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ORIGINAL ARTICLE

Parametric analysis of a combined dew point evaporative-vapour compression based air conditioning system

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KEYWORDS

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Power consumption;
Effectiveness;
Cooling load;
Conditioned space

Abstract A dew point evaporative-vapour compression based combined air conditioning system for providing good human comfort conditions at a low cost has been proposed in this paper. The proposed system has been parametrically analysed for a wide range of ambient temperatures and specific humidity under some reasonable assumptions. The proposed system has also been compared from the conventional vapour compression air conditioner on the basis of cooling load on the cooling coil working on 100% fresh air assumption. The saving of cooling load on the coil was found to be maximum with a value of 60.93% at 46 °C and 6 g/kg specific humidity, while it was negative for very high humidity of ambient air, which indicates that proposed system is applicable for dry and moderate humid conditions but not for very humid conditions. The system is working well with an average net monthly power saving of 192.31 kW h for hot and dry conditions and 124.38 kW h for hot and moderate humid conditions. Therefore it could be a better alternative for dry and moderate humid climate with a payback period of 7.2 years.

1. Introduction

Air conditioning through evaporative cooling is one of the good alternatives to conventional vapour compression air conditioning, as these systems generally consume less electric power than conventional vapour compression air conditioning systems. Therefore, these systems help to decrease the peak electricity demand and also contribute to decrease the harmful greenhouse gas emissions. Conventional evaporative cooler reduces the process air temperature: approaching the wet bulb temperature of air.

Direct and indirect are the two conventional types of evaporative cooling systems available of which direct evaporative cooling system has generally higher cooling effectiveness as compared to indirect cooling system. However, direct evaporative cooling system is applicable only to dry climates because it increases the moisture in air as shown in Fig. 1(a), which makes the condition uncomfortable due to increased humidity of air in the humid climate. On the other hand indirect evaporative cooling reduces the temperature of process air without adding the moisture into it as shown in Fig. 1(b). However, it is generally having low cooling effectiveness. Therefore, it

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Nomenclature

h	specific enthalpy of air, kJ/kg
Δh	change in enthalpy, kJ/kg
m_a	mass flow rate of air kg/s
Q	cooling load, kW
Q_r	Cooling rate to the conditioned space, kW
s	second
T	Dry bulb temperature, °C
X	bypass factor of the cooling coil
ϕ	relative humidity, %
ε	effectiveness of evaporative cooler
ω	specific humidity of air, kg/kg of dry air

Subscript

ae	condition of air after evaporative cooler
c	cooling coil
cs	conventional system
dew	dew point
i	inside design condition of conditioned space
in	inlet air
o	outside (ambient) condition
ps	proposed system
s	supply condition of air to the room
wb	wet bulb
wk	working air

Acronyms

ADP	apparatus dew point, °C
ANMPS	average net monthly power saving
CCC	conventional cooling coil
CFM	cubic feet per minute
COP	coefficient of performance
DBT	dry bulb temperature, °C
DEC	direct evaporative cooler
DPC	daily power consumption
DPEC	dew point evaporative cooler
DPT	dew point temperature, °C
IEC	indirect evaporative cooler
kW h	kilo watt hour
MPC	monthly power consumption
MPS	monthly power saving
NMPC	net monthly power consumption
NMPS	net monthly power saving
PC	power consumption
TR	tons of refrigeration
VCR	vapour compression refrigeration
WBT	wet bulb temperature, °C

cannot cool the temperature of process air to the human comfort conditions working alone.

A dew point evaporative cooler is a better substitute for the above as it decreases the process air temperature below its wet bulb temperature without adding moisture, approaching the dew point temperature of the process air. However, in real application where the ambient temperature increases beyond 40 °C and reaches up to 45 °C, the dew point evaporative cooler working alone cannot reduce the ambient temperature to human comfort condition at desired cooling rate. Therefore, conventional vapour compression air conditioner can be used in conjunction with the dew point evaporative cooler, which will change the ambient air to human comfort condition at desired cooling rate by consuming comparatively less electric power: as partial cooling load is shared by the dew point evaporative cooler.

Kim and Jeong [1] studied the energy performance of an indirect and direct evaporative cooler assisted 100% outdoor air system (IDECOAS). Results revealed that the IDECOAS operating in the two-stage mode in the intermediate season shows a 51% energy saving over the conventional variable air valve (VAV) system. However, the proposed system may consume 36% more operating energy than the conventional VAV system during the cooling season due to limited cooling performance of the indirect evaporative cooler (IEC) in hot and humid climate. Jain et al. [2] studied the financial feasibility of a hybrid direct evaporative cooler (DEC) combined with an air conditioning (AC) unit to reduce the annual expenditure on electricity usage (as against standalone AC unit to provide almost similar level of comfort). They considered four different building applications located in four different places of India. The hybrid mode operation is found financially attractive for movie theatres and waiting hall building applications for all the climatic conditions considered in the study.

Delfani et al. [3] investigated the performance of indirect evaporative cooling (IEC) system to pre-cool air for a conventional mechanical cooling system for four different locations of Iran. A combined experimental setup of an IEC unit and a packaged unit air conditioner (PUA) was designed, constructed and tested. The performance and energy reduction capability of combined system have been evaluated through an analytical method for the cooling season. The result reveals that IEC can reduce cooling load up to 75% with 55% reduction in electrical energy consumption of PUA during cooling season. Cui et al. [4] presented a hybrid system that combines indirect evaporative cooler (IEC) system and vapour compression system. The exhaust air from the conditioned room is used as the working air for IEC and outdoor fresh air is used as the product air, as a result of which the IEC unit produces pre-cooled air for vapour compression system. Two types of IEC units, namely a conventional counter flow IEC unit and a novel counter flow IEC unit based on Maisotsenko (M-cycle), have been numerically analysed. Results reveal that the humid outdoor fresh air can be pre-cooled to a temperature below its dew point temperature when the wet bulb temperature of the exhaust air is lower than the dew point temperature of the outdoor air. Wang et al. [5] investigated the Coefficient of Performance (COP)'s augmentation of an air conditioning system using an evaporative cooling condenser. The experimental setup consisted of four major components namely compressor, evaporator, thermal expansion valve, and the condenser. An evaporative cooling unit was located upstream from the condenser. Thermal parameters, such as dry bulb temperature, wet bulb temperature and rela-

