INTERNATIONAL JOURNAL OF
ENERGY AND ENVIRONMENT

Volume 2, Issue 5, 2011 pp.921-932
Journal homepage: www.IJEE.IEEFoundation.org

Comparative thermal analysis of theoretical and experimental studies of modified indirect evaporative cooler having cross flow heat exchanger with one fluid mixed and the other unmixed

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Abstract
The comparative thermal analysis of theoretical and experimental studies of modified indirect evaporative cooler having cross flow heat exchanger with one fluid mixed and the other unmixed is presented in this research paper. A heat and mass transfer mathematical model is developed to simulate the properties of indirect evaporative cooler. The theoretical result analysis was done by plotting the curves between various performance parameters. This work presents the fabrication and experiments carried out on the indirect evaporative cooler at various outdoor air conditions. The data acquired by experiment were analyzed by plotting the curves between various performance parameters. The theoretical and experimental results were compared and analyzed. The theoretical model can be used to predict the performance of modified indirect evaporative cooler.

Keywords: Effectiveness of heat-exchanger; Evaporative cooling; Humidification efficiency; Specific humidity; Ton refrigeration.

1. Introduction
The simple evaporative cooler is not useful in hot and humid summer weather conditions which prevail in coastal regions like Mumbai, Chennai etc. as it increases the moisture content of the space to be cooled. In such conditions an indirect evaporative cooling system has the potential for widespread applications in industrial and commercial buildings because it cools the air without adding moisture to it. Camargo [1] presented the basic principles of the evaporative cooling process for human thermal comfort, the principles of operation for the direct evaporative cooling system and the mathematical development of the equations of thermal exchanges, allowing the determination of the effectiveness of saturation in his paper. This is achieved with the use of some form of heat exchanger [2, 3] that uses the cool moist air, produced through evaporative cooling, to lower the temperature of drier air. This cool dry air is then used to cool the environment, and the cool moist air is expelled to ambient. The indirect evaporative cooling process requires no energy input apart from that required for the fan and water

pump. The coefficient of performance of this system is therefore likely to be high. The principle of an indirect evaporative cooler using porous ceramic as the cooling source and a heat pipe as the heat transfer device is presented Riffat and Zhu [4] in their paper. In a novel counter flow heat and mass exchanger the cooling (dew point and wet bulb) effectiveness and energy efficiency are largely dependent on the dimensions of the airflow passages, air velocity and working-to-intake-air ratio, and less dependent on the temperature of the feed water [5]. It is observed that the materials used as heat and mass transfer medium in the indirect evaporative cooling systems, the thermal properties of the materials, i.e., thermal conductivity and water-retaining capacity (porosity), have little impact on system heat/mass transfer, and therefore, these two parameters play low keys in terms of material selection. Instead, shape formation/holding ability, durability, compatibility with water-proof coating, contamination risk as well as cost, are more important concerns in this regard [6]. Qureshi and Zubair [7] demonstrated in their research paper that thermal effectiveness of the evaporative heat exchangers degrades significantly with time indicating that, for a low risk level (pZ0.01), there is about 66.7% decrease in effectiveness for the given fouling model. Furthermore, it is noted that there is about 4.7% increase in outlet process fluid temperature of the evaporative fluid cooler. The application of adsorbents, would allow the evaporative cooling system to be applied in hot, humid climates, in addition to hot climates with low humidity [8]. The energy cost, the environmental impact, the green house effect due to CO_2 emissions and the destruction of the ozone layer due to the emission of the coolant, the necessity of fulfilling established protocols, etc. force to reduce energy demand. This can be achieved using more energy efficient equipment. Within this context, evaporative cooling offers an alternative for enhancing the performance of the air conditioning installations [9]. The evaporative cooling system has been widely used for hot and dry climates but for humid climates and during rainy seasons, in the regions where summers are hot and dry, humidity levels are quit high, rendering evaporative cooling ineffective and restricting the widespread use of evaporative cooling in countries like India. This deficiency can be overcome by using desiccants to remove the bulk of moisture and evaporative cooling to reduce temperature [10].

A modified indirect evaporative cooler having cross flow heat-exchanger with one fluid mixed and the other unmixed is presented in this paper. The system is shown schematically in Figure 1.The schematic diagram shows the various components of the modified indirect evaporative cooler eg. cross flow heat exchanger, supply fan, exhaust fan, water tank, water pump and pipe with nozzles to spray water on the top of the heat exchanger. The early experiments used air and water in concurrent flow downward together in the heat-exchanger vertical passages. That encouraged rivulet, not film, flow of water and potentially dry areas. Therefore, the air flow was reversed; upward against the water, it slows its downward flow and spreads it film-wise, increasing the wetted surface. Thus, return air enters below the heat-exchanger, passes through the drizzle of water draining from it, and is drawn upward into its vertical wet side passages.

