Thermodynamic Performance of Year Round Air-Conditioning System for Variable Room Sensible Heat Factor

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℃ dry bulb temperature (DBT) and 50% relative humidity. In the present paper, for maintaining room condition thermodynamic simulation is being done. .The paper shows the variation in supply condition of air, volume of cellulose cooling pad of evaporating cooler, temperature of cooling coil in hot and dry as well as hot and wet weather conditions, temperature of preheat and reheat coil in cold and dry weather condition with respect to the RSHF (0.6 to 0.9).

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

1. Introduction

The Air conditioning system plays a significant role in industry, infrastructure and domestic use. For dissimilar weather condition dissimilar air conditioning systems are used. There is no such single air conditioning system which gives constant output (comfort) during the whole year. There are following psychometric process for air conditioning. (a)Cooling and humidification (b) Cooling and dehumidification. (c)Heating and humidification (d) Heating and dehumidification. Cooling and humidification process is used generally in summer air conditioning to cool and humidify the air. In which air is first partial cool and humidify by evaporating cooling and after that is cool by sensible cooling from cooling coil. Cooling and dehumidification process is used in summer (rainy) season air conditioning to cool and dehumidify the air. The humidification is done by two methods. (1)By desiccant wheel and (2) By cool the cooling coil up to dew point temperature. Heating and dehumidification process for dehumidification, desiccant wheel is used and for heating purpose heating coil is used. Heating and humidification process is used in cold and dry weather condition, for humidification, evaporative cooler is used and heating purpose heating coil is used. The above problem of human body due to exiting air conditioning system and psychometric process, 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. F.Moukalled et al [3] 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.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. 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 . Fatemeh et al[7]shown the modeling of a desiccant wheel used for dehumidifying the ventilation air of an air-conditioning system. 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. The electrical energy necessity can be very low comparing with conventional refrigeration system. 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. It also shows that, the moisture can be removed from the desiccant by heating it to temperature around 60°C-90°C and revealing it to a regenerative air stream. 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..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
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	di	desiccant wheel inlet
T_c	Cooling coil temperature	do	desiccant wheel outlet
T_{rh}	Temperature of re-heat coil	ei	evaporative cooler inlet
T_{ph}	Temperature of pre-heat coil	eo	evaporative cooler outlet
T_{rd}	Regeneration temperature of desiccant wheel	ci	cooling coil inlet
P	% mixing of re-circulated air to atmospheric air	co	cooling coil outlet
M_m	Mass flow rate of air in duct after mixing (kg/s) of ambient air and re-circulating air	rhi	re-heat coil inlet
		rho	re-heat coil outlet
RDW	Radius of desiccant wheel	a	ambient
dt	Thickness of desiccant coating(mm)	s	supply
Dh	Hydraulic diameter of a channel (mm)	r	room
N	Desiccant wheel speed	m	mixed air

Subscripts

Ph	pre-heating coil
rh	re-heating coil
c	cooling coil
phi	pre- heating coil inlet
pho	pre –heating coil outlet

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)Axial molecular diffusion within the desiccant is negligible.

(2)The channels are assumed adiabatic and impermeable.

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

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

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

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

2.2. Equations

2.2.1 Equation of DBT for mixed air [15].

$$T_m = \frac{(h_m - 2.5 \times W_m)}{(1.01 + 0.00189 \times W_m)} \qquad (1)$$

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

$$T_{ph} = \frac{(B_{ph} \times T_m - T_{pho})}{(B_{ph} - 1)} \qquad (2)$$

2.2.3 Equation of enthalpy for supply air [15].

(2.2.3.1) For cool and dry weather condition.

$$h_s = \frac{(M_m \times h_r + RTHO)}{M_m} \quad (3)$$

(2.2.3.2) For hot and dry as well as hot and wet weather condition.