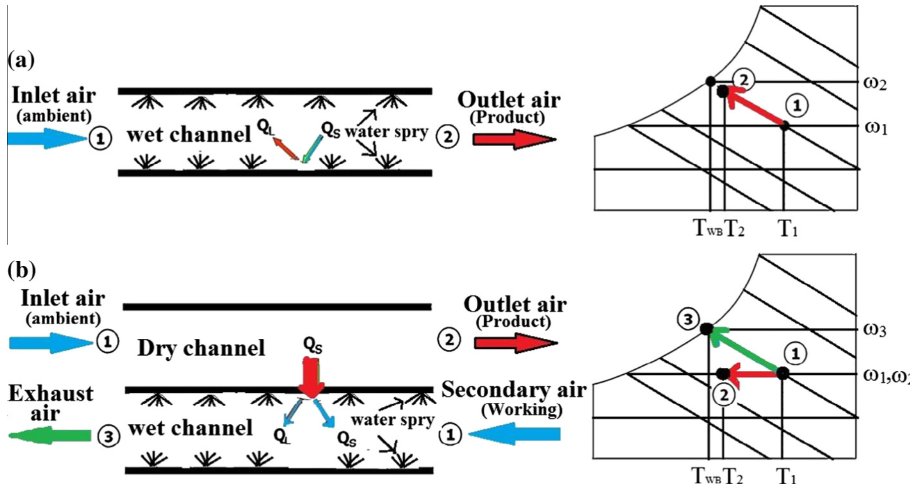

Figure 1 (a) Direct evaporative cooling. (b) Indirect evaporative cooling.

tive humidity were measured to find out the effect of in-direct evaporative cooling on the system's COP.

Chauhan and Rajput [6] proposed an evaporative-vapour compression based combined air conditioning system for providing good human comfort conditions at a low cost working under hot and dry climate. They also compared the system on the basis of saving (%) in cooling load on the cooling coil for the same sensible cooling rate to the conditioned space from the conventional vapour compression air conditioner working on 100% fresh air assumption. The saving of cooling load on the coil was found to be maximum with a value of 64.19% in the month of March due to lower outside temperature and it is minimum for the month of May with a value of 27.36% due to higher outside temperature. The proposed system worked well for hot and dry climate with a net power saving of 646.8 kW h from March to May for a small capacity application. Riangvilaikul and Kumar [7] studied a novel dew point evaporative cooling system for sensible cooling of the ventilation air for air conditioning application. The outlet air condition and the system effectiveness at different inlet air conditions were determined by conducting the experiment. The key results indicated that wet bulb effectiveness spanned between 92% and 114% and the dew point effectiveness between 58% and 84%. Frank Bruno [8] conducted an experiment which takes the advantage of evaporative cooling to reduce the temperature of air without the addition of moisture. They also presented the results obtained from testing a prototype cooler installed in both commercial and residential applications for a wide range of ambient conditions. The performance characteristic of the indirect evaporative cooler with regard to its outlet temperatures and electrical energy efficiency is presented. Riangvilaikul and Kumar [9] presented a paper on theoretical performance of a novel dew point evaporative cooling system which operates under various inlet air conditions and influence of major operating parameters. Heat and mass transfer processes in a dew point evaporative cooler have been simulated through the model. The results predicted by the model using numerical method have been validated with the experimental findings and with the recent literature.

Anisimov and Pandelidis [10] developed a numerical model based on the modified e-NTU method to perform thermal calculations of the indirect evaporative cooling process. The model was validated against experimental data available from the literature. Results showed satisfactory agreement with the result of experimental measurements. The results of computer simulation showed high efficiency gains that are sensitive to various inlet conditions, and allow for estimation of optimum operating conditions. Anisimov et al. [11] tested a novel cross-flow HMX (heat and mass exchanger) which uses the M-cycle (Maisotsenko cycle) for dew point indirect evaporative cooling for evaluating the performance in terms of thermal effectiveness and specific cooling capacity under various operating conditions. Apart from this, the operational performance of the investigated HMX was also examined on the basis of developed model. The results obtained from the model have been compared with the experimental data. The analysis shows attractiveness and high efficiency of the novel M-cycle HMX used for indirect evaporative cooling in air conditioning units. Pandelidis et al. [12] investigated a mathematical model of the heat and mass transfer in the two different Maisotsenko Cycle (M-Cycle) heat and mass exchangers used for the indirect evaporative cooling in different air-conditioning systems. A mathematical model has been developed to perform the thermal calculations of the indirect evaporative cooling process. The model was also validated against the experimental data.

Cui et al. [13] theoretically investigated the performance of a novel dew-point evaporative cooler. The evaporative cooler, based on a counter-flow closed-loop configuration, is able to cool air to temperature below ambient wet bulb temperature and approaching dew-point temperature. A computational model for the cooler has been developed and validated by comparing the temperature distribution and outlet air conditions against experimental data from the literature. Simulated results showed that the wet bulb effectiveness ranged from 122% to 132% while dew-point effectiveness spanned 81%–93%. Caliskan et al. [14] presented the energy and exergy analyses and sustainability assessment of the novel evaporative air cooling system based on Maisotsenko cycle which allows the product fluid to be cooled to a dew point temperature of the incoming air. The exergy input, exergy output, specific flow exergy, exergy destruction, exergy loss, exergy efficiency, exergetic COP, primary exergy ratio and entropy generation rates are determined for various cases. Key results reveal that maximum exergy efficiency is found to be 19.14% for a reference temperature of 23.88 °C where the optimum operation takes place. Zhan et al. [15] presented a performance based comparative study of cross-flow and counter-flow M-cycle heat exchangers for dew point cooling. Cross-flow and counter-flow heat exchangers were theoretically and experimentally investigated to know the difference in cooling effectiveness of both under the parallel structural/operational conditions. The results indicate that the counter-flow exchanger offered greater (around 20% higher) cooling capacity, as well as greater (15–23% higher) dew-point and wet-bulb effectiveness when equal in physical size and under the same operating conditions.

Cui et al. [16] presented simulation results on a novel dew-point evaporative air conditioner which was designed based on a counter-flow closed-loop configuration consisting of separated working channels and product channels. To investigate the performance of the evaporative air cooler under a variety of conditions, the Eulerian–Lagrangian computational fluid dynamics (CFD) model was adopted. They also validated the model by comparing the temperature distributions and outlet air conditions against experimental data. Simulation results have indicated that the novel dew-point evaporative air conditioner achieves a higher wet-bulb and dew-point effectiveness with lower air velocity, smaller channel height, larger length-to-height ratio, and lower product-to-working air flow ratio. Kulkarni and Rajput [17] studied about the theoretical performance analysis of cooling pads of different materials for evaporative cooler. The material has been considered, rigid cellulose, corrugated paper, corrugated high density polythene and aspen fibre. Result shows that the saturation efficiency decreases with increasing mass flow rate of air. It has also concluded that material with higher wetted surface area gives higher saturation efficiency. Camargo et al. [18] studied the direct evaporative cooler operating during summer in a Brazilian city. A mathematical model for direct evaporative cooling system has been developed and presented the experimental results of the tests performed in a direct evaporative cooler which took place in the air conditioning laboratory at the University of Taubate Mechanical Engineering Department, Brazil. Nada et al. [19] theoretically investigated the performance of proposed integrative air-conditioning (A/C) and humidification–dehumidification desalination (HDD) systems for the purpose of energy saving of the air conditioning system and at the same time utilizing the system in fresh water production for the large capacity air conditioning systems. Cianfrini et al. [20] proposed an integrated energy-recovery system basically consisting of indirect evaporative cooling equipment combined with a cooling/reheating unit to reduce the energy demand of air-conditioning installations.

