

1 **A Novel Evaporative Cooling System with a Polymer Hollow Fibre Spindle**

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14 **Abstract:** A novel evaporative cooling system, in which the hollow fibre module constitutes as the
15 humidifier and evaporative cooler, is proposed. With the aim to avoid the flow channelling or shielding
16 of adjacent fibres the fibres inside each bundle were made into a spindle shape to allow maximum
17 contact between the air stream and the fibres. This novel hollow fibre integrated evaporative cooling
18 system will provide a comfortable indoor environment for hot and dry area. Moreover, the water vapour
19 can permeate through the hollow fibre effectively, and the liquid water droplets will be prevented from
20 mixing with the processed air. Under various inlet air dry bulb temperatures (27°C, 30°C, 33°C, 36°C
21 and 39°C), and various inlet air relative humidity (23%, 32% and 40%), the cooling performances of
22 the proposed novel evaporative cooling system were experimentally investigated. The variations of
23 outlet air dry bulb temperature, wet bulb effectiveness, dew point effectiveness and cooling capacity
24 with respect to different incoming air dry bulb temperature were studied. The effects of various
25 incoming air Reynolds number on the heat and mass transfer coefficients, heat flux and mass flux across

26 the polymer hollow fibre module were analysed. Experimentally derived non-dimensional heat and
27 mass transfer correlations were compared with other correlations from literature. Due to the spindle
28 shape of proposed hollow fibre module, the shielding with hollow fibre bundles could be avoided
29 greatly, therefore the mass transfer performance of the proposed system demonstrated significant
30 improvement compared with other devices reported in literature.

31

32 **Key words:** Polymer hollow fibre, evaporative cooling, heat transfer, mass transfer, experiment

33 **Nomenclature**

34 A Heat transfer area (m^2)

35 C_p Specific heat ($kJ/kg\ K$)

36 d Fiber diameter (m)

37 h Heat transfer coefficient (W/m^2K)

38 h_v Enthalpy of saturated water vapour (kJ/kg);

39 k Mass transfer coefficient (m/s)

40 L Characteristic length of hollow fibre bundle (m)

41 m Mass flow rate (kg/s)

42 n Number of fibres inside the heat exchanger

43 N Mass flux ($mg/m^2\ s$)

44 Nu Nusselt number

45 Pr Prandtl number

46 q Heat flux (W/ m^2)

47	Re	Reynolds number
48	Sh	Sherwood number
49	Sc	Schmidt number
50	T	Temperature ($^{\circ}\text{C}$)
51	U	Overall heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)
52	V	Volumetric flow rate of the incoming air, m^3/h ;
53	Q	Sensible cooling capacity (W)
54	Greek Letters/Subscripts	
55	a	Air
56	dew	Dew point
57	dry	Dry bulb
58	e	Evaporated water
59	φ	Packing fraction of the module
60	ε	Effectiveness
61	H	Heat transfer
62	λ	Thermal conductivity (W/mK)
63	M	Mass transfer
64	ρ	Density of the fluid (kg/m^3)
65	u	Incoming air velocity, m/s
66	ν	Dynamic viscosity of the fluid (kg/ms)

67 ω Humidity ratio of the air (kg/kg)

68 wb wet bulb

69

70 **1. Introduction**

71 Global energy demand is soaring during past few decades due to the rapid worldwide economy
72 development and urban sprawl. According to the research report produced by International Institute of
73 Refrigeration, air conditioning system accounts for 45% of the total energy consumption for domestic
74 and commercial buildings[1]. Overall, air-conditioning system takes up approximately 15% of the total
75 energy consumption around the world[1]. In the Middle East, where the climatic condition is dry and
76 humid, air conditioning system consumes as high as 70% of energy required for buildings and around
77 30% of the total energy[2]. With the impact of global warming, the demands of effective air-
78 conditioning system which consumes less energy and provide higher cooling performance is massive.

79 The current widely-used vapour compression system plays dominant role in the market. However,
80 vapour compression system has the disadvantages of intensive energy consumption and low
81 performance in hot and humid climate. Moreover, the possible leakage of high GWP refrigerants will
82 lead to the depletion of Ozone Layer, which further contributes to the global warming and other
83 associated environmental and social changes. Hence, the development of more energy efficient and
84 environmental benign cooling systems remains to be the research topics for scientific researches.

85 In the past few decades, evaporative cooling system arouses great attentions among the researchers due
86 to the fact that it is more environmentally friendly (use of the water as working fluids), simple in
87 structure configuration, and less consumption in primary energy. Direct evaporative cooling system
88 works under the following principle: the incoming hot and humid air gets direct contact with the
89 circulating water, causing the evaporation of the water and the air temperature will be reduced
90 accordingly. Consequently, the evaporated water, in the form of vapour will be absorbed by the air,
91 which leads to the humidity increase of the outlet air.

92 Recently, the research interests of this topic are focused on pad incorporated evaporative cooling
93 system[3-6], desiccant based evaporative cooling system[7, 8], and dew point based evaporative cooling
94 system[9-12]. Due to the large contact surface area, porous pad incorporated evaporative cooling
95 systems have attracted more attentions. Wu et al.[13] presented a simplified mathematical model to
96 describe the heat and moisture transfer between water and air in a direct evaporative cooler, with pad
97 thickness of 125mm and 260mm, the cooling efficiency reached 58% and 90% respectively. Franco et
98 al. [14] studied the influence of water and air flows on the performance of cellulose media. The results
99 showed that with a thickness of 85mm, a plastic grid pad could offer a cooling efficiency of 65% at
100 wind speed of 1.5m/s. However, since water is directly in contact with the incoming air in the closed
101 system, there is the potential for microbial growth due to the supply of stagnant water. This may provide
102 an opportunity for the spread of liquid phase-born bacterial diseases for occupants[15].

103 In order to solve this problem, a hollow fibre integrated evaporative cooling system has been proposed.
104 Compared with porous pad media, hollow fibre materials provide several advantages as follows: 1)
105 allow selective permeation of moisture: with pore sizes less than $0.1\mu\text{m}$, hollow fibre material will allow
106 the water vapour transfer but eliminate the bacteria and fungi penetration[16]; 2) provide large surface
107 area per unit volume[17], which is favourable for enhanced heat and mass transfer. Detailed descriptions
108 about hollow fiber materials and their applications are summarized in the literature[18]. According to
109 Chen et al.[19], the overall heat transfer coefficients could reach $1675\text{W}/\text{m}^2\text{K}$ with a fibre diameter of
110 $550\mu\text{m}$. Kachhwaha and Preahhakar[20] analysed heat and mass transfer performance for a direct
111 evaporative cooler using a thin plastic plate. The experimental testing results indicated that the outlet
112 air temperatures were between 21°C and 23°C , at the inlet dry bulb temperature of $24.8\text{-}28.4^\circ\text{C}$, the air
113 humidity ratio of $2.3\text{-}5.8\text{g}/\text{kg}$ and air mass flow rate of $0.13, 0.2, 0.3$ and $0.4\text{g}/\text{s}$. Zhang[21] proposed
114 the theoretical investigations on a rectangular cross-flow hollow fibre membrane module for air
115 humidification. With 2600 fibres (fibre outside diameter 1.5mm) inside the module, the outlet air
116 temperature could reach 21.5°C when the inlet dry bulb temperature was 30°C . Johnson et al.[15]
117 studied the heat and mass transfer of a hollow fibre membrane evaporative cooling system. With a
118 different range of fibre bundles (9, 19, 29 fibre bundles), the heat transfer area was in the range of

119 0.35m²-1.13m², and around 0.4°C temperature drop could be observed from the experiments. The above
120 publications are mainly concentrated on the theoretical analysis on the polymer hollow fibre integrated
121 evaporative cooling system. The available experimental results were limited to the variation of outlet
122 air temperatures with respect to different air flow rates. In addition, as stated by Johnson et al.[15], due
123 to the shielding from adjacent fibres, the heat and mass transfer performance will decrease when using
124 a large number of fibres inside one module.

125 A summary of the recent experimental and modelling works on evaporative cooling system is presented
126 in Table 1. Literature review indicates that the previous published papers were mainly concentrated on
127 the theoretically modelling of evaporative cooling system. For the limited experimental investigations
128 reported in the literature, the evaporative coolers were mainly made from porous paper materials. This
129 paper presents a novel evaporative cooling system with a hollow fibre evaporative cooler. Instead of
130 previously reported cross flow configurations[13, 15, 21], five fibre bundles (each contains 100 fibres)
131 with the distance of 5cm were placed normal to the air stream, with detailed configuration shown in
132 Figure 2. In order to avoid the flow channelling or shielding of adjacent fibres, the fibres inside each
133 bundle were made into a spindle shape to allow maximum contact between the air stream and the fibre.
134 As a subsequent work of previous research[22], this research work extends the previous experimental
135 testing conditions to a wider range, with the incoming air temperature up to 39°C and relative humidity
136 up to 40%. The variations of outlet air dry bulb temperature, wet bulb effectiveness, dew point
137 effectiveness and cooling capacity were studied by varying the incoming air dry bulb temperature from
138 27°C to 39°C and RH from 23% to 40%. The effects of various incoming air Reynolds number on the
139 heat and mass transfer coefficients, heat flux and mass flux across the polymer hollow fibre module
140 were analysed. Two sets of experimentally derived non-dimensional heat and mass transfer correlations
141 were summarized, which could be favourable for the future design of polymer hollow fibre integrated
142 evaporative cooling system.

