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# MAISOTSENKO CYCLE FOR COOLING PROCESSES

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The Maisotsenko Cooling cycle combines the thermodynamic processes of heat exchange and evaporative cooling in a unique indirect evaporative cooler resulting in product temperatures that approach the dew point temperature (not the wet bulb temperature) of the working gas. This cycle utilizes the enthalpy difference of a gas, such as air, at its dew point temperature and the same gas saturated at a higher temperature. This enthalpy difference or potential energy is used to reject the heat from the product. Consider the cooling gas to be air and the liquid to be water; the Maisotsenko Cycle allows the product fluid to be cooled in temperature ideally to the dew point temperature of the incoming air. This is due to the precooling of the air before passing it into the heat-rejection stream where water is evaporated. For purposes of this paper, the product fluid is air. At no time is water evaporated into the product airstream. When exhausted, the heat rejection airstream or exhaust air is saturated and has a temperature less than the incoming air, but greater then the wet bulb temperature. This cycle is realized in a single apparatus with a much higher heat flux and lower pressure drop than has been realizable in the past due to its efficient design.

*Keywords:* Dew point; Wet bulb; Evaporative cooling; Effectiveness, Maisotsenko cycle

## INTRODUCTION

For ease of explanation, we will consider the cooling gas to be air and the fluid to be cooled also to be air. To help understand the Maisotsenko cycle a direct evaporative process will be examined, then a typical indirect evaporative process, and finally the Maisotsenko cycle.

## DIRECT EVAPORATIVE COOLING

In the past, evaporative coolers have been used to lower the temperature of air by using the latent heat of evaporation, changing water to vapor (see Fig. 1). In this system, the energy in the air does not change. Warm dry air is changed to cool moist air. Heat in the air is used to evaporate water; no heat is added or removed, making it an adiabatic process (fan heat gain or pump energy is ignored in this evaluation). This also assumes the water entering the system to be evaporated is at the wet bulb temperature of the entering air and there is no excess

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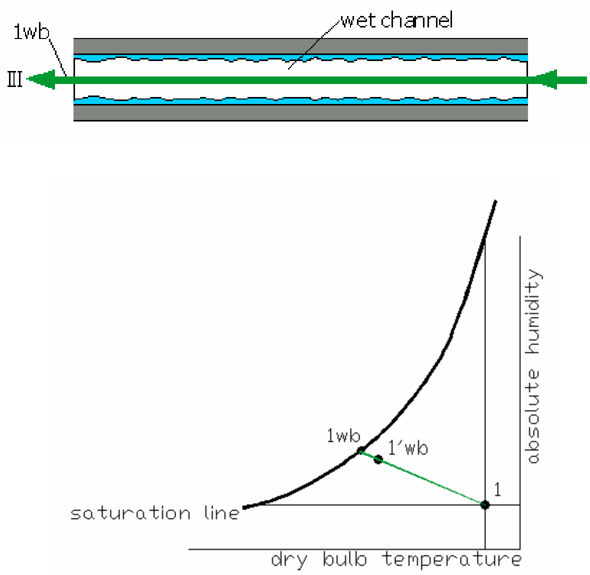


FIGURE 1 Scheme and process of direct evaporative cooling.

water or it has negligible effect on the adiabatic process. The enthalpy of the system does not change [see Eqs. (1)–(3)],

$$h = (C_p * t) + (h_s * W) \tag{1}$$

$$h_{in} = h_{out} \tag{2}$$

The lower temperature and higher vapor content of the air can then be expressed by

$$G_{air} * C_p * (t_{in} - t_{out}) = G_{air} * h_s *(W_{out} - W_{in}) = G_{water} * h_s \tag{3}$$

(sensible heat loss) (latent heat gain) (latent heat gain)

These direct evaporative systems vary from 70% to 95% effective ($E$) in temperature reduction to the incoming air's wet bulb temperature (Watt and Brown, 1997), (see Fig. 1) where

$$E = \frac{t_1 - t'_{1\ \text{wet bulb}}}{t_1 - t_{\text{wet bulb}}}. \tag{4}$$

## INDIRECT EVAPORATIVE AIR COOLING

For many years, indirect evaporative air coolers have been used with little success. Because of the poor heat transfer rates, units that have been commercialized

