Testing and performance analysis of a hollow fiber-based core for evaporative cooling and liquid desiccant dehumidification

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\textbf{ABSTRACT}

In this study, an innovative heat and mass transfer core is proposed to provide thermal comfort and humidity control using a hollow fiber contactor with multiple bundles of micro-porous hollow fibers. The hollow fiber-based core utilizes 12 bundles aligned vertically, each with 1,000 packed polypropylene hollow fibers. The proposed core was developed and tested under various operating and ambient conditions as a cooling core for a compact evaporative cooling unit and a dehumidification core for a liquid desiccant dehumidification unit. As a cooling core, the fiber-based evaporative cooler provides a maximum cooling capacity of 502 W with a wet bulb effectiveness of 85%. As a dehumidification core and employing potassium formate as a liquid desiccant, the dehumidifier is capable of reducing the air relative humidity by 17% with an overall dehumidification capacity of 733 W and humidity effectiveness of 47%. Being cheap and simple to design with their attractive heat and mass transfer characteristics and the corresponding large surface area-to-volume ratio, hollow fiber membrane contactors provide a promising alternative for cooling and dehumidification applications.

\textbf{KEYWORDS}

Cooling capacity; evaporative cooling; hollow fiber membrane; liquid desiccant dehumidification

1. Introduction

As an alternative and environmentally friendly cooling technology, evaporative cooling has a large potential to provide thermal comfort in occupied spaces due to the simple design, cheap materials and high coefficient of performance in addition to the efficient operation compared to conventional vapor compression-based cooling systems (Delfani et al. 2010; Hammoud, Ghali, and Ghaddar 2014; Maheshwari, Al-Ragom and Suri 2001; Uçkara et al. 2013). Evaporative cooling systems have a coefficient of performance in the range of 8–20 which is much higher than that of the conventional vapor compression coolers, and thus requiring less electrical power to operate providing high economic feasibility (Anisimov, Pandelidis, and Danielewicz 2014; Cui et al. 2014; Zhan et al. 2011). In addition, such evaporative cooling systems can avoid 44% of the carbon dioxide emissions produced by vapor compression systems with a very high energy efficiency ratio (EER) reaching 80 (Duan et al. 2012). As a comparison, typical values of EER for different air conditioning techniques are presented in Table 1 (Afonso 2006). However, conventional evaporative coolers such as those disclosed in US Patents No. 5,971,370 and 6,079,365 (Galabinski 1999; Medlin and Wilkins 2000), supply water vertically through multiple pads allowing water evaporation and process air cooling. Such evaporative coolers are relatively expensive to manufacture and install and have major drawbacks with a high possibility of drawing water through the pads into the building’s interior. This will have negative impacts on the building structure and the indoor air quality including damage to adjacent equipment, high rates of microbial growth and joint and respiratory infections.

Nevertheless, high air relative humidity in buildings affects the occupants breathing and skin evaporation rate, in addition to encouraging mould and germs growth and building construction and equipment decay. Thus it is favorable to remove water vapor from air through dehumidification. Conventional cooling systems, mainly vapor compression units, are extensive energy consumers using cold coils to reduce air temperature below the dew point temperature allowing water vapor condensation and latent heat release. Recently, a large body of research has been presented regarding the use of solid and liquid desiccant materials and their potential in air dehumidification and cooling applications (Kumar, Chaudhary and Yadav 2014; Mei and Dai 2008; Qiu and Riffat 2010; Uçkara et al. 2014). The major factor governing moisture absorption by a desiccant material is process air-desiccant material surface vapor pressure gradient. As the vapor pressure of the desiccant surface is less than that of the air, moisture absorption continue allowing air dehumidification until the equilibrium in the vapor pressure between air and desiccant is attained (Lowenstein 2008). Liquid desiccant air dehumidification units have significant potential providing different advantages compared to solid desiccant units including lower consumption of energy, higher coefficient of performance, higher flexibility with the ability to transport the liquid desiccant between various system units including the dehumidifier and regenerator (She, Yin and Zhang 2014; Daou, Wang and Xia 2006; Oliveira et al. 2000; Qi, Lu and Huang 2014; Yutong and Hongxing 2010). On the other hand, conventional liquid desiccant units still suffer from major drawbacks especially the serious problem of liquid desiccant entrainment...
by the process air which could affect the indoor air quality and has negative impacts on the thermal comfort of the occupants (Jradi and Riffat 2014). Therefore, it is reported that using semi-permeable micro-porous contactors in liquid desiccant units could help in eliminating the liquid desiccant entrainment problem (Isetti, Nannei, and Magrini 1997).

