Research Article

Humidification Technique Using New Modified MiniModule Membrane Contactors for Air Cooling

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An experimental study is conducted to cool the ambient air using a new humidification technique. A wind tunnel is built with a test section formed by four modified MiniModule membrane contactors. An ambient air passes over the membrane contactors (cross flow) while water pumps through the contactors. Air temperature and relative humidity are measured upstream and downstream of the membrane contactors array which was used to humidify and cool the outdoor air. Five average air velocities (3.03, 3.33, 3.95, 4.52, and 5.04 m/s) and four water flow rates (0.0, 0.013, 0.019, and 0.025 kg/s) are used. Air velocity is measured at different locations along the centerline of the cross section. Using the modified MiniModule membrane contactors array dropped the air temperature by a maximum and minimum of $10.77^\circ C$ and $3.44^\circ C$, respectively, depending on the outdoor air. The corresponding maximum increase of the relative humidity is 4.65% which depends on the ambient condition. It is noticed that the evaporation process does not follow the isenthalpic lines therefore; heat transfers from the air as latent and sensible heats.

1. Introduction

This paper describes the potential of using the MiniModule membrane contactors array in evaporative cooling of ambient air which could be used in gas turbine applications. Cooling the air at compressor inlet is a well known technology used to increase the gas turbine capacity and efficiency. Air humidification can be used to cool the air at compressor inlet. This technology is inexpensive and simple to apply and its power consumption is low. Now, the humidification is carried out by spraying water in air flow upstream of the compressor inlet. Using this method requires high quality water to avoid corrosion and erosion of the compressor blades and scale off composition on compressor blades. Furthermore, droplets drift can increase water consumption in humidification process. Membrane evaporation is a new technology used in many applications including desalination juice concentration. Using this technology in air humidification eliminates blade problems mentioned above and droplets drift. In addition, low quality water can be used in humidification process. Inlet air cooling markedly enhances the performance of combustion turbines [1–6]. The turbine power increases at a lower cost per kW, and as an added bonus the heat rate also improves. Various approaches to cool the turbine inlet air have been employed. The two most common approaches (evaporative cooling and mechanical refrigeration) have been extensively applied and are well developed and documented. Combustion turbines have ambient temperature sensitivity: both the capacity and the efficiency decrease as the ambient temperature increases. The power demand of the compressor section of the turbine is proportional to the absolute temperature of the inlet air. The compressor capacity is proportional to the density of the inlet air, which is inversely proportional to the absolute temperature. Therefore higher ambient temperatures negatively affect both capacity and efficiency of the turbine. The turbine manufacturers supply curves detailing with both the power output and the heat rate as a function of ambient temperature.

Erickson et al. [3] reported that a 300-refrigeration ton aqua ammonia refrigeration unit is required to cool the inlet of a 5 MW gas turbine from $35^\circ C$ to $5^\circ C$. This cooling increases the power output by 1 MW, and the added power...
is at a marginal efficiency of 39%, compared to 29% for the base turbine power. Alhazmy and Najjar [5] reported that the spray coolers appear to be capable of boosting the power and enhancing the efficiency of the gas turbine power plant in a way that is less expensive than cooling coils. Although the performance of spray coolers is deeply influenced by the ambient temperature and humidity, they operate efficiently during hot and dry climatic conditions. The analysis of Alhazmy and Najjar [5] has shown that the spray cooler reduces the temperature of incoming air by 3–15°C, enhancing the power by 1–7% and improving the efficiency by 3%.

The membrane evaporation is a new technology which utilizes the evaporative cooling technique in air conditioning, water desalination, juice concentration, and other applications [7–12]. Microporous hydrophobic membranes have been examined by Loeb [7] for possible use as containers in the evaporative cooling of water, particularly in desert climates. The potential of using hollow fiber membranes in evaporative cooling applications for space air conditioning has been reported by Johnson et al. [8]. Their results showed that reasonable numbers of fibers and membrane surface area could provide cooling effectiveness comparable to the conventional evaporative cooling equipment. Experimental and theoretical investigations of air humidification/dehumidification processes were carried in a hollow-fiber membrane contactor by Bergero and Chiari [13]. Their experimental results indicate high mass transfer efficiency for both humidification and dehumidification.

