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Performance analysis of two-stage evaporative coolers

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Abstract

An experimental rig of two-stage evaporative cooling unit is constructed and tested in the Kuwait environment. The system is formed of an indirect evaporative cooling unit (IEC) followed by a direct evaporative cooling unit (DEC). The system is operated during the summer season of Kuwait with dry bulb temperatures higher than 45 °C. The system is operated as a function of the packing thickness and water flow rate of the DEC unit. Other parameters include the water flow rate to the IEC unit and the mode for operating the IEC heat exchangers. Results show that the efficiency of the IEC/DEC varies over a range of 90–120%. Similarly, the efficiency of the IEC unit is varied over a range of 20–40%. The efficiency of the DEC unit varies over a range of 63–93%. The data is used to correlate the Nusselt number for the air stream outside the IEC heat exchange unit. Results show that the Nusselt number varies over a range of 150–450, which corresponds to a heat transfer coefficient of 0.1–0.4 kW/m^2 K. These results are consistent with literature data and should be of great value in designing of IEC units. In summary, the results and analysis presented here indicate the attractiveness of the EC systems for indoor air conditioning. High efficiency is obtained irrespective of the high intake air temperature. It is highly recommended to continue research investigations in this area in search for low energy consumption air conditioning units as these would contribute to a cleaner environment and would conserve the limited resources of fossil fuel.

Keywords: Evaporative cooling; Indirect/direct evaporative cooling; Mechanical vapor compression; Air conditioning

1. Introduction

Global developments and technological progress during the twentieth century is associated with massive urban expansion. Large buildings, shopping areas, hospitals, and schools necessitated adoption of indoor air conditioning systems. Urban developments in Kuwait and the Gulf States were also associated with massive use of indoor air conditioning. This is because of the long duration of the hot. summer season, which extends for more than 8 months. In addition, long spells of high temperature and high humidity conditions are encountered. Considering the above, a major part of the energy consumed in Kuwait and the Gulf States is due to indoor air conditioning. As a result, the air conditioning units consume more than 70% of the installed capacity of power generating units.

The most effective and common form for air conditioning is the mechanical vapor compression system (MVC). This system uses chlorofluorocarbons (CFCs) as the working fluid or refrigerant. The MVC system provides the required cooling range for various ambient conditions that includes hot, arid, or humid. A main disadvantage of the MVC system is the use of CFCs compounds as the working fluid. CFCs have harmful effects on the environment. Also, CFCs contribute to the green house effect that result in global warming and increase in the ambient temperature. In addition CFCs destroy the upper ozone layer, which shields the earth surface from ultraviolet rays. Advantages of the MVC system include the following:

- Wide cooling range, which is suitable for hot, arid, or humid conditions.
- Availability in various sizes and cooling capacities. MVC units are available to control the indoor air temperature for a single room, an entire house, office buildings, shopping malls, hospitals, etc.

Irrespective of the wide spread for the MVC systems, its main disadvantage is the high specific power consumption. As a result, many of the power generating units required for operating the MVC units during the summer time remain idle during the winter season. This situation has motivated many researchers to consider the more efficient, sustainable, and environmental friendly sources for power generation and air cooling. In this regard, evaporative cooling (EC) configurations provide large energy savings over the MVC system.

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Nomenclature

A	heat transfer area of the IEC heat exchangers (m^2)
d	tube diameter of the IEC heat exchanger (m)
G	air mass flow rate across the DEC packing (kg/s)
h_a	heat transfer coefficient of the air stream ($kW/m^2 K$)
L	water mass flow rate across the DEC packing (kg/s)
Nu	Nusselt number, $Nu = h_a d/\lambda_a$ (dimensionless)
Re_w	Reynolds number for the water stream flowing inside the heat exchange tubes of the IEC unit, $Re_a = \rho_a V_a d/\mu_a$ (dimensionless)
Re_a	Reynolds number for the air stream flowing outside the heat exchange tubes of the IEC unit, $Re_a = \rho_a V_a d/\mu_a$ (dimensionless)
T_1	dry bulb temperature of the inlet air (°C)
T_2	wet bulb temperature of the inlet air (°C)
T_7	dry bulb temperature of the air after the second IEC heat exchange unit (°C)
T_{12}	wet bulb temperature of the outlet air (°C)
T_{w7}	wet bulb temperature at after the IEC heat exchangers (°C)
V	velocity (m/s)

Greek symbols

δ	thickness of the DEC packing (m)
ε_{IEC}	efficiency of the IEC unit $\varepsilon_{IEC} = (T_1 - T_7)/(T_1 - T_2)$
ε_{DEC}	efficiency of the IEC unit, $\varepsilon_{DEC} = (T_7 - T_{11})/(T_1 - T_{w7})$
$\varepsilon_{IEC/DEC}$	efficiency of the IEC/DEC unit, $\varepsilon_{IEC/DEC} = (T_1 - T_{11})/(T_1 - T_2)$
λ	thermal conductivity (kW/m K)
μ	viscosity (kg/ms)
ρ	density (kg/m^3)

Subscripts

a	air
w	water

Also, hybrids of MCV and EC are found to give lower specific power consumption than the stand-alone MVC units. In summary, gains to be made from adopting such schemes include the following:

- Reduction in the required power generation capacity. This would reduce the associated expenses of the capital and operating cost of newly installed power plants.
- Reduction in the amount of power consumption, which would allow for use of the saved power in other applications or exporting this power to neighboring countries.
- Reduction in power generation is beneficial since it reduces the combustion rates of fossil fuels and reduces associated emissions of air pollutants.
- EC units use water as the working fluid, which has no harmful effect on the environment. This is one of the main advantages over the MVC system, which uses CFCs.
- The EC manufacturing technologies are simple and can be adopted by all countries.

Irrespective of the above merits for the EC system, it is used on a limited scale due to the following:

- Limited number of manufacturers, design, and cooling capacities.
- The stand-alone EC units have a narrow cooling range for hot and humid ambient conditions.

The following is a summary for a number of literature studies on the performance of the IEC and DEC systems:

- Brown [1] showed drastic reduction in the total energy cost in the air conditioning system, upon proper selection and combination of the evaporative cooling units.
- The analysis performed by Supple and Broughton [2] showed that achieving the required indoor comfort condition depends on the following factors:
  - Type of the conditioned space, i.e. offices, homes, or storage areas.
  - The wet bulb temperature of the ambient air.
  - The type of air conditioning unit, i.e. combinations of direct evaporative cooling, indirect evaporative cooling and mechanical refrigeration.
- Supple and Broughton [2] suggested the following configurations to achieve indoor comfort for office space.
  - Direct evaporative cooling for wet bulb temperatures below 15 °C.
  - Direct/indirect evaporative cooling for wet bulb temperatures between 15 and 18 °C.
  - Indirect evaporative cooling coupled with mechanical refrigeration for wet bulb temperatures between 18 and 24 °C.
  - Mechanical refrigeration for wet bulb temperatures above 24 °C.
- Giabaklou and Ballinger [3] studied passive evaporative cooling, which makes use of natural draft for air movement rather than the use of motor driven fans. This design is more suitable for residential buildings and is more attractive than the stand-alone natural draft or mechanical air conditioning systems. The system resulted in a maximum indoor air temperature of 25.9 °C in comparison with a maximum ambient temperature of 35.8 °C. The average air velocity is 0.28 m/s, average relative humidity

is 73.2%, and average temperature is 24.18 °C, which is close to the comfort range.