Above literature review reveals that no work has been done as proposed in this paper to determine the performance of the combined system for a wide range of ambient temperature and specific humidity for 100% fresh air to conditioned space. However, some researchers have tried to reduce the power consumption of mechanical cooling coil by incorporating the direct and indirect evaporative cooler along with mechanical cooling coil as shown in Table 1.

Looking to the above, a system has been proposed in this paper which is the combination of dew point evaporative cooler and conventional vapour compression air conditioner. This paper presents the theoretical analysis of a proposed system with a key objective to reduce the temperature of air without adding moisture which creates the human comfort condition at a faster rate with less electric power consumption as compared to the conventional vapour compression air conditioning system.

2. System description

2.1. Dew point evaporative cooler

The process of dew point evaporative cooler is shown in Fig. 2 (a) and (b). The ambient air (1) is drawn into the dry channel where it gives sensible heat to the wet channel as shown in psychometric process 1–2. Consequently, the outlet air (2) leaving the dry channel is at a lower temperature than the ambient. It would be beneficial to supply some fraction of this air in the wet channel to act as the working air which will further reduce the temperature of air in the dry channel. The working air is humidified and absorbs the heat from the dry channel as shown in process 2–3. This hot and humid air is rejected to atmosphere. Thus, the outlet air temperature is reduced below the ambient wet bulb temperature without any moisture addition or removal. The outlet temperature of the air can be reduced theoretically approaching the dew point temperature of intake air in an ideal process.

The relationship between the outlet air temperature and other parameters can be derived by taking enthalpy difference of incoming and outgoing streams to system as shown in Fig. 2 (a).

$$m_{in}(h_1 - h_2) = m_{wk}(h_3 - h_2) \quad (1)$$

The enthalpy change is only because of sensible cooling to the process air in the dry channel. Thus, the enthalpy difference between point 1 and 2 in psychometric process of Fig. 2 (b) can be written as follows:

$$(h_1 - h_2) = C_{pm}(t_1 - t_2) \quad (2)$$

Now by solving Eqs. (1) and (2) for finding the outlet air temperature

$$t_2 = [t_1 - r(h_3 - h_2)/C_{pm}] \quad (3)$$

where r is the mass ratio of working air to intake air, C_{pm} is the specific heat of moist air, m_{wk} is the mass flow rate of working air and m_{in} is the mass flow rate of inlet air.

The inlet and outlet conditions of the process air can be used to evaluate the performance of dew point evaporative cooler using two indices, namely wet bulb and dew point effectiveness. The wet bulb effectiveness is defined as the ratio of the difference between inlet and outlet air temperature to the difference between inlet air temperature and its corresponding wet bulb temperature. The wet bulb effectiveness is expressed as follows:

$$\varepsilon_{wb} = (T_{a,in} - T_{a,out})/(T_{a,in} - T_{wb,in}) \quad [7] \quad (4)$$

Similarly, the dew point effectiveness is defined as the ratio of the difference between inlet and outlet air temperature to the difference between inlet air temperature and its corresponding dew point temperature. The dew point effectiveness is expressed as follows:

$$\varepsilon_{dew} = (T_{a,in} - T_{a,out})/(T_{a,in} - T_{dew,in}) \quad [7] \quad (5)$$

Table 1 Summary of work done related to the proposed work in the literature.

Sr. No.	Author	Cooling technique	Applicable climate	Description	Findings
1	[1]	IEC + CCC	Hot and humid	Used in series for 100% fresh air to conditioned space.	Effectiveness varies for IEC from a range of 31.1% to 44.3%, while it consumes 36% more energy than conventional system because of limited sensible cooling at IEC.
		IEC + DEC	Hot and dry	Used in series for 100% fresh air to conditioned space.	Effectiveness varies for IEC + DEC from a range of 83.4% to 97.8%, while it saves 51% operational energy as compared to conventional system.
2	[2]	DEC/CCC	All seasons	Either DEC or CCC: as per the requirement with air recirculation.	Cost analysis has been performed for the four cities in India with a maximum payback period of 6.0 years at Indore and minimum of 3.6 years at Akola.
3	[3]	IEC + CCC	Hot and dry and hot and humid	Used in series with partial air recirculation.	IEC can provide about 75% of cooling load during cooling season. Also about 55% saving in electrical energy consumption can be obtained in proposed system.
4	[4]	IEC & CCC	Hot and humid	IEC is used for pre cooling of supply air through the exhaust air. Used for partial air circulation.	Computational model has been developed and validated against experimental data from the literature.
5	[5]	DEC & CCC	Hot and dry	DEC is used for cooling the condenser of CCC for full air recirculation.	DEC condenser increases saturation temperature drop through the condenser from 2.4 °C to 6.6 °C which resulted in the increase in COP from 6.1% to 18%.
6	[6]	DEC + CCC	Hot and dry	Used in series for 100% fresh air to conditioned space in a single unit with same fan or blower for air flow.	Combined system worked well with a net power saving of 646.8 kW h from March to May with a payback period of 6.6 years.
7	Present study	DPEC + CCC	Hot and dry, hot and moderate humid	Used in series for 100% fresh air to conditioned space with two different units.	Proposed system worked well with a maximum net monthly power saving of 240.28 kW h at 46 °C and 6 g/kg specific humidity of ambient air with a payback period of 7.2 years.

Abbreviation: IEC: indirect evaporative cooler, DEC: direct evaporative cooler, CCC: conventional cooling coil, DPEC: dew point evaporative cooler, COP: coefficient of performance, kW h: kilo watt hour.

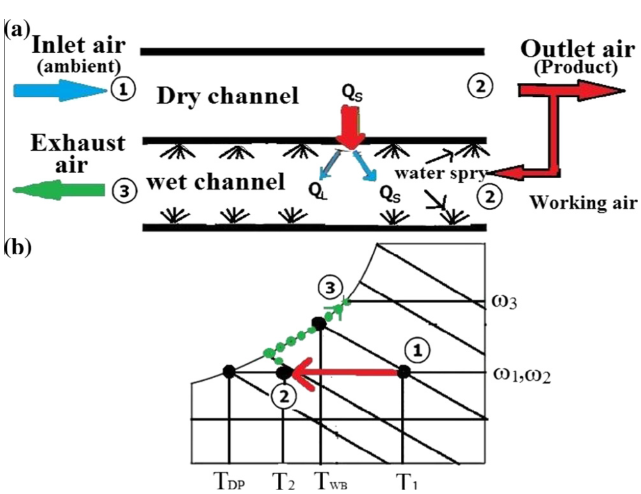

Figure 2 (a) Dew point evaporative cooling. (b) Psychometric representation of dew point evaporative cooling.