In order to address these issues and to improve the indoor air quality, we are proposing an innovative dehumidification and cooling system using hollow fibers and employing liquid desiccant as the dehumidification working fluid. Compared to the conventional cooling and dehumidification techniques, the proposed system has the following innovative features:

- Providing thermal comfort through the use of selective hollow fibers, with high packing density and large heat and mass transfer surface area, allowing no direct contact between air and liquid desiccant and thus no droplets carryover by the air.
- Employing environmentally friendly working fluids, water and potassium formate liquid desiccant, to reduce the environmental negative impacts of space air conditioning.
- Presenting an innovative evaporative cooling core using hollow fiber membrane as a wetting medium with no water-air interaction and thus better indoor air quality.
- Using inexpensive materials, hollow fibers and cheap containers in addition to the simple and compact integrated system configuration, resulting into low system capital and running costs.

### 2. Hollow fiber contactor technology

A hollow fiber contactor is composed of bundles of micro-porous hollow fiber membranes having numerous fine pores across the fiber wall with a very small diameter (Gabelman and Hwang 1999). Hollow fiber-based membranes, shown in Figure 1 (SpinTek 2014), have the ability to act as a passive barrier between two heterogeneous fluid phases without dispersion, where one of those fluids occupies the pores void volume on the surface of the membrane. Hollow fiber contactors are typically made of materials with high hydrophobic effect including: polyethylene (PE), polypropylene (PP), polytetrafluoroethylene (PTFE) and polyvinylidene fluoride (PVDF) (Rajabzadeh et al. 2009).

Different companies now are manufacturing hollow fiber membrane contactors as an economic and cost effective alternative solution in different fields as shown in Figure 2 (Mitsubishi 2014). Employing cheap materials and having attractive mass transfer characteristics in addition to their large surface area/volume ratio, hollow fibers have been successfully employed as membrane contactors in various applications including: chemical engineering separation, liquid-liquid extraction, gas absorption, microfiltration processes, brackish water desalination, potable water purification, wastewater treatment, drying processes and biofuels separation (Boributh et al. 2012; Cath et al. 2005; Mansourizadeh and Ismail 2009).

The selection of the hollow fiber membrane contactor for a specific application is based on different factors including fiber surface pore size, surface pores distribution, separation layer thickness in addition to the physical, chemical and mechanical properties of the fiber material (Pabby and Sastre 2013). (Pabby and Sastre 2013) presented a comprehensive review on the recent research and developments in the field of hollow fiber contactor technology and membrane-based extraction processes. They reviewed different hollow fiber membrane aspects including performance, mass transfer modeling, stability issues, applications and the development in the hollow fiber membrane-based separation techniques. In addition to the commercial advancement in the field of hollow-fiber based membrane contactors, different researchers have investigated the performance of such type of contactors employing different configurations and fluid flow patterns, with specific concentration on the heat and mass transfer phenomena within the hollow fiber membrane (Bui, Vu, and Nguyen 2010; Huang et al. 2013; Huang and Yang 2013; Zhang et al. 2012). Very few research studies have investigated the use of hollow fibers in cooling and dehumidification applications. A theoretical and experimental study of a liquid desiccant air dehumidification system utilizing a hollow-fiber based membrane core was presented by (Zhang and Zhang 2014). A compression heat pump was used to simultaneously heat and cool the liquid desiccant solution to enhance the system efficiency by passing the solution through the heat pump evaporator and condenser. A dehumidification efficiency of 0.3–0.5 was attained and a satisfactory system performance was reported even in hot and humid conditions. (Dijkink et al. 2004) carried out an experimental investigation of polyetherimide (PEI) hollow fiber membrane contactor coated with a thin non-porous silicone layer on the inside and using dilute aqueous glycerol.