143 2. Heat and mass transfer of hollow fibre evaporative cooling system

144 The 3D model of the hollow fiber module is shown in Figure 1 together with the temperature
 145 and humidity change profile. As illustrated in Figure 1, the incoming hot and humid air gets
 146 in contact with the porous hollow fiber module, inside which the water will be circulating
 147 around. As water evaporates through the hollow fibers, it will extract energy from the
 148 incoming air causing the temperature to drop. As reported by Johnson et al.[15], the
 149 resistance of fiber materials is very small, therefore it could be neglected. According to
 150 Rawangkul et al[5],

151 The sensible cooling capacity (Q_{\square}) supplied by the incoming air of such novel evaporative cooling
 152 system can be calculated by:

$$153 \quad Q_{\square} = m_a C_{pa} (T_1 - T_2) = \frac{C_{pa} \rho_{\square} V (T_1 - T_2)}{3.6} \quad \text{Eq.(1)}$$

154 Where, Q is the flow of transferred heat (W);

155 m_a is the mass flow rate of the incoming air (kg/h);

156 C_{pa} is the specific heat of air at constant pressure, kJ/(kg K);

157 T_1 is the dry bulb temperature of the incoming air ($^{\circ}\text{C}$);

158 T_2 is the dry bulb temperature of the outgoing air ($^{\circ}\text{C}$);

159 ρ_{\square} is the density of the air, kg/m^3 ;

160 V is the volumetric flow rate of the incoming air, m^3/h ;

161

162 The rate of water evaporation is:

$$163 \quad m_e = m_a (\omega_2 - \omega_1) \quad \text{Eq.(2)}$$

164 Where m_e is the flow of evaporated water(kg/h);

165 m_a is the mass flow rate of the incoming air (kg/h);

166 ω_1 is the humidity ratio of the incoming air;

167 ω_2 is the humidity ratio of the outgoing air.

168 The rate of heat transfer (Q) and the rate of water evaporation (m_e) can be given as the
169 product of heat transfer coefficient and the mean logarithmic difference in temperature (ΔT),
170 and the product of the mass transfer coefficient and the mean logarithmic difference in the
171 water vapour density ($\Delta\rho_V$). These can be expressed in the following two equations:

172 $Q = h_H A_s \Delta T = q A_s$ Eq.(3)

173 $m_e = h_M A_s \Delta\rho_V = N A_s$ Eq.(4)

174 Where h_H is the coefficient of heat transfer (W/m^2K);

175 h_M is the coefficient of mass transfer (W/m^2K);

176 q is the heat flux (W/m^2);

177 N is the mass flux (mg/ m^2s);

178 A_s is the total surface area of the polymer hollow fibre (m^2);

179 The mean logarithmic difference in temperature (ΔT) and water vapour density ($\Delta\rho_V$) can be
180 calculated using following equations:

181
$$\Delta T = \frac{(T_2 - T_1)}{\ln\left(\frac{T_2 - T_{wb}}{T_1 - T_{wb}}\right)}$$
 Eq.(5)

182
$$\Delta\rho_V = \frac{(\rho_{V2} - \rho_{V1})}{\ln\left(\frac{\rho_{V1} - \rho_{wb}}{\rho_{V2} - \rho_{wb}}\right)}$$
 Eq.(6)

183 Where ρ_{V1} and ρ_{V2} are the water vapour density on entering and leaving the hollow fibres
184 (kg/m^3);

185 ρ_{wb} is the saturated water vapour density at the wet bulb temperature (kg/m^3).

186 The heat and mass transfer coefficients could be calculated from:

$$187 \quad h_H = \frac{Nu * k}{d_h} \quad \text{Eq.(7)}$$

$$188 \quad h_M = \frac{Sh * D_{AB}}{d_h} \quad \text{Eq.(8)}$$

189 Where k is the thermal conductivity of the air (W/mK);

190 d_h is the hydraulic diameter of the hollow fibre module, m. According to [23], d_h can be
191 calculated as :

$$192 \quad d_h = \frac{4 \text{ cross sectional area of flow}}{\text{wetted perimeter}} \quad \text{Eq. (9)}$$

193 According to the hollow fiber bundle configuration[24], d_h can further be expressed as:

$$194 \quad d_h = \frac{4 A_{flow}}{n\pi d_0 + \pi D} = \frac{(1-\varphi)D^2}{\pi d_0 + D} \quad \text{Eq. (10)}$$

195 Where φ is the packing fraction of the module;

196 d_0 is the outer diameter of a single fiber;

197 D is the diameter of the module shell (m);

198 The relationship between Reynolds number (Re), Prandtl number (Pr) and Nusselt number
199 (Nu), and the relationship between Reynolds number (Re), Schmidt number (Sc) and
200 Sherwood number (Sh) can be expressed by:

$$201 \quad Nu = C_1 Re^{m_1} Pr^{1/3} \quad \text{Eq.(11)}$$

202 $Sh = C_2 Re^{m_2} Sc^{1/3}$ Eq.(12)

203 Where C_1, C_2, m_1, m_2 are constants for the hollow fibre bundles.

204 Reynolds number can be calculated by:

205 $Re = \frac{u*d_h}{\nu}$ Eq.(13)

206 Where u is the incoming air velocity, m/s;

207 ν is the viscosity of the incoming air, m²/s.

208 From the previous study [22], using the experimental data, general empirical correlations for the non-
209 dimensional heat and mass transfer data for the proposed system are derived using mathematical data
210 regression techniques. In this research, as the experimental testing conditions are extended to a large
211 range, the applicability of these equations will be verified in section 4.

212 $Sh = 1.275 Re^{1/3} Sc^{1/3}$ Eq. (14)[22]

213 $Nu = 0.958 Re^{1/3} Pr^{1/3}$ Eq. (15)[22]

214 The wet bulb effectiveness (ϵ_{wb}) is an important expression used to characterise the air saturation
215 capacity of the polymer hollow fibre bundle. This is defined as the ratio between the thermal
216 difference on passing through the hollow fibre bundle ($T_1 - T_2$) and the maximum thermal difference
217 that would occur if the air were saturated ($T_1 - T_{wb}$):

218 $\epsilon_{wb} = \frac{T_1 - T_2}{T_1 - T_{wb}}$ Eq.(16)

219 Similarly, the dew point effectiveness (ϵ_{dew}) is defined as the ratio between the incoming air and the
220 outgoing air to the difference between the incoming air and its dew point temperature, as indicated by
221 the following expression:

222 $\epsilon_{dew} = \frac{T_1 - T_2}{T_1 - T_{dew}}$ Eq.(17)

223 **3. Experimental testing rig**

224 A lab scale experimental testing rig is developed, which integrates the hollow fiber evaporative cooler
225 with the evaporative cooling system. Polyvinylidene fluoride (PVDF) hollow fibres (manufactured by
226 ZENA Ltd.) with outside diameter of 0.8mm and inside diameter of 0.6mm, an effective pore size of
227 0.5 μ m and a porosity of 50% were used for the fabrication of the polymer hollow fibre module. The
228 polymer hollow fibre module consists of 5 fibre bundles (each contains 100 fibres), which were
229 connected at each bundle ends using T piece plastic tubing. The hollow fibre module was incorporated
230 into a circular aluminium tunnel, whose cross section diameter was 0.15m. In order to avoid the flow
231 channelling or shielding of adjacent fibres, the fibres in each bundle were compressed from both ends
232 to make the bundle into a spindle shape to allow maximum contact between the air stream and the fibres.
233 The detailed physical properties of the polymer hollow fibre module were summarized in **Table 2**.

234 The testing rig consists of a polymer hollow fibre module, an air tunnel, a water pump, a fan and a water
235 tank. The detailed schematic diagram is shown in Figure 2. A 5-litre water tank was used to provide
236 water circulation inside the fibre. In order to avoid any particle blockage within the polymer hollow
237 fibres, a water filter was allocated to improve the purity of the incoming water into the fibres. A flow
238 meter and a ball valve were included in the water circulation cycle with the aim to control the water
239 flow rate inside the fibre. In order to minimize the experimental testing errors, four humidity and
240 temperature sensors (EK-H4, Sensirion, UK) were located at the inlet (point 1 in Figure 2) and outlet
241 (point 2 in Figure 2) of the tunnel respectively, to measure the inlet and outlet conditions of the air
242 stream. Additional K type thermocouples were used to measure the water temperature entering and
243 leaving the hollow fibre module (point 3 and 4 in Figure 2). The aluminium tunnel was connected with
244 a variable frequency drive centrifugal fan, which was linked directly with the environmental chamber.

