Chapter 5

Experimentation on DUAL-MODE Evaporative Cooler

5.1. Introduction

The dual mode evaporative cooler is a novel concept where this heat and mass exchanger can be utilized with little modifications. The climatic conditions varied drastically from April (hot and dry) to August (hot and humid) in composite climates. The direct evaporative coolers, which are very effective in dry months, are non-performing in humid outdoor conditions. Whereas the indirect regenerative evaporative coolers may effectively work in humid months. Hence the device change is needed to achieve effective space cooling in different climatic conditions at the same location. In this situation, a dual-mode (direct and indirect) cooling device is proposed to eliminate the need for two devices simultaneously. Which is a two in one device and work as DEC in dry ambient conditions, whereas work as REC in humid ambient conditions. This novel dual mode evaporative cooler is fabricated and experimented in this work. Novelties of the device are dual-mode operation and intermittent water supply. The device is tested by varying the operating parameters (air inlet temperature, air inlet velocity, air inlet specific humidity, water inlet temperature, water flow rate, and extraction ratio) and compared in direct and regenerative modes of operation. The performance parameters are plotted against the operating parameters to analyze the advantage of operating modes.

5.2. Experimentation methodology

The regenerative (indirect) and direct evaporative coolers are combined in one device with little modification of REC. The experimental setup is designed based on the configuration A of the previously discussed chapter. The main component of the indirect cooler is heat and mass exchanger. The dual-mode cooling system consists of a heat exchanger similar to regenerative coolers with the additional arrangement (vanes) to stop flow in dry channels during direct mode operations. The regenerative mode operation ensures entry of the outdoor air in dry channels and exhaust through wet channels. The additional axial fan is used for this purpose to extract a portion of dry air in the wet channels. The direct mode operation is achieved by stopping the axial fan and making the outdoor air to entry in wet channels of the heat exchanger. This additional arrangement is useful in operating one device in both modes.

5.2.1 Fabrication of dual-mode heat and mass exchanger

To maximise heat/mass exchange, the thermal and physical qualities of materials composing a heat/mass exchanger should match the certain requirements: The material should have high thermal conductivity and less thickness to avoid conduction resistance. In terms of mass transfer, wetting surfaces of materials should have adequate water absorption and retention capabilities. The best possible locally available materials are considered in this regard. The dual-mode heat exchanger is made by vertically stacking very thin aluminium plates. This vertical construction helps in the better wettability and easy flow in the excess water. The dry channel and wet channel are placed alternate in the acrylic cover. This casing material makes heat exchanger robust enough for experimentation. Fig. 5.1 shows the heat exchanger and its fabricated dry and wet channel in the lab. The alternate wet channels are formed by gluing pure cotton textile fiber on both sides of wet channel of the aluminium foil. The borders of the rectangular aluminium plate are glued by the foam material of thickness 5 mm to form it proper channels. The channel gap inside the channels is maintained by using spacers at different locations. The detailed dimensions of the testing unit heat exchanger are mentioned in Table 5.1.

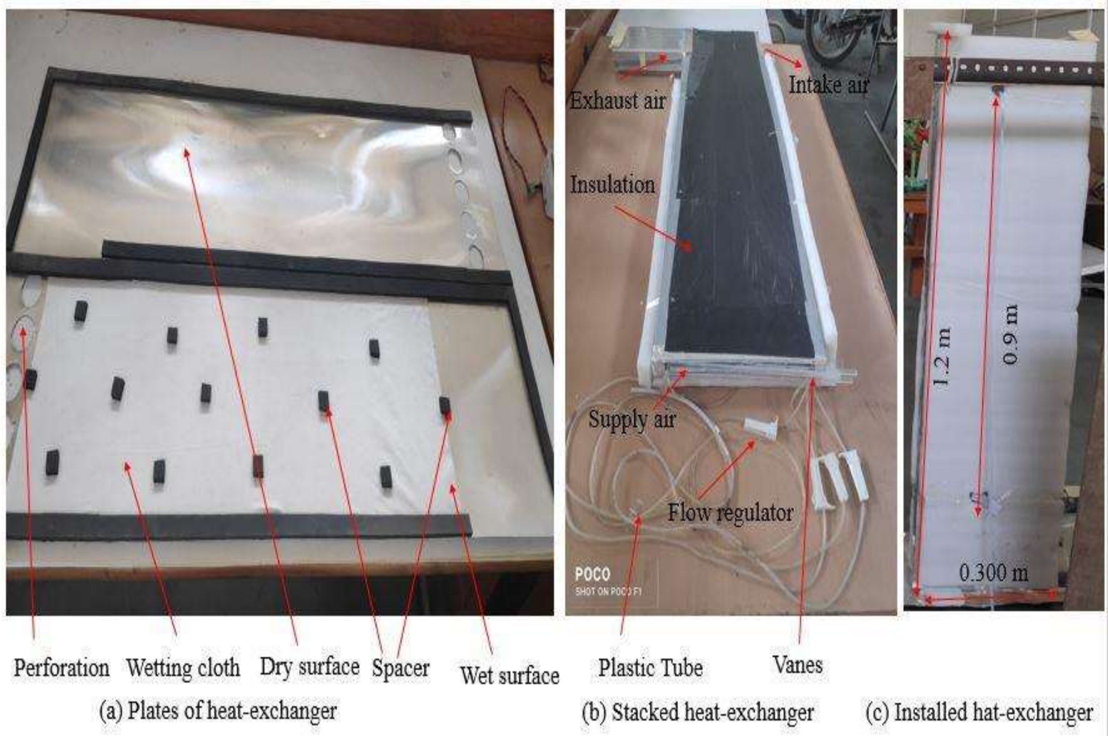

Fig. 5.1: Fabrication of heat and mass exchanger

Table 5.1: Specifications of dual-mode heat exchanger.

Effective length of both channels	0.9 m
Effective Width of channels	0.250 m
Channel gap between the dry channels	5 mm
Channel gap between the wet channels	5 mm
Thickness of the aluminum sheet	0.208 mm
Number of channels (dry+ wet)	4
Size of one hole at the top of the sheet	3.8 mm
Number of holes	5

The water distribution over the wet surface is obtained by using small diameter tubes at both sides of the wet channels. The water carrying tube is connected at the top of the channels in each wetting surfaces. The water flows natural gravity driven downward direction and collected at the bottom. The sliding thumb lock flow controller is used to regulate the water flow rates inside the tubes of the wet channels. The circular holes are made at the top of the plates to accomplish regeneration in the regenerative mode of the evaporative cooler. The air leak-proof duct tape is used to hold and press sheets together. The acrylic cover sheet is used at the front and back of the heat exchanger to provide its strength. The whole heat exchanger is covered with thick insulation material to prevent heat leaks.

5.2.2 Experimental test setup and instrumentation

The schematic diagram of the experimental test unit is shown in Fig. 5.2. The complete test unit consists of the tangential fan, air heater, humidifier, heat and mass exchanger, upper water tank, plastic tubing, lower water tank, and axial fan. The air heater and humidifier are used to obtain desirable inlet conditions at the heat exchanger. The prototype dual mode cooling testing unit was installed in a laboratory room with stable air temperature and humidity. The rotational speed of the fan is varied to achieve desirable flow rates of air. The air heating is achieved through the Variac controlled resistance air heater. The humidifier is installed in between the air heater and heat exchanger. The humidity addition in the air is also controlled by the voltage governed Variac. The variable speed axial fan is installed to control the supply to exhaust air flow ratio. The axial fans are installed in the proper extension ducts, which is fabricated to adjust the dimensions of the heat exchanger and fan diameter. The complete experimental setup with measuring instruments is shown in Fig. 5.3. The list of instruments used to measure the different parameters at different locations is given in Table 5.2. The K-type thermocouple is used at the inlet, outlet (supply), and exhaust position of the test unit. The data logger is connected to record the temperature simultaneously at all three-point. The capacitive type humidity probe is used at the inlet, supply, and exhaust position to measure the relative humidity of the air. The averaged air velocity is measured at a rectangular cross-section (at inlet and exhaust) with the help of a hot wire anemometer. The pressure loss in the dry and wet channels is measured by connecting the differential pressure transmitter to the attached tapped tubes of the channels. The digital wattmeter is used to record the pump power consumption.

Table 5.2: List of instruments used to measure the different parameters

Parameters	Instruments	Measurement range	Accuracy	Resolution
Temperature	K type thermocouple	-20 to 70 ºC	±0.5 ºC	0.1 ºC
Relative humidity	Humidity probe-capacitive type	0 to 100 % RH	±2 % RH	0.1 % RH
Velocity	Hot-wire anemometer	0 to 30 m/s	± 0.03 m/s + 4 % of mean value	0.01 m/s
Pressure drop	Differential pressure transmitter	-200 to 200 Pa	± 0.5 % ± 2 Pa	1 Pa

The data logger used in the experimentation is the NI 9214 is a high-density thermocouple module. The specification of the data logger is listed in Table 5.3. The instruments are purchased with calibration as per factory calibrated protocol. The company Testo (for anemometer and humidity probe) provided calibration certificates (certificates of conformity) along with measured values and validation period. The humidity probe is calibrated at the reference value of 44% RH with the permissible tolerance of ± 2 % RH. The Actual value obtained was 44.3 % RH. The anemometer is calibrated at four different velocities (1 m/s, 3 m/s, 6 m/s, 25 m/s). At the one m/s of reference value with the tolerance limit of ± 0.07 m/s, the actual value obtained was 0.98 m/s. Differential pressure transmitter is calibrated through PC using serial converter by company (Redix) norms

Table 5.3: The specification of the data logger

Number of channels	16 thermocouple channels, 1 internal auto zero channel
ADC resolution	24 bits
Timing Mode	High-resolution - Conversion Time (Per Channel) 52 ms , Sample Rate (All Channels) 0.96 S/s High-speed-Conversion Time (Per Channel) 735 µs, Sample Rate (All Channels) 68 S/s
Thermocouple signal input bandwidth	High-resolution mode 14.4 Hz High-speed mode 80 Hz
Temperature Measurement Accuracy (For K type thermocouple)	High-resolution mode-0.01 °C High-speed mode-0.10 °C
Power Requirements (Active mode)	Power consumption from chassis -300 mW maximum Thermal dissipation (at 70 °C)- 630 mW maximum

