Dew point evaporative cooling: Technology review and fundamentals

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Dew Point Evaporative Cooling: Technology Review and Fundamentals

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ABSTRACT

Used throughout history, evaporative cooling is an effective means of air conditioning in hot and dry climates. Despite its effectiveness, there is not substantial market penetration versus vapor compression systems in more humid climates. This is historically the case, as in its most common form, the direct evaporative cooler, evaporative cooling suffers from substantial water consumption, humidification of supply air, and limited cooling to ambient wet bulb temperatures. The recent development of several innovative evaporative cooling cycles have broken through these traditional technical barriers.

Dew point evaporative cooling, using a novel heat exchanger and flow path arrangement, can deliver unhumidified air below wet bulb temperatures consuming less water than direct evaporative and vapor compression coolers. Supply air temperatures approaching the dew point temperature are achieved in a single-stage unit with cooling capacity independent of the ambient air dry bulb temperature. Recently, a prototype 5 ton rooftop unit delivered 80% energy savings relative to a conventional vapor compression system, demonstrating the potential for dew point evaporative cooling in Zero Energy Design. This paper describes the technology fundamentals of dew point evaporative cooling through said heat exchanger and its context in the technology evolution of evaporative cooling, ranging from direct to multi-stage indirect-direct evaporative cooling, with performance comparisons under common operational conditions.

INTRODUCTION

The primary method of air conditioning currently is a refrigerant-based vapor compression system (VCS) at over 90% of the market (Westphalen 2001). While vapor compression for refrigeration has patents dating back to the early 19th century, it was Willis Carrier who employed this cycle first for air temperature and humidity control in 1902 (*History of the Carrier Corporation* 2006), and the fundamental air conditioning process has changed little since then. The longevity and widespread application of VCS is a testament to its effectiveness.

This widespread use of VCS is not without its drawbacks. Residential and commercial VCS consume 1,304 TWh (Westphalen 2001; DOE 2009) of primary energy, which results in the release of 1,357 million metric tons of carbon dioxide (CO_2) from residential sector and 1,196 million metric tons of CO_2 from the commercial sector. In addition, the common refrigerants currently used in VCS for space cooling are a growing source of concern for their contribution to the climate change. Typical VCS refrigerants include hydrochloroflourocarbons (HCFC) and hydroflourocarbons (HFC), which have global warming potentials (GWP) several orders of magnitude above that of CO_2. For example, two common refrigerants in air-conditioning R-410a and R-134a have a GWP of 2,088 and 1,430 respectively (Leck 2010), where the GWP of a compound is its impact toward climate change scaled relative to CO_2.

An alternative to VCS is evaporative cooling, which has been used throughout history as an effective means of air

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conditioning in hot and dry climates. Evaporative coolers generally have substantially reduced power consumption and installed cost, as both the compressor and copper heat exchanger are eliminated from the system. Additionally the use of high-GWP refrigerants is also eliminated, as the phase change of water is what drives cooling[1].

Despite its effectiveness and relative simplicity, there is not substantial market penetration versus VCS in more humid climates. This is historically the case as in its most common form, the Direct Evaporative Cooler, evaporative cooling suffers from substantial water consumption, humidification of supply air, and limited cooling to ambient wet bulb temperatures. These systems facilitate a direct swap of latent for sensible cooling, thus operate ideally under adiabatic conditions. More complex and non-adiabatic evaporative cooling systems mitigate the drawbacks of the Direct Evaporative Cooler, achieving reduced supply air humidification, enhanced cooling, and more efficient operation. This comes at a cost of increased materials, water consumption, and a greater proportion of rejected "working" air, the fraction of ambient air that is humidified and rejected to drive sensible cooling, requiring increased fan energy. These advanced evaporative cooling systems achieve these improvements with a flow arrangement combining some or all of the following three processes:

- *Direct evaporation* – a humidifying channel that cools through a swap of latent for sensible heat.
- *Indirect evaporation* – two channels, exchanging heat through a common wall, where the channel that contains the product fluid (e.g. supply air) is cooled sensibly by the other channel, a direct evaporator channel.
- *Dehumidification* – increasing the evaporative cooling potential of the working air upstream of the direct and/or indirect evaporator, using a desiccant and/or selective membrane.