245 The experimental prototype image is shown in Figure 3. **The basic working principle of the experiment**
246 **is as following: The hot and humid air from the environmental chamber will be blown out by the blower**
247 **into the air tunnel. With the help of the circulation pump, water is circulating from the water tank into**
248 **the hollow fibre bundles to ensure that the fiber surface will get wetted throughout the tests. The**
249 **incoming hot and humid air will get in contact with the hollow fibre module. As water evaporates**

250 through the porous hollow fibres, the temperature of incoming air will be reduced, accompanied by the
251 increased relative humidity.

252 At the beginning of each test, the environmental chamber was set to the required temperature and
253 humidity level. As soon as the temperature and humidity reached the desired values, the water pump
254 for the fibre module and the fan in the air stream direction was switched on. The air velocity was
255 measured at the five different positions along the cross sections of the outlet aluminium tunnel, using
256 the air velocity probes connected to a recorder (Testo 454). The dynamic pressures in the upstream and
257 downstream of the air flow were recorded using Pressure transducers (Ge UNIK 5000).

258 For each test, the temperature and humidity values were recorded every 20 seconds until the time when
259 the system reach steady states as indicated by the humidity and temperature sensor readings. The
260 accuracy of the measuring instruments used was: $\pm 0.2\%$ for temperature, $\pm 0.5\%$ for pressure, $\pm 2\%$ for
261 air velocity, and $\pm 2\%$ for relative humidity.

262 4. Results and discussion

263 As indicated in Table 3, the experiments were carried out for the proposed novel evaporative cooling
264 system with various inlet air dry bulb temperatures (27°C, 30°C, 33°C, 36°C and 39°C), and various
265 inlet air relative humidity (23%, 32% and 40%). Figure 4 presents the variation of outlet air dry bulb
266 temperatures with respect to various inlet air dry bulb temperatures under different incoming air
267 relative humidity. For incoming air temperature in the range of 27-39°C and the incoming air relative
268 humidity varying from 23%, 32% to 40%, it can be observed that the outlet air temperature is
269 dramatically affected by inlet air relative humidity at constant incoming air temperature. At the same
270 inlet air dry bulb temperature, the higher incoming air relative humidity will lead to the lower outlet
271 air dry bulb temperature. For instance, at the inlet air dry bulb temperature of 30°C, the outlet air dry
272 bulb temperature were 24.8°C, 25.7°C and 26.4°C respectively for RH of 23%, 32% and 40%. This
273 shows that the proposed novel evaporative cooling system has great potential to be used in hot and dry
274 climatic conditions. On the other hand, Figure 4 shows that the outlet air dry bulb temperature tends to
275 form a linear relationship with the inlet air dry bulb temperature at the same inlet air relative

276 humidity. The slopes of the outlet air dry bulb temperature at constant relative humidity are in the
277 range of 0.7-0.74. This means that, when increasing the inlet air temperature by 10°C, the outlet air
278 temperature will be improved by 7-7.4°C.

279 Figure 4 also compares the proposed experimental testing data with the outlet dry bulb temperature
280 presented by Dohnal, et al [25], for a compact bundle of 600 polypropylene hollow fibres placed
281 horizontally down the air stream, with the inlet air dry bulb temperature of 24.6°C, and the RH= 25%.
282 It can be found that the presented experimental results agrees well and even shows better cooling
283 performance compared with the testing results obtained by Dohnal et al [25]. This might be due to the
284 fact that the proposed evaporative cooling system adopted the novelty of using spindle shape fibre
285 bundles to allow maximum contact between the air stream and the fibre. While in the experiments
286 proposed by Dohnal et al. [25], the shielding from adjacent fibre in the entire bundles resulted in
287 decreased heat transfer performance.

288 The wet bulb effectiveness of the proposed novel evaporative cooling system is illustrated in Figure 5.
289 It can be found that, with the incoming air dry bulb temperature in the range of 27-39°C, and RH of
290 23%, 32% and 40%, the wet bulb effectiveness varies from 0.32-0.45. Higher inlet air dry bulb
291 temperature leads to greater wet bulb effectiveness, due to the fact that larger temperature depression
292 is obtained for higher inlet air temperature. In addition, lower inlet air relative humidity will result in
293 higher wet bulb effectiveness. The reason is due to the fact that drier incoming air with small relative
294 humidity actually represents larger driving force of vapour pressure difference between the inlet and
295 outlet air condition. Thus the drier incoming air can potentially absorb more moisture during the water
296 evaporation process. Therefore, more latent heat will be required during the water evaporation
297 process. Consequentially, a larger amount of sensible heat of the processed air will be transferred
298 from the incoming dry air to the outgoing wet air. This leads to a much lower outlet air temperature
299 compared with air of higher incoming relative humidity. Comparable value of wet bulb effectiveness
300 obtained by Dohnal et al. [25] is also included in Figure 5. With the inlet air dry bulb temperature of
301 24.6°C, and the RH= 25%, the wet bulb effectiveness achieved by Dohnal, et al. [25] was 0.354,
302 which showed a good agreement with the presented experimental results.

303 Figure 6 illustrates the variations of dew point effectiveness with respect to different inlet air
304 conditions. When air inlet dry bulb temperature increases from 27°C to 39°C, the dew point
305 effectiveness is in the range of 0.18-0.3. At the same inlet air RH, higher inlet air temperature leads to
306 higher dew point effectiveness, due to the fact that larger temperature depression is obtained for
307 higher inlet air temperature. Moreover, when the inlet air dry bulb temperature maintains at the same
308 level, lower inlet RH will result in higher dew point effectiveness. The reason is similar to what we
309 present in last paragraph, as drier incoming air with smaller relative humidity actually represents
310 larger driving force of vapour pressure, thus leads to higher cooling performance. The dew point
311 effectiveness **obtained** in this experimental study is slightly better than the results achieved by Dohnal
312 et al. [25], due to the fact that the at lower air velocity, the individual fibres within the bundles were
313 shielded from the air stream, as the majority of the air will go past the outer layer of the fibre bundles.
314 Therefore, such intra-bundle shielding effect in Dohnal et al. [25]'s testing configurations might lead
315 to reduced heat transfer performance.

316 Figure 7 depicts the variations of cooling capacity with respect to various inlet air dry bulb temperatures
317 under different incoming air relative humidity. Generally, the cooling capacity increases with the
318 improvement of the inlet air dry bulb temperature from 27°C to 39°C. With the same inlet air dry bulb
319 temperature, lower inlet air RH will lead to higher cooling capacity. For instance, when inlet air dry
320 bulb temperature is fixed at 30°C, the cooling capacities are 125.2W, 109.9W and 83.1W respectively
321 for the inlet air RH equals to 23%, 32% and 40%. The reason is due to the fact that, as shown in **Eq. (1)**,
322 the cooling capacity is proportional related to the differences between the inlet air and outlet air dry
323 bulb temperature. Drier incoming air with small relative humidity actually represents larger driving
324 force of vapour pressure difference between the inlet and outlet air condition. Thus the drier incoming
325 air can potentially absorb more moisture, which leads to a much lower outlet air temperature compared
326 with air of higher incoming relative humidity. Therefore, the cooling capacity shows decreased trend
327 when increasing the inlet air RH from 23% to 32% and 40%.

328 Figure 8 shows the experimental obtained heat flux under different incoming air relative humidity. It
329 can be observed that the heat flux will increase with the improvement of incoming air Reynolds

330 number. At the same Reynolds number, lower inlet air relative humidity will lead to greater heat flux.
331 For instance, at Reynolds number of 100, the heat flux increases from 798W to 1512W respectively
332 for RH equals to 40% and 23%. This means that approximately 1.8 times more heat is
333 transferred in the process when RH decreased from 40% to 23%. The reason is because, as
334 shown in Eq. (3), the heat flux is positively related to the inlet and outlet air dry bulb
335 temperature difference. Moreover, as shown in Figure 5, lower inlet air relative humidity will
336 yield much lower outlet air dry bulb temperature, which means greater inlet and outlet air dry bulb
337 temperature difference. This will consequentially lead to higher heat flux. Similar trend could be
338 found in Figure 9, showing the variations of mass flux with respect to Reynolds number under
339 different incoming air RH. It is obvious that higher Reynolds number will contribute to greater mass
340 flux under the same RH. While for the same Reynolds number, lower RH will lead to higher mass
341 flux.

