TESTING OF AN EVAPORATIVE COOLING SYSTEM THAT SUPPLIES AIR NEAR THE DEW POINT TEMPEREATURE

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Abstract

For many years it has been known that an evaporative cooling system can be designed to supply cool air close to the dew point temperature of the ambient air. Prototypes have been built to demonstrate the principles and temperatures achievable, but the technology has never been developed due to its high cost and complexity. A new counterflow indirect evaporative cooler has been developed which addresses the disadvantages of previous systems.

This paper gives details of the system and explains the operation to achieve supply air close to the dew point temperature of the inlet air. Furthermore, experimental results are presented and the performance characteristics of the cooler in regards to its outlet temperatures and electrical energy efficiency are discussed.

1. Introduction

Traditional direct evaporative coolers provide cooling in dry climates with low energy through the direct evaporation of water in evaporative media pads. As air is drawn through the evaporative pads, water is evaporated thereby cooling the air to temperatures which may approach the wet bulb temperature of the incoming air. However, while the temperature of the air is reduced, the water evaporated from the pads raises the humidity of the air, partially offsetting the improved comfort achieved by the reduced temperature.

The process of cooling in a direct evaporative cooler is described as adiabatic cooling – a process which takes place with no net energy exchange to the surroundings. In this process, sensible heat in both air and water becomes latent heat in the entrained vapour, and temperatures fall and equalise. This can be explained on a psychrometric chart, which displays the properties of moist air as determined by the temperature and humidity ratio as shown in Figure 1.

An important parameter for consideration of the performance of a direct evaporative cooler is the wet bulb temperature, defined as the temperature measured by a temperature sensor covered with a water moistened wick. When measured correctly, the wet bulb temperature approximates the temperature of adiabatic saturation and so is the lowest temperature that can be obtained by evaporating water into the air. A direct evaporative cooler can only ever produce air which approaches the wet bulb temperature. The extent to which a particular cooler can approach the wet bulb temperature is its “effectiveness”. A well made direct evaporative cooler will have an effectiveness of approximately 85%, and produce temperatures and humidity’s represented by the line in Figure 1.

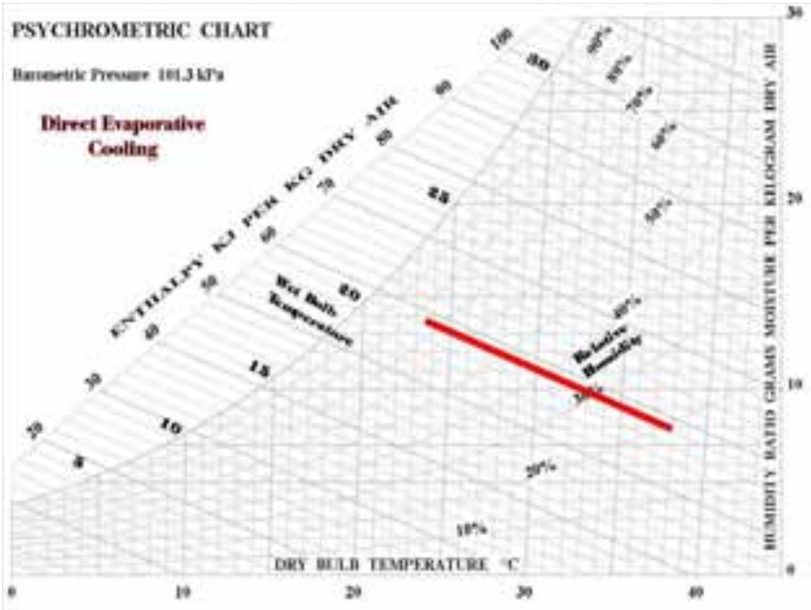

Figure 1: Psychrometric Chart showing direct evaporative cooling

Technologies have existed for some time which utilise the advantages of evaporative cooling to reduce the temperature of air without the addition of moisture [1]. These technologies invariably use some form of heat exchange media between the evaporation of water process and cooling of air. The low temperatures created by the evaporation of water provide a temperature differential to enable the cooling of air across a heat exchange barrier, which enables the transfer of heat from the air stream to be cooled without any direct contact with wet surfaces. The conditioned air is thereby cooled without the addition of moisture to the air stream. This technology is generally known as indirect evaporative cooling.

