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Energy, exergy and economic assessments of the dual-mode evaporative cooler for various international climate zones

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Abstract

The conventional desert cooler is effective for dry seasons and the regenerative evaporative cooler (REC) is an effective device for humid seasons in composite climate zones. Hence, the dual-mode evaporative cooler (a two-in-one device) is an intelligent choice for air conditioning, which can operate in both direct and regenerative modes depending on the seasonal climatic condition. The exergy and economic analyses of this novel device for global climatic conditions are performed to check the suitability in different regions of the world. An experimental prototype of a dual-mode evaporative cooler is developed and tested to validate the simulation model. The effectiveness, coefficient of performance, exergy destruction, exergy efficiency, operating cost, and specific total cost (STC) are evaluated for both (direct and regenerative) modes of operation. The annual and month-wise performances of dual-mode evaporative cooler have been assessed for five cities of international climate zones. The operating cost of both modes is compared by considering electricity charges in different countries. The dual-mode device is compared with the single-mode device as well. The specific cost is similar for both modes in most of the ASHRAE climatic zones. The present study reveals that significant energy and cost savings are possible by using the dual-mode evaporative cooler.

Practical application: This article considers the application of a dual-mode evaporative cooler (direct as well as regenerative mode) in different climate zones and, through investigating the exergy and economic performances, allows designers and operators to understand the potential benefits of employing various operating modes in particular climates.

Keywords

Dual-mode evaporative cooler, experimentation, coefficient of performance, wet bulb effectiveness, exergy destruction, specific total cost

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Introduction

For most of the hot and humid regions in the world, about 6 months remain uncomfortable without using the air-conditioning system. Today's cooling technology is dominated by a vapor-compression air-

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conditioning system, which contributes to the significant sharing of building energy consumption as well as negative environmental effects. Hence, eco-friendly, as well as low energy consumption substitute, needs to be incorporated, such as evaporative cooling. Evaporative cooling technologies are further classified into direct evaporative cooling (DEC) and indirect evaporative cooling (IEC). Direct evaporative cooling is a mature technology and used worldwide. However, it has a limitation of wet bulb temperature and becomes ineffective in high humidity conditions.[1] Direct evaporative cooling adds moisture to the air, which makes occupants uncomfortable. Indirect evaporative cooling is developed to cool air at a constant humidity condition. Indirect evaporative cooling has been further modified as a regenerative evaporative cooler (REC), which is effective in wet conditions.[2] This modification has been done by extracting a fraction of dry air and utilizing it as wet channel air. Regenerative evaporative cooler consists of a heat and mass exchanger (HME) in which alternate dry and wet channels are separated by non-permeable sheets. The dry channel air gets cooled at a constant specific humidity by exchanging heat with wet channels, and a portion of this cooled air is redirected in the wet channels. However, it has a disadvantage of low cooling energy produced as compared to DEC. To trade-off between suitability and running cost of these coolers with seasonal climate variation, the present authors have recently proposed a dual-mode evaporative cooler, which can operate as DEC as well as REC as per requirement.[3]

The exergy analysis of the evaporative cooler helps to identify the inefficiency of both heat and mass transfer processes. The forms of exergy involved in evaporative cooling are thermal exergy, chemical exergy, and mechanical exergy. Direct evaporative cooling cools air at the cost of increased humidity and hence the chemical exergy losses to obtain the thermal exergy. Regenerative evaporative cooler cools the air by preserving the chemical exergy; that is why it can be further cooled below the wet bulb temperature. This preserved exergy is utilized in the wet air channel and transferred to dry air. The evaporative cooling process utilizes humidity potential (chemical exergy) to obtain cool air (thermal exergy). However, in this conversion process, total exergy (sum of thermal, mechanical, and chemical exergies) deteriorates due to irreversibility, which is calculated in terms of exergy destruction. The ratio of useful output exergy to total exergy incoming is called exergy efficiency. The exergy analysis helps to compare the performance of the evaporative coolers, identify the inefficiencies, and improvement measures.

Within the last decade, many exergy and economic analyses have been performed on IEC. Chengquin et al.[4] performed exergy analysis and compared five different cases of evaporative cooling schemes and found that REC performs better as compared to the others. Farahani et al.[5] performed exergy analysis on the direct, indirect, and two-stage combined evaporative coolers and found two-stage cooler is a good choice for hot and semi-humid climates. Caliskan et al.[6] performed exergoeconomic, environmental, and sustainability analyses of the M-cycle evaporative cooler. Duan et al.[7] conducted an experiment on the counter flow sub-wet bulb cooler for the different climatic conditions of China and calculated the reduction of cooling load. Jafarian et al.[8] numerically simulated the counter flow REC with the new boundary conditions at the separating wall and found good prediction using a two-dimensional model (with error less than 4.5%). Lin et al.[9] conducted exergy analysis on the counter flow dew point evaporative coolers and found that exergy destruction is about 10–25% of inlet air exergy. Sohani et al.[10] performed an economic analysis of a two-stage indirect and direct evaporative cooler (DEC) and optimized pad thickness, inlet velocity, specific contact area, and working air ratio. Bakeem et al.[11] did energy, exergy, and economic analyses of DEC and the effect of hourly variation of temperature on effectiveness and capital cost is investigated. Arun and Mariappan[12] presented experimental results of an ultrasonic REC and used an ultrasonic ceramic transducer. Pakari et al.[13] compared one-dimensional and three-dimensional models of REC with the experimental result and concluded that the one-dimensional model is adequate enough to predict the performance. Cui et al.[14] simulated multistage evaporative cooler and found some trade-off between performance and fan power. Wang et al.[15] conducted a combined energy–exergy analysis of IEC and optimized for thermal performance. Kashyap et al.[16] numerically simulated the different configurations of REC by changing the airflow and water flow direction and obtained the best configuration based on exergy, economic, and sustainability analyses. However, exergy and economic analyses have not been conducted for a dual-mode evaporative cooler.

ASHRAE divides the world into eight different international climatic zones: four are hot and four are cold zones.[17] For hot zones, DEC is the proven technology for dry seasons and REC is a better option for humid seasons. Hence, to abstain from the use of two coolers for the same purpose, a dual-mode evaporative cooler is already proposed[3] that can operate in dual mode (direct as well as a regenerative indirect mode). However, further detailed analysis of this novel device is required for various world climate zones (cities) for global acceptability. Hence, in this study, exergy and economic analyses of this novel device are performed to check its exergy and economic feasibilities. Its performance (both modes) for four different international climatic hot zones is also investigated to check worldwide acceptability. For model validation, the experimental test rig of a dual-mode evaporative cooler has been developed and tested. The cooling energy produced, effectiveness, coefficient of performance (COP), exergy destruction, exergy efficiency, and specific total cost (STC) of the device are investigated for both operational modes.