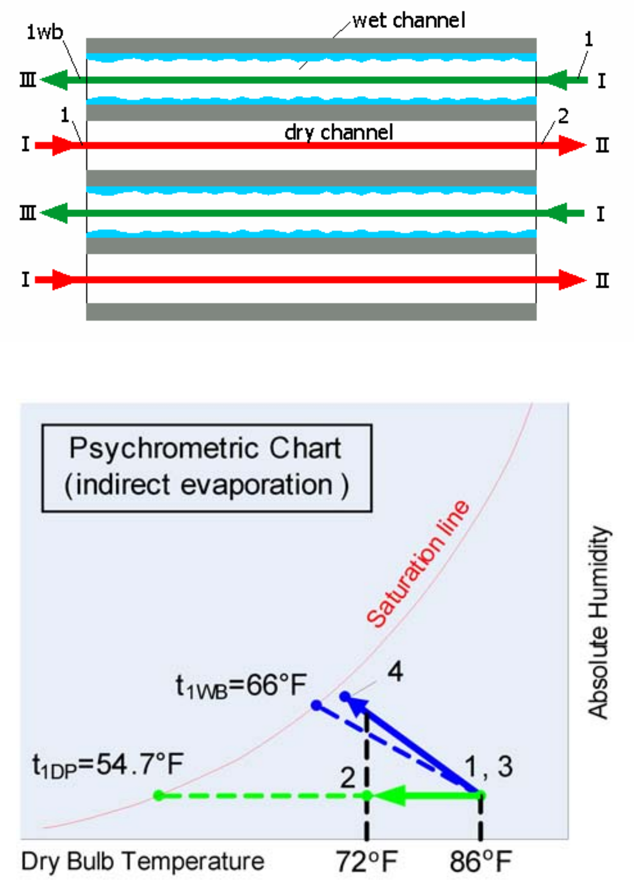


FIGURE 2 Scheme and process of indirect evaporative cooling.

have not been able to produce a cooling capacity that justifies the excessive material and manufacturing costs.

Thermodynamically, an indirect evaporative air cooler passes primary or product air over the dry side of a plate and secondary or working air over the opposite wet side of a plate. The wet side absorbs heat from the dry side by evaporating water and therefore cooling the dry side with the latent heat of vaporizing water into the air. The ideal and real conditions for indirect evaporative cooling are represented in Figure 2. The air with temperature $t_1$ on the dry side of the plate travels in counterflow to the air on the wet side. Ideally, the product air temperature on the dry side of the plate could reach the wet bulb temperature $t_{1\text{wet bulb}} = t_2$ of the incoming air (in reality only $t_2'$).

Ideally, the working air on the wet side of the plate would increase in temperature from its incoming air wet bulb temperature to the incoming product air dry bulb temperature and be saturated. Of course, this would require a balancing

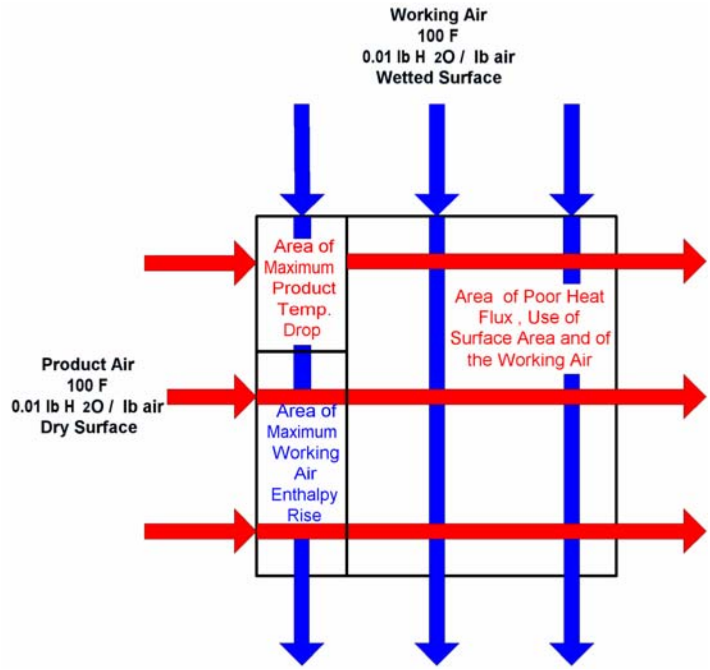


FIGURE 3 Typical cross flow indirect evaporative cooler.

of the product and working airflow rates with an infinite amount of surface area and pure counterflow. Equations (5) show the balance of energy (excluding fan or pump losses and water temperature entering gains or losses) for any indirect evaporative cooling system, ideal or not, where $Q_o$ represents the cooling capacity.

$$Q_o = G_{\text{product}}\ (h_{\text{product in}} - h_{\text{product out}})$$

$$= G_{\text{exhaust}}\ (h_{\text{exhaust out}} - h_{\text{exhaust in}}) \tag{5}$$

$$Q_o = G_{\text{product}} * C_p\ (t_{\text{product in}} - t_{\text{product out}})$$

$$= G_{\text{exhaust}} * [h_s(W_{\text{exhaust out}} - W_{\text{exhaust in}}) + C_p(t_{\text{exhaust in}} - t_{\text{exhaust out}})] \tag{5a}$$

In practice, it is not possible to have pure counterflow because the air must enter and leave from the same sides. This geometry of plate exchangers forces indirect evaporative coolers to be in crossflow. The effectiveness $E$ of these types of coolers is reported to approach 54% (Watt and Brown, 1997) of the incoming air wet bulb temperature (see Fig. 2), where

$$E = \frac{t_1 - t_2'}{t_1 - t_2} = \frac{t_1 - t_2'}{t_1 - t_{1\ \text{wet bulb}}}. \tag{6}$$

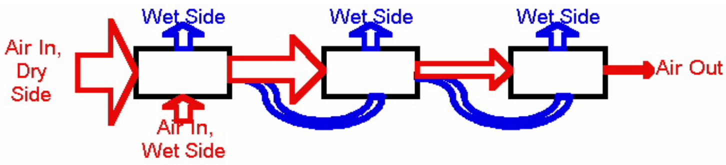


FIGURE 4 Several indirect evaporative coolers for approaching the dew point temperature.