Above the heat-exchanger are multiple nozzles spraying water down on it and into its narrow, presumably water lined, passages. About 30 cm below the heat-exchanger a broad water sump catches drippings for recirculation. It contains a float valve to admit new water and drain and bleed-off outlets, etc.; outside at one end is an electric pump to power the sprays. At this end of the cabinet are large, outside-air entry grilles protected by very efficient dust filters. Inside the grilles supply air fan is fitted and above it on the right top corner a exhaust fan is fitted. The upper fan (Exhaust) draws air in through the return-air opening and up through the wet-side exchanger passages, down which water films are moving. After absorbing heat in the heat-exchanger, that secondary air, now containing much evaporative water, is discharged outdoors by the fan. In modified indirect evaporative cooler a part of exhaust humid air is re-circulated as per the requirement. Thus, the wet-side operates re-generatively on return air. Meanwhile, the lower fan (supply) draws in outside air through the filters and pushes it into corresponding horizontal dry-air passages. Here, it surrenders heat to the air-water mixture in the wet-air passages and becomes cooled. Then, the fan pushes it out into any provided delivery ducts. The cooling process of modified indirect evaporative cooler is shown on psychrometric chart in Figure 2.

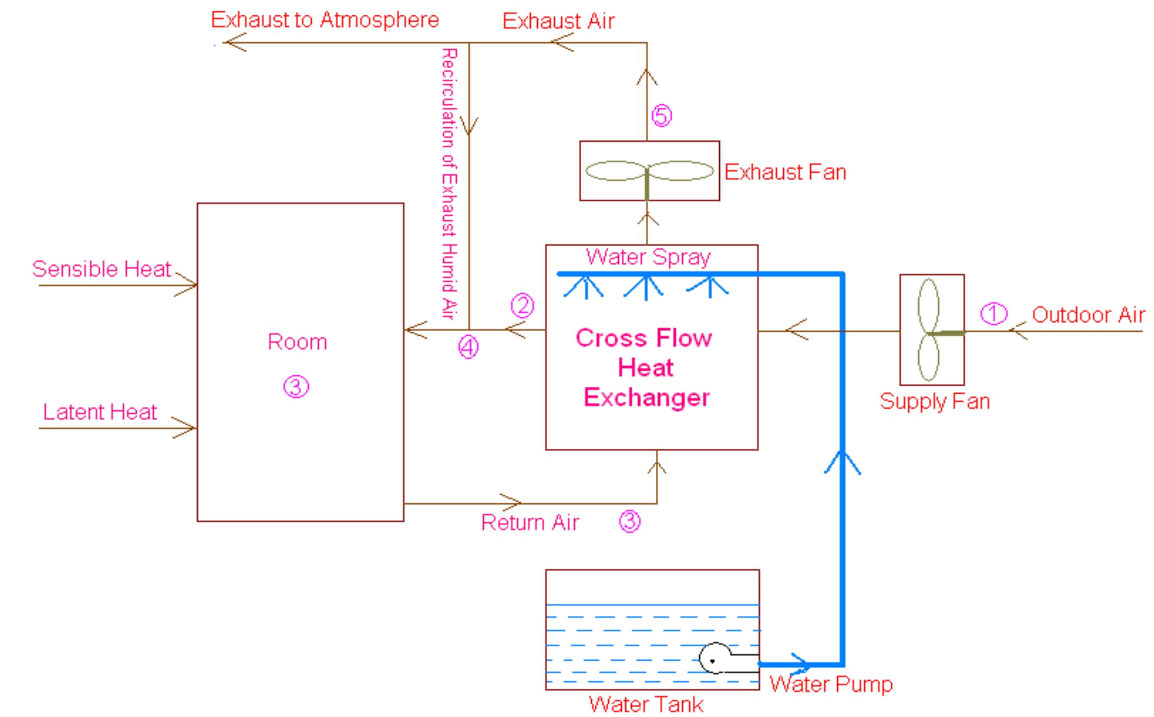

Figure1. Schematic diagram of modified indirect evaporative cooler

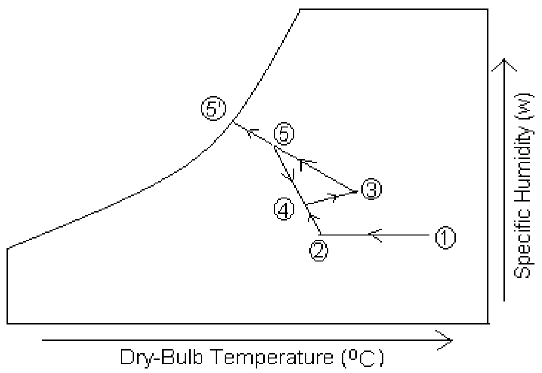

Figure 2. Psychrometric representation of cooling process

Process 1-2	represents sensible cooling of outdoor air.
Process 3-5-5'	represents adiabatic saturation of the return air from room.
Point 4	represents the condition of supply air to the room (mixing point).

2. Mathematical model

Now to get the thermal analysis of the cooler, theoretically and experimentally, its mathematical model is prepared on the basis of the mass and energy balance between the fluids flowing.

The outdoor air gets sensibly cooled (process 1-2) and in this process only energy transfer takes place while during adiabatic humidification of return air (process 3-5) both mass and energy transfer takes place as shown in Figure 3. The effectiveness of cooler or heat exchanger is assumed to be equal to 70% and heat capacity ratio is taken equal to 1. The humidification efficiency is also assumed equal to 70%. These are the assumptions made for making the mathematical model of the cooler. The effectiveness of cooler or heat-exchanger (ε) is defined as the ratio of actual reduction in dry-bulb temperature of ambient air to the maximum possible reduction in temperature or the ratio of actual heat transfer to the maximum possible heat transfer.