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

2.2.4 Equation of DBT for supply air [15].

(2.2.4.1)For cool and dry weather condition.

$$T_s = T_r + \frac{(RSHO)}{(1.148314607 \times M_m)} \quad (5)$$

(2.2.4.2) For hot and dry as well as hot and wet weather condition.

$$T_s = T_r - \frac{RSH}{(1.148314607 \times M_m)} \quad (6)$$

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

$$\eta = \frac{(T_{ei} - T_{eo})}{(T_{ei} - T_{wei})} \quad (7)$$

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 $T_{wdo} = T_{wei} = T_{weo}$

2.2.6 Equation for outlet DBT of desiccant wheel [17].

$$T_{do} = p_1(N)\, p_2(T_m)\, p_3(dt) p_4(T_{rd}) p_5\, (W_m) p_6(D_h) p_7(U) \quad (8)$$

Where

$P_1(N) = -0.0002N^2 + 0.0112N + 0.420$

$p_2(T_m) = -0.0001\,(T_m)^2 + 0.0275\,T_m + 0.7993$

$p_3(dt) = -18.79(dt)^2 + 7.92dt + 1.75$

$p_4(T_{rd}) = -0.0004(T_{rd})^2 + 0.1255T_{rd} + 0.6757$

$p_5(W_m) = 594.48(W_m/1000)^2 + 26.76(W_m/1000) + 3.79$

$p_6(D_h) = -0.039(D_h)^3 + 0.026(D_h)^2 + 0.603D_h + 0.0912$

$p_7(U) = -0.060U + 0.7973$

3.2.15 Equation for efficiency of desiccant wheel [16].

$$\varepsilon = k_1(N) k_2(T_m)\, k_3(dt) k_4(T_{rd}) k_5\, (W_m) k_6(D_h) k_7(U) \quad (9)$$