The components of the dew point evaporative cooler comprised of many dry and wet channels separated by polymer sheets to avoid penetration of water and the water feeding system. The water feeding should saturate all surfaces only in the wet channels by using a vertical configuration as shown in Fig. 3. By this method, the water could be easily supplied from the reservoir to the outlet portion of wet channels at the top side and travels vertically down to saturate the wet surfaces continuously. The supplied water can be regulated precisely using a control valve. One fan is used to transport the intake and working air for the cooling system.

2.2. Combined system

The ambient air from the outside is first sensibly cooled in a dew point evaporative cooler as shown in Fig. 4, where it decreases its sensible heat to the working air channel. The process occurs in such a way that the specific humidity of the air before the evaporative cooler is the same as specific humidity of the air after the evaporative cooler as shown in Fig. 5. This sensibly cooled air when blows across the cooling coil further

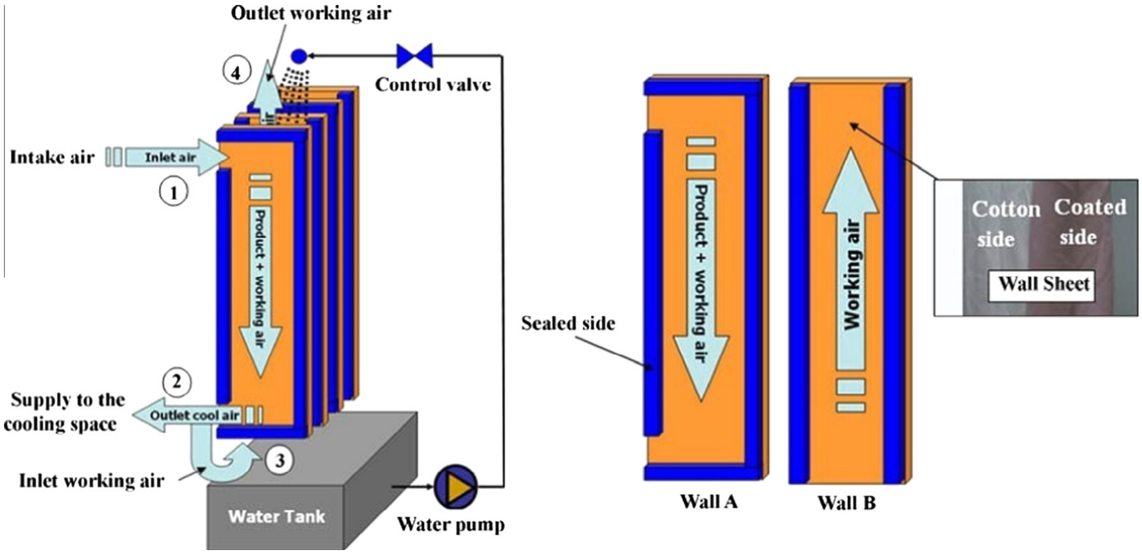

Figure 3 Dew point evaporative cooler adapted from [7].

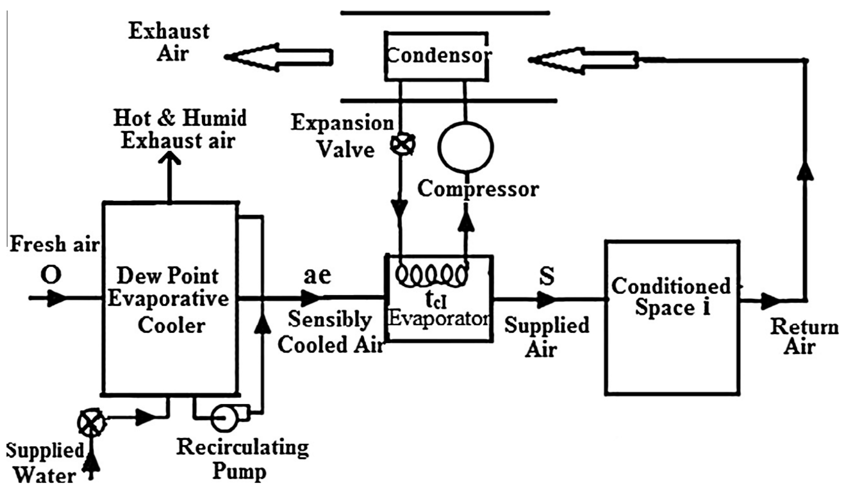

Figure 4 Proposed system for 100% fresh air to conditioned space.

reduces its temperature to sufficiently low value, which is capable of producing the human comfort conditions at a faster rate with less cooling load on the cooling coil as compared to the conventional vapour compression air conditioner as shown in Fig. 6. This air passing through the cooling coil is supplied to the conditioned space, where cooling is required. The fresh ambient air from the outside is taken again by the dew point evaporative cooler, while the exhaust air from the conditioned space is utilized in cooling the condenser of vapour compression air conditioner which further increases the performance of the combined system and this way the process is repeated.

3. Parametric analysis

3.1. Assumptions

- The mass flow rate of air in conventional system is taken as 0.28 kg/s [21].
- Inside design condition is taken as 25 °C DBT and 50% ϕ.
- The wet bulb effectiveness (ε_{wb}) of a dew point evaporative cooler is taken as 1 (average value) [7] irrespective of water.
- Bypass factor of the cooling coil is taken as 0.2.
- Apparatus dew point (ADP) of the cooling coil is taken as 2 °C for the calculation.
- The electric power consumption for fan and circulating pump in case of dew point evaporative cooler is taken as 300 W [18].
- 100% fresh air is supplied to the room: no recirculation of air.
- Cooling load calculation is done after achieving the steady state of the conditioned space.

3.2. Equations

- Temperature of air after evaporative cooler in proposed system:

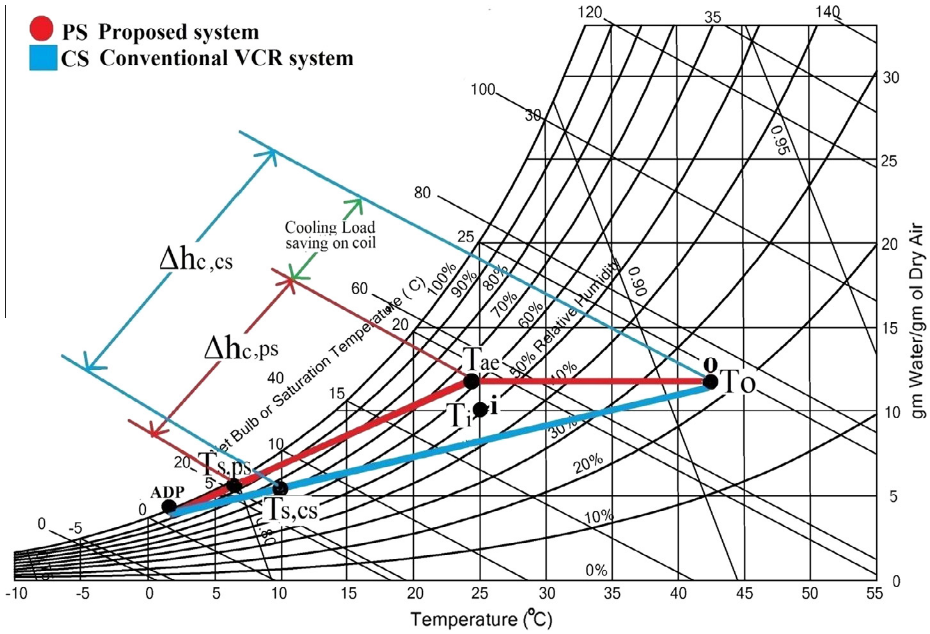

Figure 5 Psychometric representation of cooling process in the proposed and in conventional VCR system.