Effectiveness of cooler or heat exchanger

$$(\varepsilon) = Q_{actual}/Q_{max.} = (t_1 - t_2)/(t_1 - t_{wbt1}) \quad (1a)$$

$$t_2 = t_1 - \varepsilon(t_1 - t_{wbt1}) \tag{1b}$$

where t_1= outdoor dry-bulb temperature(^{0}C), t_2= dry-bulb temperature after sensible cooling(^{0}C) and t_{wbt1}= outdoor wet-bulb temperature (^{0}C).

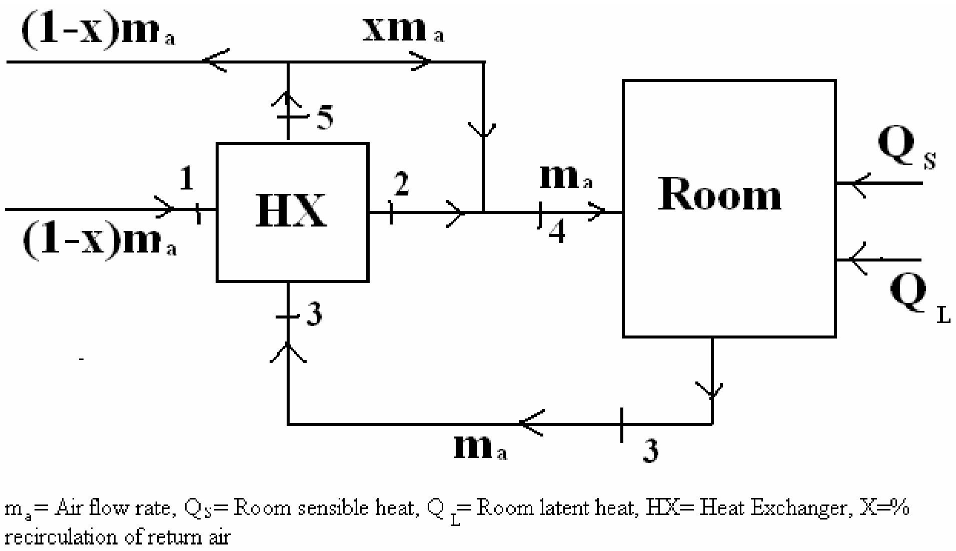

Figure3. Line diagram of modified indirect evaporative cooler

The total heat load of the room or Ton Refrigeration (TR) is equal to the sum of sensible heat and latent heat gain by room per minute. The total heat load of the room is taken up by the supply air at temperature, t_4 to keep the temperature of the room at t_3.
Thus, Room Total Heat or Ton Refrigeration

$$(TR) = m_a c_{pa}(t_3 - t_4)/210 \tag{2a}$$

$$t_4 = t_3 - (210TR)/m_a c_{pa} \tag{2b}$$

where t_3= dry-bulb temperature of room (^{0}C), t_4= dry-bulb temperature at supply point (^{0}C), m_a= mass flow rate of air (kg/min) and C_{pa}= specific heat of air (1.005 kJ/kg K).
The return air from room entering the vertical passages of the cross flow heat-exchanger at point 3 as shown in figure 1. The return air gets humidified and cooled during the process 3-5. The partial adiabatic saturation of the air can be used to reduce its dry-bulb temperature in hot weather conditions. The lowest possible dry-bulb temperature is equal to the wet-bulb temperature of the entering air. The complete humidification of air is not possible. To study the performance of spray chamber of the cross flow heat-exchanger, the humidifier efficiency is defined as the ratio of actual humidification to the maximum possible humidification.
Humidifier efficiency (η_h) = actual humidification / maximum possible humidification

$$\eta_h = (w_5 - w_3)/(w_{5'} - w_3) \tag{3a}$$

$$w_5 = w_3 + \eta_h(w_{5'} - w_3) \tag{3b}$$

where w_3 = specific humidity at room temperature (kg w.v./kg of dry air), w_5 = specific humidity after adiabatic humidification and $w_{5'}$ = specific humidity at saturation point.

$$(w_5 - w_3) = m_w / m_a \tag{4a}$$

So, mass of make up water,

$$m_w = m_a(w_5 - w_3) \tag{4b}$$

where m_w = mass of make up water (kg/min).
The fresh outside air (1-X) m_a getting sensibly cooled and the room return air (m_a) is getting humidified. So, the sensible heat loss by the outside air is equal to the heat absorbed by water.

$$\text{Heat loss by fresh air } = (1-X)m_a c_{pa}(t_1\text{-}t_2) \tag{5}$$

where X = percent recirculation of exhaust humid air.

$$\text{Heat absorbed by water } = m_a c_{pa}(t_3 - t_5) + m_w(h_{fg}) \tag{6}$$

where t_5 = dry-bulb temperature after adiabatic humidification (^{0}C) and h_{fg} = latent heat of evaporation of water (2500 kJ/kg).
Heat loss by fresh air = heat absorbed by water. So,

$$(1-X)m_a c_{pa}(t_1\text{-}t_2) = m_a c_{pa}(t_3 - t_5) + m_w(h_{fg}) \tag{7}$$

By comparing equations (5) and (6), the percentage recirculation of return humidified air X m_a can be calculated.

$$X = 1-(t_3 - t_5)/(t_1\text{-}t_2) - m_w(h_{fg})/m_a c_{pa}(t_1\text{-}t_2) \tag{8}$$