Recently new products have reversed this trend and broken through these traditional technical barriers, such as supply air at temperatures below the ambient wet bulb. This sub-wet bulb cooling, hereafter called "dew point evaporative cooling", can be achieved using a novel heat exchanger and flow path arrangement, delivering unhumidified air below wet bulb temperatures while consuming less water than other evaporative coolers. In some cases, supply air temperatures approaching the dew point temperature are achieved in a single-stage unit with cooling capacity independent of the ambient air dry bulb temperature. Recently, a prototype 5 ton rooftop unit delivered 80% energy savings relative to a conventional vapor compression system, demonstrating the potential for dew point evaporative cooling in Zero Energy Design (Kozubal 2009). This paper describes the technology fundamentals of dew point evaporative cooling through said heat exchanger and its context in the technology evolution of evaporative cooling, ranging from direct to multi-stage indirect-direct evaporative cooling, with performance comparisons under common operational conditions.

EVAPORATIVE COOLING TECHNOLOGY SURVEY

Direct Evaporative Cooling technology is a simpler and cheaper alternative to VCS. Used in "swamp coolers" for air conditioning, warm and dry air is passed by a wetted surface, and the latent heat of water vaporization cools and humidifies the air. As defined, the theoretical cooling limit is the wet bulb temperature (t_{WB}) of the incoming air stream. Ideally, this is an adiabatic process in which there is a direct energy swap of the latent heat of vaporization of water to sensible cooling of the supply air and as such there is *no net cooling capacity*[2]. While a simple process, often cooling *and* humidifying the air is not desirable. A diagram of Direct Evaporative Cooling with example psychrometric conditions is shown in Figure 1.

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1 As employed in current evaporative cooling systems, water is not technically a refrigerant as the cycles are open loops and water comes into contact with the cooled fluid (air). At the time of writing, the UN Framework Convention on Climate Change does not recognize water vapor as a greenhouse gas (GHG), thus a Global Warming Potential (GWP) has not been agreed upon.

2 In other words sensible cooling and latent heating are coincident, thus Direct Evaporative Cooling is enthalpy neutral.

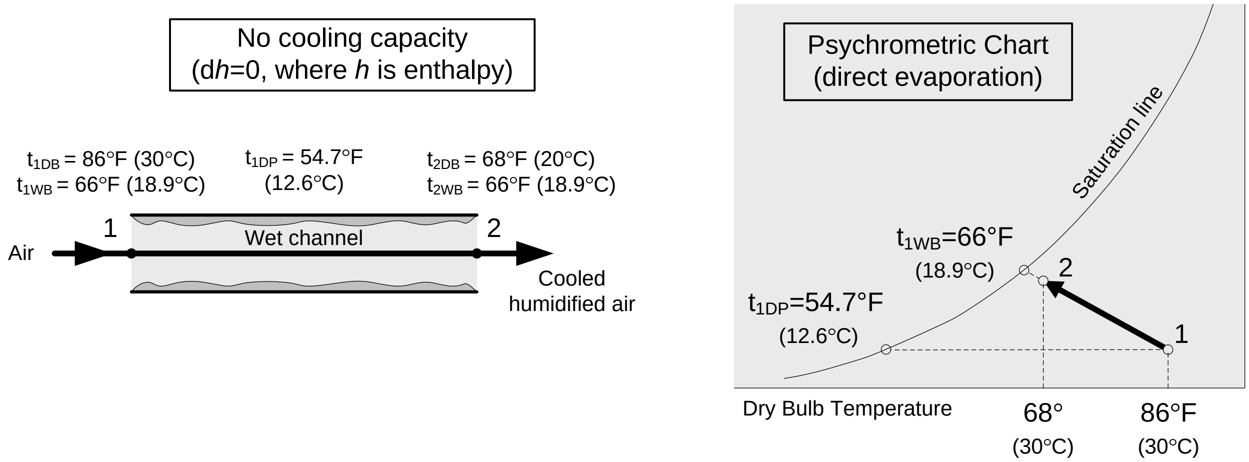

Figure 1 Diagram and example psychrometric chart for Direct Evaporative Cooling

Slightly more complex systems that prevent humidification of the supply air employ **Indirect Evaporative Cooling**. This improvement upon Direct Evaporative Coolers comes at a cost of increased water consumption, decreased efficiency, and increased installed and operating cost. Indirect Evaporative Coolers avoid humidifying the product air through use of a dual-channel sensible heat exchanger. This is also a direct swap of latent for sensible energy, however, the heat exchanger prevents the product air from being humidified. Using the example diagram in Figure 2, the outside air, at points 1 and 3, is split into the following two channels:

- The **wet channel** is effectively a Direct Evaporative Cooler, which ideally humidifies the air to its wet bulb temperature t_{1WB} = 66°F (18.9°C). On the psychrometric chart in Figure 2, this is the dashed line of constant wet bulb temperature from Point 3 to 66°F (18.9°C) on the saturation line. In reality, the air is humidified to higher wet bulb temperature t_{4WB} at Point 4 because of sensible heat transferred from the dry channel.
- The **dry channel** is cooled through the heat exchanger to 72°F (22.2°C) at Point 2. On the chart this moves horizontally at constant absolute humidity from a dry bulb of 86°F (30°C) to 72°F (22.2°C)

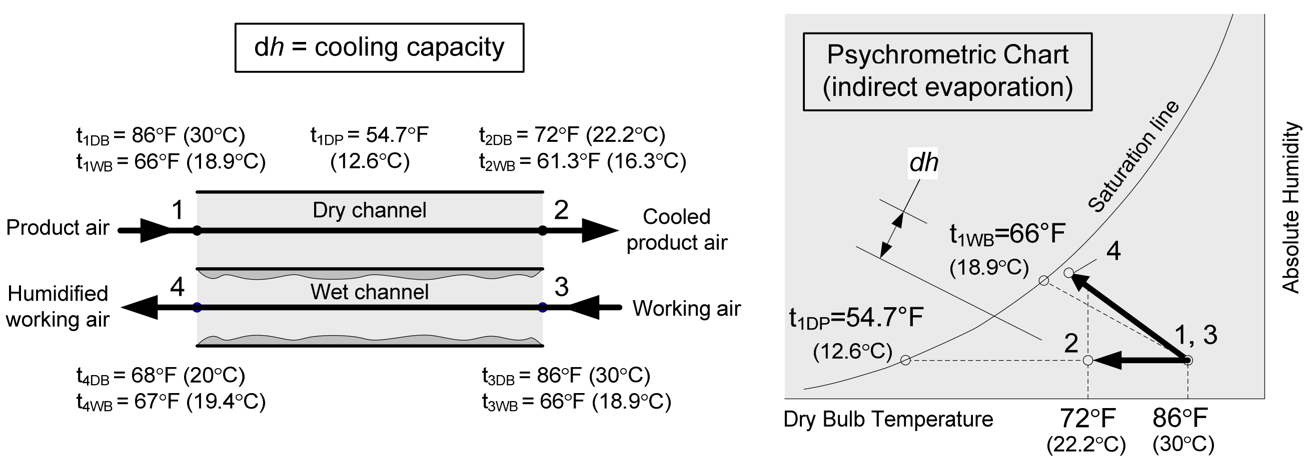

Figure 2 Diagram and example psychrometric chart for Indirect Evaporative Cooling.

Like Direct Evaporative Cooling, Indirect Evaporative Cooling is limited by the wet bulb temperature of the incoming fluid. As such, they often are commercialized as "hybrid" systems that employ a secondary VCS stage to address this capacity limitation, at an additional increased operating and installed cost. While an efficient design, wholly indirect evaporative systems are more energy efficient in providing significant energy savings with sub wet-bulb temperatures.

Dew point cooling (sub-wet bulb) is currently achieved predominantly with **Indirect-Direct Evaporative Cooling**, which combines the two abovementioned evaporative processes in series. With a simple arrangement, Indirect-Direct Evaporative Cooling reroutes the dry channel through a second wet channel. It takes advantage of the fact that when a parcel of air is sensibly cooled, the saturated water vapor pressure decreases, reducing its wet bulb temperature thus increasing its evaporative cooling potential. Note that in the psychrometric chart on Figure 2 this occurred as well; as the wet bulb temperature at Point 2 was lower than that of Point 1. This process is shown in Figure 3 and by examining the psychrometric chart, it is apparent how through this staged sensible (Point 1 to 2) *then* latent cooling (Point 2 to 3), a supply air dry bulb temperature (t_{3DB}) below the incoming wet bulb temperature (t_{1WB}) is achieved. The limit for cooling now is the dew point temperature ($t_{1,2DP}$), which is indicated by the horizontal dotted line extending from Point 2 to the saturation line.