### Table 1. Typical EER values for standard air conditioning techniques.

| Cooling Technique | Vapor Compression | Absorption | Adsorption | Desiccants | Ejector | Thermoelectric |
|-------------------|-------------------|------------|------------|------------|---------|----------------|
| EER (Btu/Wh)      | 7–17              | 2–3.4      | 0.7–2.7    | 1.7–5.1    | 0.9–2.7 | 1.7–3.4        |

Data from Afonso 2006.
solution as a liquid desiccant. (Johnson, Yavuzturk, and Pruis 2003) investigated experimentally hollow fiber based evaporative cooling systems and recommended the use of hollow fiber membranes with larger pore sizes, thinner membrane walls and low tortuosity to increase mass transfer rates in evaporative cooling systems. (Das and Jain 2013) studied the performance of air–liquid indirect membrane contactors for liquid desiccant cooling systems using hollow fibers. The maximum vapor flux attained was about 1295 g/m$^2$.h with dehumidification effectiveness between 23% and 45% using LiCl as a desiccant solution. Based on these analytical and numerical investigations and the results reported, it is shown that hollow fiber-based membrane contactors have favorable heat and mass transfer characteristics and possess a large potential to serve dehumidification and cooling applications.

### 3. Proposed hollow-fiber-based core

Having favorable hydrophobic specifications, large surface area-to-volume ratio, attractive heat and mass transfer characteristics, simple and maintenance-free operation, in addition to employing cheap materials that are corrosion resistant and high-temperature and pollution tolerant, hollow fiber-based membrane presents a cost effective and environmentally friendly alternative solution to serve as a core in evaporative cooling systems and liquid desiccant dehumidification systems. In this work an innovative hollow fiber-based core is proposed to provide thermal comfort and humidity control and improve indoor air quality in occupied spaces. The presented core consists mainly of a hollow fiber-based membrane contactor that could be employed as a dehumidification core in liquid desiccant dehumidification units and as a cooling core in evaporative cooling units. As shown in Figure 3, the hollow fiber-based core utilizes 12 bundles aligned vertically, each with 1000 packed hollow fibers. The specifications of the hollow fibers utilized in the study are presented in Table 2.

The fibers are assembled and packed using short pieces of a plastic tube and potted at both ends of the bundle using epoxy resin and silicone sealant as shown in Figure 4(a). In addition, the fiber bundles are attached to the plastic water/liquid desiccant distribution network at the top of the unit as shown in Figure 4(b). Water/liquid desiccant is circulated in the system and a small pump (6 W electric power consumption), shown in Figure 5(a) is utilized to feed the fluid from a plastic collection tank, placed at the bottom of the cooling unit, through a pipe connected to the 12 fiber bundles. Water/liquid desiccant is circulated at a flow rate ranging between 0.2 and 0.7 l/min where the fluid flows through hollow fibers and drops back into the collection tank. An air duct is employed where fibers are densely spread inside and extend through both ends of the duct. Air is introduced employing a small AC blower, shown in Figure 5(b), through the duct in a horizontal direction to the vertical fluid flow pattern in the fiber membrane.

The developed hollow fiber-based core configuration allows cross-flow heat and mass exchange between the air flow in the duct and fluid flow in the hollow fibers. The semi-permeable membrane hollow fibers employed are water vapor permeable and liquid tight, allowing moisture transport with no direct contact between the liquid flowing inside the fibers and the air passing across the external fibers surface. The mass transport of water vapor depends mainly on the water vapor pressure difference between both sides of the fiber membrane wall. When used as a dehumidifier, the humid air is introduced through the duct and gets in contact with the external surface of the fibers, where the liquid desiccant flows inside the fibers absorbing water vapor through the pores distributed along the surface and dehumidifying the process air. Similarly when the core is utilized as an evaporative cooler, hot air is introduced to flow in the duct where water flows inside the fibers, and a portion of the water is evaporated leading to a drop in the air temperature. The large surface area-to-volume ratio provided by the compact and simple design of the fiber membrane enhances heat and mass transfer between the air flowing on the membrane external surface and the fluid flowing inside the hollow fibers. Neither water nor liquid desiccant carryover occurred during
cooling and dehumidification processes providing thermal comfort and good indoor air quality.