- Evaporative cooling is used to lower temperature and humidity of air in greenhouses, Vollebregt and de Jong [4]. The unit includes air-to-air plate type heat exchange system. The airside of the greenhouse flows on the dry side of the plates. The other side of the plates is covered by wet cotton wicking to generate the evaporative effect in the secondary air stream. Data show the evaporative cooling unit results in reduction of greenhouse air temperature as well as its humidity. The dehumidification of the greenhouse air is more efficient at low air flow rates; at such conditions the latent heat transfer is more than twice the sensible heat transfer.
- Peterson and Hunn [5] studied the performance of a simulated air conditioning loop, which includes two loops for secondary and primary air. Each loop includes a circulation fan, cooling coil, a heating coil, a humidification/dehumidification unit. The two loops are attached through the indirect evaporative cooler unit, where the secondary is exposed to a stream of water spray. Results show that, the system has a higher thermal efficiency by 70% over conventional air conditioning units. In addition, the required system capacity to condition the same volume is 12% less than conventional systems.
- Hunn and Peterson [6] studied the effect of the building type and location on performance of indirect evaporative cooler. The study is conducted in three locations in Texas, USA, at different climate conditions, which include Dallas, El-Paso, and. Houston. The building types include schools, restaurants, and retail stores. The study shows high energy savings in climates with low wet bulb temperatures, El-Paso, and in addition savings are highest in restaurants, followed by retail stores and schools. This is because schools are shut clown during the summer months.
- Scofield and DesChamps [7] studied characteristics of direct and indirect evaporative cooling units, which utilize plate type air-to-air heat exchanger. The first stage of the system contains an indirect evaporative cooling unit, which includes a plate type heat exchanger. In this unit, ambient air, with low wet bulb temperature is sprayed with water before it flows in the plate heat exchanger against indoor air (primary air). This results in reduction of the primary air temperature. Further conditioning of the primary air is achieved in a conventional cooling tower. Operation of this system shows monthly savings of 30% in the energy cost over conventional refrigeration systems.
- Goswami et al. [8], Mazzei and Palombo [9], reported similar results for performance enhancement of a mechanical refrigeration unit in Florida, USA, and Italy, respectively. In both studies, evaporative cooling is combined with the existing mechanical refrigeration system. Energy savings of 20% are realized in the combined system, which allows for payback periods of 1.47–2.15 years for the evaporative cooling system. These results are found to depend on the design value of dry and wet bulb temperatures.
- Al-Juwayhel et al. [10] studied the performance of an indirect/direct evaporative cooling system and the effect of coupling the system with a cooling tower. Results show that the highest thermal efficiency is obtained for the combined system, which is followed by a two-stage indirect/direct evaporative cooling unit. The lowest thermal efficiency is reported for the direct evaporative cooling system. In the combined system, the cooling tower removes the thermal load added to the system during air precooling and as a result higher thermal efficiency is achieved.

The main objective of this study is to evaluate the performance of EC units in the Kuwait, environment. This study includes experimental and theoretical analysis of a small-scale EC system. This includes evaluation of system efficiency as a function of operating conditions and evaluation of the heat transfer coefficient for the exchange unit in the IEC unit. These results will be of great value in the design of energy efficient configurations for indoor air conditioning in hot climates.

2. Apparatus

The air cooling system is shown in Fig. 1. The system comprises a metal frame, a water basin, a circulation pump, an air blower, packing material, two heat exchangers, water spray nozzles, siding sheets of Plexiglas, connection tubes, valves, base wheels, and a number of measuring devices. The measuring devices include water flow meters and temperature thermocouples.

The system dimensions are 0.83 m × 2.6 m × 2 m in width, length, and height respectively. The water basin has dimensions of 0.83 m × 0.6 m × 0.32 m in width, length, and height. The amount of water accumulated in the basin is limited to 0.15 m^3. This water volume is necessary to maintain steady state operation, since the water floater is adjusted to a water height of 0.3 m. This is necessary to limit the effects of warmer make-up water, which may affect the steady state conditions of the system.

As is shown the water loop starts at the circulation pump, which transports water from the water basin to the water spray nozzles of the DEC unit and the heat exchangers of the IEC unit. The system of nozzles allows for break down of the water stream into fine mist, which is evenly distributed over the packing material. Therefore, the water flow from the top of the packing towards the basin is in a cross flow direction to the air stream. The air loop is simple and is controlled by a constant speed air blower, which is located at end of the system. The air blower moves the air across the heat exchangers of the IEC unit and then through the DEC unit.

As is shown in Fig. 1 the packing material is divided into two layers. Each layer has the same cross-section area as the metal frame. This is to minimize air bypassing, which

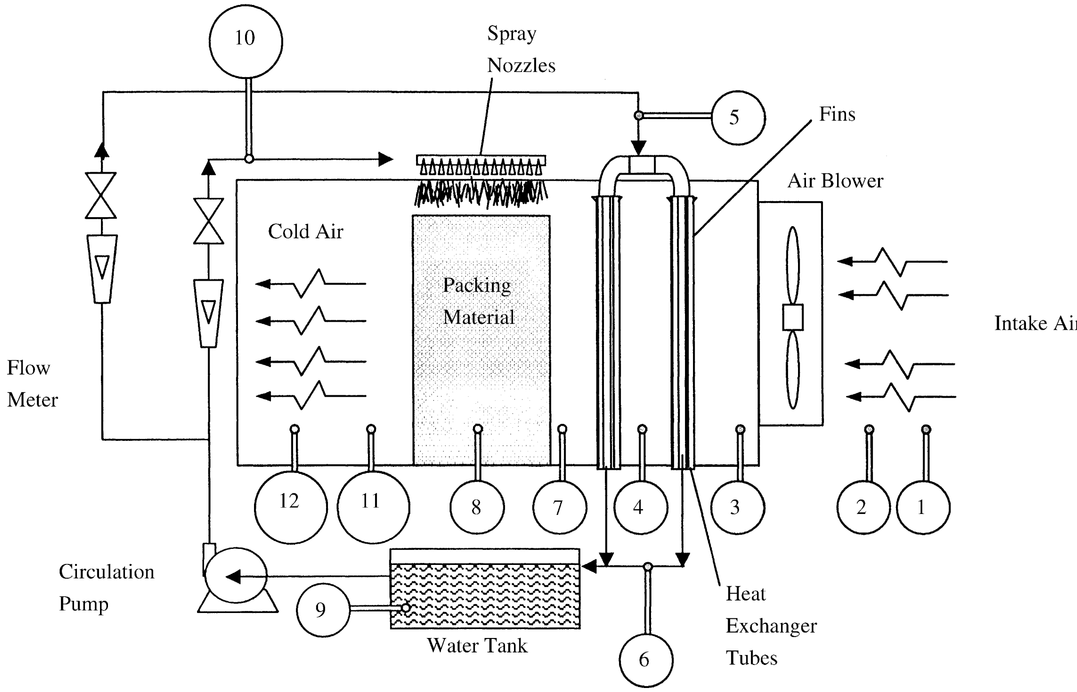

Fig. 1. Schematic of the IEC/DEC evaporative cooling system.

would result in reduction of the cooling efficiency. The packing thickness is varied during measurements with values of 0.2, 0.3, and 0.4 m. Also, this thickness does not generate a high pressure drop within the system. The packing material is selected to provide proper distribution of the water stream. Also, the packing material gives sufficient contact area between the air and water streams.

