system

3.3.1 Influence of inlet air temperature

Fig. 3.17 shows the outlet air temperature and COP varies of inlet air temperature from 25 to 45 °C.

The outlet air temperature was nearly linear depending on the inlet air temperature. However, the system had better performance, according to the COP profile. At the range of 25 to 30 °C, the COP was maintained around 2.3, while with the inlet air temperature increase after 30 °C, the COP dropped significantly.

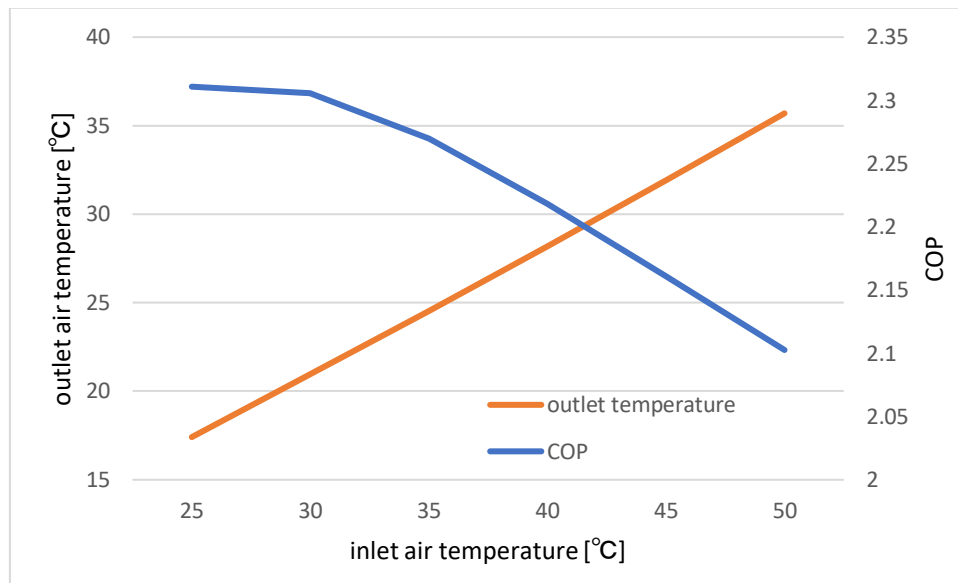


Fig. 3. 17 Impact of inlet air temperature on outlet air temperature and COP for whole cooling system.

3.3.2 Influence of inlet air humidity ratio

Fig. 3.18 reflects that the Relative humidity caused a critical effect on the whole system performance. It is distinct that the COP sharply decreased as the inlet air get moisture. On the other hand, similar to the influence of inlet air temperature, the outlet air temperature had a linear relation with the relative humidity.

In high humidity conditions, dehumidification makes the MDC provide an air of lower temperature compared to standalone MDC. While in low humidity conditions, although the COP exceeded 3, utilizing MDC independently is a more economic choice. The air of low humidity maintains a large cooling capacity for evaporation.

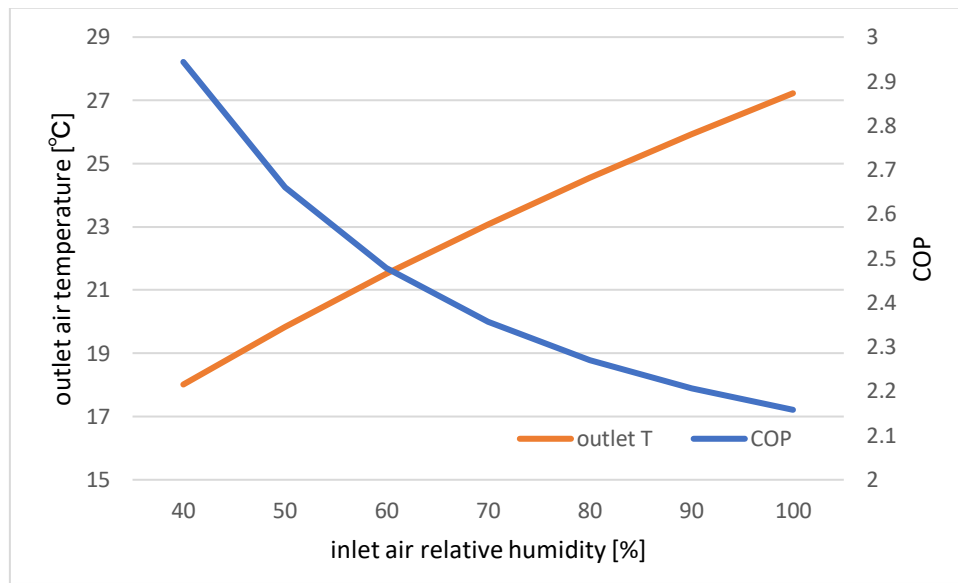


Fig. 3. 18 Impact of relative humidity on outlet air temperature and COP for whole cooling system