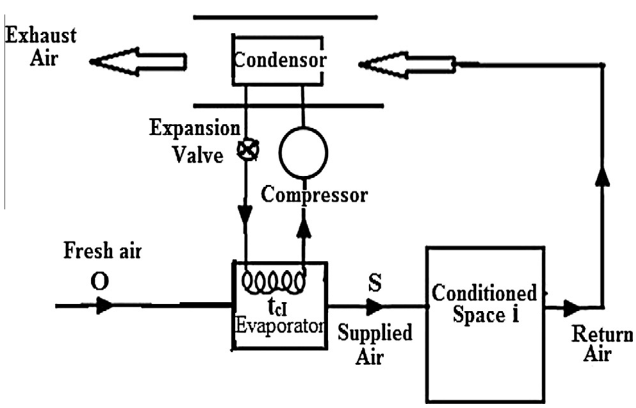

Figure 6 Conventional VCR system for 100% fresh air to conditioned space.

$$T_{ae} = [T_o - \{(\varepsilon_{wb}/100) \times (T_o - T_{wb,o})\}] \quad [22] \tag{6}$$

- Temperature of air supplied to conditioned space in proposed system:

$$T_{s,ps} = [\mathrm{ADP} + \{X \times (T_{ae} - \mathrm{ADP})\}] \quad [22] \tag{7}$$

- Specific enthalpy of air supplied to the space in PS:

$$h_{s,ps} = [h_{\mathrm{ADP}} + \{X \times (h_{ae} - h_{\mathrm{ADP}})\}] \quad [22] \tag{8}$$

- Cooling load on the cooling coil in PS:

$$Q_{c,ps} = m_a \times (h_{ae} - h_{s,ps}) \quad [22] \tag{9}$$

- Cooling rate to the conditioned space in PS:

$$Q_{r,ps} = m_a \times (h_{s,ps} - h_i) \quad [22] \tag{10}$$

where h_i is same in both the cases

- Temperature of air supplied to conditioned space in conventional system:

$$T_{s,cs} = [\mathrm{ADP} + \{X \times (T_o - \mathrm{ADP})\}] \tag{11}$$

- Specific enthalpy of air supplied to the space in CS:

$$h_{s,cs} = [h_{\mathrm{ADP}} + \{X \times (h_o - h_{\mathrm{ADP}})\}] \tag{12}$$

- Cooling Load on the cooling coil in CS:

$$Q_{c,cs} = m_a \times (h_o - h_{s,cs})$$

$$: \text{ where } h_o \text{ is same in both cases} \tag{13}$$

- Cooling rate to the conditioned space in CS:

$$Q_{r,cs} = m_a \times (h_{s,cs} - h_i) \tag{14}$$

- Saving of cooling load on the cooling coil for same cooling rate:
- Saving in

$$Q_c(\%) = [\{(Q_{c,cs} - Q_{c,ps})/Q_{c,cs}\} \times 100] \tag{15}$$

3.3. Methodology

The parametric analysis of the proposed system is done for wide range of temperature and specific humidity as shown in Table 2.

Calculation has been performed on the excel solver for different values of temperature taken from the Table 2 at a constant specific humidity for each value of temperature. Proposed system has also been compared from the conventional system on the basis of cooling load on the cooling coil for the same cooling rate to the conditioned space. The energy saving potential and cost analysis have also been done for the proposed system.

3.4. Calculation procedure

The mass flow rate of air and coefficient of performance (COP) are taken as 500 CFM (0.28 kg/s) and 3.0 respectively from the

Table 2 Operating parameters for the analysis.

Sr. No.	Parameter	Value
1	Ambient temperature T_o (°C)	30 °C, 34 °C, 38 °C, 42 °C, 46 °C
2	Specific humidity ω_o (g/kg)	6, 10, 14, 18, 22
3	ADP	2 °C
4	COP of VCR system	3
5	Pump and fan power	300 W
6	Coil bypass factor (X)	0.2
7	Wet bulb effectiveness ε_{wb}	1
8	Mass flow rate of air m_a (kg/s)	0.28

source [21] for a 1.5 tonne of refrigeration (TR) window air conditioner. The cooling load on the cooling coil is determined by multiplying the mass flow rate with the change in enthalpy across the cooling coil for both the systems. The two systems have been compared on the basis of same cooling rate to the condition space for a given outside condition for 100% fresh air to the conditioned space as shown in Table 3. The load on the each cooling coil is divided by coefficient of performance (COP) in order to get power consumption by the compressor and finally it is multiplied by the actual hours of compressor operation in order to determine the kW h for energy saving potential and cost analysis.

4. Energy saving potential

- The energy saving of the proposed system as shown in Table 3 has been calculated for small office working from 10:00 AM to 06:00 PM (8 h cooling duration).
- Assume the actual operation time of compressor is 75 % [21] of the total cooling duration.
- Power consumption by the compressor in proposed system: $PC_{ps} = Q_{c,ps}/COP$ in kW.
- Power consumption by the compressor in conventional system: $PC_{cs} = Q_{c,cs}/COP$ in kW.
- Daily power consumption $(DPC) = PC \times$ Actual workinghours of compressor (6 h).
- Monthly power consumption $(MPC) = DPC \times 30$(Days in a month) in kW h.
- Net monthly power consumption $(NMPC_{ps}) = [(MPC_{ps} \times 0.85) + 54\text{ kW h}]$ considering the 15% power reduction due to heat recovery in cooling the condenser [5] and 300 W fan and pump power for dew point evaporative cooler [18].
- Net monthly power consumption $(NMPC_{cs}) = [(MPC_{cs} \times 0.85)]$.
- Net Monthly power saving in the proposed system.
- $(NMPS) = NMPC_{cs} - NMPC_{ps}$ in kW h.

5. Cost analysis

The cost analysis has been done with reference to the Bhopal, India. The temperature and specific humidity for ambient condition have been taken as the average of daily maximum temperature and humidity for hot and dry and for hot and moderate humid season from the source [23]. The ambient temperature and specific humidity are shown in Table 4.

Cost of 1 unit of electricity (1 kW h) in Bhopal, India = 7 Rupees.