Where

$k_1(N) = -0.0001N^2 + 0.0042N + 0.4474$

$k_2(T_m) = -0.0001(T_m)^2 - 0.0031T_m + 0.8353$

$k_3(dt) = -21.67(dt)^2 + 6.93dt + 1.34$

$k_4(T_{rd}) = -0.0001(T_{rd})^2 + 0.0355\,(T_{rd}) - 0.4924$

$k_5(W_m) = 592.77(W_m/1000)^2 - 41.23(W_m/1000) + 1.283$

$k_6(D_h) = -0.0572(D_h)^3 + 0.0933(D_h)^2 + 0.6139D_h - 0.0922$

$k_7(U) = -0.0611U + 0.8376$

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

$$W_{do} = (W_m - \varepsilon W_m) \tag{10}$$

3. System Description

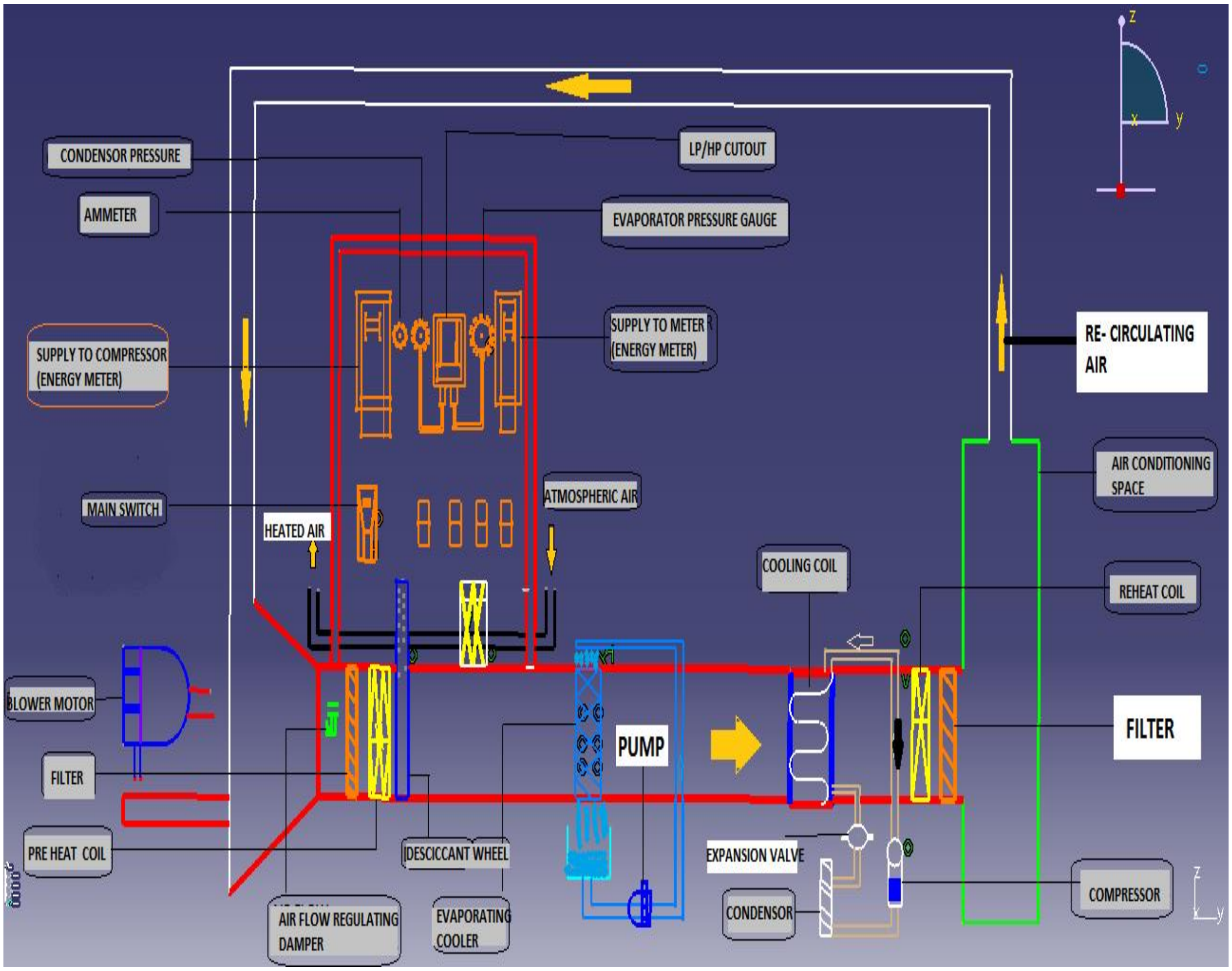


Fig.1. Diagram of year round air conditioning system [1]

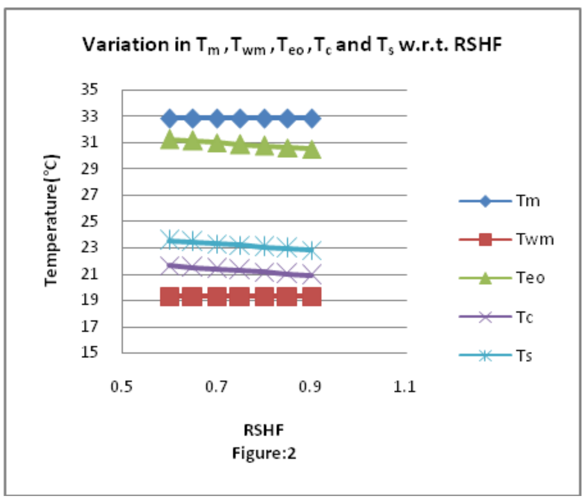
The positions of equipments used in year round air conditioning system are shown in the diagram of year round air conditioning system(Fig.1).The analysis is done on the concept that, supply air condition is depends on the room conditions, mass flow rate, CLC, and RSHF. We take the room conditions be constant. So the supply condition will be depends on mass flow rate, CLC and RSHF

4. Result and discussion

4.1: Hot and dry weather condition.

Table1: variation of output parameter w.r.t. room sensible heat factor (RSHF) at 44.8°C DBT of ambient air, altitude be 523m , 25°C DBT and 50% relative humidity of room,60 % mixing of re-circulated air ,0.2 BPF factor,1TR CLC and 1.25kg/s mass flow rate.

RSHF	0.60	0.65	0.70	0.75	0.80	0.85	0.90
T_m(°C)	32.81	32.81	32.81	32.81	32.81	32.81	32.81
T_{eo}(°C)	31.22	31.11	31.00	30.83	30.72	30.61	30.50
T_c(°C)	21.60	21.48	21.35	21.24	21.11	20.99	20.86
T_s(°C)	23.52	23.40	23.28	23.16	23.03	22.91	22.79
W_m(g/kg)	9.50	9.50	9.50	9.50	9.50	9.50	9.50
W_{eo}(g/kg)	10.04	10.09	10.14	10.19	10.24	10.29	10.33
W_s(g/kg)	10.04	10.09	10.14	10.19	10.24	10.29	10.33
η(%)	11.77	12.59	13.40	14.65	15.47	16.28	17.09
V(cm³)	250.54	269.05	287.72	316.94	336.07	355.39	374.90