In the following sections, the preliminary testing of the fiber membrane-based core as an evaporative cooler and as a liquid desiccant dehumidification unit is presented and the results are reported.

4. Evaporative Cooling Unit Performance

The hollow fiber-based core developed and was tested and investigated as a cooling core for a compact evaporative cooling unit. A climatic chamber was utilized to control the inlet temperature and relative humidity to the cooling core. In addition, two HMP50 temperature and humidity probes were employed to record the intake air and supply air humidity and temperature. The temperature measurement range of these sensors is –40 to +60°C with ±0.3°C accuracy where the relative humidity measurement range is 0-98%. In addition, a K-type thermocouple of ±0.25% accuracy and a maximum temperature measurement of 250°C, were employed to measure the water inlet temperature to the fiber membrane. The recording sensors were connected to a Datataker DT 80 data logger for data monitoring and recording. Throughout the cooling unit preliminary testing sessions, intake air speed of 2.4 m/s was employed.

The overall cooling capacity of the evaporative cooling unit can be given by:

\[ Q_{\text{cooling}} = \dot{m}_{\text{su}}(h_{\text{in}} - h_{\text{su}}) \]  

\[ m_{\text{su}} \] is the supply air mass flow rate in kg/s, \( h_{\text{in}} \) and \( h_{\text{su}} \) are the respective air specific enthalpy at the inlet and outlet of the evaporative cooler.

The cooler coefficient of performance is given by the ratio of the cooling capacity provided to the electric power consumption. The total electric power consumption, \( W_{\text{ele,cooler}} \) in W, includes the power required to operate the water circulation pump for and the power required to run the fan employed for air supply. The system overall COP is represented by:

\[ \text{COP}_{\text{cooler}} = \frac{Q_{\text{cooling}}}{W_{\text{ele,cooler}}} \]  

The cooling effectiveness of the evaporative cooler can be expressed in terms of the wet bulb effectiveness and the dew point effectiveness given by equations (3) and (4) respectively.

\[ \varepsilon_{\text{wb}} = \frac{T_{\text{in,db}} - T_{\text{su,db}}}{T_{\text{in,db}} - T_{\text{in,wb}}} \]  

\[ \varepsilon_{\text{dp}} = \frac{T_{\text{in,db}} - T_{\text{su,db}}}{T_{\text{in,db}} - T_{\text{in,dp}}} \]  

\( T_{\text{in,db}} \) is the inlet air dry bulb temperature, \( T_{\text{su,db}} \) is the supply air dry bulb temperature, \( T_{\text{in,wb}} \) is the inlet air wet bulb temperature and \( T_{\text{in,dp}} \) is the inlet air dew point temperature.

4.1 climatic chamber set temperature of 30°C

Figure 6(a) presents the variation in the inlet and outlet air temperatures across the evaporative cooler employing different water volumetric flow rates. Based on the data recorded, the average temperature drop across the cooler is about 6.1°C, 6.9°C and 6.7°C for water volumetric flow rate of 0.2, 0.4 and 0.5 l/min, respectively. In addition, Figure 6(b) presents the
variation in the inlet and outlet air relative humidity at different flow rates. It is shown that the average increase in the outlet air relative humidity is about 22.6%, 24.5%, and 24.6% at water volumetric flow rate of 0.2, 0.4 and 0.5 l/min.