Table 3 Energy saving potential for 100 % fresh air to condition space.

Sr. No.	t_o (C)	ω (kg/kg)	$Q_{c,ps}$ (kW)	$Q_{c,cs}$ (kW)	Q_r (kW)	DPC_{ps} (kW h)	DPC_{cs} (kW h)	$NMPC_{ps}$ (kW h)	$NMPC_{cs}$ (kW h)	NMPS (kW h)
1	30	0.006	3.86	7.12	9.31	7.73	14.2	251.1	363.63	112.45
2	34	0.006	4.17	8.04	9.24	8.35	16.0	266.9	410.24	143.29
3	38	0.006	4.46	8.95	9.16	8.93	17.9	281.9	456.85	174.94
4	42	0.006	4.74	9.87	9.09	9.49	19.7	296.1	503.46	207.27
5	46	0.006	5.01	10.7	9.03	10.0	21.5	309.7	550.07	240.28
6	30	0.01	6.85	9.42	8.57	13.7	18.8	403.5	480.49	76.919
7	34	0.01	7.12	10.3	8.50	14.2	20.6	417.6	527.45	109.82
8	38	0.01	7.39	11.2	8.49	14.7	22.5	431.1	574.4	143.29
9	42	0.01	7.64	12.1	8.37	15.2	24.3	444.0	621.35	177.34
10	46	0.01	7.88	13.1	8.31	15.7	26.2	456.3	668.3	211.95
11	30	0.014	10.0	11.7	8.11	20.1	23.4	567.9	597.36	29.424
12	34	0.014	10.2	12.6	7.98	20.5	25.2	577.3	644.54	67.18
13	38	0.014	10.4	13.5	7.85	20.8	27.1	586.3	691.84	105.48
14	42	0.014	10.6	14.4	7.73	21.2	28.9	594.6	739.13	144.49
15	46	0.014	10.7	15.4	7.62	21.5	30.8	603.2	786.54	183.33
16	30	0.018	13.3	14.3	7.81	26.6	28.6	732.5	730.56	−2.021
17	34	0.018	13.4	14.9	7.69	26.9	29.9	741.4	764.41	22.969
18	38	0.018	13.6	15.8	7.58	27.2	31.7	749.5	809.39	59.812
19	42	0.018	13.7	16.8	7.47	27.5	33.6	757.5	857.03	99.453
20	46	0.018	13.9	17.7	7.36	27.8	35.4	765	904.67	139.67
21	30	0.022	16.6	17.1	7.54	33.2	34.3	901.9	876.65	−24.44
22	34	0.022	16.7	17.8	7.43	33.5	35.7	909.0	910.92	1.8348
23	38	0.022	16.9	18.5	7.32	33.8	37.0	916.8	944.19	27.396
24	42	0.022	17.0	19.1	7.22	34.1	38.3	924.0	977.04	52.957
25	46	0.022	17.1	20.0	7.12	34.3	40.1	931.0	1022.9	91.827

Table 4 Ambient temperature and Specific humidity for Bhopal [23].

Sr. No.	t_o (C)	ω (kg/kg)	Climate	Applicable month	ANMPS (kW h)	Total power saving during season (kW h)
1	38	0.009	Hot and dry	March–May	151.20	$151.20 \times 3 = 453.60$
2	34	0.014	Hot and moderate humid	Sept–Nov	67.18	$67.18 \times 3 = 201.54$
3	Total yearly power saving in proposed system					$453.60 + 201.54 = 655.14$

Cost of 1.5 TR window air conditioner after discount = 25,000 Rupees [21].
Cost of medium size dew point evaporative cooler and accessories = 8000 Rupees (from an Indian market).
Cost of water = 0 Rupees (Freely available in India).
Total cost of proposed system = 33,000 Rupees.
Net saving of money for hot and dry, and for hot and moderate humid season from March to May and from Sept to Nov per year $= 655.14 \times 7 = 4585.98$ Rupees per year.
Payback period $= 33,000/4585.98 = 7.20$ years.

6. Results and discussion

Variation of saving in cooling load Q_c (%) on the cooling coil in the proposed system as compared to the conventional system for the same cooling rate (Q_r) to the conditioned space with respect to ambient temperature and specific humidity has been shown in Fig. 7. Significant saving in the load on the cooling coil can be seen from Fig. 7. Load saving on the cooling coil is increasing with increasing the temperature while it is decreasing with increasing the specific humidity of ambient air due to increase in the specific enthalpy of ambient air. Load saving is maximum with a value of 60.93% at 6 g/kg specific humidity and 46 °C temperature while it is minimum with a value of 8.93% at 22 g/kg specific humidity and 30 °C temperature of ambient air. The slope of the line is increasing on increasing the specific humidity, which results in greater change in percentage saving in cooling load on the coil at higher specific humidity of ambient air.

Variation of cooling load (Q_c) on the cooling coil for the same cooling rate (Q_r) of 9.317 kW to the conditioned space with respect to temperature for different specific humidities has been shown in Fig. 8. The load on the coil (Q_c) is increasing with increasing the temperature and specific humidity of the ambient air. It is found maximum with a value of 22.49 kW at 22 g/kg specific humidity and 46 °C temperature, while it is found minimum with a value of 3.86 kW at 6 g/kg specific humidity and 30 °C temperature of ambient air.

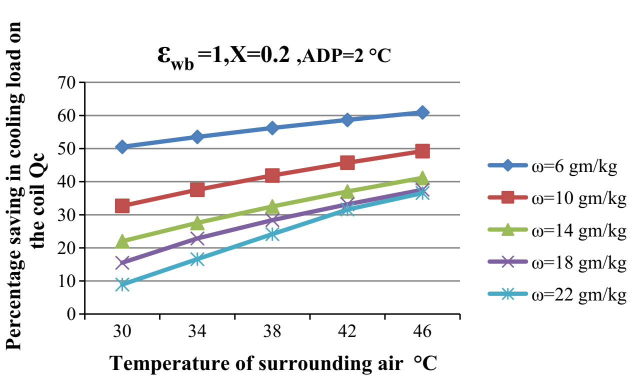

Figure 7 Variation of saving in cooling load (%) with respect to ambient temperature and specific humidity.

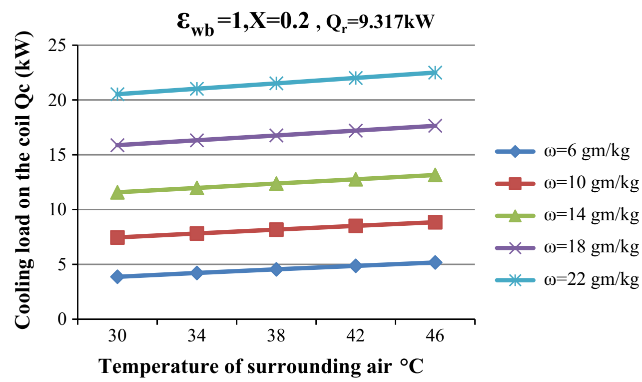

Figure 8 Variation of cooling load on the coil with respect to ambient temperature and specific humidity.