Two heat exchangers are installed in the system. The configuration allows for operation of the two units in series or parallel as well as operation of a single unit. The two heat exchangers are identical and each has dimensions of $0.89\,\text{m} \times 0.6\,\text{m} \times 0.16\,\text{m}$ in length, height, and width, respectively. Each heat exchanger has 18 copper tubes that has a double U-shape, which gives three rows. The tube inner diameter is 0.0254 m and the wall thickness is 0.0015 m. The average length for each tube row is 0.83 m, which gives a total surface area of $4\,\text{m}^2$.

Structured packing material made of high density polythene with a specific area density of $420\,\text{m}^2/\text{m}^3$ is used in the DEC unit. The packing is originally acquired in the form of large blocks with dimensions of $3\,\text{m} \times 3\,\text{m} \times 0.05\,\text{m}$ in length, width, and thickness. The blocks are cut into the proper dimensions, width and length, which fits the column dimensions. Also, the cut block is assembled to the desired thickness with values of 0.1, 0.2, and 0.4 m. The packing material is light and has a high specific surface area as well as low pressure drop per unit length. These characteristics are reflected positively on the design and performance of the DEC unit. The low density of the packing material simplifies the details of the system metal structure. The high specific surface area also implies use of a small packing volume to achieve the desired thermal efficiency. Finally, the low pressure drop through the packing results in a low power requirement for the air blower.

The packing material has been used in previous studies for performance evaluation of indirect/direct evaporative coolers and cooling towers, El-Dessouky [11], and Al-Juwayhel et al. [10]. The material proved to be resilient and does not promote scaling or fouling. The material has sufficient mechanical strength, which keeps its integrity and does not break down or generate fragmented matter.

Twelve thermocouples are used to measure the following temperatures:

(1) The ambient air dry bulb.
(2) The ambient air wet bulb.
(3) The dry bulb temperature of the air before the IEC unit.
(4) The dry bulb temperature of the air after the first IEC unit.
(5) The water temperature entering the IEC unit.
(6) The water temperature leaving the IEC unit.
(7) The dry bulb temperature of the air after the second IEC unit.
(8) The water/air temperature in the packing of the DEC unit.
(9) The temperature in the water tank.
(10) The water temperature entering the DEC unit.
(11) The dry bulb temperature of the air after the DEC unit.
(12) The wet bulb temperature of the air after the DEC unit.

Other instrumentation includes the following:

- Water flow meter for the IEC unit. The flow meter range is from 0.05 to 0.5 kg/s.

- Water flow meter for the DEC unit. The flow meter range is from 0.05 to 0.5 kg/s.
- Water level controller for the water tank.
- Air velocity meter.

System operation includes the following steps:

- Automatic calibration of the data logging system and the thermocouples connected to the unit.
- Filling of the water tank.
- Operation of the air suction blower and the water circulation pump.
- Adjustment of water flow rates to the IEC and DEC systems.
- Selection of the desired heat exchange configuration, which includes single heat exchanger, two heat exchangers in series, or two heat exchangers in parallel.
- Data collection is performed at an interval of 1 min, where all the measured temperatures are logged sequentially.
- Each setting for the heat exchanger configuration and flow rates is kept constant for a period of 6 h. Subsequently, a new setting is made and a new data set is collected.

The experiments are conducted for the following conditions:

- Four flow rates for the DEC system, which include 0.1, 0.133, 0.167, and 0.2 kg/s.
- Four flow rates for the IEC system which include 0.1, 0.133, 0.167, and 0.2 kg/s.
- Three heat exchange configurations, which includes a single heat exchanger, two heat exchangers in series, and two heat exchangers in parallel.

3. Definition of system efficiency

The cooling efficiency of the IEC and DEC units is given by the following equation:

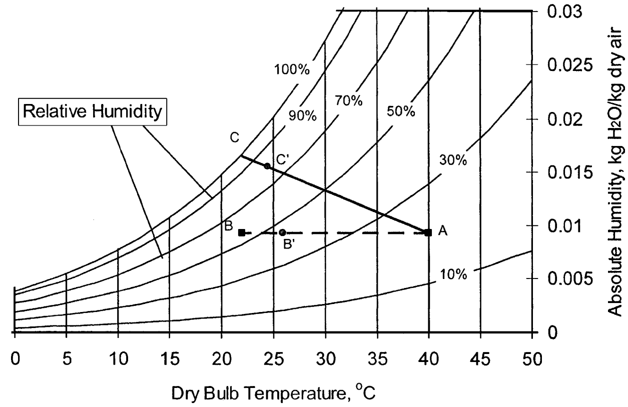

Fig. 2. Cooling paths for indirect evaporative cooler. Paths AB and AC correspond to 100% efficiency. Paths AB′ and AC′ correspond to an actual operating efficiency.

$$\varepsilon = \frac{T_{d_i} - T_{d_o}}{T_{d_i} - T_{w_i}} \tag{1}$$

where ε is the cooling efficiency, T_{d_i} and T_{d_o} are the inlet and outlet dry bulb temperatures of the air stream, and T_{w_i} the inlet wet bulb temperature of the air stream.

Figs. 2–4 show the cooling paths of the IEC, DEC, and IEC/DEC systems, respectively. The cooling paths AB, AC, and ABC correspond to ideal operation, where the outlet air temperature is equal to the wet bulb temperature of the intake air. In actual operation, the cooling efficiency is lower. This because of heat losses with the surroundings, incomplete saturation of the air stream, and bypass of the air or water streams. The actual cooling paths correspond to the segments A′B′, A′C′, and A′B′C′. Therefore, the cooling efficiency of the stand-alone IEC and DEC units are lower than one. On the other hand, the cooling efficiency of the combined system may be greater than one. This is because the outlet dry bulb temperature of the air stream can be lower than the inlet wet bulb temperature.

