MODELLING AND ANALYSIS OF THREE STAGE INDIRECT/DIRECT EVAPORATIVE COOLING SYSTEM TO MAKE USE OF COLD CONDENSATE WATER

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ABSTRACT: *A three stage alternate cooling system with core objective of reduced energy consumption is analyzed in this work. The system consists of (1) a water to air cooling coil (2) an indirect evaporative cooler and (3) a direct evaporative cooler. The condensate water generated in AHUs of 17 storeys corporate office building with a 500TR cooling load, is collected in a thermally insulated storage tank and used as a cooling and evaporating water in the first and last two stages of the system respectively. This system is modeled on TRNSYS, a transient system simulation software for the weather conditions of the city of Lahore. The results indicate that analyzed cooling system has considerable potential of energy and cost savings as compared to conventional VCC based split air conditioners for a single room for months of summer season having lower relative humidity.*

Keywords: Cold condensate water collection, Indirect-direct evaporative cooler, energy savings, TRNSYS16

1. INTRODUCTION

As Pakistan lies in the hot and humid climate region, the major energy consumption in the buildings, both residential and commercial sector in the summer season is due to the air conditioning systems. This is a universal trend where nearly 33% of the generated energy is consumed by HVAC systems for achieving comfort zone in the buildings [1]. The rising problem of global warming has developed the critical issue of temperature rise, which consequently adds up to the cooling load requirements of the buildings. This alongwith the depleting natural resources puts a responsibility on the HVAC systems designers to look deeply in to energy efficiency and conservation possibilities in these systems [2-4].

It is a matter of common observation that whenever an air conditioner is running, water is continuously drained out of a pipe attached with the indoor unit. This is the condensate water being generated due a psychrometric process called "Simultaneous cooling and dehumidification". Air is sensibly cooled to decrease its dry bulb temperature (DBT) and dehumidified to remove the latent load by removing the water vapors in the air in condensed form thus lowering its humidity ratio (W) and increasing its Relative humidity (RH). In usual practice, this water is wasted, but if collected in a well-insulated storage tank, as it is at much less temperature than the outside hot air in the summers, this water can be used to harvest its cooling potential. J. Loveless et al. [5] indicated that South East Asia is one the regions where substantial water condensate rates (gal/cfm/year) can be achieved from air conditioners. According to their results, annual condensate collection potential of Karachi city in Pakistan is 19.652 gal/cfm/year. The highest value is for city of Manilla in Philippines with a value of 41.2 gal/cfm/year. S. A. khan et al. [6] reported that nearly 368 $l.Hr^{-1}$ of condensate can be collected through fresh air handling units for high rise buildings using the ventilation rates criteria of ASHRAE standard 62.1 for residential buildings of 16 floors (ground floor + 15 upper floors). This condensate water was used by them to pre cool the condenser's cooling ambient air.

Condensate water from air conditioners can be a good source of drinking water after a little purification [7]. The standards and clauses in the energy efficient design guidelines for buildings highlight the utilization of chilled water condensate [8, 9].

Indirect evaporative cooling (IDEC) is a modified form of the most common and oldest cooling technique; evaporative cooling also called direct evaporative cooling (DEC). IDEC has two air streams flow, one is the secondary air and other is primary air. In IDEC, as the name implies, there is no direct contact between the supply air and water. The secondary air is cooled as in the DEC, then this air cools the primary air (product or supply air) through indirect contact, thus the moisture content (W) and Dew Point Temperature (DPT) of the primary air remains unchanged with a decrease in the dry bulb temperature (DBT) and wet bulb temperature (WBT). Thus it becomes an air to air heat exchanger comprising of passages alternatively occupied by the primary and secondary air, transferring the heat across the separating walls of respective air channels. This is schematically shown in the Fig. 1.

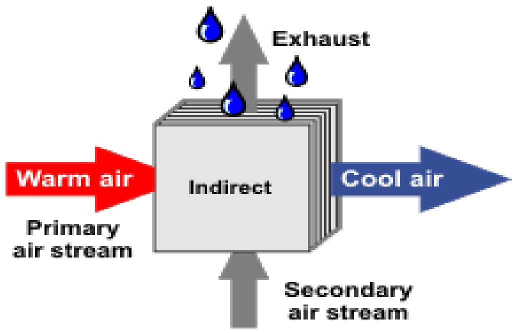

Fig. 1 Schematic diagram of Indirect Evaporative Cooler

Free cooling systems that are actually the systems using the direct and indirect evaporative cooling are a potential option when it comes to energy and cost savings for appropriate climatic conditions [10, 11].

To get increased effectiveness and temperature drop through evaporative cooling, many researchers have worked on systems incorporating a combination of Indirect-Direct Evaporative cooling. Heiderinejad G. et al. [12] experimentally evaluated the cooling performance of two stage indirect/direct evaporative cooling for a variety of simulated climatic conditions. Their results show that more than 60% power saving is achievable as compared to conventional mechanical vapor compression systems (VCC air conditioners). Dilip Jain [13] developed and tested a two stage evaporative cooler and reported an effectiveness in the range of 1.1-1.2 of two stage system over single stage direct evaporative cooler.

Yi Chen et al. [14] worked on the system shown in the Fig. 2. The condensate water from an Air handling unit (AHU) was utilized to act as the cooling water in the IDEC unit placed before AHU. Moreover, the exhaust air from AHU was put into the IDEC as secondary air, thus precooling the fresh intake primary air.

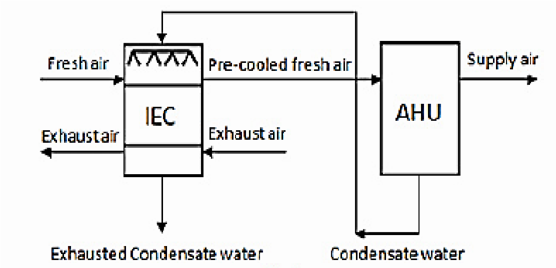

Fig. 2 Schematic diagram of system used in [14]

M Farmahini-Farahani et al. [15] used a water to air cooling coil to pre cool the hot ambient air before entering into the indirect evaporative cooler. The water used in the coil was itself cooled by radiative cooler panels by using the night time cooling. M. Farmahini-Farahani and Heiderinejad. G [16] studied a multi stage cooling system of nocturnal cooling for a water to air pre cooler and indirect-direct evaporative cooling. The energy savings of this multi- step system were found to be in range of 75-79% as compared to VCC systems. Hajidavalloo et al. [17, 18] studied the application of evaporative cooling to precool the inlet hot ambient air to air cooled condensers. They reported around 50% improvement in COP of the air conditioners. A. Khalid et al. [19] developed an experimental setup to evaluate a system of solar assisted pre-cooled hybrid desiccant cooling system and also simulated this system on TRNSYS software for climatic conditions of cities of Karachi and Lahore in Pakistan. Their results indicated that Lahore has higher potential for indirect-direct cooling of air. In present work, a large commercial office building is selected. The cold condensate water from the AHUs in the building is collected in a thermally insulated storage tank. This stored water is used to pre cool the hot ambient air through a cooling coil. The pre-cooled air then passes through an indirect and direct evaporative coolers combination. This 3 stage system is expected to increase overall effectiveness as compared to either using a direct evaporative cooler or an indirect-direct evaporative cooler pair alone. The TRNSYS software is used to simulate this system for the city of Lahore for six months' summer duration (April to September).