A configuration for indirect evaporative cooling comprising counterflow heat exchange in adjacent wet and dry air passages was proposed as early as 1976 by Maisotsenko et al in Soviet Patent No. SU620745 lodged 9 September 1976 [2]. The configuration presented demonstrated that temperatures of the incoming air could approach its dew point temperature, which is considerably lower than the wet bulb temperature which is the limit achieved with direct evaporative cooling. Over the years research has been conducted and prototypes built to demonstrate the principles and temperatures achievable, but the technology has never been developed due to its high cost and complexity [3,4].

The original principles demonstrated by Maisotsenko to maximise the heat exchange process and achieve super-cooling (i.e. cooling below the wet bulb temperature) of the delivered air have been adapted in a new cooler. The construction of the plate heat exhanger uses a special medium with the characteristics of high water retention and wickability as the wet channel, while the dry channel uses a moisture-impervious membrane. The construction allows for automation of the manufacturing process for the “core” heat exchanger, and allows for intense heat exchange and super-cooling via water evaporation, to take place within the core. The result is a construction which is effective as an indirect evaporative cooler and has a low cost to manufacture.

In the configuration used to construct the novel cooler, a proportion of the air which has been heat exchanged to a lower temperature is returned along the wet channel as illustrated schematically in Figure 2.

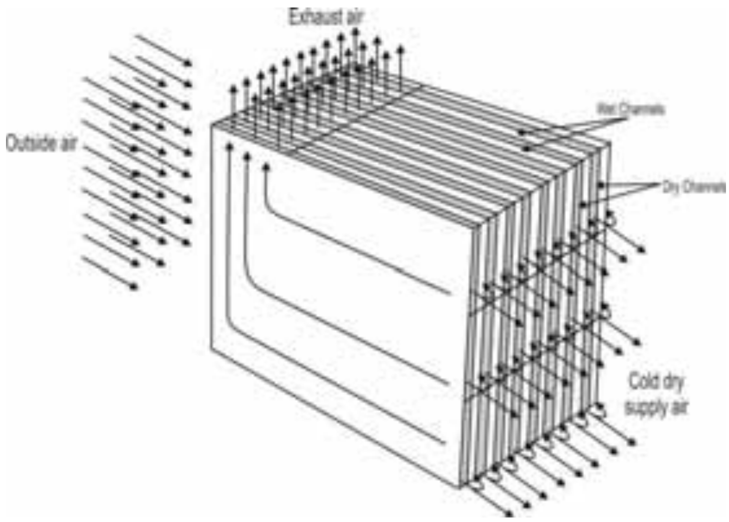

Figure 2: Indirect cooler schematic.

Since the air returned has a depressed wet bulb temperature, evaporation on the wet surfaces of the wet channel will produce temperatures approaching the now lower wet bulb temperature. This significantly lower temperature then further intensifies the heat exchange to the dry channel, further reducing the wet bulb temperature of the proportion returned to the wet channel. This process continues throughout the heat exchanger matrix continually intensifying the heat exchange and evaporation processes until the exit temperatures start to approach the dew point of the incoming air. The cooling process is represented on a psychrometric chart in Figure 3.