Figure 7(a) shows the variation in the cooling capacity delivered by the evaporative cooler and the cooler coefficient of performance (COP) at different water volumetric flow rates. The maximum cooling capacity produced by the system increases from 164 W at 0.2 l/min water volumetric flow rate to about 191 W at 0.5 l/min volumetric flow rate. Correspondingly, the COP increases from an average of 5.3 to about 6.3 as the water volumetric flow rate increases from 0.2 to 0.5 l/min. In addition, the estimated wet bulb and dew point effectiveness of the system is presented in Figure 7(b). It is shown that the wet bulb effectiveness increases from 64% to 69% with the increase of volumetric flow rate of water from 0.2 to 0.5 l/min, accompanied with an increase in the dew point effectiveness from 41% to 46%.

### 4.2 Climatic chamber set temperature of 35°C

Increasing the climatic chamber set temperature from 30°C to 35°C, Figure 8(a) shows the variation in the inlet and outlet air temperatures across the evaporative cooler. As shown in the figure, the maximum temperature drop across the cooler is about 7.7°C, 8.2°C, and 8.9°C for water volumetric flow rate of 0.4, 0.5, and 0.7 l/min, respectively. In addition, Figure 8(b) shows the variation in the inlet and outlet air relative humidity at different flow rates. It is reported that the average increase in the outlet air relative humidity is about 22.3%, 23.2%, and 26.1% at water volumetric flow rate of 0.4, 0.5, and 0.7 l/min.

Figure 9(a) presents the cooling capacity produced by the evaporative cooler and the cooler COP at different water volumetric flow rates. It is shown that the maximum cooling capacity produced by the system increases from 278 W at 0.4 l/min water volumetric flow rate to about 334 W at 0.7 l/min volumetric flow rate. The evaporative cooler COP follows an increasing trend as the water volumetric flow rate increases. As shown in the figure, the maximum COP attained increases from 10.7 to 12.9 as the volumetric flow rate increases from 0.4 to 0.7 l/min. Moreover, the evaporative cooler wet bulb and dew point effectiveness data is presented in Figure 9(b). The system maximum wet bulb effectiveness increases from 71% to 79% as the water volumetric flow rate increases from 0.4 to 0.7 l/min. Similarly, the system dew point effectiveness increases from 50% to 57%.

### 4.3 Climatic chamber set temperature of 45°C

With a climatic chamber set temperature at 45°C, Figure 10(a) presents the variation in the evaporative cooler inlet and outlet air temperatures at different water volumetric flow rates. It is shown that the maximum temperature drop across the cooler is

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**Figure 5(a-b).** (a) Fluid pump, (b) AC air blower.

**Figure 6(a-b).** Variation of air (a) temperature and (b) relative humidity across the cooler at 30°C set temperature.

**Figure 8(a).** Variation in the cooling capacity and COP at different water volumetric flow rates.

**Figure 8(b).** Variation in the inlet and outlet air relative humidity at different flow rates.

**Figure 9(a).** Cooling capacity produced by the evaporative cooler and COP at different water volumetric flow rates.

**Figure 9(b).** Wet bulb and dew point effectiveness data.
about 8.2°C, 8.7°C, and 9.2°C for water volumetric flow rate of 0.2, 0.4, and 0.5 l/min respectively. In addition, Figure 10(b) shows the variation in the inlet and outlet air relative humidity at different flow rates. The average increase in the relative humidity of the supply air is about 22.2%, 23.3%, and 24.5% at water volumetric flow rate of 0.2, 0.4, and 0.5 l/min.

Figure 11(a) shows the evaporative cooler cooling capacity in addition to the COP employing different water volumetric flow rates and at a climatic chamber set temperature of 45°C. As the volumetric flow rate of water increases from 0.2 to 0.5 l/min, the maximum system cooling capacity increases form 427 W to 502 W. The COP follows a similar trend and increases from 16.1 to a maximum of 19.3 at water volumetric flow rate of 0.2 and 0.5 l/min respectively. In addition, the system calculated wet bulb effectiveness increases from 78% to 85% with the increase of water volumetric flow rate from 0.2 to 0.5 l/min, accompanied by an increase in the dew point effectiveness from 60% to 65%.