2. SYSTEM DESCRIPTION

Fig. 4 shows the schematic of the system under study. The system consists of water condensate piping from air handling units (AHU) of the building, a thermally insulated condensate water storage tank, water pump, a water to air cooling coil, an indirect-direct evaporative cooler and a small electrical equipment room. As a result of simultaneous cooling and dehumidification of air in the AHUs of the building, water condensate is generated, which is collected in the insulated storage tank as soon as the air conditioning plant of the building starts operating in the steady state. The temperature of cold water after collection in the storage tank is taken as 20℃ (68℉). From the performance data log sheet of the refrigeration plant of the building, the average chilled water temperature is 11℃ (52℉). As condensate water sticks to coil outer surface before dropping out, the condensate water is nearly at the EST of the coil. After all the thermal losses from AHU to storage tank, a temperature of 20℃ is a reasonable approximation. This stored water acts as a cold fluid in the cooling coil that is connected to storage tank via water circulation pump. This cooling coil is a compact cross flow heat exchanger (both fluids unmixed), which precools the hot ambient air. As this is a sensible cooling process, the humidity ratio and dew point of air remains constant with a decrease in the dry bulb and wet bulb temperature of air. This precooled air then enters in to the two stage evaporative cooler. In the first stage, i.e. indirect evaporative cooler, the dry bulb and wet bulb temperatures are further lowered with no addition in the moisture content of product air because it is also a sensible heat exchange between the product air and secondary air. In the last stage of the cooling system, air passes through a direct evaporative cooler, having combined heat and mass transfer between the air and evaporating water thus increasing the humidity ratio and dew point of the air with a decrease in dry bulb temperature at constant specific enthalpy of air. The air from the last stage of system enters in to the room under consideration as supply air to remove the cooling load of the room. The psychrometric of the three stage cooling is shown in Fig. 3.

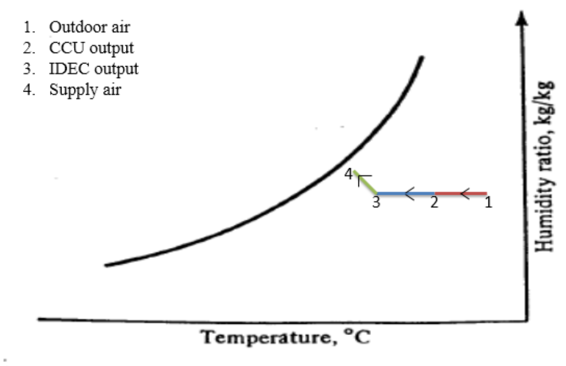

Fig. 3 Psychrometric process of three stage cooling system

3. METHODOLOGY

Fig. 4 shows the schematic of system under study. The mathematical formulation of each component is described in detail in following sections.

3.1 Condensate Collection

In order to calculate the condensate rate from air conditioners, Psychrometric Analyzer software (version 6.8) was employed [20]. The input parameters are shown in Table 1. The same was validated with the help of a laboratory equipment named "Recirculating Air Conditioning Unit A771" (Fig. 5) based on vapor compression cycle employing R134a as a refrigerant. The features of unit A771 are shown in the Table 2. The condensate rate is calculated as follows [21]

$$m_w = m_a *(W_i - W_o) \quad (1)$$

$$m_a = 0.0757 \sqrt{\frac{z}{v_a}} \quad (2)$$

Where, m_w and m_a are mass flow rates of condensate water in kg/s and air to be treated respectively. Subscripts i and o show the inlet and outlet conditions respectively. W is the humidity ratio of air. z is differential pressure in mm of H_2O at inlet to apparatus and v_a is specific volume of air in m^3/kg read from psychrometric chart.

3.2 The Selected Building

A multi storeys commercial office building was selected for further analysis as shown in the Fig. 6 with details as in Table 3.

Table 1 Input parameters in Psychrometric Analyzer Program

Property	Values
Inside Design Condition	*25℃ DBT, 50% Relative Humidity
Outside Design Condition	38℃ DBT, 37% Relative Humidity
Return Air condition	25℃ DBT, 50% Relative Humidity
Supply air Condition	14.5℃ DBT, 13.2℃ WBT
Ventilation air Quantity/person	*15CFM (7L/s) per person

*As per ASHRAE criteria [21]

Table 2 Specifications of Recirculating Air Conditioning Unit A771

Equipment Name	Recirculating Air Conditioning Unit A771
Working cycle	Vapor compression cycle
Refrigerant	R134a
Compressor	Hermetically sealed compressor,1.37KW work input
Condenser type	Air cooled condenser
Throttling device	Thermostatic expansion valve (TEV)
Arrangement of components	Intake orifice, mixer, preheating coil, steam injectors, cooling and dehumidification unit, reheater coils, supply fan, split damper (to control the amount of recirculated air with outdoor air)

Table 3 Details of selected building

Parameter	Value
Latitude	31.560 N
Longitude	74.327 E
Building Type	Corporate office
Construction Type	Modern High Rise
No. of Blocks	2
Total floors	17 (15 office floors+2 general use ground floors)
Ceiling Height/floor	11’
Avg. Occupancy/floor	60
Total cooling load	500TR
Type of System	Vapor Absorption Cycle
No. of Chillers	2 steam fired chillers of 250TR capacity each
No. of AHUs	17

3.3 Cooling Coil Design

The e-NTU is used to design the heat exchanger because the inlet temperature of both air and water is known. The cooling coil is designed as per following procedure [22]. The overall heat transfer coefficient U_o calculated on basis of air side heat transfer area is given by

$$\frac{1}{U_o} = \frac{1}{h_o \eta_{so}} + \frac{1}{h_i \frac{A_i}{A_o}} \quad (3)$$

Where h_o and h_i are outside (air side) and inside (water side) convective heat transfer coefficients, A_o and A_i are heat transfer areas on outer and inner sides and η_{so} is surface effectiveness of fins.

The water side heat transfer coefficient h_i is calculated by the famous Dittus-Boelter equation [22, 23]

$$h_i = 0.023 \frac{K}{D} Re_D{}^{0.8} Pr^{0.3} \quad (4)$$

Where K is thermal conductivity of water, D is inner diameter of tubes, Re_D is the Reynolds Number and Pr is Prandtl Number.