3. Theoretical analysis and results

In order to make the theoretical analysis, the indoor comfort conditions are fixed and the effectiveness of the heat-exchanger and humidification efficiency is assumed. For this the "Refrigeration and Air-Conditioning" Data-Book by Manohar Prasad [11] was referred.
Theoretical analysis was done for different atmospheric conditions, based on mathematical model prepared. The data acquired are presented here in the form of curves from Figures 4 to 10.
The room comfortable conditions are taken as, (t_3) = 26.5^{0}C and R.H. =55%. At these inside comfort conditions, the specific humidity are (w_3) = 0.0118 kg/kg of dry air, (w_5) = 0.0136 kg/kg of dry air, ($w_{5'}$) = 0.0144 kg/kg of dry air, (t_5) = 22.2 ^{0}C at w_5. For water tank capacity of 40 liters, the prescribed volume air flow rate of supply air is 2000 m^3/hr, taken from R.A.C., data book. So, mass flow rate of supply air, (m_a) = (2000 x 1.2) / 60 = 40 kg/min. The mass of make up water, $m_w = m_a(w_5\text{-}w_3)$ = 0.072 kg/min.

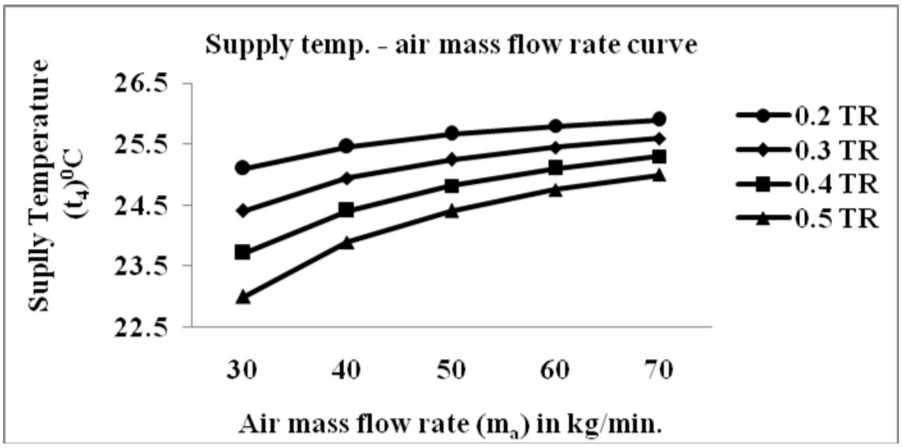

Figure 4. Supply air temperature (t_4) v/s mass flow rate (m_a) curve

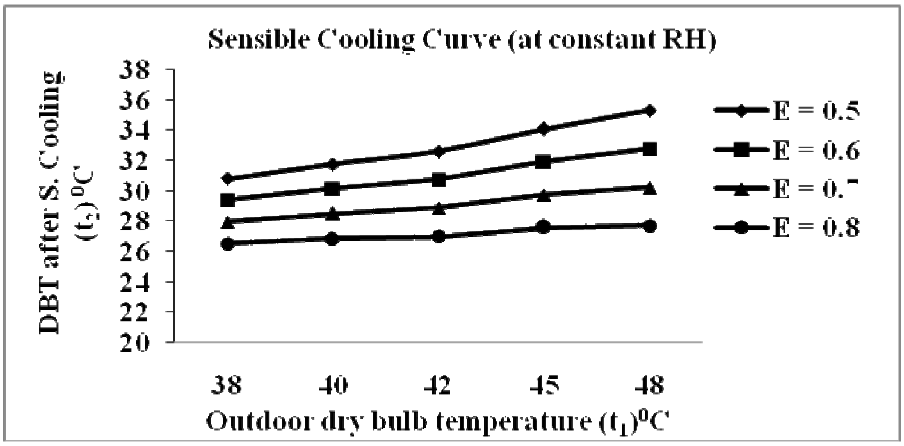

Figure 5. Sensible cooling curve (at constant air outdoor relative humidity)