4.4 Effect of inlet air temperature

Employing the climatic chamber to control the inlet air temperature to the evaporative cooler allows investigating the impact of inlet air temperature on the overall performance of the cooling system. As presented in the previous sections, three different chamber set temperatures were employed, 30°C, 35°C, and 45°C. Figure 12(a) shows the variation in the
temperature of the inlet air and outlet air of the cooling system at different set temperatures, employing water volumetric flow rate of 0.5 l/min. It is shown that the maximum temperature drop across the evaporator is about 7.2°C, 8.1°C, and 9.1°C at an operating air inlet temperature of 30°C, 35°C, and 45°C, respectively. In addition, Figure 12(b) shows the variation in the cooling capacity delivered by the evaporative cooler and the COP of the cooling system employing different inlet air temperatures. It is shown that the cooling capacity produced by the system is directly proportional to the inlet air temperature. The maximum evaporative cooler cooling capacity increases from about 191 W at a set temperature of 30°C up to 502 W at a set temperature of 45°C. In addition, the system COP follows the same trend as the cooling capacity and increases with the increase in the inlet air temperature. As shown in Figure 12(b), the evaporative cooler maximum COP is about 7.4, 11.2 and 19.3 at a set inlet air temperature of 30°C, 35°C, and 45°C. Regarding the water temperature, Figure 12(c) shows the variation in the water temperature during the evaporative cooler operation at different inlet air set temperatures. As the water, initially at a temperature between 18 and 20°C, is recirculated from the tank to the fibers and then back to the tank, its temperature increases with time when it gets in contact with the relatively hot air. As shown in the figure, there is a slight increase from 18.7°C to 19.1°C in the water temperature when the inlet air has a temperature in the range of 30°C. However, the water temperature increase is much more significant, increasing from about 19.2°C to 23.6°C, employing an inlet air of temperature 45°C. It is shown that the cooling capacity delivered by the evaporative cooling system and coefficient of performance are inversely proportional to the inlet water temperature. Therefore, to attain the system maximum cooling capacity, the inlet water temperature needs to be maintained at 18–19°C throughout the operation period. This could be done through replenishing the water introduced to the fiber-based evaporative cooling system or using a continuous water supply source.

5. Dehumidification unit performance

In addition to investigating the developed hollow fiber-based core as a cooling core for indirect evaporative cooling applications, the hollow fiber membrane was investigated and tested as a dehumidification core for a liquid desiccant dehumidification unit. The same experimental setup described in section 3 is employed but potassium formate HCOOK liquid desiccant of 74% mass concentration was circulated and introduced to the fiber membrane instead of water. Potassium formate was used due to its various environmental, physical and thermodynamic advantages compared to conventional liquid desiccants (LiCl, LiBr) with lower density, viscosity and being less corrosive (Longo and Gasprella 2005; Riffat, James and Wong 1998). Humid air was drawn in the duct from the climatic chamber to be dehumidified by the action of the fiber bundles. Throughout the dehumidification unit preliminary testing sessions, intake air speed of 2.6 m/s was employed.

The dehumidification capacity of the liquid desiccant dehumidification system can be represented as:

Figure 10(a-b). Variation of air (a) temperature and (b) relative humidity across the cooler at 45°C set temperature.

Figure 11(a-b). Evaporative cooler (a) cooling capacity and COP, (b) wet bulb and dew point effectiveness at 45°C set temperature.
Dehumidification unit.

The humidity effectiveness ($\varepsilon_w$) of the liquid desiccant dehumidification system, is the actual air humidity ratio change over the maximum possible change:

$$\varepsilon_w = \frac{w_{a,in} - w_{a,out}}{w_{a,in} - w_{eq}}$$ (6)

$h_{a,in}$ and $h_{a,out}$ are the respective air specific enthalpy at the inlet and outlet of the dehumidification unit.

$w_{eq}$ is the air humidity ratio in equilibrium with the liquid desiccant in kg$H_2O$/kg$air$. This humidity ratio is the ideal minimum level to which air can be dehumidified and is given in terms of the desiccant partial vapor pressure $P_v$ by equation (7):

$$w_{eq} = \frac{0.62197 \frac{P_v}{1.013 \times 10^5} - P_v}{1}$$ (7)

The enthalpy effectiveness ($\varepsilon_h$) is defined as the actual change in air enthalpy over the maximum ideal enthalpy change and can be given by:

$$\varepsilon_h = \frac{h_{a,in} - h_{a,out}}{h_{a,in} - h_{eq}}$$ (8)

$h_{eq}$ in J/kg, is the air enthalpy in equilibrium with the liquid desiccant and can be obtained in terms of the air equilibrium humidity ratio $w_{eq}$ at an air temperature equivalent to that of the liquid desiccant employed.