The air side heat transfer coefficient is calculated by the expression

$$h_o = J G_c c_p \left(\frac{\mu\, c_p}{k}\right)^{-2/3} \quad (5)$$

J is a dimensionless factor, G_c is mass velocity of air at narrowest cross section, c_p is specific heat of air, μ and k are dynamic viscosity and thermal conductivity of air respectively.

The surface effectiveness η_{so} is calculated for typical aluminum fins by following expression

$$\eta_{so} = 1 - \frac{A_f}{A}(1 - \eta) \quad (6)$$

Where η is fin efficiency

The $\frac{A_i}{A_o}$ is ratio of water side heat transfer area to air side heat transfer area. It is calculated as

$$\frac{A_i}{A_o} = \frac{\pi D_i}{x_a x_b\, \sigma} \quad (7)$$

χ_a is vertical (center to center) distance between tubes rows and χ_b is horizontal (center to center) distance between tube rows and σ is ratio of minimum free flow area of finned passage to frontal area of exchanger.

The geometric configuration of the cooling coil is found as follows

$$\dot{Q}_s = \dot{m}_a \, x \, c_p \, x \, \Delta T_a \quad (8)$$

$$\Delta T_w = \frac{\dot{Q}_s}{\dot{m}_w \, x \, c_p} \quad (9)$$

$\dot{Q}_s$ is sensible heat of air to be removed

The fluid capacity rates are determined as

$$C = c_p \, x \, \dot{m} \quad (10)$$

And the ratio C_{min} / C_{max} is determined.

The effectiveness is calculated as

$$\varepsilon = \frac{Actual\ temperature\ difference}{Max\ possible\ temperature\ difference} \quad (11)$$

Using value of ratio C_{min} / C_{max} and ε, number of thermal transfer units NTU is determined and the area A_o is calculated from following expression.

$$NTU = \frac{U_o A_o}{C_{min}} \quad (12)$$

The total volume V of heat exchanger is

$V = \frac{A_o}{\alpha}$ (13) $A_{fr} = \frac{Q^{\cdot}}{V_{fr}}$ (14)

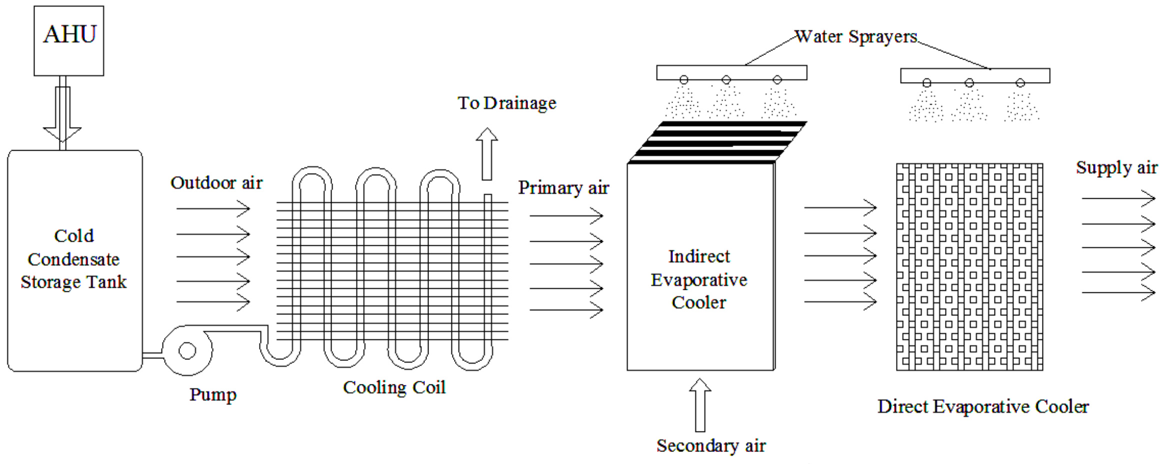

Fig. 4 Schematic diagram of three stage system of cooling coil and indirect-direct evaporative cooler combination

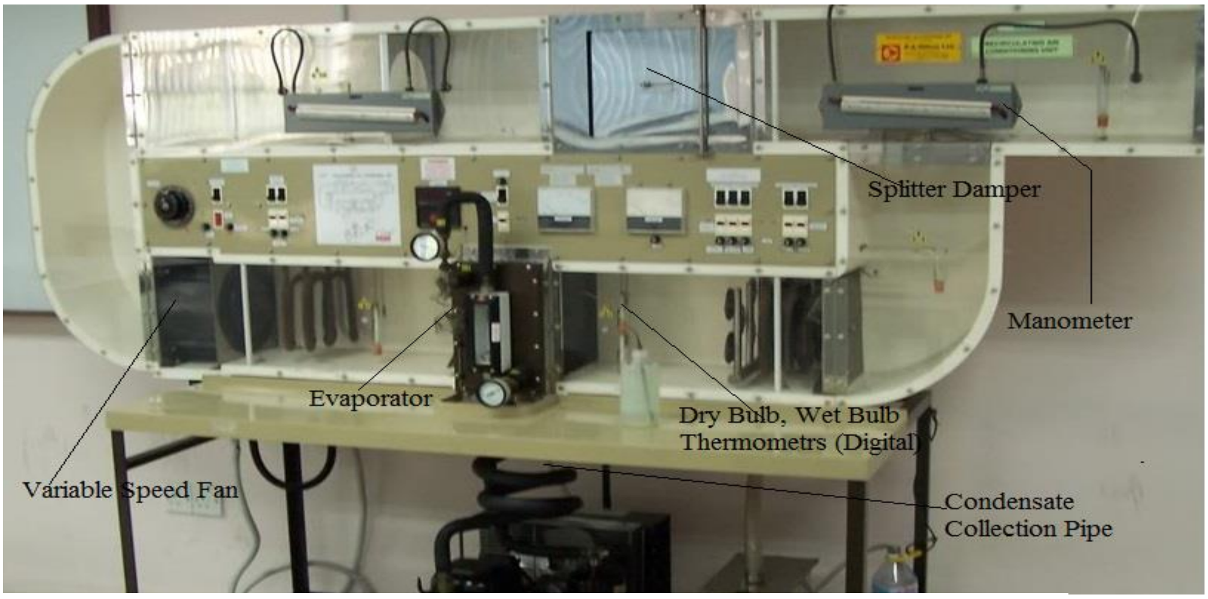

Fig. 5 Recirculating Air Conditioning Unit A771

The frontal area of the heat exchanger is

Where, $Q^{\cdot}$ is volume flow rate of air in CFMs and V_{fr} is frontal velocity of air. The depth of heat exchanger L is

$$L = \frac{V}{A_{fr}} \quad (15)$$

And number of tube rows Nr is

$$N_r = \frac{L}{x_b} \quad (16)$$

The air side pressure loss ΔP_0 is calculated by

$$\Delta P_0 = \frac{{G_c}^2}{2g\rho_1}\left[(1+\sigma^2)\left(\frac{\rho_1}{\rho_2}-1\right)+f\left(\frac{A}{A_c}\right)\left(\frac{\rho_1}{\rho_m}\right)\right] \quad (17)$$

