Simulation of Year Round Air-Conditioning System for Variable By-pass Factor of Heating and cooling coil

Dr. Shankar Kumar[1]

[1]*Mechanical Engineering Department, Lakshmi Narain College of Technology Excellence Bhopal, (India)*

Abstract

This paper presents a study on different kinds of air conditioning systems in comparison to existing one to use through of the year. Mainly the system imparts all three regular weather conditions. Like hot and dry, hot and wet and cool and dry. For this the out let condition will be fixed 25°C dry bulb temperature (DBT) and 50% relative humidity. In the present paper, for maintaining room condition thermodynamic simulation is being done. If BPF of cooling coil or heating coil changed, the year round air conditioning equipment like cooling coil or heating coil change own temperature to maintain the room condition . The cooling coil temperature decreases with increase in BPF value for hot and dry as well as hot and wet weather condition. The value of re-heating coil increases with the increase in BPF. The value of pre-heating coil increases with increase in BPF.

Keywords: *Cooling coil, Cellulose cooling pad, Desiccant wheel, Heating coil, Air Supply condition.*

1. Introduction

The Air conditioning system plays an important role in industry, infrastructure and domestic use. For different weather condition different air conditioning systems are used. There is no such single air conditioning system which gives constant output (comfort) throughout the whole year .The existing air conditioning system works better only in the specified weather condition. While human body requires comfort throughout the year. So solving the problem of human body due to exiting air conditioning system, year round air conditioning system is required. It have following components.(1) Fan(2)Desiccant wheel (3)Cooling pad(4)Cooling coil(5)Heating coil.

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.Zainab Hasson et al [2] presented an efficient methodology to design modified evaporative air –cooler for winter air-conditioning in Baghdad city. The performance is reported in terms of effectiveness of DEC, saturation efficiency of DEC, outlet temperature of air and cooling capacity.F.Moukalled et al [3] reported, about the use of CFD for predicting and improving the performance of air conditioning equipment. D.La.et al [4] study, rotary desiccant air conditioning system, which combines the technology of desiccant dehumidification and evaporative cooling. It evaluates the status of rotary desiccant dehumidification and air conditioning system in the following two aspects; (1) Improvement of advanced desiccant materials and (2) Optimization of system arrangement. Kulkarni and Rajput [5] studied about the theoretical

performance analysis of cooling pads of different materials for evaporative cooler. 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 [6] 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. Fatemeh et al [7] shown the modeling of a desiccant wheel used for dehumidifying the ventilation air of an air-conditioning system. By the numerical method, the performance of an adiabatic rotary dehumidification is parametrically studied and the optimums rotational speed is determined by examine the outlet adsorption –side humidity profiles. D.G. Waugaman et al [8] studied about the advantages and disadvantages of desiccant wheel. It states that the main advantage of desiccant cooing is significant potential for energy reduction and reduced utilization of fossil fuels. Kang and Maclain-cross [9] showed that dehumidification is the key factor of desiccant cooling system and the cooling COP can be significant enhanced by enhancing the performance of component.J.J. Jurinak et al and F.E.Nia et [10] premeditated about the desiccant process that in this process, fresh air is dehumidified and then sensibly and evaporatively cooled before being sent to the conditioned space. This method works without conventional refrigerants, and also it allows the use of low-temperature heat to drive cooling cycle. Camrago, J. R., et al [11] 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. [12] obtained the heat and mass transfer coefficient by experimentally for the evaporative cooling process through various thicknesses of rigid impregnated cellulose evaporative medium. E.V.Gomez et al [13] 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.[14] 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
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
T_{rh}	Temperature of re-heat coil
T_{ph}	Temperature of pre-heat coil
T_{rd}	Regeneration temperature of desiccant wheel
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
RDW	Radius of desiccant wheel

dt	Thickness of desiccant coating(mm)
N	Desiccant wheel speed
Subscripts	
Ph	pre-heating coil
rh	re-heating coil
c	cooling coil
phi	pre- heating coil inlet
pho	pre –heating coil outlet
di	desiccant wheel inlet
do	desiccant wheel outlet
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)
η	Saturation efficiency of evaporative cooler (%)
ε	Efficiency of desiccant wheel.

2. Methodology

Following assumptions and equations have been taken for analysis.

2.1. Assumption

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

(2)The adsorption heat per kilogram of adsorbed water is constant.

