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A comprehensive review of the Maisotsenko air conditioning systems; evaluation methods, obtained results, industrial status and future research direction

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Abstract: Maisotsenko cycle (M-cycle) is a promising air cooling technique which can reduce the temperature of air flow until dew point which was not possible either in direct contact techniques or former indirect evaporative methods. M-cycle systems have been employed previously on gas turbines, air conditioning systems, cooling towers, electronic cooling etc. Simultaneous consideration of all of them prevents detailed presentation. To that reason and because of the wide application of air conditioning systems, this paper focuses only on the use of M-cycle on air conditioning systems. Moreover, former types of indirect evaporative air coolers which do not work based on Maisotsenko cycle are not considered in present study. Researchers have evaluated the M-cycle characteristics via different methods including analytical solution, numerical simulation, statistical design methods and experimental-techniques all of which is divided into several categories as well. All said methods are organizedly discussed and compared in this paper. It has been tried to provide an evolutionary viewpoint for analytical solutions of M-cycle. Thus, analytical solutions were reorganized with unique abbreviations in order to become more understandable and comparable with each other. All M-cycle parameters (which have been analyzed via numerical or experimental ways) are coherently systematized and then a comprehensive-compact view of obtained results is presented. Finally, current status of M-cycle industry is summarized and the future research direction on M-cycle is proposed.

Keywords: Heat exchanger, Maisotsenko cycle, Evaporative, Dew-point, Wet-bulb, Air conditioner

Contents

Nomenclature

A area (m^2)

c Specific heat of moist air (Kj/Kg. ºC)

h Specific enthalpy (Kj/Kg)

h_g Specific enthalpy of water vapor (J/Kg)

h_{fg} Latent heat of vaporization of water (J/Kg)

k_p Thermal conductivity of plate (W/m ºC)

L Length (m)

Le Lewis factor

q Heat transfer rate (W)

R Thermal resistance (m^2K/W)

T Temperature (ºC)

T_f Water film temperature (ºC)

U Overall heat transfer coefficient (Kw/m^2 ºC)

$\dot{m}$ Mass transfer rate (Kg/s)

w Humidity ratio (Kg moisture /Kg dry air)

W Mass transfer between water film and moist air (Kg/s)

Special characters

ζ Ratio between the change of enthalpy and wet-bulb temperature

δ_p Thickness of plate (m)

δ_w Thickness of water film (m)

α Convective heat transfer coefficient (W/m^2 K)

β Convective mass transfer coefficient (W/m K)

Subscripts

wa Working air (secondary air).

Pa Primary air (product air)

f water film

wb wet-bulb

s saturated

1. Introduction

International Energy Outlook (2017) [1] has recently reported notable information regarding to energy consumption by buildings. Rising standards of living in non-OECD (Organization for Economic Co-operation and Development) countries increase the demand for appliances, personal equipment, and commercial services [1]. It is projected that electricity use in buildings grows 2% annually, while total energy consumptions in buildings would increase by 32% between 2015 and 2040. The buildings sector (both commercial and residential), would account for almost twenty one percent of the world's delivered energy consumption in 2040 [1]. Air conditioning systems are key energy consumer, using nearly fifty percent of the total consumed electricity in the buildings [2-4]. Hence, recognizing efficient air conditioning systems is required to reduce energy use in buildings.

Current air coolers especially compressor-based coolers increase significantly the electrical consumption in warm seasons, as they require electrical power to make cooling and circulate air. Most power-plants have to work with their maximum capacity in hot seasons in order to produce required extra electrical power which is mostly due to air conditioning systems. Moreover, increment of air temperature reduces gas-turbine-plants (one of the common powerhouses) performance which aggravates their working conditions. Subsequently, electrical power outage may occur in some regions of each country. Regardless the extra applied load on electrical power-plants, current air coolers increase the electrical power consumption cost for both home-users and industrial consumers as well. Hence, some governments have to allocate specific subsidies for electrical power consumers for some hot-weather regions of their country in summers. Obviously, this policy has its own economic issues and can't be considered as a permanent solution. Irrespective of economic features, all air coolers which work based on refrigerants may cause irreparable damages on our environment (particularly Ozone layer). Furthermore, burning of fossil fuels to generate extra electrical power causes air pollution too.

Employment of air-water direct contact coolers (Fig. 1a) instead of compressor based systems may seem as a solution, as they only require electrical power for air circulation and cooling is made by water evaporation (into the air) without external power input. Although electrical consumption of direct contact evaporative air coolers (Fig. 1a) is usually less than the compressor-based coolers, they have some unavoidable disadvantages as below.

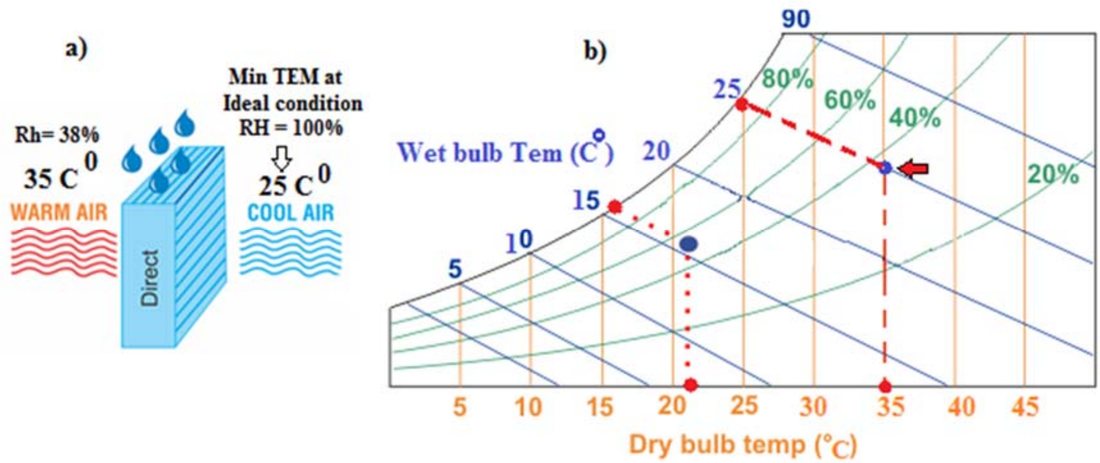


Fig. 1. Air-water direct contact mechanism a) general view b) process in psychometric chart

1. They are not able to reduce the air flow temperature less than wet-bulb temperature of inlet air. In other words, the minimum theoretical ideal temperature which can be achieved is wet-bulb temperature of inlet air (Fig.1b).
2. The mechanism causes reduction of air temperature (adding moisture) not only results in people's discomfort but also is harmful for electrical devises.
3. This type of evaporator does not work on humid climate and the performance of such direct evaporative mechanisms significantly is affected by climates.
4. The water consumption of direct contact coolers is high and there is a possibility of health issues which can be due to unclean droplets of water fluid are evaporated by direct contact with the air.

Because of different problems in both compressor-refrigerant based cooling techniques and direct-evaporative method (as mentioned above), researchers enthusiastically started to study on indirect evaporative coolers (see Fig. 2a). Although the initial type of indirect evaporative cooler which basically combines a DEC and a heat exchanger (HE) as shown in Fig. 2(a) does not add any moisture to the product air, ie. overcomes the disadvantage 2 of DEC mentioned above, its performance is low in comparison with former direct contact evaporators. The outlet temperature of product air could reach the wet-bulb temperature of the incoming air theoretically [5]. Moreover, in completely ideal condition, outlet temperature of wet side of air flow could increase from its inlet wet-bulb temperature into the product air inlet dry bulb temperature (at saturated condition). Said ideal condition requires infinite amount of surface area and pure counter flow configuration. However, in real condition, temperature of dry side reaches only point “c” in Fig. 2b. Hence, for many years, indirect evaporative air coolers were not commercialized because of poor heat transfer rate which does not justify the excessive material and manufacturing cost.

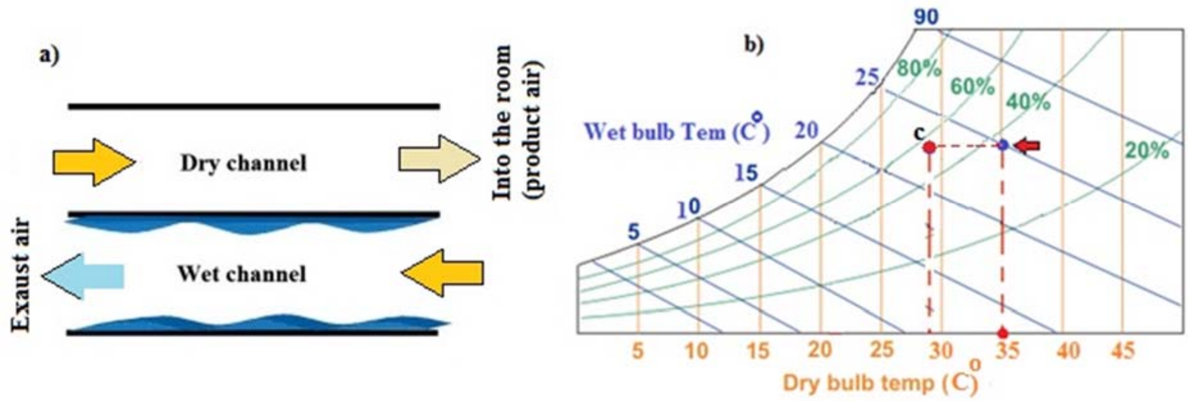


Fig. 2. Air-water indirect contact mechanism a) general view b) real process in psychometric chart

Finally, Maisotsenko (at around 2000) presented a novel form of indirect evaporative exchanger in which all aforesaid problems of indirect evaporative method were solved. Based on Maisotsenko cycle, the wet side air fluid is pre-cooled before entering the wet channel. However, this precooling process can be occurred via different techniques. Fig.3 (modified from [6]) illustrates different counter flow configurations of precooling process of wet side air flow by own wet channel. In Fig. 3(a), wet channel air fluid is pre-cooled via another dry channel on the other side of wet channel. However, in Fig. 3(b), a partial of air fluid of the main dry channel (which has been cooled) is returned into the wet channel at the end of the dry channel which is termed regenerative heat and mass exchanger. Maisotsenko technology causes reduction of dry side outlet air temperature under we-bulb temperature (until dew-point temperature) without adding any moisture. All other problems of direct contact evaporators and also former kind of indirect evaporative techniques have been solved in this novel form of indirect evaporative system. Psychometric chart related to Fig. 3(b) is illustrated in Fig. 3(c) form [13]. Fig. 4 [from4] shows a perforation type of working channel in M-cycle heat exchanger which causes reduction of pressure drop across the exhaust channel [5].

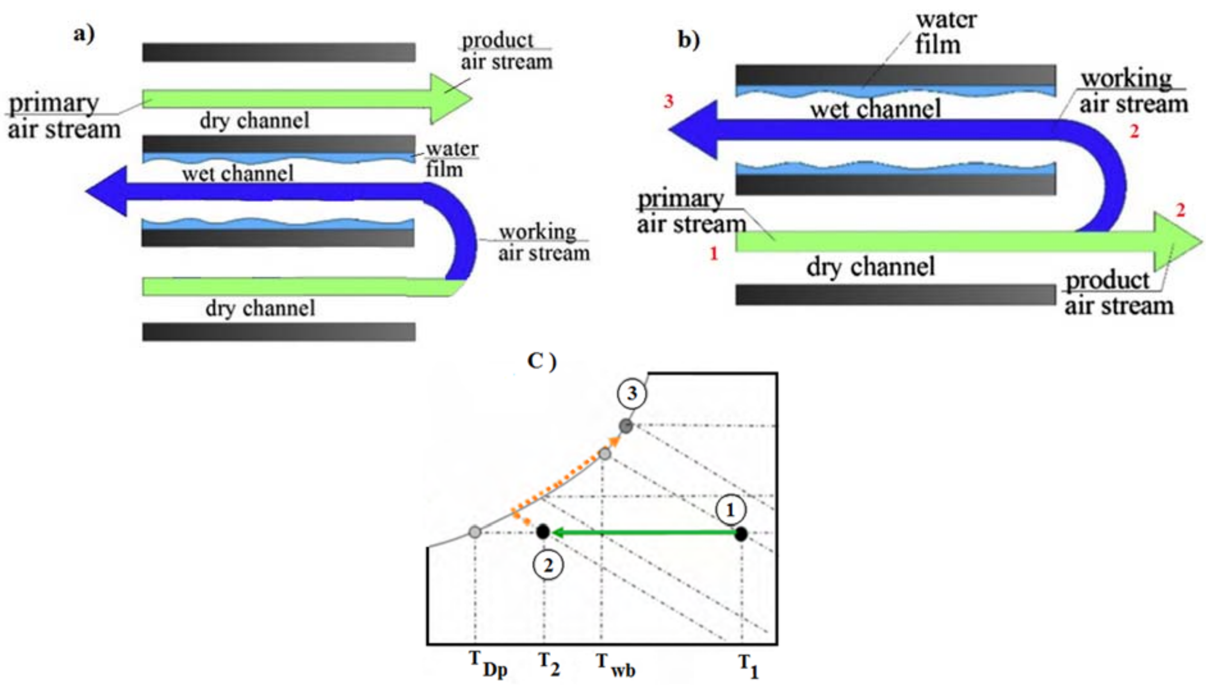


Fig. 3. Two main counter flow configurations of indirect evaporative based on main idea of M-cycle [6]

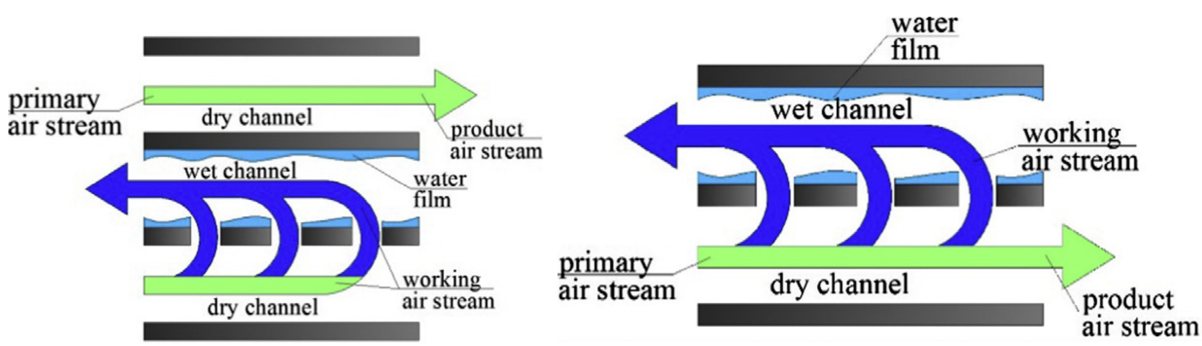


Fig. 4. Perforation type of counter flow configuration of indirect evaporative based on M-cycle [6]

Nonetheless, counter flow configuration of M-cycle air coolers was not commercialized for many years. Indeed, pure counter-flow configuration in a plate heat exchanger cannot be fabricated because of the geometry of the plates with air entering and evacuating on the same direction [5&7]. Hence, cross flow arrangement was presented which is easy to manufacture as a unit (see Fig. 5) and is produced by Coolerado Corporation [8] (see Fig. 6 [9] in which "1" is the primary air flow in dry channel, "2" is the working air stream which at first flow along the dry duct then it is delivered to the wet-channel, and "3" is the secondary air wet channel [4]).

