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Thermodynamic Analysis of hybrid Air-Conditioning System for Variable percentage mixing of re-circulates air

(Hot and Dry weather)

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ABSTRACT

This paper presents a study on hybrid air conditioning systems. Mainly the system has two types of cooling system like VCRS and evaporating cooling system. In this paper the air conditioned space will be kept 25℃ dry bulb temperature (DBT) and 50% relative humidity. For maintaining space condition thermodynamic analysis is being done for 44.8℃ DBT and 12% relative humidity of environment. If percentage mixing of re-circulated air changed, the hybrid air conditioning equipment like cooling coil change own temperature to maintain the room condition .The paper also shows the variation in supply condition of air, volume of cellulose cooling pad of evaporating cooler, temperature of cooling coil in hot and dry weather conditions, with respect to the percentage mixing of re-circulated air (50% to 80%).

Keywords: *Cooling coil, Cellulose cooling pad Evaporative cooling, VCRS, and Air Supply condition.*

1. INTRODUCTION

The scope of air conditioning is very wide and its applications are very diverse and literally thousands of scientists and engineers have contributed towards its development. A substantial amount of work has been done on various aspects of air-conditioning systems. The Air conditioning system plays an vital role in production, transportation and household use. The human body can be considered as thermal machine with 20% thermal efficiency. The remaining 80% must be disposed off from body to the surrounding which depends upon temperature and humidity of surrounding otherwise the accumulation of heat result and causes discomforts in the form of body pain, heat cramp or shocks, heat stroke, heat exhaustion. The psychometric process for air conditioning for hot and dry weather is cooling and humidification. In which air is first partial cool and humidify by evaporating cooling and after that is cool by

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sensible cooling from cooling coil. For this the hybrid air conditioning system is required .It have following components.(1) Fan(2)Desiccant wheel (3)Cooling pad(4)Cooling coil(5)Filter (6)Air flow damper.

Shankar Kumar et al [1] worked on the actual position of equipment used in year round air conditioning system. They also studied about the parameters on which the system depends. Shankar Kumar et al[2] studied the out let condition of exit air with different atmospheric relative humidity of year round air conditioning system. Shankar Kumar [3,4] also worked on outlet condition of air corresponding to variable values of BPF of cooling coil as well as RSHF .F.Moukalled et al [5] reported, about the use of CFD for predicting and improving the performance of air conditioning equipment .They reported a full numerical model for the concurrent forecast of velocity, temperature and humidity of air flowing in an air conditioning unit.

Kulkarni and Rajput [6] studied about the theoretical performance analysis of cooling pads of different materials for evaporative cooler. The material have been considered, rigid cellulose, corrugated paper, corrugated high density polythene and aspen fiber .It has been observed that the saturation efficiency decreases with increasing mass flow rate of air .It also seen that material with higher wetted surface area gives higher saturation efficiency. E.velasco et al [7] study about actual evaporating cooling method .They explain that when in an isolated system water and air supposed to be in contact, if air gains enthalpy then water lose it, being cooled, while if air looses enthalpy, water would be heated .Thus in a process where air and water are in contact, water will always tend to adiabatic saturation temperature. Lechner[8] found in this study that, direct evaporative cooling method is the oldest and the simplest type of evaporative cooling in which the outside air is brought into direct contact with water. Xuan et al. [9] did theoretical study on evaporative cooling to understand the heat and mass transfer processes in evaporative cooling for predicting the process outputs under different working conditions. Many studies were conducted on numerical simulation of heat and mass transfer of DEC. Zhang and Chen analyzed [10] the heat and mass transport processes in DEC and developed a basic physical model for the DEC, in which the process air was forced to flow above a wet plate with simultaneous heat and mass transfer. Qiang et al. [11] recognized a neutral network model to predict the air handling performance of DEC under different working conditions. The direct cooling technology using water evaporation is broadly used for environmental control in agricultural buildings. Zhang [12] studied the heat and mass transfer characteristics of a wet pad cooling apparatus by considering whole evaporation of the spraying water and also did comparable analysis of the heat and mass transfer processes in DEC and cooling tower. Du et al. [13, 14] obtained formula of efficiency for DEC as a function of the heat transfer coefficient, pad thickness; face velocity and specific pad surface. A mathematical model of DEC and its related boundary conditions were established and the distributions of the humidity and velocity were calculated with the SIMPLER method. J. R., et al [15] studied about the fundamental ideology of the evaporative cooling process for human thermal comfort, the ideology of operation for the direct evaporative cooling system and the mathematical development of the equations of thermal exchanges, allowing the purpose of the effectiveness of saturation. Dowdy, J.A., Karbash, N.S. [16] obtained the heat and mass