Where f is friction factor determined on the basis of a dimensionless factor FP

$$\mathrm{FP} = {Re_D}^{-0.25}\left(\frac{D}{D^*}\right)^{0.25}\left[\frac{\chi\,a-D}{4(s-y)}\right]^{-0.4}\left[\frac{\chi\,a}{D^*}-1\right]^{-0.5} \quad (18)$$

Fig. 6 The selected Multi Storeys Building

Here, $\frac{A}{A_f}$ is ratio of total area to minimum flow area inside coil and ρ_m is average of air densities at inlet and outlet to the coil. The subscripts i and o designate the inlet and outlet of the coil respectively.

3.4 Indirect Evaporative Cooler (IDEC)

The performance of IDEC is evaluated according to following parameters as per ANSI/ASHRAE Standards (133-2008/143-2007) [24-26].

The wet-bulb effectiveness, ε_{wb} an index of fact that how close the IEC can make the DBT of primary air to the WBT of incoming secondary air.

$$\varepsilon_{wb} = \frac{t_{p,DBT,i} - t_{p,DBT,o}}{t_{p,DBT,i} - t_{p,WBT,i}} \tag{19}$$

$$Q_T = c_p m_p (t_{p,DBT,i} - t_{p,DBT,o}) \tag{20}$$

The coefficient of performance COP of the IDEC is the ratio of total cooling capacity to its power consumption

$$COP = \frac{Q_T}{W_{in}} \tag{21}$$

The rate of water evaporation V_w in the wet channels of IEC is dependent on the mass flow rate and the increase in the humidity ratio of secondary air.

$$V_w = \frac{V_s \rho_s (w_{s,o} - w_{s,i})}{\rho_w} \tag{22}$$

"Evaporative Cooler Efficiency Ratio" (ECER) which is ratio of cooling delivery volume to power consumption of IDEC is calculated as, [26, 27]

$$ECER = \frac{m_p c_p (t_r - \varepsilon_{wb}(t_{p,DBT,i} - t_{p,WBT,i}))}{W_{in}} \tag{23}$$

Where, t_r is the room temperature.

3.5 Direct Evaporative Cooler

Various parameters influence the working of direct evaporative cooler like

- Wet bulb depression (WBD) of inlet air
- The material and geometry of evaporative pad
- Inlet air face velocity
- Temperature of evaporating water

The outlet dry bulb temperature of air from the DEC is calculated as follows

$$T_{DBT,o} = T_{DBT,i} - \varepsilon_{wb}(T_{DBT,i} - T_{WBT,i}) \tag{24}$$

4. MODELLING ON TRNSYS

In order to analyze the performance of the three stage system for a small electrical equipment room, the system is modeled on the TRNSYS (version 16) as shown in Fig. 7. The first component is a weather file module which has built in weather data for many cities of world. The component which contains the weather data for city of Lahore in Pakistan is Type109-TMY2. The name of weather data file for Lahore is PK-Lahore-416400.tm2.

The next component is Type5e which is used as cooling coil. It is a zero capacitance sensible heat exchanger, which demands the inlet temperatures and flow rates of the cold and hot fluids as input parameters and calculates the outlet temperatures of both fluids along with the effectiveness of this heat exchanger for a given overall heat transfer coefficient.

Two new components named as Indirect E.C and Direct E.C are developed on TRNSYS Simulation Studio for indirect and direct evaporative coolers respectively to determine the outlet Dry bulb temperature of air across these stages of system. The inputs, outputs and parameters of the IDEC and DEC components are shown in the Table 4 and Table 5 respectively.

The component Type56a is used to calculate the cooling load and temperature of the room before and after the cooling. The building description is read by this component from a set of external files having the extensions like *.bui. The files are generated based on user supplied information by running the preprocessor program called TRNBuild. The description includes the properties of walls, windows, roof, internal heat gains like occupancy, lighting, electrical equipment and infiltration and ventilation needs. The indoor design dry bulb temperature t_r for electrical equipment room is 27℃ and the allowable relative humidity RH is up to 65%. The parameters describing geometrical configurations, construction materials and heat gains of the room are shown in the Table 6.

Table 4 Variables for new component "In direct E. C"

*Sr. No.	Name	Symbol Used in C++	Units
1	Inlet DBT of Primary air	db_p	C
2	Inlet WBT of Primary air	wb_p	C
3	Inlet Humidity ratio of Primary air	w_p	-
4	Effectiveness of IDEC	e	-
5	Air density	a_d	Kg/m^3
6	Specific Heat of air	c	KJ/kg.K
7	Volume flow rate of Primary air	vap	m^3/s
8	Power consumption of IDEC	p	KW
9	Water Density	w_d	Kg/m^3
10	Volume flow rate of Secondary air	vas	m^3/s
11	Room Temperature	t_r	C
12	Outlet DBT of Primary air	db_po	C
13	Outlet WBT of Primary air	wb_po	C
14	Outlet humidity Ratio of Primary air	w_po	-
15	Energy Efficiency Ratio	EER	-
16	Total Cooling capacity of IDEC	Q	KW
17	Water Evaporation rate	water_rate	m^3/s

* The variables from serial No. 1-3 are Inputs, from 4-11 are Parameters and from 12-17 are Outputs

Table 5 Variables for new component "Direct E.C"

*Sr. No.	Name	Symbol used in C++	Units
1	Inlet DBT of air	i.db	C
2	Inlet WBT of air	i.wb	C
3	Effectiveness of DEC	E	-
4	Outlet DBT of air	o.db	C

*The variables from serial no. 1, 2 are Inputs, 3 is a Parameter and 4 is the Output

Two psychrometric calculators, Type33e and Type33c are used to determine the unknown psychrometric properties at required stages. Type33e is used to determine humidity ratio of ambient air as the dry bulb temperature and relative humidity is known from Type109-TMY2 component. Type33c provides the wet bulb temperature of precooled air as humidity ratio and dry bulb temperature for this air is known from cooling coil outputs.

After placing all components, they are linked with each other through appropriate properties and creating an input-output relationship between the components. Finally, a graphical interface component called online plotter Type65a, is used to get the results of simulation. The analysis of system is done on average temperatures of every day, from 9am to 5pm which are the working hours of the offices as well as the room under consideration. This gives a more realistic and meaning full presentation of data analysis as the concern is to evaluate the performance of the system in working hours of the building. For this purpose, firstly, the analysis is done hourly basis for 6 months, a total of 4392 hours (from hour No.2161-6552 of the year). Then data for 8 hours is extracted for 183 days of the summer duration to convert that hourly data into daily data based on average of 8 working hours.