Figure 3 shows a typical crossflow indirect evaporative cooler. Examining the cooler across one plate shows that the highest heat flux will be where the dry channel inlet and wet channel inlet cross. This is due to the wet channel working air having its lowest wet bulb temperature and of course the dry channel entrance having its highest temperature at this point within the heat exchanger. The highest enthalpy of the working air leaving the exchanger (maximum work for the quantity of working air) will be where the dry channel product air enters, since it is the hottest, and the wet channel working air leaves, since it will have reached its highest temperature and humidity. This means the cooling across the entire inlet of the product air dry channels is the most efficient portion of the heat exchanger. The further across the dry channel the product air travels, the less heat transfer (work) is accomplished by the working air in the wet channels simply due to the crossflow characteristics. This means that about 10% of the working air and 10% of the surface area performs about 70% of the cooling.

Looking at product air flow change in energy or enthalpy, where no water is added to the product,

$$G_{\text{product}} * (h_{\text{product in}} - h_{\text{product out}}) = G_{\text{product}} * C_p\ (t_{\text{product in}} - t_{\text{product out}}). \qquad (7)$$

If an indirect evaporative cooler is 54% efficient, then the change in energy of the working air or the change in enthalpy is

$$G_{\text{exhaust}}\ (h_{\text{exhaust out}} - h_{\text{exhaust in}}) = G_{\text{product}} * C_{\text{p}}\ 0.54\ (t_{\text{product in}} - t_{\text{wet bulb}}). \qquad (8)$$

It has been suggested that several indirect evaporative coolers can be used in series to lower the product airstream to the dew point temperature of the air entering by directing a portion of the precooled air from the first cooler to the working of the next cooler, etc. (see Fig. 4). This is good in theory, but is impractical since each exchanger is at best only 54% efficient. The result would be that each subsequent heat exchanger would require more product air to be used than working air. In the end, a dew point temperature might be obtained; however, the amount of product airflow exiting the final heat exchanger would be minimal, making such a system unrealistic for cooling. The high-pressure drop encountered due to the entrance and exit losses of each unit would also make such a system impossible for commercial use.

## HEAT AND MASS TRANSFER OFF A ONE-SIDE-WETTED POROUS SURFACE

One of the key components for good heat and mass transfer in an indirect evaporative cooler is having a thin, wetted, porous surface on the wet side of the plate. The other side of the plate must be impervious to water. The obvious reasons are as follows:

- The entire surface will be wetted creating the maximum evaporation surface.
- The heat from the dry side does not pass directly through the plate to the air on the wet side.
- Being thin, it takes up less space.
- The thinness helps lower the heat transfer rate.

A less obvious reason, but very important, is that the wetted surface temperature will be at the wet bulb temperature of the adjoining air. The air temperature near the plate will then also be at its wet bulb temperature and saturated (Luikov, 1963; Smolsky and Sergeyev, 1962). Heat added to the wetted surface through the plate from the dry side causes water to absorb the heat. The water then evaporates into the air at the partial pressure of water vapor of the surrounding air. The partial pressure of the water vapor also defines the airs wet bulb temperature. In effect, the heat added through the plate to the water on the wetted surface forces the water to evaporate off into the air's low partial pressure and into a saturated condition. As a comparison, in an adiabatic evaporation process, such as in direct evaporating cooling, the air draws the moisture off at a much slower rate and never reaches saturation.

The working air on the wetted side cannot rise in temperature directly from the heat on the dry side of the plate, since it is adjacent to the wet side, which is at the wet bulb temperature. To raise the temperature of the saturated working air, the moisture being forced from the wick into the air must also force a higher wet bulb temperature on the wick surface, then the air can rise in temperature to accept additional water vapor. Because the air is saturated, its dry bulb, wet bulb, and dew point temperatures are the same. The air leaving an efficiently wetted and designed indirect evaporative cooler is saturated or very close to it.

The pores' wick heat transfer surface eliminates excess water. Without excess water, the evaporative latent heat goes to cooling the air, not the excess water.

This also means that the water temperature entering has little effect on the cooling capacity since the water sensible heat gain or loss in reaching the wet bulb temperature of the adjacent air is small relative to the latent heat of evaporation.

Most indirect evaporative coolers do not have a wick surface or even a thin film of water covering the entire plate, but instead have areas of heavy wetting and areas that are dry. In these conditions, the air is humidified without absorbing heat directly from the dry side from a combination of heavy wetted areas and being heated sensibly through the dry sections. The air does not necessarily leave the heat exchanger saturated.

## MAISOTSENKO CYCLE

The Maisotsenko cycle uses the same wet side and dry side of a plate as described in the above indirect evaporative cooler, but with a much different airflow creating a new thermodynamic cycle. This cycle allows the product air to be cooled below the wet bulb temperature and toward the dew point temperature of the incoming working air.

The Maisotsenko cycle (first embodied in Russian patents No. 571669, 979796, and 2046257, and U.S. patents No. 4350570, 4842052, 4971245, 4976113, 4977753, and 5453223) utilized the psychrometric energy (or the potential energy) available from the latent heat in an evaporating gas. The Maisotsenko cycle was realized in a uniquely designed plate wetting and channel system, which achieved optimum cooling temperatures and saturated working air with the highest enthalpy possible for the exhausted working air temperatures obtained.