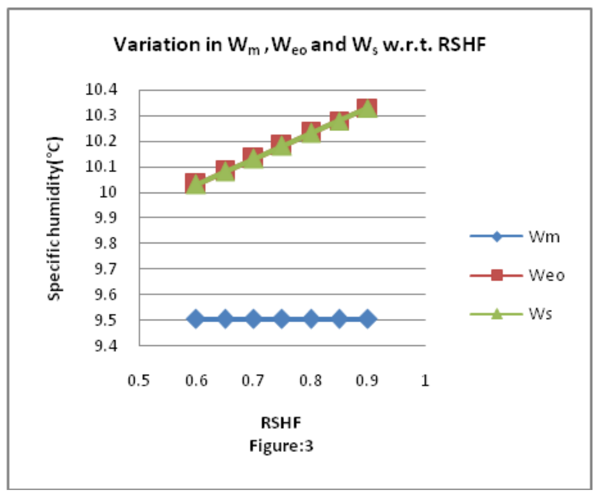


Figure (3) and Figure (4) indicates variation in temperature and specific humidity RSHF respectively. Findings from the results are being discussed as follows;

❖ The variation in RSHF has no effect on the inlet condition of evaporative cooler. Because inlet condition of evaporative cooler only depends on the room condition, ambient condition and mixing ratio of air.

- ❖ The DBT of supply air decreases with increase of RSHF value, since if RSHF value is more, then room sensible heat will be larger i.e. rise in temperature of air- conditioned room will be more, so larger amount of cooled air is required to maintained the room temperature but the mass flow rate of air is taken to be constant and which is only possible that DBT of supply air will be less.
- ❖ Temperature of cooling coil is decreases since supply air temperature decreases and BPF of cooling coil be constant.
- ❖ The DBT of evaporative cooler outlet air (T_{eo}) decreases with increases the value of RSHF.
- ❖ The value of W_s and W_{eo} is same, because only sensible cooling is done by cooling coil.

The table (1) shows that saturation efficiency increases with the increase in value of RSHF because inlet DBT of evaporative cooler(Tm) is constant and DBT of evaporative cooler outlet air (Teo) decreases. The volume of cooling pad increases with increase in RSHF value because increasing in value of saturation efficiency means the contact time of water with flowing air is more, and which is only possible when volume of cooling pad will be large.

4.2: Hot and wet weather condition.

Table2: variation of output parameter w.r.t. room sensible heat factor (RSHF) at 30°C DBT and 95 % of relative humidity of ambient air, 523m of altitude , 25°C DBT and 50% relative humidity of air conditioned room , 60% mixing of re-circulated air, 1TR CLC Value, 0.2 BPF, 1.25kg/s mass flow rate of air,15 rph and 90°C of regeneration temperature of desiccant wheel.

RSHF	0.60	0.65	0.70	0.75	0.80	0.85	0.90
T_m(°C)	27.57	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	59.91
T_{eo}(°C)	49.48	49.37	49.25	49.09	48.98	48.87	48.75
T_c(°C)	17.04	16.91	16.79	16.68	16.55	16.42	16.30
T_s(°C)	23.52	23.40	23.28	23.16	23.03	22.91	22.79
W_m(g/kg)	17.38	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	5.81
W_{eo}(g/kg)	10.04	10.09	10.14	10.19	10.24	10.29	10.33
W_s(g/kg)	10.04	10.09	10.14	10.19	10.24	10.29	10.33
ε	0.66	0.66	0.66	0.66	0.66	0.66	0.66
η(%)	29.29	29.60	29.94	30.39	30.70	31.01	31.35
V(cm^3)	438.15	443.69	449.75	457.89	463.51	469.16	475.35