### 5.1 Set conditions: $T_{ain}$ 30°C and $RH_{ain}$ 80%

Employing climatic chamber settings of 30°C temperature and 80% relative humidity, Figure 13(a) shows the variation in the air temperature at the inlet and outlet sections of the dehumidifier in addition to the change in the potassium formate liquid desiccant temperature introduced to the hollow fibers-based system. With the inlet air temperature fixed at about 30°C, the minimum air temperature attained at the outlet of the dehumidifier is about 27.6°C where the maximum drop in the air temperature across the dehumidifier is about 2.4°C. In addition, the liquid desiccant temperature increases with time from 19.3°C to about 26.4°C. This temperature increase is mainly due to the temperature difference between the relatively colder liquid desiccant flowing inside the hollow fibers and the relatively hotter air flowing in direct contact with the external surfaces of the fibers. In addition, the temperature drop across the dehumidifier is inversely proportional to the increase in the liquid desiccant temperature as shown in Figure 13(a). Figure 13(b) shows the variation in the relative humidity of air at the inlet and outlet of the dehumidifier. While fixing the inlet air relative humidity at around 80%, the average relative humidity of the outlet air is around 66.1%. The maximum drop in the air relative humidity across the dehumidifier is about 16.6%. In addition, Figure 13(c) presents the variation of the air enthalpy at the inlet and outlet of the dehumidifier along with the cooling capacity delivered by the system. It is shown that the average inlet and outlet air enthalpy is about 85.5 kJ/kg and 69.4 kJ/kg, respectively, where the maximum cooling capacity provided by the dehumidifier is about 733 W with an average capacity of 673 W throughout the experimental session. Figure 13(d) shows the variation in the system humidity and enthalpy effectiveness along with the variation in the liquid desiccant temperature. It is obvious that the dehumidification system effectiveness is inversely proportional to the liquid desiccant temperature. The maximum humidity and enthalpy effectiveness attained is about 47% and 44%, respectively.
5.2 Effect of inlet air relative humidity

Employing the climatic chamber, multiple testing sessions for the liquid-based dehumidification system was carried out to investigate the effect of the inlet air relative humidity on the overall system performance. Figure 14(a) shows the variation in the air relative humidity at the inlet and outlet of the dehumidifier at three inlet air relative humidity settings, 60%, 70%, and 80%. It is shown that the drop in the air relative humidity is directly proportional to the relative humidity of the inlet air, with a maximum drop of 4.9%, 9.3%, and 16.3% at inlet air relative humidity of 60%, 70% and 80% respectively. In addition, the average air relative humidity at the outlet of the dehumidifier is about 56%, 62%, and 66% for a respective inlet air relative humidity of 60%, 70%, and 80%. In addition, Figure 14(b) presents the variation in the dehumidification system cooling capacity in addition to the variation of the liquid desiccant temperature at the three investigated relative humidity settings. The maximum cooling capacity delivered by the dehumidification system is about 426 W, 519 W and 733 W for relative humidity setting of 60%, 70%, and 80%. It is shown that the desiccant temperature increases as the temperature of the liquid desiccant increases, where the system cooling capacity is directly proportional to the inlet air relative humidity. Similarly, the dehumidification system humidity effectiveness and enthalpy effectiveness is directly proportional to the inlet air relative humidity as shown in Figure 14(c). The maximum reported wet bulb and dew point effectiveness are (40%, 37%), (43%, 40%), and (47%, 44%) at inlet air relative humidity setting of 60%, 70%, and 80%, respectively.