342 Experimental determined overall heat transfer coefficient (h_H) and mass transfer coefficients (h_M) with
343 respect to Reynolds number under different inlet air relative humidity are illustrated in Figure 10 and
344 Figure 11. An increase in Reynolds number yields better heat and mass transfer between the air stream
345 and the water inside polymer hollow fibre. Despite of different inlet air relative humidity, h_H follows
346 the same linear relationship with Reynolds number, with variations less than 4.1% during the
347 experiments. The similar linear relationship between Reynolds number and h_M is illustrated in Figure
348 11. Further inspection of Figure 10 and Figure 11 indicate that, by increasing Reynolds number from 0
349 to 220, h_H changes from around 60 W/m²K to 250 W/m²K (about 4.2 times), while h_M improves from
350 0.01m/s to 0.25m/s respectively (about 2.5 times). This indicates the changes of Reynolds number has
351 more significant impact on the heat transfer coefficients than the mass transfer coefficients for the
352 proposed polymer hollow fibre integrated evaporative cooling system.

353 Figure 12 and Figure 13 respectively illustrated the non-dimensional heat and mass transfer data. It is
354 evident that the three different testing conditions yield more or less the same linear relationship
355 between the logarithm value of $Sh/Sc^{1/3}$, $Nu/Pr^{1/3}$ and logarithm value of Reynolds number.

356 The heat and mass transfer data are correlated by an empirical equation as shown in Eq. (11)-(12).
357 The Reynolds number exponent, α , in the Eq. (11)-(12) is an indication of the flow conditions. For
358 example, $\alpha=1/3$ indicates laminar flow with fully developed velocity profile. According to
359 literature[26], for $\alpha=0.5$, the indication is for developing velocity and concentration profiles, which is
360 in the entry region condition[27]. For α approaching 0.8-1.0, the flow is in the turbulent flow regime.
361 For the proposed novel evaporative cooling integrated hollow fibre system, the low Reynolds number
362 (less than 300) clearly shows the laminar flow regime. Therefore, the exponent α , for this study will
363 be chosen as $1/3$.

364 After obtaining the heat and mass transfer coefficients h_H and h_M , based on Eq. (7) and Eq. (8), Nu
365 and Sh number can be calculated accordingly. Reynolds number can be obtained by inserting
366 experimental measured air velocity and hydraulic diameter of the hollow fiber module into Eq. (13).
367 Pr number of is taken as 0.713 under the air temperature of 20°C . Based on the experimental obtained
368 data, general empirical correlations for the non-dimensional heat and mass transfer data of the
369 proposed system are derived using mathematical data regression techniques:

$$370 \quad Sh = 1.134Re^{1/3}Sc^{1/3} \quad \text{Eq. (18)}$$

$$371 \quad Nu = 0.976Re^{1/3}Pr^{1/3} \quad \text{Eq. (19)}$$

372 The above experimental determined correlation for the non-dimensional heat and mass transfer
373 data of the polymer hollow fibre integrated evaporative cooling system are compared with the
374 results (Eq. (20) and Eq. (14)- (15) obtained by Yang and Cussler[28] and X. Chen et al.[22].
375 Data from Yang and Cussler[28] and X. Chen et al.[22]'s experiments showed that the
376 exponent of Reynolds number were both equal to $1/3$, while the constant for heat transfer
377 correlations were 1.38 and 1.275 respectively. The deviation of the constants for Eq. (18) and
378 Eq.(20) is 21.7%. This is due to the fact that Yang and Cussler[28]'s experimental testing rig
379 contained 750 fibers (outside diameter 0.4mm) inside the module for a liquid-gas heat and mass
380 transfer process, while in the proposed system the fiber number is reduced to 500 with larger

381 outside diameter of 0.8mm. Such difference in testing configurations could lead to the
382 discrepancy between the two constants in the equations. On the other hand, different from the
383 subsequent work of previous research[22], the present work offers a wider range of testing
384 conditions, for inlet air temperature and relative humidity up to 39°C and 40%. However, the
385 deviations of the two heat transfer constants and mass transfer constants for the present research
386 and the previous study are very small, with only 12.3% and 2.1%, as indicated in Eq. (18)-(19)
387 and Eq. (14)-(15). Such deviations could be due to the unavoidable experimental errors. For
388 instance, performing the uncertainty analysis proposed by Moffat[29] for Equation (13), the
389 obtained uncertainty of the Reynolds number is 5.3%. Therefore, it can be concluded that for a
390 wider range of testing conditions, the obtained heat and mass transfer correlations actually
391 agree well with the previous research[22]. This further proves that the testing conditions do not
392 have significant effects on the non-dimensional heat and mass transfer data.

$$393 \quad Sh = 1.38Re^{0.4}Sc^{1/3} \quad \text{Eq. (20)[28]}$$

394 Experimental obtained mass transfer data in comparison with other correlations from literature is
395 presented in Figure 14. Different groups of mass transfer correlations developed by Zhukauska[30],
396 Yang and Cussler[28], Cote et al.[31], Chen et al.[22] and Johnson et al.[15] were used to perform the
397 comparisons. The results obtained from the proposed novel evaporative cooling system is similar to
398 Cote et al.[31]'s correlation, where the fibre bundle was used to transport oxygen through water. The
399 presented results are also very close to Johnson et al.[15]'s correlation, which was obtained from the
400 fibre array integrated evaporative cooler. It is also possible to observe that the mass transfer data for the
401 present research agrees relatively well with those obtained from the previous research[22]. Furthermore,
402 Figure 14 demonstrated that the mass transfer performance for the fibre bundles (29 fibres included)
403 achieved by Johnson et al.[15] are the worst compared with other fibre configurations. In the present
404 study, the fibre module consists of 5 fibre bundles which contain 100 fibres individually. However, the
405 mass transfer performance of such proposed fibre module shows significant improvement compared
406 with the mass transfer conditions of fibre bundles presented by Johnson et al. [15]. The reason is due to

407 the fact that, the spindle shape in the proposed system helps to avoid any over-shielding effect within
408 the fibre bundle. By compressing the fibre from both ends, the loosed spindle shape fibre bundle could
409 be obtained, which helps to enable better heat and mass transfer between each individual fibre and the
410 incoming air. For normal shape fibre bundle, such shielding effect is more significant at lower Reynolds
411 number, when the air will go past the outside of the fibre bundle, leaving the majority of the fibre inside
412 the bundle with very little contact with the incoming air. As the Reynolds number increases, better
413 contact between the fibres inside the bundle could be achieved, which leads to better mass transfer
414 performance, as shown in Figure 14.

415

416 5. Conclusions

417 A novel evaporative cooling system with hollow fiber bundles in the spindle shapes is proposed in this
418 research. With the aim to avoid the flow channelling or shielding of adjacent fibres, the fibres were
419 compressed into a spindle shape to allow maximum contact between the incoming air and the fibres.
420 This novel hollow fibre integrated evaporative cooling system will provide a comfortable indoor
421 environment for hot and dry area. Under various inlet air dry bulb temperatures (27°C, 30°C, 33°C,
422 36°C and 39°C), and various inlet air relative humidity (23%, 32% and 40%), the cooling performances
423 of the proposed novel evaporative cooling system were experimentally investigated. The variations of
424 outlet air dry bulb temperature, wet bulb effectiveness, dew point effectiveness and cooling capacity
425 were studied by varying the incoming air dry bulb temperature. Some conclusions can be found:

- 426 1) Increase the inlet air dry bulb temperature will lead to the increase of the outlet air dry bulb
427 temperature, cooling capacity, wet bulb effectiveness and dew point effectiveness. By keeping
428 the inlet air dry bulb temperature at constant value, increase the inlet air relative humidity will
429 lead to the decrease of cooling capacity, wet bulb effectiveness and dew point effectiveness;
- 430 2) The heat and mass transfer coefficients remain to be in linear relationships with respect to the
431 Reynolds number, despite of various inlet air relative humidity. With the Reynolds number in
432 the range of 10-220, the heat and mass transfer coefficients were in the range of 50-250W/m²K

433 and 0.05-0.3m/s. Two sets of non-dimensional heat and mass transfer correlations with respect
434 to Reynolds number were deviated from the experimental results, which showed good
435 agreements with other correlations from literature;

436 3) With inlet air dry bulb temperature being the same, increase the inlet air relative humidity will
437 lead to the decrease of heat flux and mass flux. With the inlet air relative humidity being equal,
438 increase the Reynolds number of incoming air will lead to the increase of heat flux and mass
439 flux;

440 4) Experimental obtained mass transfer data are illustrated and compared with other correlations
441 from literature. Due to the spindle shape hollow fibre module, the shielding with hollow fibre
442 bundles could be avoided greatly, therefore the mass transfer performance of the proposed
443 system demonstrated significant improvement compared with other devices reported in
444 literature. The non-dimensional heat and mass transfer data comparisons indicate that the
445 variations of experimental testing conditions have very limited impact on the heat and mass
446 transfer correlations.