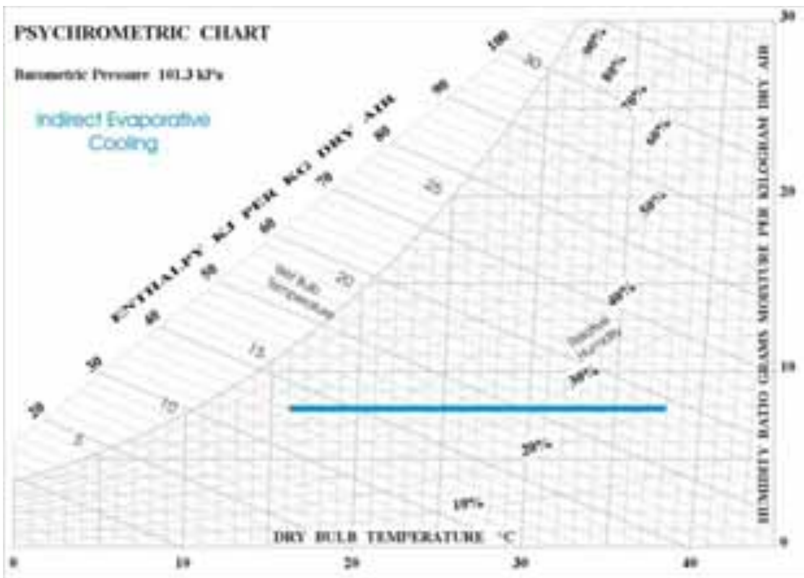

Figure 3: Psychrometric Chart showing indirect evaporative cooling.

In the example shown in Figure 3, air has entered the cooler at the nominal design condition of 38°C DB/21°C WB corresponding to a humidity ratio of 8 g/kg. Cooling has taken place along a constant humidity ratio line with the final delivered air condition of 16°C DB/13°C WB and the humidity ratio still 8 g/kg. The final delivered temperature is actually below the entering wet bulb temperature, resulting in a wet bulb effectiveness of greater than 100%. In the limit, with ideal heat exchange and evaporation, the temperature delivered can approach the dew point of the incoming air.

The cooler is shown schematically in Figure 4 with its major components. Outside air is drawn in to a backward curved centrifugal fan, which pressurises the air ahead of the indirect cooling core. The pressurised air passes through alternate passages within the core created from a moisture impervious membrane. These passages are the dry passages. Air emerges from the core into a plenum space, with flow to the delivery restricted by an orifice plate. The restriction provided by the orifice plate creates a static pressure in the plenum space, which drives a proportion of the air emerging from the dry passages back through the passages separating the dry passages. These passages have a moisture absorbent surface kept wet by a water distribution system, and are known as the wet passages. Air in the wet passages passes through these passages until it emerges from the exhaust, and is then discarded.

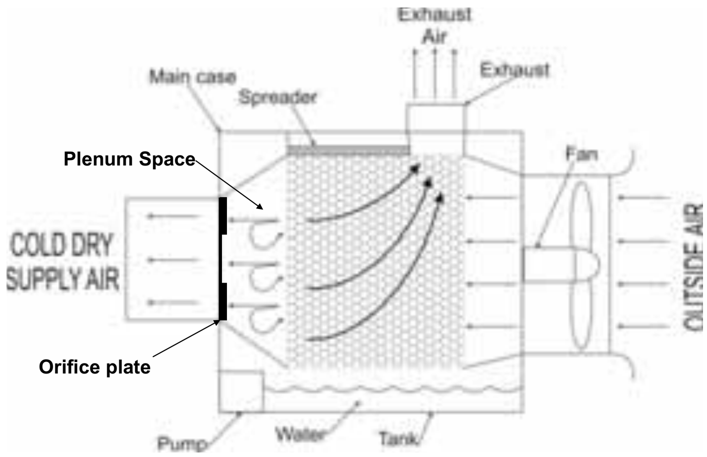

Figure 4: Indirect Cooler schematic

The cooling capacity $\dot{Q}$ of the cooler in kW is defined as:

$$\dot{Q} = \dot{V} \times \rho \times C_p \times (T_{db,in} - T_{db,out}) \quad (1)$$

where $\dot{V}$ = air flow rate delivered by cooler (excluding exhaust) (m^3/s)
ρ = air density (kg/m^3)
C_p = specific heat of air = 1.023 (kJ/kgK)
$T_{bd,in}$ = dry bulb temperature into the cooler (°C)
$T_{db,out}$ = dry bulb temperature delivered by the cooler (°C)