Recently, Zhang [14] has reported numerical and experimental study about parallel-plates membrane cores used in air-to-air heat exchangers for fresh air heat and moisture recovery. His results indicated that for those membrane structures, when the channel pitch is below 2 mm, the flow distribution is quite homogeneous and the sensible and latent heat performance deteriorations due to flow mal distribution are below 9% and can be neglected. However, when the channel pitch is larger than 2 mm, the maldistribution is quite large and the consequent thermal and latent performance can be deteriorated by 28%. More recently, a numerical simulation for mass transfer through a porous membrane of parallel straight channels has been reported by J. Lu and W.-Q. Lu [15]. In their study, two types of flows, channel flow and ultrafiltration flow, have been physically described. Their results have displayed the flow and solute distribution patterns inside channels, described the ultra-filtration profiles along the surface of the porous membrane, and disclosed an existent nanoscale reverse osmosis problem. Ceramics and ceramic matrix composites for heat exchangers in advanced thermal systems have been reviewed by Sommers et al. [16]. In their paper, the current state of the art of ceramic materials for use in a variety of heat transfer systems has been reported. Coupled heat and mass transfer in a counter flow hollow fiber membrane module for air humidification has been reported numerically and experimentally by Zhang and Huang [17]. Their results showed good matching between the numerical and experimental humidification and cooling effectiveness. In another study, Zhang [18] has shown the heat and mass transfer characteristics of fibers bundle in an aligned and staggered arrangement in air cross flow. The flow maldistribution and the consequent performance deterioration in a cross flow hollow fiber membrane module used for air humidification has been studied by Zhang et al. [19]. Their results showed that the packing fraction affects the flow maldistribution substantially. It should also be mentioned that other methods of cooling the inlet air is known as wet media evaporative cooling technology, which offer 85 to 90% of evaporation efficiency, may not require high water quality but they need huge amount of water. These methods offer reduced risk of erosion to the compressor blades and corrosion to turbine inlet duct structure. It should be noted that the current suggested method of research presents a new interesting engineering concepts to enhance the performance of the gas turbine in spite of its low efficiency. Therefore, further researches need to be performed to improve the efficiency of the proposed technology, such as increasing the number of MiniModule membrane contactors array, which leads to the increase of exchange surface between the membrane and airflow. Another advantage is the possibility of using recycled water (chemically treated water) since the fresh quality water in desert area such as Saudi Arabia is mostly available through the desalination plants and commonly used for human beings supply.

In this paper, evaporation technology is used to humidify the air for cooling before it enters the compressor of a gas turbine. A wind tunnel is built and a matrix of four MiniModule membrane contactors is used as a test section where water pumps through (cross flow heat exchange). The relative humidity, temperature, and pressure losses are measured before and after the test section. The air velocity over the test section and the water flow rates through the contactors are measured and the effectiveness of cooling and humidification is also determined.

2. Experimental Setup

The experimental apparatus consists of a wind tunnel and four MiniModule membrane contactors serving as a test section. The wind tunnel consists of three parts. The first part is the conical adaptor of length 150 cm that converts the blower circular section of 50 cm in diameter to rectangular section of 28 cm × 14 cm as seen in Figure 1. The blower
Figure 2: Schematic of the wind tunnel, (a) wind tunnel (dimensions in cm); 1-test section, 2-conical adaptor, 3-variable speed fan, 4-intake duct, 5-inlet air and (b) complete wind tunnel assembly.

Figure 3: The test section showing the MiniModule membrane contactors array.

Figure 4: Dimensions of the MiniModule membrane contactor (as provided by the manufacturer).

Part has a fan of 2.2 kW (1710 RPM) as seen in Figure 1 and schematically in Figure 2(a) and the complete wind tunnel assembly is shown in Figure 2(b). The second part is the test section and the third part is the settling or the straightener section. This settling section consists of three rectangular ducts each is 102 cm length with a cross section of 28 cm × 14 cm in addition to one more duct of 50 cm length with the same cross section that makes the total settling length of 356 cm (Figure 2(a)). A form of honeycomb is used at the entrance section to provide more uniform flow. The second part is the test section that consists of four MiniModule membrane contactors connected in parallel as seen in Figure 3. The characteristics of the membrane contactors are given in Table 1 (as specified by the manufacturer, Membrana-Charlotte A Division of Celgard, LLC., USA). Figure 4 shows the dimensions of the MiniModule contactors. It should be noted that modification has to be done on the design of the outer shell of the contactors in order to make a direct contact between the flowing air and the membrane inside the contactors. This was done by making two slits (window like) through the outer cylindrical shell. The area of each window is equivalent to 18.3 cm (height length) × 5.1 cm (arc length). Figure 5 shows the contactors after the modification. The measurement devices such as digital low range water flow meters, velocity and temperature sensors, pressure gauges, and relative humidity devices are installed on the wind tunnel and connected to data acquisition systems. Relative humidity measuring devices and temperature sensors are installed in their housing on the duct before and after the test section. Variable speed fan of the wind tunnel is used to achieve variable air flow velocities. Two computers are used; one is connected to the data acquisition system for...
Table 1: Characteristics of the MiniModule membrane contactors (as supplied by the manufacturer).