## COUNTERFLOW ADIABATIC HEAT AND MASS EXCHANGER

To explain how the original Maisotsenko cycle works, thermodynamically, we have started with the simple adiabatic model shown in Figure 5. This shows the cross section of plates with wet sides together and dry sides together. In this example, the incoming air I is passed over the dry side of the plates and then turned as the air II passes over the wet side of the plates and then exhausted out as air III. As the air passes over the dry side of the plate, it is cooled by the water evaporating on the wet side, or the latent heat of vaporization absorbs the heat from the plate. The airstream in the dry channels is cooled by the same airstream in the wet channels, reducing its wet bulb temperature. The enthalpy at the point where the air turns from the dry channel to the wet channel II is at the dew point temperature of the incoming air $h_{\text{dew}}$ point. This precooled air that turns to

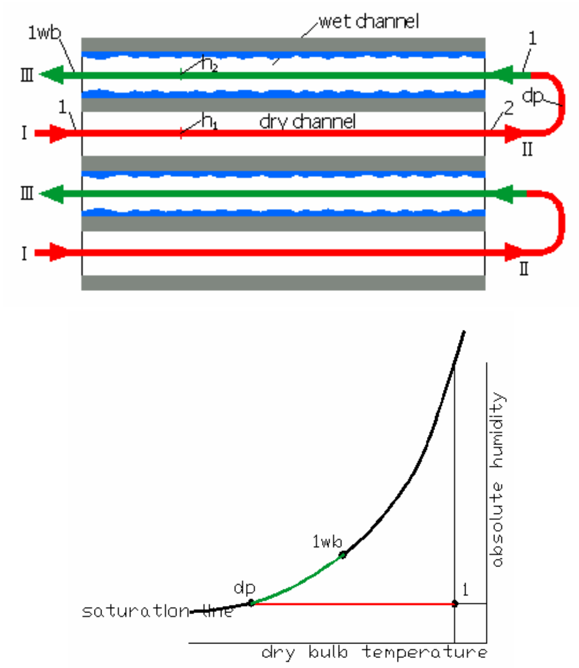


FIGURE 5 Scheme and process of regenerative adiabatic evaporation.

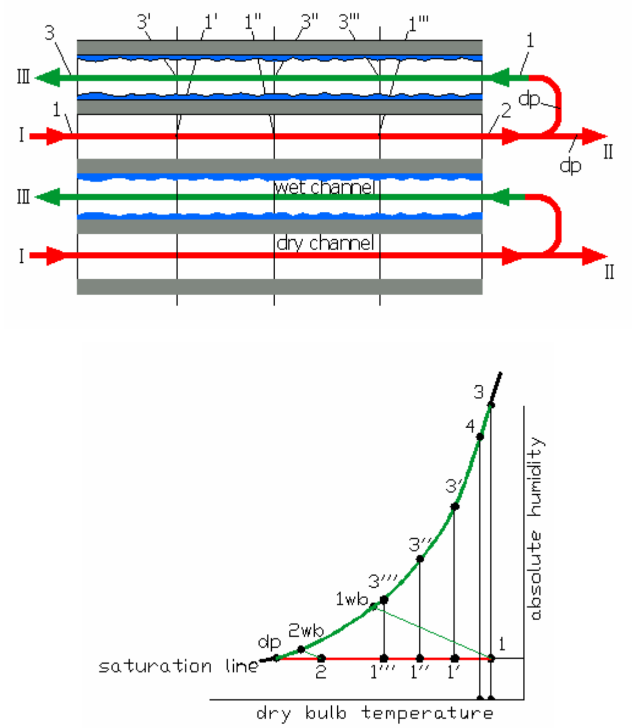


FIGURE 6 Scheme and process of the Maisotsenko cycle with partial extraction of air as the product cold air.

enter the wet channels is then at the dew point temperature of the entering airstream. The energy balance would then be (see Fig. 5)

$$G\,(h_1 - h_{\text{dew point}}) = G\,(h_{1\text{wet bulb}} - h_{\text{dew point}}) \quad \text{or} \quad h_1 = h_{1\text{wet bulb}} \tag{9}$$

(dry side) (wet side)

at any point across the plate $h_1 = h_2$ since there is no heat being added to or removed form the system.

These theoretical results have been achieved on a regular basis in several prototype models, confirming that the temperature of the airstream, after passing along the dry side of the plate, approaches the dew point temperature. In this process, the cooling capacity is equal to zero because the product air is the working air; there is only one stream of air or $h_{\text{in}} = h_{\text{out}}$.

Analysis of this counterflow adiabatic heat exchanger shown in Figure 5 shows that the airstream leaving the wetted channels or plates possesses additional cooling capacity. This is characterized by the psychrometric difference in temperature between the air entering and leaving the system. The air leaving the system has the potential of being at the saturation line ($\varphi = 1$) with its associated moisture to the maximum of point 3 of Figure 6, where $t_3 = t_1$ and $\varphi = 1$. Heat to drive the exhaust temperature and moisture up must be obtained from the dry side of the plates, since the wet side is in heat transfer only with the dry side, or $h_3 - h_{\text{dew point}} > h_1 - h_{\text{dew point}}$. Consequently, the flow rate of the exhaust airstream becomes less than the flow rate of the entering airstream. If the reduction in the exhaust airstream flow rate would occur before entering the wetted channel where the air is coolest, then the air diverted could be used as a useful product. In this case, the described heat exchanger in Figure 5 is transformed to Figure 6.