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transfer coefficient by experimentally for the evaporative cooling process through various thicknesses of rigid impregnated cellulose evaporative medium. E.V.Gomez et al [17] studied about the comparison of high grade energy required in air-conditioning system and evaporating cooling system. The weather data for analysis has taken from IMD Bhopal[19] for the year 2013.Neti, S., Wolfe,E.I.[18] calculated the effectiveness in a silica gel rotary exchanger for 0.5to 2.5m/s process air flow velocity and 20 to 30°C temperature ranged with 30 to 100% relative humidity.

NOMENCLATURE

T Dry bulb temperature of air
h Enthalpy of air(kj/kg)
T_w Wet bulb temperature
W Specific humidity (g/kg)
R Ratio of ambient air mass to Re-circulated air mass
RTH Total heat inter from atmosphere to room (kW)
RTHO Total heat out from room to atmosphere (kW)
RSHF Room sensible heat factor
RSH Room sensible heat
RSHO Room sensible heat out
CLC Cooling load Capacity (TR)
C_{pa} Specific heat of air at constant pressure (j/kg-k)
C_{pv} Specific heat of vapour at constant pressure (j/kg-k)
V Volume of cooling pad
B Bi-pass factor
T_c Cooling coil temperature
P % mixing of re-circulated air to atmospheric air
M_m Mass flow rate of air in duct after mixing (kg/s) of ambient air and re-circulating air

Subscripts
c cooling coil
ei evaporative cooler inlet
eo evaporative cooler outlet
ci cooling coil inlet
co cooling coil outlet
a ambient
s supply
r room
m mixed air

Greek Symbols

φ Relative humidity of air(%)
ρ Density of air(kg/m³)
 Saturation efficiency of evaporative cooler (%)

2. METHODOLOGY

Following assumptions and equations have been taken for analysis.

2.1. ASSUMPTION

(1) Axial heat conduction and water vapour diffusion in the air is negligible.

(2)The channels are assumed adiabatic and impermeable.

(3)The heat and mass transfer coefficient are constant.

(4)The wet bulb temperature before and after the evaporative cooler will be constant.

(5)The cooling coil only used for sensible cooling.

(6)The mass flow rate (or air velocity) will be same throughout the duct.

2.2. EQUATIONS

2.2.1 Equation of enthalpy for mixed air [2].

$$h_m = \frac{R \times h_a + h_r}{1 + R} \quad (1)$$

2.2.2 Equation of specific humidity for mixed air.

$$W_m = \frac{R \times W_a + W_r}{1 + W_a} \quad (2)$$

2.2.3 Equation of DBT for mixed air[2].

$$T_m = \frac{(h_m - 2.5 \times W_m)}{(1.01 + 0.000189 \times W_m)} \quad (3)$$

2.2.4 Equation of enthalpy for supply air [2].

$$h_s = \frac{(M_m \times h_r - RTH)}{M_m} = \frac{(M_m \times h_r - 3.5168525 \times CLC)}{M_m} \quad (4)$$

2.2.5 Equation of DBT for supply air [2].

$$T_s = T_r - \frac{RSH}{(1.148314607 \times M_m)} \quad (5)$$

2.2.6 Equation for saturation efficiency of evaporative cooler [3,4].

$$\eta = \frac{(T_{ei} - T_{eo})}{(T_{ei} - T_{wei})} \quad (6)$$

Where

$$T_{ei} = T_m \text{and } T_{wm} = T_{wei} = T_{weo}$$

2.2.7 Equation for volume of cooling pad.[2]

$$V_s = \frac{(C_{pa} + W_{ei} \times C_{pv}) \times M_m \times \ln(1 - \eta)}{(-h_c \times 370)} \quad (7)$$

(b)For hot and dry weather condition.