447 5) The wet bulb effectiveness achieved in the proposed research was between 0.3 and 0.45 with
448 the packing fraction of 0.028. In the literature, the packing fraction was 0.28 in Zhang [21],
449 which was about 10 times higher than what the hollow fiber module presented in this paper.
450 However, the cooling effectiveness was only 1.5-2 times higher than that is obtained in this
451 research. This means that the design of this hollow fiber integrated evaporative cooler prototype
452 could provide comparably cooling performance as presented by other researchers, but with
453 lower packing fraction factor (fewer fibers included) . The future work could be concentrated
454 on increase the packing fraction by inserting more fibers into the one bundle, in order to
455 potentially increase the cooling effectiveness.

456 **Acknowledgement**

457 The authors would like to acknowledge the financial support and contributions from Innovate
458 UK (project code: 131821).

461 **Table 1 Summary of some published research results on evaporative cooling systems**

Reference	Research method	Evaporative cooler type/materials	Inlet air temperature (°C)	Outlet air temperature (°C)	Wet bulb effectiveness	Conclusions
Wu et al. [13]	Simulation (simplified model)	Direct evaporative cooler/Porous honeycomb paper	27-37	23-28	0.6-1.0	1) Frontal air velocity and pad thickness of the module are two key factors for the evaporative cooling system 2) The optimum frontal velocity was 2.5m/s
Lin et al. [32]	Experiment+ Simulation (ϵ -NTU method)	Dew point evaporative cooler/hydrophobic material	22-30	25-18	0.6-1.0	1) The saturation point of the working air is influenced by the working air ratio and channel height; 2) Overall heat transfer coefficient could achieve higher than 100W/m ² K
Zhang [33]	Experiment+ Simulation (fractal theory)	Direct evaporative cooler/hollow fiber membrane	NA	NA	NA	1) Experimental obtained relationship between Sherwood and Reynolds number was established using fractal model; 2) Membrane module with higher packing fraction could lead to better heat transfer performance.
Franco et al. [14]	Simulation	Direct evaporative cooler/ Porous paper	NA	NA	0.6-0.8	1)Comparisons of the cooling performance of five different porous materials revealed that the plastic grid block produced highest efficiency of 82.6%; 2) Higher efficiency will lead to lower specific water consumption
Zhao et al. [34]	Simulation	Dew point evaporative cooler/ Porous polygonal stack	28°C	NA	0.5-1.3	1)Cooling effectiveness increased with the increase of the working-to-intake air ratio; 2)Under UK summer design condition, the wet-bulb and dew-point effectiveness could reach up to 1.3 and 0.9.

463

464

Table 2 Geometric and physical properties of the polymer hollow fibre evaporative cooler

Property	Symbol	Values	Unit
Duct cross section diameter		0.15	m
Duct total length	L	0.9	m
Fibre number inside the module	N	5*100=500 (5 bundles)	
Fibre outside diameter	d_o	0.8	mm
Fibre inside diameter	d_i	0.6	mm
Nominal pore size		0.2	μm
Fibre porosity		0.6	
Packing density		10.67	m^2/m^3
Packing fraction	φ	0.028	
Polymer hollow fibre thermal conductivity	k	0.17	W/mK

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466

467

Table 3 Testing conditions of the novel polymer hollow fiber integrated evaporative cooling

468

system

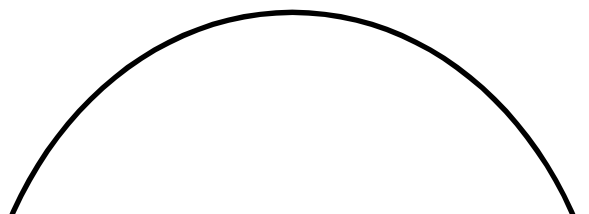
Property	Symbol	Values	Unit
Incoming air velocity	v_a	0.1-5.0	m/s
Water flow rate inside fibre		0.05	l/m
Incoming air temperature	T_a	27-39	$^{\circ}\text{C}$
Incoming air humidity	RH	23-40	%

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Figures

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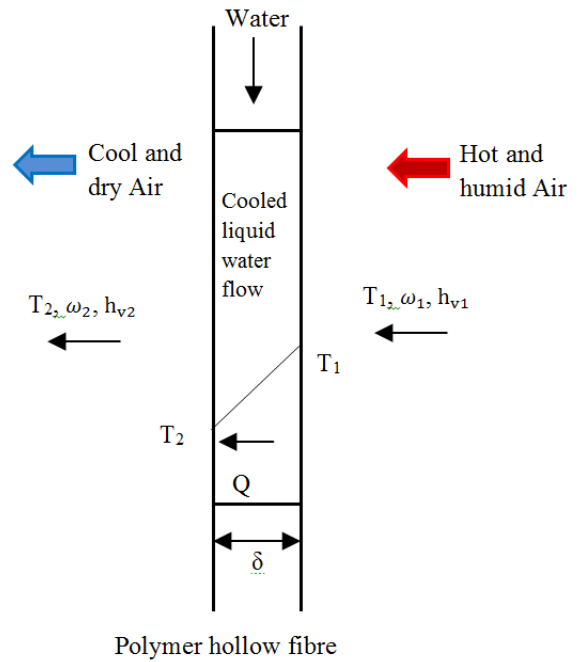
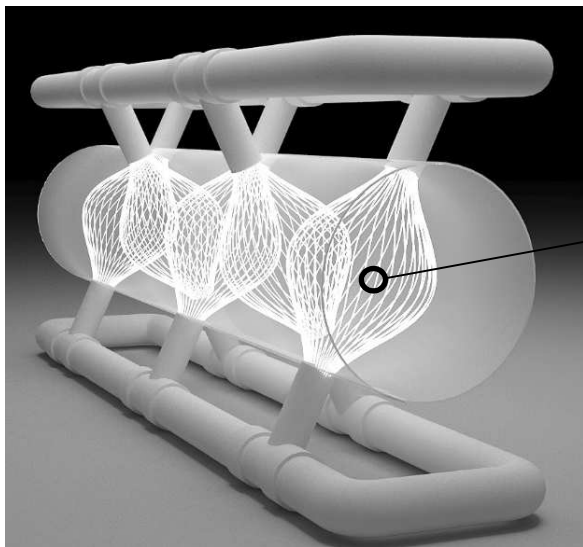
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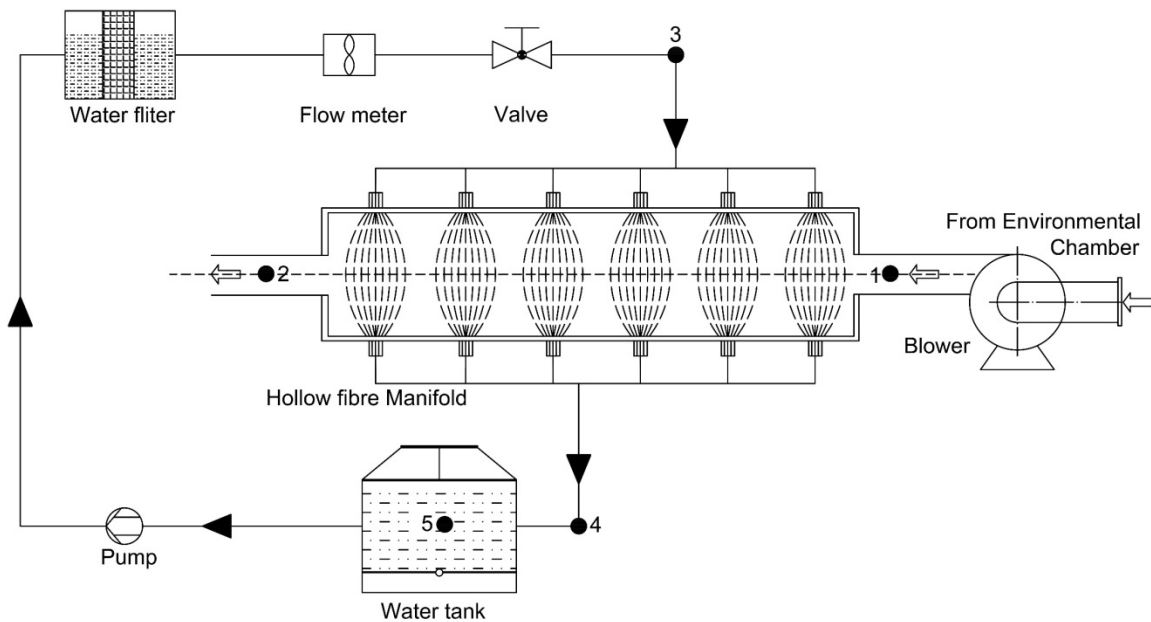
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Figure 1 The 3-D model and the temperature and humidity change profile in the hollow fibre module