Working of dual-mode evaporative cooling device

The dual-mode cooler works in either REC or DEC mode, depending on the outside condition. The system works on an airflow circuit consisting of a wet channel flow for DEC mode and a dry side flow for regenerative IEC mode. The HME used in the evaporative cooling device is similar to the regenerative indirect evaporative cooler (IEC) with an additional vane. In REC mode, the outdoor air enters the device at point 1, as shown in Figure 1. The air passes through the dry channel and gets cooled sensibly. The dry channel and wet channel are separated by a non-permeable plate. A fraction of this cooled dry air is sent to the cooling space at point 2. In this mode, vane 2 remains fully opened, while

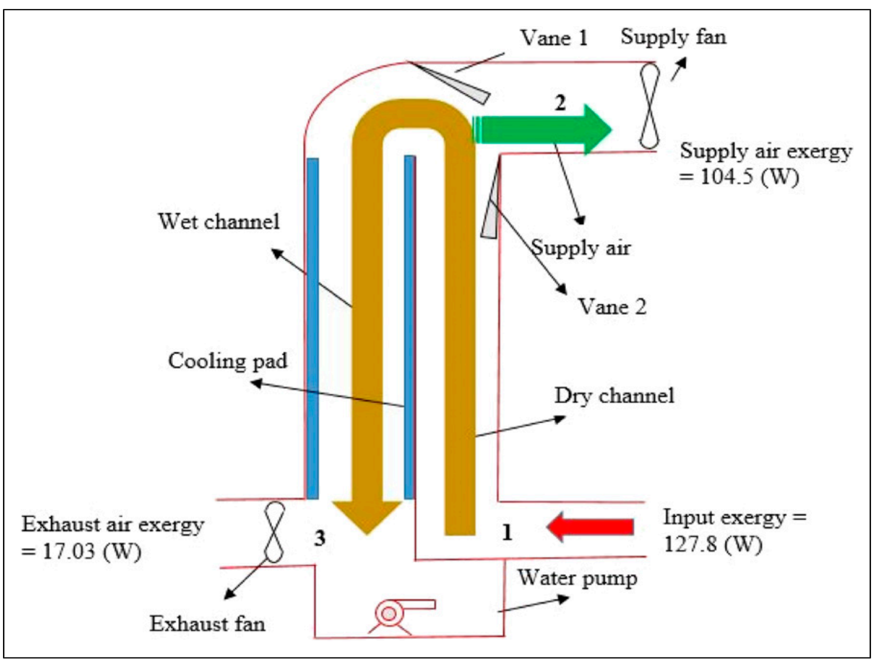

Figure 1. Dual-mode evaporative cooler in indirect mode regenerative evaporative cooler.

vane 1 controls the extraction ratio (wet channel air to dry channel air). In the wet channel, the air gets humidified and it is then discharged into the atmosphere at point 3. In DEC mode operation of the device, vane 1 remains fully opened while vane placed at the end of the dry channel (vane 2) remains fully closed. Hence, there is no flow of air through the dry channel. The atmospheric airflows in the device at point 1, as shown in Figure 2. While passing through the wet channel, the air gets humidified and cooled. This air is supplied to the conditioning space at point 4.

The flow process of air inside a dry channel and wet channel for indirect mode is shown on the psychometric chart in Figure 3. Outdoor air sensibly cools in the dry channel during the process 1–2, and the flow process of redirected air in the wet channel is shown by 2–3. The flow process of air in the direct mode is also represented on the psychometric chart by the process 1–4 in Figure 3. Control over the two different modes is achieved by controlling the vanes. Vane 2 remains fully opened in REC mode and remains fully closed in DEC mode. Vane 1 remains partially opened (depending upon extraction ratio) in REC mode while remains fully opened in DEC mode. Hence, by controlling vane 1 and vane 2, the dual-mode evaporative cooler can be switched from DEC mode to REC mode or vice versa depending on the outdoor air condition change.

Numerical model and simulation

The mathematical model presented here is based on the steady-state energy conservation in both dry and wet channels and mass conservation in the wet channel.[13] Conservation equations are written according to the flow direction. The x coordinate is taken in an upward direction[3] for numerical modeling. Some assumptions are used to simplify and develop the model:

(i) The outer walls of the evaporative cooler are assumed to be adiabatic by considering proper insulation, so there is only heat exchange inside the device.
(ii) The airflow inside both channels is considered laminar as the flow rate is taken as very small (high velocity is not desirable).

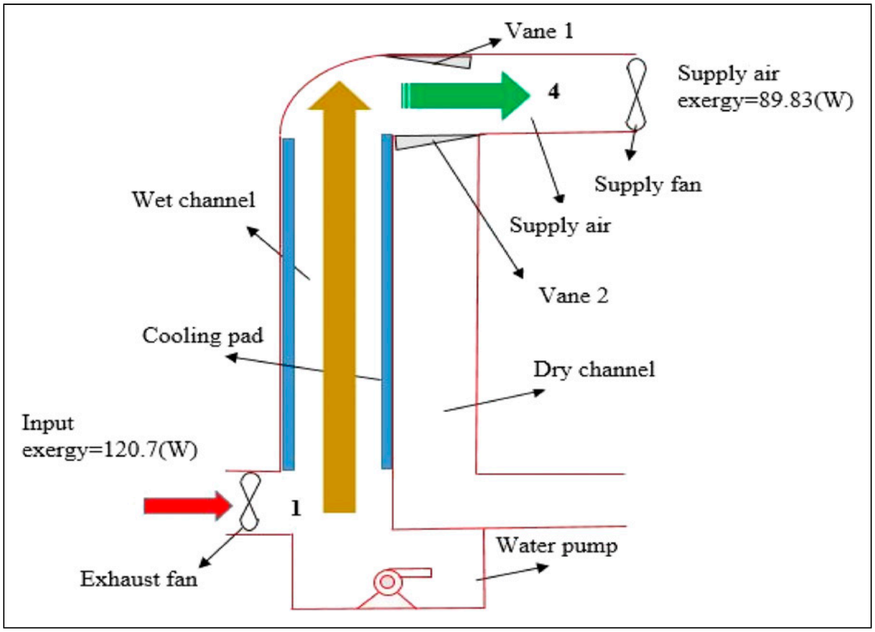

Figure 2. Dual-mode evaporative cooler in direct mode direct evaporative cooler.

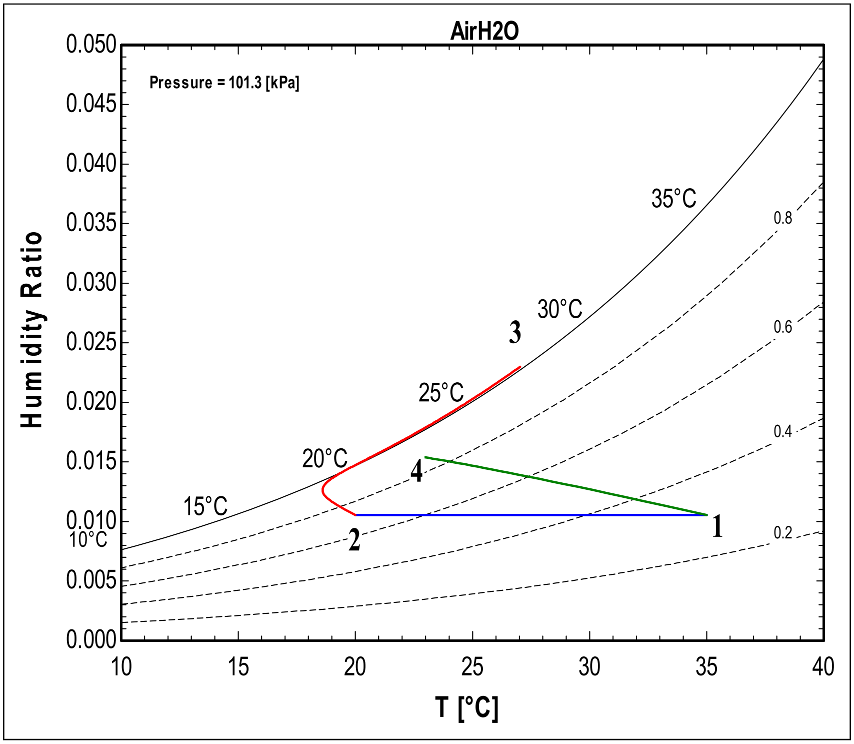

Figure 3. Flow processes in both DEC and REC modes on psychometric chart. Note: DEC, direct evaporative cooler; REC, regenerative evaporative cooler.