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

(4)The cooling coil and heating coil be only used for sensible cooling and sensible heating.

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

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

2.2. Equations

2.2.1 Equation of enthalpy for mixed air [15].

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

2.2.2 Equation of DBT for mixed air[15].

$$T_m = \frac{(h_m - 2.5 \times W_m)}{(1.01 + 0.00189 \times W_m)} \quad (2)$$

2.2.3 Equation of pre-heat coil temperature [15].

$$T_{ph} = \frac{(B_{ph} \times T_m - T_{pho})}{(B_{ph} - 1)} \quad (3)$$

2.2.4 Equation for saturation efficiency of evaporative cooler [5, 16].

$$\eta = \frac{(T_{ei} - T_{eo})}{(T_{ei} - T_{wei})} \quad (4)$$

Where (a) For cool and dry weather condition.

$T_{ei} = T_{pho}$ and $T_{wei} = T_{weo} = T_{wpho}$

(b)For hot and dry weather condition.

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

(c) For hot and wet weather condition.

$T_{ei} = T_{do}$ and $T_{wdo} = T_{wei} = T_{weo}$

2.2.5 Equation for volume of cooling pad.[5]

$$V_s = \frac{(C_{pa} + W_{ei} \times C_{pv}) \times M_m \times \ln(1-\eta)}{(-h_c \times 370)} \quad (5)$$

Where (a) For cool and dry weather condition.

$W_{pho} = W_m = W_{ei}$

(b)For hot and dry weather condition.

$W_m = W_{ei}$

(c) For hot and wet weather condition.

$$W_{do} = W_{ei} \quad (6)$$

2.2.6 Equation of re-heats coil temperature.

$$T_{rh} = \frac{(B_{rh} \times T_{so} - T_s)}{(B_{rh} - 1)} \quad (7)$$

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

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

2.2.8 Equation for temperature of cooling coil.

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

2.2.9 Area of desiccant wheel in which mixed air is passed

$$A = \frac{\pi \times (RDW)^2}{2} \quad (10)$$

2.2.10 Equation for velocity of air

$$U = \frac{M_m}{(\rho_a \times A)} \quad (11)$$

2.2.11 Equation for specific humidity of outlet air for desiccant wheel [17, 18]

$$W_{do} = (W_m - \varepsilon W_m) \quad (12)$$

3. System Diagram

The positions of equipments used in year round air conditioning system are shown in the diagram of year round air conditioning system (Fig.1).