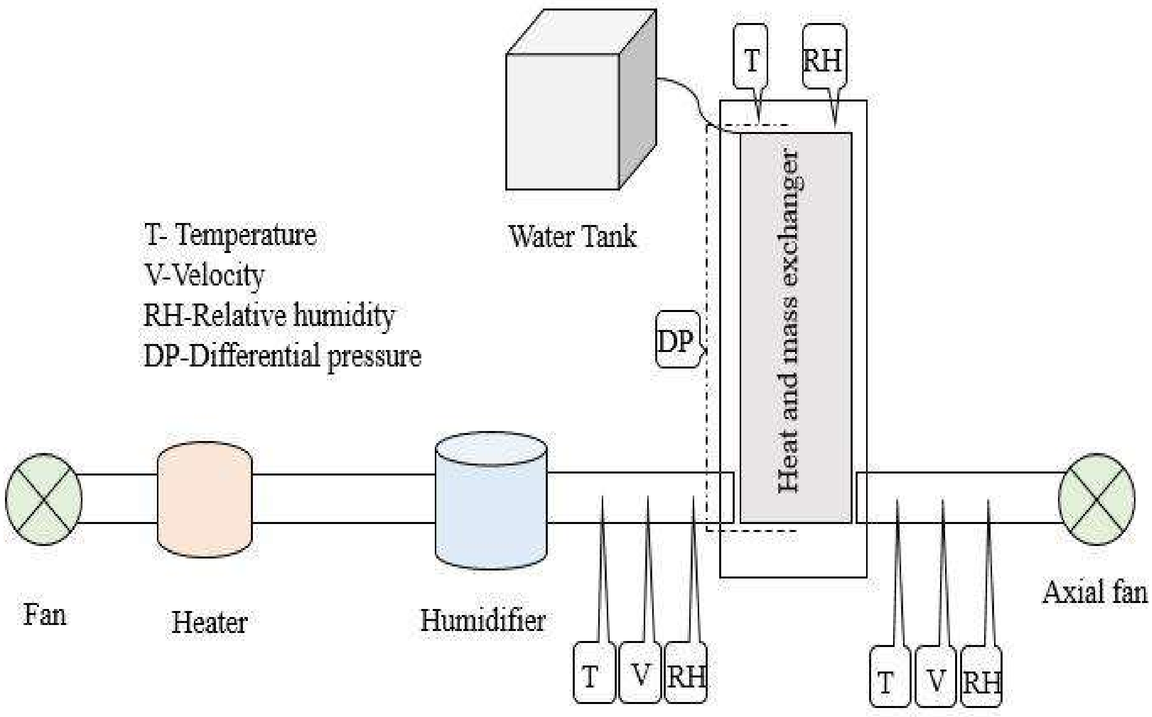

Fig. 5.2: Layout of the experimental setup for dual mode device

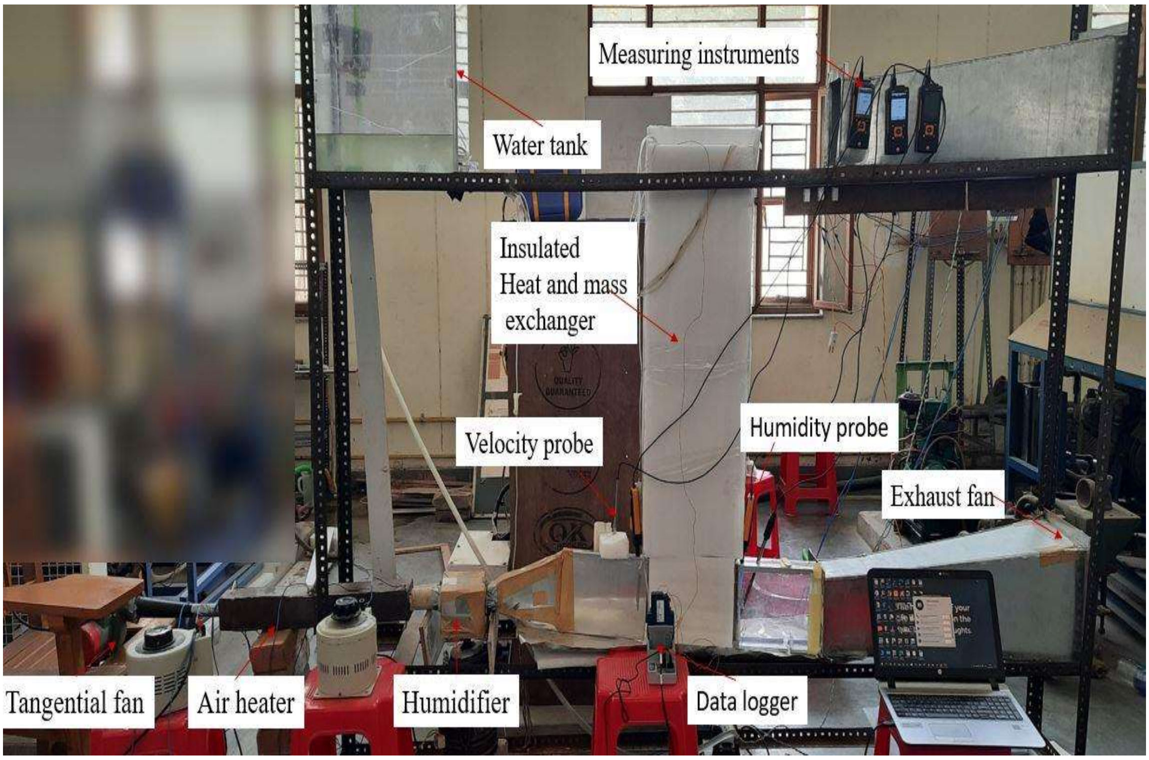

Fig. 5.3: Photograph of the experimental setup for dual mode device

5.2.3 Experimental procedure

The bottom of the vertical mounted dual mode heat exchanger is connected with the pre-conditioning unit. The Variac is used to control the heating temperature and moisture addition rate of the blowing air. The vanes at the top of the heat exchanger are handled manually to block air flow in the dry and wet channels (as per the mode). In the regenerative testing mode, the inlet opening of HMX is connected to pre-conditioning unit while the exhaust opening to the exhaust fan. Wet channels are blocked at the top by vanes, and air enters in the wet channel only through circular holes. The wet channels at the bottom is connected to exhaust fan. The rotational speed of the exhaust fan is varied to adjust the extracted air in the wet channel. A regulator (for speed variation) is connect to get this RPM (round per minute) variation. In the direct testing mode, HMX is rotated and exhaust opening is now connected with the pre-conditioning unit. The exhaust fan is now disconnected, and dry channels are blocked by vanes. So the supply air is obtained from the top of the HMX, in both REC and DEC modes of testing.

Preparation is part of the testing procedure which includes following points: (1) The temperature and relative humidity of the air were established and recorded in the environmental outdoor conditions once it was turned on. (2) The data recorder and computer were turned on, and all of the experimental devices were checked for accuracy and correct reading. (3) The water tanks of the testing unit is filled with the water by external pump. (4) The fan was started to circulate air from the environment to heat and mass exchanger through the pre-conditioning unit until the system was stable. (5) By managing the humidifier and heater linked to the intake duct, the system's inlet air temperature and humidity were adjusted to predicted inlet conditions. (6) It was necessary to run the cooling device for half an hours before testing to ensure that the heat exchanger's wet surfaces were thoroughly wetted. (7) The data logger started recording the data when the desired inlet air conditions were obtained and stabilised for at least 15 minutes.

The velocity is measured at the rectangular cross-section by the trivial method of grid measurement. The cross-section is divided into four equal-sized areas, and measurement was done at the center of each grid. The arithmetic averaging is done to obtain the mean value at the particular cross-section. The variation of the relative humidity along the cross-section is negligible; hence the center of the duct value is assumed to be averaged relative humidity value. The NI data logger is used to get the time-averaged value of the temperature every 5 seconds to increase the reliability of the experiment. The experimentation is done in the static conditions to obtain a steady state at the specific inlet conditions. The extraction ratio is varied in the range of 0.31-0.7, inlet temperature in the range of 46.4-28.3 ºC, inlet velocity in the range of 0.5-2.5 m/s, water temperature 12.6-20 ºC water flow rate 11.2-19.8 lph.

5.2.4 Evaluation of experimental performance parameters

The recorded temperature, pressure, and specific humidity at different locations are used in the calculation of performance parameters. The air goes to the conditioning space from the outlet of the device. The performance of the cooling device is determined by the state of the air at the outlet of the device.

The extraction ratio (ER) is defined as the portion of dry channel air (primary air) is redirected into the wet channels (as secondary air) and expressed as:

$$ER = \frac{\dot{m}_{sa}}{\dot{m}_{pa}} \tag{5.1}$$

As per the ASHRAE Standard 143 (ASHRAE 2000), when the indirect evaporative systems use outside air as the intake air of systems, the cooling capacity of coolers can be evaluated using the sensible cooling of intake air. The cooling capacity has been evaluated by, for both modes,

$$Q_{system} = \dot{m}_{out} c_{p,out} (T_{in} - T_{out}) \tag{5.2}$$

The dew-point effectiveness (DPE), has been evaluated by, for both modes,

$$\varepsilon = \frac{T_{in} - T_{out}}{T_{in} - T_{dew-point}} \tag{5.3}$$

The mass flow rate of air is calculated by measuring the average velocity at the desired cross-section. The coefficient of performance (COP). It is expressed as, for both modes,

$$\mathrm{COP} = \frac{\mathrm{Q_{system}}}{\mathrm{P_{fan}} + \mathrm{P_{pump}}} \tag{5.4}$$

The fan is driven by electrical power and controlled by an attached variac to regulate the rpm of the fan. The same inlet tangential fan is connected through pre-conditioning equipment and a heat exchanger. So the best way to calculate fan power is through pressure drop measurement. The pressure drop between the channels is measured through taps connected at the end of the channels. The total power consumption includes the fan power and pumps power. The fan power is calculated as:

$$\mathrm{P_{fan}} = \frac{\Delta \mathrm{P_{pa}} \mathrm{V_{pa}} + \Delta \mathrm{P_{sa}} \mathrm{V_{sa}}}{\eta_{\mathrm{fan}}} \tag{5.5}$$

The volume flow rate in the respective channel is calculated with the velocity measurement. The pump power is obtained by wattmeter reading directly.

For DEC and REC modes, the irreversibility has been calculated by, respectively,

$$m_{sa}\psi_{sa-in} + m_w\psi_w - m_{sa}\psi_{sa-out} - \dot{I} = 0 \tag{5.6}$$

$$m_{pa}\psi_{pa-in} + m_w\psi_w - m_{pa-out}\psi_{pa-out} - m_{pa}\psi_{pa-out} - \dot{I} = 0 \tag{5.7}$$

Ψ has been evaluated by correlation presented in equation 3.19. (Chapter 3)

Where $m_w = m_{sa}(\omega_{sa-out} - \omega_{sa-in})$ and $\psi_w = -R_v T_0 \ln \varphi_0$

Where φ_0 is the relative humidity at the dead state, in this study, the atmospheric saturation state is assumed as the dead state.

The exergy efficiency for the DEC mode of the cooler is defined as,

$$\eta_{ex} = \frac{m_{sa}\psi_{sa-out}}{m_{sa}\psi_{sa-in} + m_w \psi_w} \tag{5.8}$$

The exergy efficiency for the REC mode is given by,

$$\eta_{ex} = \frac{m_{pa-out}\psi_{pa-out}}{m_{pa}\psi_{pa-in} + m_w \psi_w} \tag{5.9}$$

The sustainability index for both modes has been calculated by,

$$\text{Sustainability index} = \frac{1}{1 - \Phi} \tag{5.10}$$

Where has been evaluated by correlation presented in equation 3.34(chapter 3)

The cooler's initial cost and running cost are estimated for economic analysis of the dual-mode evaporative cooler. The evaporative cooler contains a heat exchanger, fan, and pump as main components. The capital (initial) cost is calculated as:

$$C_{Capitalt} = C_{fan} + C_{pump} + C_{HMX} + C_{frame} + C_{Upper\ tank} + C_{Lower\ tank} + C_{piping} + C_{labour} \tag{5.11}$$

The cost of each individual component is given in Table 5.4. The total cost is given by:

$$C_{Total} = C_{Capital} + C_{Operating} \tag{5.12}$$

The capital and labor cost are acquired in the Indian rupee, but it is converted in USD by the current conversion rate. The USD is a globally accepted currency that helps readers of all across the globe to understand the economic analysis of the cooling device. The running cost of each individual mode is calculated by the energy (electricity) needed to run the pump and fan. The electricity consumption charge is taken as a global average value of 0.139 USD/kWh. The operating cost of the device is calculated for ten working hours in a day over the period of one year. The specific total cost (STC) is defined as the total cost per unit cooling rate of the cooling device, and hence,

$$STC = \frac{C_{Total}}{Q_{system}} \tag{5.13}$$

Table 5.4: Cost of the different component

Components	unit	USD
Tangential fan	1	79.15
Axial fan	1	16.38
Water pump	1	70.96
Heat and mass exchanger	1	95.53
Acrylic casing	1	10.92
Upper tank	1	20.47
Lower tank	1	9.55
Water piping	1	2.05
Frame		34.12
labour		68.23

5.2.5 Uncertainty analysis

The uncertainty calculation involves the estimation of error indirectly measured or calculated parameters of the experiment. The uncertainty in this test is calculated as explained by Holman (2001). Let, the result (Z) is a function of the independent variables p_1, p_2, p_3,……,p_n. Thus,

$$Z = Z\left(p_1^{i_1}, p_2^{i_2}, p_3^{i_3}, \ldots\ldots, p_n^{i_n}\right) \tag{5.14}$$

Let U_Z be the uncertainty in the result and i_1, i_2, i_3,……,i_n being the uncertainties in the independent variables, then the uncertainties in the result are given by,

$$U_Z = \left[\left(\frac{\partial Z}{\partial P_1} i_1\right)^2 + \left(\frac{\partial Z}{\partial P_2} i_2\right)^2 + \ldots\ldots\ldots\ldots + \left(\frac{\partial Z}{\partial P_n} i_n\right)^2\right]^{\frac{1}{2}} \tag{5.15}$$

The maximum uncertainties of the measured and calculated parameters are listed in Table 5.5. The highest uncertainty is obtained in the COP calculation.

Table 5.5: The maximum uncertainty of the measured and calculated parameters.

Temperature (ºC)	±1.00%
Relative humidity	±1.92%
Air velocity (m/s)	±3.00%
Pressure difference (Pa)	±1.00%
Dew point effectiveness	±2.61%
Cooling capacity (W)	±3.97%
Coefficient of performance	±4.09%

5.3. Results and discussion

The dual-mode evaporative coolers testing unit is tested for both regenerative and direct modes. The device operated in the regenerative mode with the additional axial fan located at the end of the wet channels. The Variac is used to control the rpm of both the inlet tangential fan and exhaust axial fan. The extraction ratio is controlled by the regulating rpm of the exhaust axial fan, connected at the end of the wet channels. The Variac of 0-270 V output is connected with the fan and used to vary the flow rate. The direct mode is tested by connecting the pre-conditioning unit (tangential fan + air heater

+ humidifier) to the wet channels and restricting the airflow only in wet channels. The reference operating conditions for the dual-mode heat exchanger is inlet air temperature is 45 ºC, and inlet specific humidity is 10 g/kg, inlet velocity is 1.2 m/s, water temperature is 20 ºC and water flow rate is 11.6 lph and extraction ratio 0.5. Table 5.6 compares the results of both modes at the reference state. The comparative result of data of flow, heat, and moisture content in each part of the device is mentioned in the table. The important parameters of the dual-mode evaporative are varied to investigate the effect on the performance of evaporative coolers. The results of the direct mode and regenerative mode are compared by varying one parameter while the rest of the parameters are made to be on reference state.