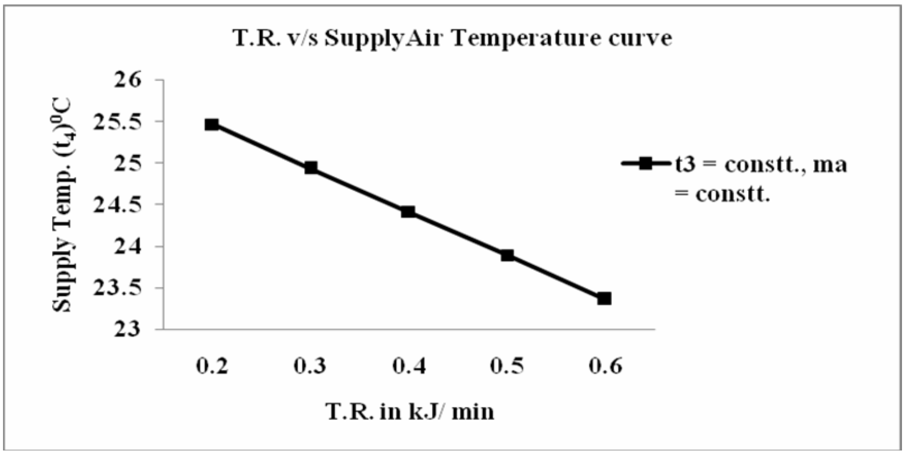

Figure 6. Ton Refrigeration (T.R.) v/s Supply Air Temperature (t_4) curve

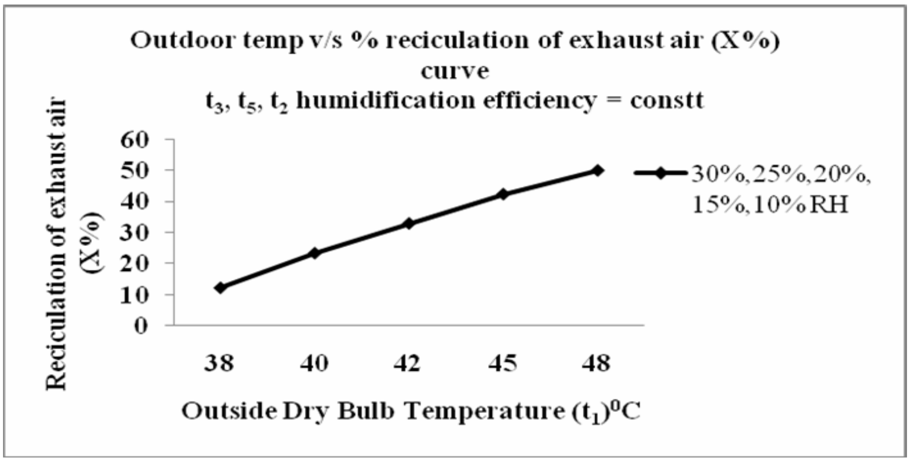

Figure 7. Out-door dry-bulb temperature (t_1) v/s recirculation of exhaust humid air (X) curve

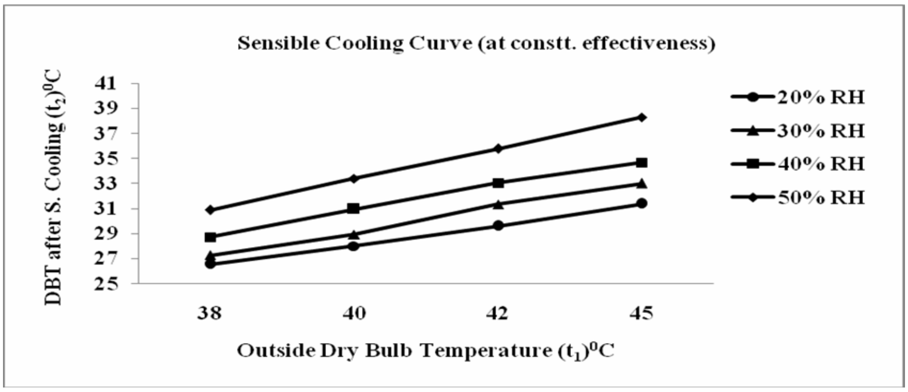

Figure 8. Sensible cooling curve (at constant effectiveness of heat exchanger)

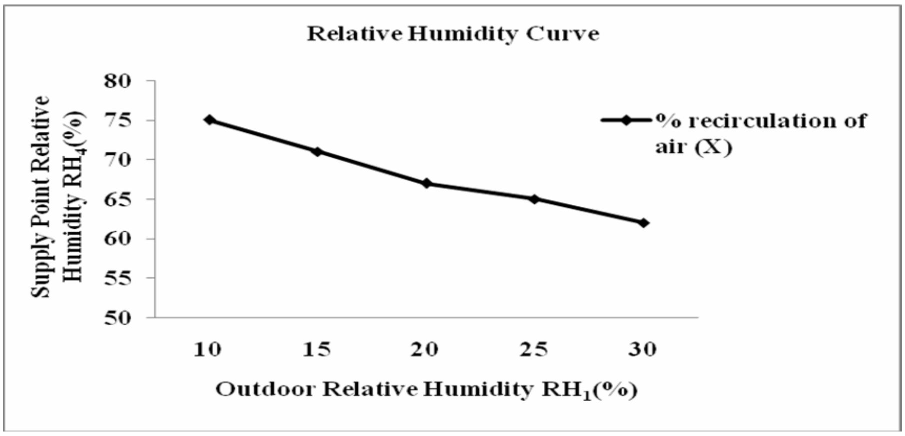

Figure 9. Outdoor Relative Humidity $(RH)_1$ v/s supply point Relative Humidity $(RH)_4$ curve

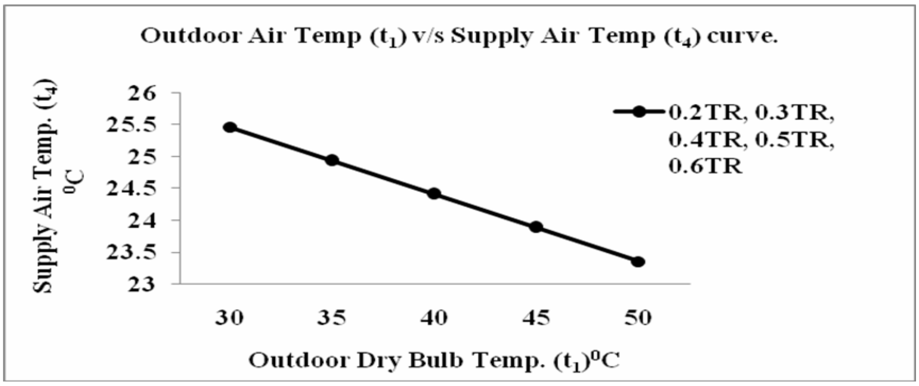

Figure 10. Outdoor dry-bulb temperature (t_1) v/s Supply air temperature (t_4) curve

4. Experimental study

Experimental study was done to get the actual performance behaviour of the modified indirect evaporative cooler**.** The results obtained were analyzed and the curves were plotted between various parameters.