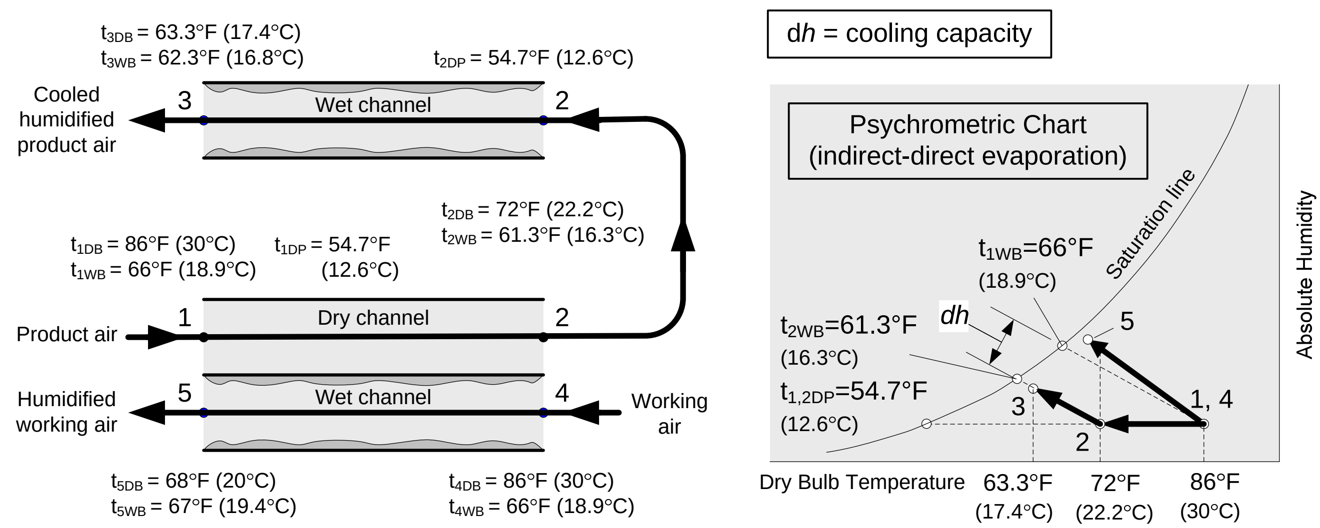

Figure 3 Diagram and example psychrometric chart for Indirect-Direct Evaporative Cooling.

One could add another dry channel atop the supply air wet channel (Point 2 to 3) and perform a second latent for sensible energy swap, which could then facilitate dew point cooling without humidification. This process of adding additional dry-to-wet channels could theoretically repeat *ad infinitum* until the supply air is cooled to the ambient dew point temperature ($t_{1,2DP}$). This is both inefficient with energy and space however, as each combined dry/wet channel pairing must be thermally separated to remain adiabatic, thus increasing the flow path and pressure drop. Additionally fan energy increases, as with each additional dry/wet channel pairing a greater fraction of the incoming air is humidified and rejected, as done at Point 5. Commercially available systems often employed a dehumidification step, drying the working air and/or the cooled and humidified supply air. Generally, installations are engineered systems and this dehumidification step is accomplished through use of a solid desiccant wheel. While delivering high performance, the added dehumidification step adds system cost and the increased water consumption of the Indirect-Direct Evaporative system is not reduced. These thermally activated desiccant technologies are proven to be a viable alternative to VCS, however, the desiccant regeneration process is energy intensive.

These three general classes of evaporative coolers: Direct, Indirect, and Indirect-Direct, represent the majority of the market. Each has their pros and cons, as summarized in Table 1.

Table 1. Pros and Cons for Direct, Indirect, and Indirect-Direct Evaporative Cooling

Evaporative Cooling Type	Relative Cost	Pros	Cons
Direct	Low	- Effective and used throughout history	- No cooling capacity - Cooling limited to the ambient t_{WB}
Indirect	Mid	- Has cooling capacity - Benefits of Direct Evaporative Cooling without supply air humidification	- Cooling limited to the ambient t_{WB} - Working air must be rejected, increasing fan work - Additional material without cooling benefit is often not justifiable
Indirect-Direct	High	- Has cooling capacity - Cooling is not limited to incoming t_{WB}	- Supply air is humidified - Working air must be rejected, increasing fan work - Indirect to direct channels must be separated increasing material and space requirements