Table 5.6: Data of flow and humidity at a different point in the device at the reference state

Mode	T_{supply} (ºC)	DPE	Cooling capacity (W)	V_{inlet} (m^3 /min)	V_{supply} (m^3 /min)	$T_{exhaust}$ (ºC)	$V_{exhaust}$ (m^3 /min)	Exhaust air humidity (kg/kg)	Supply air humidity (kg/kg)	COP
REC	24.8	0.65	34.23	0.18	0.09	26.5	0.09	0.02088	0.010	52.9
DEC	28.1	0.54	57.35	0.18	0.18	-	-	-	0.019	89.4

5.3.1 Effect of extraction ratio

The increases in the extraction ratio reduce the supply air temperature and cooling capacity, as shown in Fig. 5.4. The increased mass flow of air in wet channels works as increased heat sink capacity to cool dry air. The disadvantage of the increased extraction ratio is to reduce the supply air mass flow rate, which reduces the cooling capacity drastically. The cooling capacity of the regenerative device was reduced from 45.7 to 21.9

W, with an increase in the extraction ratio from 0.31 to 0.7. Fig. 5.5 shows the dew point effectiveness and COP variation with the extraction ratio. The effectiveness improves with the increase in the extraction ratio. The reduction in supply air mass with an increased extraction ratio is the more dominating factor in reducing the cooling capacity of the regenerative mode of the cooler. The reduced cooling capacity and increased pressure loss in the wet channel reduce the COP value with an increase in extraction ratio. The COP diminishes by 52% with an increase in the extraction ratio of 0.31 to 0.7, while dew point effectiveness improves by only 10%. Hence high extraction ratio is not recommended, but its value should be such that supplied cool air lies in the thermal comfort for the occupants with maximum cooling capacity. The specific humidity of the supply remains the same as inlet-specific humidity conditions.

Fig. 5.6 and Fig. 5.7 shows the variation of exergy destruction, sustainability index, exergy efficiency and specific cost with the fraction of the secondary air volume flow rate. The exergy destruction of REC mode of the developed device rises with the higher extraction ratio. The higher secondary to primary air mass (extracted air) leads to the high mass available to directly contact the wet surface and more destruction of the exergy. The highest exergy destruction of 2.5 W is achieved at the extraction ratio of 0.7. The sustainability of the REC decreases with the increases in the extraction ratio. The increased extraction ratio improves wet-bulb effectiveness, but cooling potential diminishes due to reduced outlet air (supply air) flow to cooling space. The exergy ratio decreases with the higher portion of the extracted air in the device. The increased exhaust air flow decreases the exergy ratio. The specific cost of the evaporative cooler hiked with the increase of the extracted air. The cooling capacity decreases very sharply with increased secondary air mass, which increases the specific total cost of the cooling device.

The exergy ratio decreases by 9.6%, and the specific total cost increases by two times with the increase of the extraction ratio from 0.31 to 0.7.

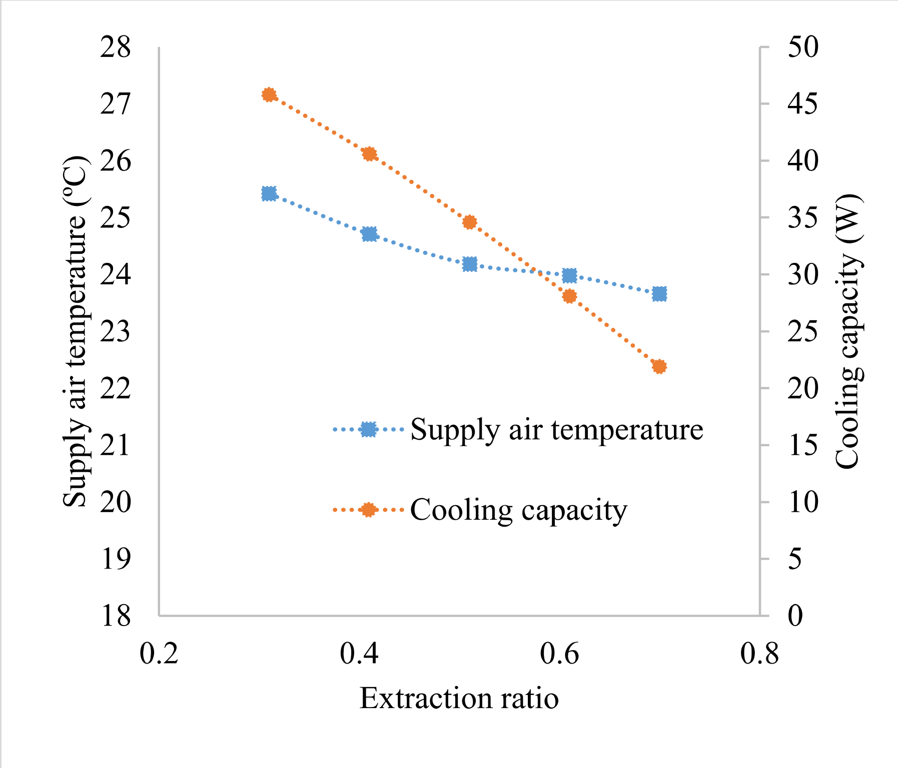

Fig. 5.4: Supply air temperature and cooling capacity variation with an extraction ratio

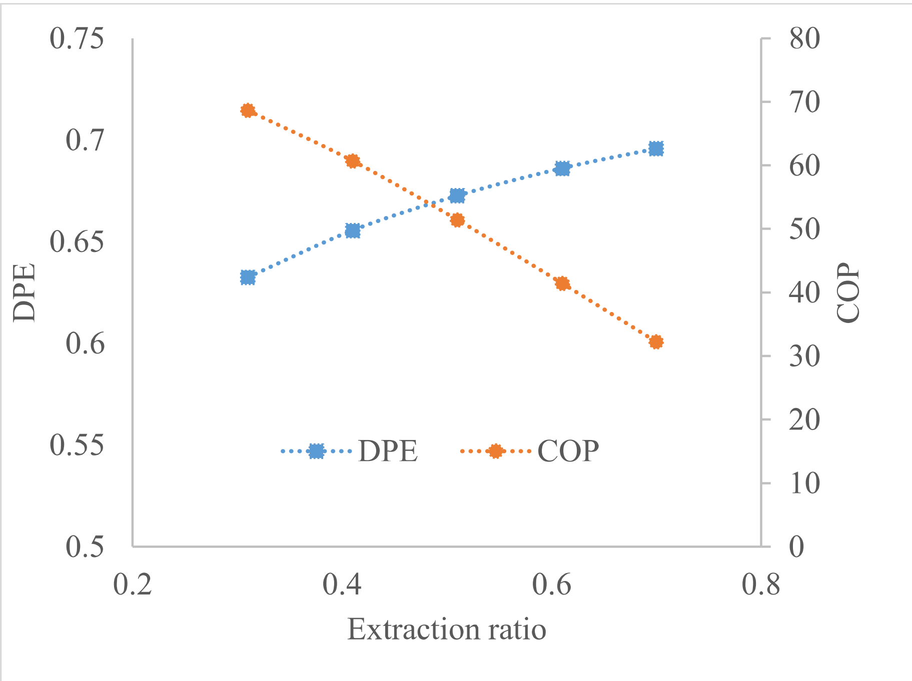

Fig. 5.5: Dew point effectiveness and COP variation with extraction ratio

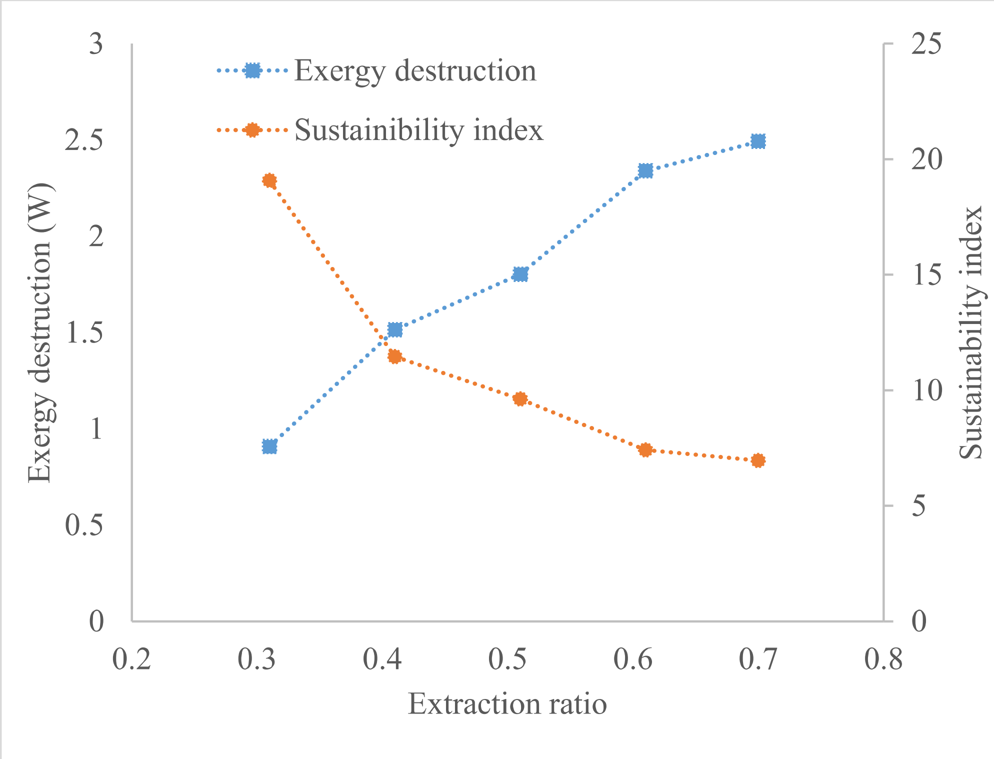

Fig. 5.6: Variation of exergy destruction and sustainability index with extraction ratio

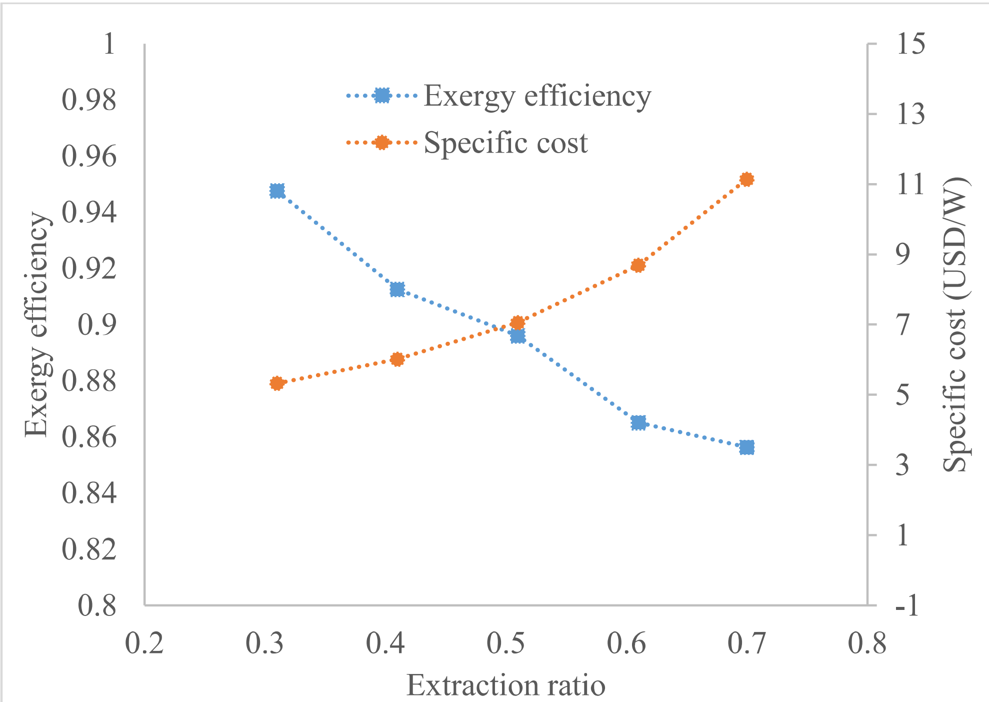

Fig. 5.7: Variation of exergy efficiency and specific cost with extraction ratio

5.3.2 Effect of inlet air velocity

The supply air temperature increases with the increases in the inlet velocity, as shown in Fig. 5.8. The increased velocity inside the channels reduces the time duration for heat and mass exchange between the airs, which results in higher supply air temperature of the evaporative cooler in both modes. Higher temperature is obtained for the direct mode as compared to the regenerative mode. The supply air temperature in regenerative mode increases from 20.9 to 27.7 ºC with the increase in velocity from 0.5 to 2.5 m/s. The cooling capacity of the direct mode is 74.0% higher than the regenerative mode at the lowest inlet velocity (0.5 m/s). The cooling capacity also increases sharply as more mass of air is supplied to the conditioned space. The cooling capacity increase is sharper in direct mode after 1 m/s inlet velocity. The cooling capacity here is also dominated by the increased mass of the supply air over the reduced temperature drop. The dew point effectiveness of both modes decreases while cop increases with the increases in the inlet velocity, as shown in Fig. 5.9. The reduced temperature drop decreases the effectiveness of the device in both direct and regenerative modes. The COP is enhanced by the increased cooling capacity, and it is dominated over the increased pressure loss due to higher mass flow rates inside the same channels. The effectiveness of the cooler, decreases by 28.2 % and 29.7 % in regenerative and direct mode, respectively, with an increase in velocity from 0.5 to 2.5 m/s. The highest cop of the regenerative mode is 78.28, which is obtained at the highest operating inlet velocity. Both modes of operation show COP increase by approximately 1.8 times with an increase of the velocity 0.5 to 2.5 m/s. The device should be operated at maximum inlet velocity to get comfortable cooling air.