As the working principle of M-cycle IEC, it could theoretically overcome disadvantages 1, 2 and 4 of DEC [10-12]. Hence, the main advantages of the novel M-cycle air conditioning systems can be described as below.

1. M-cycle deliver cooling air temperature lower than either direct or indirect evaporative cooling systems without adding moisture [14] until dew point (lower than wet-bulb).
2. M-cycle technology uses much less energy than conventional compression air-cooler [14]
3. There is no direct contact between water and product air which removes the possibility of health issues from contaminated water.
4. M-cycle's cooling capacity enhances with increment of incoming air temperature [14]
5. M-cycle is free from CFC and can use approximately 50% less water [14].
6. It has a very competitive initial cost and its operating cost is in half [14].
7. M-cycle required electrical power can be produced by a compact solar panel [15].
8. M-cycle coolers Provides healthier indoor atmosphere by incorporating 100% fresh air.

Nonetheless, some researchers believe that M-cycle is not appropriate for humid climates yet and it should be combined with desiccant systems in order to get higher efficiency. Thus, combination of liquid-desiccant or solid desiccant with M-cycle has been recently argued. These systems comprises of two processes: moisture removal by dehumidifier and sensible heat removal by M-cycle. Obviously, the effectiveness of first stage impresses on the working quality of the second stage. Required power to drive desiccant system can be obtained by low-grade heat sources such as solar energy [16-19].

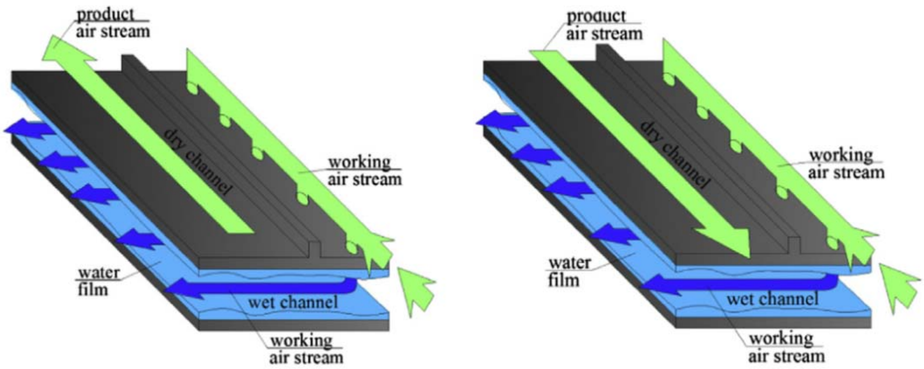


Fig. 5. Cross flow arrangement of M-cycle [6]

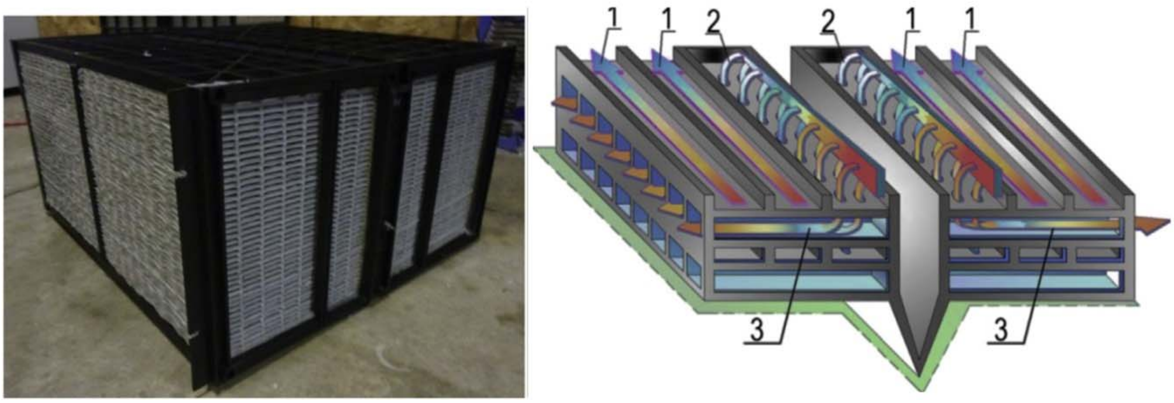


Fig. 6. Cross flow M-cycle HMX of Coolerado Corporation from [9]

Despite the fact that the first idea of M-cycle evaporative was developed around 1980, this type of M-cycle air coolers started to be developed and commercialized in this century [10-12]. Although Maisotsenko cycle is used in different applications such as gas turbine [20-31], cooling towers [32-35] and electronic cooling, [36&37] this paper only focuses on the employment of M-cycle on air conditioning systems. Mahmood et al. [7] has presented the only review paper in any reputed journal on M-cycle systems in which the application of Maisotsenko cycle has been classified into 3 main parts (HVAC, cooling tower, gas turbine) and each part has been fundamentally discussed. However, present paper discusses only on Maisotsenko air conditioning systems in order to provide extra detail on this major application of M-cycle. Different evaluation methods of M-cycle air conditioning systems including analytical solution, numerical simulation, statistical design methods and experimental techniques which have been proposed by researchers are organized and discussed in this research. It has been tried to provide an evolutionary viewpoint for analytical solutions of M-cycle. Thus, analytical solutions were reorganized with unique abbreviations in order to become more understandable and comparable with each other. All M-cycle parameters (which have been analyzed via numerical or experimental ways) are systematized and then a comprehensive-compact view of obtained results is presented. Finally, the status of current M-cycle industry and the future research direction for M-cycle technology are discussed.

2. Evaluation Methods of indirect evaporative coolers

Generally, evaluation techniques of indirect M-cycle evaporative can be classified in four main groups as shown in Fig. 7. Eight main analytical solutions of M-cycle are reorganized with unique abbreviations in order to become comparable with each other. The relationship between analytical solutions is discovered and discussed. Actually, it is tried to provide an evolutionary viewpoint for analytical solutions of M-cycle. M-cycle characteristics which have been evaluated by numerical or experimental techniques are discussed by some graphical representations.

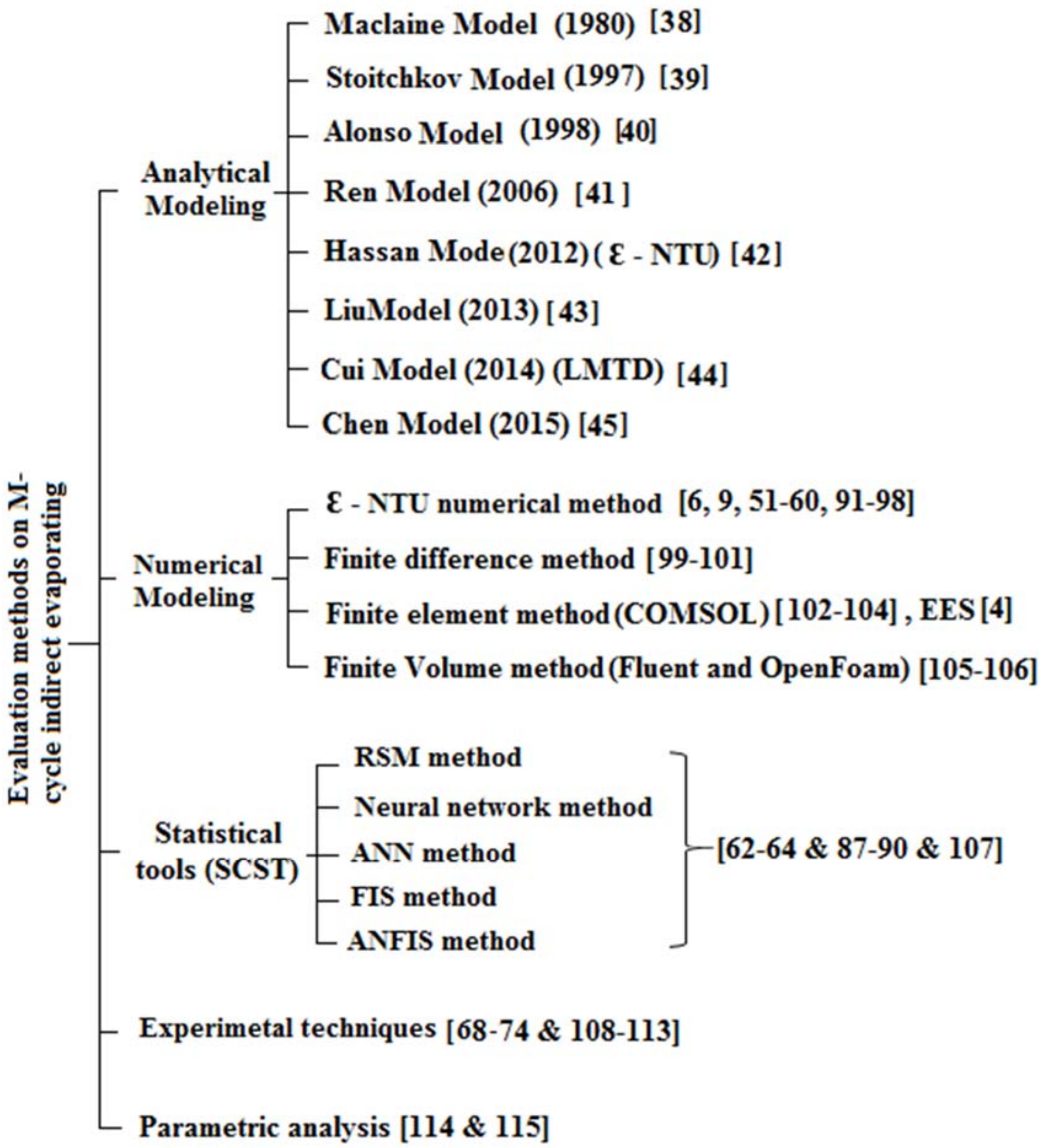


Fig. 7. Evaluation methods of M-cycle air conditioning systems

3. Analytical approaches

The core of the M-cycle IEC modeling is to model/analyze heat and mass transfer on wet surface. Although some researches (Mickley [46] at 1949 and Pescod [47] at 1968) provided some basic theories on wet surface heat exchangers, it can be said that Makline-Cross theory [38] at 1980 is the leading general modeling of wet surface heat exchangers and its application to regenerative evaporative cooling. The main difference among analytical models is related to their assumptions. Indeed, some researchers prefer to simplify their modeling by considering some conditions while other researchers would not like to sacrifice accuracy for simplicity of solution. Nonetheless, some of them have unique or new formulations (compared to the Maclaine model) which will be discussed with more detail of their formulations. Table 1 shows assumptions used in all analytical models reviewed, in which assumptions used in each modeling have been identified by “*” mark.

Lewis factor (assumption 7 in Table 1) plays a key role in evaluation of heat and mass transfer between liquids and gases. Lewis factor (Le) is dimensionless number and generally is defined as the ratio of thermal diffusivity to mass diffusivity. Lewis [48] stated that the Le is approximately equal with “1” for air/water mixtures. Although according to [49 & 50] the proof given by Lewis was not strictly correct, this assumption has been used in most analytical models.

$$\mathrm{Le} = \frac{\alpha}{\beta c_{wa}} = 1 \qquad (1)$$

Thus: $\alpha = \beta\, c_{wa}$ (2)

Where c_{wa} is the specific heat of moist air and $c_{wa} = 1.006 + 1.86w$. Eq. (2) creates a relationship between convective heat transfer coefficient (α) and convective mass transfer coefficient (β).

It should be mentioned that, each analytical model may focus on particular geometry (configuration) of direct contact heat exchangers. Thermal-flow configuration of each model has been provided in Table2.

Table1. Main assumptions of analytical modeling of indirect evaporative systems

No	Assumptions	Maclaine [38]	Stoitchkov[39]	Alonso [40]	Ren [41]	Hassan [42]	Liu [43]	Cui [44]
1	Zero fluid thermal and moisture diffusivity in the flow directions	*	*	*	*	*	*	*
2	No heat transfer to the surrounding occurs	*	*	*	*	*	*	*
3	The passage walls are impervious to mass transfer	*	*	*	*	*	*	*
4	Pressures and mass flow rates are constant and uniform for both streams	*	*	*	*	*	*	*
5	c, h, α and U are constant and uniform;	*	*	*	*	-	-	*
6	The passage geometry is uniform throughout the heat exchanger	*	*	*	-	-	-	*
7	The Lewis relation is satisfied (Eq. (1))	*	*	-	-	*	*	*
8	The specific enthalpy of moist air h_{wa} is a linear function of T_{wa} and w_{wa}, thus $h_{wa} = a + c_{wa}T_{wa} + h_g w_{wa}$	*	-	-	-	-	-	-
9	The evaporating water film is stationary and continuously replenished at its surface with water at the same temperature.	*	-	-	-	-	-	-
10	The humidity ratio w_f of the air in equilibrium with the water surface is a linear function of the water surface temperature T_f so that the model saturation line is given by $w_f = d + eT_f$	*	-	-	*	-	-	-
11	Other type of 9: The water is distributed uniformly all over the channels and wets all the surface	*	-	-	-	-	*	*
12	Interface temperature is assumed to be the bulk water temperature	-	-	-	*	-	-	-
13	The interface between moist air and water film is saturated at the wate film temperature T_f	-	-	-	-	*	-	*
14	Air flow is laminar and also fully developed	-	-	-	-	-	-	*

Table2. Heat exchanger type of each analytical model

Models	Geometry of Heat exchanger
Maclaine [38]	Parallel and counter flow heat exchanger
Stoitchkov [39]	Cross flow plate heat exchanger
Alonso [40]	Cross flow heat exchanger
Ren [41]	Parallel and counter flow heat exchanger
Hassan [42]	Counter/parallel and usable for cross flow
Liu [43]	Counter flow heat exchanger
Cui [44]	Counter flow and expandable to cross flow
Chen [45]	Counter flow heat exchanger

3.1. Maclaine-Cross Model

Maclaine and Banks [38] presented a general theory of wet surface heat exchanger and its application to regenerative evaporative cooling. They proposed a liner approximate model of wet surface heat exchanger by analogizing with dry surface heat exchanger [45]. Maclaine method proposed a linear approximate and graphical representation which can be employed to evaluate the wet surface heat exchanger effectiveness. A general view of wet surface heat exchanger is shown in Fig. 8.