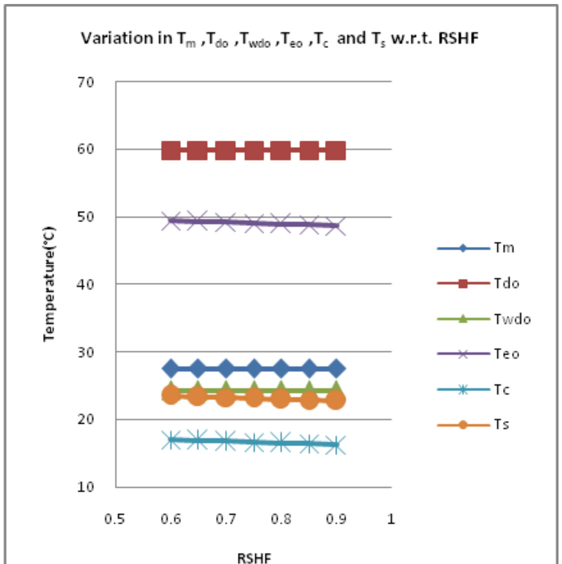


Figure:4

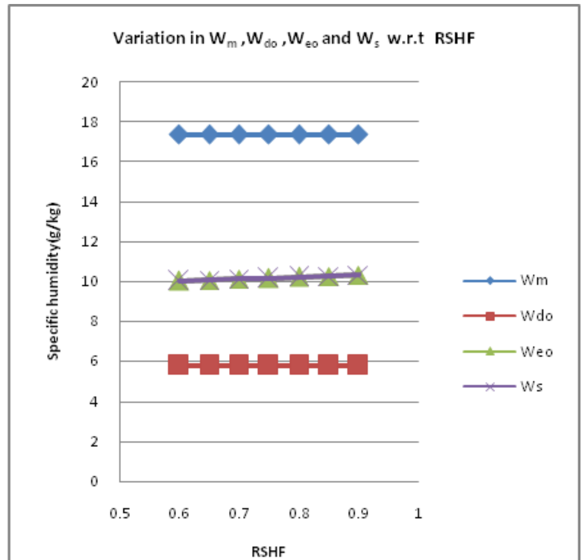


Figure:5

Figure (4) and Figure (5) indicates variation in temperature and specific humidity w.r.t RSHF 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 RSHF 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 DBT of supply air decreases with increase of RSHF value, since if RSHF value is more, then room sensible heat will be larger i.e. rise in temperature of air- conditioned room will be more, so larger amount of cooled air is required to maintained the room temperature but the mass flow rate of air is taken to be constant and which is only possible that DBT of supply air will be less.
- ❖ Temperature of cooling coil is decreases since supply air temperature decreases and BPF of cooling coil be constant.
- ❖ The DBT of evaporative cooler outlet air (T_{eo}) decreases with increases the value of RSHF.
- ❖ The value of W_s and W_{eo} is same, because only sensible cooling is done by cooling coil.

The table (2) shows that saturation efficiency increases with the increase in value of RSHF because inlet DBT of evaporative cooler(T_m) is constant and DBT of evaporative cooler outlet air (T_{eo}) decreases. The volume of cooling pad increases with increase in RSHF value because increasing in value of saturation efficiency means the contact time of water with flowing air is more, and which is only possible when volume of cooling pad will be large. The efficiency of desiccant wheel remains constant because RSHF have no effect on inlet as well as outlet condition of desiccant wheel.

4.3: Cold and dry weather condition.

Table 3: variation of output parameter w.r.t. room sensible heat factor (RSHF) at 7°C DBT and 60 % of relative humidity of ambient air, altitude be 523m , 25°C DBT and 50% relative humidity of room,60 % mixing of re-circulated air ,0.2 BPF of heating coil ,3.5KW RTHO value , 1.25kg/s mass flow rate and 16°C wet bulb temperature of outlet air for pre-heat coil.

RSHF	0.60	0.65	0.70	0.75	0.80	0.85	0.90
T_m(°C)	17.90	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	26.22
T_{ph}(°C)	28.29	28.29	28.29	28.29	28.29	28.29	28.29
T_{eo}(°C)	18.28	18.44	18.56	18.67	18.78	18.89	19
T_{rh}(°C)	28.50	28.62	28.74	28.86	28.99	29.11	29.24
T_s(°C)	26.46	26.58	26.70	26.82	26.95	27.07	27.19
W_m(g/kg)	7.92	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	7.92
W_{eo}(g/kg)	11.17	11.12	11.08	11.03	10.98	10.93	10.88
W_s(g/kg)	11.17	11.12	11.08	11.03	10.98	10.93	10.88
η(%)	77.69	76.12	74.95	73.87	72.79	71.72	70.64
V(cm^3)	2526.48	2412.26	2331.41	2260.55	2192.56	2127.21	2064.30

The table (3) shows that saturation efficiency decreases with the increase in value of RSHF because inlet DBT of evaporative cooler (T_{pho}) is constant and DBT of evaporative cooler outlet air (T_{eo}) increases. The volume of cooling pad decreases with increase in RSHF value because decreasing in value of saturation efficiency means the contact time of water with flowing air is less, and which is only possible when volume of cooling pad will be reduced.