The higher intake velocity decreases the exergy destruction of the evaporative cooler in REC and DEC modes. DEC mode exergy destruction is greater than the REC mode, as shown in Fig. 5.10. The high velocity makes the heat and mass exchange process faster, which increases the irreversibility of the cooling process. The exergy destruction of direct mode is sharper at high intake velocity. The sustainability index of both modes declines with the increment of the intake air mass flow in the device. The exergetic performance of the regenerative mode is better than the direct mode, which makes REC a high sustainable mode. The exergy ratio of the duel-mode evaporative cooler decreases with the rise in the inlet air velocity, as shown in Fig. 5.11. The increased velocity decreases wet-bulb effectiveness but increases cooling capacity. The increased exergy destruction due to the increased mass of air in the system decreases the exergy ratio. The increased cooling capacity of the system in both modes reduces the specific cost of the device drastically at a higher velocity. The specific cost of the direct mode is reduced from 8 to 2 USD/W when inlet velocity is increased from 0.5 to 2.5 m/s.

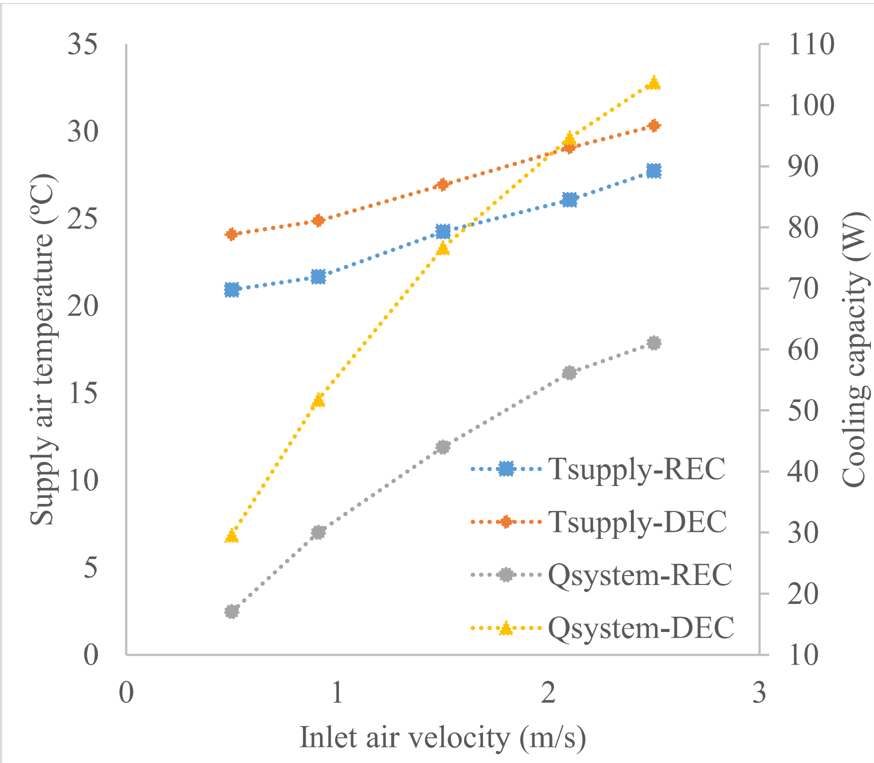

Fig. 5.8: Supply air temperature and cooling capacity variation with inlet velocity

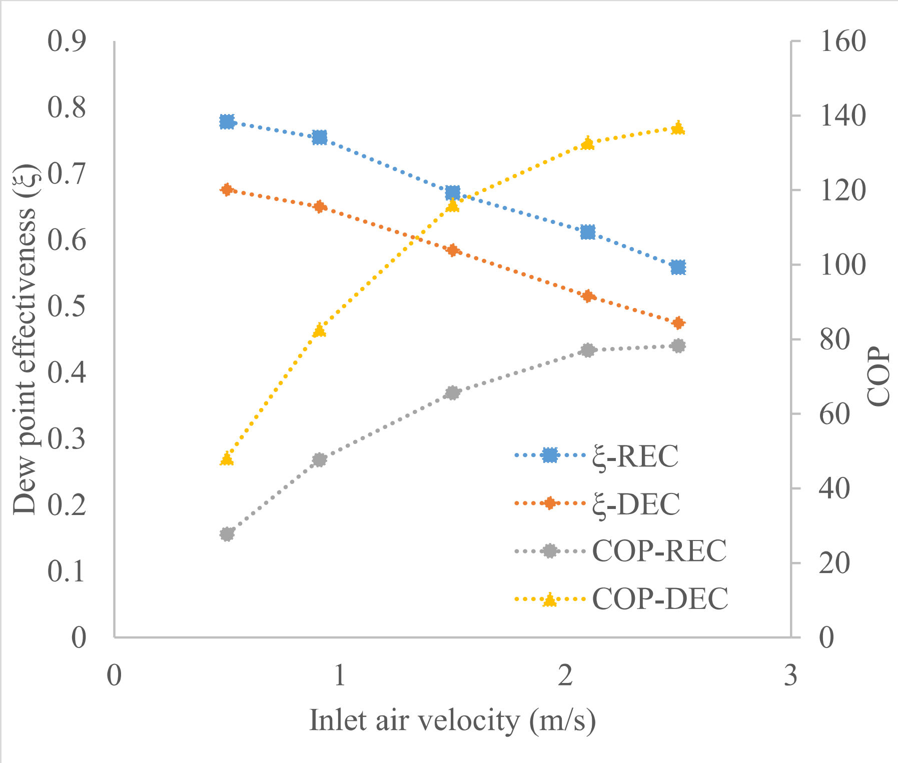

Fig. 5.9: Dew point effectiveness and COP variation with inlet velocity

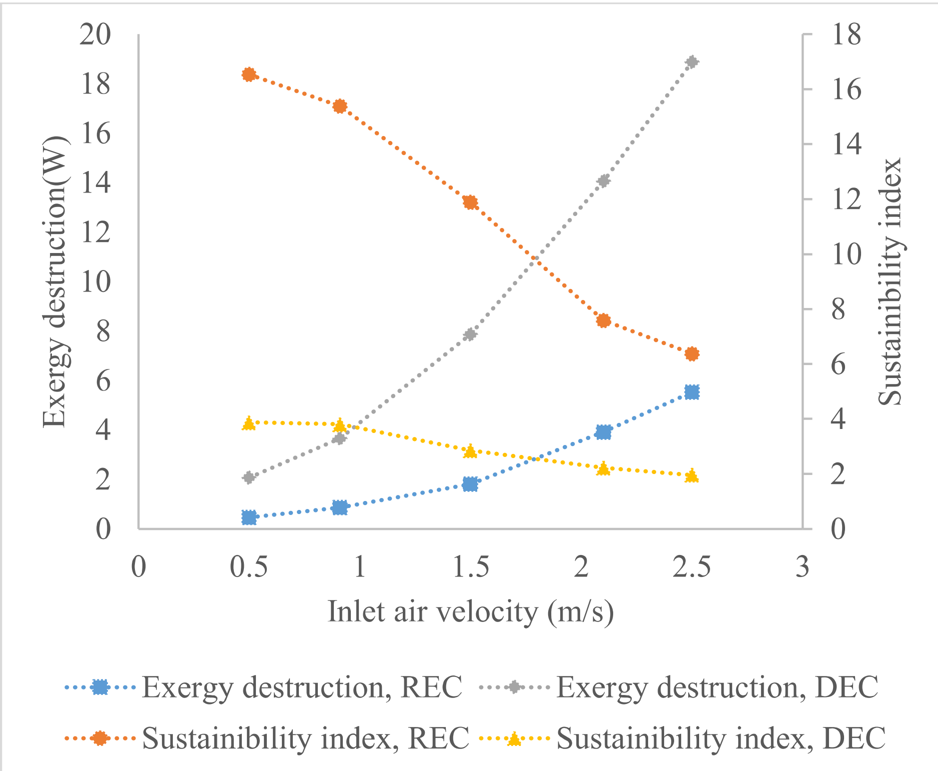

Fig. 5.10: Exergy destruction and sustainability index variation with air inlet velocity

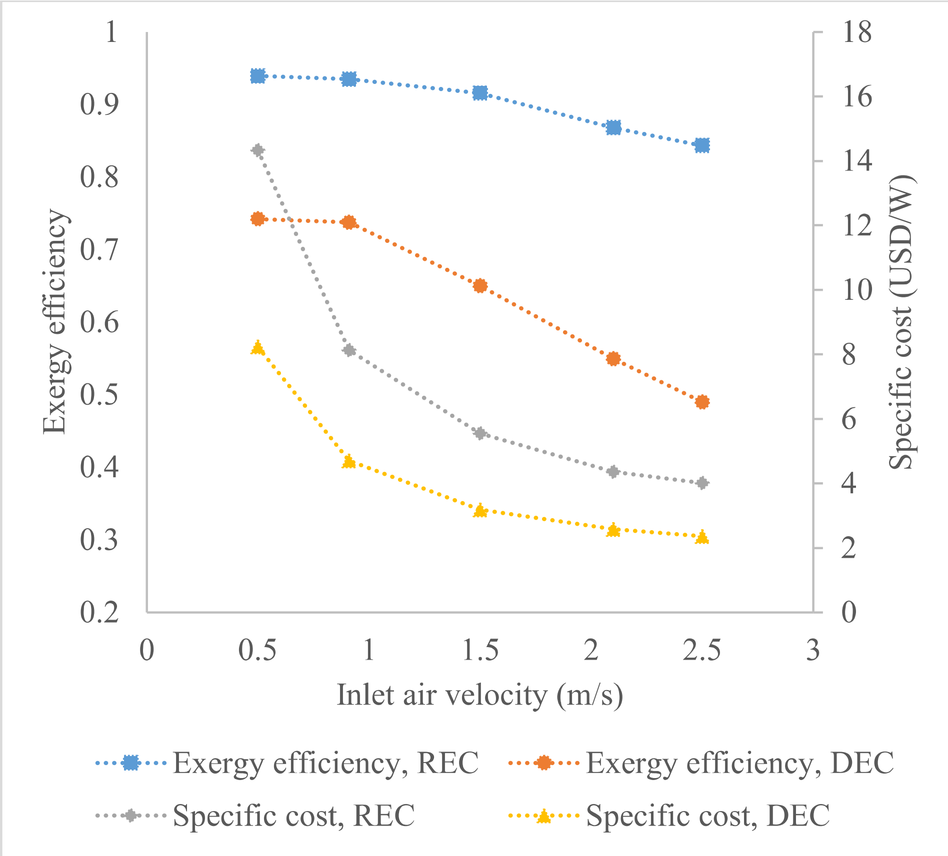

Fig. 5.11: Exergy efficiency and specific cost variation with air inlet velocity

5.3.3 Influence of air inlet conditions

The supply air temperature decreases with the decrease in the inlet temperature at constant humidity conditions. Fig.5.12 compares the result of direct and regenerative modes at three different humidity conditions. The difference between direct and regenerative modes is significant at high inlet conditions. The increase in inlet humidity increase's the supply air temperature at the same inlet temperature. The increased inlet humidity of the intake air reduces the potential of more moisture addition in the air hence reducing the cooling potential of the cooler in direct and regenerative modes. The direct mode temperature varied from 33.2 to 24.2 ºC while the regenerative mode temperature was 32.7 to 24 ºC. The direct mode and regenerative mode operation provide almost the same temperature at low inlet temperature conditions for all three tested humidity conditions. The indirect mode supplies air without humidity addition, while the direct mode relative humidity is above 90%. The high inlet temperature at the same specific humidity provides high wet-bulb depression, which increases the scope of cooling. The cooling capacity increases with the increase of the inlet temperature, as shown in Fig.5.13.The cooling capacity of the direct mode evaporative cooler is significantly higher than the regenerative mode evaporative cooler. This difference in the cooling capacity of the direct and regenerative mode increases with increases in wet-bulb depression. The more mass of supply-air in the direct mode dominates the cooling capacity of the device. The low specific humidity of air has a high potential to get cooled and so high cooling capacity too. The highest cooling capacity of 60 W is found for direct mode at the highest dew point depression and driest intake conditions. The lowest cooling capacity of the device is obtained for the regenerative mode at the lowest dew point depression and at 0.024 kg/kg of intake humidity conditions. The dew point effectiveness of both modes of the evaporative cooling device increases with the increases of the inlet temperature (inlet dew point depression), as shown in Fig. 5.14. The effectiveness of the device increases with the increase in the inlet-specific humidity of the air. The highest dew point effectiveness of 74 % is obtained for the regenerative mode at the highest investigated humidity conditions and inlet temperature. The dew point effectiveness of the regenerative mode decreased by 25%, while for direct mode, it decreased by 23% with the decreases in the inlet dew point depression at 0.015 kg/kg of humidity conditions. The COP of the device is high at the low specific humidity conditions in both direct and regenerative modes, as shown in Fig.5.15. The high cooling capacity at the same power consumption of the device increases the COP of the cooler at the drier inlet conditions. The highest cop of the cooler in regenerative mode is 48.2 while in direct mode 92.3 at the specific humidity of 15g/kg of dry air. The COP of the cooler is decreasing with the decrease in the inlet temperature at the same humidity conditions. COP of both modes decreases approximately 75% with the decrease in inlet dew point depression at the 0.015 kg/kg of intake humidity conditions.