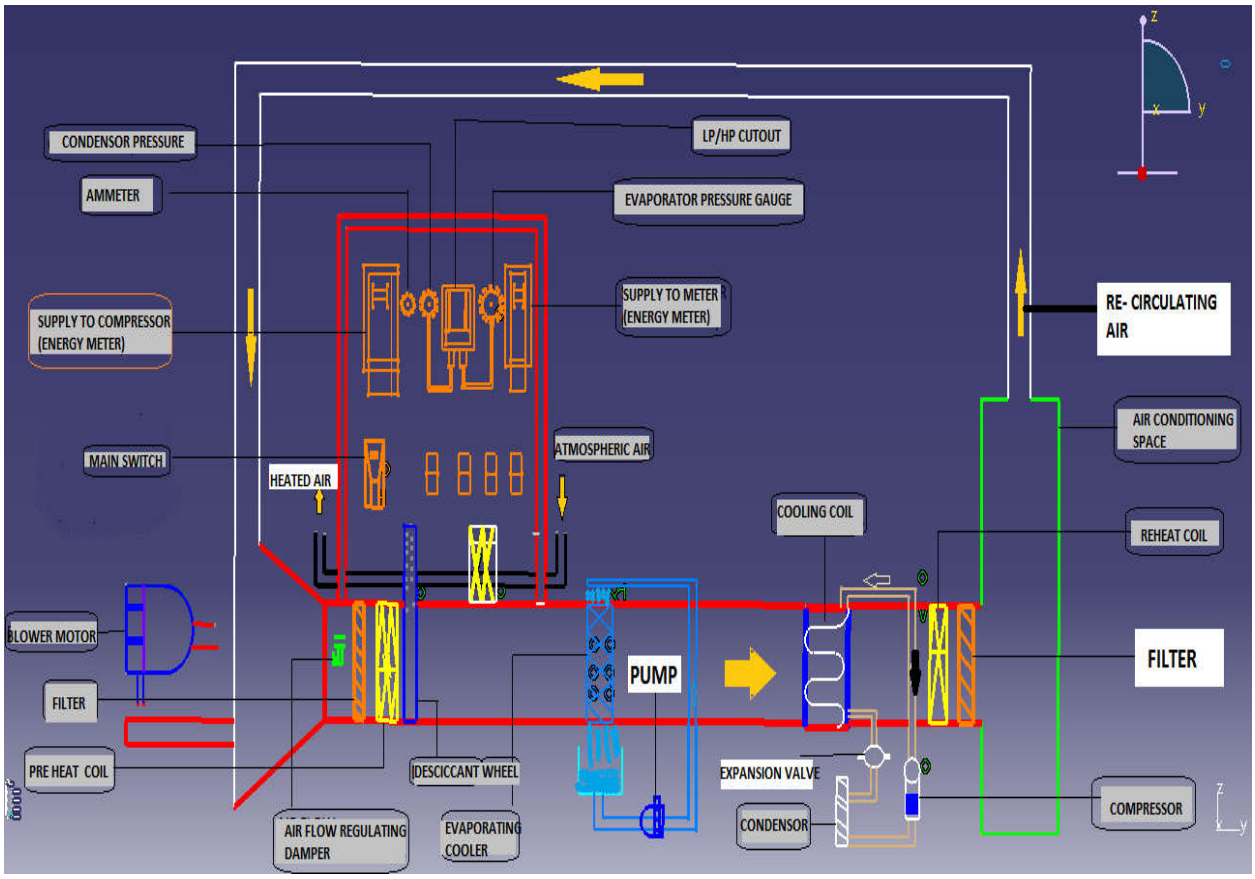


Fig.1. Diagram of year round air conditioning system

4. Result and discussion

4.1: Hot and dry weather condition.

Figure (2) and Figure (3) indicates variation in temperature and specific humidity w.r.t BPF respectively. Findings from the results are being discussed as follows;

- Our result shows that only cooling coil temperature changes w.r.t. the BPF. The cooling coil temperature decreases ,when BPF increases, because BPF will be more then more air will pass through the cooling coil without contacting the surface of it. The remaining air must have lower temperature, which is only possible when cooling coil temperature will be lower.

- T_m, T_{eo} and T_s are kept constant with increase of BPF.
- W_m, W_{eo} and W_s are also constant with increase of BPF.

Table1: variation of output parameter w.r.t. bypass factor of cooling coil (BPF) at 44.8°C DBT and 12 % relative humidity of ambient air , 523m altitude , 25°C DBT and 50% relative humidity of room, 60 % mixing of re-circulated air, 1.25kg/s mass flow rate of air , 1TR CLC and 0.9 RSHF .

BPF	0.10	0.15	0.20	0.25	0.30	0.35
T_m(°C)	32.81	32.81	32.81	32.81	32.81	32.81
T_{eo}(°C)	30.50	30.50	30.50	30.50	30.50	30.50
T_c(°C)	21.93	21.43	20.86	20.22	19.49	18.64
T_s(°C)	22.79	22.79	22.79	22.79	22.79	22.79
W_m(g/kg)	9.50	9.50	9.50	9.50	9.50	9.50
W_{eo}(g/kg)	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
η(%)	17.09	17.09	17.09	17.09	17.09	17.09
V(cm³)	374.90	374.90	374.90	374.90	374.90	374.90