(iii) The gap between the two channels is very small and the gap to width ratio is very less, so it is treated as a 1D problem.[3]
(iv) Airflow inside the dry and wet channel is assumed to be incompressible and steady.
(v) Thermal and physical properties of water vapor and flowing air are assumed to be temperature-dependent.
(vi) Properties of water are temperature-dependent and there is no axial conduction in water film.

Energy and mass conservations for direct evaporative cooler mode

The mass and energy balances for DEC mode are given by, respectively

$$m_{aw}\frac{d\omega_{aw}}{dx} = \alpha_m W\left(\omega_{wf} - \omega_{aw}\right) \tag{1}$$

$$m_{aw}c_{p,aw}\frac{dT_{aw}}{dx} = -\alpha_{aw}\left(T_{aw} - T_{wf}\right)W \tag{2}$$

The conservation of energy for the water film is given by

$$m_{wf}c_{p,wf}\frac{dT_{wf}}{dx} = \alpha_{aw}\left(T_{aw} - T_{wf}\right)W - h_{fg}\alpha_m\left(\omega_{wf} - \omega_{aw}\right)W \tag{3}$$

Energy and mass conservations for REC mode

The differential equation for energy conservation in the dry channels is given by

$$m_{ad}c_{p,ad}\frac{dT_{ad}}{dx} = -U\left(T_{ad} - T_{wf}\right)W \tag{4}$$

where the overall heat transfer coefficient between dry air and water (U) is obtained by

$$\frac{1}{\mathrm{U}} = \frac{1}{\alpha_{ad}} + \frac{t_{\text{plate}}}{k_{\text{plate}}} + \frac{1}{\alpha_{wf}} \tag{5}$$

The mass and energy balance equations in the wet channel is given by, respectively

$$m_{aw}\frac{d\omega_{aw}}{dx} = \alpha_m W\left(\omega_{wf} - \omega_{aw}\right) \tag{6}$$

$$m_{aw}c_{p,aw}\frac{dT_{aw}}{dx} = -\alpha_{aw}\left(T_{aw} - T_{wf}\right)W \tag{7}$$

The conservation of energy at the water film is given by

$$m_{wf}c_{p,wf}\frac{\partial T_{wf}}{\partial x} = U\left(T_{ad} - T_{wf}\right)W + \alpha_{aw}\left(T_{aw} - T_{wf}\right)W - h_{fg}\alpha_m\left(\omega_{wf} - \omega_{aw}\right)W \tag{8}$$

Heat and mass transfer coefficients

The small channels gap results in a small hydraulic diameter and hence the low velocity and small hydraulic diameter make airflow laminar. Hence, the heat transfer coefficients of air in both channels and water have been calculated by[18,19]

$$\alpha_{ad} = 7.54(k_{ad}/d_{h,d}) \tag{9}$$

$$\alpha_{aw} = 7.54(k_{aw}/d_{h,w}) \tag{10}$$

$$\alpha_{wf} = 0.023\mathrm{Re}_{wf}^{0.8}\mathrm{Pr}^{0.3}\left(k_{wf}/d_{h,wf}\right) \tag{11}$$

The mass transfer coefficient in the wet channel is determined by[20,21]

$$\alpha_m = \alpha_{aw}/\left(0.87^{2/3}c_{p,aw}\right) \tag{12}$$

Boundary conditions

For DEC mode, inlet temperature and specific humidity are known and specified as

$$T_{\text{aw}} = \text{Tin at } x = 0 \tag{13}$$

$$\omega_{\text{aw}} = \omega_{\text{in}} \text{ at } x = 0 \tag{14}$$

$$T_{\text{wf}} = T_{\text{wf,in}} \text{ at } x = L \tag{15}$$

Similarly for REC mode, inlet temperature and specific humidity are specified as[3]

$$T_{\text{ad}} = T_{\text{in}} \text{ at } x = 0 \tag{16}$$

$$\omega_{\text{ad}} = \omega_{\text{in}} \text{ at } x = 0 \tag{17}$$

$$T_{\text{aw}} = T_{\text{ad}} \text{ at } x = L \tag{18}$$

$$\omega_{\text{aw}} = \omega_{\text{ad}} \text{ at } x = L \tag{19}$$

$$T_{\text{wf}} = T_{\text{wf,in}} \text{ at } x = L \tag{20}$$

Numerical simulation

The governing differential equations are discretized with the help of a finite difference scheme. These equations are implemented in the engineering equation solver (EES).[22] Properties of water, air, and water vapor are calculated with in-built functions in the EES and then heat and mass transfer coefficients have been estimated. Systems of equations in each grid are solved using a variation of Newton's method. Then, performance parameters have been evaluated. It has been observed from the grid-independent test that the solution is converged after about 150 nodes.

Energy calculation

The wet bulb effectiveness and cooling energy produced are important factors in the evaporative cooling technology. Hence, the wet bulb effectiveness is given by

$$\varepsilon = \frac{T_{\text{in}}\text{-}T_{\text{out}}}{T_{\text{in}}\text{-}T_{\text{wet-bulb}}} \tag{21}$$

The cooling energy produced or cooling capacity of the evaporative cooling devices is defined as the sensible cooling of the supply air and is given as[10]

$$Q_{\text{system}} = m_{\text{out}}c_p\left(T_{\text{in}} - T_{\text{out}}\right) \tag{22}$$

where m_{out} and T_{out} are air mass flow rate and outlet temperature, respectively.

The pressure drop of REC mode consists of pressure loss in the dry channel and pressure loss in the working channel.[23] The pressure drop of dry air consists of sudden contraction loss, frictional loss, line flow loss, and outlet sudden expansion loss

$$\Delta p_{ad} = 0.25\rho_{ad}u_{ad}^2 + \frac{48L\rho_{ad}u_{ad}v_{ad}}{d_h^2} + 0.05\rho_{ad}u_{ad}^2 + 0.5\rho_{ad}u_{ad}^2 \tag{23}$$

The pressure drop of the wet channel consists of sudden contraction, frictional loss in the dry channel, branch flow loss, diversion loss coefficient, frictional loss in a wet channel, and outlet sudden expansion loss

$$\Delta p_{aw} = 0.25\rho_{ad}u_{ad}^2 + \frac{48L\rho_{ad}u_{ad}v_{ad}}{d_h^2} + 0.45\rho_{aw}u_{aw}^2 + 0.75\rho_{aw}u_{aw}^2 + \frac{48L\rho_{aw}u_{aw}v_{aw}}{d_h^2} + 0.5\rho_{aw}u_{aw}^2 \tag{24}$$

In DEC mode, pressure drop calculation is similar to the dry air pressure drop as described above. Theoretical fan power: $P_{\mathrm{fan}} = \Delta p_{\mathrm{ad}} \times V_{\mathrm{ad}} + \Delta p_{\mathrm{aw}} \times V_{\mathrm{aw}}$. The actual fan power consumption is 120–170% more than the theoretical power.[24] Theoretical pump power = mgH, where $m = n \times m_{\mathrm{aw}} \times (\omega_{\mathrm{aw-out}} - \omega_{\mathrm{aw-in}})$, g = acceleration due to gravity, and H is the height to be raised. Actual pump power (P_{pump}) is theoretical pump power by pump efficiency, which is taken as 80%. The COP is given by

$$\mathrm{COP} = \frac{Q_{\mathrm{system}}}{P_{\mathrm{fan}} + P_{\mathrm{pump}}} \tag{25}$$

Exergy calculation

The total flow exergy of the humid air per kg of dry air is expressed as[4]

$$\psi = c_pT_0\left(\frac{T}{T_0} - 1 - \ln\frac{T}{T_0}\right) + (1 + 1.608\omega) \times R_aT_0\ln\frac{p}{p_0} + R_aT_0\left\{(1 + 1.608\omega) \times \ln\frac{1 + 1.608\omega_0}{1 + 1.608\omega} + 1.608\omega\ln\frac{\omega}{\omega_0}\right\} \tag{26}$$

where T_0, p_0, and ω_0 are the temperature, pressure, and humidity at the reference state, respectively. R_a is the specific gas constant of dry air. This total exergy is separated into thermal exergy, mechanical exergy, and chemical exergy. Exergy balance for the direct mode of an evaporative cooler is given by[5]

$$m_{aw}\psi_{aw-\mathrm{in}} + m_{wf}\psi_{wf} - m_{aw}\psi_{aw-\mathrm{out}} - I = 0 \tag{27}$$

where $m_{wf} = m_{aw}(\omega_{aw-\mathrm{in}} - \omega_{aw-\mathrm{out}})$ and $\psi_{wf} = -R_vT_0\ln\phi_0$.