DEW POINT EVAPORATIVE COOLING

Rather than add active dehumidification to an Indirect-Direct Evaporative Cooler, an alternative is a Dew Point Evaporative Cooler arrangement that takes advantage of the previously discussed wet-bulb depression in a non-adiabatic design. A compact hybrid (evaporative cooling with supplemental vapor compression) dew point evaporative cooling system was demonstrated with a sensible energy efficiency ratio (EER) in excess of 40, corresponding to a Coefficient of Performance (COP) of 12.0 (Kozubal 2009). When operating with the VCS portion disabled, it maintained a dry bulb temperature difference of 5°F (2.8°C) between exhaust and supply air. For reference, the minimum Seasonal Energy Efficiency Ratio (SEER) for an air conditioning system sold in the U.S. is 13, with Energy Star® systems rated at exceeding a SEER of 14. These SEERs correspond to COPs of 3.4 and 3.7. The Dew Point Evaporative Cooler is an alternative to the Indirect-Direct arrangement, whereby sub-wet bulb cooling of the supply air is performed without humidification. Shown in Figure 4, the Dew Point Evaporative Cooler uses the same wet and dry side of a plate as described in the Indirect Evaporative Cooler but with different airflow arrangement creating a new thermodynamic cycle (often identified as the so-called "Maisotsenko Cycle"). Similar to Indirect-Direct Evaporative Cooling, the t_{DP} is the theoretical limit of cooling.

Two manifestations of Dew Point Evaporative Cooling are shown in Figure 4: (*a*) a Partial Extraction of Air arrangement, which fractions off a portion of the dry channel as usable product and (*b*) a Product Air Cooling arrangement which keeps working and product dry channels separated. Both arrangements are psychrometrically identical while arrangement (*b*) with product channel has essential advantages as compared to arrangement (*a*). Thus, any gas or liquid matter (for example hot waste product) can be used as a product flow in arrangement (*b*) while arrangement (*a*) is limited to the same gas for both channels. Moreover, using the product channel allows reduced pressure drop in the system.

Moving from Indirect-Direct Evaporative Cooling, this Dew Point Evaporative Cooler differs through integration of the working wet and dry channels, in which the working fluid participates in a latent to sensible energy swap *with itself*. The two key characteristics of Dew Point Evaporative Cooling, which facilitate its effective and simple design, are (1) at steady state, the coolest point throughout the cycle is always near the transition from working dry to wet channels, at state 2 in the working dry channel and (2) saturation is reached rapidly and maintained within the working wet channel.

Momentarily ignoring the temperatures shown on the psychrometric chart at Point 2 in Figure 4, consider the following:

1. Worst case scenario, assume that the wall between the working dry and wet channels is infinitely insulated and preventing the sensible cooling of the dry channel, rendering Points 1 and 2 identical. Like Direct Evaporative Cooling, Point 3 is saturated, with a t_{DB} equal to the incoming t_{WB}, 66°F (18.9°C). Thus, the best cooling offered to the Product Dry Channel would be slightly hotter, at 68°F (20°C) t_{DB}, no better than Indirect Evaporative Cooling.