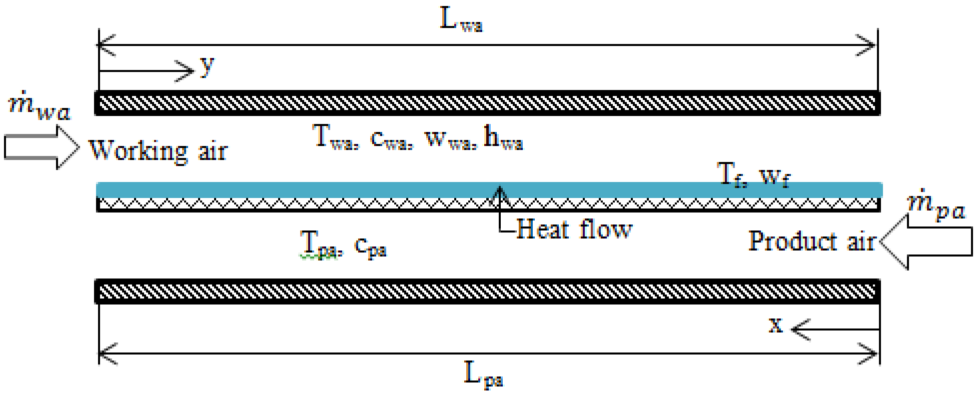


Fig. 8. A general view of wet surface heat exchanger (redrawn modified from [38])

Maclaine model is based on eight equations which four of them are based on assumptions (Table 1) and rest of them are based on conservation of energy or mass as below.

Conservation of energy for product air stream:

$$UA\,(T_f - T_{pa}) = \dot{m}_{pa} c_{pa} \frac{\partial T_{pa}}{\partial x} L_{pa} \quad (3)$$

Energy balance between two channels:

$$\alpha_{wa} A_{wa}\,(T_{wa} - T_f) + \beta A_{wa} h_{fg}(w_{wa} - w_f) + UA(T_{pa} - T_f) = 0 \quad (4)$$

Conservation of water vapor in the moist air stream

$$\dot{m}_{pa}\, L_{wa} \frac{\partial w_{wa}}{\partial y} = \beta\, A_{wa}\,(w_f - w_{pa}) \quad (5)$$

Conservation of energy for working air:

$$\dot{m}_{wa} \frac{\partial h_{wa}}{\partial y} L_{wa} = \alpha_{wa} A_{wa}\,(T_f - T_{wa}) + \beta\, A_{wa}\, h_g\,(w_f - w_{pa}) \quad (6)$$

According to assumption 7, 8 and 10:

$\alpha = \beta\, c_{wa}$ (7)

$h_{wa} = a + c_{wa}T_{wa} + h_g w_{wa}$ (8)

$w_f = d + eT_f$ (9)

The approximation psychometric equation is written as below in which $/$ indicates the adiabatic saturation state defined using the linearized or model saturation line of assumption 10.

$w_{wa} = w_{wa}^{/} + (T_{wa}^{/} - T_{wa})\,\frac{c_{wa}}{h_{fg}}$ (10)

Inlet working air temperature and humidity ratio and also product air inlet temperature are constant and known as the boundary conditions of solving aforesaid eight equations.

Regarding to Eq. (9), although the actual saturation line in real psychometric chart is not linear, Maclaine assumed a linear behaviour (Eq. (9)) for saturation line (see Fig. 9). Indeed, if the constants "d" and "e" in Eq. (9) are chosen to give an approximate least square fit to the actual saturation line over the range of water surface temperature, this model can present actual performance. The recommended values for "d" and "e" by Maclaine are as below.

$e = \frac{W_{f,max} - W_{f,min}}{T_{f,max} - T_{f,min}}$ (11)

$d = \frac{2(W_{f,min} + W_{f,mean}) - W_{f,max}}{3 - eT_{f,min}}$ (12)

Where $T_{f,\ min}$ and $T_{f,\ max}$ are the estimates of the minimum and maximum water surface temperature. $T_{f,\ mean}$ is the average of $T_{f,\ min}$ and $T_{f,\ max}$.

See Fig. 9 to understand graphical concepts of these values. Indeed, the estimated values of T_{fmin}, $T_{f\ max}$ and $T_{f\ mean}$ are used to plot the points W_{min}, W_{max} and W_f on the actual saturation line. The straight line joining W_{min} and W_{max} is drawn, then, parallel to this line and two thirds of the way from it towards point $W_{\ mean}$, another line is drawn. This is Maclaine model saturation line [39]. Maclaine solved these eight equations by some methods found in literature in order to determine the thermal performance of characteristics of indirect evaporative exchanger.

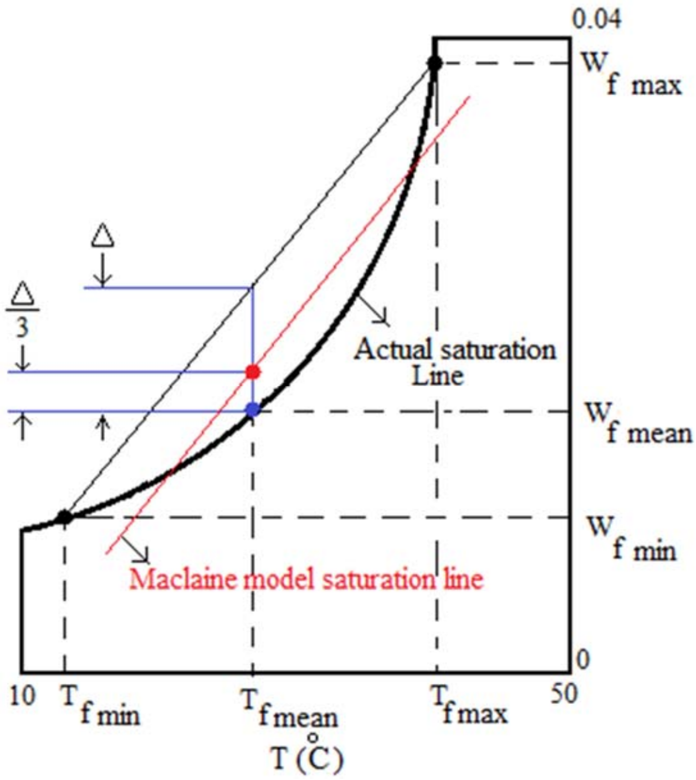


Fig. 9. Maclaine Model saturation line (modified redrawn from [38])

3.2. Stoitchkov Model

Stoitchkov [39] indicated three deficiencies for Maclaine model as listed below:

- Maclaine did not show how the mean surface temperature should be estimated.
- The values of total to sensible heat ratio for different mean surface temperatures given in a table are calculated for a barometric pressure of 101325 Pa. Hence, for other pressures, the outcomes will have a source of error.
- In Maclaine model, the evaporating water film is stationary and continuously replenished at with water at the same temperature (assumption 9 in Table 1). However, in real heat exchanger, the evaporated water is much less than the mass flow rate of sprayed water which causes reduction of effectiveness of the wet surface heat exchanger compared to the Maclaine assumption.

Hence, Stoitchkov further improved the Maclaine model. However, this model focuses on cross flow plate heat exchanger. Stoitchkov and Dimitrov improved Maclaine method by following considerations:

- Determination of the mean surface temperature for any defined thermal and geometrical specification.
- Presenting a correlation (approach) to estimate the barometric pressure [39].

3.3. Alonso Model

Alonso provided a user-friendly simplified model by providing an equivalent water temperature [45] in Maclaine solution. The model obtained based on the models developed by Maclaine [38].

3.4. Ren Model

Ren and Yang [41] believed that previous simplified models sacrifice accuracy for simplicity of solution. Most simplified models assume a unity Lewis factor and neglect the water losses due to evaporation. They indicated following deficiencies for all previous modeling. All these deficiencies were applied to simplify the former models.

- Moisture content of air has been considered a linear function of the water surface temperature.
- Lewis factor is satisfied
- The evaporating water film was stationary and continuously replenished with water.
- Wet surface is assumed completely wetted.

Hence, they expanded an analytical model for indirect evaporative cooler with parallel/counter flow configuration with variable surface wettability and Lewis factor [45]. Their model is sophisticated which is due to non-unity Lewis factor, surface wettability, varying spray water temperature and spray water enthalpy change [44]. Briefly, Ren model consists of below characteristics which can’t be found in former analytical modes.

- Incomplete surface wetting condition: The wall surface cannot be entirely wetted because of low distance between surface and high value of sprayed water which creates more tension. This leads to reduction of mass transfer area.
- Non-unity Lewis factor: Lewis factor is not necessary equal with unity even for entirely wetted surface [7].
- Consideration of spray water evaporation
- Consideration of spray water temperature variation
- Consideration of spray water enthalpy change through the heat exchanger

Nonetheless, humidity ratio is steel assumed to have a linear relationship with water surface temperature. However, the error of this condition can be minimized by choosing suitable values of “e” and “d” in Eq. (9). Assumptions of this model can be seen in Table 1. The effect of condition 12 (in Table 1) is minor because of very large heat transfer coefficient between water film and air-water interface [8, 50].

After development of model, Ren compared the performance of four different configurations of indirect evaporative air cooler as shown in Fig. 10. The results evaluated by Ren model reviled higher performance for case “a” compared to the three other cases. However, with negligible spray water flow rate and complete wetting surface, case “a” and case “b” showed the same performance.

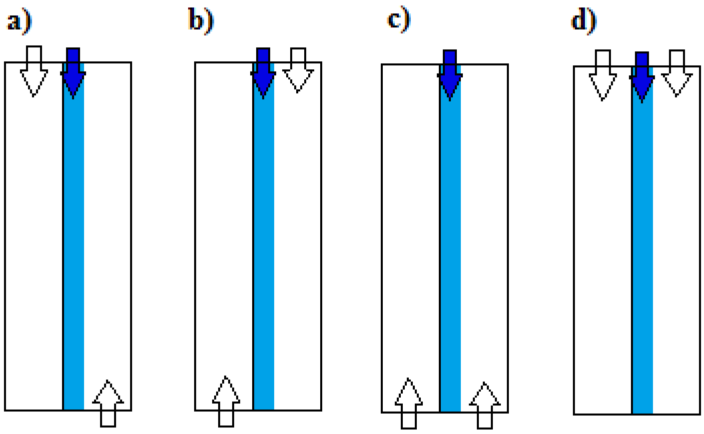


Fig. 10. Different arrangements evaluated by Ren analytical model [41]

3.5. Cui Model (LMTD)

Conventional LMTD method is a well-known method of evaluation of heat exchangers which deal with sensible heat transfer. The final aim in this technique is finding a correlation for total Q as $Q = A\ U\Delta T_{LMTD}$ in which the value of ΔT_{LMTD} is evaluated only by inlet and outlet temperatures. LMTD is appropriate for investigations in which temperature distribution is not considered as a main factor. In other words, LMTD works based on inlet/outlet, surface and some other bulk characteristics of heat exchangers. Contrary to numerical methods, this technique is neither cumbersome nor high time required. Nonetheless, conventional LMTD method should be modified in order to become usable in indirect evaporative systems. Indeed, latent heat transfer is an intrinsic feature of indirect evaporators which has not been applied in basic format of LMTD. To this reason, Cui et al. [44] presented a modified LMTD method to analyze counter flow indirect evaporative heat exchangers. This method provides better precision if the temperature variation in the heat exchanger is not linear and the maximum temperature difference at one end of the heat exchanger is higher than twice the temperature difference at the other end of the heat exchanger [44].

3.5.1 Assumptions of Cui method (LMTD)

As briefly stated in Table 1, the assumptions of this method are as below.

1) The distribution of water on wet surface is uniform and even.
2) Heat and mass transfer occur in direction normal to the air flow.
3) Air flow is laminar and also fully developed and the system is well insulated.
4) Water fluid in the wet channel behaves as a thin static film because the water flow rate by convection is negligible (heat and mass transfer between water film and plate is assumed to be occurred only by conduction).
5) Thin moist air layer at the water-air interface is saturated at the water film temperature.

6) It is reasonable to assume that the enthalpy difference between two points has a linear function with wet bulb temperature difference for small operating range of temperatures. Indeed, on Psychrometric chart, the wet-bulb temperature lines are nearly parallel with constant enthalpy lines for small operating range of temperatures [44 & 45]. Hence, the parameter ξ is defined as the ratio between the change of enthalpy and the change of wet-bulb temperature ($\xi = \frac{\Delta h}{\Delta T_{wb}}$) and is estimated by input and output condition of wet channel ($\Delta h = \xi\ \Delta T_{wb}$) .

7) Lewis factor is unity

3.5.2 Mathematical development

Fig. 11 shows the defined computational element which comprises half of the product channel and working channel (due to geometrical symmetry) of a plate type indirect evaporative cooler [44]. Where U_{pa}, T_{pa}, T_f, T_{wa}, h_{fg} and α are overall heat transfer coefficient of primary air, temperature of primary air, temperature of water film, temperature of secondary air, latent heat of water evaporation and convection heat transfer coefficient respectively. δ_p, k_p, δ_w, k_w are thickness of plate, thermal conductivity of plate, thickness of water-film and thermal conductivity of water respectively.

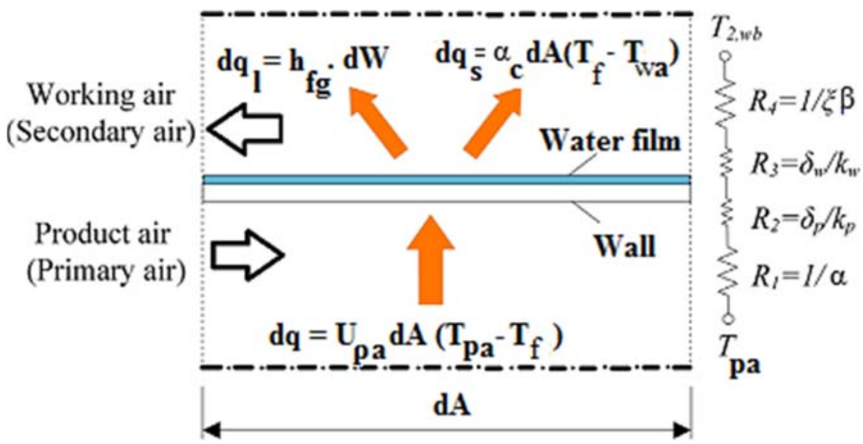


Fig. 11 One dimensional cross flow indirect heat exchanger [modified from44]

According to conservation principle of energy for primary air, dry side heat transfer rate can be calculated from Eq. (13).

$dq = -\ \dot{m}_{pa}\ c_{pa}\ dT_{pa} \xrightarrow{\text{yields}} dT_{pa} = -\frac{dq}{\dot{m}_{pa}\ c_{pa}}$ (13)

Besides, based on conservation principle of energy for working air and assumption "6", wet side heat transfer rate can be calculated from Eq. (14).

$dq = -\ \dot{m}_{wa} dh_{wa} \approx -\ \dot{m}_{wa}\ \xi\ dT_{wa,wb} \xrightarrow{\text{yields}} dT_{wa,wb} = -\frac{dq}{\dot{m}_{wa}\ \xi}$ (14)

Subtracting Eq.(14) from Eq.(13) gives Eq.(15).

$d(T_{pa} - dT_{wa,wb}) = -\left(\frac{1}{\dot{m}_{pa}\ c_{pa}} - \frac{1}{\dot{m}_{wa}\ \xi}\right) dq$ (15)

Eq. (15) can't be integrated unless a correlation is found for dq. Hence, attempts are made to provide a correlation for dq as below.