Variation of cooling load (Q_c) on the cooling coil in the proposed system for same cooling rate (Q_r) of 9.22 kW to the conditioned space with respect to wet bulb effectiveness of the dew point evaporative cooler for different specific humidities of ambient air has been shown in Fig. 9. The load on the cooling coil is decreasing with increasing the ε_{wb}, while it is increasing with increasing the specific humidity of ambient air. The load on the coil is found maximum with a value of 22.57 kW at 0.9 ε_{wb} and 22 g/kg specific humidity, while it is minimum with a value of 4.22 kW at 1.1 ε_{wb} and 6 g/kg specific humidity of ambient air.

Variation of ε_{dew} of the evaporative cooler in the proposed system with respect to the ε_{wb} for different specific humidities at a constant ambient temperature has been shown in Fig. 10. It is increasing with increasing the ε_{wb} and specific humidity of

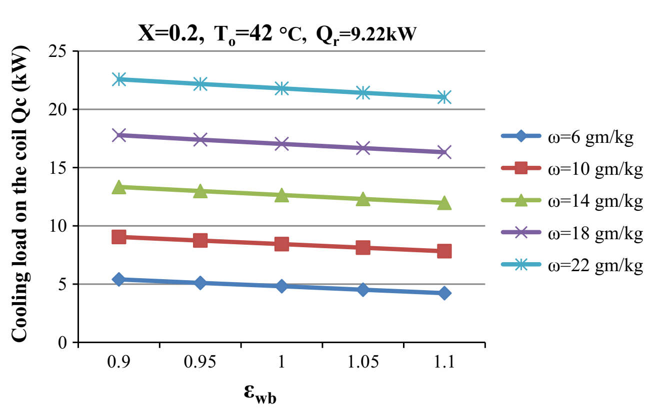

Figure 9 Variation of cooling load on the coil with respect to wet bulb effectiveness and specific humidity.

ambient air. ε_{dew} is found maximum with a value of 0.85 at 1.1 ε_{wb} and 22 g/kg specific humidity, while it is minimum with a value of 0.55 at 0.90 ε_{wb} and 6 g/kg specific humidity of ambient air.

Variation in the outlet temperature of evaporative cooler: temperature of air supplied to the coil with respect to ε_{wb} has been shown in Fig. 11. The temperature is decreasing with increasing the ε_{wb}, while it is increasing with increasing the specific humidity of ambient air. The temperature is found maximum with a value of 30.36 °C at 0.9 ε_{wb} and 22 g/kg specific humidity, while it is found minimum with a value of 17.31 °C at 1.1 ε_{wb} and 6 g/kg specific humidity of ambient air.

Variation of ε_{dew} with respect to temperature ambient for different specific humidities has been shown in Fig. 12. Its value is increasing with increasing the temperature and specific humidity of ambient air. ε_{dew} is maximum with a value of 0.78 at 46 °C and 22 g/kg while it is minimum with a value of 0.58 at 30 °C and 6 g/kg of ambient air.

Variation of outlet temperature of evaporative cooler with respect to ambient temperature for different specific humidities has been shown in Fig. 13. The outlet temperature of evaporative cooler is increasing with increasing the temperature of ambient air and also increases with increasing the specific humidity. It is found maximum with a value of 29.88 °C at 46 °C and 22 g/kg specific humidity, while it is minimum with a value of 15.68 °C at 30 °C and 6 g/kg specific humidity of ambient air.

Variation of the cooling load (Q_c) on the cooling coil in the proposed system with respect to bypass factor (X) of the cooling coil for different specific humidities has been shown in Fig. 14. The load on the cooling coil is decreasing with increasing the bypass factor, while it is increasing on increasing the specific humidity of ambient air. It is found maximum with a value of 18.65 kW at 0.1 X and 22 g/kg specific humidity, while it is found minimum with a value of 4.15 kW at 0.3 X and 6 g/kg specific humidity of ambient air.

Variation of mass flow rate (m_a) of air supplied to the conditioned space for same cooling rate of 9.317 kW with respect to ambient temperature for different specific humidities has been shown in Fig. 15. The mass flow rate is increasing with increasing the temperature of ambient air and also increasing with increasing the specific humidity of ambient air. It is found maximum with a value of 0.366 kg/s at 46 °C and 22 g/kg specific humidity, while it is found minimum with a value of 0.28 kg/s at 30 °C and 6 g/kg specific humidity of ambient air.

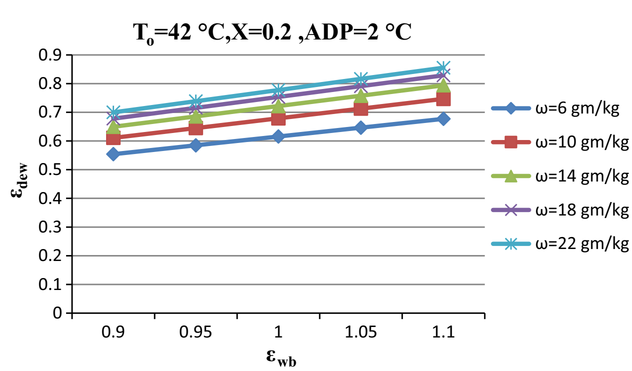

Figure 10 Variation of dew point effectiveness with respect to wet bulb effectiveness and specific humidity.

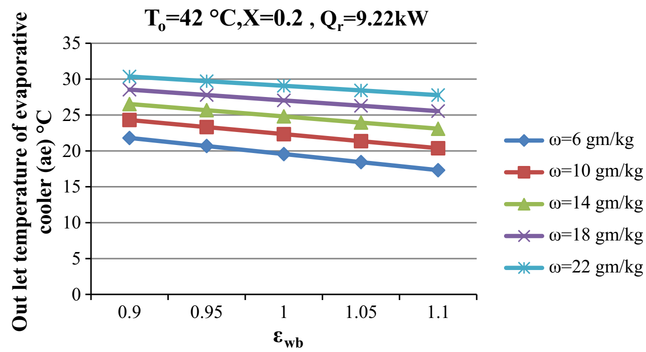

Figure 11 Variation of outlet temperature of dew point evaporative cooler with respect to wet bulb effectiveness and specific humidity.

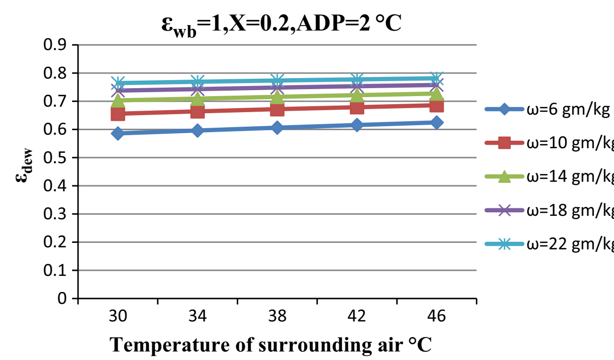

Figure 12 Variation of dew point effectiveness with respect to ambient temperature and specific humidity.