Fig. 5.16 shows that the rise in the exergy destruction in DEC mode as well as REC mode at the higher intake temperature while specific humidity remains constant. The exergy destruction of the regenerative mode operation is lower than the DEC mode. The higher intake humidity at the constant inlet temperature conditions decreases the exergy destruction of the evaporative cooling device. The exergy destruction of the direct mode is sharper at a higher temperature, while for regenerative mode, it is almost linear. The highest exergy destruction of the 4.1 watts is obtained for DEC at the investigated driest air conditions. The sustainability index variation with the intake temperature of the cooling device is presented in Fig. 5.17.The higher ambient dry-bulb temperature and lower ambient specific humidity make the device more sustainable. The sustainability of the regenerative mode operation is higher than the direct mode of the cooler. The evaporative coolers are most effective in dry conditions; hence low inlet-specific humidity increases sustainability. The highest sustainability index of 8.8 is obtained for 46.4 ºC at the driest investigated conditions. Fig. 5.18 presents that the exergy ratio of the cooler is increased with the increment in the intake temperature. The dry inlet conditions help to improve both energetic as well as exergetic performance of the device. The REC mode shows a 20 % higher exergy ratio as compared to the DEC mode at the intake specific humidity of 0.015 kg/kg. The exergy ratio improves by 13 % at 28.3 ºC when inlet-specific humidity is changed from 0.015 to 0.019 kg/kg. The humid air shows a higher exergy ratio in direct mode at high inlet temperature conditions. The specific cost of the device decreases with the rise of the intake temperature conditions (Fig. 5.19). The specific cost of the DEC is less than the REC due to the higher cooling capacity. The lowest specific cost of 4 USD/W is obtained for the DEC mode at the intake specific humidity of 0.015 kg/kg.