## Counterflow Heat and Mass Exchanger

Moving to Figure 6, if a portion of the total airstream I has the product stream II split off, then the *working airstream* III will receive additional heat from the dry channel since the total airstream I is greater then the working airstream III (Maisotenko *et al.*, a). This forces additional evaporation and now $h_{\text{total in}} > h_{\text{product out}}$ or $h_1 > h_{\text{dew point}}$ or there is cooling. The working airstream temperature will increase in the saturated condition moving up the saturation line $\varphi$ on the psychrometric chart shown in Figure 6. At any point across a plate there is cooling of the total dry air $G_{\text{total}}$ (point 1) and evaporation of the *working air* $G_{\text{working}}$ (point 3). For an ideal cycle, the temperature of the working air leaving the exchanger will equal the temperature of the air entering the dry side or $t_1 = t_3$.

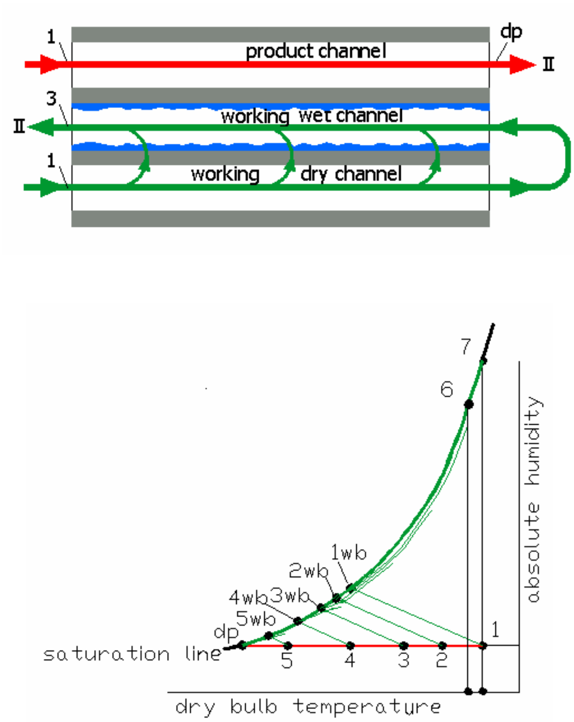


FIGURE 7 Scheme and process of the Maisotsenko cycle with product air cooling.

The ideal cycle heat balance, ignoring fan, pump energy loss, and entering water temperature heat gain or loss, is (see Fig. 6)

$$G_{\text{total}}\ (h_{1\text{in}} - h_{\text{dew point}}) = G_{\text{working}}\ (h_3 - h_{\text{dew point}}) \tag{10}$$

or

$$G_{\text{product}}\ (h_{1\text{in}} - h_{\text{dew point}}) = G_{\text{working}}\ (h_3 - h_{1\text{in}}) \tag{10a}$$

Because of the steep curve of the saturation line $\varphi$, a small increase in the working air humidity on the saturation curve create a larger temperature decrease of the product. Therefore, the flow rate of the working airstream becomes less than the flow rate of the product airstream. The amount of energy that the working air can then remove in an ideal cycle is $h_{3\text{saturated}} - h_1$ and that is also the amount of cooling. In practice, the exhaust enthalpy will always be less the ideal, $h_4$ saturated.

It is interesting to note that the performance of the cycle is thermodynamically dependent on the inlet air conditions since these determine the ideal coldest product temperature (the dew point temperature of the working air), and the ideal hottest temperature the working air can reach (the product air inlet temperature).

A variation of this cycle is to use perforations in the working air channels, shown in Figure 7, to reduce the pressure drop across the exhaust channels. In an ideal cycle, the working air leaving the cooler reaches the temperature of the air entering the dry channel and is saturated. The product air will be at the dew point temperature. The cooling capacity $Q_o$ for this ideal model is

$$Q_{o\ ideal} = G_{product}\ (h_{1\ in} - h_{dew\ point}) \tag{11}$$

The heat balance of the ideal model is

$$G_{total}\ (h_{1in} - h_{dew\ point}) = G_{working\ ideal}\ (h_3 - h_{dew\ point}) \tag{12}$$

or

$$G_{product\ ideal}\ (h_{1in} - h_{dew\ point}) = G_{working\ ideal}\ (h_3 - h_{1in}) \tag{12a}$$

Others have tried to commercialize the concepts of the cycle's inventor, Valeriy Maisotsenko, as shown in Figure 6, but have failed. The problems encountered are that pure counterflow in a plate heat exchanger is not possible due to the geometry of the plates with air entering and leaving on the same sides. The wetting process for working air channels was limited to vertical wicking. The pressure drop for the product and working airstreams were excessive. The plates were required to be ridged enough to give the unit structure, making the materials and manufacturing process too expensive.

## Perforated Crossflow Heat and Mass Exchanger

The Maisotsenko cycle has now been embodied in a perforated heat exchanger (allowing crossflow) to provide both the method and apparatus for a commercially viable dew point evaporative cooler (Maisotenko *et al.*, b; Wicker, 2003). This new system uses the previously described thermodynamic process capable of sensibly cooling not only outside air, but also any fluid below the wet bulb temperature and approaching the dew point temperature of the working gas, or in this case air. The process realizes the objectives of the above counterflow Maisotsenko cycle, but without many of the problems. This cooler is now being commercialized and is available for use in a wide range of heat rejection and recovery applications.