$W_m = W_{ei}$

2.2.8 Equation for percentage mixing of re-circulated air with atmospheric air.

$$P = \frac{100}{(1+m_1/m_2)} = \frac{100}{(1+R)} \quad (8)$$

2.2.9 Equation for temperature of cooling coil.

$$T_c = \frac{(T_s - B \times T_{eo})}{(1-B)} \quad (9)$$

2.2.10 Equation for velocity of air

$$U = \frac{M_m}{(\rho_a \times A)} \quad (10)$$

3. SYSTEM DESCRIPTION

The positions of equipments used in hybrid air conditioning system are shown in the diagram (Fig.1).The analysis is done on the concept that, supply air condition is depends on the room conditions, mass flow rate, CLC, and RSHF. We take the room conditions be constant. So the supply condition will be depends on mass flow rate, CLC and RSHF. For hot and dry weather condition the supply air condition is also the outlet condition of cooling coil. Because from cooling coil only sensible cooling is done, so the inlet and outlet specific humidity of air for cooling coil will be same. The inlet condition of cooling coil is same the outlet condition of the evaporative cooler, and the WBT of inlet and outlet air of evaporating cooler will be same, which is equal to the WBT of mixed air .So the outlet condition of evaporative cooler is find with the online psychometric calculator [20]. Hence we find out the saturation efficiency of evaporative cooler and the volume of cellulose cooling pad with the help of inlet and outlet condition of evaporating cooler.

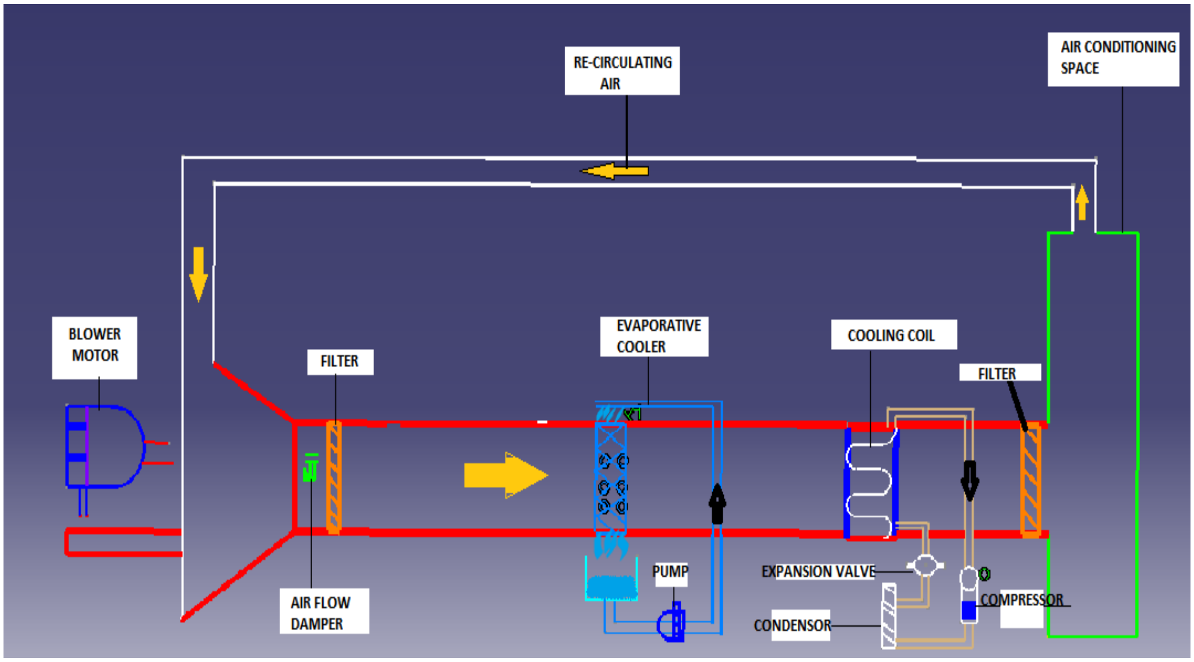

Fig.1. Diagram used for hot and dry weather condition

4. RESULT AND DISCUSSION

Figure 2 and Figure 3 indicates variation in temperature and specific humidity w.r.t. percentage mixing of re-circulated air (P) respectively. The inlet temperature of evaporative cooler or mixed air temperature decreases with the increase in value of percentage mixing of re-circulated air. The value of T_{we} also decreases with increase in mixing percentage of re-circulated air. The result shows that the supply condition is constant because the supply condition depends on the mass flow rate ,Room conditions, RTH or cooling load and RSHF value and these parameters are taken as constant . The value of specific humidity of outlet air of evaporative cooler is constant, because the specific humidity of supply air is constant and it is equal to specific humidity of outlet air of evaporative cooler. The cooling coil temperature increases, since the supply air DBT is constant and the inlet of cooling coil i.e. at the outlet of evaporative cooler DBT of air decreases and the bypass factor is constant.