Obviously, heat transfer rate between a fluid flow and a plate can be evaluated via Newton's Law of Cooling too. Thus, heat transfer on the dry side of heat exchanger (which is only sensible and occurs between product air and water-film) is evaluated by.

$dq = U_{pa}\ dA\ (T_{pa} - T_f)$ (16)

$U_{pa} = \frac{1}{\frac{1}{\alpha} + \frac{\delta_p}{k_p} + \frac{\delta_f}{k_f}}$ (17)

In wet side of heat exchanger, there are two types of heat transfer as below:

$dq_{sensible} = \alpha\ dA\ (T_f - T_{wa})$ (18)

$dq_{latent} = h_{fg}\ .\ dW$ (19)

dq_{latent} is due to water evaporation in which dW is mass transfer rate between the water film and moist air and is calculated from Eq. (20) (β is mass transfer coefficient and w is humidity ratio).

$dW = \beta\ dA\ (w_f - w_{wa})$ (20)

Hence, latent heat transfer is calculated by:

$dq_{latent} = h_{fg}\ \beta\ dA\ (w_f - w_{wa})$ (21)

Total heat transfer rate on wet side is sum of the sensible heat and latent heat as below.

$dq = dq_{sensible} + dq_{latent} = \alpha\ dA\ (T_f - T_{wa}) + h_{fg}\beta\ dA\ (w_f - w_{wa})$ (22)

According to Eq. (2) (from assumption "7"), Eq. (22) can be rewritten as below.

$$dq = \beta c_{wa} dA(T_f - T_{wa}) + h_{fg}\ \beta\ dA(w_f - w_{wa}) = \beta\ dA\ [(c_{wa}T_f + h_{fg}w_f - (c_{wa}T_{wa} + h_{fg}w_{wa})]$$

$$\xrightarrow{h=cT+h_{fg}w} dq = \beta\ dA[h_s(T_f) - h_{wa}] \qquad (23)$$

$h_s(T_f)$ is saturation enthalpy of air fluid in water film temperature. According to Eq. (23), it can be said that, the driving force of total heat transfer in wet channel is calculated by enthalpy variation between the saturated air at water surface and the main moist air stream [11]. Eq. (23) can be rewritten based on assumption "6" ($\Delta h = \xi\ \Delta T_{wb}$) as follow (this is why modified thermal resistance in Fig. 10 is $1/\xi\ \beta$).

$dq = \xi\ \beta\ dA\ (T_f - T_{wa,\ wb}) = U_{wa}\ dA\ (T_f - T_{wa,\ wb})$ (24)

U_{wa} is modified overall heat transfer coefficient in wet channel. If Eq. (16) is rearranged based on T_f and then is substituted for T_f in Eq. (24), yields,

$$dq = \frac{U_{pa}\ U_{wa}}{U_{pa} + U_{wa}}\ dA\ (T_{pa} - T_{wa,wb}) = U\ dA\ (T_{pa} - T_{wa,wb}) \qquad (25)$$

U is the modified overall heat transfer coefficient which is related to both dry and wet channel.

$$U = \frac{1}{\frac{1}{\alpha_{pa}} + \frac{\delta_p}{k_p} + \frac{\delta_w}{k_w} + \frac{1}{\xi\ \beta}} \qquad (26)$$

Now, Eq. (25) can be substituted for dq in Eq. (17) which yields Eq. (27).

$$\frac{d(T_{pa} - dTwa,wb)}{T_{pa} - T_{wa,wb}} = -U\left(\frac{1}{\dot{m}_{wa}\ c_{wa}} - \frac{1}{\dot{m}_{wa}\ \xi}\right) dA \qquad (27)$$

Eq. (27) can now be integrated over the entire surface as below.

$$\int_0^A \frac{(T_{pa} - T_{wa,wb})}{T_{wa} - T_{wa,wb}} = -\int_0^A U \left(\frac{1}{\dot{m}_{wa}c_{wa}} - \frac{1}{\dot{m}_{wa}\xi}\right) dA \qquad (28)$$

$$\mathrm{Ln}\ (T_{pa} - T_{wa,wb})\Big|_0^A = -U \left(\frac{1}{\dot{m}_{wa}c_{wa}} - \frac{1}{\dot{m}_{wa}\xi}\right) A\Big|_0^A \qquad (29)$$

$$\mathrm{Ln}\,\frac{(T_{pa}-T_{wa,wb})_A}{(T_{pa}-T_{wa,wb})_0} = -\,U\Big(\frac{1}{\dot{m}_{sa}c_{sa}} - \frac{1}{\dot{m}_{sa}\xi}\Big)A \qquad (30)$$

For a counter flow IEHX for example, Eq. (30) can be rewritten as below.

$$\mathrm{Ln}\,\frac{T_{pa,outlet}-T_{wa,wb,inlet}}{T_{pa,inlet}-T_{wa,wb,outlet}} = -\,U\Big(\frac{1}{\dot{m}_{wa}c_{wa}} - \frac{1}{\dot{m}_{wa}\xi}\Big)A \qquad (31)$$

Integrating Eq. (13) and Eq. (14) over the entire length of channel yields:

$$Q = \dot{m}_{pa}\,c_{pa}\,(T_{pa,inlet} - T_{pa,outlet}) \xrightarrow{\text{yields}} \frac{1}{\dot{m}_{pa}c_{pa}} = \frac{Q}{T_{pa,inlet} - T_{pa,outlet}} \qquad (32)$$

$$Q = \dot{m}_{wa}\,\xi\,(T_{wa,wb,outlet} - T_{wa,wb,inlet}) \xrightarrow{\text{yields}} \frac{1}{\dot{m}_{wa}\xi} = \frac{Q}{(T_{wa,wb,outlet} - T_{sa,wb,inlet}} \qquad (33)$$

Substituting Eq. (32) and Eq. (33) in Eq. (31) and after some simplification yields:

$$Q = U\,A\,\frac{(T_{pa,inlet}-T_{wa,wb,outlet})-(T_{pa,outlet}-T_{wa,wb,inlet})}{\mathrm{Ln}\,(\frac{T_{pa,inlet}-T_{wa,wb,outlet}}{T_{pa,outlet}-T_{wa,wb,inlet}})} \qquad (34)$$

Eq. (34) is the final format of heat transfer rate of a counter flow indirect heat exchanger based on LMTD method (comparable with conventional LMDT method) in which:

$$\mathrm{LMTD} = \frac{(T_{pa,inlet}-T_{wa,wb,outlet})-(T_{pa,outlet}-T_{wa,wb,inlet})}{\mathrm{Ln}\,(\frac{T_{pa,inlet}-T_{wa,wb,outlet}}{T_{pa,outlet}-T_{wa,wb,inlet}})} \qquad (35)$$

3.6. Ɛ - NTU Model (Hassan [42])

Conventional Ɛ-NTU is one of the well-known methods for solving heat transfer problems for sensible heat exchangers. Hasan [42] presented a modified version of this technique which can be used for indirect evaporative heat exchangers. Sensible format of this technique can’t be employed for indirect heat exchanger because of the existence of two gradients [42] including 1: temperature gradient between the air fluid in dry channel and water film and 2: enthalpy gradient between the saturated air-water film interface and the moist air. In the modified Ɛ-NTU method attempts are made to create a connection between temperature of dry side channel and “h” in the wet channel by a unique gradient. As mentioned before in Table 1, the assumptions of this model are listed as below.

1) The cooler is well insulated to the surrounding.

2) Thermal conduction in the wall through longitudinal is neglected.

3) Heat and mass transfer coefficient inside each passage are constant.

4) Lewis number is unity.

5) Interface between moist air and water film is saturated at the water film temperature (T_f).

3.6.1. Mathematical development of Ɛ - NTU Model

The same equations (16 to 23) form LMTD method are the beginning mathematical process in this method too. Hence, Eq. (23) is rewritten here.

$dq = \beta\, dA[h_s(T_f) - h_{wa}]$ (36)

$h_s(T_f)$ is saturation enthalpy of air fluid in water film temperature. Totally, it can be assumed that, there is a linear relation between air saturation temperature and its enthalpy (saturated enthalpy) as below.

$h_s(T) = aT + b \overset{so}{\Rightarrow} h_s(T_f) = aT_f + b$ (37)

Where $h_s(T)$ is saturated enthalpy of air at temperature of T and “a” is the slope of the saturation line (this assumption should not result in a significant error for small temperature ranges). Substituting Eq. (37) into Eq, (36) gives:

$dq = \beta\, dA(aT_f + b - h_{wa})$ (38)

Parameter T_f can be replaced with from Eq. (16).

$dq = \beta\, dA\left(a\left(T_{pa} - \frac{dq}{UdA}\right) + b - h_{wa}\right) \xrightarrow{yields} dq = \frac{aT_{pa} + b - h_{wa}}{\frac{a}{U} + \frac{1}{\beta}}\, dA$ (39)

It should be noted that, according to Eq. (37), the term ($aT_{pa} + b$) is $h_s(T_{pa})$ which means air saturated enthalpy at the primary air (dry side) temperature. Thus, Eq. (39) becomes

$dq = \frac{1}{\frac{a}{U} + \frac{1}{\beta}}\, dA\left[h_s\left(T_{pa}\right) - h_{wa}\right]$ (40)

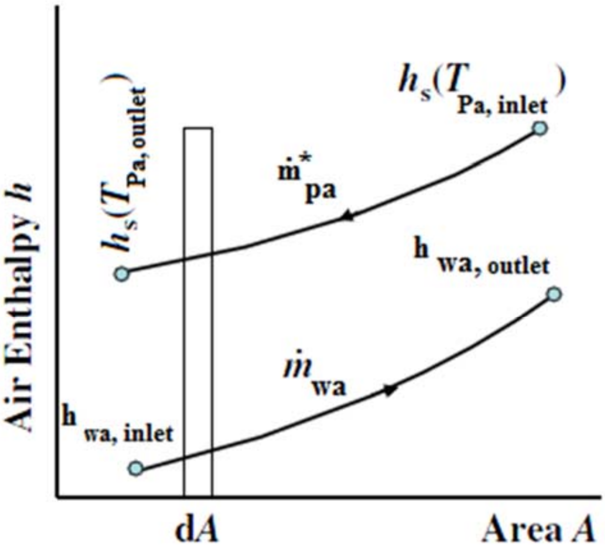
$\frac{1}{\frac{a}{U}+\frac{1}{\beta}}$ is new transfer coefficient. Eq. (40) connects T_{pa} in dry channel with h_{wa} in the wet channel by one equation based on unique enthalpy gradient ($[h_s(T_{pa}) - h_{wa}]$). It should be noted that, the modified enthalpy form $[h_s(T_{pa})]$ represents the thermal content of air fluid flow in dry side. The value of this modified enthalpy at the inlet and outlet of dry channel are $h_s(T_{pa,\,inlet})$ and $h_s(T_{pa,outlet})$. Generally, the amount of heat transfer rate through dry channel is $dq = \dot{m}c\,(T_{inlet} - T_{outlet}) = \dot{m}(h_{inlet} - h_{outlet})$. However, $h_s(T_{pa,\,inlet})$ and $h_s(T_{pa,outlet})$ are modified enthalpy (not real enthalpy). Hence, a modified mass transfer rate should be defined which can be found from heat balance on dry air side as below.

$$\dot{m}_{pa}\, c_{pa}\, (T_{pa,inlet} - T_{pa,outlet}) = \dot{m}^*_{pa}\, [h_s(T_{pa,\,inlet}) - h_s(T_{pa,outlet})] \quad (41)$$

Rearranging this equation:

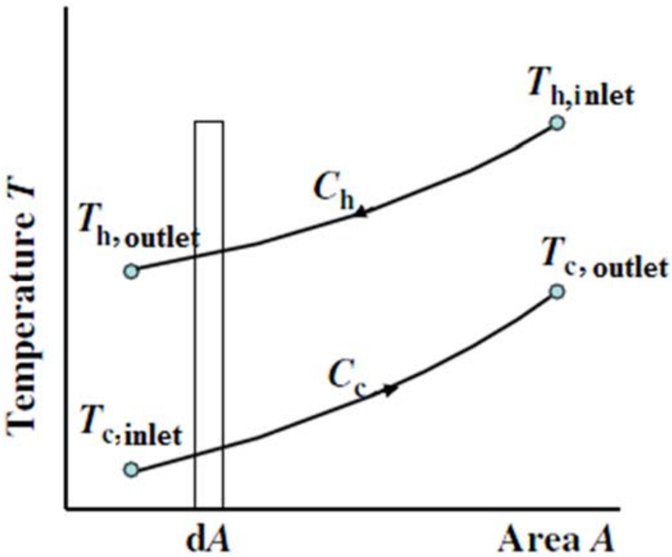
$$\dot{m}^*_{pa} = \dot{m}_{pa}\, c_{pa}\, \frac{T_{pa,inlet} - T_{pa,outlet}}{h_s(T_{pa,inlet}) - h_s(T_{pa,outlet})} = \frac{\dot{m}_{pa}\, c_{pa}}{a} \quad (42)$$

where “a” is the slope of the temperature-enthalpy saturation line. Conventional Ɛ-NTU method of sensible heat exchangers is based on (T, c and $\dot{m}$). The correlations of Ɛ-NTU method based on (T, c and $\dot{m}$) and its temperature profile is illustrated in Fig. (12). However, thermal profile of indirect evaporative was achieved based on modified enthalpy and modified mass flow rate as shown in Fig. (13a). It is mentioned that, these modified parameters can connect dry channel to wet channel via one equation and unique enthalpy gradient which is necessary in Ɛ-NTU evaluation method. And that is interest reason of this type of modified parameters. The Ɛ-NTU method can be applied for indirect heat exchanger if proper redefining of sensible parameters is made by compering Fig. 12 and Fig. 13a. These adjustments are shown in Fig. 13b. The expression for the effectiveness $Ɛ^*= f(NTU^*, C_r)$ takes similar forms as equations of sensible heat exchangers by replacing NTU by NTU^*. Heat transfer occurs from the fluid in dry side (hot air) to the fluid in wet side. Hassan method can be employed for different flow configurations in IEC coolers (regenerative, counter and parallel flow). Iteration is not necessary for counter and parallel flow configuration. However, for the regenerative configuration, iterations are needed because the wet side inlet air temperature is equal to the dry side outlet temperature which is unknown [42].