Figure 7 shows the scheme of the new cycle and processes. In this design, the working airstream $G_{working}$ can be outside air or air that is designated for exhaust from the system. It must have a dew point temperature lower then the temperature of the product stream to be cooled to. The working airstream is first passed over the dry side of a plate, where it is precooled, and after to the wet side of the plate. More particularly, the working airstream $G_{working}$ passes through the perforations in the plate over the length of the plate from its dry side to the wetted side, where it then cools the dry side by evaporating water off the wet surface. Simultaneously, the product air (or any fluid) $G_{product}$ is passed along a different portion of the dry side of the same plate as the dry working air and having a heat exchange relationship with the wet side working air. The heat balance (see Fig. 7) where the product and working air enter at the same temperature and humidity, is

$$\text{Ideal} \quad G_{product}\,(h_1 - h_{dew\ point}) = G_{working}\,(h_7 - h_1) \tag{13}$$

$$\text{Real} \quad G_{product}\,(h_1 - h_5) = G_{working}\,(h_6 - h_1). \tag{14}$$

The effectiveness of the product stream to approach the dew point temperature is

$$E = (t_1 - t_5)/(t_1 - t_{dew\ point}). \tag{15}$$

### Perforated Heat Exchanger Design

A heat exchanger consists of several sheets of a cellulose-blended fiber that is designed to wick fluids evenly. One side of each sheet is also coated with polyethylene. The sheets are stacked one on another, separated only by channel guides (see Fig. 8) that are placed on one side of the sheet. The channel guides that are attached to the polyethylene sides of the sheet run along the length of the sheet. The guides that are placed on the noncoated side run along the width of the

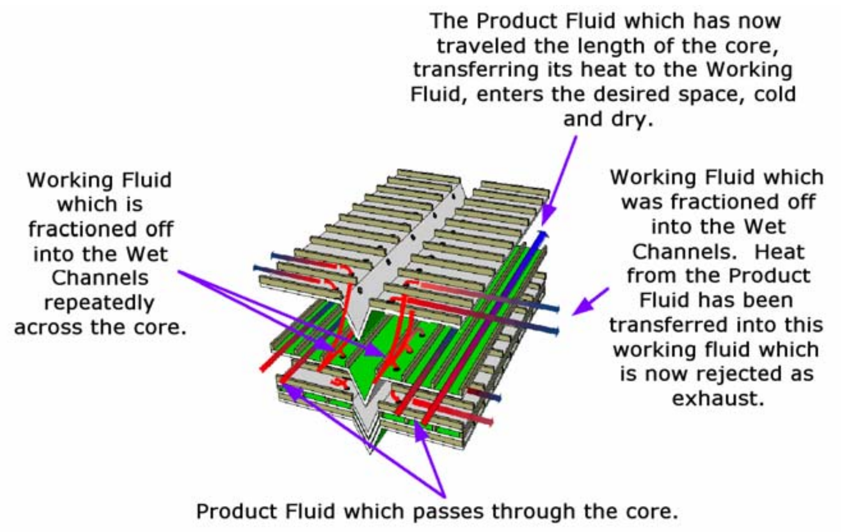


FIGURE 8 Cut away view of the heat and mass exchanger through the Maisotsenko cycle.

sheet. Their purpose is to provide structure as well as guide air movement within the heat exchanger.

When assembled, the coated side of the bottom sheet is placed facing up, while the coated side of the second sheet is placed facing down. These two-coated sheets, when placed together, form a "dry channel." Conversely, the top side of the second sheet, which is uncoated, has a third sheet placed with its un-coated surface down, to form a "wet channel." This pattern of alternating wet channels and dry channels reoccurs throughout the height of the heat exchanger. Within a single heat exchanger, air is divided into the incoming airstream working air (or any other gas in a vapor state) and product air (or any other fluid). The product air is always separate from the working air and remains within dry channels the entire length of the heat exchanger. Consequently, the product air is cooled sensibly as it travels the distance of the core and into the designated cooling space.

The working air channels (the innermost channels) are blocked at the opposite end of the inlet, preventing the air from ever reaching the product air or cooling space. The heat from the working air is removed evaporatively in the wet channels and then exhausted out of the sides of the heat exchanger as fully saturated air.

The cellulous material used in the manufacture of the heat exchanger acts as a natural capillary wick within wet channels. The natural wicking assures uniform wetting within the heat exchanger with no excess fluid, thereby focusing the energy removal on the cooling of the product airstream, not water cooling. The