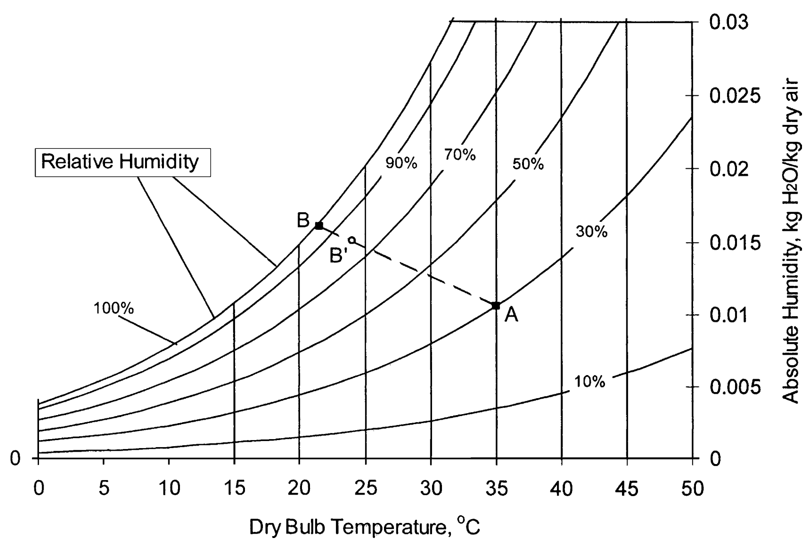

Fig. 3. Cooling path for direct evaporative cooler. Path AB correspond to 100% efficiency and path AB′ correspond to efficiency of actual operation.

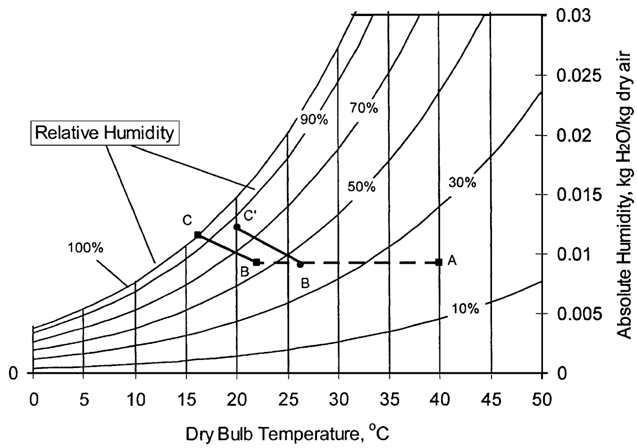

Fig. 4. Cooling path indirect/direct evaporative cooler. Paths AB and BC correspond to 100% efficiency. Paths AB′ and BC′ correspond to an actual operating efficiency.

4. Nusselt number of the IEC unit

The following assumptions are made to model the Nusselt number of the IEC unit:

- Steady state operation.
- Uniform distribution of the air stream around the outside surface of the IEC heat exchange tubes.
- Heat losses to the surroundings are negligible.
- The specific heat at constant pressure, thermal conductivity, and density of the air and water stream are temperature dependent.

The thermal load of the IEC heat exchanger is given by:

$$q_{\mathrm{IEC}} = q_{\mathrm{w}} = q_{\mathrm{a}} \tag{2}$$

where q_{IEC}, q_{w} and q_{a} are the thermal loads of the IEC heat exchanger, water stream flowing on the tube side, and air stream flowing outside the tubes, respectively.

The thermal loads of the water and air streams are given by:

$$q_{\mathrm{w}} = M_{\mathrm{w}} C_{p_{\mathrm{w}}} (T_6 - T_5) \tag{3}$$

$$q_{\mathrm{a}} = M_{\mathrm{a}} C_{p_{\mathrm{a}}} (T_3 - T_7) \tag{4}$$

where M_{w} is the mass flow rate of the water stream (kg/s), M_{a} the mass flow rate of the air stream (kg/s), $C_{p_{\mathrm{w}}}$ the specific heat at constant pressure of water (kJ/kg K), $C_{p_{\mathrm{a}}}$ the specific heat at constant pressure of air (kJ/kg K), T_5 the inlet water temperature to the IEC heat exchanger (°C), T_6 the outlet water temperature from the IEC heat exchanger (°C), T_3 the air temperature before the IEC heat exchanger (°C), T_7 the air temperature after the IEC heat exchanger (°C).

The heat transfer equation of the IEC heat exchanger

$$q_{\mathrm{IEC}} = U \times A \times \mathrm{LMTD} \tag{5}$$

where A is the heat transfer area (m^2), LMTD the logarithmic mean temperature difference (K), and U the overall heat transfer coefficient (kW/m^2 K).

The IEC heat exchanger operates in a cross flow pattern. Therefore, the LMTD in Eq. (5) is defined by:

$$\mathrm{LMTD} = \frac{T_3 - T_7}{\ln\left[\frac{R}{R+\ln(1-\mathrm{RP})}\right]} \tag{6}$$

where R and P are given by:

$$R = \frac{T_6 - T_5}{T_3 - T_7}$$

$$P = \frac{T_3 - T_7}{T_3 - T_5}$$

The overall heat transfer coefficient, U, in Eq. (5) is defined by

$$\frac{1}{U} = \frac{1}{h_{\mathrm{w}}} \frac{A_{\mathrm{o}}}{A_{\mathrm{i}}} + R_{\mathrm{f_w}} \frac{A_{\mathrm{o}}}{A_{\mathrm{i}}} \frac{A_{\mathrm{o}}\delta}{A_{\mathrm{i}}\lambda} R_{\mathrm{f_a}} \frac{1}{h_{\mathrm{a}}} \tag{7}$$

Where h_{a} is the airside heat transfer coefficient (kW/m^2 K), h_{w} the water-side heat transfer coefficient (kW/m^2 K), $R_{\mathrm{f_w}}$ the waterside fouling resistance (m^2 K/kW), $R_{\mathrm{f_a}}$ the airside fouling resistance (m^2 K/kW), λ the thermal conductivity of the tube wall (kW/m K), A_{o} the outside heat transfer area of the IEC heat exchange unit (m^2), A_{i} the inner heat transfer area of the IEC heat exchange unit (m^2), δ the wall thickness of the heat exchanger tubes (m).

The waterside heat transfer coefficient is obtained from well-known Dittus–Bolter equation:

$$h_{\mathrm{w}} = 0.023 (Re_{\mathrm{w}})^{0.8} (Pr_{\mathrm{w}})^{0.4} \left(\frac{\lambda_{\mathrm{w}}}{d}\right) \tag{8}$$

where Pr_{w} is the Prandtl number of water, $Pr_{\mathrm{w}} = \rho_{\mathrm{w}} C_{p_{\mathrm{w}}}/\mu_{\mathrm{w}}$ (dimensionless), Re_{w} the Reynolds number of water, $Re_{\mathrm{w}} = \rho_{\mathrm{w}} V_{\mathrm{w}} d/\mu_{\mathrm{w}}$ (dimensionless), V the water velocity (m/s), ρ_{w} the water density (kg/m^3), μ_{w} the water viscosity (kg/ms), λ_{w} the water thermal conductivity (kW/m K), d the tube diameter (m).

The following procedure is used to calculate the Nusselt number of the air stream on the outside of the IEC heat exchange tubes:

- The temperatures at points 3, 5, 6, and 7 are obtained from experimental measurements.
- The tube diameter, length, and wall thickness are measured as well the total number of tubes. This data is used to determine the outside and inside heat transfer areas.
- The fouling resistances are specified for water and air.
- The thermal load of the heat exchanger is obtained from either Eq. (3) or (4).
- The LMTD of the heat exchanger is obtained from Eq. (6).
- The overall heat transfer c is calculated from Eq. (5).
- The heat transfer coefficient of the water stream is obtained from Eq. (8).
- The heat transfer coefficient of the air stream is obtained from Eq. (7).
- The Nusselt number for the air stream is then calculated from $Nu_{\mathrm{a}} = h_{\mathrm{a}} d/\lambda_{\mathrm{a}}$

5. Error analysis

Uncertainty analysis for the experimental results presented in this study is estimated by the method proposed by Coleman and Steele [12]. The uncertainty in measurements gives the precision and errors in each variable as well as propagation in the experimental results. The error analysis includes measured temperature, relative humidity, water flow rate, and air velocity.