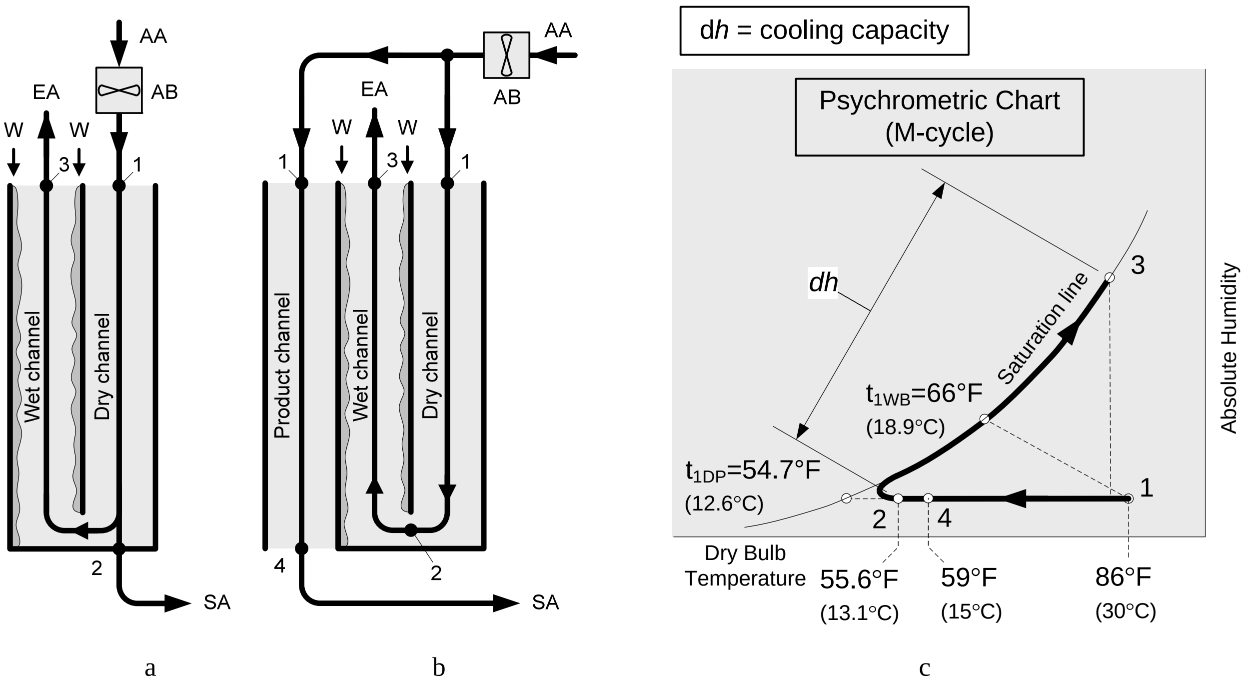

Figure 4: Diagram of Dew Point Evaporative Coolers With Partial Extraction of Air (a) and With Product Channel (b) with Example Psychrometric Chart (c): AA = Ambient Air, AB = Air Blower, EA = Exhaust Air, SA = Supply Air, and W = Water

2. Like Indirect-Direct Evaporative Cooling demonstrated, it is desirable to sensibly cool the working dry channel prior to transitioning to a wet channel, reducing the wet bulb temperature, thus increasing the evaporative cooling potential.
3. Consider now that sensible heat is exchanged across the wall between the working wet and dry channels. Assume that the working dry channel is sensibly cooled to just above the incoming wet bulb temperature, to 68°F (20°C) t_{DB}. This is certainly feasible, as the wet working channel can easily be reduced to 66 °F (18.9°C) t_{DB} as shown in worst case scenario. The t_{WB} is now decreased from Points 1 to 2, from 68°F (20°C) to 60°F (15.6°C). Finally the exiting state at Point 3 for the, is at saturated conditions with a t_{DB} equal to the Point 2 wet bulb temperature, 60°F (15.6°C).
4. Note that the basis for assumed conditions at Point 2, sensibly cooled from Point 1 down to 68°F (20°C) t_{DB}, is that the coolest portion of the working wet channel is 66°F (18.9°C) t_{DB}. Subsequently, this assumption leads to cooler temperatures at Point 2, down to 60°F (15.6°C) t_{DB}. If 60°F (15.6°C) t_{DB} is now the coolest portion of the working wet channel, why cannot Point 2 be cooler than 68°F (20°C) t_{DB}? It can and a useful feedback mechanism develops.
5. Revising the assumed state at Point 2 from 68°F (20°C) to 60°F (15.6°C) t_{DB}, to just above the coolest portion of the working wet channel, this further sensible cooling from Point 1 to 2 drives t_{WB} at Point 2 down further, to 57°F (13.9°C). Thus the working wet channel has more cooling capacity than previously estimated, which requires another revised assumption of sensible cooling from Point 1 to 2, and so on. Continuing these iterations, the coolest portion of the working wet channel moves closer and closer to the dew point of the incoming fluid.
6. Under ideal conditions at equilibrium, Point 2 will reach saturated conditions at the incoming t_{DP}. So long as it is continuously wetted, the working fluid is saturated for the remainder of the working wet channel.

Researchers are examining the fundamentals of such heat and mass transfer arrangements (Maisotsenko, et al., 2001) and alternatives to the Dew Point Evaporative Cooling arrangement shown in Figure 4 (Nasif, 2005 and Zhao, 2008). While there may be room for incremental design improvement and cost-engineering for these Indirect Evaporative Coolers, in terms of reduced pressure drops, increased contact surface area, and reduced cost materials, unavoidable performance barriers remain that include: a limit of cooling to the incoming dew point temperature, water consumption, and limited application in humid climates.