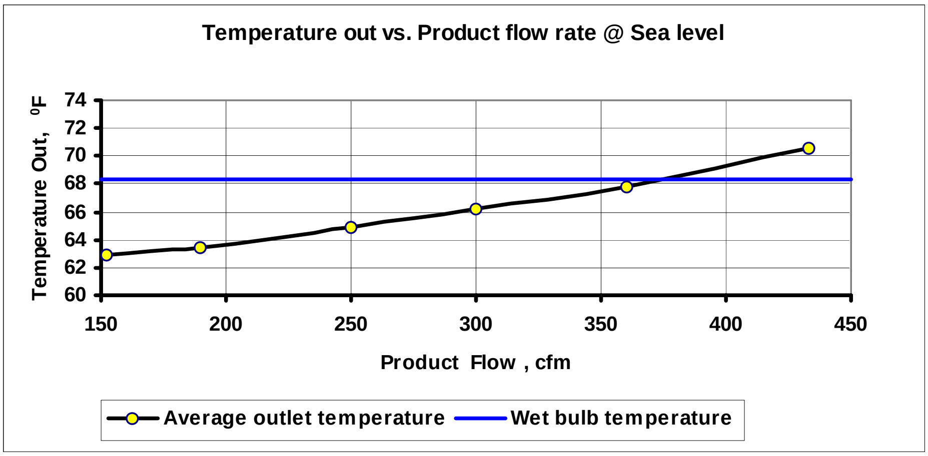


FIGURE 9 Outlet product air temperature versus product airflow at sea level. Extrapolated data for sea level condition: Air absolute humidity = 0.0075–0.008 lbm water/lbm air, exhaust/product airflow ratio = 0.41, and inlet air temperature = $100^{o}$F.

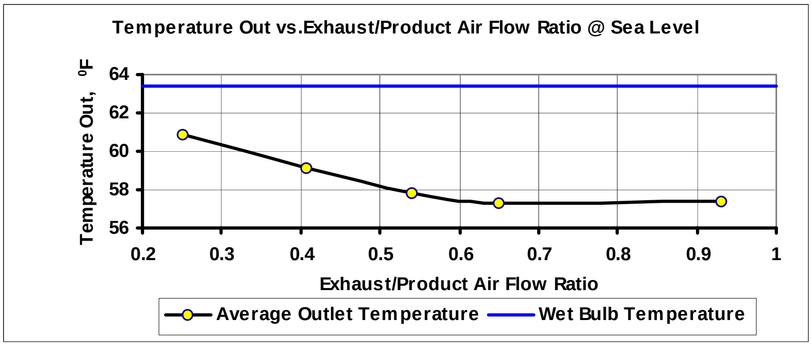


FIGURE 10 Outlet product air temperature versus ratio of the airflow between exhaust and product at sea level. Extrapolated data for sea level condition: Air absolute humidity = 0.0067–0.0071 lbm/lbm, product airflow = 250 cfm, and inlet air temperature = 100°F.

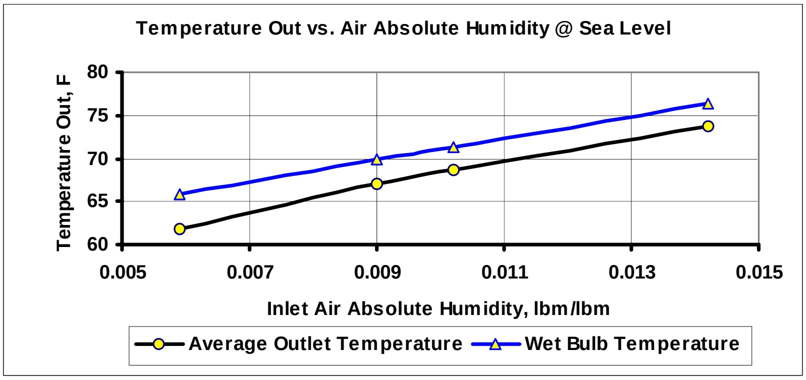


FIGURE 11 Outlet product air temperature versus product air absolute humidity at sea level. Extrapolated data for sea level condition: Product airflow = 250 cfm, exhaust/product airflow ratio = 0.38, and inlet air temperature = 100°F.

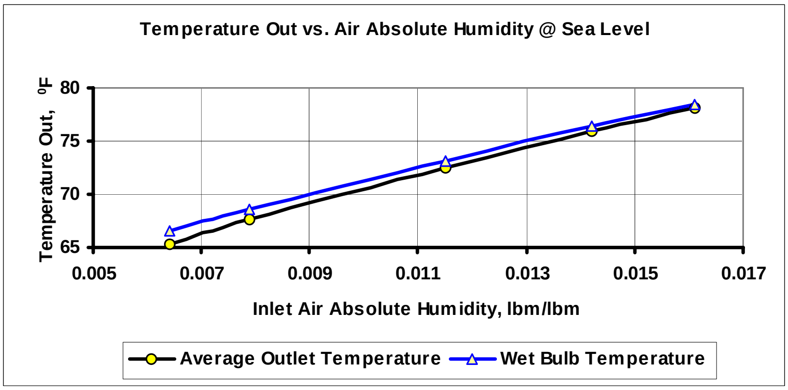


FIGURE 12 Outlet product air temperature versus product air absolute humidity at sea level. Extrapolated data for sea level condition: Product airflow = 240 cfm, exhaust/ product airflow ratio = 0.25, and inlet air temperature = 100°F.

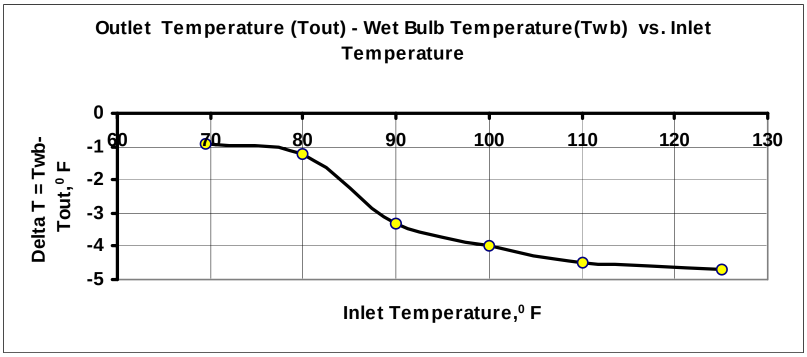


FIGURE 13 Wet bulb temperature ($T_{wb}$) — outlet product air temperature ($T_{out}$) versus inlet temperature: Product airflow = 250 cfm, exhaust/product airflow ratio = 0.38, and air absolute humidity = 0.0055–0.0075 lbm/lbm.

wicking nature of the cellulose material also helps break down the surface tension of the fluid, resulting in a higher mass and heat transfer rate.