5. RESULTS AND DISCUSSION

The investigation of a multistep alternate cooling system of a pre-cooler, indirect-direct evaporative cooling using chilled water condensate has been done for climatic conditions of Lahore (Latitude 31.55N, Longitude 74.33E, elevation from sea level 706ft). The city of Lahore experiences summers for a duration of 6 months, i.e. from April to September.

Fig. 8 shows the results of psychrometric analysis done on Psychrometric Analyzer Program in order to calculate the condensate rate. The difference between humidity ratio of mixed air condition M and supply air condition is 0.001348kg/kg. The condensate collection rate comes out to be 580 L/hr. i.e. 1.16L/hr./ton of cooling capacity. These results correspond to calculations done with Recirculating Air Conditioning unit A771, which gives a condensate collection rate of 1.35L/hr/ton. John. A Bryant et al. [7] reported a condensate collection rate of 1.2 L/hr./ton of refrigeration load, for an institutional building in Doha, Qatar (climate; hot and humid). The flow rate of water for cooling coil is taken as 0.15 L/s (540L/hr) on the basis of hourly condensate collection rate. The supply air flow rate comes out to be 860CFM (0.49kg/s) based on design cooling load of the room.

Fig. 9 shows the required supply air quantity at various supply air temperatures for the design cooling load. As this a system not using the conventional split type air conditioners, operating the system with low temperature air i.e. around 13℃ was not possible, so required supply air temperature is taken as 19℃. The effectiveness of IDEC is taken as 0.9 [26, 28] and that of DEC stage is taken as 0.77 [29].

The results of simulation of the cooling system on TRNSYS are shown in the Fig. 10 to Fig. 13.

The trend of variation of temperature of air after every stage of the cooling system for the month of April May, June, July, August and September is shown in Fig. 10(a) to Fig. 10(f) respectively. These figures also represent the temperature differences between the stages of system. For the months of May and June, the temperature difference of air across cooling coil is around 7℃ and for last three months of summer, this value is about 5.5℃. The temperature drops across two stage evaporative cooler for month of May is highest, i.e. 10.5℃ and is lowest for the months of July, August and September in range of 3-4℃ only. The average highest value of overall temperature difference i.e. 16.65℃ is for month of May and average lowest values are for months of August and September, i.e. 7.92℃ and 7.95℃ respectively. The data shows that lowest supply air temperatures are available in April. For the month of April, May and nearly half of the month of June, the system can provide the air at required supply air temperature of 19℃ or below. For the rest of summer duration, the supply air is above the required temperature. The major reason is the higher humidity in these months which makes indirect-direct cooing stage ineffective. The varying trend of two decisive parameters i.e. room total cooling load and air temperature after cooling, that evaluate the performance and suitability of system for different months are shown in Fig. 11 and Fig. 12 respectively. The highest cooling loads are observed in the month of June approaching 13000 BTU/hr and lowest cooling loads occur in April around 10,500 BTU/hr. For first two and a half months of summer duration, the three stage system maintains the room inside design temperature, i.e. 27℃ or below, whereas for months of July, August and September, the average room temperature is above 30℃.

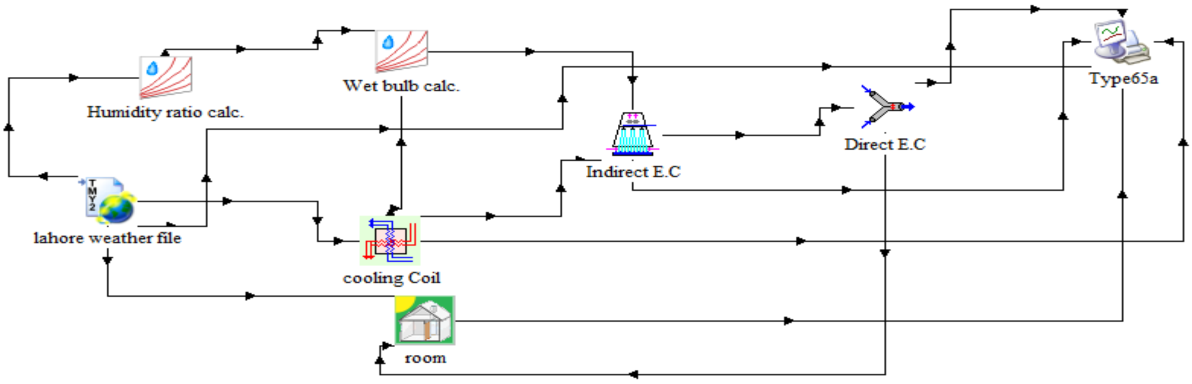

Fig. 7 The Simulation Flow diagram for the three stage cooling system

Table 6 Parameters describing the Electrical Equipment room in building model Type56a in TRNBUILD

External Heat Gains				Internal Heat Gains
Walls			Window	
Direction	Area (m^2)	Construction		
[a]East	15.33	0.5" Cement Plaster + 8" Common Brick + 0.5" Cement Plaster	No window	1. Occupancy load= 1 person, office work activity level
[a]West				2. Lighting load 5W/m^2
[b]South	10.22		Single clear glass Area 3.9 m^2	3. Electrical Equipment Load, 2 Data servers of 1000W heat dissipation each
[a]North			No window	