3.3.3 Influence of inlet air velocity

Fig. 3.19 indicates the influence of the inlet air velocity for the whole system. In Fig the inlet air velocity was in M-cycle, which was four times than velocity in IMDB. The range of inlet air velocity is from $0.6 \text{ m}\cdot\text{s}^{-1}$ to $1.5 \text{ m}\cdot\text{s}^{-1}$. And it affected the outlet air temperature apparently, where the outlet air temperature raised from 19.44 to 27.57 °C. However, the COP was almost unchanged, which is because the vacuum pump consumed most of the work input. Enhancing the inlet air velocity only requires more power for the blower. Whereas, compared to a vacuum pump, the energy consumption is not significant.

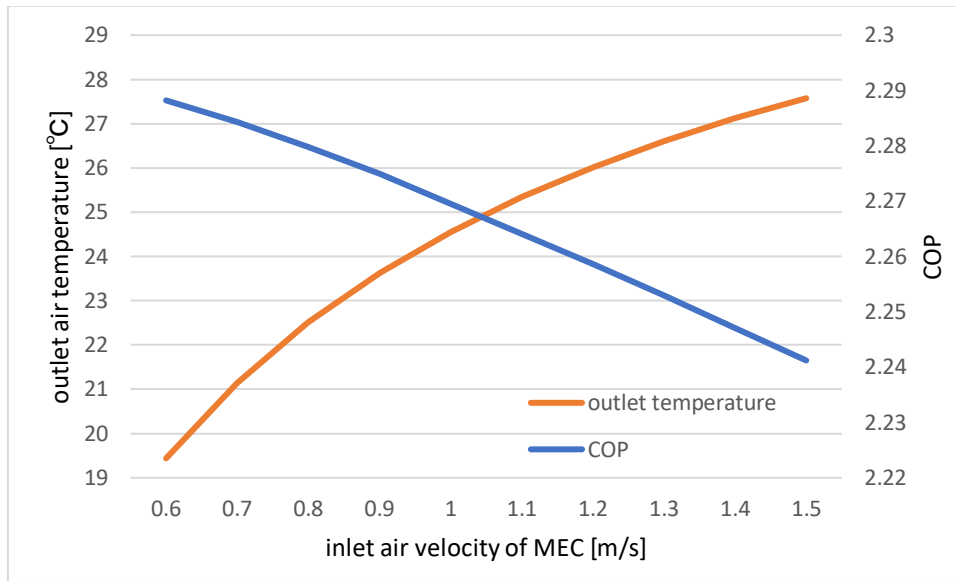


Fig. 3. 19 Impact of inlet air velocity on outlet air temperature and COP for whole cooling system.

3.3.4 Influence of channel length

Based on the total system length of 0.8 m, various combinations of IMBD and MEC were investigated. For example, if the channel length of MEC is 0.7 m, the chamber length of IMBD is 0.1m, and so on. In Fig. 3.20, different to standalone MEC, with dehumidification, MEC could give low temperature outlet air in the shorter channel. In that case, the IMBD part obtained a long chamber. The inlet air could be fully dehumidified along with the feed chamber. But there was a tradeoff. In Fig. 3.20, when MEC and IMBD length were equal to 0.4m, the outlet air temperature reached the lowest point. After that, lengthening the IMDB length would give drier air, MEC could not cool down it efficiently with the too short channel. In realistic, large-scale membrane is hard to be manufactured. And the durability of large-scale membrane needs to be studied. It is wise to choose a suitable length combination according to the required condition.