Table 1: Variation of output parameter w.r.t. percentage mixing of re-circulated air at 44.8°C DBT and 12 % relative humidity of ambient air, 523m altitude , 25°C DBT and 50% relative humidity of room air, 1.25kg/s mass flow rate of air, 1TR CLC, 0.9 RSHF and 0.2 BPF.

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P (%)	50	55	60	65	70	75	80
T_m(°C)	34.98	33.89	32.81	31.72	30.64	29.56	28.47
T_{wm}	19.66	19.47	19.28	19.09	18.90	18.70	18.50
T_{eo}(°C)	31.77	31.13	30.50	29.86	29.23	28.58	27.94
T_c(°C)	20.55	20.71	20.86	21.02	21.18	21.34	21.50
T_s(°C)	22.79	22.79	22.79	22.79	22.79	22.79	22.79
W_m(g/kg)	9.13	9.31	9.50	9.69	9.87	10.06	10.25
W_{eo}(g/kg)	10.33	10.33	10.33	10.33	10.33	10.33	10.33
W_s(g/kg)	10.33	10.33	10.33	10.33	10.33	10.33	10.33
η (%)	20.97	19.18	17.09	14.78	12.04	9.02	5.36
V(cm^3)	453.05	418.06	374.90	325.95	266.17	199.67	118.56

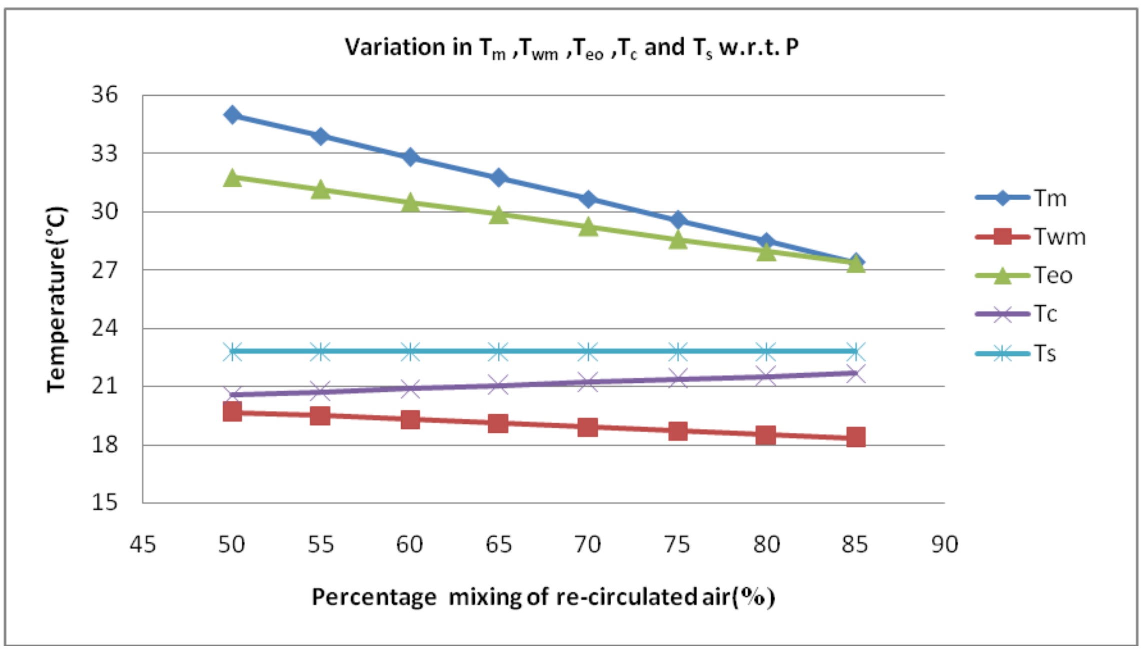

Figure 2: Variation in temperatures w.r.t. percentage mixing or re-circulated air for hot and dry weather