4.1 Fabrication of experimental set-up

The cooler consist of G.I. sheet rectangular cabinets 4 ft (1.21m) long, 3 ft (0.914m) wide, and 3.5 ft (1.07m) high. On one long side is a large cooled air outlet matching the length and height of the heat exchanger within. Under it is a return air inlet of almost equal size. The cross flow heat exchanger, approximately 2 ft (0.609m) long, 0.5 ft (0.152m) high and 1.5 ft (0.457m) wide, is mounted horizontally just inside the upper opening and delivers its cooled air horizontally through it. The main components of experimental set up (modified indirect evaporative cooler) are Supply Fan (15 inch or 0.381m), Exhaust Fan (15 inch or 0.381m), Supply air duct, Exhaust return air duct, Cross flow plate type heat exchanger, Water tank and Water pump (1.5 m head).

4.2 Experimental analysis and results

The experimental analysis of performance of the modified indirect evaporative cooler was done by performing experiments on it, at different atmospheric conditions, (Table 1), (at different outdoor dry-bulb temperature and relative humidity) and at different velocities of supply air (Table 2).

At room inside dry-bulb temperature (t_3) =26.5^{0}C and 55% Relative Humidity. The dry-bulb temperature after adiabatic humidification (t_5) = 23.5 ^{0}C and saturation temperature ($t_{5'}$) = 20 ^{0}C. The respective specific humidity are as follows:- w_3 = 0.0118 kg/kg of dry air, w_5 = 0.0131 kg/kg of dry air and $w_{5'}$ = 0.0144 kg/kg of dry air. So, the humidification efficiency (η_h) = (w_5 - w_3)/($w_{5'}$ - w_3) =0.5 or 50%.

At supply air velocity of 5 m/s the mass flow rate of supply air can be calculated as,

$$m_a = 60\rho_a V_a (\pi D^2 / 4)\,\text{kg/min} \qquad (9)$$

where D= blade diameter of the supply air fan (15 inch or 0.381m), V_a= supply air velocity in m/s, ρ_a= supply air density (1.2 kg/m^3).

Now by putting the values of all the parameters in the above equation, the mass flow rate of supply air, m_a= 41.04 kg/min. (at V_a = 5m/s).

Now, the mass of make up water (m_w) = m_a (w_5-w_3) kg/min. = 0.054 kg/min.

Table 1. Variation of various parameters with outdoor dry-bulb temperature and relative humidity

S. No.	Out-door Dry-Bulb Temp. (t_1^{0}C)	Out-door Relative Humidity (Φ%)	Temp. after sensible cooling (t_2) ^{0}C	Recirculation of humid Exhaust air(X%)	T. R kJ/min.	Supply Temp. (t_4^{0}C)
1	38	30	27.98	12.4	0.2	25.46
2	40	25	28.53	23.5	0.3	24.93
3	42	20	28.91	33	0.4	24.41
4	45	15	29.77	42.4	0.5	23.89
5	48	10	30.28	50	0.6	23.36

Table 2. Experimental data table at supply air velocity of 5 m/s

ODBT (t_1) ^{0}C	t_{wbt1} in ^{0}C	t_2 in ^{0}C	t_4 in ^{0}C	t_{wbt4} in ^{0}C	t_5 in ^{0}C	$t_{5'}$ in ^{0}C
35	22.8	28.9	24.97	20.40	23.25	20
37	23	30	24.46	20.30	23.25	20
40	23.5	31.75	23.95	20.20	23.25	20
42	23.2	32.6	23.44	20.10	23.25	20

The results acquired by performing experiments on the modified evaporative cooler were plotted in the form of curves to make the experimental analysis.

5. Comparative analysis of theoretical and experimental results

The comparative thermal analysis of modified indirect evaporative cooler was done on the basis of theoretical and experimental studies performed. The various outcomes of the studies were as follows:

- In theoretical analysis, it was assumed that the effectiveness and humidification efficiency of the cross flow heat-exchanger or cooler are 70% but experimentally it was found that the values are only 50%. So, the experimental values of effectiveness and humidification efficiency are 28.57% less as compared to the theoretical values assumed.
- By theoretical analysis of the mathematical model of modified indirect evaporative cooler, it was found that the mass flow rate of make up water is equal to 0.072 kg/min. while the mass flow rate of make up water as per experimental analysis done is equal to 0.054 kg/min. Thus, experimentally the mass flow rate of make up water is 25% less than the theoretical value.
- The variation in experimental and theoretical graphs is because of less value of effectiveness and humidification efficiency obtained experimentally.