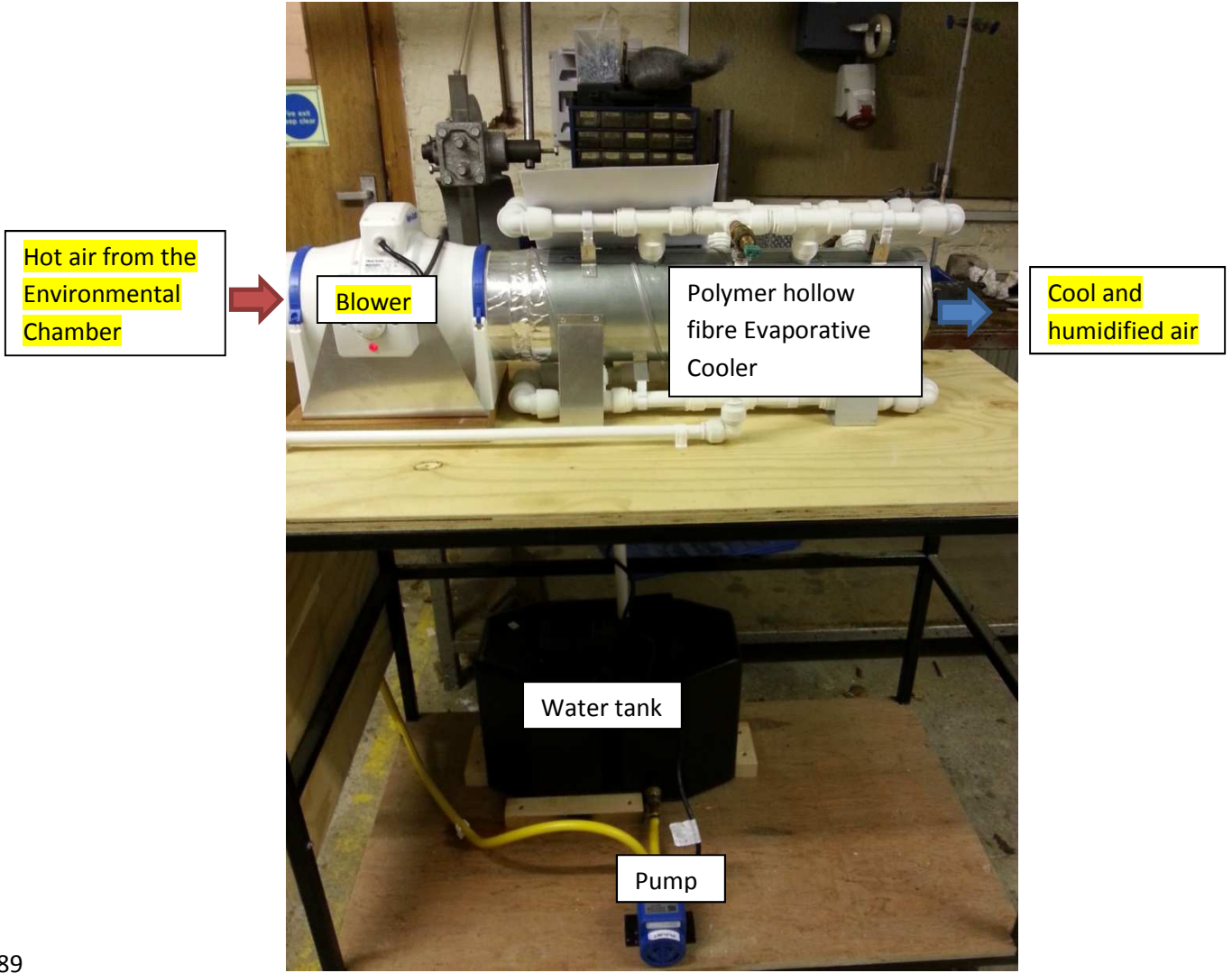


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Figure 2 Schematic diagram of polymer hollow fibre integrated evaporative cooling system

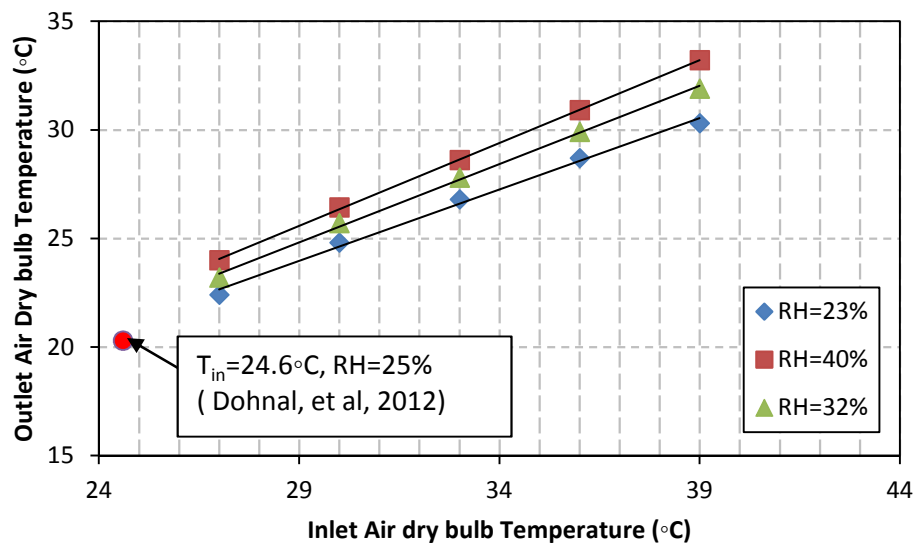
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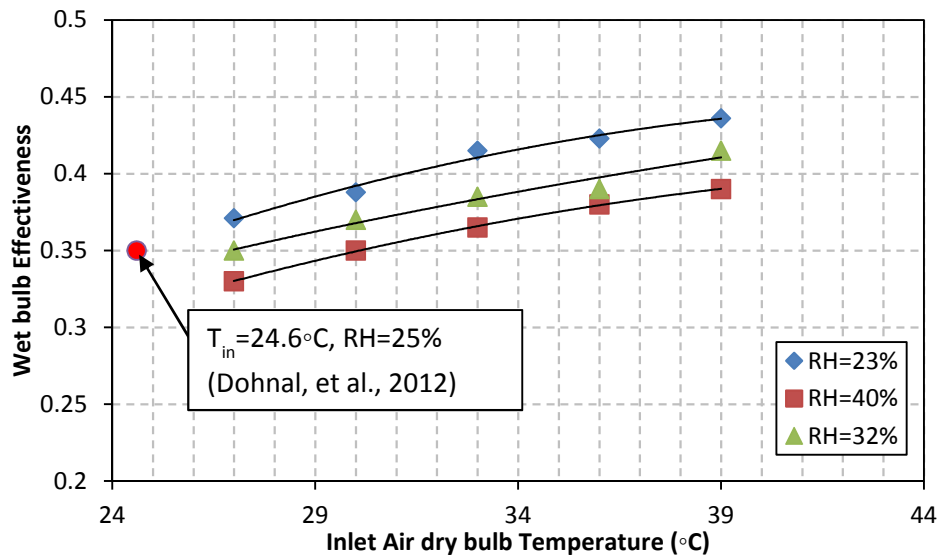
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Figure 3 Testing rig of polymer hollow fibre integrated evaporative cooling system