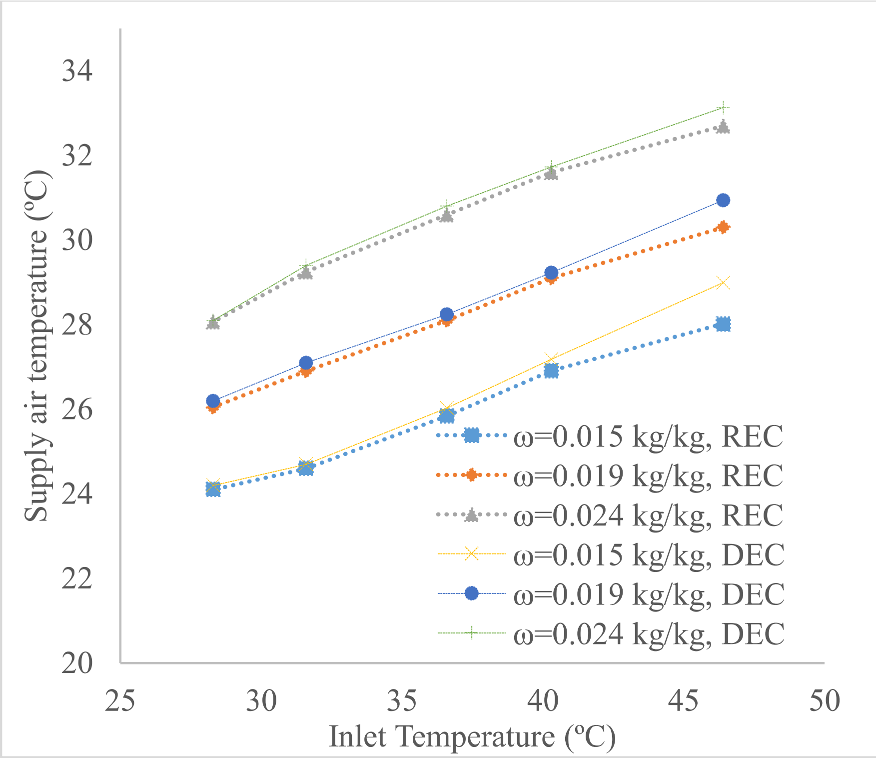

Fig. 5.12: Supply air temperature variation with inlet velocity

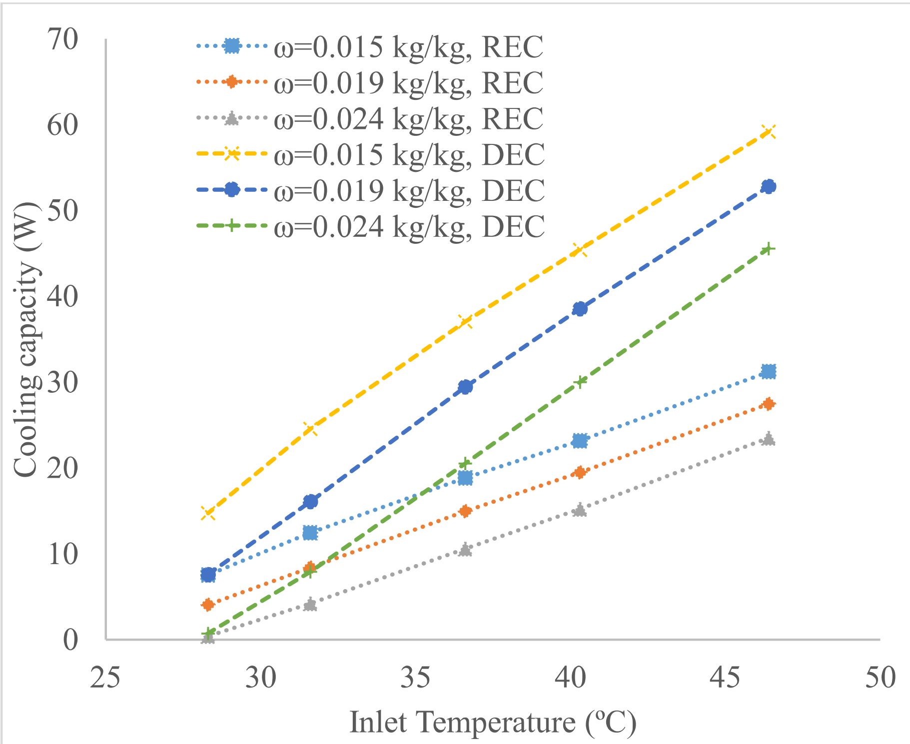

Fig. 5.13: Cooling capacity variation with air inlet temperature

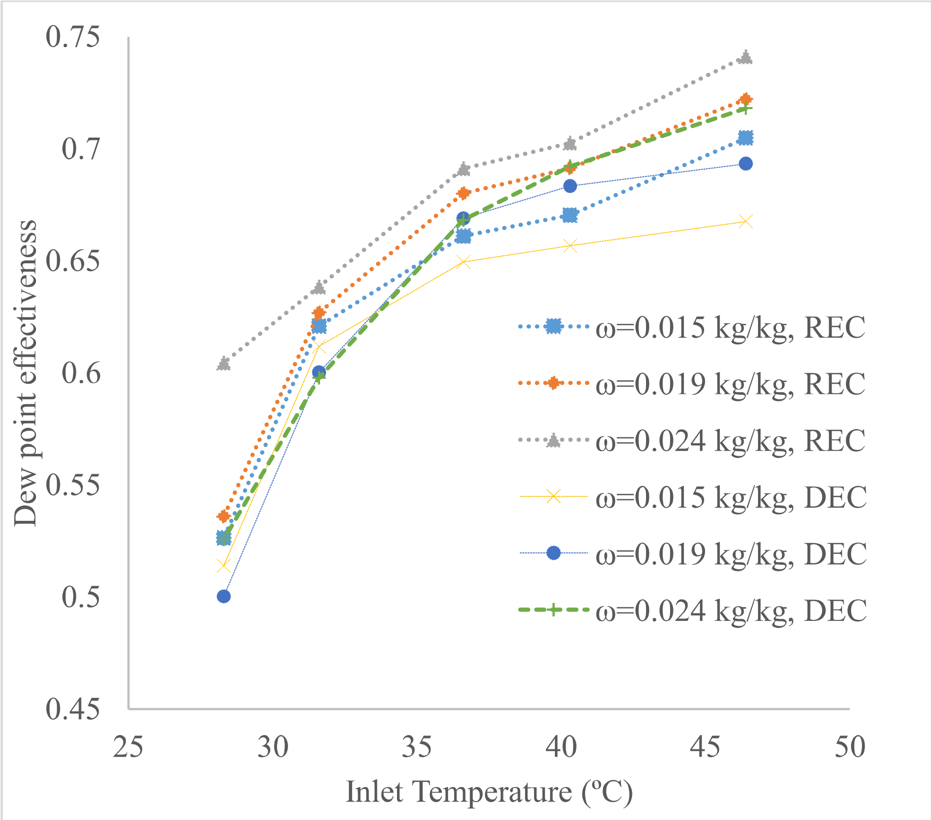

Fig. 5.14: Dew point effectiveness variation with inlet temperature

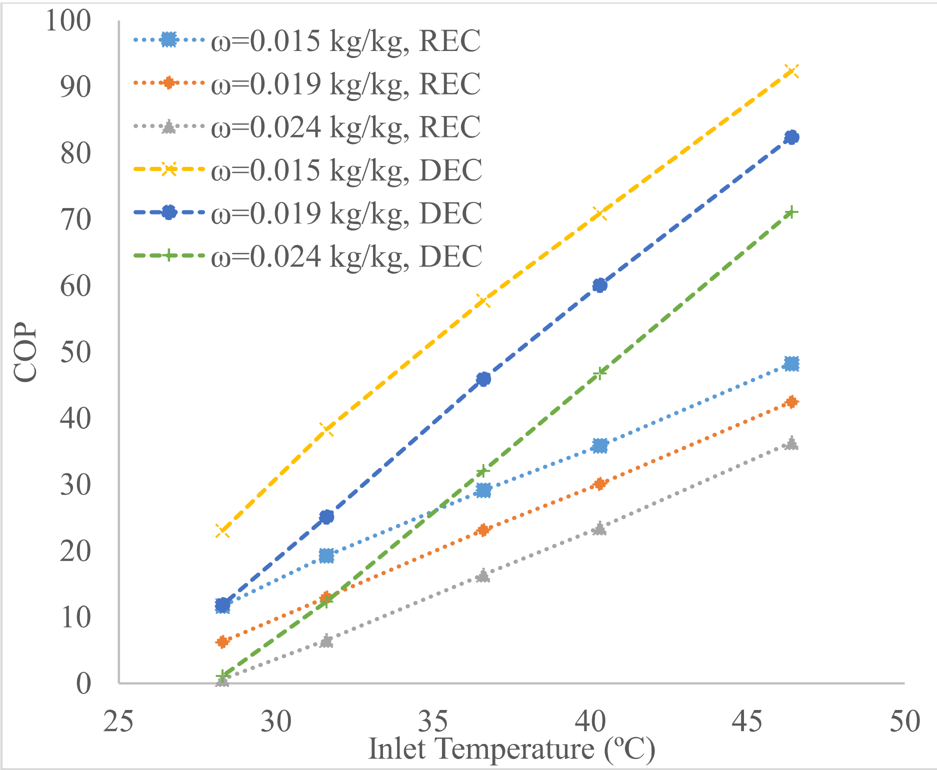

Fig. 5.15: COP variation with the inlet temperature

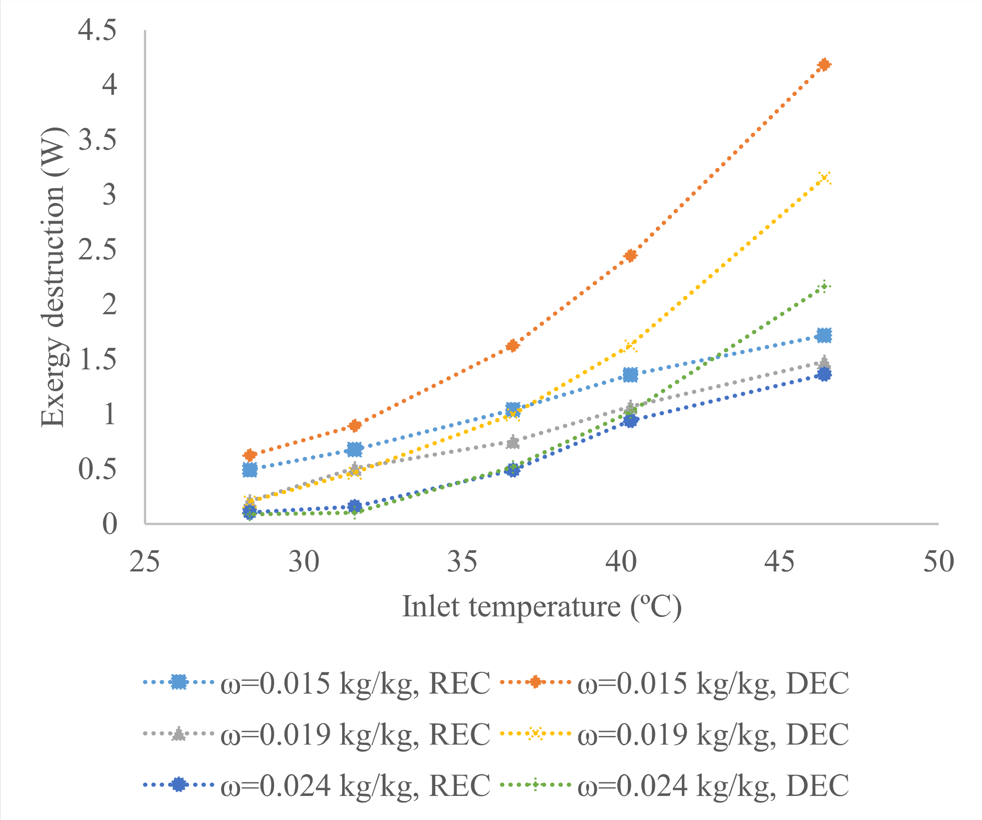

Fig. 5.16: Exergy destruction variation with air inlet temperature

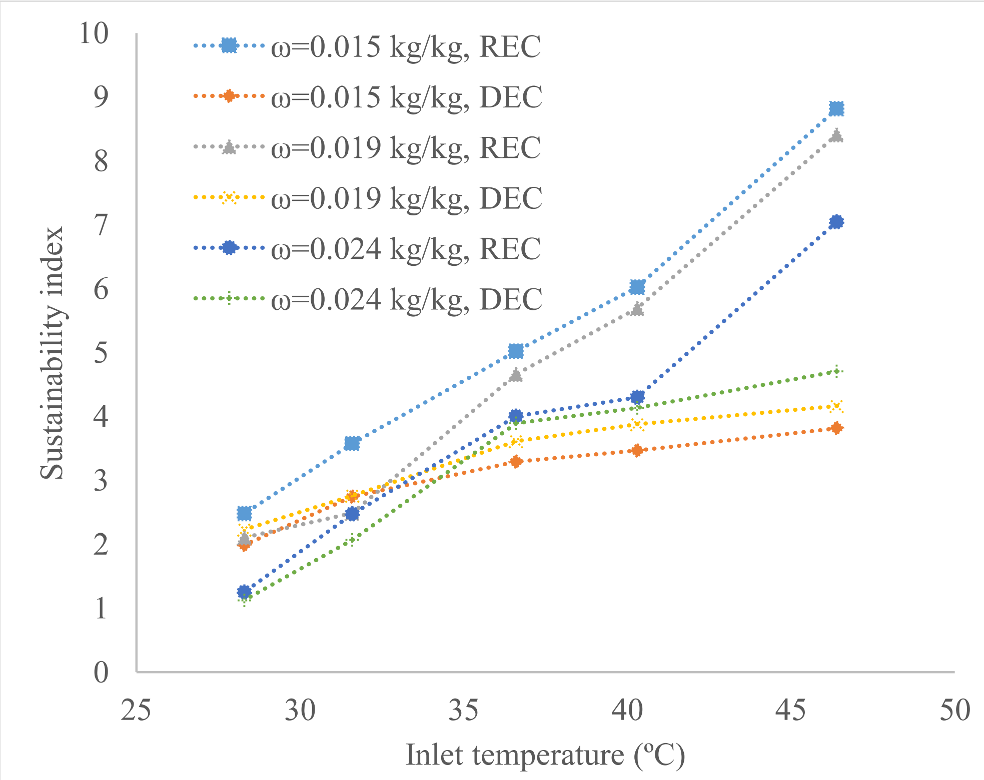

Fig. 5.17: Sustainability index variation with air inlet temperature

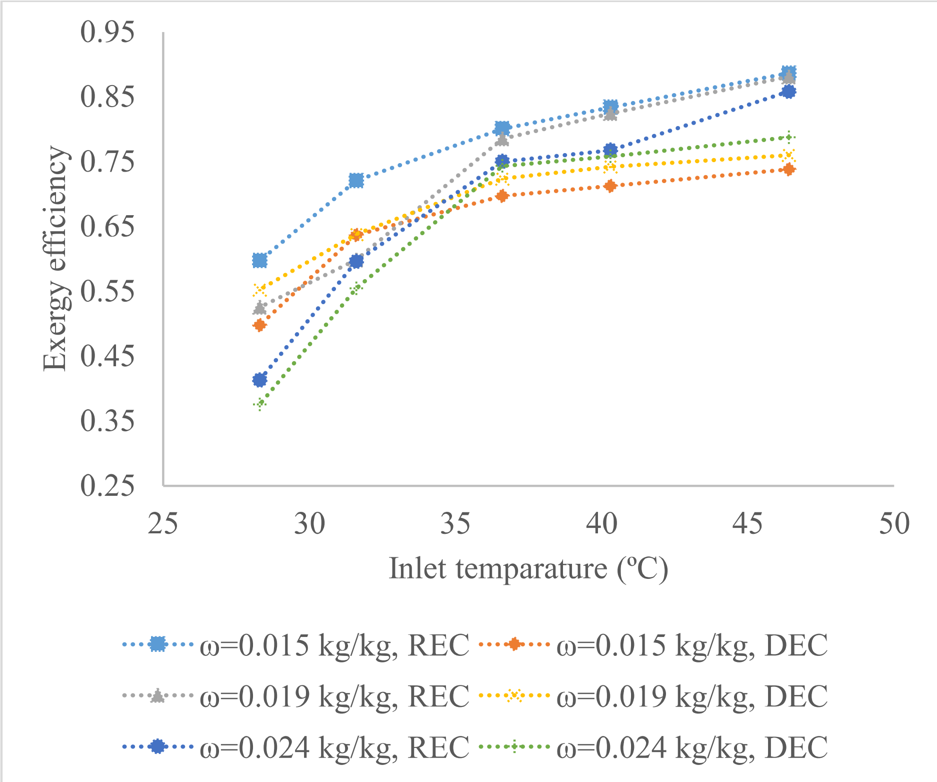

Fig. 5.18: Exergy efficiency variation with air inlet temperature

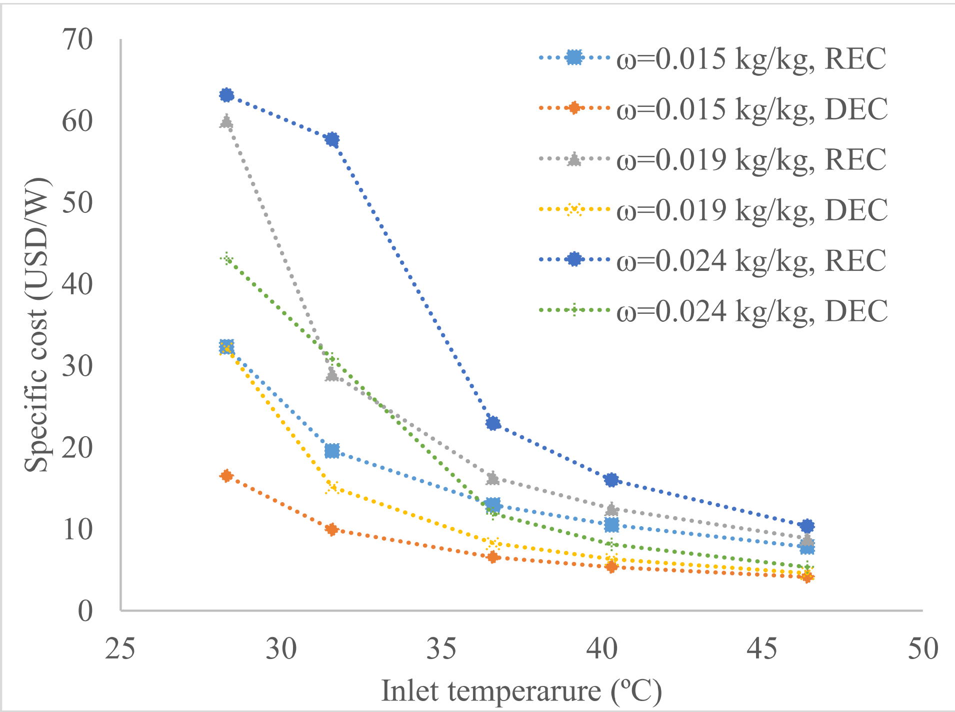

Fig. 5.19: Specific cost variation with air inlet temperature

5.3.4 Influence of water inlet temperature

Fig. 5.20 shows the effect of water temperature on the supply air and cooling capacity of the device. The chiller is used to vary water temperature from 12.6 to 19.5 ºC. The power consumed to reduce the water temperature is not included in the estimation. The presented analysis predict the cooler performance by assuming that reduced temperature water made available by external sources. The supply air temperature increases with the increases in the water temperature. The cool feed water temperature sensible cools both dry and wet channel air, which results in lower supply air temperature. The cooling capacity of the device diminishes as the water temperature increases. The water temperature affects the regenerative mode more as compared to the direct mode. The dew point effectiveness of the regenerative and direct mode decreases by 18.4 % and 6.1 %, respectively. The device performs more effectively in both modes at low water temperature, as shown in Fig. 5.21. The cooling capacity of the direct mode decreased from 70 to 66 W, while for regenerative mode, it is 42 to 34 W with an increase in the water temperature from 12.6 to 19.5 ºC. The cooling capacity decreases only due to a reduction in a temperature drop of the supply air. The COP of the direct mode is higher than the regenerative mode due to only single-pass airflow. The highest dew point effectiveness of 81.4 % is achieved at the low water temperature in the regenerative mode. The COP and cooling capacity improvement of the direct mode is marginal since the less cooing effect is observed in the device. However, the power consumption in cooling the water is not included in the calculation, so to get this improvement in the performance of the device due to cool feed water calculation, it needs to be available without energy consumption.

The groundwater or tap water can be used as the circulating water in the cooling system. The environment temperature varies significantly from April to September in composite climatic zones. The lower temperature of the circulating water reduces the exergy destruction and improves the sustainability index, as presented in Fig. 5.22. The decreased water temperature improves the exergetic performance of the developed cooling device. The low evaporative water temperature increases the effectiveness as well as the cooling capacity of the developed device. The regenerative mode is approximately twice sustainable as compared to the DEC at the water temperature of 19.5 ºC. The exergy ratio of the cooler is in both modes is almost constant with the water inlet temperature variation. The specific cost of the device is decreased with the drop in the water temperature in both modes, as shown in Fig. 5.23. The lowest specific cost of 3.4 USD/W is obtained for DEC operation at the coldest water temperature of 12.6 ºC. The highest specific cost of 7 USD/W is obtained for the regenerative mode at the water inlet temperature of 19.5 ºC.

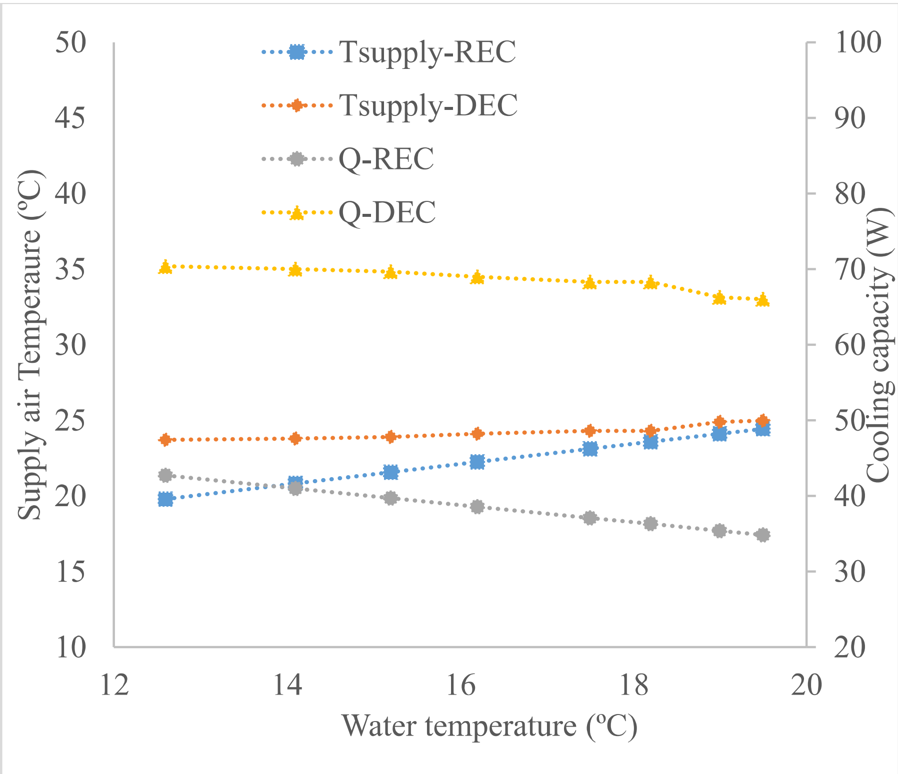

Fig. 5.20: Supply air temperature and cooling capacity variation with water temperature