| Contactor characteristics (1.7 x 8.75) |   |
|---------------------------------------|--|
| Cartridge configuration              | Parallel flow. Lumen side liquid flow |
| Liquid side flow guidelines           | < 2500 mL/min (0.66 gpm) |
| Membrane                              | ×50 fiber |
| Porosity                              | 40% porosity |
| OD/ID                                 | 300 μm OD/220 μm ID |
| Potting material                      | polyurethane |
| Active surface area                   | ID: 1.0 m² (10.7 ft²) |
| Number of fibers                      | OD: 1.4 m² (15.0 ft²) |
| Maximum temperature/pressure          | 7400 |
| Priming volume (ID)                   | 4.2 kg/cm² (4.1 bar, 60 psig) at 25°C (77°F) |
|                                      | 3.2 kg/cm² (3.1 bar, 45 psig) at 45°C (113°F) |
| Housing characteristics               | Lumen side 73 mL, shell side 132 mL |
| Material                              | Polycarbonate |
| Flange connections                    | Standard female luer lock supplied with two 1/8 inch hosebarb adaptors which mate to 1/4 inch ID tubing |
| Shell side (gas/vacuum)               | 1/4 inch FNPT |
| Lumen side (wetted surface)           |   |
| Seal options                          |   |
| Material                              | EPDM (ANSI/NSF61, FDA CFR title compliant) |
| Weight (with adapters)                |   |
| Dry                                   | 0.22 kg (0.48 lbs.) |
| Liquid full (lumen side)              | 0.29 kg (0.64 lbs.) |
| Shipping weight                       | 0.40 kg (0.88 lbs.) |

Figure 5: Modified MiniModule contactors; the white color is the X50 fiber membrane and the brown color presents the sealing between the two webs and the membrane after making the modification.

the air temperature and relative humidity measurements and the other for air velocity measurements. The air velocity is measured using and 8 channel hot wire anemometer.

3. Experimental Procedure

Air flow is established using a variable speed motor controller by adjusting the power source frequency which is fed to the fan motor. Centrifugal pump is used to pump the water through the array of contactors and the flow rate is adjusted by using water control valve. Air velocity is measured by using hot wire anemometers. Seven velocity sensors are fixed on vertical strips at vertical y-distances 7.42, 28.4, 51.24, 70.0, 88.76, 111.58, and 132.58 mm to scan the air velocity. This vertical strip holding the sensors slides along the horizontal direction of the rectangular cross section. The coordinates of the sensors are shown in Figure 6. Following this procedure, air velocities are measured at 42 points over the cross section. A data acquisition system is connected to a PC and used to

Figure 6: Locations of the velocity measurement sensors (dimensions are in mm).
collect the air velocities. The mean air velocity is estimated based on the 42 measured values. Water flow rate in the test section is measured using a turbine flow meter. Temperature along the air duct is measured using thermocouples and thermistors. K-type thermocouples are distributed along the air duct. Four thermocouples are used; two of them at the inlet and the others at the outlet to measure the dry and wet bulb temperatures, respectively. Two thermocouples are fixed before and after the test section to measure the air temperatures across the test section. Two more thermocouples are used to measure the water temperatures in the water tank and the outlet of the test section. One thermocouple is used to measure the room temperature. Humidity sensors are used to measure the relative humidity before and after the test section. Humidity sensors also equipped with thermistors to measure temperatures. Two pressure transducers are used to measure the pressure drop across the test section. Two data acquisition systems were connected to a laptop to collect the data of humidity sensors, pressure transducers, and thermocouples.

Table 2 shows the uncertainty of the measured parameters. The thermocouples and data acquisition system are calibrated at ice and boiling points. At the ice point, the error in thermocouple readings are in the range from $-0.2$ to $+0.4\, ^\circ\text{C}$ however; at the boiling point, are in the range from $0.15$ to $0.4\, ^\circ\text{C}$. The flow meter used is a turbine flow meter (FTB602) made by Omega Engineering, INC. The flow meter

| Parameter          | Uncertainty      |
|--------------------|------------------|
| Temperature        | $-0.2$ to $+0.4\, ^\circ\text{C}$ |
| Relative humidity  | $\pm2\%$ R.H     |
| Pressure           | $\pm0.25\%$      |
| Velocity           | $\pm2\%$         |
| Water flow rate    | $2\%$            |
is calibrated using a balance scale and a stop watch. The difference between the reading of the flow meter and that of the measured flow rates is $-2\%$ at most.