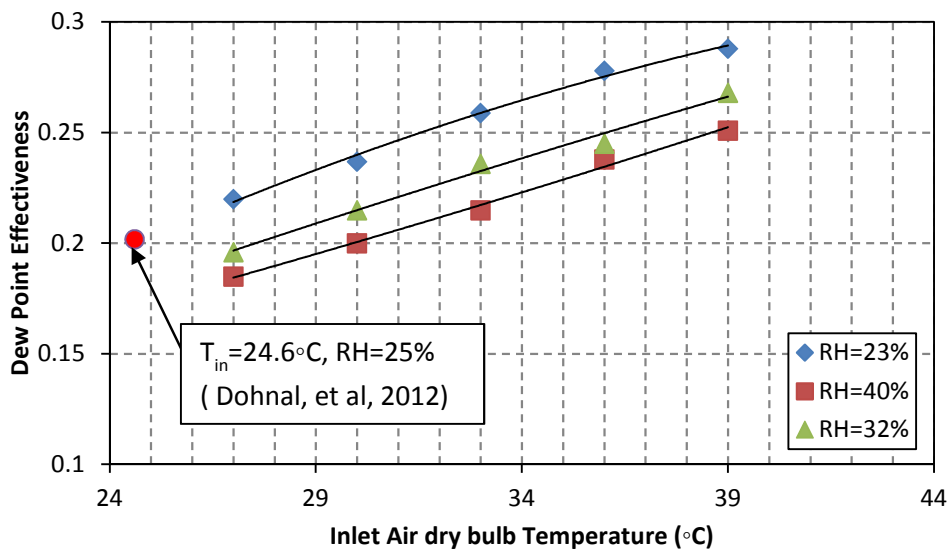


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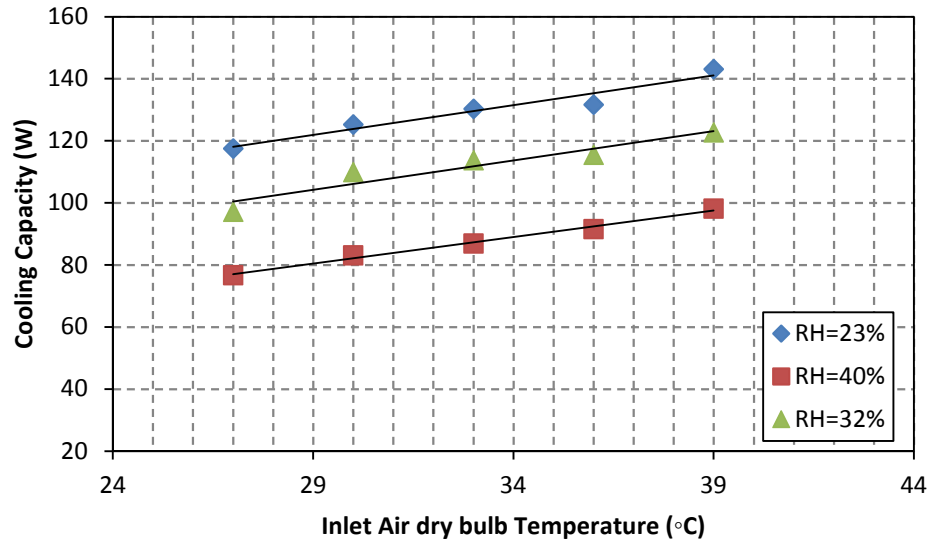
492 Figure 4 Variations of outlet air dry bulb temperatures with respect to various inlet air dry
 493 bulb temperatures under different incoming air relative humidity ($u=4.6\text{m/s}$)



494
 495 Figure 5 Variations of wet bulb effectiveness with respect to various inlet air dry bulb
 496 temperatures under different incoming air relative humidity ($u=4.6\text{m/s}$)



497
 498 Figure 6 Variations of dew point effectiveness with respect to various inlet air dry bulb
 499 temperatures under different incoming air relative humidity ($u=4.6\text{m/s}$)



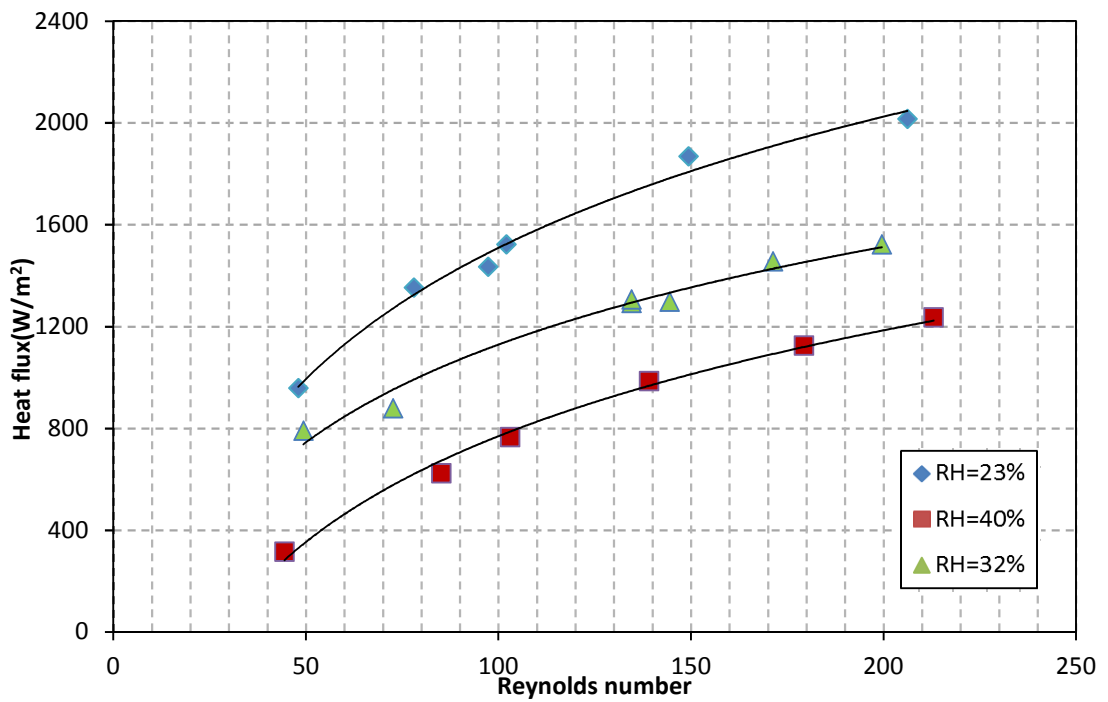
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Figure 7 Variations of cooling capacity with respect to various inlet air dry bulb temperatures

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under different incoming air relative humidity



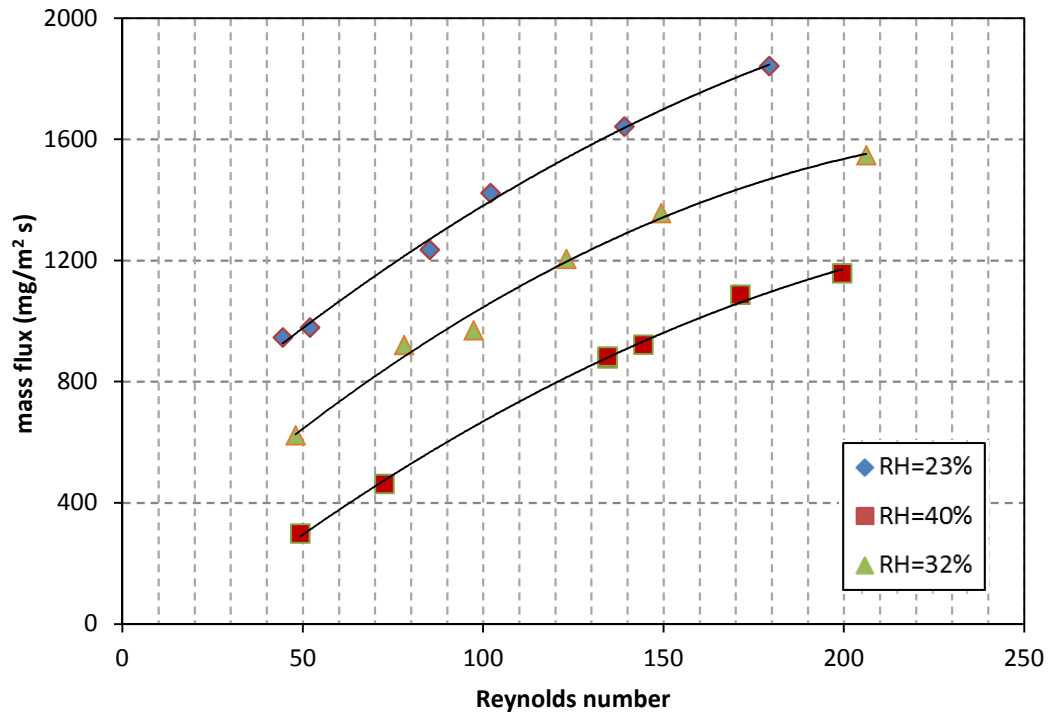
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Figure 8 Variations of heat flux with respect to Reynolds number under different incoming air