R_v is the specific gas constant of water vapor, and ϕ_0 is the relative humidity at the reference state. The atmospheric saturation state is considered the reference state.

Hence, the exergy efficiency for the direct mode is defined as[6]

$$\eta_{ex} = \frac{m_{aw}\psi_{aw-\mathrm{out}}}{m_{aw}\psi_{aw-\mathrm{in}} + m_w\psi_{wf}} \tag{28}$$

Exergy balance for the regenerative mode is given by[9]

$$m_{ad}\psi_{ad-\mathrm{in}} + m_{wf}\psi_{wf} - m_{ad-\mathrm{out}}\psi_{ad-\mathrm{out}} - m_{aw}\psi_{aw-\mathrm{out}} - I = 0 \tag{29}$$

Apart from exergy destruction, the exhaust air exergy is also not utilized. Hence, the exergy efficiency for the regenerative mode is defined as

$$\eta_{ex} = \frac{m_{ad-\mathrm{out}}\psi_{ad-\mathrm{out}}}{m_{ad}\psi_{ad-\mathrm{in}} + m_{wf}\psi_{wf}} \tag{30}$$

Economic analysis

The total cost is the sum of the operating cost and the capital cost (initial cost) of the device. The main components of the evaporative cooling device are fan, pump, HME, etc. Hence, the capital cost is given by

$$C_{\mathrm{Capital}} = C_{\mathrm{fan}} + C_{\mathrm{pump}} + C_{\mathrm{HME}} + C_{\mathrm{casing}} + C_{\mathrm{Upper\ tank}} + C_{\mathrm{Lower\ tank}} + C_{\mathrm{piping}} \tag{31}$$

where C denotes the cost. The cost of components is given in Table 1.[10] Hence, the total cost for the evaporative cooling device is given by

$$C_{\text{Total}} = C_{\text{Capital}} + C_{\text{Operating}} \quad (32)$$

The operating cost (running cost) of the cooler is obtained by the power consumption of the fan and water pump. The electricity consumption charges vary for different countries. The electricity charge for all the five different countries (five cities) are listed in Table 2 (petrolprise.com). It is assumed that, on average, coolers are required for 6 months in a year. The operating cost of the device is calculated for eight working hours in a day and 183 days for 6 months (April to June) working days. The STC is defined as the cost per unit cooling energy produced of the device and hence it is given by

$$\text{STC} = \frac{C_{\text{Total}}}{Q_{\text{system}}} \quad (33)$$

Table 1. Cost list of components of dual-mode evaporative cooler.

Component	Unit	Cost (US dollar)
Axial fan	2	$68.51\left(\frac{\text{Power}_{\text{fan}}}{70}\right)^{0.6}$
Water pump	1	$5.1\left(\frac{m_{\text{airflow}}}{1500}\right)^{0.6}$
Heat and mass exchanger	1	52.08
Acrylic casing	1	34.00
Upper tank	1	$7.5\left(\frac{\text{Volume}_{\text{tan k}}}{21}\right)^{0.6}$
Lower tank	1	$8.5\left(\frac{\text{Volume}_{\text{tan k}}}{21}\right)^{0.6}$
Water piping	1	$13\left(\frac{\text{Volume}_{\text{cooler}}}{1.32 * 0.415 * 0.913}\right)^{0.6}$

Experimentation and model validation

A testing unit of the dual-mode evaporative cooler is designed and fabricated in the lab. The schematic diagram of the testing unit is shown in Figure 4. The testing unit consists of preconditioning equipment (air blower, air heater, and humidifier), HME, water tank (upper and lower), fan, and water circulating pump. The core equipment (heat exchanger) consists of two pairs of dry and wet channels. The length and width of both channels have been taken as 0.9 m and 0.26 m, respectively. The channel gap has been taken as 5 mm. The channels are fabricated with a very thin aluminum plate. The wet channels are covered with cotton and connected with the 3 mm diameter tube for water supply. The complete test unit is shown in Figure 5 and details are given elsewhere.[25] The intake air conditions are controlled with the help of preconditioning equipment such as a heater and humidifier. All the measuring instruments have been calibrated before use and all the tests have been repeated 3 times to get reliable data. The repeatability and accuracy of the experiments are ensured by uncertainty analysis. Based on the uncertainties of the measuring parameters, the uncertainties of the evaluated parameters have been estimated. Uncertainties of dry bulb temperature, relative humidity, air velocity, cooling energy produced, dew point effectiveness, and COP are ±1.00%, ±1.92%, ±3.00%, ±3.97%, ±2.61%, and ±4.09%, respectively.

The present simulation model has been validated with the experimental results of the developed test unit. The experimental conditions of the test unit of the evaporative cooler used in the experimentation are water flow rate = 0.5 lpm, extraction ratio = 0.4, intake air velocity = 2 m/s, and water inlet temperature = 17°C. The intake air temperature is varied

Table 2. Electricity cost of countries in US dollar.

ASHRAE climatic zone	Considered city	Country	Electricity prices, 2019
1	Delhi	India	US$0.08/kWh
2	Brisbane	Australia	US$0.25/kWh
2	Brazilia	Brazil	US$0.15/kWh
3	Shanghai	China	US$0.08/kWh
4	Seoul	South Korea	US$0.11/kWh

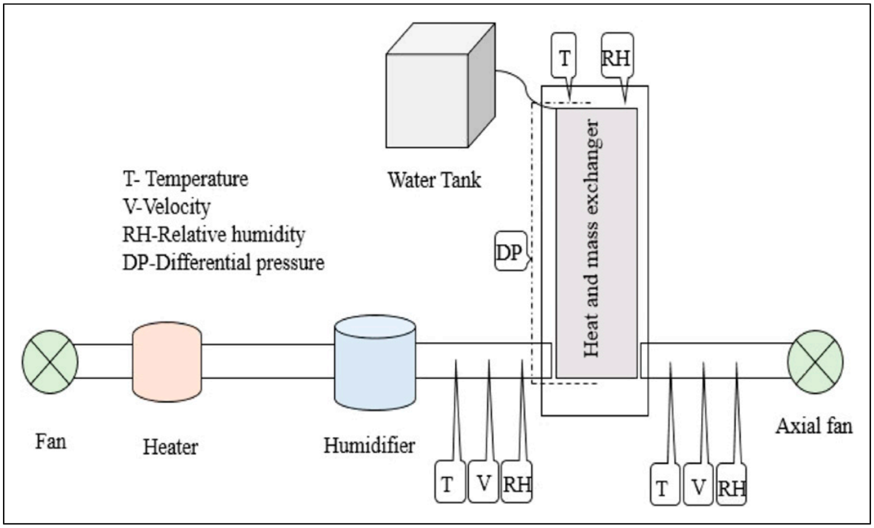

Figure 4. Schematic diagram for experimental setup of dual-mode evaporative cooler.