Note: (a) Internal/Partition wall; (b) Exposed wall

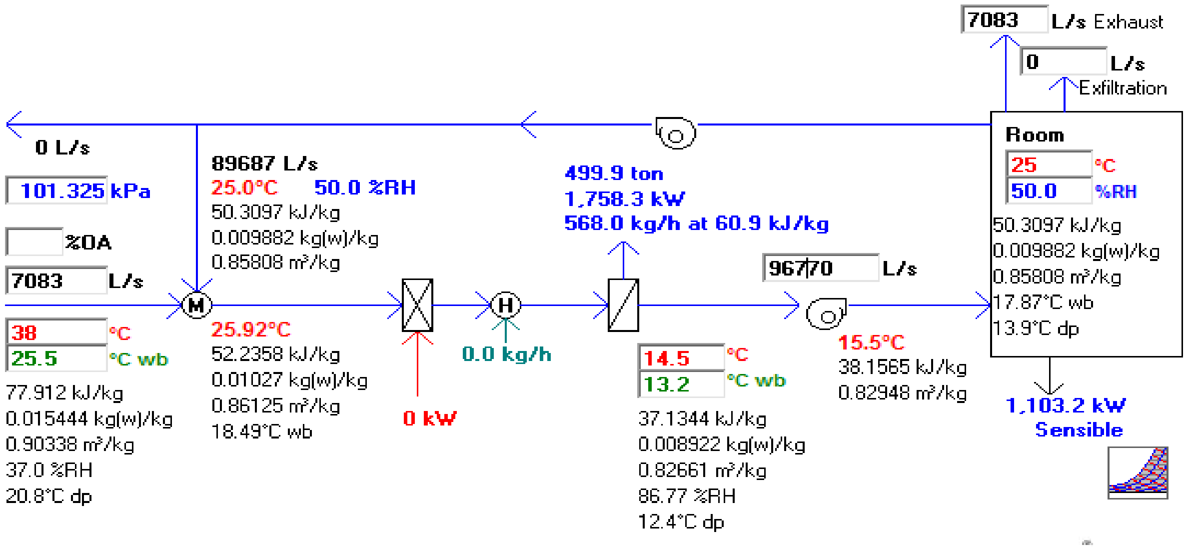

Fig. 8 Psychrometric Analyzer program showing results of analysis for 500TR cooling load

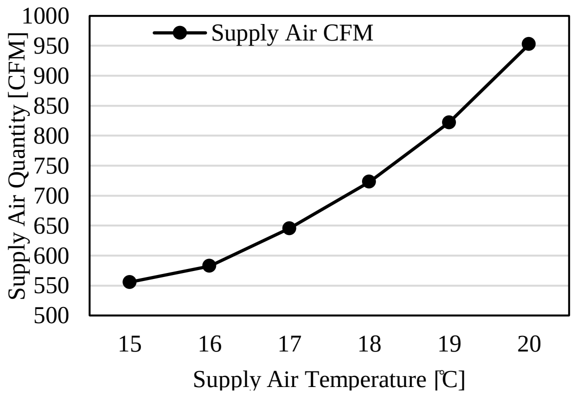

Fig. 9 Supply Air quantity at various Supply Air Temperatures for Design Cooling Load

Fig. 13 describes the variation of Energy Efficiency Ratio EER of the overall system for total sensible heat transfer from the air in the three stages. The total power consumption of the system is nearly 750W. This consumption consists of water circulation pump of 200W for cooling coil, a fan of 250W to make air flow through the three stages of system. Two pumps of 100W each for water spray and flow in indirect-direct stages. A 100W fan for working air flow

through IDEC stage. Results depict that month of May has average highest value of EER, i.e. 37.32 because it is the month with lowest relative humidity that makes indirect-direct evaporative cooler highly effective due to larger values of wet bulb depression. The months of August and September have lowest values of EER in the range of 17.8 because evaporative cooling has least contribution to

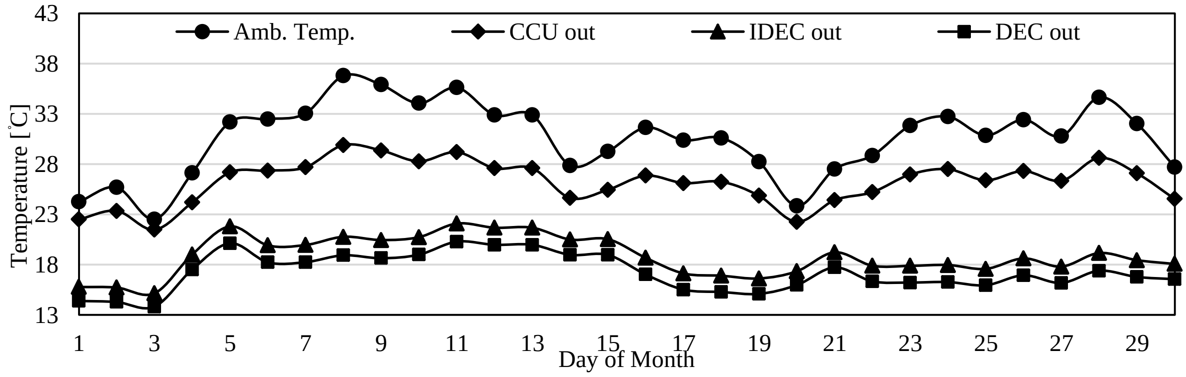

Fig. 10(a) The Temperature variation across three stages for the month of April

overall temperature drop due to higher humidity. The Seasonal Energy Efficiency Ratio, SEER of the system comes out to be 25.95.

A comparison of operational cost of the analyzed 3 staged system and conventional Vapor Compression based split air conditioners in Lahore show that this multi stage alternate cooling system has 50% less energy consumption than the conventional VCC split air conditioners. A typical 1tonn split AC consumes around 1500W. The average electricity unit price is nearly Rs.13. This means a saving of 50% of operational cost for the 2.5 months' duration when this system works efficiently to meet all the demands. The total units per month consumed by the split AC are 360 units with electricity bill of Rs.4680 per month. For 2.5 months' duration, units consumed by alternate cooling system are 180 units per month. The total savings in this duration are of Rs.5850.

Table 7 represents the summary of the results on monthly average basis.