$$C_c^* = \dot{m}_{wa}, \quad C_h^* = \dot{m}_{pa}^*$$
$$C_{min} = \min(C_c^* \,\&\, C_h^*), \; C_{max} = \max(C_c^* \,\&\, C_h^*)$$
$$C_r = C_{min}/C_{max}$$
$$NTU^* = AU^*/C_{mic}, \quad U^* = \frac{1}{a(\frac{1}{\alpha}+\frac{\delta_p}{k_p}+\frac{\delta_f}{k_f})+\frac{1}{\beta}}$$
$$\varepsilon^* = f(NTU^*, C_r)$$
$$q_{max} = C_{min}\left[h_s(T_{pa,inlet}) - h_{wa,inlet}\right]$$
$$\varepsilon^* = \frac{q}{q_{max}}$$
$$q = \varepsilon^* C_{min}\left[h_s(T_{pa,inlet}) - h_{wa,inlet}\right]$$
$$\varepsilon^* = \frac{C_h^*\left[h_s(T_{pa,inlet}) - h_{pa,outlet}\right]}{C_{min}\left[h_s(T_{pa,inlet}) - h_{wa,inlet}\right]}$$
$$\varepsilon = \frac{C_h^*(h_{wa,outlet} - h_{wa,inlet})}{C_{min}\left[h_s(T_{pa,inlet}) - h_{wa,inlet}\right]}$$

Fig. 12. Temperature profile and Ɛ-NTU correlations in sensible heat exchangers [42]



$$C_c = \dot{m}_c c_c, \quad C_h = \dot{m}_h c_h$$
$$C_{min} = \min(C_c \,\&\, C_h), \; C_{max} = \max(C_c \,\&\, C_h)$$
$$C_r = C_{min}/C_{max}$$
$$NTU = AU/C_{mic}, \quad U = \frac{1}{\frac{1}{\alpha_h}+\frac{\delta}{k}+\frac{1}{\alpha_c}}$$
$$\varepsilon = \frac{q}{q_{max}}$$
$$q_{max} = C_{min}(T_{hi} - T_{co}), \; q = \varepsilon C_{min}(T_{hi} - T_{co})$$
$$\varepsilon = \frac{C_h(T_{hi} - T_{ho})}{C_{min}(T_{hi} - T_{ci})} \quad \text{Or } \varepsilon = \frac{C_c(T_{co} - T_{ci})}{C_{min}(T_{hi} - T_{ci})}$$
$$\varepsilon = f(NTU, C_r) \begin{cases} \varepsilon = \dfrac{1-\exp[-NTU(1+C_r)]}{1+C_r} & \textbf{Parallel} \\ \varepsilon = \dfrac{1-\exp[-NTU(1-C_r)]}{1-C_r\exp[-NTU(1-C_r)]} & \textbf{Counter} \\ \textbf{Ref [16]} & \textbf{Other arrangements} \end{cases}$$

Fig. 13. Temperature profile and modified Ɛ-NTU correlations in indirect heat exchangers [42]

3.7. Liu Model

Liu [43] stated two deficiencies for Hassan Ɛ-NTU model as below:

- Hassan model introduced a coefficient “a” to present the slope of the saturation line and did not discussed how to evaluate this coefficient.
- Hassan’s model has been validated only by one operation condition without discussing its verification with other operation condition.

Thus, Lui presented a simplified thermal modeling based on Ɛ-NTU method and then validated utilizing experimental data from the literature in a wide range of operating conditions. The model highlights several improvements over the previous simplified models, including the detailed procedure for UA value calculation for laminar and turbulent IEHX channel flows [43].

The initial formulations of this model are the same equations used in Hassan model. Eq. (16) and Eq. (23) from Hassan’s model are rewritten here.

$$dq = U_{pa}\, dA\, (T_{pa} - T_f) \qquad (43)$$

$$dq = \frac{\alpha_{wa}}{c_{wa}} dA[h_s(T_f) - h_{wa}] \qquad (44)$$

Both Eq. (43) and Eq. (44) have the same functional form. If the enthalpies in Eq. (44) could be expressed as a function of temperature only, this equation can be used as a part of series heat transfer path with equation 43 [43].

For moist air, the enthalpy is expresed as Eq. (45) in which c_{pa} is specific heat of moist air and is calculating with c_{pa} = 1.006 + 1.86W (Kj/Kg C) and h_{fg} is evaporation of water at 0 celsious. However, Liu approximated the enthalpy of moist air at its wet-bulb (saturation) temperature condition as seen in Eq. (46)). Liu used simmilar assumption for calculating of $h_s(T_f)$ which is enthalpy of saturated air-water interface layer (Eq. (47)).

$$h_{wa} = c_{pa}\, T + w\, h_{fg} \qquad (45)$$

$$h_{wa} = c_{pa}\, T_{wa}(wb) + w_{wa}(wb)\, h_{fg} \qquad (46)$$

$$h_s(T_f) = c_{pa}\, T_f + w_s(T_f)\, h_{fg} \qquad (47)$$

Difference between Eq. (46) and Eq. (47) shows that $[h_s(T_f) - h_{wa}]$ in Eq. (44) can be expresed a linear equation of T_f - $T_{wa}(wb)$ as shown below.

$$dq = \frac{\alpha_{wa}}{c_{wa}} dA[h_s(T_f) - h_{wa}] \approx \frac{\alpha_{wa}}{c_{wa}} KdA_s\,(T_f - T_{wa}(wb)) \quad (48)$$

where K is the slope of the enthalpy-saturation temperaturion curve. Eq. (48) reveales that the difference between water film temperature and working air wet-bulb temperature is the driving force for energy transfer from the water film to the working air stream [43]. If the Eq. (48) and Newton's Law of Cooling is compared, it is obvious that the term K $\frac{\alpha_{wa}}{c_{wa}}$ is analogous to the local heat transfer coefficient between water film and working air. Hence, as shown in Fig. 14 (modified version of Fig. 2 from [43]), total, overall thermal resistance and its corresponding heat flux between primary air and working air are presented as Eq. (49) and Eq. (50) respectively.

$$\frac{1}{U} = \frac{1}{\alpha_{pa}} + \frac{\delta_p}{k_p} + \frac{\delta_f}{k_f} + \frac{c_{wa}}{K\,\alpha_{wa}} \quad (49)$$

$$dq = UdA\,(T_{pa} - T_{wa}(wb)) \quad (50)$$

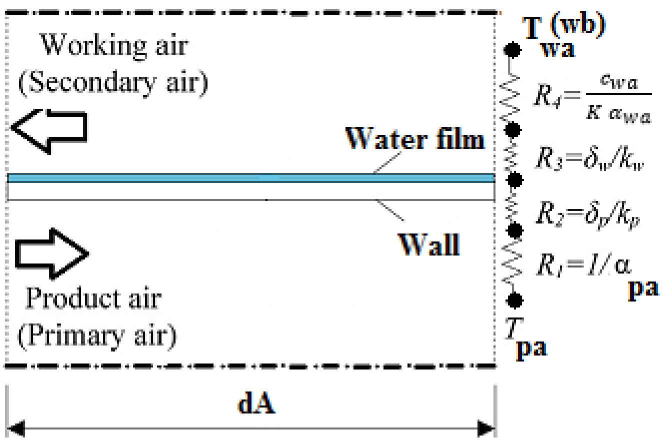


Fig. 14. Thermal resistance across the exchnager [43]

Based on conservation of energy for working air:

$q = \dot{m}_{wa}\ (h_{wa,outlet} - h_{wa,inlet})$ (51)

Parameter $\overline{K}$ (is diferent from K) is introduced and defined as the ratio of wet-bulb temperature difference between the wet side inlet and outlet and their enthalpy difference; Equation 51 can then be rewritten as:

$q = \dot{m}_{wa}\ \overline{K}\ (T^{wb}_{wa,outlet} - T^{wb}_{wa,inlet})$ (52)

Based on conservation of energy for primary air:

$q = \dot{m}_{pa}\ c_{pa}\ (T_{pa,inletr} - T_{Pa,outlet})$ (53)

Now, a modified ε-NTU method can be applied through equations 50, 52 and 53 as shown in Fig. 15. Indeed, modified C_c, C_h and other parameters of modified ε-NTU mehod is expressed by comparing equations 50, 52 and 53 with the sensible format of those equaions (for sensible heat exchnagers). The amount of α or other parametres for calculating U is definite and depends on flow regime etc. Hence, Liu [35] has discussed the method by which the value of U is evaluated for laminar or turbulante flow. Furthermore, determination of $\overline{K}$ has been presented as well by Liu for this model which can be found in [35].

For Sensible exchanger	For IEHX
$C_c = \dot{m}_c\ c_{pc}\ ,\ C_h = \dot{m}_h\ c_{ph}$	$C_c = \dot{m}_{wa}\ \overline{K}\ ,\ C_h = \dot{m}_{pa}\ c_{pa}$
$\frac{1}{U} = \frac{1}{\alpha\ (hot)} + \frac{\delta_p}{k_p} + \frac{1}{\alpha\ (cold)}$	$\frac{1}{U} = \frac{1}{\alpha_{pa}} + \frac{\delta_p}{k_p} + \frac{\delta_f}{k_f} + \frac{c_{wa}}{K\ \alpha_{wa}}$
$\mathbf{NTU} = \frac{\mathbf{A\ U}}{\mathbf{C_{min}}}$	$\varepsilon = f\,(NTU,\ C_r)$
$\mathbf{C_{min} = min\ (C_c\ and\ C_h)}$	$\mathbf{C_{max} = max(C_c\ and\ C_h)}$

Fig. 15. Liu modified ε-NTU model

3.8. Chen Model

Chen [45] believed that none of previous analytical investigations has consider the effect of condensation on the performance of IEC units. Indeed, in humid area condensation may occur in the fresh air side which causes reduction of the cooler performance [45]. Former studies has not applied this consideration because of two reasons: 1) the humidity of the fresh air is low (indirect evaporative cooler is usually used in dry regions) and 2) outdoor fresh air is used for both working air and primary air so that the plate surface temperature is higher than the dew point temperature of the air. Hence, Chen's model evaluates the performance of IEC form three viewpoints including non-condensation, totally condensation and partially condensation.

4. Numerical modelling

According to Fig. 14, M-cycle numerical modelling is classified into three main categories (based on analysis method) including Ɛ-NTU, finite element/difference/volume, Statistical Design methods. However, Statistical Design Tool can be indicated as a distinct method (as seen in Fig. 7) and in this paper is discussed as a separate part. An overall-compact view on investigated parameters of M-cycle via all numerical simulations can be seen in Fig. 14. Numerical simulations revealed that the work of M-cycle exchanger is dependent on the interaction of many important factors including aerodynamic, hydrodynamic, thermodynamic, structure and others [51-53]. Perforations decrease the pressure loss of air which may allow for a significant reduction in the unit's operation costs [51]. However, it has lower cooling capacity which is due to the warm air inlet wet channel through the entire length of heat exchanger [51].

4.1. Numerical Ɛ-NTU method

In most engineering applications, an engineer is interested in receiving bulk average values rather than variable distributions. For these cases, numerical Ɛ-NTU method is preferred to other numerical simulations methods.

Most numerical analysis of M-cycle exchanger has been performed by Ɛ-NTU technique. However, almost all Ɛ-NTU-based evaluations have been carried out by one group of authors Sergey Anisimov and Demis Pandelidis [52] who tried to find different thermal features of M-cycle with their own Ɛ-NTU modeling. Although the final formulations of the method have been presented in their publications, the references which have been cited for the origin of the formulations [74-77] are not online accessible.

A glimpse-view on the effect of different parameters on M-cycle characteristics (part A and B in Fig.14 which have been evaluated by numerical Ɛ-NTU method for cross-flow exchangers) is presented in Table 3. The signs "+" and "-"mean increment and decrement of that characteristics respectively. As can be seen in Table 3, increment of inlet humidity ratio or decrement of inlet air temperature reduces the cooling capacity of m-cycle air coolers which emphasize again the unsuitability of M-cycle coolers for humid and relatively cold climate conditions.

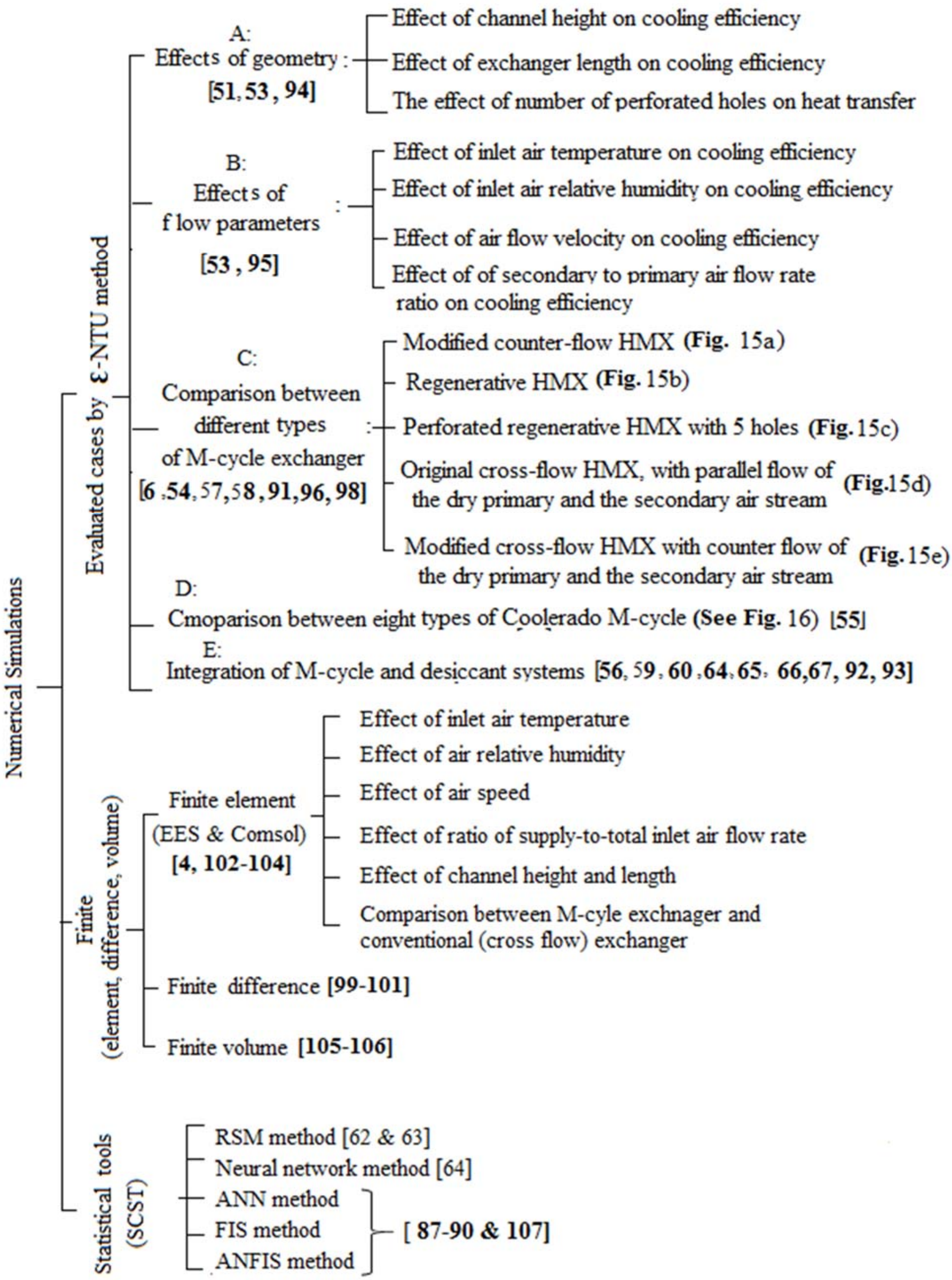


Fig. 14. A comprehensive compact view on investigated parameters of M-cycler via numerical simulations [51-63]