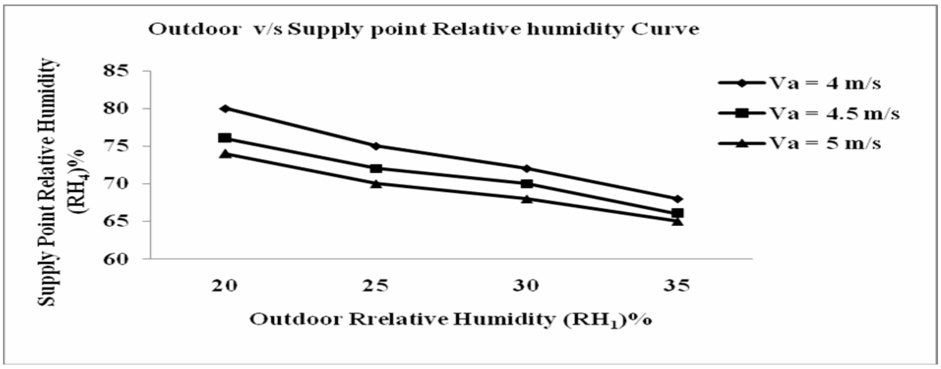

Figure 11.Outdoor Relative Humidity (RH_1) v/s Supply Point Relative Humidity (RH_4) curve

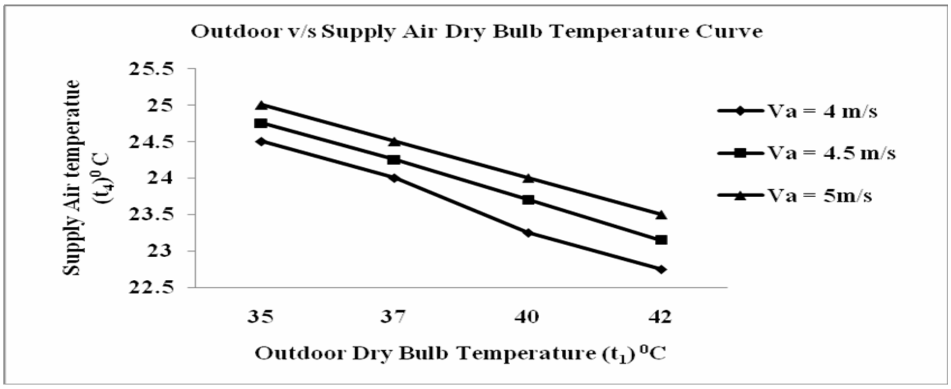

Figure 12. Outdoor v/s supply air dry-bulb temperature curve

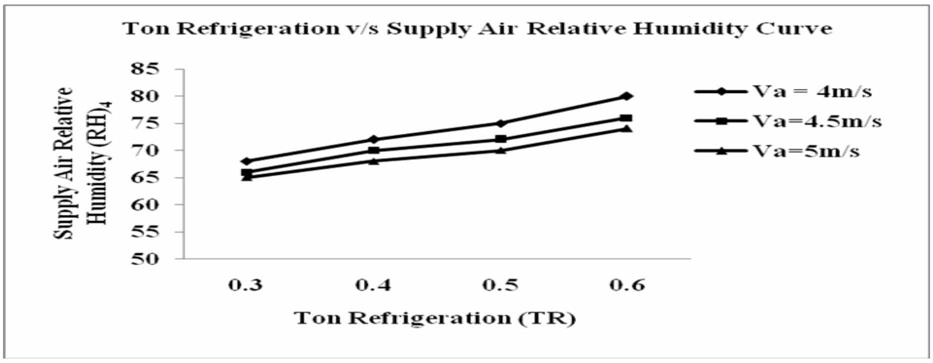

Figure 13.Ton Refrigeration (TR) v/s Supply Air Relative Humidity (RH_4) curve

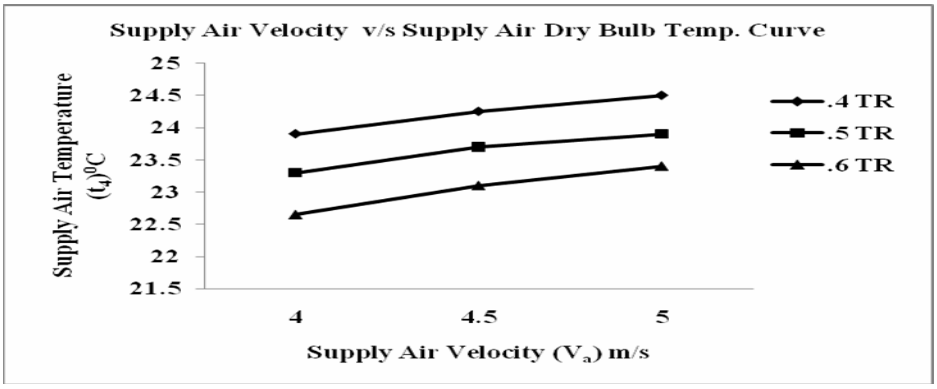

Figure 14. Supply air velocity (v_a) v/s supply air dry-bulb temperature (t_4) curve

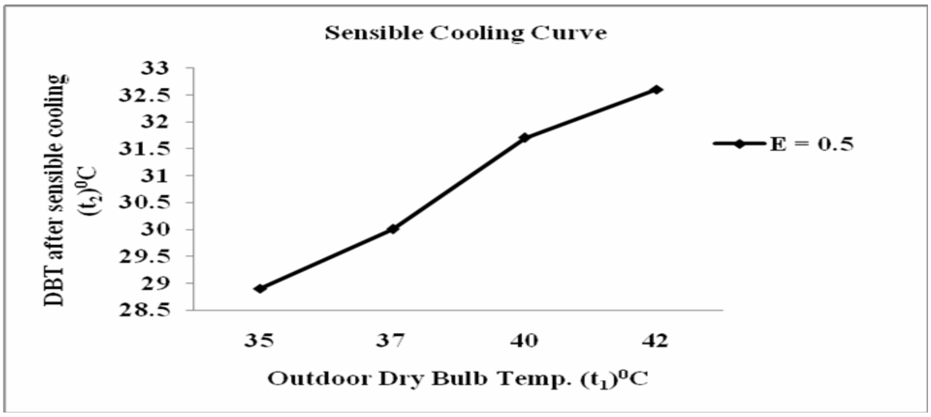

Figure 15. Outdoor dry-bulb temperature (t_1) v/s DBT after sensible cooling (t_2) curve