Applying the above method to the measured data show the following errors: 2.4% for temperature, 3.15% for water flow rate, 2.87% for air velocity, and 3.56% for the relative humidity. On the basis of these errors, the calculated uncertainties of the adiabatic efficiency are 5.4, 6.35, and 8.62% of the true values for the DEC, IEC, and IEC/DEC systems, respectively. On the other hand, the error in the Nusselt number for IEC and IEC/DEC systems are 7.86 and 11.2%, respectively.

6. Results and discussion

Experimental measurements are made for the following conditions:

- Water flow rates to IEC unit are 0.267, 0.4, 0.53, and 0.67 kg/s.
- Water flow rates to DEC unit are 0.267, 0.4, 0.53, and 0.67 kg/s.
- Air velocities are 0.5, 1, and 2 m/s, which gives a mass flow rate of 0.32, 0.64, 1.28 kg/s.
- Packing thickness in DEC unit are 0.2, 0.3, and 0.4 m.
- Two IEC heat exchangers, which can be operated in parallel or serial form. Also a single IEC heat exchanger is used in the experiments. The single, serial, and parallel heat exchangers give total heat transfer areas of 4, 8, and 8 m^2, respectively.

In dimensionless form the above parameters give the following:

- Reynolds number for water inside the IEC heat exchanger tubes, which are operated as a single heat exchanger or as two heat exchangers in series, are 17383, 17578, 18717, and 18483. These values correspond to water flow rates of 0.267, 0.4, 0.53, and 0.67 kg/s, respectively. The Reynolds numbers for the parallel configuration are 8691, 8789, 9359, 9242, respectively. These values are half those for the single and series configuration. This is because the water feed is divided equally between the two exchangers for parallel operation.
- Reynolds number for the air stream is based on the outer tube diameter of the heat exchanger tubes. The Reynolds number values for the air stream are 3000, 1500, and 750, which correspond to air velocities of 2, 1, and 0.5 m/s. It should be noted that the Reynolds number for the air stream varies slightly with temperature. This is because of limited variations in the air density and viscosity.
- The mass flow rate ratios of water to air in the DEC unit, which correspond to the above flow rates, are 0.21, 0.29, 0.31, 0.41, 0.414, 0.417, 0.625, 0.83, 1.25, 1.65, and 2.09.

The following sections include the following data analysis:

- Temperature profiles at various locations within the system.
- Efficiency of the IEC, DEC, and the IEC/DEC units.
- Correlations of the system efficiency for the IEC, DEC, and IEC/DEC units.
- The Nusselt number for the air stream outside the IEC tubes.

7. Temperature profiles

Fig. 5 shows the temperature profiles at points 1, 2, 4, 7, and 11, which correspond to the dry bulb temperature of the inlet air, the wet bulb temperature of the inlet air, the dry bulb temperature after the first heat exchanger of the IEC unit, the dry bulb temperature after the second heat exchanger of the IEC unit, and the outlet dry bulb temperature. This data is for a single IEC heat exchanger and water flow rate of 0.267 kg/s. Common features of this data and data at other operating conditions include the following:

- The highest temperature in the system is that of intake ambient air and the lowest temperature is the wet bulb temperature of the intake air.
- The temperatures at points 4 and 7 are similar. This is because in single heat exchanger configuration, the second heat exchange unit has no effect on the air temperatures.
- The air temperature decreases through the system, where the lowest temperature is that of the outlet air, which is quite close to the wet bulb temperature of the inlet air. This temperature difference decreases with the increase in the water flow rate to the IEC heat exchange unit.
- The temperature drops across the IEC and the DEC units are not equal, where higher drop occurs for the DEC unit.

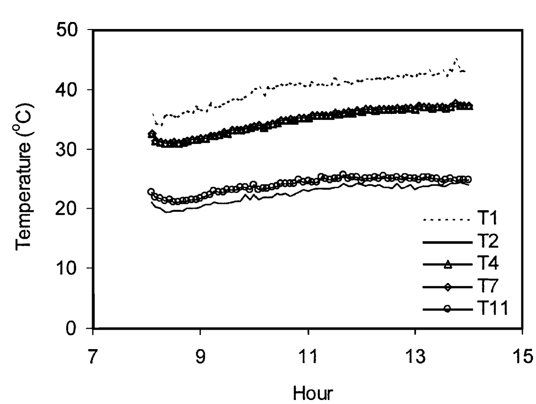

Fig. 5. Temperature transients for single IEC heat exchanger, $M_{IEC} = 0.267$ kg/s, $M_{DEC} = 0.267$ kg/s, and $M_a = 0.32$ kg/s, and $\delta = 0.2$ m.

This reflects the higher efficiency of the direct evaporative cooling process, which is primarily caused by the change in the sensible heat of the air and water streams into latent heat for water evaporation from water into air. This results in larger reduction in the water and air temperatures. As for the IEC unit its efficiency is limited by the heat exchange area and resistance to heat transfer across the heat exchange surface. Unfortunately, the heat transfer coefficient on the airside, which gives a measure for the air resistance to heat transfer, is very small. This implies high resistance to heat transfer.

- The air temperature leaving the EC unit varies with time. This is because of the increase in the ambient air temperature as well as its wet bulb temperature. As shown, the dry bulb temperature reaches high values close to 45 °C around 12–2 p.m.

8. System efficiency results

Efficiency of the IEC, DEC, and IEC/DEC systems are shown in Figs. 6–8, respectively. As shown the efficiency of various configurations reaches steady state conditions after a short period following an initial transient. The highest efficiency is that for the combined system followed by the efficiency of the DEC unit. The lowest efficiency is that for the IEC unit. The data shown in Figs. 6–8 and other measured data give an average efficiency of 30–45% for the IEC unit, 70–85% for the DEC unit, and a combined efficiency for both units well over 90%.

Average efficiencies for the IEC and DEC units are shown in Figs. 9–12. Figs. 9–11 are for the IEC system with two heat exchangers in series, two heat exchangers in parallel, and a single heat exchanger. As shown in Figs. 9–11, the IEC efficiency have stronger dependence on the air flow rate than the water flow. An increase in air velocity increases the IEC efficiency by more than 10%. This is because the major resistance for heat transfer is experienced on the airside rather than the water side. Therefore, an increase in the air velocity lowers the resistance to heat transfer.