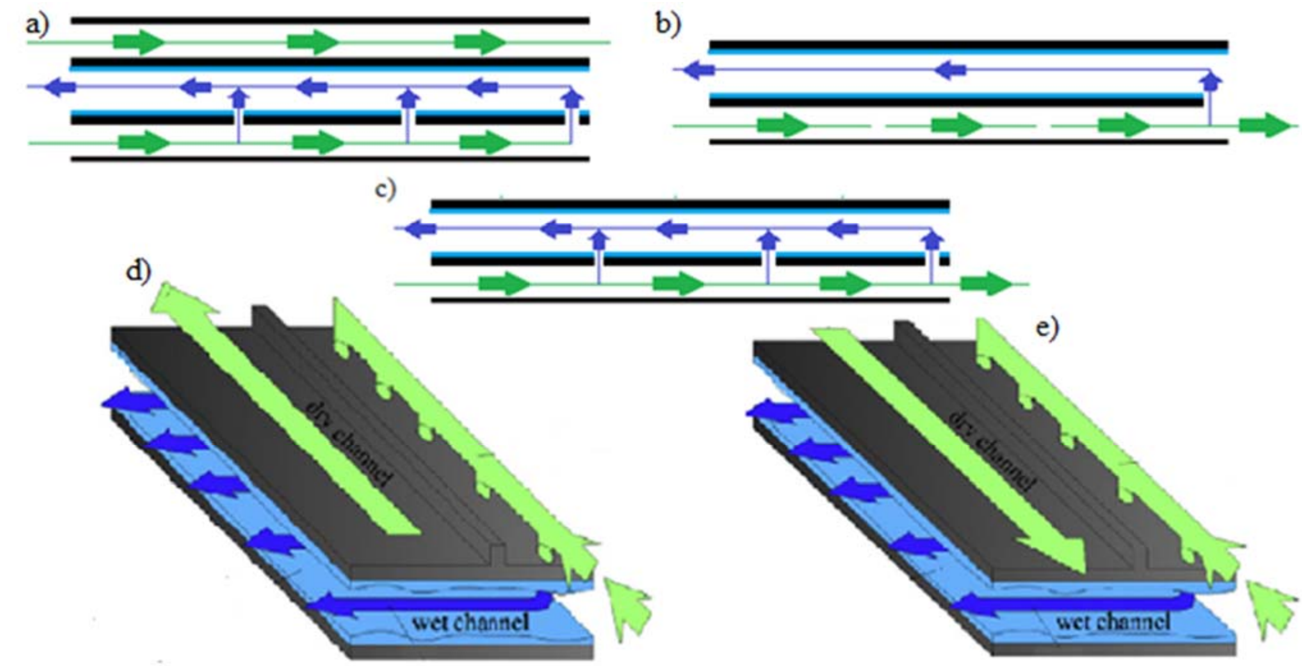


Fig. 15. Different types of M-cycle exchanger a) modified counter flow HMX, b) regenerative HMX c) perforated regenerative HMX d) cross flow HMX and e) modified cross flow HMX [6]

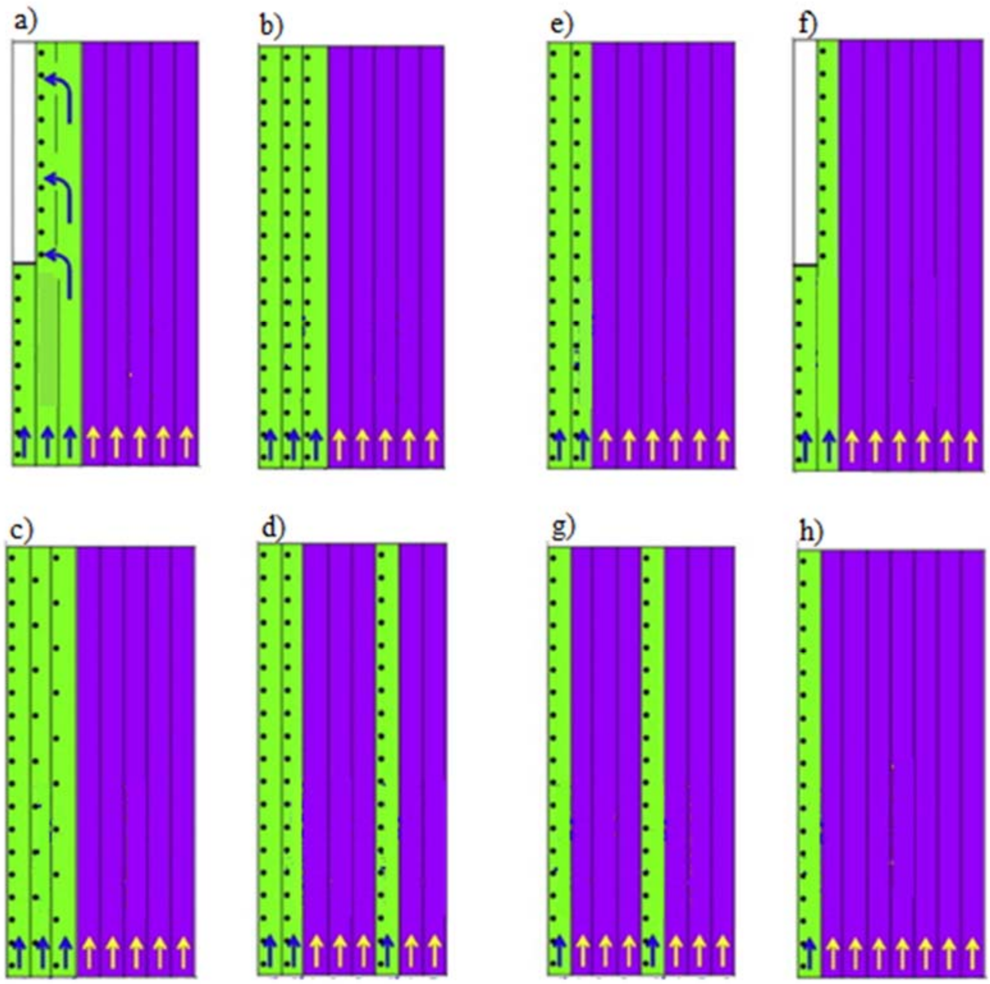


Fig. 16. Eight types of Coolerado corporation M-cycle HMX (See part D in Fig. 14) [55]

Table 3. Effect of different parameters on cross-flow exchanger characteristics which have been evaluated by numerical Ɛ-NTU method [51-58]

Parameters	**Cooler characteristics**		
	Output air temperature	Dew point effectiveness	Specific cooling capacity
Increment of air inlet temperature	+	+	+
Increment of inlet air relative humidity	+	+	-
Increment of inlet air humidity ratio	+	+	-
Increment of channel height	+	-	+
Increment of dry channel length	-	+	+
Increment of wet channel length	-	+	+
Increment of air flow velocity	+	-	-
Increment of secondary to primary air flow ratio	-	+	+
Increment of number of perforated holes	+	N	-

Regarding to part C in Fig. 14, the condition of comparison should be clearly stated because of the different structures of analysed exchangers. Comparison has been carried out under real operating condition [6] of each exchanger. All exchangers have the same dimensions, the same plate thickness and the same channel height. Nonetheless, they have different secondary to primary air ratio. According to [53] the best air ratio (which provides higher effectiveness) for cross flow exchangers equals 1. However, it is impossible to have the air ratio of 1 for regenerative exchangers (case "b" and "c" in Fig. 15) because of their construction. Indeed, all primary air streams from dry channel have to be delivered to wet side in order to have air ratio of 1 (but the cooling capacity of such would be zero). To that reason, the air ratio of case "b" and "c" was chosen around 0.3 which was suggested for these cases [13, 78-81]. For all working condition (different inlet air temperature or different relative humidity) the arrange (order) of cooling capacity is shown in Fig. 17.

(See Fig. 15)

Specific cooling capacity of case d > e > b > c > a

Wet-bulb effectiveness of case a > b > d > c > e

Output temperature of case e > c > d > b > a

Fig. 17. Comparison between different types M-cycle exchanger (see Fig. 15)

Different positions and arrangements of perforations in cross flow M-cycle (see Fig. 16) create different thermal characteristics of M-cycle. For each specific arrangement of perforation, changing of thermal-fluid condition causes creation of different output characteristics as well. However, maximum cooling capacity and wet-bulb effectiveness were obtained for case “h” (Fig. 16) and minimum values of said parameters were observed for case “a”.

4.2. Finite element/difference/volume method

Table 4 shows a compact view of the numerical M-cycle investigations which have been studied via finite element/difference/volume method.

Table. 4. Numerical studies via finite element/difference/volume method

Reference	year	Investigated method	Structure	Explanation
Zhang [4]	2011	Finite element	Cross flow	Effect of various parameters on cross-flow M-cycle performance
Cui [104]	2015	Finite element (Comsol)	Counter flow	Combination of indirect evaporative cooler and compression air cooler
Heidarianejad [100]	2015	Finite difference	Cross flow	Presenting a new model for M-cycle cross flow with consideration of spray water variation and wall longitudinal heat conduction along exchanger
Moshari [101]	2015	Finite difference (Matlab)	Counter & Cross	Performance comparison between counter flow, cross flow and four-stage indirect evaporative cooler with the same air and exchanger parameters
Cui [102]	2016	Finite element (EES)	Counter flow	Providing a performance correlation for counter flow regenerative M-cycle exchanger
Cui [103]	2016	Finite element (Comsol)	Counter flow	Combination of liquid desiccant dehumidification and regenerative M-cycle cooler
Jafarian [105]	2017	Finite Volume	Counter flow	Momentum, energy and mass transfer are simultaneously solved for counter flow M-cycle
Wan [106]	2017	Finite Volume	Counter flow	Providing a performance correlation for counter-flow M-cycle coolers

Zhang et al. [4], evaluated the specifications of the ISAW cooler [87] (which works based on M-cycle theory) using the finite-element method. The air flow through the channels was considered to be laminar. They varied each parameter while other parameters were remaining unchanged. According to finite element difference analysis of this air cooler, for desired supply air temperature (26 ^{0}C), air humidity, air velocity in dry channel and wet channels should not be higher than 65%, 1.77 m/s and 0.7 m/s respectively.

The main results (effect of different parameters on cooler characteristics) of finite element/ volume/difference analysis of M-cycle cooler can be summarized as below.

- Increment of inlet air temperature increases product air temperature (warmer supply air) but it does not mean lower efficiency or cooling capacity.
- Increment of inlet air temperature improves cooling capacity, wet-bulb effectiveness and COP which shows M-cycle suitability for warm weather.
- Increment of air relative humidity enhances wet-bulb effectiveness and supply air temperature but decreases COP and cooling capacity. These results prove that, wet-bulb effectiveness should not be considered as an independent characterize the performance of the IEC.
- Increment of air speed (air flow rate) increases wet-bulb effectiveness and supply air temperature but decreases cooling capacity and COP.
- Increment of air passage height increases supply air temperature and reduces cooling capacity wet-bulb effectiveness. However, COP showed an ascending-descending behaviour. Nonetheless, the maximum point of COP should not be considered as the best condition because of very low cooling capacity and wet-bulb effectiveness in that point.
- Increment of dry channel or wet channel length decreases supply air temperature (colder air) and COP and increases cooling capacity and wet-bulb effectiveness.
- In comparison with conventional cross-flow exchanger (in which M-cycle has not been used), this exchanger provides 16% higher wet-bulb effectiveness and around 60 W higher cooling capacity in the same characteristics.

5. Statistical design methods

Response surface methodology (RSM), Fuzzy inference system (FIS), Adaptive neuro-fuzzy inference system (ANFIS), Multiple linear regression (MLR), Genetic programing (GP), Artificial neural network (ANN) are different type of statistical design method (SCST) [63] which can be used instead of numerical models to analyse the performance of Maisotsenko exchanger. The requirements of this method are: 1) basic knowledge of parameters that affect the system characteristics and 2) enough numerical or experimental data from the system operation. Some researchers have evaluated the indirect evaporative via SCST method as shown in Fig. 7. Pandelidis and Anisimov [4&62] believed that numerical models based on partial differential and algebraic equations require higher amount of calculation time and they are complex and cumbersome for everyday use [4]. Hence, an accurate fast mathematical model based on response surface regression procedure was developed as below which may be applied for engineers.

5.1. RSM method

This technique describes the basic performance of cross-flow M-cycle exchangers and can be used for optimization of M-cycle because of its lower calculation time.

As described in [82], Response surface methodology (RSM) comprises of some basic steps as below:

1) Screening: experiments are performed with the aim of providing the vital few control parameters.

2) Modelling: experiments are performed with the aim of modelling the quality characteristic of interest (response) as a function of control parameters;

3) Optimization: response model is evaluated to find the variable settings in which optimum conditions are obtained.

Detail explanations about Response Surface Methodology can be found in [82-86]. The analysis of M-cycle exchanger by RSM method can be classified into two main group as below.

- Describing (prediction) of the basic performance of cross flow M-cycle exchanger including outlet air flow temperature, dew point effectiveness, cooling capacity and COP
- Optimization of five influence factors including inlet temperature, relative humidity, primary air mass flow rate, working to primary air ratio and relative length of the initial part in order to determine the optimal range of operational and geometrical conditions.

The results obtained from this method are the same observed in numerical simulations. However, according to [61], optimization based on single performance factor is not suitable at all for M-cycle. In other words, it is impossible to optimize on the basis of single performance factor because the optimum value for one parameter is not favourable from the view point of other parameters. Hence, a compromise multi-optimisation technique is required to determine an appropriate working condition for M-cycle air coolers. To that reason, a multi-parameter optimization was carried out in [62]. Authors [62] used the concepts of Harrington's desirability function and Compertz-curve for multivariate quality optimization (phase 3 of RSM method) in order to find suitable climate zones for the cross flow M-cycle exchanger. Gompertz curves show the effect of varying one parameter while keeping the others constant. More explanation about the concept of desirability function can be found in [82].

Based on RSM multi-optimization analysis, M-cycle is suitable for most of the typical climate condition. However, for really moist regions (more than 65% relative humidity) and also cold regions (around 25^0C) are considered as the unsuitable climates for M-cycle. Nonetheless, for hot and very moist weathers, combination of dehumidifier and M-cycle can solve the humidity problem. Cross flow M-cycle has a potential of wide application around the world [61].