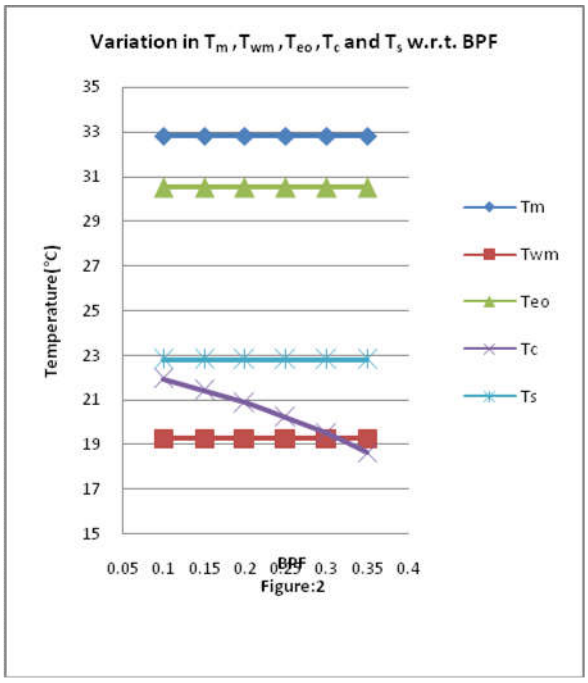


Figure:2

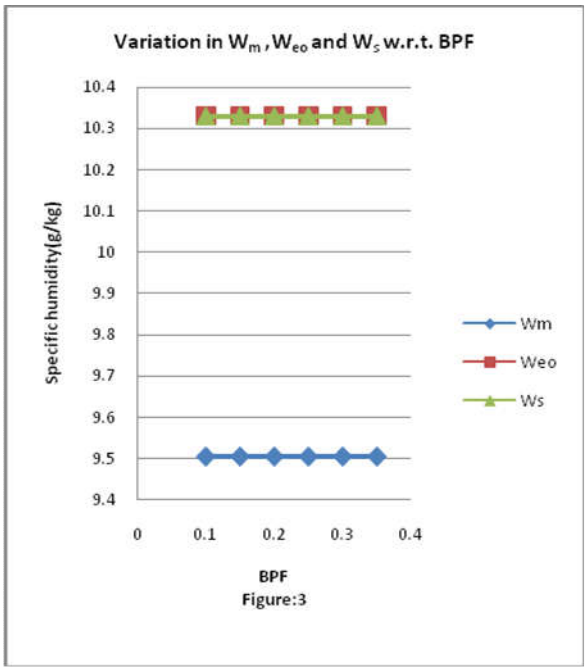


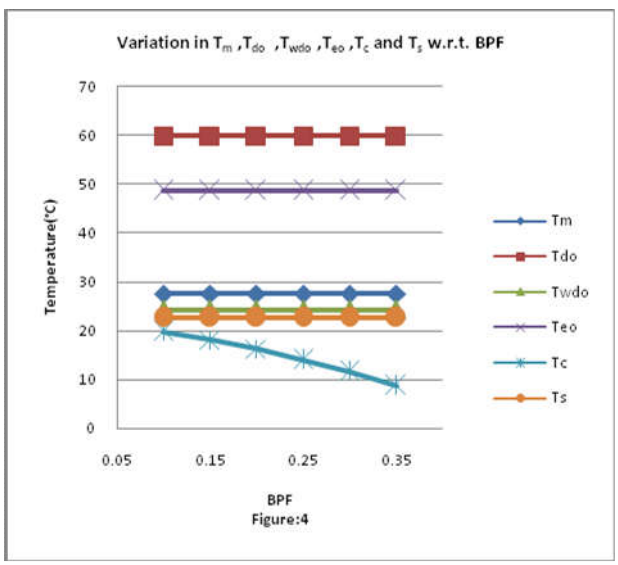
Figure:3

The table (1) shows that saturation efficiency of evaporative cooler and volume of cooling pad remain constant with increasing or decreasing value of BPF. Because the inlet and outlet condition of evaporative cooler has no effect of BPF.

4.2: Hot and wet weather condition.

Table 2: variation of output parameter w.r.t. bypass factor of cooling coil (BPF) at 30°C DBT and 95 % of relative humidity of ambient air and 523m of altitude, 25°C DBT and 50% relative humidity of air conditioned room, 60% mixing of re-circulated air, 1.25kg/s mass flow rate, 1TR CLC Value, 0.9 RSHF, 15 rph and 90°C of regeneration temperature of desiccant wheel.

BPF	0.10	0.15	0.20	0.25	0.30	0.35
T_m(°C)	27.57	27.57	27.57	27.57	27.57	27.57
T_{do}(°C)	59.91	59.91	59.91	59.91	59.91	59.91
T_{eo}(°C)	48.81	48.81	48.81	48.81	48.81	48.81
T_c(°C)	19.90	18.20	16.29	14.12	11.64	8.78
T_s(°C)	22.79	22.79	22.79	22.79	22.79	22.79
W_m(g/kg)	17.38	17.38	17.38	17.38	17.38	17.38
W_{do}(g/kg)	5.81	5.81	5.81	5.81	5.81	5.81
W_{eo}(g/kg)	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
ε	0.66	0.66	0.66	0.66	0.66	0.66
η(%)	31.35	31.35	31.35	31.35	31.35	31.35
V(cm³)	472.25	472.25	472.25	472.25	472.25	472.25