The efficiency of the DEC unit has a similar strong dependence on the packing thickness as well as the water flow rate. As shown in Fig. 12 an increase in the packing thickness increases the system efficiency. On the other hand, an increase in the water flow rate decreases the system efficiency. This behavior is explained by an increase in the (L/G)

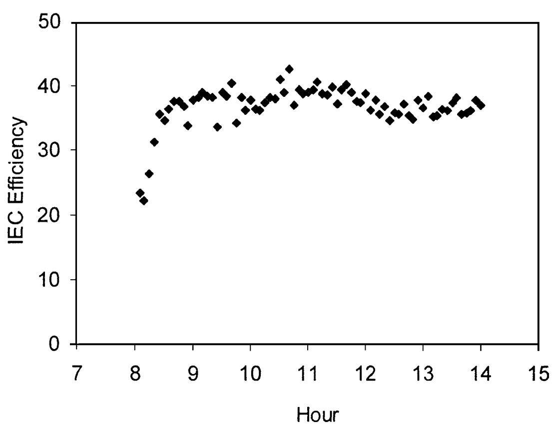

Fig. 6. Efficiency of IEC for single IEC heat exchanger, $M_{\mathrm{IEC}} = 0.4$ kg/s, $M_{\mathrm{DEC}} = 0.267$ kg/s, and $M_{\mathrm{a}} = 0.32$ kg/s, and $\delta = 0.2$ m.

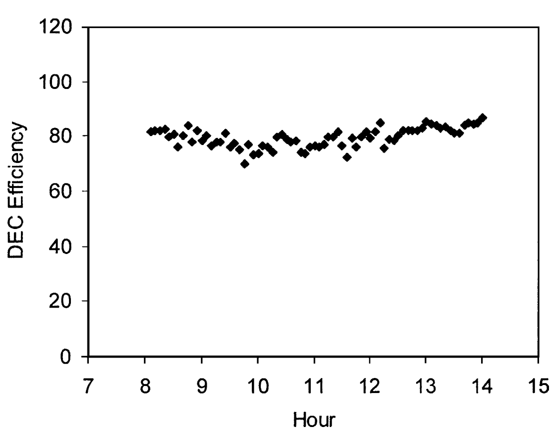

Fig. 7. Efficiency of DEC for single IEC heat exchanger, $M_{\mathrm{IEC}} = 0.4$ kg/s, $M_{\mathrm{DEC}} = 0.267$ kg/s, and $M_{\mathrm{a}} = 0.32$ kg/s, and $\delta = 0.2$ m.

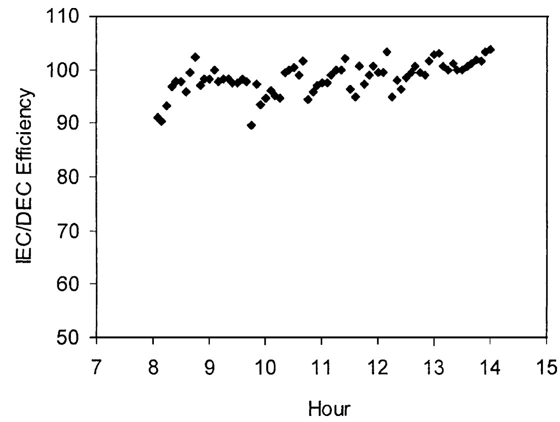

Fig. 8. Efficiency of IEC/DEC for single IEC heat exchanger, $M_{\mathrm{IEC}} = 0.4$ kg/s, $M_{\mathrm{DEC}} = 0.267$ kg/s, and $M_{\mathrm{a}} = 0.32$ kg/s, and $\delta = 0.2$ m.

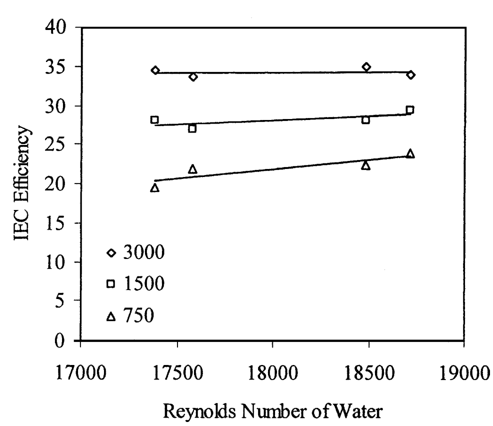

Fig. 9. Variations in the IEC efficiency with a single heat exchanger and as a function of the Reynolds number for air and water.

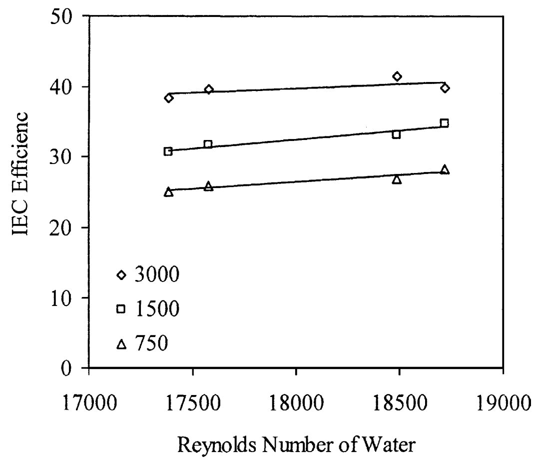

Fig. 10. Variations in the IEC efficiency with two heat exchangers in series as a function of the Reynolds number for air and water.

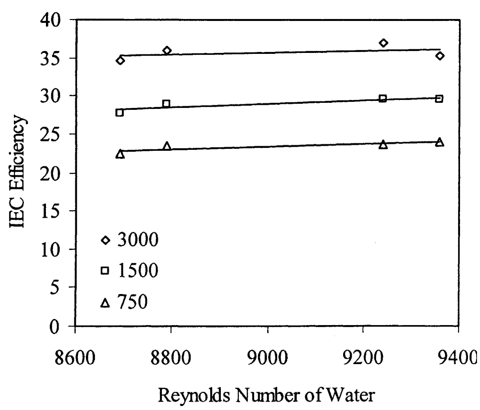

Fig. 11. Variations in the IEC efficiency with two heat exchangers in parallel as a function of the Reynolds number for air and water.

ratio, corresponding to an increase in the water flow rate to the DEC unit at constant air flow rate. Increase in the water flow increases the system thermal load. On the other hand, keeping the air flow rate constant maintains a constant rate of evaporation from the water to the air stream. Therefore, increasing the water flow rate and keeping the evaporation rate constant results in an increase in the air temperature and a reduction in the system efficiency.

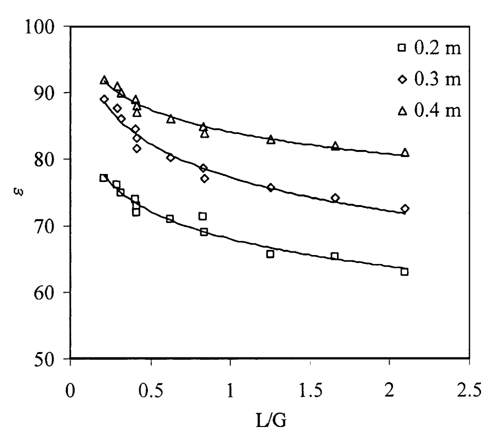

Fig. 12. Variations in the DEC efficiency as a function of the packing thickness and *L*/*G*.