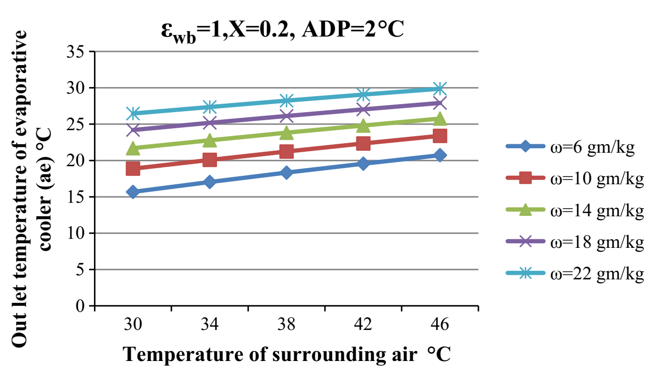

Figure 13 Variation of outlet temperature of dew point evaporative cooler with respect to ambient temperature and specific humidity.

Variation of monthly power saving (MPS) in kW h of the proposed system as compared to the conventional system with respect to ambient temperature for different values of specific humidity has been shown in Fig. 16. MPS is increasing with increasing the ambient temperature while it is decreasing on increasing the specific humidity of ambient air. It is found maximum with a value of 240.28 kW h at 46 °C and 6 g/kg specific humidity, while it is found minimum with a value of −24.44 kW h at 30 °C and 22 g/kg specific humidity of ambient air. Negative sign indicates that the proposed system is

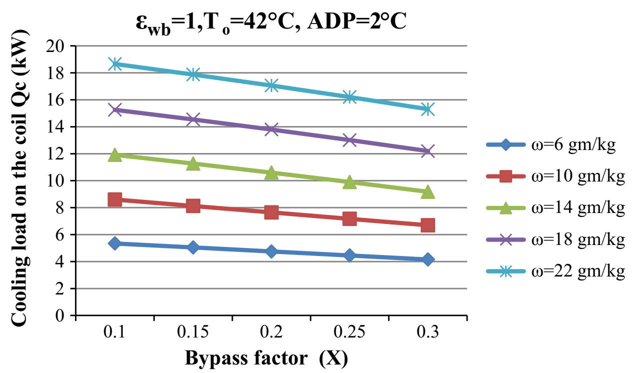

Figure 14 Variation of cooling load on the coil with respect to bypass factor of the coil and specific humidity.

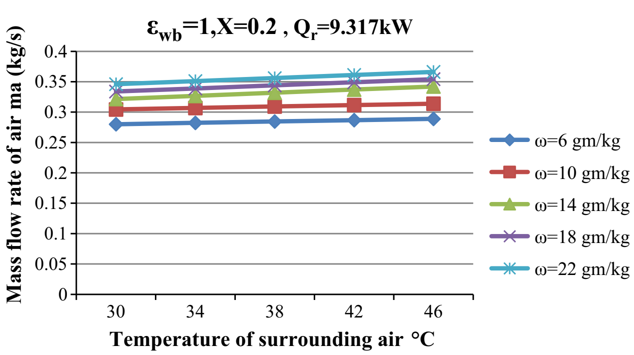

Figure 15 Variation of mass flow rate of air with respect to ambient temperature and specific humidity.

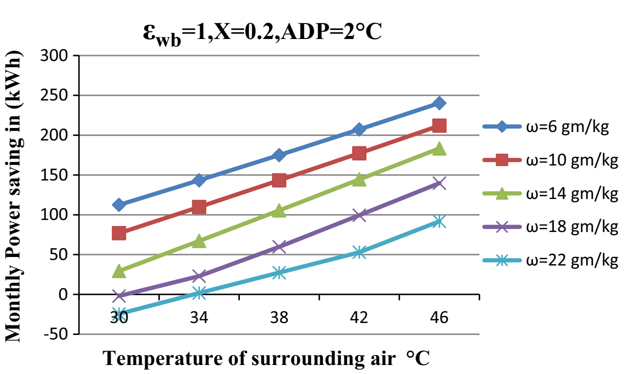

Figure 16 Variation of monthly power saving with respect to ambient temperature and specific humidity.

not suitable for very humid conditions. The power saving is decreasing with increasing specific humidity because of increasing the latent load on the coil which results in decreasing the power saving, which shows that the proposed system is working reasonably well for dry and moderate humid conditions but not suitable for very humid conditions.

7. Conclusion

The proposed system has been theoretically analysed for wide range of ambient temperatures and specific humidity under some reasonable assumptions. The proposed system has also been compared from the conventional vapour compression air conditioner on the basis of saving (%) in cooling load on the cooling coil for the same cooling rate to the conditioned space working on 100% fresh air assumption. The saving of cooling load on the coil is found maximum with a value of 60.93% at 46 °C and 6 g/kg specific humidity, while it is found minimum with a value of 8.93% at 30 °C and 22 g/kg specific humidity of ambient air. The power saving is decreasing with increasing specific humidity due to increasing the latent load on the coil, which results in decreasing the power saving. This shows that the proposed system is working reasonably well for dry and moderate humid conditions but not suitable for very humid conditions and worked well with a maximum net monthly power saving of 240.28 kW h at 46 °C and 6 g/kg specific humidity for a small capacity application. Therefore it could be a better alternative for dry and moderate humid climate with a payback period of 7.2 years.

The proposed system was also compared on the various parameters with following conclusions.

- The cooling load on the cooling coil increases with increasing the temperature and specific humidity of ambient air due to increase in specific enthalpy of ambient air, but decreases with increasing the wet bulb effectiveness of the dew point evaporative cooler and bypass factor of cooling coil for the same cooling rate to the conditioned space.
- Mass flow rate of air supplied to the conditioned space increases with increasing the temperature and specific humidity of ambient air for the same cooling rate due to increasing the specific enthalpy of ambient air.
- The dew point effectiveness of the evaporative cooler increases with increasing the wet bulb effectiveness, temperature and specific humidity of ambient air.
- The outlet temperature of evaporative cooler increases with increasing the temperature and specific humidity of ambient air but decreases with increasing the wet bulb effectiveness of the evaporative cooler.
- Monthly power saving (kW h) in the proposed system as compared to the conventional system is increased with increasing the temperature but decreases with increasing the specific humidity of ambient air.
- Cooling load on the coil is calculated at wet bulb effectiveness of 1 but this load can further be decreased by increasing the effectiveness of evaporative cooler up to 1.15 through the effective cooling pad technology [7].
- Dew point evaporative cooler is a better alternative to the direct and indirect evaporative cooler.
- DPEC is an environmentally clean and energy efficient system can be used as pre-cooling unit to reduce the size of mechanical cooling system, peak load and electrical energy consumption during the cooling season.

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