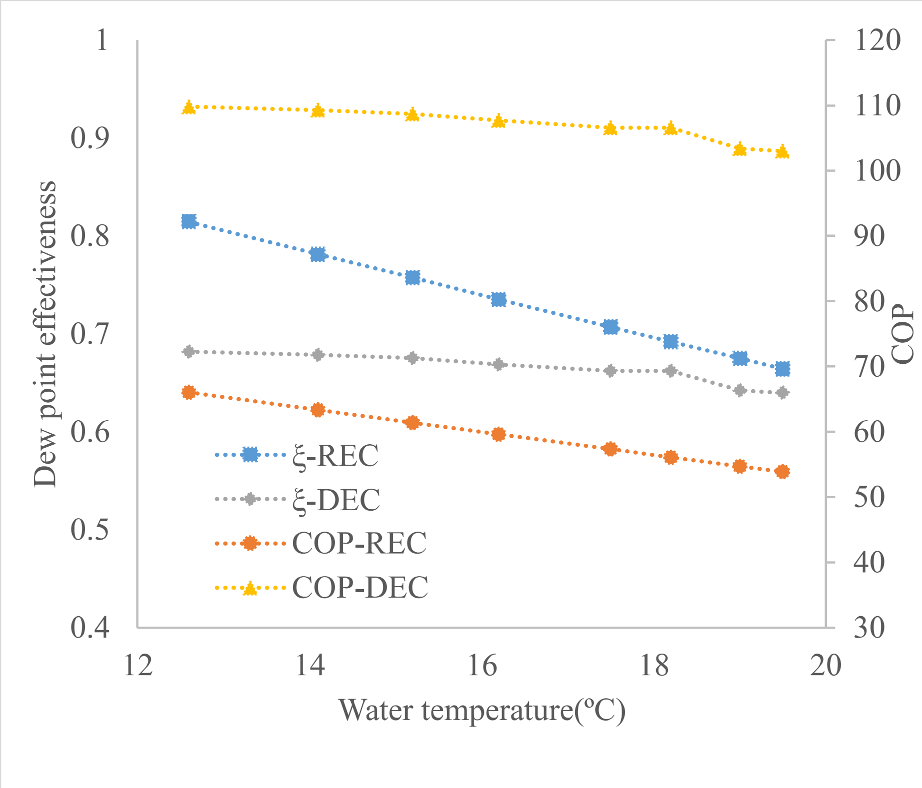

Fig. 5.21: Dew point effectiveness and COP variation with the water temperature

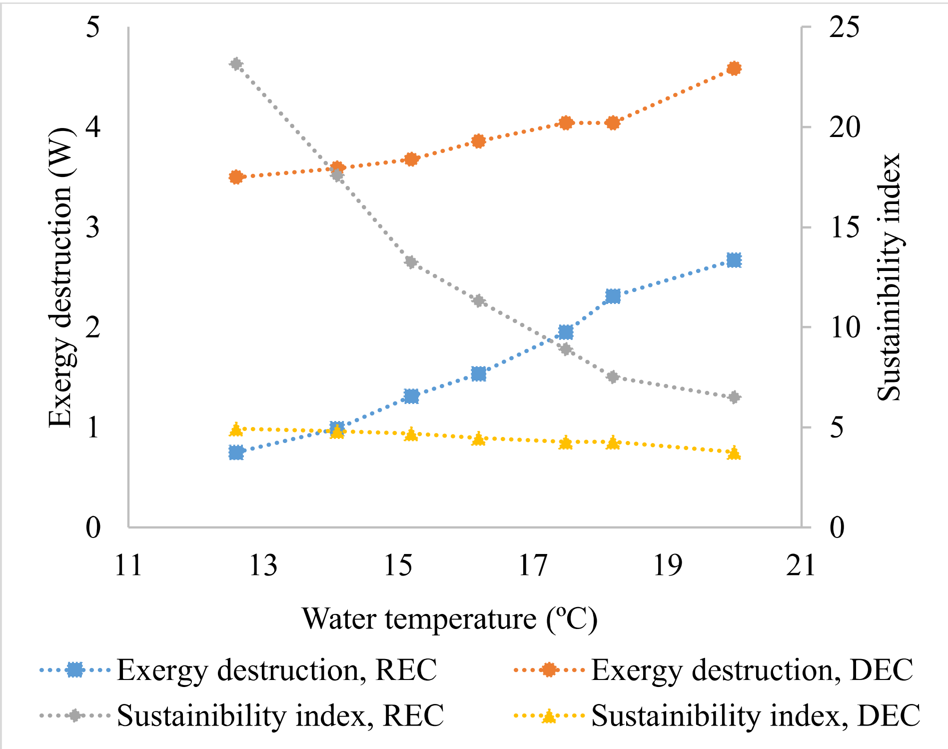

Fig. 5.22: Exergy destruction and sustainability index variation with water inlet temperature

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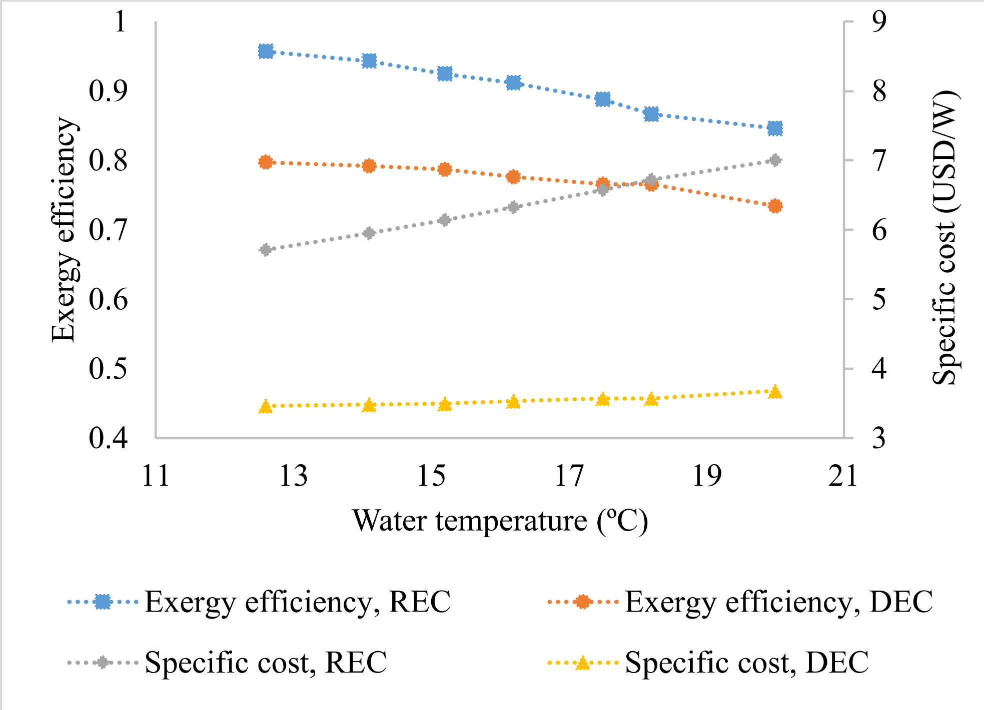

Fig. 5.23: Exergy efficiency and specific cost variation with water inlet temperature

5.3.5 Influence of water flow rate

The thumb flow controller is used to vary the flow rate at each wet surface. The increases in the water circulation rate decrease the supply air temperature slightly. The supply air temperature of the regenerative mode is decreased by 1.7 % while 2.1 % for the direct mode with an increase of 1.7 times feedwater flow rate in the wet channels of the evaporative cooler. The cooling capacity improves correspondingly, as shown in Fig. 5.24. The cooling capacity of the direct mode is 84% higher than the direct mode at 11.2 lph water flow rate. The water circulation temperature is 20 °C, which is lower than the crossing temperature (Liu et al., 2019), hence effectiveness increases with the increase of water flow rate (Fig. 5.25). The increased water flow rate increases pump work which leads to COP reduction. The COP of the regenerative mode is reduced from 34 to 20 with an increase of mass flow rate 11.2 to 19.8 lph. The effectiveness of the regenerative and direct mode is improved by 3.1 and 4.2 %, respectively. The disadvantage of a high flow rate is decreased COP, which is closer to 39.6 % decrease in both modes. The high water flow rate is not useful in these investigated conditions.

The exergy destruction in both DEC and REC increases slightly with the increases of the water volume flow rate, as presented in Fig. 5.26. DEC shows higher exergy destruction than the REC mode of cooling. The sustainability index is almost constant for both REC and DEC modes of cooling. The increased water flow rate decreases the exergy ratio marginally due to a small increment in exergy destruction. The wet-bulb effectiveness, as well as cooling capacity, of the setup, improves with the high water flow rate. The specific cost is reduces slightly due to improved cooling capacity. Fig. 5.27 shows the variation of exergy ratio and specific cost with water flow rate. The lowest specific cost of 5.7 USD/W is obtained for the direct mode at a 19.8 lph water flow rate.

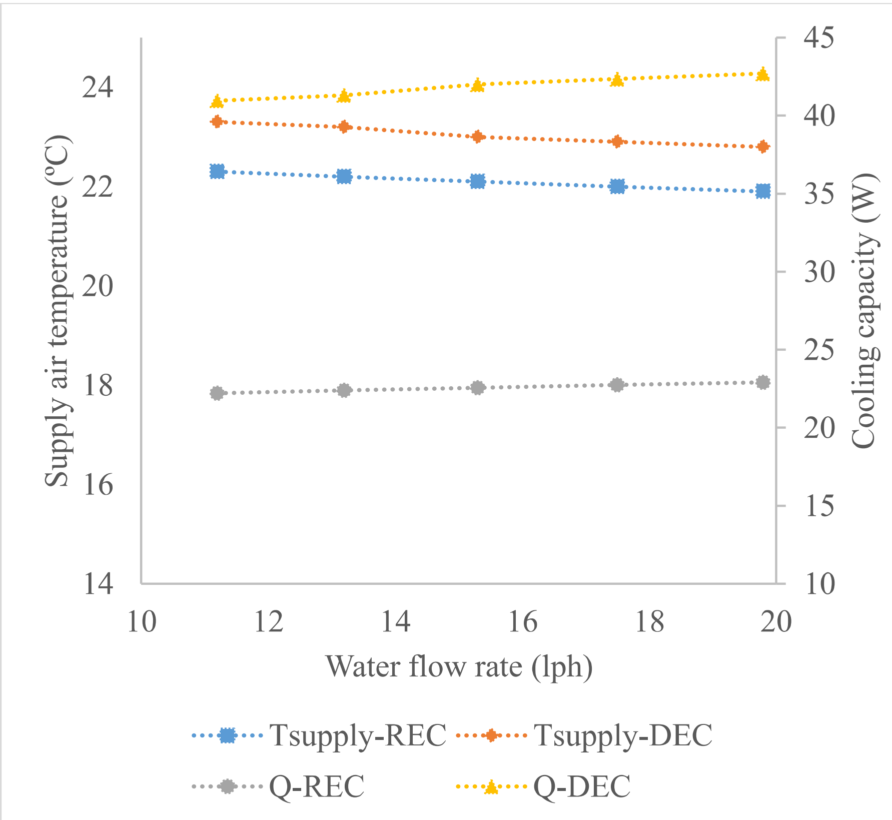

Fig. 5.24: Supply air temperature and cooling capacity variation with water flow rate

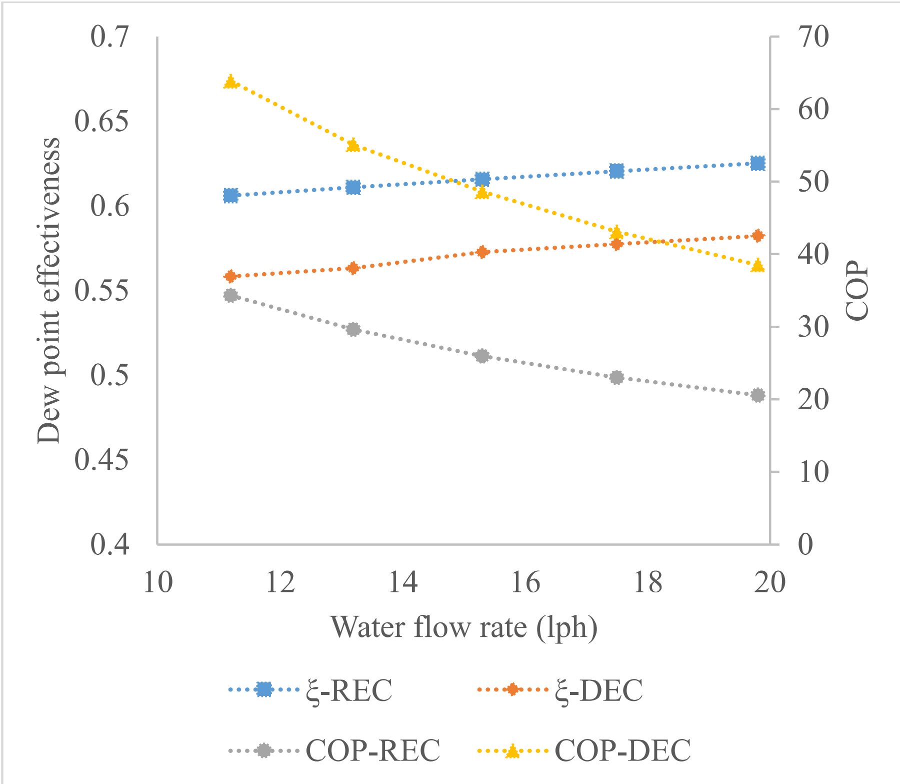

Fig. 5.25: Dew point effectiveness and COP variation with the water flow rate

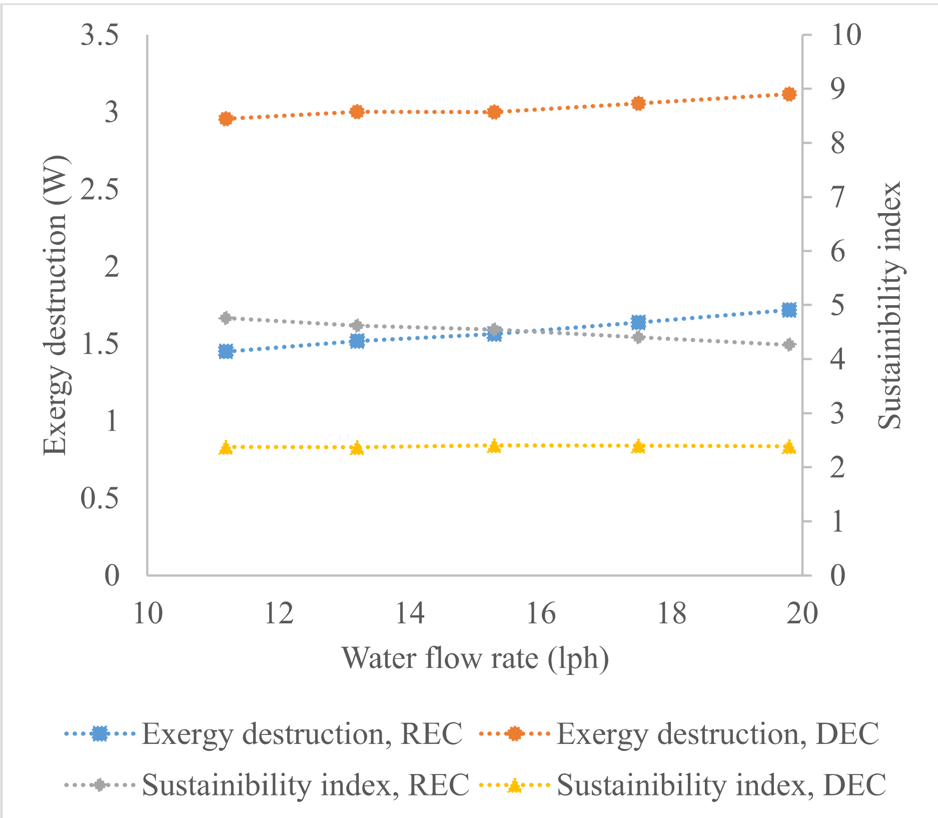

Fig. 5.26: Exergy destruction and sustainability index variation with the water flow rate