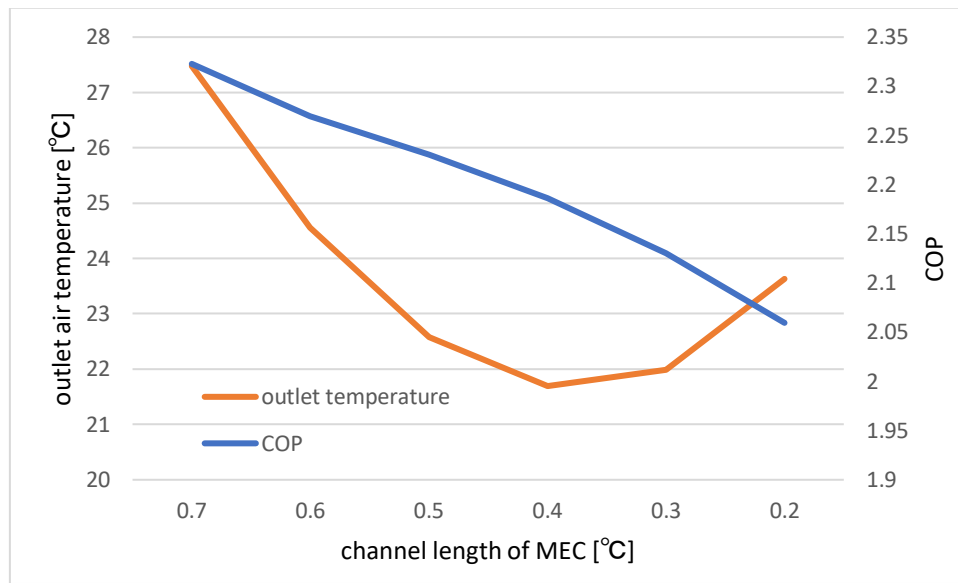


Fig. 3. 20 Impact of channel length on outlet air temperature and COP for whole cooling system.

3.3.5 Evaluation by representative cities around the world

Several representative cities were selected to evaluate the whole cooling system's performance. The temperature of hot days was calculated by 2 centigrade degrees adding to monthly highest average high temperature, round up. The relative humidity was based on monthly relative humidity data. The parameters of the system were the same as the parameters in section 3.3.

Table. 7 Performance of the cooling system in several cities

City	Temperature of Hot day °C	Relative humidity %	Supply air temperature °C	Supply air relative humidity %
Fukuoka	34	74	22.99	46.73
Pusan	32	87	23.30	54.78
Beijing	33	77	22.70	48.79
Jeddah	41	53	24.13	32.94
Sydney	28	62	17.26	42.38

According to Table.6, in humid areas, such as Fukuoka, Pusan, and Beijing, the cooling system was able to provide air at 22°C, which could satisfy most indoor refrigeration requirements. Meanwhile, in hot areas, such as Jeddah, with relatively dry ambient air, the supply air could also approach 22 °C. Thus, the cooling system was expected to adapt to a wide area of the world.

4 Conclusion

In this work, M-cycle evaporative cooler integrated with isothermal membrane-based dehumidification has been investigated. The whole system only uses air and water as working flow, instead of any organic refrigerants. A thermodynamic model for the system has been established and numerical analysis for different inlet air conditions and system operational parameters has been carried out.

- (1) Relative humidity significantly affects the supply air temperature and COP. For standalone MEC, it can hardly work in high humidity conditions. When combined with dehumidification, the system has a good cooling capacity under the COP of more than 2.2.
- (2) Energy consumed by vacuum pump is the critical factor to the total COP. On the one hand, lower permeate pressure has better dehumidification effectiveness and provides cooler supply air, at the cost of larger power input to the vacuum pump. On the other hand, during the whole process of dehumidification and evaporative cooling, removing latent heat accounts for the most part. Therefore, the total COP is close to the COP of IMBD.
- (3) Low inlet air velocity and appropriate long IMBD chamber length make cooler supply air temperature. However, for inlet air velocity, the cooling capacity declines because of the small air flux. For the chamber length of IMDB, the best cooling performance occurs when it is equal to the length of MEC. Nonetheless, the manufacturing of large-scale membrane is suspected. Membrane-based dehumanization was still in the experiment process and other properties should be further studied.
- (4) The simulation results show that the system works well in high humidity areas and is expected

to be applied worldwide. With totally green working fluid, this system is environmentally friendly and hopes it could reduce carbon emission.

Finally, there are mainly 3 parts could improve the system performance. First, membrane properties are key points that determine the whole system's performance. Larger permeability makes better dehumidification and high selectivity reduces power consumption. The second part is the creation of vacuum conditions. If the concentration difference could be maintained by better mechanical efficiency or device configuration, it would require less power. The third is about the structure, including channel shape, length, and connection. It is noted that the air velocity is different in IMBD and MEC. Thus, structure arrangement is a vital point in realistic engineering. These performances can be easily calculated by this simulation and it is expected to be a reference of manufacture in the future.

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