4. Results and Discussion

Measurements of the temperature and the relative humidity are taken before and after the MiniModule membrane contactors test section. Figures 7(a), 7(b), 7(c), and 7(d) show the evolutions of the average temperature before and after the test section for various air velocities. It should be mentioned that the contactors and all the connecting pipes are full of water at the start of the experiment with zero water flow rate which means that the water inlet and exit valves to and from the test section plumbing system, respectively, are closed. Furthermore, this experiment was done during the winter time; therefore it was necessary to heat the air before it enters to the test section in order to simulate the most of the year around temperature. Consequently, in Figure 7(a) it is clear that the temperature increases from the laboratory ambient temperature passing by the heater then to the test section. The average inlet and exit temperatures to and from the test section are $58.2$ and $47.4\,^\circ C$, respectively, where the air temperature dropped by $10.8\,^\circ C$ for air velocity of $3.03\, m/s$. This drop in the air temperature presents a $32.3\%$ cooling ratio. The effect of increasing the air velocity is to reduce the drop in the temperature across the test section as seen to be $5.5$, $4.1$, and $3.9\, ^\circ C$ as the velocity increases to $3.33$, $3.945$, and $5.039\, m/s$ as seen in Figures 7(b), 7(c), and 7(d), respectively.

Similar curves are obtained for various water flow rates through the contactors and the results are summarized in Figure 8. This figure shows that the effect of moving water through the contactors on the magnitude of the temperature at zero flow rate is almost negligible. This figure also summarizess the temperature drop between the flowing air before (TBTS) and after (TATS) the test section at all water flow rates and for different air velocity. It is also clear that the conclusion drawn earlier from Figure 7 about the reduction of the temperature drop as the air velocity increases is still valid for the other water flow rate through the contactors. The relative humidity before and after the test section for various values of air velocity and for contactors full of water corresponding to zero water flow rate is shown in Figures 9(a), 9(b), 9(c), and 9(d). As mentioned earlier this experiment was done in winter time; therefore the relative humidity decreases as the temperature increases (see Figure 7(a)) as seen in Figure 9(a).

The average inlet and exit relative humidity to and from the test section are $1.91\%$ and $6.55\%$, respectively, where the relative humidity increases by $4.64$ for air velocity of $3.03\, m/s$. This increase in the relative humidity presents a $229.14\%$ of the inlet relative humidity. The effect of increasing the air velocity is to increase the relative humidity across the test section as seen to be $7.82$, $13.30$, and $13.48$ as the air velocity increases to $3.33$, $3.945$, and $5.039\, m/s$ as seen in Figures 9(b), 9(c), and 9(d), respectively. Similar curves are obtained for various water flow rates through the contactors and the results are summarized in Figure 10. This figure shows that the relative humidity decreases as the water flow rate increases up to $0.0125$ and then it starts to increase again especially at higher values of air velocities. This figure also summarizes the decreasing and increasing rate of the relative humidity before (RHBTS) and after (RHATS) the test section at all water flow rates and for different air velocities. It is also clear that the
Figure 9: The effect of increasing the air velocity on the relative humidity before and after the test section for various air velocities, (a) $v_a = 3.03 \text{ m/s}$, (b) $v_a = 3.33$, (c) $v_a = 3.95$, and (d) $v_a = 5.04 \text{ m/s}$.

Conclusion drawn earlier from Figure 9 about the increase of the relative humidity as the air velocity increases is still recognized for other water flow rates through the contactors. Table 3 shows the performance of the experimental data at the test section using the evaporative cooling and humidification system effectiveness $\varepsilon_c$ and $\varepsilon_h$, respectively, as

$$\varepsilon_c = \frac{(T_{ai} - T_{ao})}{(T_{ai} - T_{wb})}, \tag{1}$$

$$\varepsilon_h = \frac{(\omega_{ao} - \omega_{ai})}{(\omega_{wb} - \omega_{ai})}, \tag{2}$$