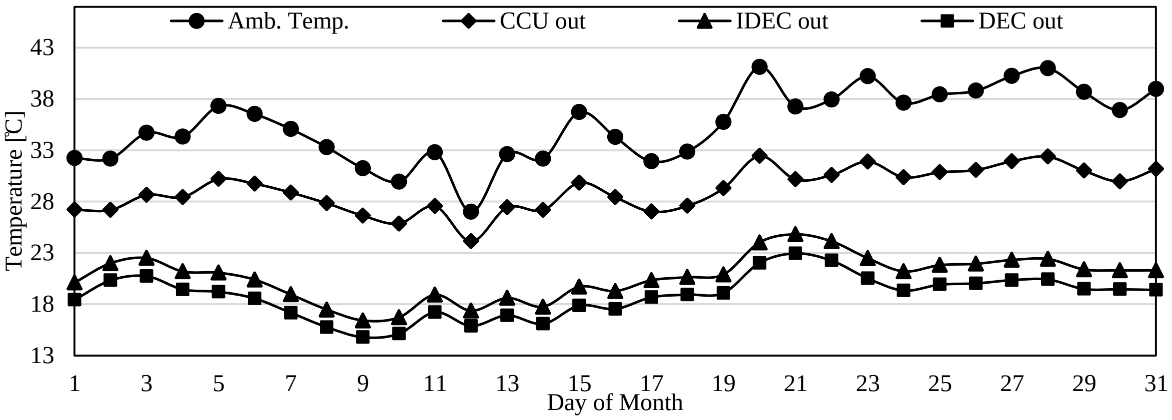

Fig. 10(b) The Temperature variation across three stages for the month of May

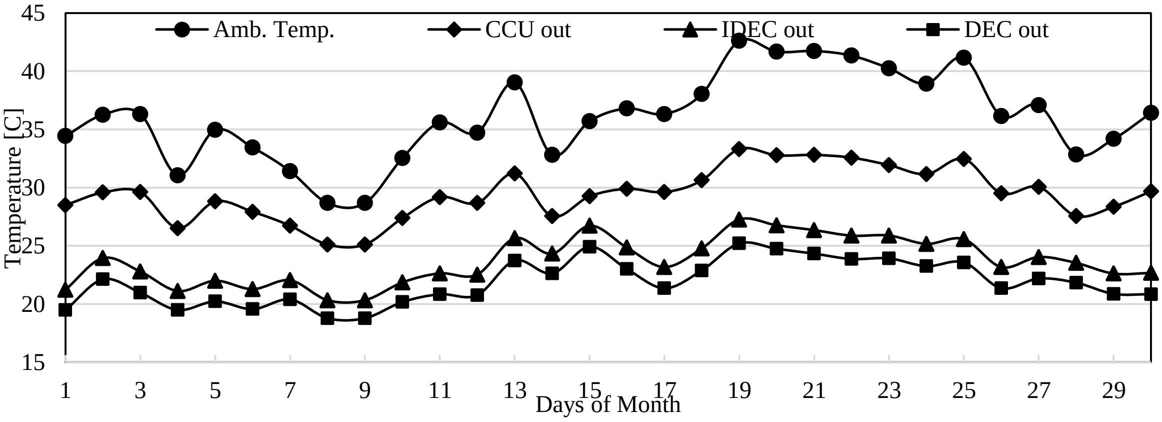

Fig. 10(c) The Temperature variation across three stages for the month of June

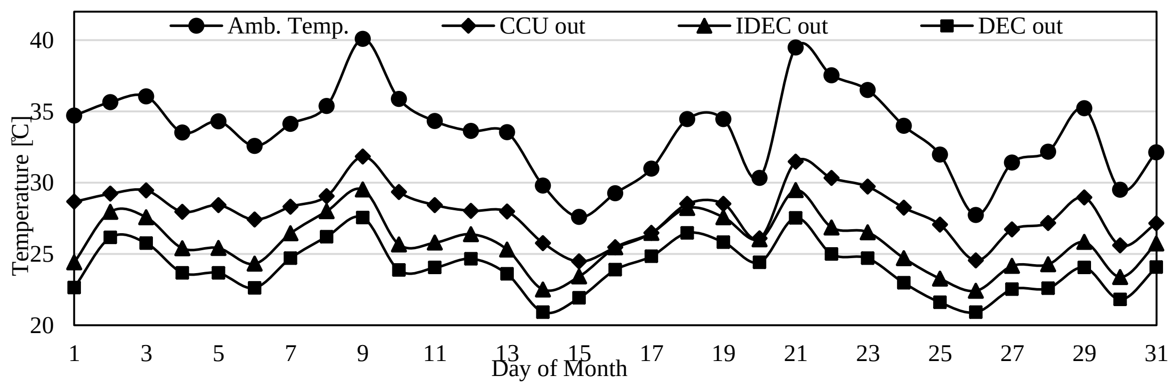

Fig. 10(d) The Temperature variation across three stages for the month of July

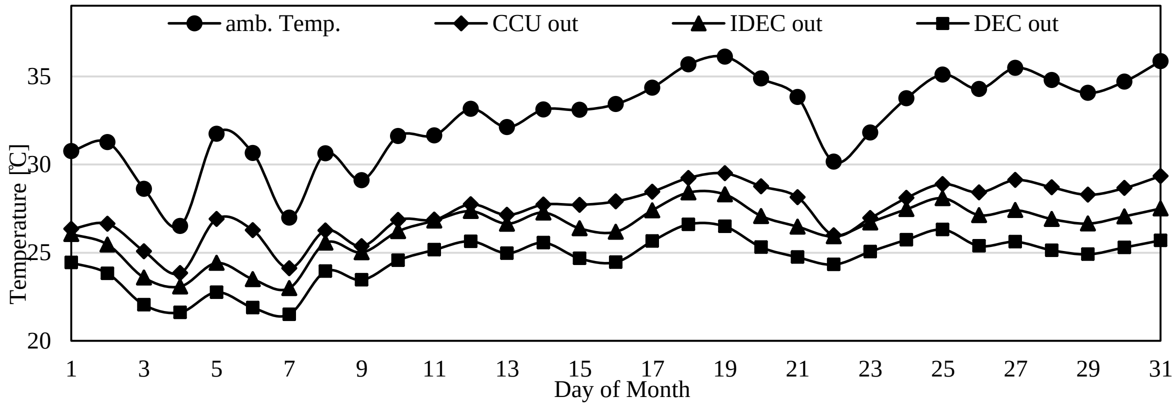

Fig. 10(e) The Temperature variation across three stages for the month of August

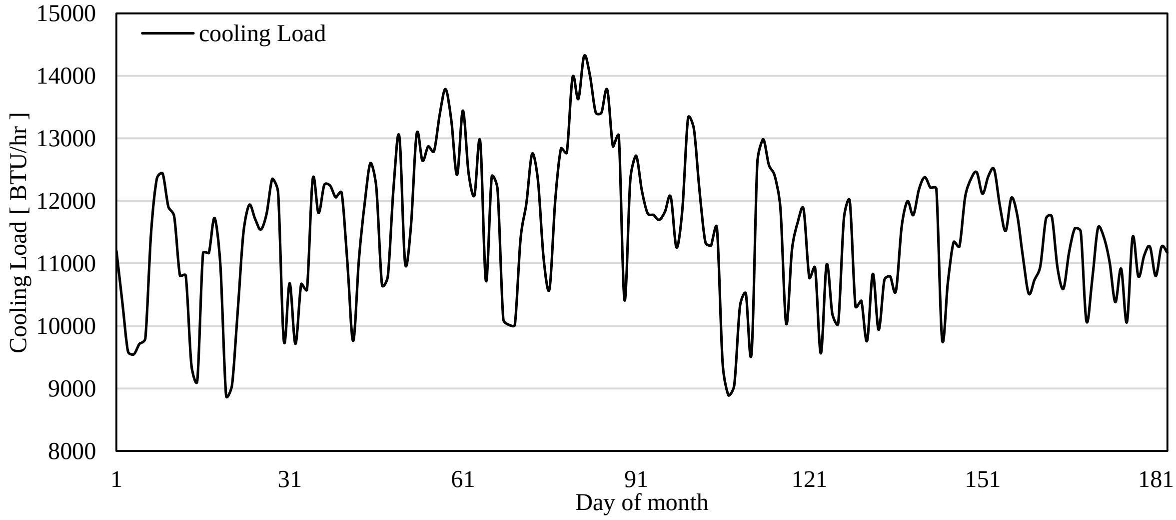

Fig. 10(f) The Temperature variation across three stages for the month of September