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relative humidity

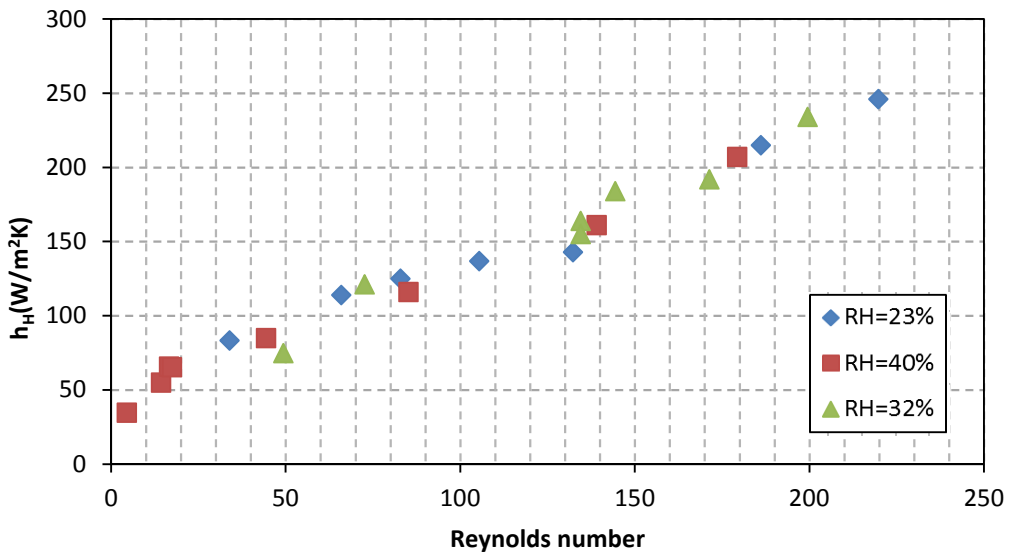


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Figure 9 Variations of mass flux with respect to Reynolds number under different incoming air relative humidity

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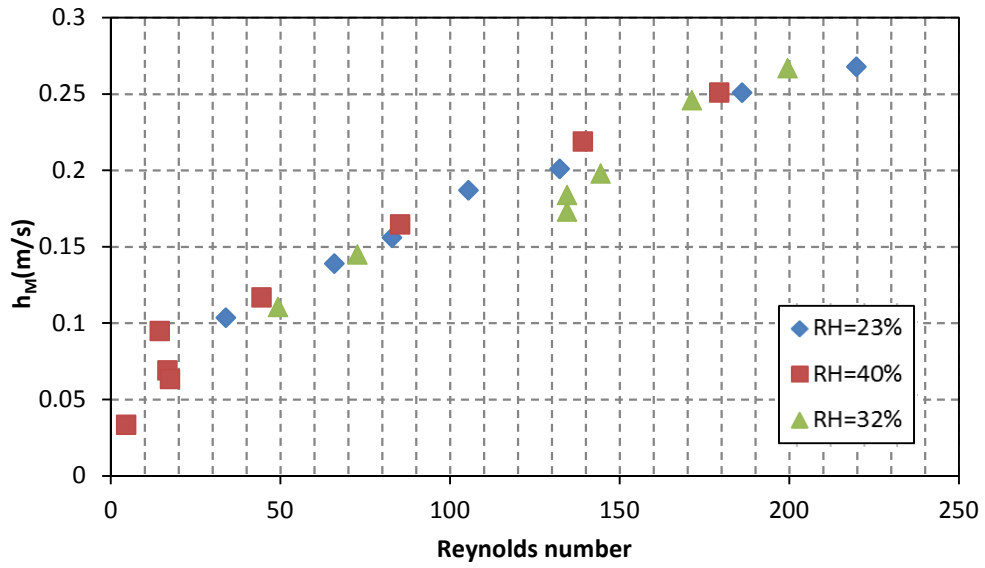
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Figure 10 Variations of heat transfer coefficients with respect to Reynolds number under different incoming air relative humidity

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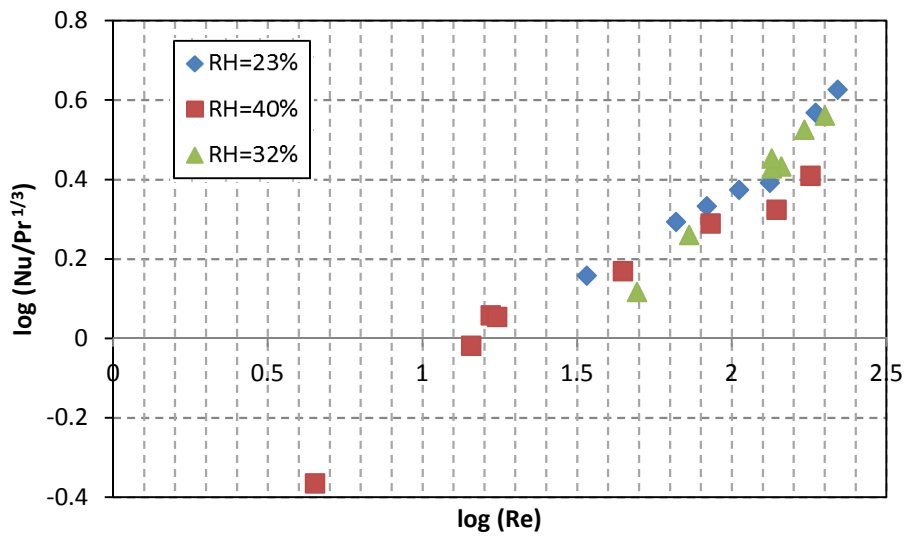


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Figure 11 Variations of mass transfer coefficients with respect to Reynolds number under different incoming air relative humidity

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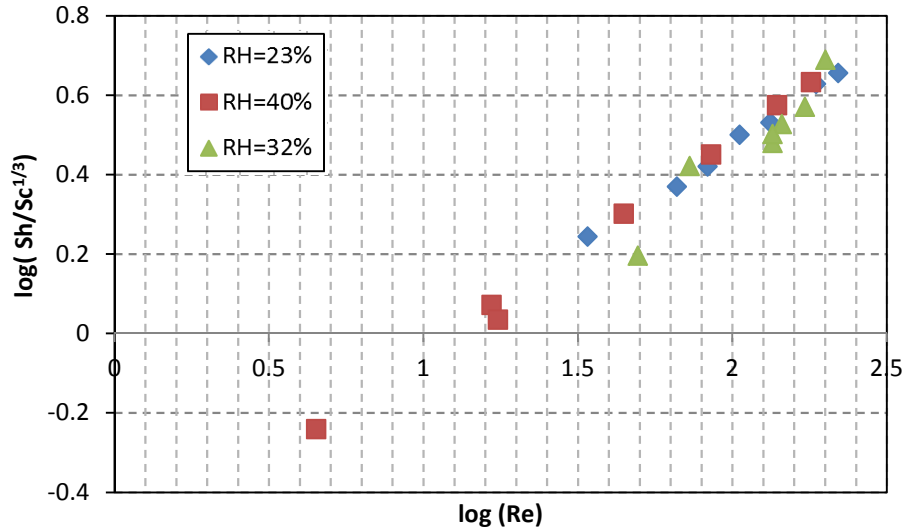
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Figure 12 Variations of Non-dimensional heat transfer data under different incoming air relative humidity

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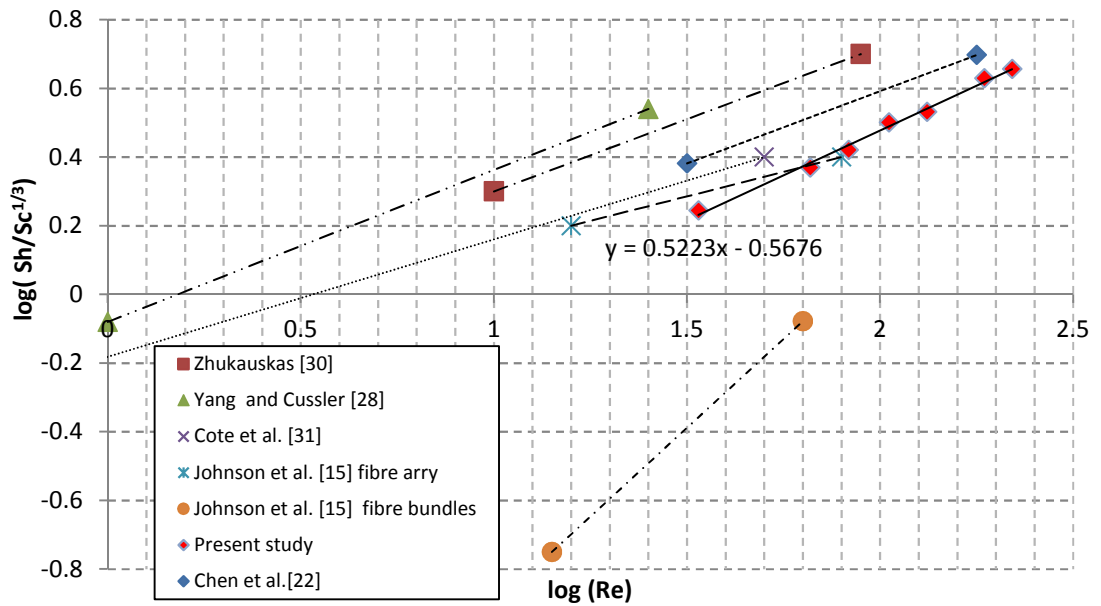
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Figure 13 Variations of Non-dimensional mass transfer data under different incoming air

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relative humidity



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Figure 14 Comparisons of mass transfer data of presented results with other correlations in

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literature

526 **Reference**

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