where the subscripts ai, ao, wbi, and wi stand for inlet air, outlet air, wet bulb inlet, and water inlet, respectively. It should be noted that expression (2) is defined following Zhang and Huang [17], where $\omega_{wi}$ is the specific humidity using the water temperature at 100% relative humidity. It should be noted that the wet bulb temperatures at the duct inlet and outlet are measured using sensors covered with wet wicks. Wet bulb temperatures at test section inlet and outlet are obtained from the psychrometric chart (EES software) assuming that the process from the duct inlet to test section inlet has the same humidity ratio and from the test section outlet to duct outlet has a constant humidity ratio. Figures 11(a) and 11(b) show the cooling and humidification system effectiveness $\varepsilon_c$ and $\varepsilon_h$, respectively, at various water flow rates. As seen in Figure 11(a), the cooling effectiveness drops sharply as the air flow rate ($m_a$) increases up to about 600 kg/hr. The fitting curve to the data is shown as a solid line and given by the following third order polynomial to cover all water flow rates:

$$\varepsilon_c = 425.584 - 2.03565m_a + 0.003324m_a^2 - 1.77892 \times 10^{-6}m_a^3,$$  \tag{3}
Figure 10: The effect of water flow rate on the relative humidity before (RHBTS) and after (RHATS) the test section for various air velocities.

Table 3: Data for air and water flow, humidity ratio RH, dry bulb temperature before and after the test section $T_{ai}$ and $T_{ao}$, specific humidity $\omega$, cooling effectiveness $\varepsilon_c$, and humidification effectiveness $\varepsilon_h$ for some runs.