With refrigeration coolers the Energy Efficiency Ratio (EER) for cooling is determined from the cooling capacity produced and input electrical power ($\dot{W}_{in}$). The same can be applied to indirect evaporative coolers using equation (1) to calculate the cooling capacity and measuring the total electrical energy consumed. Hence,

$$EER = \frac{\dot{Q}}{\dot{W}_{in}} \quad \text{in units of kW/kW.} \tag{2}$$

The dew point effectiveness (ε_{dp}) is defined as follows:

$$\varepsilon_{dp} = \frac{\left(T_{db,in} - T_{db,out}\right)}{\left(T_{db,in} - T_{dp}\right)} \quad (\%) \tag{3}$$

where T_{dp} is the dew point temperature entering the cooler.

2. Experiment equipment and test procedure

A cooler was installed to pre-cool the fresh air of an existing refrigerated air conditioning system in a commercial building in Adelaide.

A datalogger was used to monitor the dry bulb temperature of air at the inlet and outlet, humidity of air at the inlet and outlet, electrical power consumption, water consumption and air flow rate delivered to the building. The data collected enabled a number of performance parameters to be determined, namely the supply air flow rate, cooling capacity, power consumption, EER and wet bulb effectiveness.

3. Results and discussion

Figure 5 is a typical graph obtained for inlet & outlet dry bulb air temperatures on a hot dry day (9th March, 2008). The maximum ambient temperature reported by the Bureau of Meteorology about 5 km from the site was 40.2^{0}C. The cooler was able to cool the air from above 40^{o}C down to around 15^{o}C.

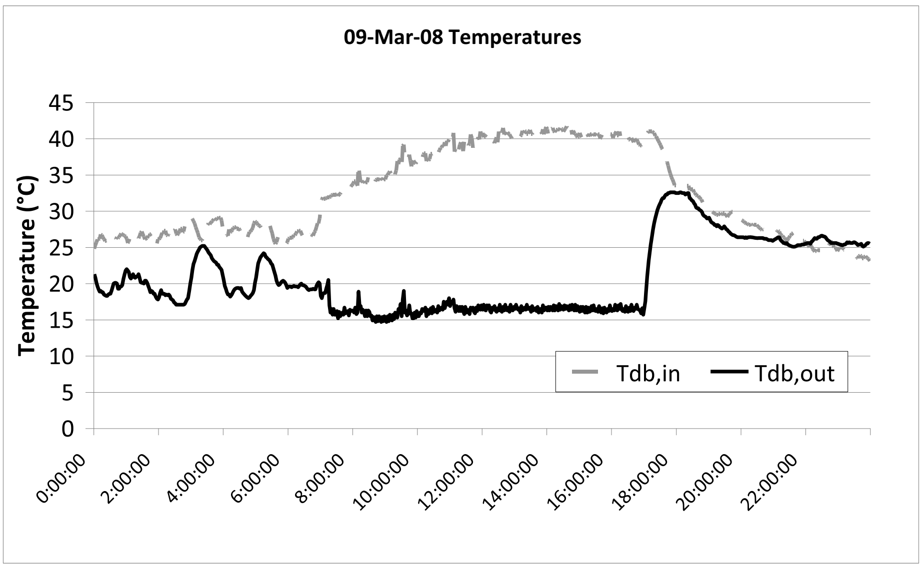

Figure 5: Inlet & outlet dry bulb air temperatures on a hot day.

Figure 6 is the typical corresponding graph for the cooling capacity and EER for the conditions above. The cooling capacity and EER are referenced to the air temperature into the cooler. The cooling capacity tends to start around 15 kW in the morning and increases to over 20 kW in the afternoon. The EER tends to start around 8 and increases to over 12 in the afternoon. The average cooling capacity for the day was 19.7 kW and the average EER was 11.5. Assuming an EER of 3.0 for the refrigeration plant, the net input energy saving from using the indirect cooler for this day was 96.8 kWh.