SYSTEM PERFORMANCE COMPARISON

Performance of each evaporative cooling system discussed is estimated, including a baseline Energy Star VCS system, in delivering 600 cfm (1019 m^3/hr) of supply air at 61°F (16.1°C) t_{DB}, for typical Western U.S. conditions of 90°F (32.2°C) t_{DB} and 67°F (19.4°C) t_{WB} at sea level (Note: conditions do not correspond to Figures 1-4). All systems are assumed to operate adiabatically with respect to the surrounding environment. System psychrometric conditions, cooling capacity, air flows, and water consumption is summarized in Table 2. Direct and Indirect Evaporative Coolers are unable to meet target supply air conditions, while Indirect-Direct, VCS, and Dew Point Evaporative systems are.

Table 2. Summarized Performance of Evaporative Cooling Methods and Baseline VCS

Evaporative Cooling Type	Supply Air Out	Working Air Out	Air Flow	Direct Water Consumption
Direct	t_{DB} = 67°F (19.4°C) t_{WB} = 67°F (19.4°C) t_{DP} = 67°F (19.4°C)	N/A	Product Air = 44.1 lb dry gas/min (20 kg dry gas/min) Working Air = N/A	Working Wet Channel = N/A Product Wet Channel = 14.4 lb/hr (6.53 kg/hr)
Indirect	t_{DB} = 72°F (22.2°C) t_{WB} = 61°F (16.1°C) t_{DP} = 54°F (12.2°C)	t_{DB} = 71°F (21.7°C) t_{WB} = 71°F (21.7°C) t_{DP} = 71°F (21.7°C)	Product Air = 44.1 lb dry gas/min (20 kg dry gas/min) Working Air = 55.4 lb dry gas/min (25.12 kg dry gas/min)	Working Wet Channel = 24.6 lb/hr (11.16 kg/hr) Product Wet Channel = N/A
Indirect-Direct	t_{DB} = 61°F (16.1°C) t_{WB} = 61°F (16.1°C) t_{DP} = 61°F (16.1°C)	t_{DB} = 71°F (21.7°C) t_{WB} = 71°F (21.7°C) t_{DP} = 71°F (21.7°C)	Product Air = 45.1 lb dry gas/min (20.45 kg dry gas/min) Working Air = 57.2 lb dry gas/min (25.94 kg dry gas/min)	Working Wet Channel = 25.8 lb/hr (11.7 kg/hr) Product Wet Channel = 7.2 lb/hr (3.27 kg/hr)
Dew Point Evaporative	t_{DB} = 61°F (16.1°C) t_{WB} = 57°F (13.9°C) t_{DP} = 54°F (12.2°C)	t_{DB} = 88°F (31.1°C) t_{WB} = 88°F (31.1°C) t_{DP} = 88°F (31.1°C)	Product Air = 45.1 lb dry gas/min (20.45 kg dry gas/min) Working Air = 14.8 lb dry gas/min (6.71 kg dry gas/min)	Working Wet Channel = 18 lb/hr (8.16 kg/hr) Product Wet Channel = N/A
VCS (COP = 3.7)	t_{DB} = 61°F (16.1°C) t_{WB} = 61°F (16.1°C) t_{DP} = 61°F (16.1°C)	t_{DB} = 105°F (40.6°C) t_{WB} = 72°F (22.2°C) t_{DP} = 54°F (12.2°C)	Product Air = 45.1 lb dry gas/min (20.45 kg dry gas/min) Working Air = 87.9 lb dry gas/min (39.86 kg dry gas/min)	N/A (see below)

While conventional VCS do not directly consume water directly, they do indirectly through electricity consumption. Electricity produced in the U.S. consumes an estimated 2.0 gallons (0.00757 m^3) of water/kWh delivered to the site (Torcellini, 2003). Therefore the VCS with a COP of 3.7 will indirectly consume 25.3 lb (11.47 kg) per hour at performance conditions. The Dew Point Evaporative Cooler, with a COP of 12.0, will indirectly consume 8.0 lb (3.63 kg) water per hour through electricity consumption. Both Indirect and Indirect-Direct Evaporative Coolers require greater direct water consumption than the Dew Point Evaporative Cooler, and as both the air flows and channel lengths (thus pressure drops) are likely to be are greater in magnitude, the COP will be reduced, increasing indirect water consumption. Comparatively, water consumption is lowest for Direct Evaporative Cooling, however the system does not have a cooling capacity, saturating the supply air without meeting target conditions. Similarly, Direct Evaporative Cooling consumes the least energy with the lowest total air flow, however the abovementioned drawbacks persist.