5.2. GMDG type neural network method

Group method of data handling-type neural network (GMDH) is a self-organizer predictive approach which is a subset of artificial neural network (ANN) family [63]. This technique was employed to perform a multi-objective performance optimization of Coolerado M50 M-cycle unit. [63]. Average annual values of COP and cooling capacity were simultaneously maximized while inlet air velocity and working to air ratio were decision variables of optimizations. This optimization was carried out for twelve Koppen-Geigers classification. Koppen Geigers is one of the most widely used climate classification systems in all over the worlds. The GMDH model was used to predict the product air temperature as a function of inlet temperature, inlet humidity, inlet velocity and non-dimensional channel length. According to GMDH model, M-cycle is applicable for cooling season of all 12 classes of Koppen-Geigers climate classification system. Other results can be briefly described as below.an experimental study

- Optimized velocity decreases with decrement of inlet air temperature
- Optimized velocity decreases with increment of inlet relative humidity
- M-50 Coolerado velocity is very close to the obtained optimum velocity range by GMDH model. Hence, there is no need to change the system's fan.
- Optimized working to air ratio decreased with reduction of inlet temperature or humidity.
- M-50 Coolerado working to air ratio is significantly higher than the achieved working to air ratio by GMD model.

6. Experimental investigations

The main experimental investigations on Maisotsenko type of indirect evaporative coolers ate illustrated in Table 5. It should be notes that, former type of indirect evaporative coolers (which are not worked based on Maisotsenko cycle) are not covered. As can be seen in Table 5, the previous experimental studies on M-cycle air coolers are mainly discussed below considerations.

- Comparison of cross flow and counter flow under various operating condition

- Evaluation of the current existence commercialized M-cycle coolers
- Investigation on the combination of M-cycle and desiccants
- Sensitivity (parametric) analysis of different types of M-cycle air cooler
- The use of existence M-cycle cooler for a defined region with specific climate
- Effect of wettability factor and dehumidification

Table 5. Experimental studies of M-cycle air conditioning systems

Experimental investigations	Year	Description
Riangvilaikul & Kumar [78]	2010	Experimental study of a novel indirect evaporative air cooler
Zhan et al. [68]	2011	Comparison between counter-flow and cross-flow exchanger of M-cycle
Zube & Gillan [71]	2011	Evaluating a commercialized type of M-cycle air cooler
Gao et al. [73]	2014	Combination of liquid desiccant and indirect M-cycle cooler
Rogdakis et al. [72]	2014	Analysis of the M-cycle air cooler for Greek climate condition
Khalid et al. [69]	2016	Design and analysis of counter flow exchanger for M-cycle
Khalid et al. [70]	2016	Investigation of an improved M-cycle cooler under low velocity condition
Duan et al. [108]	2016	Analysis of the M-cycle air cooler for China climate condition
Antonellis et al. [109]	2016	Analysis of a cross flow indirect evaporative
Xu et al. [110]	2017	The use of an innovative exchanger for M-cycle air cooler
Duan et al. [111]	2017	Operational performance and impact factors of a counter-flow regenerative M-cycle
Antonellis et al.[112]	2017	Effect of wettability factor and adiabatic humidification on a cross flow heat exchanger
Lin et al. [113]	2017	Effect of dehumidification on cross-flow M-cycle cooler
Shahzad et al.[79]	2018	Combination of solid desiccant with cross flow M-cycle cooler

Operating condition of each experimental investigation is different from each other. Characteristics of each experimental study and their operating parameters and key results are illustrated in Table 6. Type of cooling device, inlet condition including velocity and temperature, geometric specifications, obtained wet-bulb effectiveness and dew-point effectiveness are presented in this table.

Table 6. Experiment condition of each experimental study of M-cycle

	Arrangement	Inlet air Tem	Inlet air humidity	Inlet air volume	Channel gap	Channel length	Channel width	Outlet Tem	Wet-bulb Eff	Dew-point Eff
Riangvilaikul [78]	Counter flow	25-45 ^{0}C	7-26 g/Kg	1.5-6 m/s	5 mm	1200 mm	80 mm	15-32 ^{0}C	92-114%	58-84%
Zhan [68]	Counter &Cross flow	25-45 ^{0}C	11 g/kg	2000 m^3/h	5 mm	1000 mm	-	Counter: 16-18 ^{0}C Cross: 18 -20 ^{0}C	Counter: 130-138% Cross: 110-120%	Counter: 70-80% Cross: 75-80%
Zube [71]	Couner	40^{0}C	0.128m^3/s	-	-	-	-	-	-	-
Gao [73]	Cross flow	24-38 ^{0}C	14 g/kg	0.2kg/s	2.2mm	500 mm	500 mm	-	-	90-140%
Rogdakis [72]	Cross flow	32-36 ^{0}C	-	-	-	-	-	21-22.3 ^{0}C	-	-
Khalid [69]	Counter flow	25-45^{0}C	12-18 g/kg	0.88-1.5m/s	4 mm	900 mm	40 mm	18-25 ^{0}C	100-120%	70-80%
Khalid [70]	Cross flow	25-45^{0}C	11-19 g/kg	0.5-1.1 m/s	4 mm	Dry:058 mm Wet:203 mm	25 mm	18-24 ^{0}C	90-120%	60-80%
Duan et al. [108]	Counter flow	27-37^{0}C	-	-	6 mm	900 mm	314 mm	18-28 ^{0}C	74-82%	-
Antonellis et al. [109]	Cross flow	30-36^{0}C	10 g/kg	1400 m^3/h	3.35 mm	500 mm	500 mm	23-24 ^{0}C	65-80%	-
Xu et al. [110]	Counter flow	30-37.8^{0}C	-	750 m^3/h	-	1000 mm	800 mm	-	67-76%	-
Duan et al. [111]	Counter flow	24-34^{0}C	-	0.6-3 m/s	6 mm	900 mm	314 mm	19-24 ^{0}C	38-80%	20-45%
Antonellis et al.[112]	Cross flow	29-35^{0}C	10-11 g/kg	1400 m^3/h	3.21 mm	470 mm	470 mm	21-23 ^{0}C	-	-
Lin et al. [113]	Cross flow	30 ^{0}C	12-13 g/kg	2800 m^3/h	3 mm	-	-	20-28 ^{0}C	85%	62%
Shahzad [79]	Cross flow	25-45^{0}C	12-18 g/kg	660kg/h	5 mm	900 mm	280 mm	17-25 ^{0}C	-	-

Some researchers (for example [71]) have employed the commercialized M-cycle cooler for their experiments. However, most researchers have preferred to use unique test-rig for the test. A schematic view of some of the previously used experimental test-rigs is shown in Fig. 17.

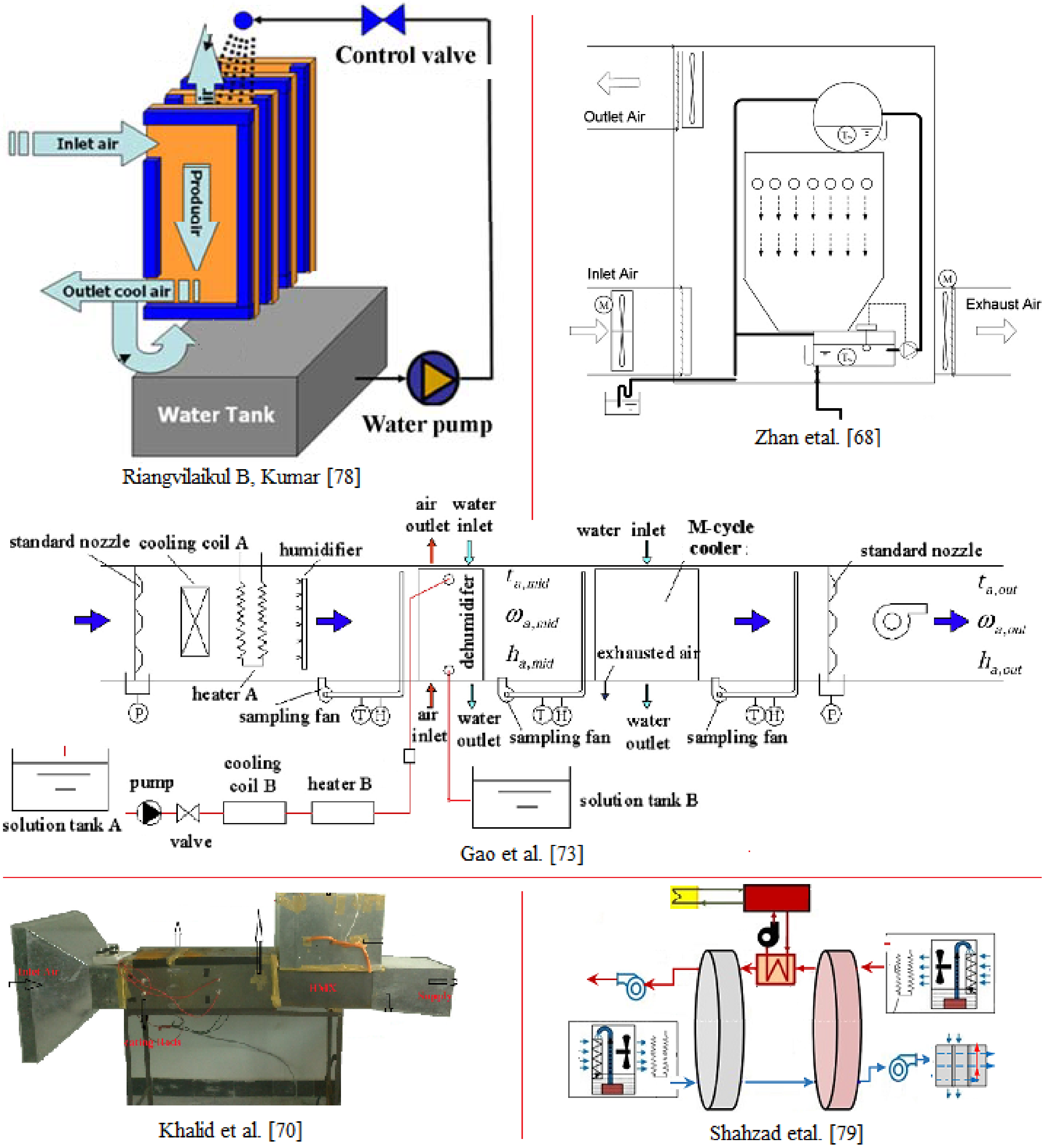


Fig. 18. Schematic view of some of the previously used experimental test-rigs

Although each experimental study has different flow, thermal and geometrical condition, the curve behaviour of sensitive analysis (parametric study) are the same for all of them through a specific type of air cooler (single M-cycle or combination of M-cycle and desiccants). Hence, it is possible to provide a compact-comprehensive view of the results of parameter analysis of M-cycle air conditioning systems. In this regard, M-cycle experimental investigations can be classified into two main categories. In First category, the researchers have focused on the effect of various parameters on single M-cycle characteristics and in the second category, the effect of said parameters were studied on combined M-cycle-desiccant systems. A comprehensive compact view of the first and second categories and their results are shown in Fig. 19 and Fig. 20 respectively which were extracted from [68-73 & 78-79]. For example, according to Fig.19, increment of inlet air humidity causes enhancement of product air temperature and dew point effectiveness and causes reduction of wet bulb effectiveness. The results obtained from experiments are comparable with the numerical results mentioned above (in Table 3).

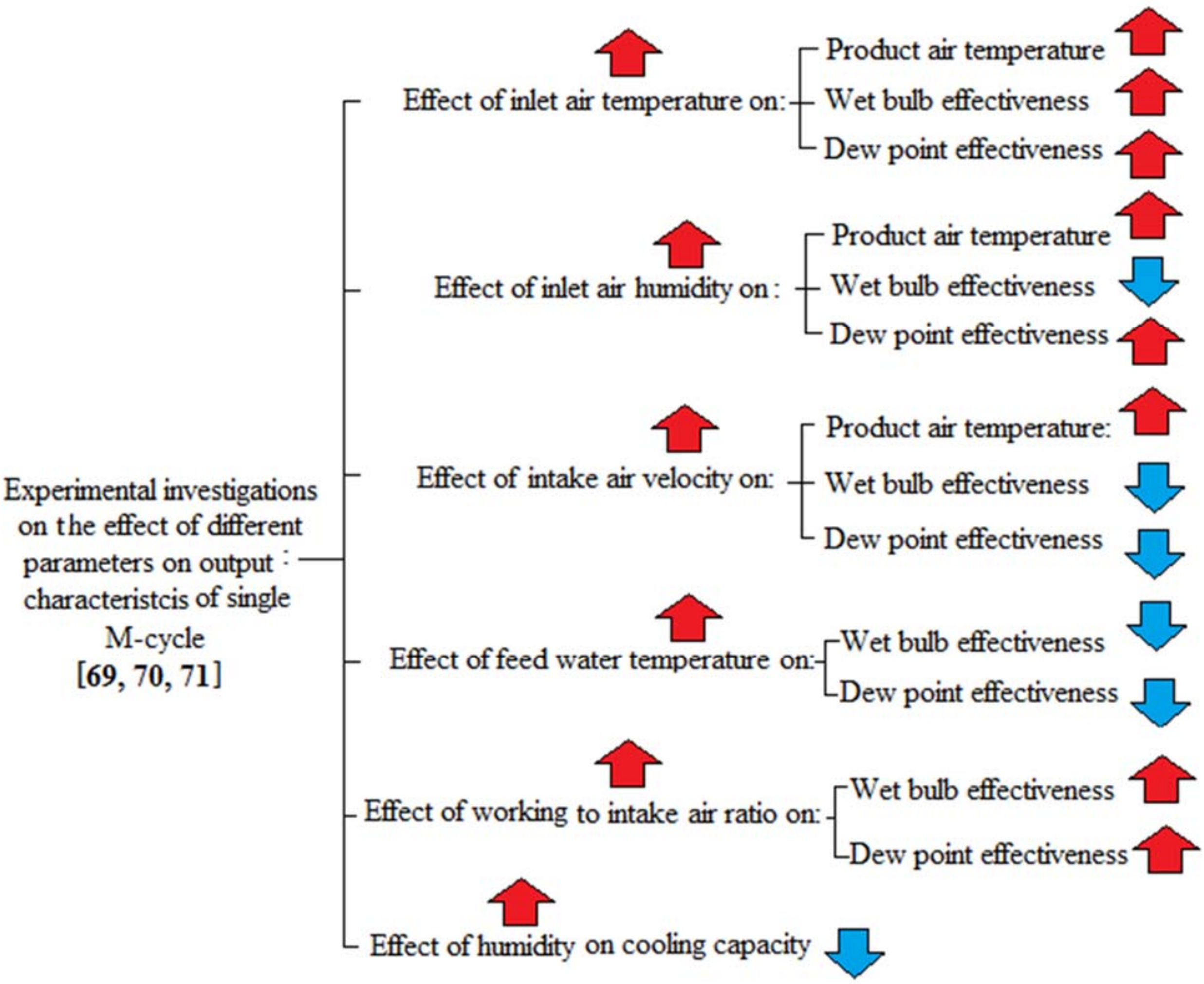


Fig.19 The effect of various parameters on single M-cycle characteristics (experimental investigations)