| Runs | $m_a$ | $m_w$ | $V_a$ | $m_w$ | $T_{ai}$ | RH | $\omega$ | $T_{ao}$ | $\omega$ | RH | $T_w$ | $\omega_w$ | $\omega_{wb}$ | $\varepsilon_c$ | $\varepsilon_h$ |
|------|-------|-------|-------|-------|---------|-----|---------|---------|---------|-----|-------|-----------|-------------|--------------|--------------|
| 1    | 398.6 | 0     | 3.03  | 0     | 58.19   | 1.91| 0.0023  | 21.62   | 47.42   | 0.0047| 6.56  | 39.41     | 0.0512      | 0.0176       | 29.45        | 4.96        |
| 2    | 481.2 | 0     | 3.33  | 0     | 51.15   | 3.85| 0.0033  | 20.33   | 45.68   | 0.0052| 7.82  | 34.42     | 0.0382      | 0.0162       | 17.76        | 5.26        |
| 3    | 585.5 | 0     | 3.95  | 0     | 41.4    | 9.18| 0.0049  | 18.51   | 37.3    | 0.0057| 13.3 | 34.17     | 0.0377      | 0.0144       | 17.93        | 2.44        |
| 4    | 672.0 | 0     | 4.52  | 0     | 40.58   | 8.3 | 0.0042  | 17.74   | 36.36   | 0.0053| 12.98| 30.91     | 0.0310      | 0.0137       | 18.46        | 3.89        |
| 5    | 756.1 | 0     | 5.04  | 0     | 37.15   | 9.9 | 0.0042  | 16.58   | 33.32   | 0.0046| 13.48| 28.1      | 0.0262      | 0.0127       | 18.65        | 1.95        |
| 6    | 429.9 | 45    | 3.03  | 0.013 | 57.27   | 0.37 | 0.0004 | 20.02   | 48.78   | 0.0004| 0.55 | 21.69     | 0.0176      | 0.0159       | 22.81        | –0.02       |
| 7    | 479.4 | 45    | 3.33  | 0.013 | 51.93   | 0   | 0.0000  | 18.06   | 45.85   | 0.0007| 1.05 | 23.72     | 0.0200      | 0.014        | 17.94        | 3.48        |
| 8    | 583.1 | 45    | 3.95  | 0.013 | 42.08   | 1.82 | 0.0010 | 15.59   | 37.62   | 0.0020| 4.61 | 23.53     | 0.0198      | 0.0119       | 16.85        | 5.29        |
| 9    | 669.2 | 45    | 4.52  | 0.013 | 40.8    | 3.22 | 0.0016 | 15.71   | 35.73   | 0.0025| 6.5  | 22.96     | 0.0191      | 0.012        | 20.2         | 5.06        |
| 10   | 753.2 | 45    | 5.04  | 0.013 | 36.94   | 5.27 | 0.0022 | 14.82   | 33.5    | 0.0030| 8.84 | 23.97     | 0.0203      | 0.0114       | 15.54        | 4.69        |
| 11   | 430.9 | 67.5  | 3.03  | 0.019 | 57.12   | 0.11 | 0.0001 | 19.76   | 48.18   | 0.0006| 0.75 | 21.04     | 0.0169      | 0.0156       | 23.92        | 2.55        |
| 12   | 481.0 | 67.5  | 3.33  | 0.019 | 50.4    | 0   | 0.0000  | 17.57   | 44.93   | 0.0006| 0.98 | 22.9      | 0.0190      | 0.0136       | 16.68        | 3.26        |
| 13   | 582.5 | 67.5  | 3.95  | 0.019 | 42      | 1.94 | 0.0011 | 15.62   | 38.06   | 0.0021| 4.76 | 23.08     | 0.0193      | 0.012        | 14.95        | 5.77        |
| 14   | 671.2 | 67.5  | 4.52  | 0.019 | 40.16   | 3.44 | 0.0017 | 15.53   | 35.99   | 0.0028| 7.19 | 26.15     | 0.0233      | 0.0119       | 16.96        | 5.29        |
| 15   | 755.5 | 67.5  | 5.04  | 0.019 | 36.88   | 5.73 | 0.0024 | 14.96   | 32.95   | 0.0029| 8.78 | 25        | 0.0217      | 0.0115       | 17.93        | 2.88        |
| 16   | 430.8 | 90    | 3.03  | 0.025 | 56.93   | 0.22 | 0.0003 | 19.79   | 48.26   | 0.0014| 1.8  | 21.76     | 0.0177      | 0.0156       | 23.36        | 6.28        |
| 17   | 483.6 | 90    | 3.33  | 0.025 | 49.9    | 0.3  | 0.0002 | 17.6    | 44.14   | 0.0015| 2.5  | 21.1      | 0.0170      | 0.0136       | 17.81        | 7.63        |
| 18   | 587.8 | 90    | 3.95  | 0.025 | 40.32   | 3.36 | 0.0017 | 15.56   | 36.87   | 0.0024| 5.88 | 22.47     | 0.0185      | 0.0119       | 13.93        | 4.52        |
| 19   | 675.4 | 90    | 4.52  | 0.025 | 39.75   | 4.09 | 0.0020 | 15.63   | 34.84   | 0.0027| 7.4  | 22.93     | 0.0191      | 0.012        | 20.36        | 4.47        |
| 20   | 756.8 | 90    | 5.04  | 0.025 | 36.82   | 5.98 | 0.0020 | 15.02   | 32.77   | 0.0031| 9.45 | 23.97     | 0.0203      | 0.0115       | 18.59        | 3.64        |
with correlation coefficient of determination $R = 94.3\%$. The humidification effectiveness is shown in Figure 11(b) where the data is scattered based on the water flow rates and on the ambient inlet condition of the outdoor specific humidity as defined by (2). Table 3 indicates that the evaporation cooling system effectiveness reaches a maximum of 29.45% when only four MiniModule membrane contactors are used. It should be noted that the current suggested method of research presents new interesting engineering concepts to enhance the performance of the gas turbine in spite of its low efficiency. Therefore, further researches need to be performed to improve the efficiency of the proposed technology, such as increasing the number of MiniModule membrane contactors, which leads to an increase of exchange surface between the membrane of the contactors and the air flow. Another advantage is the possibility of using recycled water (chemically treated water) since the fresh quality water in desert areas such as Saudi Arabia is mostly available through the desalination plants and commonly used for human beings supply. On the other hand, it is noticed that the evaporation process on the psychometric chart does not exactly follow the isenthalpic as mentioned earlier.

5. Conclusions

Using the modified MiniModule membrane contactors enhances the cooling process of the air. This enhancement is reflected by a maximum and minimum temperature drop of 10.77°C and 3.44°C, respectively. The results show that as the air velocity increases the drop in the air temperature decreases due to the differences in the inlet conditions. The relative humidity downstream of the test section is noticed to increase by a maximum of 4.65% which of course depends on the ambient condition as seen in Table 3. The results also show that the evaporation process does not follow the isenthalpic lines; therefore, heat transfers from the air as latent and sensible heats. The maximum evaporation cooling effectiveness is 29.45 which is considered low; however, more research need to be performed to improve the effectiveness of the proposed technology. The experimental data shows that the modified MiniModule membrane contactors could be implemented for gas turbine inlet air cooling without concerns of compressor blades erosion due to water droplets associated with water spray technology.

Conflict of Interests

The authors report no conflict of interests. The authors alone are responsible for the content and writing of the paper.
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