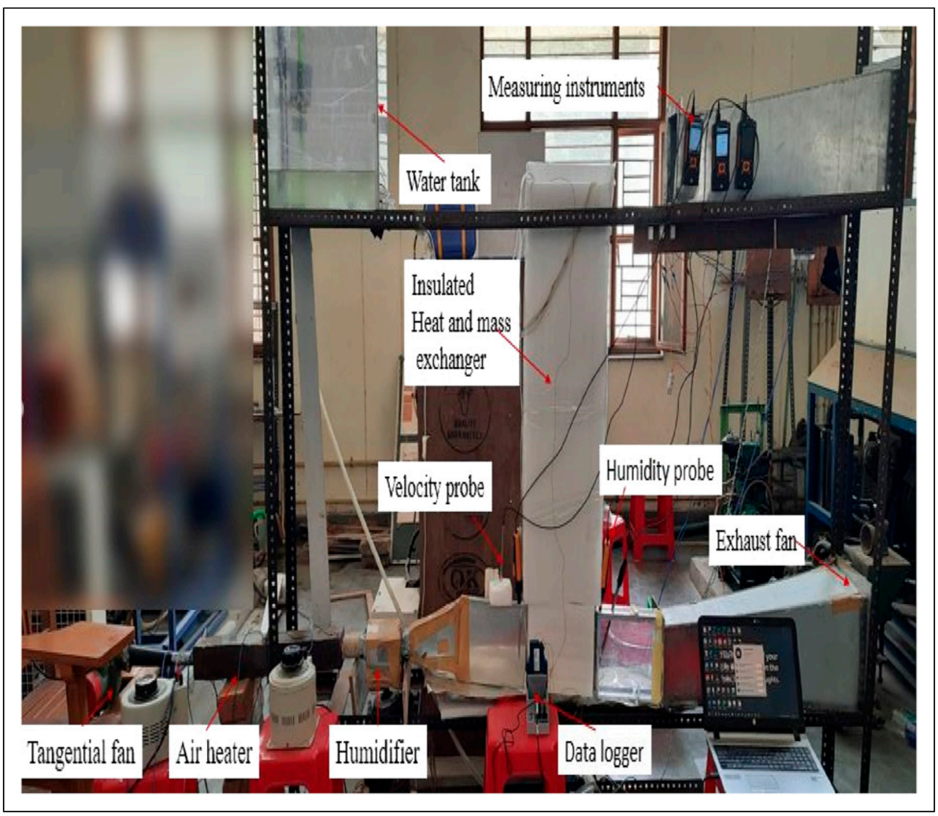

Figure 5. Testing unit of dual-mode evaporative cooler.

from 28.3°C to 46.4°C at the fixed inlet humidity conditions. The two sets of data (for specific humidity of 0.024 kg/kg and 0.010 kg/kg) are recorded for the regenerative mode operation of the cooler testing unit. Figure 6 shows the comparison of the experimental and numerical results. The maximum percentage deviation between the experimental result and the modeled result is 8.67% at the lower intake temperature.

Results and discussion

The performance of the dual-mode evaporative cooler has been assessed for the various ASHRAE climatic conditions. The following geometrical specifications have been used for the dual-mode evaporative cooler: channel length = 1.2 m, the width of the channel = 0.5 m, channel gap = 5 mm (for both wet and dry channels),[26] and the total number of channels = 140 (70 each dry and wet channels). The common operating conditions used are: air inlet velocity = 2 m/s and extraction ratio = 0.33 (as the range of 0.3–0.36 is recommended in the literature[27]). The exergy parameters are calculated by taking an atmospheric air saturation state as a dead state.[9]

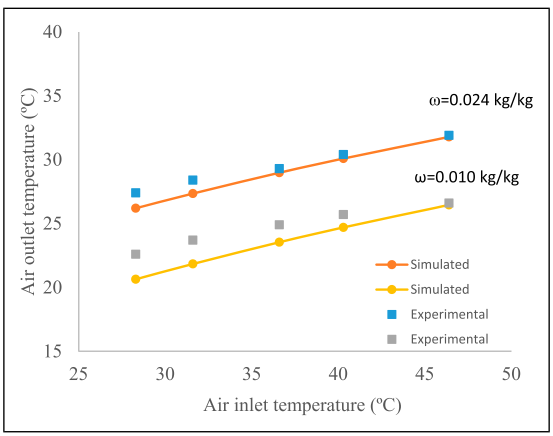

Figure 6. Validation of numerical model with experimental data.

Table 3 compares the results of REC and DEC modes for the air inlet temperature of 35°C and inlet wet bulb temperature of 20°C. The direct mode shows more cooling energy produced and COP as compared to the regenerative mode operation. The regenerative mode shows higher wet bulb effectiveness and exergy efficiency. The pressure drop is more in the regenerative mode, so more fan power is required in REC mode leading to lower COP. The exergy efficiency of REC mode is higher due to less exergy destruction. Figures 1 and 2 also indicate the exergy distribution of the evaporative cooler for REC and DEC modes. The total exergy at a point includes the thermal (due to temperature difference), chemical (humidity difference), and mechanical (due to pressure difference) exergies. The inlet air is at the dead state (outdoor air) temperature, so thermal exergy is zero at the inlet of the device in both modes. The inlet air is not saturated, so it contains only chemical and mechanical exergies. Chemical exergy contribution at the inlet of REC mode is 97.8%, while in DEC mode, it is 98.9%. The chemical exergy

Table 3. Comparisons of results of both modes of evaporative cooler.

Parameters	REC mode	DEC mode
Cooling energy produced (W)	4614	5312
Fan power (W)	46.72	15.26
COP	98.52	251.1
Wet bulb effectiveness	1.127	0.8690
Exergy (W)	127.8 (inlet), 104.5 (supply), 17.0 (exhaust)	120.7 (inlet), 89.83 (supply)
Irreversibility (W)	6.52	31.1
Exergy efficiency	0.7635	0.7138
Vane position	Vane 1 = partially open Vane 1 = fully open	Vane 2 = fully open Vane 2 = fully open

contributes to the maximum percentage of the total exergy. The chemical exergy of the air is sacrificed to obtain thermal exergy. The pressure difference between the inlet and exhaust is higher in REC mode as compared to DEC mode. Hence, the mechanical exergy contribution at the inlet is higher for REC mode (2.1%) as compared to that for DEC mode (1.05%), which causes higher air inlet exergy for REC mode.

ASHRAE provides climatic design data from different stations across the world.[17] There are two kinds of data: one is for annual cooling, dehumidification, and enthalpy design conditions and the other for monthly climatic design conditions. Annual evaporation (wet bulb and mean coincident dry bulb temperature) data are recommended for the design of evaporative coolers. There are eight international climatic zones, classified by ASHRAE, out of which four zones are selected for analysis. The five cities, Delhi (India) from the very hot–humid zone (Zone number-1), Brisbane (Australia) and Brasilia (Brazil) from the hot–humid zone (Zone number-2), Shanghai (China) from the warm–humid zone (Zone number-3), and Seoul (South Korea) from the mixed humid zone (Zone number-4) are selected for investigations from one to four climatic zones. Month-wise climatic data are listed in Table 4.