The polyethylene is used because of its low thermal resistance through its thickness while maintaining a high thermal resistance along its width and length. Subsequently, the heat transferred from the product air to the working air is concentrated at many points through its thickness, producing a sharp contrast in temperature all along the width and length. This heat transfer occurs multiple times in a short physical space within the same heat exchanger, resulting in a progressively colder temperature as the product air continues to flow across the working air.

## Experimental Research

Experimental research was performed on more than two hundred different designs of this new heat exchanger over the last three years. This research was conducted using a pass-through double test chamber.

An extensive array of thermocouples was used to measure heat transfer rates across each channel within the heat exchanger as well as working air and product fluid. The test results below demonstrate the performance of this new heat exchanger. This data validates the energy savings that can be realized by cooling below the wet bulb temperatures without a compressor or refrigerant.

Several heat and mass exchanger designs make it possible to fit the different applications that are needed in air cooling. In Figures 9–13 are the test results from testing a model 85–20 × 18 × 10 heat and mass exchanger.

Heat exchanger geometrical data:

- Exchanger consist of 85 each 20 × 18 in plates
- Channel spacing, (including plate thickness): dry channel, 0.13 in; and wet channel, 0.11 in

- Stacked 10 in height
- Ten 1 in product channels
- Six 1 in working channels
- Twenty 1 in exhaust channels
- Heat-mass transfer surface = 378 $ft^2$ (35 $m^2$)
- The core has 3 deg slope, V-shape

This test data demonstrate the heat exchanger's efficiency. Note that with higher inlet temperature, the heat flux increases. This is due to the high exhaust enthalpy, especially at the inlet, and is another reason the temperature of the water entering this cooler has little effect on the product temperature. The small amount of water cooling needed is soaked up in the high exhaust air enthalpy.

Consequently, the hotter the temperature of the incoming air, the greater the cooling capacity for any system, using this new heat and mass exchanger.

## CONCLUSION

This newly designed heat and mass exchanger is not directly comparable to any existing heat-rejection/recovery cycle. This heat exchanger combines the advantages of a series of indirect evaporative coolers into a single efficient unit. This new heat and mass exchanger core can be designed to cool any fluid, below the wet bulb temperature and approach the dew point temperature of the working air, without a compressor or refrigerant.

The advanced heat rejection characteristics of this cost-effectively produced heat exchanger can revolutionize the comfort cooling and industrial heat rejection market. Because of its superior heat flux, this new heat exchanger is also ideally suited for use in a desiccant air-conditioning system. Because the air exiting a desiccant wheel (or any liquid or solid desiccant system) can vary in temperature from 90$^o$F, if a heat recovery wheel is used, to 180$^o$F, it must be cooled before it can be used in any conditioned space. Today, desiccant systems are used in series with conventional cooling systems to cool the hot air exiting the desiccant system. This new heat and mass exchanger core can now cool the air with far less energy. The greatest benefit will be realized above freezer cases in grocery stores, in hospitals, schools, and restaurants, where very large quantities of fresh makeup air are required. The National Renewable Energy Laboratory (NREL) is the nation's primary laboratory for renewable energy and it recommends the Maisotsenko cycle, which "significantly reduces electric demand for any cooling applications" (U.S. Department of energy, 2007).

Today the M-cycle assists federal agencies reach their energy-use reduction goals and it has been successfully tested and researched for cooling applications by NREL (FEMP), Delphi, SMUD, PG&E, Sanwa (Japan), etc. Since then, this product received wide recognition from all over the world: Coolerado Cooler won the International 2004 R&D 100 award, the U.S. Green Builder 2006 Top Ten Product award, the 2007 Sustainable Business Silver Medal of Honor award, and just recently, the History Channel and Invent Now Award 2007, the BUILDING PRODUCTS Top 100 Winner for 2008, Governor's Excellence in Renewable Energy Award for 2008 and, just recently, the first certified winner of the UC Davis "Western Cooling Challenge", 2009.

The "Popular Science" announced Coolerado's new hybrid air conditioner through the Maisotsenko cycle, the H80, won a "Best New Green Technology" award (2009).

Coolerado air conditioners can be found in markets around the world — in Japan, Europe, Australia, and South America, Singapore as well as in the USA from Washington to Florida.

## *Nomenclature*

| | |
|---|---|
| $C_p$ | specific heat of the air |
| $E$ | effectiveness |
| $G$ | mass flow rate of dry air |
| $h$ | enthalpy of the air |
| $h_s$ | enthalpy of the water vapor or latent heat from the evaporation of water |
| $Q$ | cooling capacity |
| $t$ | temperature |
| $W$ | humidity ratio or the weight of the water vapor in the air divided by the weight of the dry air |

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