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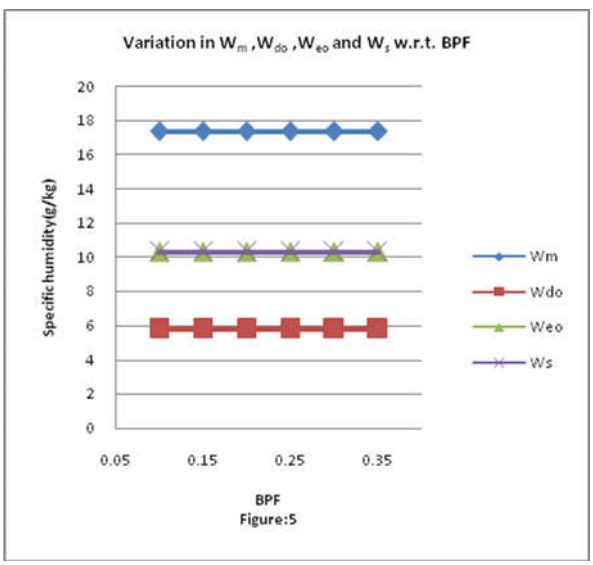

Figure:5

Figure (4) and Figure (5) indicates variation in temperature and specific humidity w.r.t BPF of cooling coil respectively. Findings from the results are being discussed as follows;

- The inlet temperature of desiccant wheel or mixed air temperature remains constant with the increase in BPF of cooling coil.
- The DBT and specific humidity of desiccant wheel outlet air becomes constant because both of desiccant wheel inlet air is constant.
- The result shows that the supply condition be constant because the supply condition is depends on the mass flow rate ,Room conditions, RTH or cooling load and RSHF value and these parameters are taken to be 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 decreases, when BPF increases, because BPF will be more, then more air will pass through the cooling coil without contacting the surface of it. The remaining air must have lower temperature, which is only possible when cooling coil temperature will be lower.

The table (2) shows that saturation efficiency of evaporative cooler and volume of cooling pad remain constant with increase of BPF. Because the inlet and outlet condition of evaporative cooler has no effect of BPF. With same region the efficiency of desiccant wheel is also remain constant with increase in BPF of cooling coil.

4.3: Cold and dry weather condition.

Table 3: variation of output parameter w.r.t. bypass factor of heating coil (BPF) at 7°C DBT of and 60 % of relative humidity of ambient air and 523m of altitude, 25°C DBT and 25% relative humidity or room, 60 % mixing of re-circulated air, 1.25kg/s mass flow rate, 3.5KW RTHO Value, 0.9 RSHF and 16°C wet bulb temperature of outlet air for pre-heat coil.

BPF	0.10	0.15	0.20	0.25	0.30	0.35
T_m(°C)	17.90	17.90	17.90	17.90	17.90	17.90
T_{pho}(°C)	26.22	26.22	26.22	26.22	26.22	26.22
T_{ph}(°C)	27.14	27.68	28.29	28.99	29.78	30.69
T_{eo}(°C)	19.00	19.00	19.00	19.00	19.00	19.00
T_{rh}(°C)	28.10	28.64	29.24	29.92	30.70	31.60
T_s(°C)	27.19	27.19	27.19	27.19	27.19	27.19
W_m(g/kg)	7.92	7.92	7.92	7.92	7.92	7.92
W_{pho}(g/kg)	7.92	7.92	7.92	7.92	7.92	7.92
W_{eo}(g/kg)	10.88	10.88	10.88	10.88	10.88	10.88
W_s(g/kg)	10.88	10.88	10.88	10.88	10.88	10.88
η(%)	70.64	70.64	70.64	70.64	70.64	70.64
V(cm³)	2064.30	2064.30	2064.30	2064.30	2064.30	2064.30