5.3 Effect of inlet air temperature

Testing sessions were carried out to investigate the effect of inlet air temperature on the dehumidifier various performance parameters. Figure 15(a) shows the variation in the air temperature at the dehumidifier inlet and outlet at two inlet air temperature settings: 30°C and 35°C, where the set relative humidity is fixed at 80%. As shown in the figure, the average air temperature drop across the dehumidifier is about 1.6°C and 1.1°C at 30°C and 35°C inlet air set temperature. It is shown that the drop in the air temperature decreases with time due to the fact that the liquid desiccant temperature introduced increases with time. This effect is more significant at higher air temperatures where the desiccant temperature exhibits fast increase compared to relatively lower temperatures.

In addition, Figure 15(b) shows the variation in the air relative humidity across the dehumidifier under the two employed inlet air temperature settings. It is shown that the relative humidity drop across the dehumidifier is inversely proportional to the inlet air temperature. The average relative humidity drop across the dehumidifier is about 11.5% and 14.6% at 30°C and 35°C, respectively. Figure 15(c) shows the variation in the liquid desiccant temperature and the cooling capacity delivered by the fiber-based dehumidification system at two different inlet air temperature settings. It is shown that the liquid desiccant temperature increases from 19.3°C to 26.4°C at inlet air temperature setting of 30°C, where the desiccant temperature increase rate is more significant at an inlet air temperature setting of 35°C, increasing form about 21.4°C to about 30.9°C. In addition, it is shown that the dehumidification system cooling capacity is inversely proportional to the inlet air temperature where the average cooling capacity attained is around 673 W and 612 W at inlet air.
6. Conclusion

With the increase in the conventional fuel prices and the global warming problem, a growing body of research has been presented to investigate efficient and environmentally friendly alternative technologies and solutions to provide thermal comfort and good indoor air quality. In this work, an innovative hollow fiber-based energy core is proposed and investigated to provide thermal comfort and humidity control in air-conditioned spaces. The presented core comprises a hollow fiber contactor having multiple bundles of micro-porous hollow fibers packed and assembled together, providing large surface area-to-volume ratio with favorable heat and mass transport characteristics. The employed fiber membrane is cheap, simple and compact in design and corrosion resistant with good hydrophobic properties. The use of the innovative semi-permeable fiber-based membrane eliminates any water or liquid desiccant droplets carryover by the air allowing better indoor air quality. The proposed fiber-based core was tested in the Built Environment laboratories at the University of Nottingham to serve as a cooling core in evaporative cooling systems and a dehumidification core in liquid desiccant systems. As a cooling core, the maximum cooling capacity provided by the fiber-based evaporative cooler was about 502 W with a COP of around 19, wet bulb effectiveness of 85% and dew point effectiveness of 65%. Using potassium formate as a liquid desiccant, the dehumidification core was able to decrease the humid air relative humidity by about 17% with a dehumidification capacity of 733 W and humidity effectiveness of 47%. The satisfactory preliminary testing results reported demonstrate the potential of using hollow fiber-based energy cores for cooling and dehumidification applications allowing technical, economic and environmental benefits compared to conventional cooling and dehumidification systems.

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Nomenclature

| Symbol | Description             |
|--------|-------------------------|
| AC     | alternating current     |
| COP    | coefficient of performance |
| EER    | energy efficiency ratio (Btu/Wh) |
| h      | enthalpy (J/kg)         |
| HCOOK  | Potassium Formate       |
| LiBr   | Lithium Bromide         |
| LiCl   | Lithium Chloride        |
| m      | mass flow rate (kg/s)   |
| P_v    | vapor pressure (mbar)   |
| PE     | polyethylene            |
| PEI    | polyetherimide          |
| PP     | polypropylene           |
| PTFE   | polytetrafluoroethylene |
| PVC    | polyvinyl chloride      |
| PVDF   | polyvinylidene fluoride |
| RH     | relative humidity (%)   |
| LiCl   | Lithium Chloride        |
| Q      | cooling capacity (W)    |
Greek

ε effectiveness

Subscripts

a air
db dry bulb
Deh dehumidification
dp dew point
ele electric
eq equilibrium
in input
out output
su supply
wb wet bulb

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