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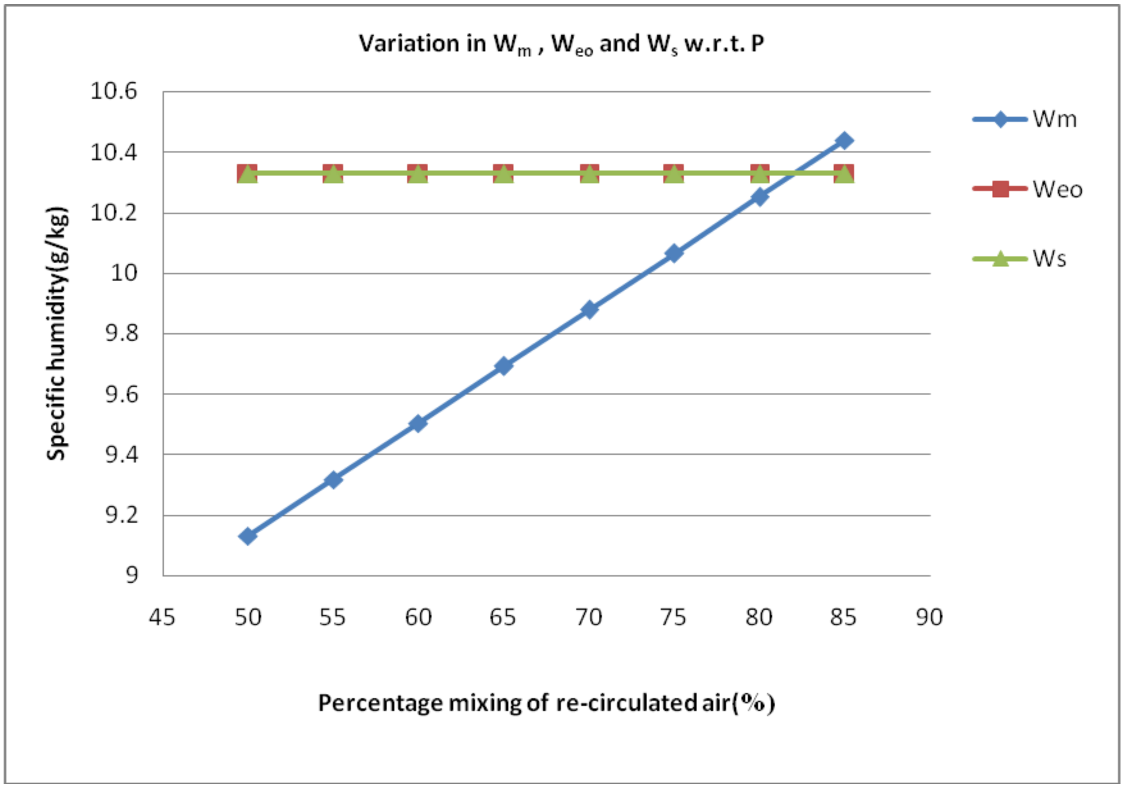

Figure 3: Variation in specific humidity w.r.t. percentage mixing of re-circulated air for hot and dry weather

The volume of cooling pad decreases with increase in percentage mixing of re-circulated air (shown in table 1), since mixed air specific humidity or inlet air specific humidity of evaporative cooler increases and outlet specific humidity is constant and which is only possible when volume of the cooling pad decreases. Because when volume or thickness decreases then the air has lesser contact time with water for same the mass flow rate. The increase in relative humidity is only allowed up to the limit at which the specific humidity of inlet air of evaporative cooler will be less than the specific humidity of outlet air of evaporative cooler. If both are same then in that condition the volume of evaporative cooler becomes zero, i.e. no requirement of evaporative cooler in this case. If inlet specific humidity of air is more than outlet specific humidity then other system is applicable.

5. CONCLUSION

The analysis of hybrid air conditioning system depends mainly on the performance of the evaporative cooler and cooling coil. The DBT of supply air with variation in percentage mixing of re-circulated air (p) is shown for hot and dry weather. It is remain constant and equal to 22.79°C with the increase P. The specific humidity of supply air also remain constant with respect to increases in P and its value is 10.33 g/kg for dry and hot weather condition for constant mass flow rate of 1.25kg/s. The volume of cellulose cooling pad decreases from 453.05to 118.56cm^3 for hot and dry with increase in P (50% to 80%). The saturation efficiency of evaporative cooler decreases from 20.97 to 5.36% for hot and dry weather condition with increase P (50% to 80%).

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