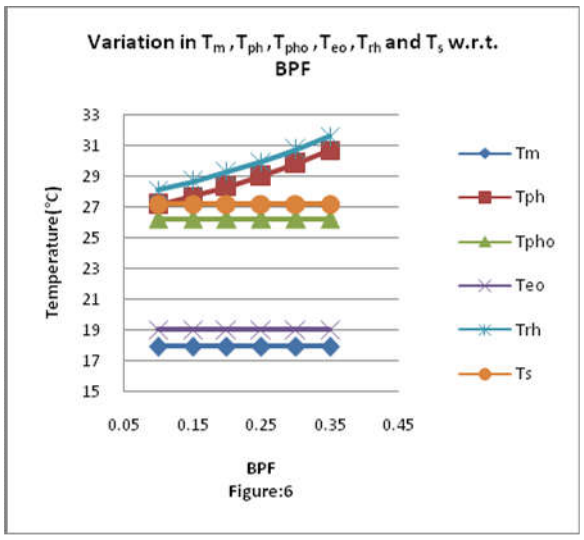

Figure:6

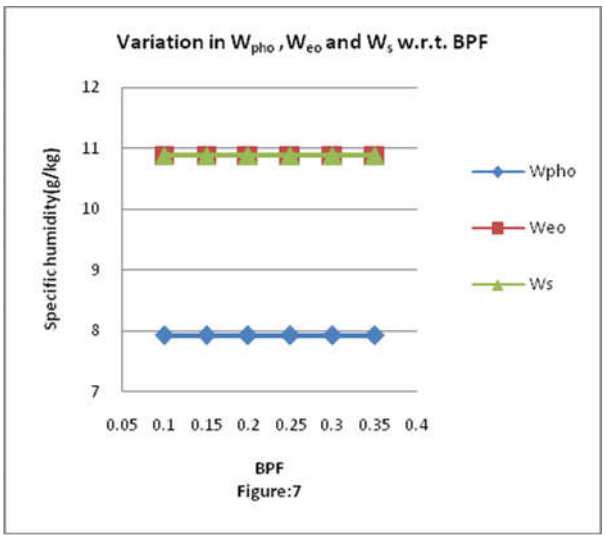

Figure:7

Figure (6) and Figure (7) indicates variation in temperature and specific humidity w.r.t BPF respectively. Findings from the results are being discussed as follows;

- The inlet temperature of pre-heating coil or mixed air temperature remains constant with the increase in BPF of cooling coil.
- For fixed WBT of outlet air of pre-heating coil, T_{pho} is constant with increase of BPF.
- The result shows that the supply condition be constant because the supply condition is depends on the mass flow rate ,room conditions, RTH or cooling load and RSHF value and these parameters are taken to be 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 temperatures of heating coils increase, when BPF increases, because BPF will be more, then more air will pass through the heating coils without contacting the surface of it. The remaining air must have higher temperature, which is only possible when heating coils temperature will be higher.

The table (3) shows that saturation efficiency of evaporative cooler and volume of cooling pad remain constant with increase of BPF. Because the inlet and outlet condition of evaporative cooler has no effect of BPF. With same region the efficiency of desiccant wheel is also remain constant with increase in BPF of cooling coil.

5. Conclusion

The analysis of year round air conditioning system depends mainly on the performance of the desiccant wheel, evaporative cooler, heating coil and cooling coil. The cooling coil temperature decreases with increase in BPF value for hot and dry as well as hot and wet weather condition. The value of re-heating coil increases with the increase in BPF. The value of pre-heating coil increases with increase in BPF.The supply air temperature remains same corresponding to increase in BPF of cooling or heating coil because its effect is reduced by changing in its own

parameter of equipment used for this system like cooling coil or heating coil temperature. Its value is 22.79°C for hot and dry as well as hot and wet weather condition and 27.19°C for dry and cool weather condition. The specific humidity of supply air remain constant with respect to increases in BPF of cooling or heating coil and its value is 10.33 g/kg for dry and hot as well as hot and wet weather condition where as 10.88g/kg for cold and dry weather condition at constant mass flow rate of 1.25kg/s. The volume of cellulose cooling pad remains constant w.r.t increase in BPF(0.1 to 0.35) for all the weather condition and its values are 374.9cm^3(hot and dry),472.2cm^3 (hot and wet),and 2064.3cm^3(cool and dry) respectively. The saturation efficiency of evaporative cooler remains constant w.r.t increase in BPF(0.1 to 0.35) for all the weather condition and its values are 17.09% (hot and dry),31.35% (hot and wet),and 70.64%(cool and dry) respectively. The efficiency of desiccant wheel remain constant (66.56%) with the increase in RSHF as well as BPF value.

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