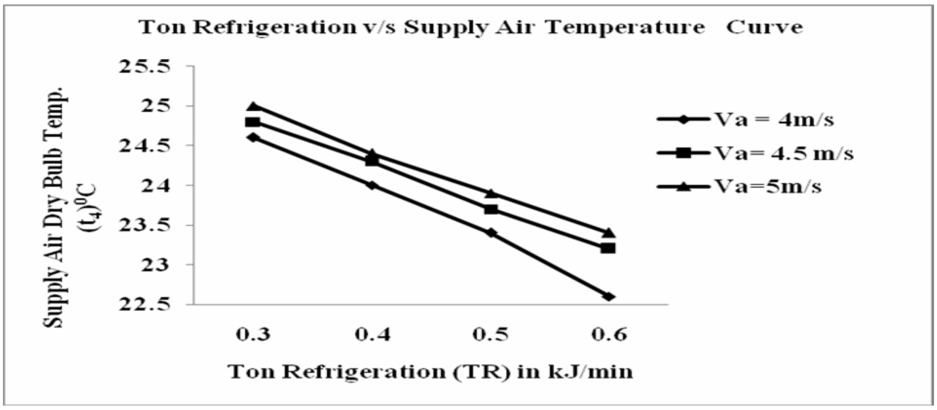

Figure 16. Ton Refrigeration (T.R.) v/s supply air dry-bulb temperature (t_4) curve

6. Results and discussion

The outcome of various graphs obtained from theoretical and experimental analysis of modified indirect evaporative cooler is as follows:

- With increase in supply air temperature (t_4), the mass flow rate of air (m_a) also increases accordingly in order to meet the fixed comfortable conditions inside the space to be cooled as shown in figure 4. The supply air has to take the total heat load of the room. If the supply air temperature (t_4) increases, it is not sufficient to take the total heat load of the room. This deficiency is overcome by supplying more air per minute. The experimental results show that, at constant ton refrigeration (TR), the supply air dry-bulb temperature (t_4) increases with increase supply air velocity (v_a) as shown in Figure 15.
- At constant effectiveness of heat exchanger, the difference in temperature of air entering (t_1) and of air leaving (t_2) remains constant. So, with increase in outdoor dry-bulb temperature (t_1), the temperature obtained after sensible cooling (t_2) increases to keep the difference of temperatures constant as shown in Figure 5.
- At constant room temperature and mass flow rate of air the total heat load of room (TR) is to be taken by supply air. Therefore with increase in the total heat load of room (TR), the supply air temperature (t_4) decreases or they are inversely proportional to each other as shown in Figure 6.
- With increasing outdoor dry bulb temperature (t_1) and decreasing relative humidity (%), the percentage recirculation of return humidified air is increased as shown in Figure 7. to meet the fixed comfortable conditions inside the space to be cooled because in hot and dry summer weather conditions the relative humidity of outdoor air is less and to acquire the 55% relative humidity inside the room(comfortable condition ,as per R.A.C., data book) the percentage recirculation of return humidified air is increased.
- Sensible cooling capacity decreases with increase in relative humidity of outdoor air at constant dry-bulb temperature.

- With increase in relative humidity of outdoor air, the relative humidity of air at supply point decreases as shown in Figure 9. Experimental results show that, at constant outdoor relative humidity (RH_1), relative humidity at supply point (RH_4) decreases with increase in supply air velocity (Va) as shown in Figure 12.
- At constant outdoor relative humidity, with increase in supply air velocity, the supply air relative humidity decreases as shown in Figure11 and with increase in cooling load (TR) the supply air relative humidity increases as shown in Figure13.
- For constant cooling load (TR), the supply air temperature increases with increase in supply air velocity as shown in Figure14 and at constant supply air velocity, with increase in cooling load (TR), the supply air temperature(t_4) decreases in order to meet the fixed comfortable conditions inside the space to be cooled as shown in Figure16.
- With increase in outdoor dry bulb temperature (t_1), supply point temperature (t_4) decreases as shown in figure 10. The comfortable condition inside the room is 26.5^0C and 55% Relative Humidity as taken from RAC, Data Book by Manohar Prasad. So, in order to take the sensible heat load of the room at increased outdoor dry-bulb temperature, the supply point temperature (t_4) is decreased.

7. Conclusion

After going through thermal analysis of theoretical and experimental studies of modified evaporative cooler, at different atmospheric conditions, the following conclusions were made:

- The evaporative cooler gives best performance in hot and dry summer weather conditions.
- In hot and humid climate like in Mumbai, Chennai etc. the modified indirect evaporative cooler with only sensible cooling and without recirculation of return exhaust humid air is suitable.
- Theoretical study can be used to predict the performance of modified indirect evaporative cooler.
- It is recommended that to improve the performance of modified indirect evaporative cooler, thermal conductivity between supply air and heat exchanger plates should be increased.
- Higher cooling effect can be obtained by increasing the effectiveness of heat exchanger.
- The recirculation of return exhaust humid air in required percentage (X) increases cooling effect of modified indirect evaporative cooler.

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