9. System efficiency correlation

The efficiency correlations are developed for the IEC, DEC, and IEC/DEC systems. The correlation for the IEC system is given by the following relation in which efficiency is a function of the heat transfer area of the heat exchangers, the Reynolds number for the water stream inside the heat exchanger tubes, and the Reynolds number for the air stream outside the heat exchanger tubes.

$$\varepsilon_{\mathrm{IEC}} = 1.0819A^{0.185}(Re_{\mathrm{w}})^{0.0181}(Re_{\mathrm{a}})^{0.254} \quad (9)$$

where $\varepsilon_{\mathrm{IEC}}$ is the efficiency of the IEC unit, $\varepsilon_{\mathrm{IEC}} = (T_1 - T_7)/(T_1 - T_2)$, T_1 the dry bulb temperature of the inlet air (°C), T_7 the dry bulb temperature of the air after the second IEC heat exchange unit (°C), T_2 the wet bulb temperature of the inlet air (°C), A the heat transfer area of the IEC heat exchangers (m^2), Re_{w} the Reynolds number for the water stream flowing inside the heat exchange tubes of the IEC unit, $Re_{\mathrm{w}} = \rho_{\mathrm{a}}V_{\mathrm{a}}d/\mu_{\mathrm{a}}$ (dimensionless), Re_{a} the Reynolds number for the air stream flowing outside the heat exchange tubes of the IEC unit, $Re_{\mathrm{a}} = \rho_{\mathrm{a}}V_{\mathrm{a}}d/\mu_{\mathrm{a}}$ (dimensionless), d the tube diameter of the IEC heat exchanger (m).

The coefficient of determination for Eq. (9) is 93.4%. The above correlation is valid over the following ranges: $8700 \leq Re_{\mathrm{w}} \leq 18700$, $550 \leq Re_{\mathrm{a}} \leq 3100$, and $4 \leq A \leq 8$.

Results for this correlation and the measured values, given by the above definition of $\varepsilon_{\mathrm{IEC}}$, are shown in Fig. 13. As is shown in the above correlation the IEC efficiency has positive dependence on the three parameters. This implies increase in the Reynolds number for the air or the water streams increase the IEC efficiency. Also, increase in the heat transfer area increases the IEC efficiency. However, the dependence of the IEC efficiency is stronger on the air mass flow more than on the water mass flow rate. This is because air is the controlling fluid for heat transfer.

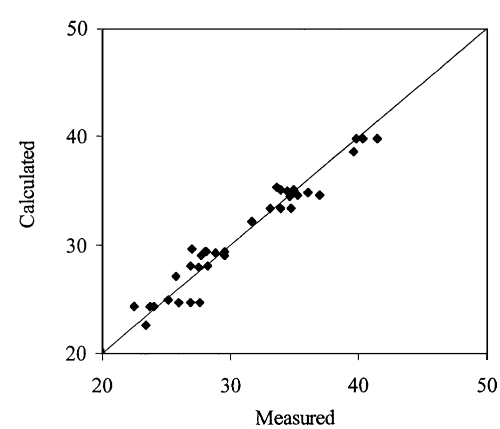

Fig. 13. Measured and calculated efficiency for the IEC unit.

The correlation for the efficiency of the DEC unit is developed as function of the ratio of the water to air mass flow rates and packing thickness in the DEC unit. The correlation is given by:

$$\varepsilon_{\mathrm{DEC}} = 108.8\left(\frac{L}{G}\right)^{-0.077}\delta^{0.287} \tag{10}$$

where $\varepsilon_{\mathrm{DEC}}$ is the efficiency of the DEC unit, $\varepsilon_{\mathrm{DEC}} = (T_7 - T_{11})/(T_7 - T_{\mathrm{w7}})$ (dimensionless), T_{w7} the wet bulb temperature at point 7 (°C), L the water mass flow rate across the DEC packing (kg/s), G the air mass flow rate across the DEC packing (kg/s), δ the thickness of the DEC packing (m).

The coefficient of determination for this correlation is to 97.3%. The above correlation (10) is valid over the following ranges: $0.21 \le L/G \le 2.09$, $0.2 \le d \le 0.4$.

Results for this correlation and the measured values, given by the above definition of $\varepsilon_{\mathrm{DEC}}$, are shown in Fig. 14. It should be noted that the dry and wet bulb temperature for the inlet air of the DEC unit are those measured at point 7 and not point 1 as displayed in Fig. 1. This may explain why the DEC efficiency is higher than the efficiency reported in literature for the stand-alone DEC units.

The efficiency correlation for the combined system is given in terms of the heat transfer area of the IEC heat exchanger, the Reynolds numbers for water and air inside and outside IEC heat exchange tubes, the thickness of the DEC packing, and the mass flow rate ratio of water and air in the DEC packing. This relation is given by:

$$\varepsilon_{\mathrm{IEC/DEC}} = 0.95A^{0.743}(Re_{\mathrm{w}})^{-0.112}(Re_{\mathrm{a}})^{0.247} \tag{11}$$

where $\varepsilon_{\mathrm{DEC}}$ is the efficiency of the DEC unit, $\varepsilon_{\mathrm{DEC}} = (T_1 - T_{11})/(T_1 - T_{12})$ (dimensionless), T_{12} the wet bulb temperature of the outlet air (°C).

The coefficient of determination for this correlation is to 93.4%. Results for this correlation and the measured values, given by the above definition of $\varepsilon_{\mathrm{IEC/DEC}}$, are shown in Fig. 15. It should be noted that Eq. (11) is valid for the same range of parameters of Eqs. (9) and (10).

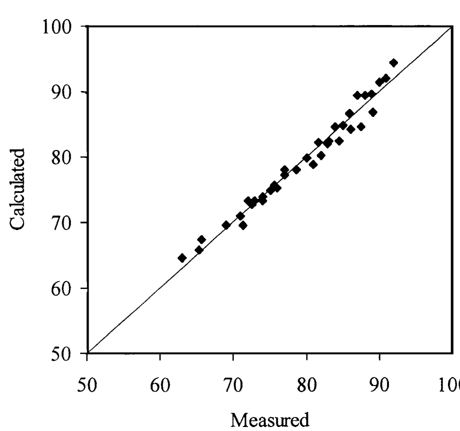

Fig. 14. Measured and calculated efficiency for the DEC unit.

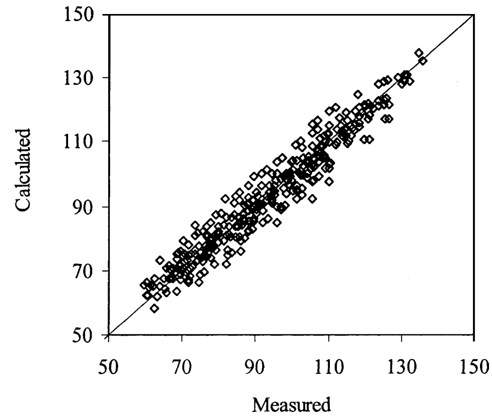

Fig. 15. Measured and calculated efficiency for the IEC/DEC system.