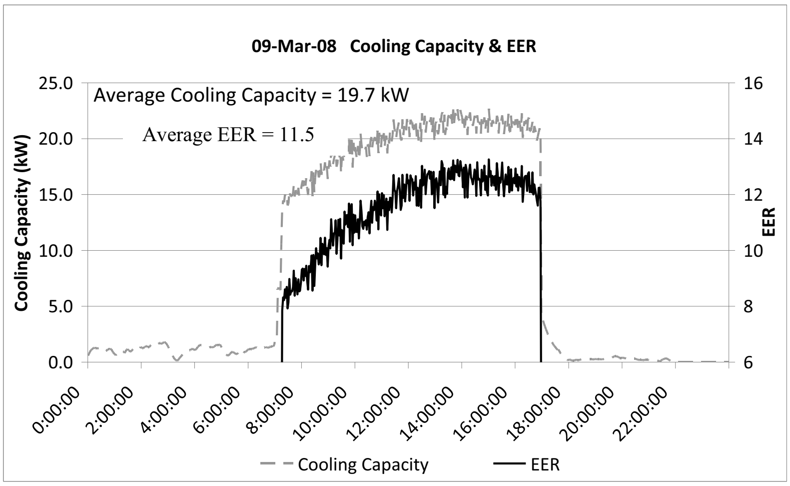

Figure 6: Cooler cooling capacity and EER (referenced to inlet air temperature) on a hot day.

Figure 7 summarises all the data collected for the cooler installed in Adelaide. A total of 44 days was analysed and each point represents one day. The daily average ambient temperature (whilst the cooler was operating) varied from 22.5°C up to 40.3°C. The daily average ambient humidity (whilst the cooler was operating) varied from 10% up to 55%. The x-axis in Figure 7 is the average difference of dew point and dry bulb temperature of ambient air for the day while the unit was operating. The y-axis is the corresponding average EER measured for the day whilst the unit was operating. The graph shows that there is a very good linear relationship which can be represented by the following equation:

$$EER = 0.3433 \times (T_{db,in} - T_{db,out}) - 0.117 \qquad (4)$$

The linear relation suggests that this equation can be used to calculate the EER for any condition where the dry bulb and dew point temperature is known. This equation is only valid for the installation at this site and only when the unit is operating at the constant full speed setting used for this cooler. The average cooling capacity during operation can be determined by multiplying the EER calculated in equation (4) by the motor input power of 1.73 kW.

Figure 8 summarises the daily average outlet air temperature versus the corresponding daily average ambient dew point temperature for the cooler installed in Adelaide. The lowest average daily outlet temperature obtained was 14.7°C. Most of the days the air temperature delivered was below 18°C. There were only 2 days where the average outlet temperature was above 20°C (being 20.3°C and 21.8°C). On these days the ambient air was humid and the daily maximum temperature was about 32°C. The average air temperature delivered by the cooler for all 44 days (for which data was recorded) was 17.3°C.

Figure 9 summarises the average dew point effectiveness measured for all the days tested, which ranged from 0.57 to 0.74, and an overall average of 0.65.

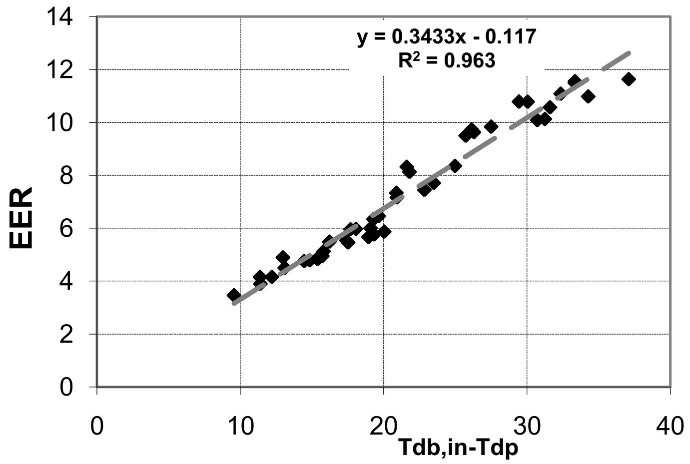

Figure 7: EER versus the difference of dew point and dry bulb temperature of ambient air. Each point corresponds to the average for the day whilst the unit was operating.