The annual climatic design data are used to compare the performance of both modes for all five cities. Cooling energy produced depends on the mass flow rate of air and temperature difference between inlet and outlet. In REC mode, the mass flow rate is reduced due to the extracted air, but it produces a lower outlet temperature than that of DEC mode. The cooling energy produced of DEC mode is more as compared to REC mode for all the five different cities of the international climatic zones. The cooling energy produced gets reduced when we move from climatic zone 1 to four for both modes (Table 4). Figure 7 compares the wet bulb effectiveness in both modes. The regenerative mode cools air to a temperature between the wet bulb and dew point temperature. Direct mode delivers air in between 80 and 90% of the wet bulb temperature. The highest wet bulb effectiveness in REC mode is obtained for the Shanghai city (1.18), while Delhi (0.84) is in DEC mode. The Brazilia city shows the lowest effectiveness in both modes. There is a possibility to restrict the outlet air up to the desired (comfort zone) relative humidity by governing the other parameters. The five cities averaged effectiveness of the regenerative mode is 30% higher than the direct mode. In climatic zone 3, July and August months show higher cooling energy produced in both the modes. Similar results are also obtained for climatic zone 4, such as lower temperature by REC and higher cooling energy produced by DEC mode (Table 4). It may be noted that the specific cooling capacity of REC mode is higher due to more temperature drop obtained as compared to DEC mode. However, as the supply air is less due to extraction of intake air in REC mode, the cooling capacity in DEC mode is more than that in REC mode for the same intake airflow.

In the direct mode, only one exhaust fan is needed, and pressure drop is reduced. Hence, the COP is much higher for DEC mode as compared to REC mode. The annual power consumption for the dual-mode evaporative cooler is higher than that for single-mode DEC and lower than that for single-mode REC for the same cooling energy produced. Figure 8 compares the COP of the device for both modes of operation. The maximum COP is obtained for the direct mode of operation of the device in climatic zone 1. The COP goes down from climatic zone 1 to 4 for both modes. The dual-mode evaporative cooler yields higher COP in Brisbane as compared to that in Brazilia (both cities belong to the climatic zone 2) for both DEC and REC modes of operation. COP of the device gets reduced by 30% in REC mode and 29% in DEC mode when moving from zone 1 to zone 4. Seoul city shows the lowest among all four climate zones in both REC (26) and DEC (86) modes.

Figure 9 shows the exergy destruction rate of the evaporative cooler for both modes of operation. The climatic zone 1 shows the highest exergy destruction for DEC mode of operation. The exergy destruction is more for the direct mode of operation as compared to REC mode for all the five cities. The mixing of air and water vapor is an irreversible process and irreversibility is more in DEC as compared to REC mode of operation. In REC mode, some exergy is recovered in sensible cooling from wet to dry channel heat

Table 4. Results of dual-mode cooler for five cities of different climatic zones.

			REC				DEC			
	T_{in} (°C)	RH_{in} (%)	T_{out} (°C)	RH_{out} (%)	Q (W)	I (W)	T_{out} (°C)	RH_{out} (%)	Q (W)	I (W)
Delhi										
Apr	41.6	15.05	22.33	44.8	5144	211.6	24.24	77.6	6924	449.1
May	44.1	12.9	22.82	42.4	5634	246.8	25.06	76.5	7533	532.7
June	44.2	15.77	23.77	49.3	5421	214.9	26.12	78.0	7166	464.6
July	40.2	32.32	25.95	71.9	3855	92.02	28.04	84.6	4914	200.8
Aug	37.1	43.07	26.18	80.0	2992	53.83	27.97	87.8	3736	116.5
Sept	36.4	39.92	25.09	76.1	3097	62.98	26.76	86.9	3945	131.7
Brisbane										
Apr	29.4	46.21	21.44	74.1	2220	47.77	22.17	88.6	3013	83.88
May	27.1	37.73	18.79	62.4	2326	65.94	18.93	86.3	3413	113
June	24.7	50.38	18.8	72.2	1667	40.60	18.83	89.8	2476	61.61
July	24.2	39.48	17.22	60.7	1971	59.97	16.89	86.9	3082	96.4
Aug	27.0	33.91	18.22	57.8	2457	75.93	18.24	85.2	3660	131.9
Sep	30.1	26.23	18.86	51.4	3110	106.4	19.19	82.5	4508	198.7
Brazilia										
Apr	30.8	37.22	21.01	66.4	2714	69.48	21.77	86.1	3736	128.1
May	30.0	32.48	19.78	59.7	2833	83.36	20.29	84.6	4021	152.9
June	28.8	26.64	18.21	50.4	2941	102.4	18.32	82.7	4346	187.8
July	29.2	22.69	17.82	45.0	3153	119.8	17.86	81.3	4692	223.7
Aug	32.0	18.18	18.48	40.6	3711	151.4	18.84	79.4	5392	295.4
Sep	33.2	16.18	18.68	38.2	3968	168.6	19.18	78.5	5724	334.5
Shanghai										
Apr	29.2	46.01	21.28	46.0	2210	48.18	21.97	88.6	3014	84.2
May	32.2	33.81	21.32	33.8	3003	80.94	22.23	85.0	4108	153.8
June	35.2	52.83	26.59	52.8	2378	33.32	28.21	90.4	2884	71.47
July	38.0	42.59	26.71	42.5	3087	54.91	28.61	87.7	3835	121.5
Aug	38.0	45.52	27.29	45.5	2933	47.08	29.2	88.5	3596	106.5
Sep	34.1	56.98	26.45	56.9	2120	27.54	27.96	91.4	2541	57.29
Seoul										
Apr	25.5	30.15	16.85	51.2	2426	85.01	16.48	84.0	3780	146.7
May	29.1	32.73	19.29	58.9	2727	81.60	19.65	84.7	3924	147.4
June	31.7	35.47	21.29	65.5	2878	75.20	22.16	85.6	3939	141.4
July	34.0	51.9	25.55	84.3	2339	35.68	27.01	90.2	2888	72.64
Aug	34.3	45.48	24.68	79.2	2654	48.58	26.12	88.5	3370	97.94
Sep	30.5	46.28	22.2	75.5	2310	47.56	23.1	88.7	3077	86.04

exchange. Exergy destruction is directly related to the wet bulb depression.[4] The wet bulb depression is less at the inlet of the wet channel in REC mode and hence there is less destruction of exergy. The cities (Brisbane and Brazilia) of climatic zone 2 yield the highest exergy destruction in REC mode. The city of climatic zone 4 shows the lowest exergy destruction for DEC mode. The exergy destruction decreases from zone 1 to 4 in the direct mode of operation. The exergy destruction rate varies for all months depending upon inlet humidity and temperature conditions, as listed in Table 4.

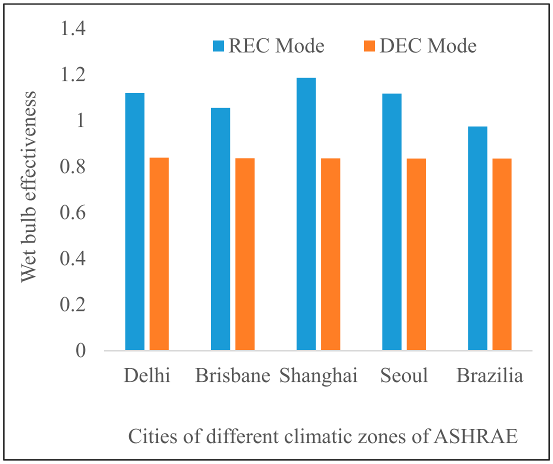

Figure 7. Wet bulb effectiveness of both modes of evaporative cooler.

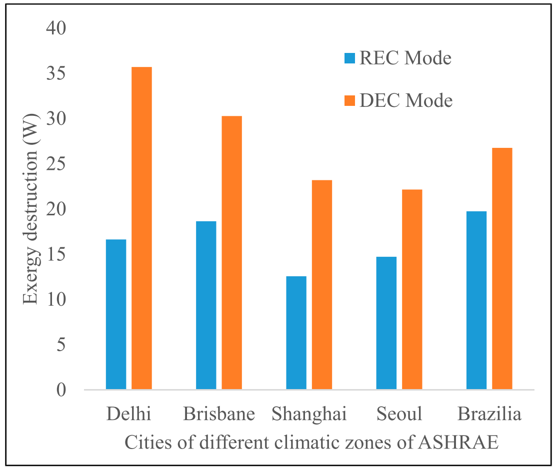

Figure 9. Exergy destruction in both modes of evaporative cooler.