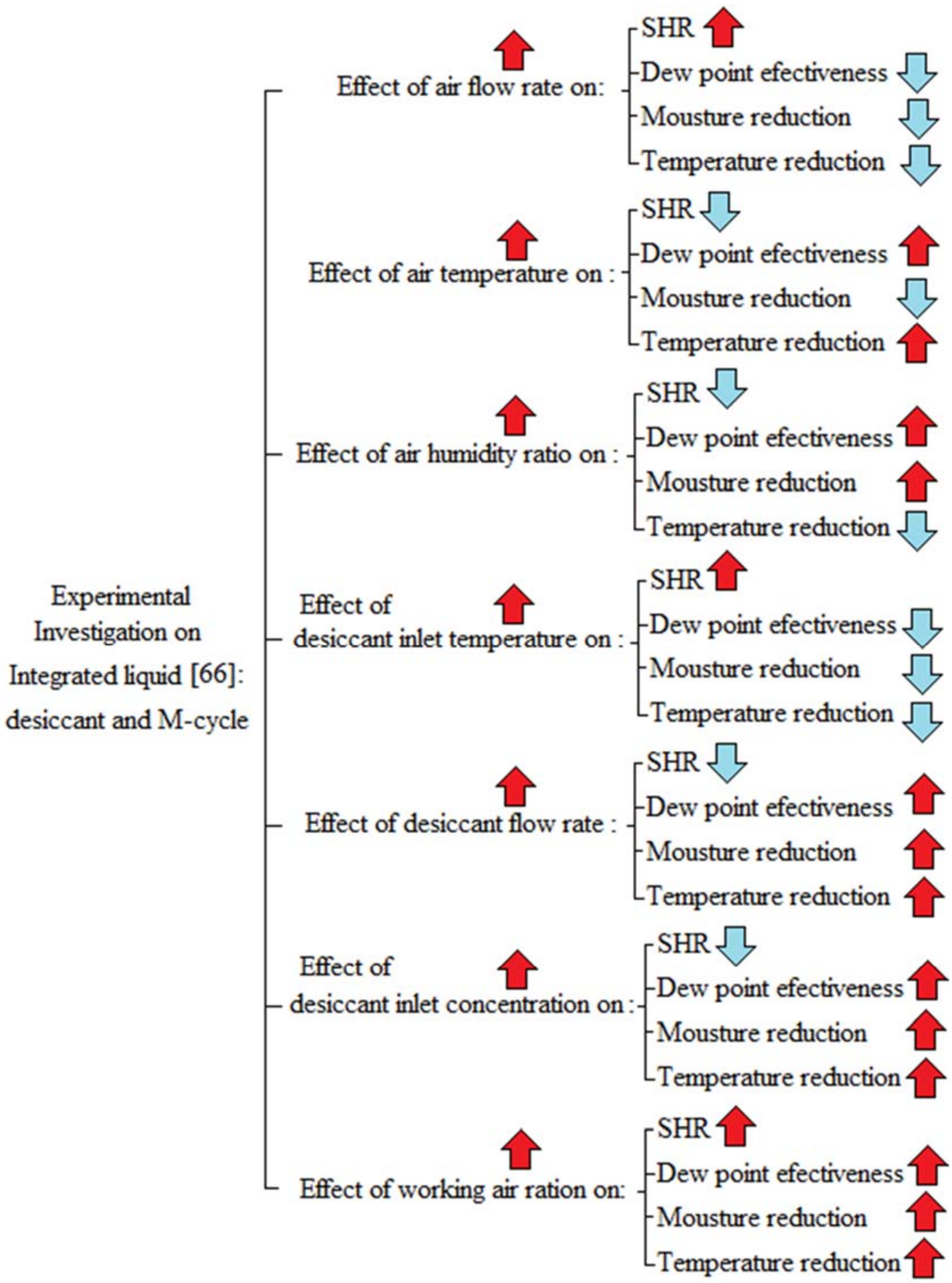


Fig.20 The effect of various parameters on single desiccant M-cycle characteristics (experimental investigations)

7. Industrial status of M-cycel air coolers

Maisotsenko cycle is protected by more than 200 patents all over the world [15]. Coolerado cooperation produced the first practical realization of the M-cycle cooler technology [17] for different aims including commercial, residential, solar and hybrid M-cycle air coolers. According to the experiments which were carried out by National Renewable Energy Laboratory (NREL), Coolerado's H-80 air cooler used 80% less energy than compressor-based air conditioning systems under hot and dry climate condition [17]. Coolerado air conditioners can be found in markets around the world. Coolerado products are classified into three main categories including HMX, M50 and C60. A general view of these air coolers can be seen in Fig. 20 to Fig. 22 which was extracted from Coolerado catalogs. Nowadays, other corporations such as Climate Wizards (Adelaide, Australia) etc. produced different types of M-cycle indirect evaporative air coolers particularly for big halls. Their applicability for big halls (which is installed on the roof of halls) is more prevalent than those for residential. Indirect evaporative cooler installed in the Hub central at the University of Adelaide is the first large IEC system utilization in Australia [89].

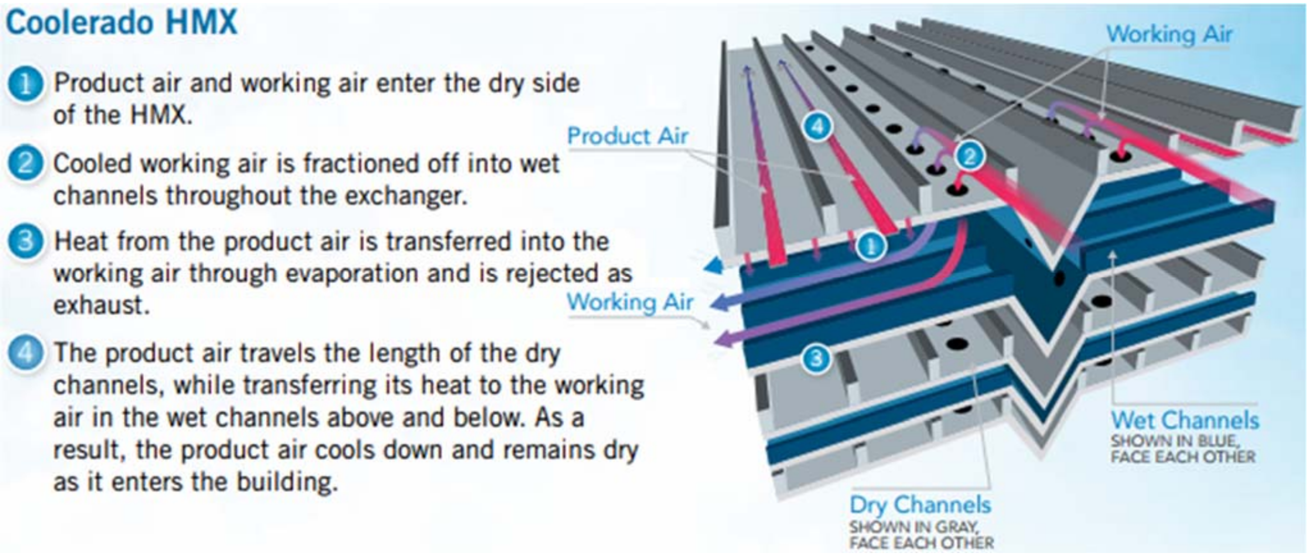


Fig. 20 HMX Coolerado air cooler [from brochure]

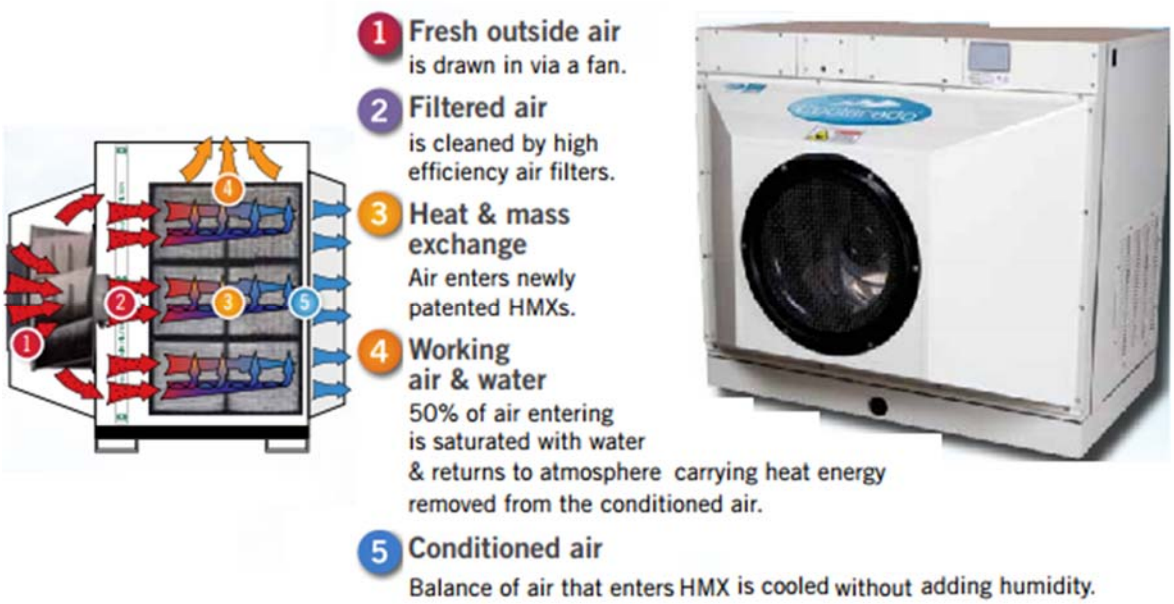


Fig. 21 C60 Coolerado air cooler [from brochure]

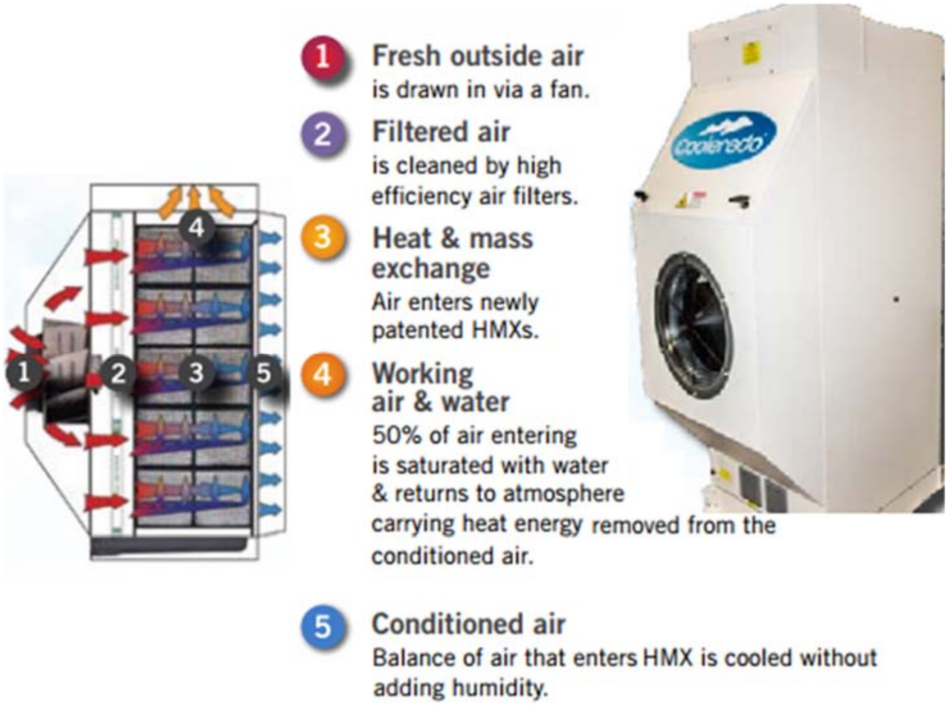


Fig. 22 M50 Coolerado air cooler [from brochure]

8. Future research direction

Through the literature review described above, it was found that, most of the previous analytical solutions of M-cycle air cooler have made several assumptions to simplify their model and reduce the calculation time, but sacrificing their accuracies. Moreover, previous analytical models are developed only on the simplest structure (without perforation) of M-cycle coolers. Therefore, researchers may interested in develop new analytical models for perforated or other more complex structures of M-cycle. Perforated M-cycle for example are investigated only via numerical or experimental methods which requires further time and cost. Obviously, formulations of current analytical model cannot easily be employed for complicated structures of M-cycle coolers.

Furthermore, there is no solution (and even application) for non-parallel-plate arrangements of M-cycle in literature review. In other words, M-cycle system's thermal-frictional behaviors of various configurations of non-parallel plates are completely unknown. As a result, lack of analytical solution for perforated M-cycle coolers and also non-parallel-plate structures can be considered as some of the future research direction.

In addition, despite the importance of the second law of thermodynamic consideration in any thermodynamic process, extremely few analyses have been carried out for M-cycle coolers from the view point of the second law of thermodynamic. Hence, clarification of exergetic aspect of M-cycle system and developing (via either numerical or experimental techniques) is another major research direction which should be bridged. This analysis includes the impacts of operational-geometrical parameters on exergetic characteristics of M-cycle which leads to the improving of performance of M-cycle systems. Therefore, the research directions of the M-cycle cooler can be briefly described as below.

- Lack of high accurate thermal analytical model for other structures of M-cycle cooler.
- Lack of enough M-cycle evaluation from the view point of the second law of thermodynamic.
- Lack of enough experimental study of the exergetic characteristics of M-cycle air cooler.
- Lack of enough exergetic sensitive analysis of M-cycle.
- Lack of the evaluation of the effect of non-parallel plates on M-cycle cooler.

Furthermore, it seems that, extra investigation is required to determine how M-cycle can be appropriately developed for residential aims in additional to big public places. In other words, it may be tried to reduce the size (weight) of M-cycle coolers without reduction of their performance so that it can be employed easily for any area. Moreover, the results of some investigations showed that the specifications of commercialized M-cycle coolers are not the best-optimized ones. Hence, further analysis and optimization via other methods is required. Although the use of perforation through the separator plate reduces the pressure drop, it reduces thermal performance and causes complexity of manufacture and increment of production cost. Thus, other simple novel techniques should be developed in order to reduce the pressure drop along the exchanger. As a general result a lot of investigations should be carried out to find the best operational condition, geometry and other aspects of M-cycle air conditioning systems.

9. Conclusion

The present study provides a comprehensive review on Maisotsenko air conditioning systems, regarding its evaluation methods, obtained results, industrial situation and future research direction. The main analytical solutions of M-cycle (indirect evaporative) were evolutionary and briefly discussed. The relationship between analytical solutions was provided. The significant results which have been obtained from numerical simulation or experimental techniques were graphically presented. Statistical Design Tool is another analysis method for M-cycle which has been employed by some researchers. The application of said methods and their features (advantages) were explained. The current industrial situation of M-cycle air coolers was demonstrated. Coolerado-corporation which is the first corporation produced M-cycle coolers was chosen for this aim. The study concludes that, despite the commercialization of Maisotsenko air coolers, a lot of exact researches are required to improve the M-cycle coolers. A geometrical modification can be used instead of perforations to reduce the pressure drop through the exchanger. M-cycle should be evaluated from the view point of the second law of thermodynamic in addition to the first law of thermodynamic. Extremely few experimental investigations have been performed compared to the other methods.

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