For the remaining systems with cooling capacities, the Dew Point Evaporative Cooler achieves the highest energy efficiency at steady state through its two key characteristics: (1) the coolest point throughout the Dew Point Evaporative Cooling is always near the transition from working dry to wet channels and (2) saturation is reached rapidly and maintained within the working wet channel. The former facilitates sub-wet bulb cooling without humidification of the supply air, driving up the system cooling capacity, and the latter rejects working air at a saturated condition slightly below the ambient t_{DB}, increasing working air enthalpy thus reducing necessary working air flows and water consumption. Overall, this arrangement is an attractive alternative to Indirect-Direct Evaporative coolers that also deliver air at 61°F (16.1°C) dry bulb, as it delivers unhumidified supply air requiring 45% less water consumption and 74% less working air. For the remaining evaporative systems, the working air is saturated however heat rejected is limited to incoming wet bulb temperatures. The VCS is limited in that it uses ambient air for heat rejection on the condenser side. In both cases, a larger working air flow is necessary for heat rejection to the working air, requiring greater fan work and water consumption.

CONCLUSION

The primary theory and performance of dew point evaporative cooling through a compact heat and mass exchanger is discussed in the context of the technology evolution of evaporative cooling, ranging from direct to multi-stage indirect-direct evaporative cooling. Performance comparisons under common operational conditions are estimated, including a baseline vapor compression system, which suggest that Dew Point Evaporative Cooling systems yield energy and water consumption savings in hot and dry climates without supply air humidification. This design discussed achieves improved results through (1) sensibly cooling dry working air prior to the transition to wet channel, thus depressing the t_{WB} and subsequently (2) retaining saturated conditions at the heat sink, air within the working wet channel.

NOMENCLATURE

t = Temperature; *Subscripts: DB* =Dry Bulb, *WB* = Wet Bulb, *DP* = Dew Point

REFERENCES

ASHRAE. 2009. ASHRAE Handbook: Fundamentals.

History of the Carrier Corporation, *www.corp.carrier.com* (2006).

Kozubal, E. and Slayzak, S. "Technical Report: Coolerado 5 Ton RTU Performance – Western Cooling Challenge Results." National Renewable Energy Laboratory (8/09). http://www.nrel.gov/docs/fy09osti/46524.pdf

Gillan, L. "Maisotsenko Cycle for Cooling Processes" Clean Air Journal, Vol. 9, pp. 1-18 (2008).

Leck, T. "Development and Evaluation of High Performance, Low GWP Refrigerants for AC and Refrigeration". DuPont Flourochemicals R&D. Int'l Symposium on Next-Generation Air Conditioning and Refrigeration Technology (2/10).

Maisotsenko et al, U.S. patents No 6,497,107; 6,581,402; 6,705,096; 6,776,001; 6,779,351; 6,854,278; 6,948,558; 7,007,453; 7,197,887; 7,228,669.

Maisotsenko V., Gillan L., "Evaporative cooling fundamentals. "Idalex" coolers advantages and achievements", Proceedings of the 2nd Int'l Workshop: Non-Compression Refrigeration & Cooling, Odessa State Academy of Refrigeration, Ukraine, p. 50-55, (1/01).

Nasif, M. et al. "Membrane based enthalpy heat exchanger performance in HVAC system". Applied Membrane Science & Technology 2, 31-46 (12/05).

Torcellini, P. et al. "Consumptive Water Use for U.S. Power Production". National Renewable Energy Laboratory (12/03).

Westphalen, W. and Koszalinkski, S. "Energy Consumption Characteristics of Commercial Building HVAC Systems, Volume 1: Chillers, Refrigerant Compressors, and Heating Systems", Arthur D. Little Report For Office of Building Technology State and Community Programs, Department of Energy (4/01).

Zhao, X. et al. "Numerical Study of a novel counter-flow heat and mass exchanger for dew point evaporative cooling". Applied Thermal Engineering 28, 1942-1951 (2008).