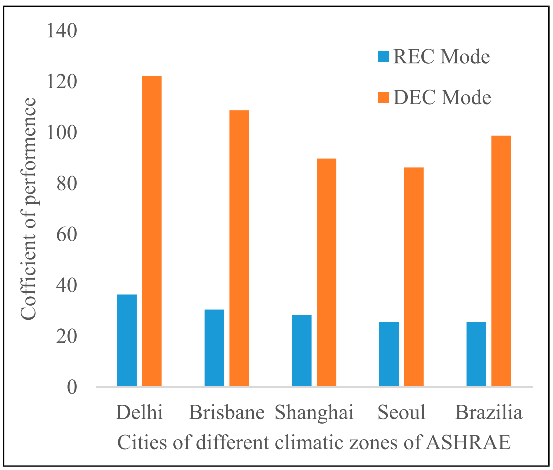

Figure 8. Coefficient of performance of evaporative cooler for both modes.

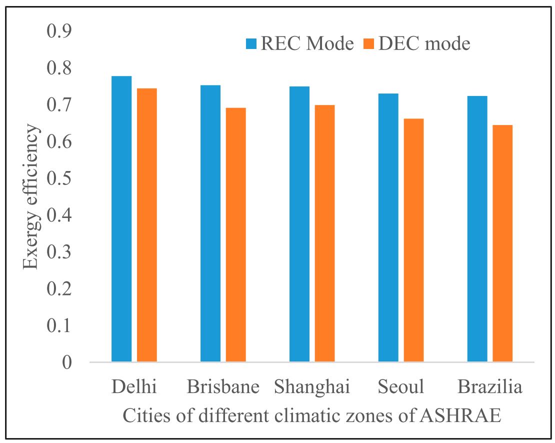

Figure 10. Exergy efficiency in both modes of evaporative cooler.

The exergy efficiency of the regenerative mode is higher than that of the direct mode for all the five considered cities due to higher specific cooling capacity and less exergy destruction in the regenerative mode, as shown in Figure 10. The difference of exergy efficiency between REC and DEC modes is almost constant for all five cities. The averaged exergy efficiency of DEC mode is 5.8% lesser than that of REC mode. Brazilia city shows the lowest exergy efficiency in both modes. The exergy efficiency of direct mode decreases by 8% from zone 1 to 4. The month-wise exergy efficiency of REC mode is presented in Figure 11. The exergy efficiency is highest for Shanghai and lowest for Brisbane in 4 months (June to September). The low exergy destruction rate ensures a high exergy efficiency of the device. The city of zone 1 shows marginal increases in the exergy efficiency from April to September in REC mode. The exergy efficiency of Brisbane and

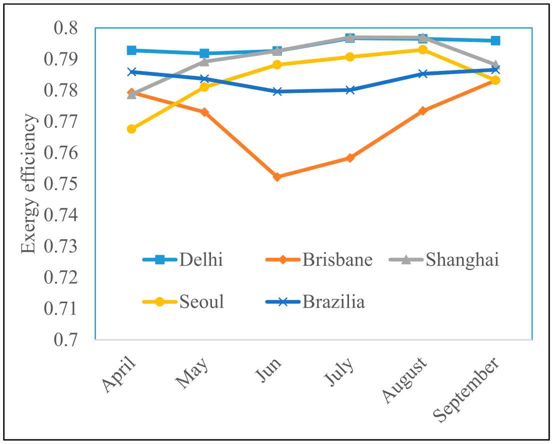

Figure 11. Month-wise variation of exergy efficiency in regenerative evaporative cooler mode.

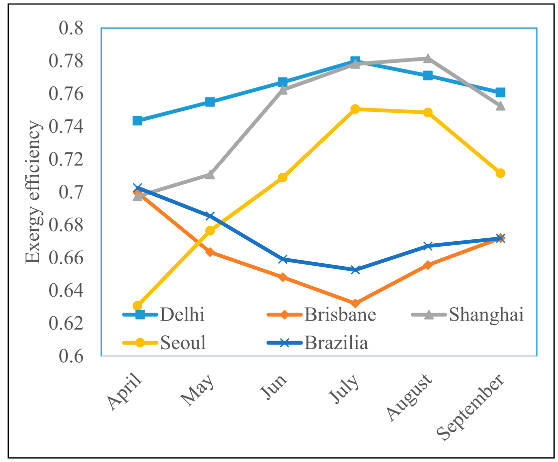

Figure 12. Month-wise variation of exergy efficiency in firect evaporative cooler mode.

Brazilia is lowest in June and July. Delhi shows the highest exergy efficiency in DEC mode (Figure 12) during dry months; however, its value decreases in July, August, and September. The exergy efficiency during July and August is highest for Seoul, while lowest for Brazilia and Brisbane. The climatic zone 4 shows the increase (12%) in exergy efficiency from April to July, which further decreases up to September.

The operating cost of the device consists of the electricity consumption charge of the fan and water pump. The electricity charge varies for different countries. As shown in Figure 13, the running cost of the direct mode is much less as compared to the regenerative mode since only one fan needs to be operated. The operating cost of climatic zone 2 is the highest among all the climatic zones due to high electricity charges. The capital cost of a single-mode DEC device is around US$127, which will not be effective in humid months. The capital cost of single-mode REC is around US$202, which provides low cooling capacity and high specific cost in dry months. The capital cost of a dual-mode cooler is around US$207, which can be operated at the best-suited modes depending upon the humidity conditions. Hence, the combined capital cost of two single-mode devices will be US$329, which is much higher than that of a dual-mode device. If it is assumed that DEC runs for 50% time while REC for

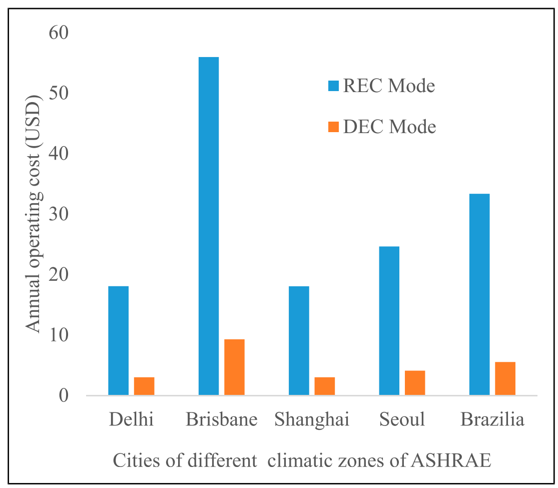

Figure 13. Annual operating cost of evaporative cooler for all climatic zones.

50% time, the annual total costs will be US$339.5 (Delhi), US$361.6 (Brisbane), US$339.5 (Shanghai), US$343.4 (Seoul), and US$348.4 (Brazilia) by using both single-mode devices; whereas, US$217.5 (Delhi), US$239.6 (Brisbane), US$217.5 (Shanghai), US$221.4 (Seoul), and US$226.4 (Brazilia) by using the dual-mode device. The STC of the dual-mode evaporative cooler in the individual modes (DEC and REC) is presented in Figure 14. The STC

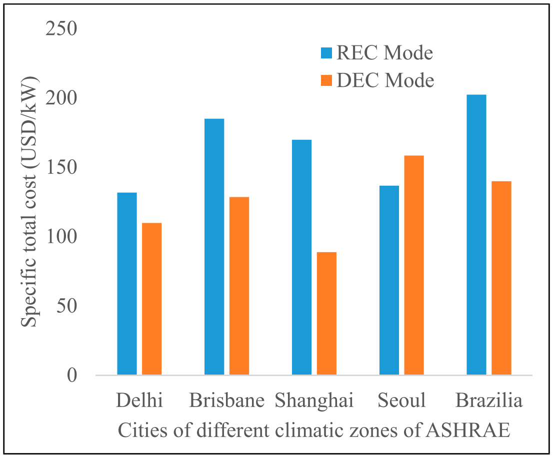

Figure 14. Specific total cost of evaporative cooler for both modes of operation.