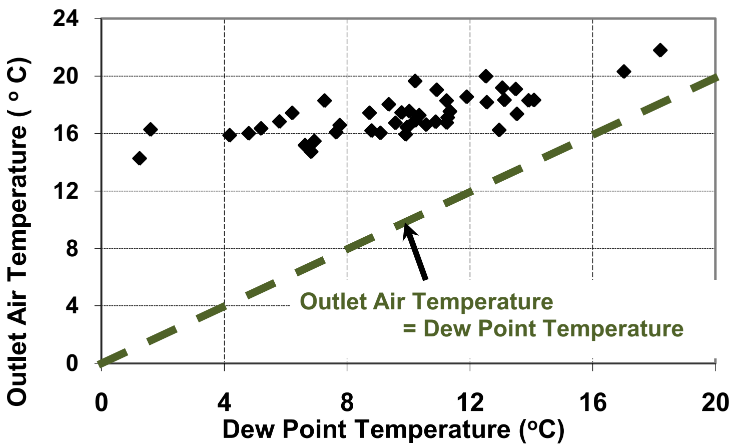

Figure 8: Daily average outlet air temperature versus the daily average ambient dew point temperature for the days analysed in Adelaide.

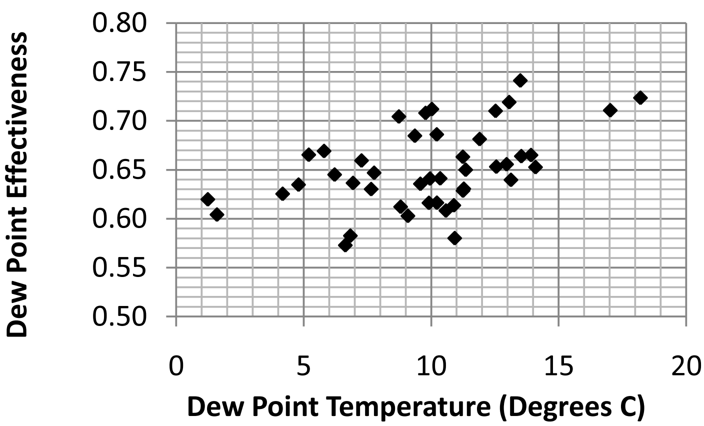

Figure 9: Daily average dew point effectiveness versus the daily average ambient dew point temperature for the days analysed in Adelaide.

4. Conclusion

Experimental results have been presented for the operation of a counterflow indirect evaporative cooler. The cooler was used to pre-cool the fresh air supplied to an existing refrigerative air conditioning system of a commercial building in Adelaide.

On days with a maximum temperature above 35°C, the average cooling capacity over the day was in the range of 12.4 to 19.9 kW. The EER was in the range of 7.2 to 11.5. On warm days with a maximum temperature of around 27°C, the cooling capacity of the unit was around 7.2 to 8.5 kW and the EER was in the range of 4 to 5. The lowest average daily outlet temperature from the cooler obtained was 14.7°C. Most of the days, the air temperature delivered was below 18°C. There were only 2 days where the average outlet temperature was above 20°C (being 20.3°C and 21.8°C). On these days the ambient air was humid and the daily maximum temperature was about 32°C. The average air temperature delivered by the cooler for all 44 days analysed was 17.3°C.

The average EER for a given day ranged from 3.5 to 11.6. The performance of the coolers, expressed in terms of EER, was found to be a linear relationship with the difference between air temperature and dew point. This relationship demonstrates that cooling performance increases dramatically when weather conditions are both hot and dry. This direct correlation enables the performance of the unit to be calculated for different locations using the dry bulb and dew point temperature.

It has been demonstrated that the counterflow indirect evaporative cooler can supply air at a temperature comparable to refrigeration systems at higher thermal efficiencies. This technology is one solution for reducing the energy and associated greenhouse gas emissions for space cooling.

Acknowledgement

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