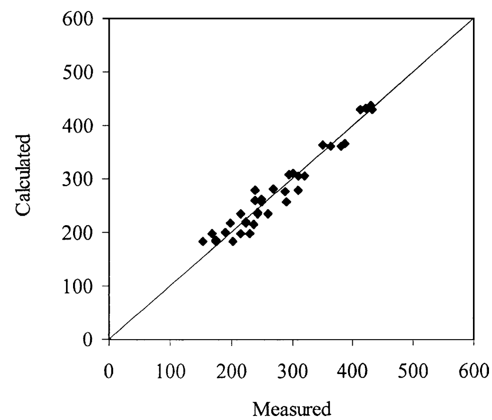

Fig. 16. Measured and calculated Nusselt number for IEC heat exchange unit.

10. Nusselt number correlation

The Nusselt number correlation for the IEC unit is given by the following relation:

$$Nu = 38.108A^{0.743}(Re_{\mathrm{w}})^{-0.112}(Re_{\mathrm{a}})^{0.247} \tag{12}$$

where Nu is the Nusselt number, $Nu = h_{\mathrm{a}}d/\lambda_{\mathrm{a}}$ (dimensionless), λ_{a} the thermal conductivity of the air stream (kW/m K), h_{a} the heat transfer coefficient of the air stream (kW/m^2 K).

The coefficient of determination for the above correlation is equal to 91.8%. Comparison for the predictions of the above equation against the measured values of the Nusselt number, which are given by Eqs. (2)–(8) is shown in Fig. 16. The above correlation is also valid for the same experimental ranges given by Eqs. (9)–(11).

11. Conclusions

The performance of an experimental unit for indoor air conditioning is evaluated in the hot and humid environment of Kuwait. The unit is composed of indirect and direct evaporative cooling units. Measurements are made as a function of various operating and design parameters, which includes number of heat exchangers in the IEC unit, flow direction of water inside the IEC heat exchange units, flow rate of the air stream outside the heat exchange units and across the DEC packing, thickness of the DEC packing, water flow rate across the DEC packing. Measurements include the air temperature across the evaporative cooling unit as well as the water flow rates in the IEC and DEC units. The system performance is reported in terms of the temperature efficiency, which gives a measure for the cooling degrees of the ambient air. Also, the Nusselt number of the air stream across the IEC heat exchanger is correlated as a function of design and operating conditions.

Results show that the efficiency of the IEC/DEC varies over a range of 90–120%. This implies that the dry bulb temperature of the outlet air is lower than the wet bulb temperature of the ambient (or intake) air. Similarly, the efficiency of the IEC unit is varied over a range of 20–40%, which is less than that of the DEC unit. The efficiency of the DEC unit varies over a range of 63–93%. Such values are quite higher than those reported in literature for the stand-alone DEC systems, which commonly vary over a range of 50–75%. This is because of the cooling effect of the IEC unit. Also, it should be noted that the efficiency of the IEC is also lower than that for those reported in literature, which commonly vary over a similar range for the stand-alone DEC system. This is because results reported in literature are for an IEC system combined with an external cooling tower. However, in the system studied here, water cooling is accomplished by the DEC unit only.

The Nusselt number for the air stream varies over a range of 150–450, which corresponds to a heat transfer coefficient of 0.1–0.4 kW/m^2 K. These results are consistent with literature data and should be of great value in designing of IEC units.

In summary, the results and analysis presented here indicate the attractiveness of the EC systems for indoor air conditioning. High efficiency are obtained irrespective of the high intake air temperature. It is highly recommended to continue research investigations in this area in search for low energy consumption air conditioning units, as this would contribute towards a cleaner environment and would conserve the limited resources of fossil fuel.

Acknowledgements

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Appendix A

$$\rho = 10^3(A_1F_1 + A_2F_2 + A_3F_3 + A_4F_4) \tag{A.1}$$

where $A_1 = 1.9011225$, $A_2 = -0.0560935$, $A_3 = -0.0078725$, $A_4 = 0.00038$, $A = (2T - 200)/160$, $F_1 = 0.5$, $F_2 = A$, $F_3 = 2A^2 - 1$, $F_4 = 4A^3 - 3A$.In the above equations ρ is the water density (kg/m^3) and T the water temperature (°C). This correlation is valid for a temperature range of $10 \le T \le 180\,°C$.

The water thermal conductivity is given by:

$$\log_{10}(\lambda) = 2.38 + 0.434\left(2.3 - \frac{343.5}{T + 273.15}\right) \times \left(1 - \frac{T + 273.15}{647.3}\right)^{1/3} \tag{A.2}$$

where λ is the thermal conductivity (W/m K) and T the temperature (°C). The above correlation valid for a temperature range of $20 \le T \le 180\,°C$.

The water specific heat at constant pressure is given by the following correlation:

$$C_p = (4206.8 - 1.1262T + 1.2026 \times 10^{-2}T^2 + 6.8777 \times 10^{-7}T^3) \times 10^{-3} \tag{A.3}$$

where C_p is in kJ/kg K and T in °C. The above correlation is valid for a temperature range of $20 \le T \le 180\,°C$.

The correlation for the dynamic viscosity of saturated liquid water is given by:

$$\mu = \exp\left(\frac{-3.79418 + 604.129}{139.18 + T}\right) \times 10^{-3} \tag{A.4}$$

where μ is in kg/ms and T in °C. The above correlation is valid for a temperature range of 10–115 °C.

The air density is calculated from ideal gas law, which is valid for low pressures

$$\rho = \frac{P \times M}{R \times T} \tag{A.5}$$

In the above equation ρ is the density in kg/m^3, P the pressure in Pa, M the molecular weight (for air, $M = 28.966$), and R the universal gas constant and is equal to 8315 Pa m^3/kmol K.

The dynamic viscosity of air is given by the following correlation, which is valid over a temperature range of $250 < T < 600$ K,

$$\mu = 1 \times 10^{-6}(-0.98601 + 9.080125 \times 10^{-2}T - 1.17635575 \times 10^{-4}T^2 + 1.2349703 \times 10^{-7}T^3 - 5.7971299 \times 10^{-11}T^4) \tag{A.6}$$

In the above equation T is in K and μ in kg/ms.

The thermal conductivity of air is given by the following correlation, which is valid over a temperature range of $250 < T < 1050$ K, Rohsenow et al. (1998).

$$\lambda = 1 \times 10^{-3}(-2.276501 \times 10^{-3} + 1.2598485 \times 10^{-4}T - 1.4815235 \times 10^{-7}T^2 + 1.73550646 \times 10^{-10}T^3 - 1.066657 \times 10^{-13}T^4 + 2.47663035 \times 10^{-17}T^5) \tag{A.7}$$

In the above equation T is in K and λ in kW/m K.

The specific heat at constant pressure of air is given by the following correlation, which is valid over a temperature range of $250 < T < 1050\,\mathrm{K}$, Rohsenow et al. (1998).

$$C_p = 1.03409 - 2.84887 \times 10^{-4} T + 7.816818 \times 10^{-7} T^2 - 4.970786 \times 10^{-10} T^3 + 1.077024 \times 10^{-13} T^4 \quad \text{(A.8)}$$

In the above equation T is in K and C_p in kJ/kg K.

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