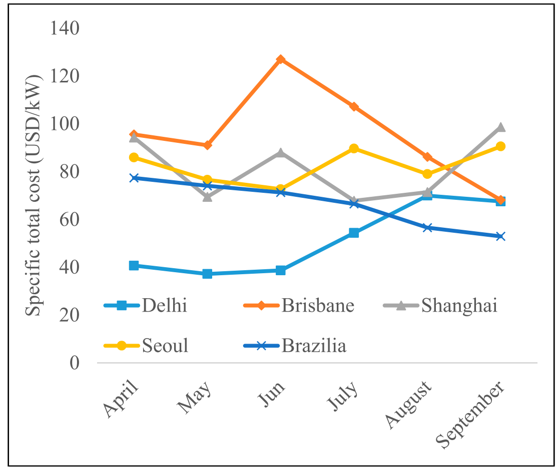

Figure 15. Month-wise variation of specific total cost in regenerative evaporative cooler mode.

is least for climatic zone 1 and highest for climatic zone 4 in both modes of operation. The STC of the REC mode is lower for the climatic zone 1 due to the high cooling energy produced. Seoul city shows a little higher specific cost in direct mode as compared to regenerative mode. A significant difference (around 1.9 times) between the REC and DEC mode-specific cost is obtained for Shanghai city.

The month-wise STC for REC and DEC modes is presented in Figures 15 and 16, respectively. The STC of Brisbane is highest in both modes due to the high operating cost of the device. Delhi city shows a low STC from April to July due to high cooling energy produced and low electricity charges. The STC of DEC mode is less in all zones for all 6 months. The specific cost of Shanghai city keeps decreasing and increasing trend from April to August in both modes. The specific cost for zone 1 increases, while for zone 2, it decreases from April to September. The specific cost for zones 2 and 3 do not show too many fluctuations from April to September. In climatic zone 2 (Brisbane city), REC gives lower temperature and cooling energy produced while DEC shows lower STC. Brazilia city has the lowest total specific cost in the month of August and September.

The single-mode DEC is available in the market and its wet bulb effectiveness range is similar to that of the dual-mode cooler in DEC mode. REC mode

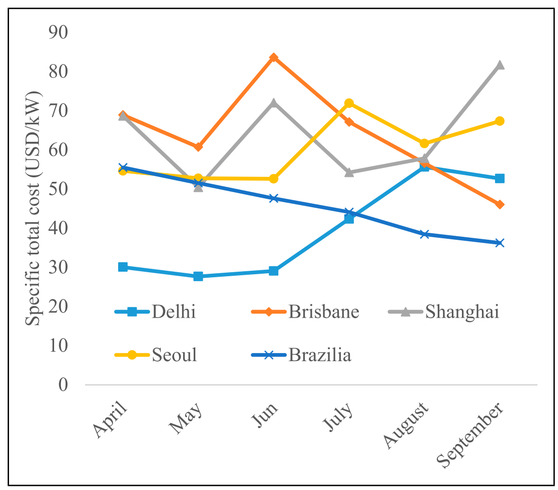

Figure 16. Month-wise variation of specific total cost in direct evaporative cooler mode.

performance of the dual-mode evaporative cooler is similar to the REC presented in the literature, where effectiveness lies closer and above one. The cooling capacity of the dual-mode cooler can be improved by increasing the height (number of plates) of the HME. The dual-mode evaporative cooler is best for composite climate (DEC is suitable for some months and REC for others) and about 40% annual cost saving is possible by using this device instead of two single-mode (DEC and REC) devices. The operating mode selection depends upon the climatic (inlet temperature and inlet humidity) conditions. The wet bulb temperature includes both performances influencing properties (temperature and humidity). The analysis shows that the device should be operated in REC mode if the wet bulb temperature of intake conditions is greater than 24°C. The dual-mode device should be switched to DEC mode below this wet bulb condition.

Conclusions

In this study, energy, exergy, and economic analyses of the dual-mode evaporative cooler are presented for five different international climate zones. Annual as well as monthly performances are investigated based on ASHRAE climatic data for both DEC and REC modes. The experimental setup has been developed and test data have been used for model validation. The dual-mode device is also compared with single-mode devices. From results and discussion, the main conclusions are as follows:

- ➢ The monthly design results show that the cooling energy produced of DEC mode is higher than REC. So, the direct mode is found to be more suitable for high sensible heat removal of the conditioning space.
- ➢ The very hot–humid zone has higher wet bulb depression, which results in higher exergy destruction for DEC mode as compared to REC mode. The exergy efficiency for REC mode is higher than DEC mode and independent of the climatic zone.
- ➢ The annual running cost for the REC mode is always higher than DEC mode. The difference between the running cost of REC and DEC is significantly affected by electricity charges.
- ➢ The STC is more for climate zone 4. The difference in STC between REC and DEC modes is marginal for climatic zone 1 and climatic zone 4; whereas, significant (US$46/kW) for climatic zone 3.
- ➢ REC mode is better in terms of wet bulb effectiveness and exergy efficiency; whereas, DEC mode is better in terms of COP and STC.
- ➢ The dual-mode evaporative cooler can be automated to operate in DEC or REC mode by sensing outdoor conditions (DEC mode can be switched to REC mode if the wet bulb temperature of intake condition is greater than 24°C and vice versa).
- ➢ About 40% annual cost saving is possible by using the dual-mode cooling device instead of two single-mode (DEC and REC) devices for the composite climate.

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Appendix 1

Notation

A Heat transfer area, m^2
c_p Humid air specific heat at constant pressure, kJ/kgK
d_h Hydraulic diameter, m

D_{va}	Water vapor diffusion coefficient, m^2/s
E_x	Exergy, W
h_{fg}	Enthalpy of evaporation, kJ/kg
I	Exergy destruction/irreversibility
k_{paper}	Thermal conductivity of the moisture-retaining paper, W/mK
k_{plate}	Thermal conductivity of the channel separating plate, W/mK
L	Channels length, m
m	Mass flow rate of air, kg/s
Nu	Nussult number
P	Power, W
Q	Cooling energy produced, W
T	Air temperature, °C
T_{wf}	Temperature of water film, °C
t_{plate}	Thickness of plate separating channels, m
t_{paper}	Thickness of moisture-retaining paper, m
U	Overall heat transfer coefficient, W/m^2K
u	Air velocity, m/s
V	Volume flow rate, m^3/s
W	Width of the channel, m
y	Gap in the channels, m

Greek letters

α	Heat transfer coefficient, W/m^2K
α_m	Mass transfer coefficient in the wet channel, kg/ms
η_{ex}	Exergy efficiency
ν	Kinematic viscosity, m^2/s
ω	Humidity ratio, kg/kg
ω_{wf}	Saturation humidity at the water film temperature, kg/kg
ρ	Density, kg/m^3
ε	Wet bulb effectiveness
Ψ	Flow exergy per kg of dry air, J/kg

Subscripts

ad	Dry channel
aw	Wet channel
in	Incoming
out	Outgoing
wf	Water film
0	Reference state