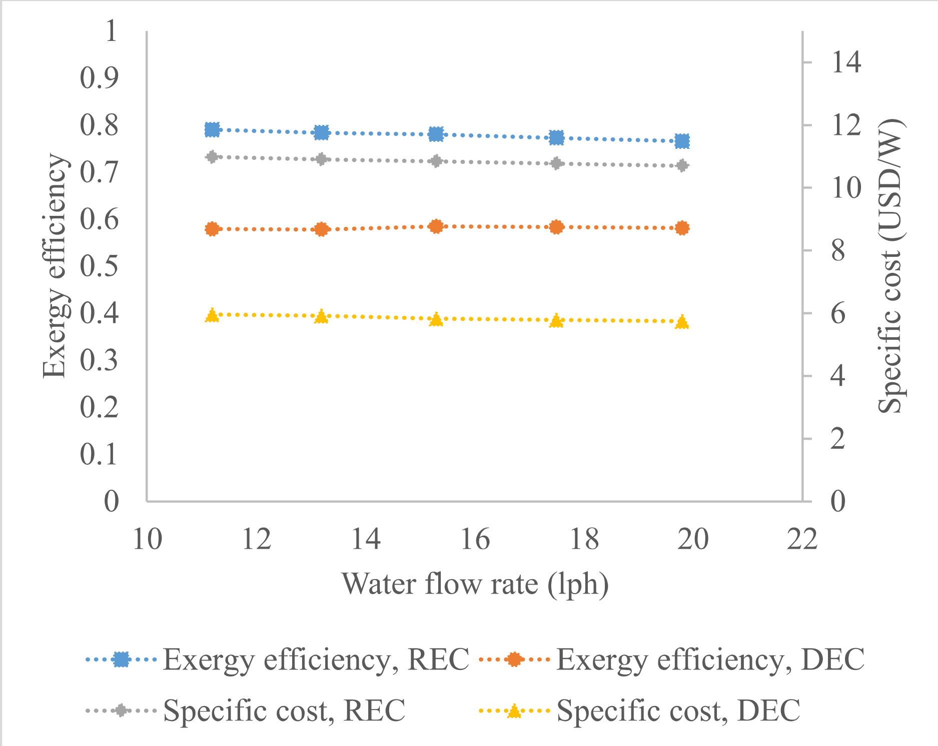

Fig. 5.27: Exergy efficiency and specific cost variation with the water flow rate

5.3.6 Thermal comfort assessment

The psychrometric representation of the different outlet conditions by varying the above-investigated air inlet conditions is shown in Fig. 5.28. The points shown are left, and the bottom of the thermal comfort zone is suited conditions for thermal comfort. These points can be easily finished in the thermal comfort zone by just increasing the velocity or other varying other in-hand parameters. The point depicted above the comfort zone is not suited for evaporative cooling. The experimentation is done mostly in dry conditions (humidity of 0.010 kg/kg). Almost all investigated outlet conditions are in thermal comfort except a few cases. The high-velocity conditions (2 and 2.5 m/s) in the direct mode supply the air out of thermal comfort. The supply air temperature for inlet humidity of 0.019 and 0.024 kg/kg is out of the thermal comfort in regenerative mode. The direct mode is also not able to cool air in the thermal comfort range for intake air humidity of 0.015, 0.019, 0.024 kg/kg. In some cases (REC mode at the 0.015 kg/kg humidity), supplied air is out of thermal comfort, but it can be forced into the comfort zone by reducing the velocity. The specific inlet conditions for the switch in the mode from DEC to REC vary with the variation of temperature and humidity (outdoor climatic conditions). The DEC mode is suited best for the 0.010 kg/kg humidity for all investigated temperature points except 46.4 C. The direct mode is found to be suitable up to 0.015 kg/kg humidity and temperature 31.6 ºC. The higher humidity for all investigated temperature range is not suited for direct mode. The investigated conditions show that if the wet-bulb temperature of outdoor conditions is less than 24 ºC, the operating mode should be DEC. The dual-mode device should be switched to REC mode after this wet bulb condition. However, the supply air condition for inlet humidity above 0.019 kg/kg is out of thermal comfort and hence not suitable for both modes of operation.

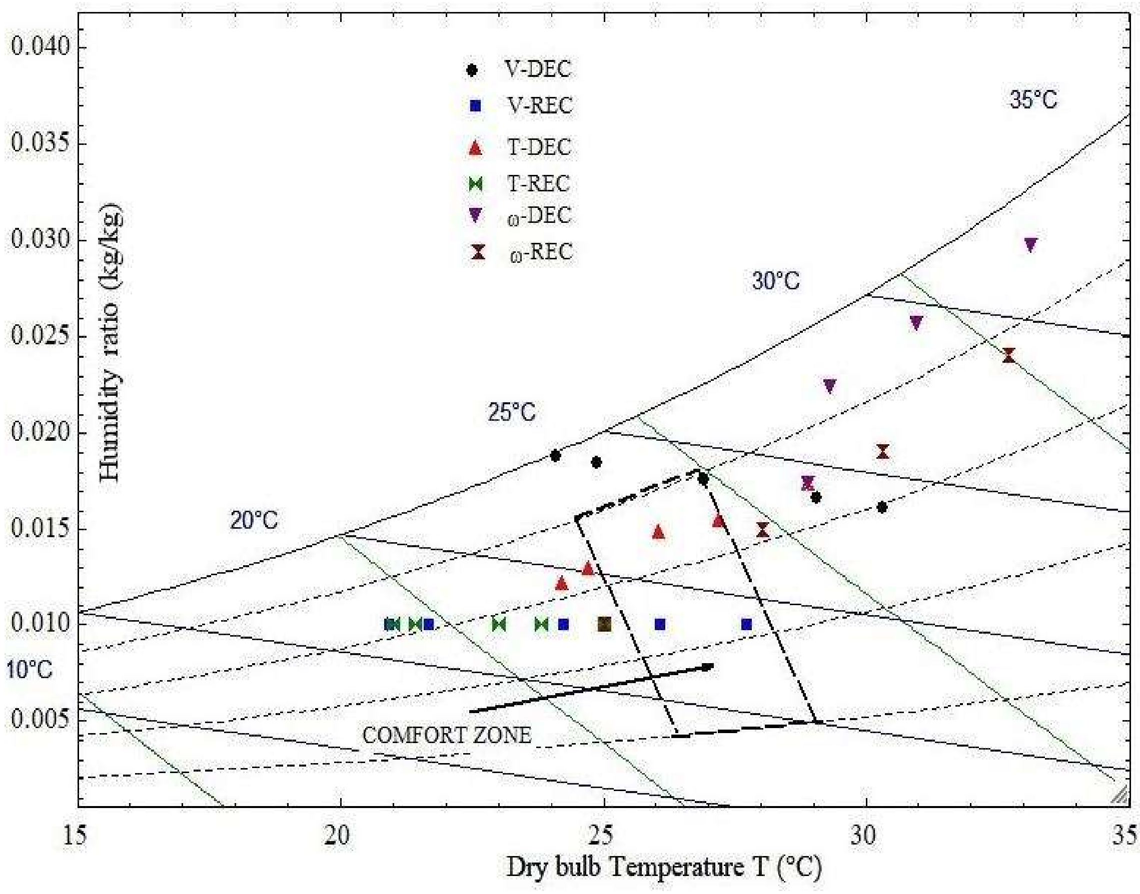

Fig. 5.28: Psychrometric chart representation of supply air conditions with inlet air velocity, temperature, and specific humidity variations

5.3.7 Techno-economic Usefulness

The developed dual-mode evaporative cooler is economically compared with a vapor compression-based air conditioner available in the market, which is conventionally used for comfort cooling purposes. The fabricated cooling device is only a prototype testing unit of a full-fledged dual-mode evaporative cooling device. The investigated cooling unit has a very low cooling capacity (34 W for REC and 57 W for DEC at mean operating conditions). The initial cost of the device comes to around 242.2 USD. The cost of 1.5-ton air conditioner in India is around 536 USD. The fabrication of a full-fledged dual-mode cooler will increase the cost of the heat and mass exchanger, and it will come close to the cost of the conventional air conditioner. The major difference in cost comes in operating these two cooling devices (conventional air conditioner and dual-mode evaporative cooler). The COP of a 5 star rated air conditioner is 4.51, but the dual-mode cooler shows 52.9 (for REC at mean conditions). The full-fledge dual-mode device can be assumed to operate nearby this COP. The running cost for an air conditioner is 292.6 USD, while for a dual-mode cooler is just 24.94 USD when both devices operate for 180 days in a year. This cost difference shows the economic usefulness of the developed evaporative cooling device.

The presented experimental setup is a prototype with only four channels for investigating purposes. Hence this device has a low cooling capacity, but the border version (which has more number channels) of this device may be suited best for cooling purposes. The conventional alternates available in the market are vapor-compression-based air conditioners. These two cooling devices can be compared directly when both are of almost the same cooling capacity. But an outline of comparison is presented in Table 5.7. The comparison presented in the table clearly shows that this cooler may be a very useful and efficient device in the present scenario.

Table 5.7: comparison between Vapor compression air – conditioning system and Dual mode evaporative coolers

Characteristics	Vapor compression AC	Dual mode cooler
Coolant	CFCs/HFCs	Water
Ventilation	20% Outside air	100% outside air
Power consumption	High	Significantly Lower than AC
Water consumption	Moderate water needed at power plant	High
Fabrication	Complicated	Easy
Weather	Applicable in all climate types	Better in hot, dry less humid climates.
COP	low	High
Cost	costly	cheaper
Emission	Very high	emission saving

5.4. Important findings

A dual-mode evaporative cooler is fabricated and tested in this work. The evaporative cooling device is operated in both direct and regenerative modes. The performance of the developed cooler is compared in DEC and REC modes. The performance parameters are calculated by varying the operating parameters of the device. The key outcomes are as follows:

- The higher extraction ratio in regenerative mode increases the effectiveness of the cooler but reduces cooling capacity sharply. 52% decrease in COP while 10% increase in effectiveness is obtained within the studied range. The minimum extraction ratio value should be decided such that supply air temperature lies in the thermal comfort.
- The high extraction ratio in REC operation raises the exergy destruction and specific cost of the cooler. The sustainability of the device also decreases with the extraction ratio increases.
- The increased velocity reduces effectiveness but improves cooling capacity. The effectiveness decreases by 29 %, while cooling capacity increases drastically in both modes. The inlet velocity should be maximized to achieve thermal comfort cooling. The high inlet velocity of the device increases sustainability and reduces the specific cost of the coolers in both modes. The specific cost reduction of 72% in REC while 75% in DEC is achieved.
- The dew point effectiveness of the device improves with an increase in the inlet temperature (dew point depression) at constant specific humidity conditions.
- The exergy efficiency improves with the increases in the intake air temperature in both modes. The hot (high inlet temperature) and dry (low specific humidity) conditions show higher sustainability and low specific cost for the cooler in both modes. The coolers proved to be more useful in these conditions
- The lower water temperature increases the cooling performance of the device in both modes. The effectiveness decreased by 6.1 % in direct mode. The regenerative mode is more sensitive to the water inlet temperature. The cooler water enhances the exergetic performance of the cooling unit in both direct and regenerative modes. The exergy destruction increased by 1.3 times in direct mode while 3.5 times in

regenerative mode with an increase in water inlet temperature. The REC is more sensitive to the water inlet temperature.

- The increase in water flow rate (at the temperature of 20 ºC) increases effectiveness slightly, but COP of the cooler diminishes. The cooling capacity improvement is marginal, while COP decrease is closer to 39%. Hence high flow rate is not recommended in the investigating conditions. The higher flow rate of water enhances the exergy destruction and reduces sustainability marginally.
- DEC mode is applicable for inlet wet bulb temperature below 24ºC, while REC mode is applicable for the air inlet condition above 24ºC wet bulb temperature and below about 0.019 kg/kg specific humidity.
- This device is more economical in operation in both modes when compared to a conventional vapor compression-based air conditioner.

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