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

- ❖ The variation in RSHF has no effect on the inlet condition of pre-heat coil. Because inlet condition of evaporative cooler only depends on the room condition, ambient condition and mixing ratio of air.
- ❖ The DBT of supply air increases with increase of RSHF value, since if RSHF value is more, then room sensible heat will be larger i.e. fall in temperature of air- conditioned room will be more, so larger amount of heated air is required to maintained the room temperature but the mass flow rate of air is taken to be constant and which is only possible that DBT of supply air will be more.
- ❖ Temperature of re-heat coil is increases since supply air temperature increases and BPF of re-heating coil be constant.

- The pre-heating coil temperature is constant, since the inlet air DBT of pre-heating coil is constant and the outlet of pre-heating coil that is inlet of evaporative cooler DBT of air is constant and the bypass factor be also constant.
- The DBT of evaporative cooler outlet air (T_{eo}) increases with increases the value of RSHF.
- The value of W_s and W_{eo} is same, because only sensible cooling is done by cooling coil.

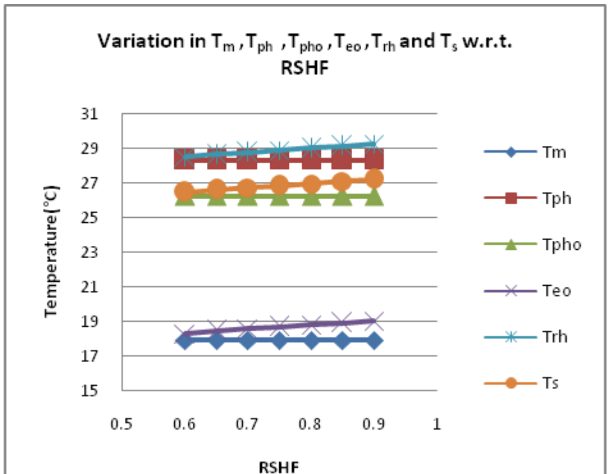


Figure:6

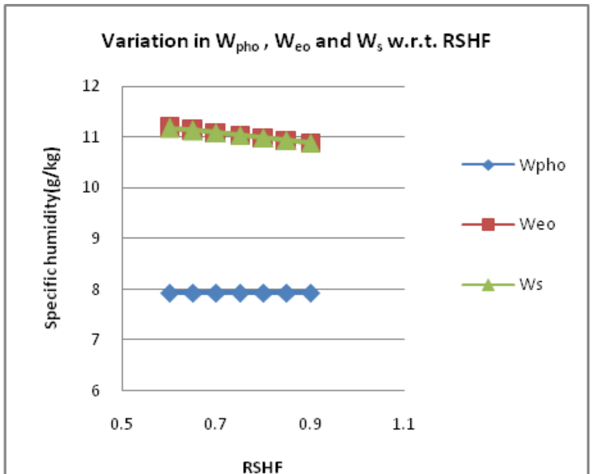


Figure:7

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 variation in DBT of supply air with RSHF is shown, for all weather condition that is hot and dry, hot and wet and dry and cool. The DBT of supply air decreases with the increase of RSHF for hot and dry along with hot and wet weather condition. The DBT of supply air vary from 23.53°C to22.79°C in both hot and dry as well as hot and wet weather condition, where as it vary from 26.46°C to27.19°C for RSHF value 0.6 to 0.9 . 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 vary from 10.04 to 10.33g/kg for hot and dry weather, as well as hot and wet weather condition and 11.67 to 10.88g/kg for cold and dry weather condition according to RSHF value 0.6 to 0.9. The volume of cellulose cooling pad increases from 250.5to 374.9cm^3 for hot and dry, and from 438.1 to 475.3cm^3 for hot and wet weather condition whereas decreases from 2526.4 to 2064.3cm^3 for cold and dry weather condition with increase in RSHF (0.6 to 0.9) . The saturation efficiency of evaporative cooler increases from 11.77to 17.09% for hot and dry, and from 29.29 to 31.35% for hot and wet weather condition whereas decreases from 77.69 to 70.64% for cold and dry weather condition with increase in RSHF (0.6 to 0.9). The efficiency of desiccant wheel remain constant (66.56%) with the increase in RSHF. The cooling coil temperature decreases with increase in RSHF for hot and dry as well as hot and wet weather condition.

The value of re-heating coil increases with the increase in RSHF. The value of pre-heating coil remains constant with increase in RSHF value.

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