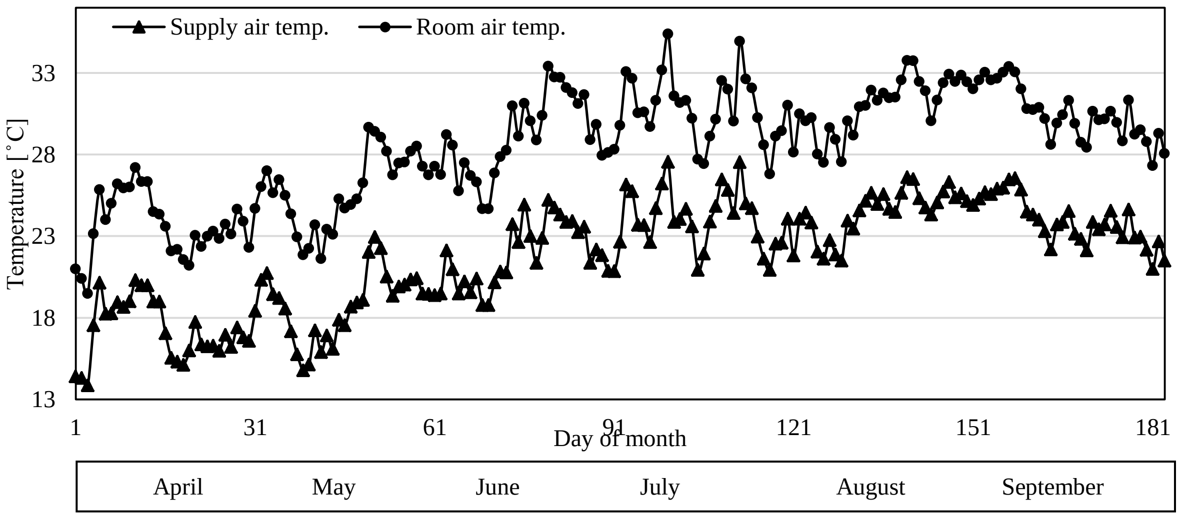

Fig. 11 Variation of Room Cooling Load for six months

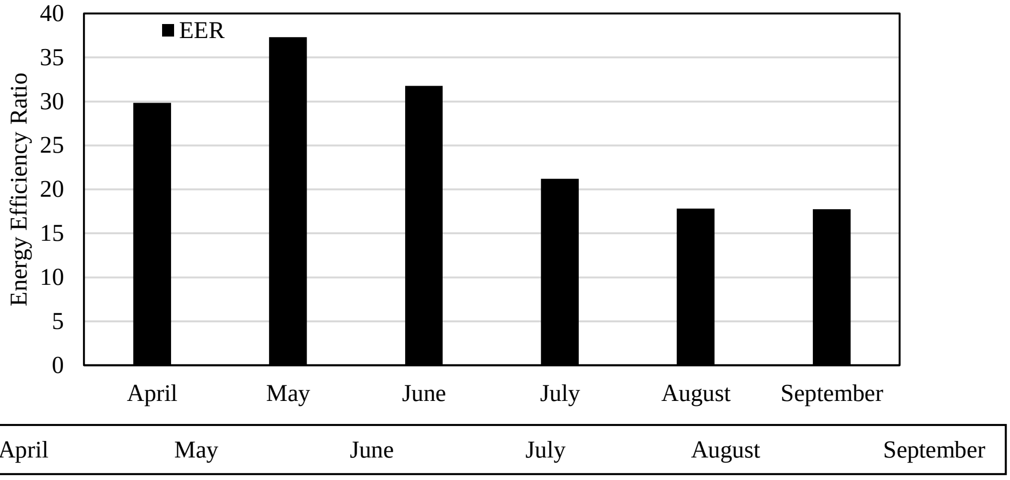

Fig. 13 Monthly Variation of EER of the total system

Table 7 Summary of results

		Outlet Dry Bulb Temperatures			Temperature Differences b/w Stages				Room Conditions	
Month	Amb. Temp.	CCU	IDEC	DEC	Amb.-CCU	CCU-IDEC	IDEC-DEC	Amb.-Supply air	Cooling load	Room temp.
	℃	℃	℃	℃	℃	℃	℃	℃	BTU/hr	℃
April	30.6	26.2	18.8	17.2	4.3	7.4	1.6	13.3	10655	23.5
May	35.5	29.1	20.6	18.8	6.4	8.5	1.8	16.7	11938	25.8
June	36.0	29.4	23.7	21.9	6.6	5.8	1.8	14.2	12285	29.1
July	33.5	27.9	25.8	24.0	5.5	2.2	1.7	9.5	11417	30.7
August	32.6	27.4	26.3	24.6	5.2	1.1	1.7	8.0	11262	31.2
September	31.7	26.9	25.5	23.8	4.8	1.4	1.7	7.9	11128	30.4
6 Months Average	33.3	27.8	23.4	21.7	5.5	4.4	1.7	11.6	11448	28.5

6. CONCLUSIONS

The performance analysis of an alternate three stage evaporative cooling system has been carried out with objective of reduced energy consumption as compared to conventional small unit air conditioners for achieving inside design conditions. The simulation results (on daily and monthly basis) of the cooling system on TRNSYS have been obtained. Following are the conclusions of the analysis.

- The city of Lahore has highly variable summer weather conditions that change from April to September such that the months April, May and half of the month of June have lower relative humidity with temperature rising gradually from April to June and getting to peak values in the month of June. Then from July to September, the temperature starts decreasing slowly with rapid increase in values of humidity which adds latent loads.
- The variation of psychrometric properties of air e.g. Dry bulb temperature, Wet bulb temperature, humidity ratio and relative humidity greatly influence the performance of system because it operates on 100% outdoor air.
- The cooling system has best performance parameters in the month of May. The overall temperature difference and EER have their peak values of 16.7℃ ad37.32 respectively in this month. The average value of room air temperature is 25.8℃. All this is because May is the driest month with relative humidity in the range of 20-30%, so evaporative cooling plays a major role in temperature drop of air i.e. 10.3℃, which adds to the total cooling done by system.
- The months of August and September have the poorest performance of system because of highest humidity ratios at relatively lower temperatures than June. That is why the lowest values of overall temperature drop and EER, i.e. 8℃ and 17.8 respectively exist in these months. Moreover, three stage system is unable to achieve the inside design temperature of 27℃ indicating that these months require conventional air conditioners for cooling.
- For the months in which the alternate cooling system performs well to fulfill all the demands, more than 50% energy and cost savings are achieved as compared to conventional air conditioners.
- The most important fact is that the higher relative humidity, not the higher temperature, is major cause of the inefficient operation of three stage system in the months of July, August and September

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