American University of Beirut

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MECH 673

Project 2022

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I. Given Data:
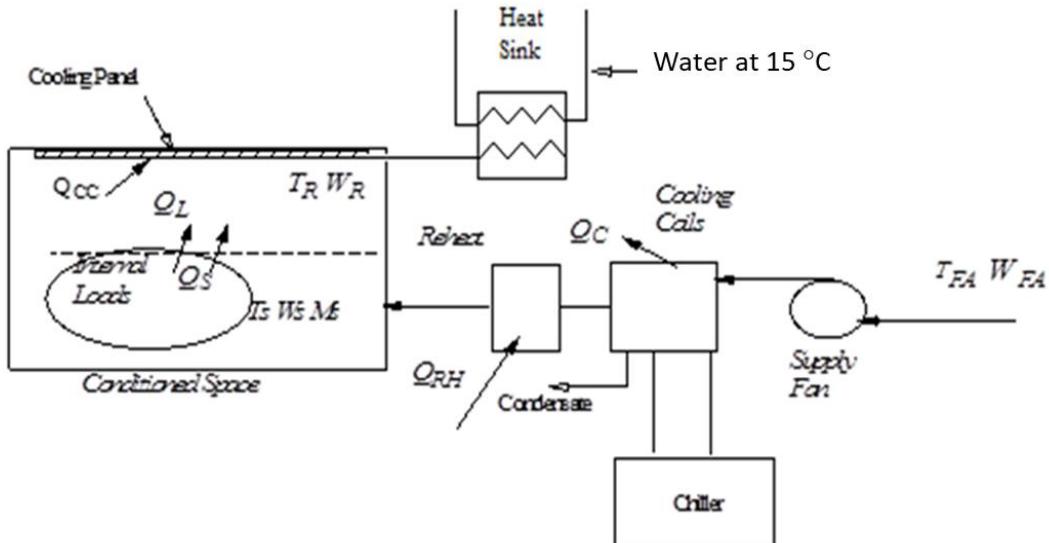
Figure 1: CCDV System Schematic

Hour	1	2	3	4	5	6	7	8	9	10	11	12
No. of people	0	0	0	0	0	0	1	4	5	10	10	12

Hour	13	14	15	16	17	18	19	20	21	22	23	24
No. of People	12	10	10	10	8	7	7	6	4	2	0	0

Table 1: Occupancy Schedule

Hour	1	2	3	4	5	6	7	8	9	10	11	12
Ambient Temperature (°C)	24.9	24.5	24.1	23.7	23.4	23.3	23.7	24.5	25.8	27.3	28.7	29.9
Ambient Humidity (kg H2O/kg air	0.01253	0.01229	0.01207	0.01184	0.01171	0.01162	0.01184	0.01229	0.01310	0.01388	0.01475	0.01546

Hour	13	14	15	16	17	18	19	20	21	22	23	24
Ambient Temperature (°C)	30.7	31.3	31.6	31.6	31.1	30.1	29.1	28	27.2	26.5	25.9	25.4
Ambient Humidity (kg H2O/kg air	0.01622	0.01661	0.01680	0.01664	0.01637	0.01569	0.01503	0.01431	0.01378	0.01350	0.01306	0.01286

Table 2: Weather conditions

- The office space dimensions are 5 m x 5 m x 3 m.
- The internal load due to equipment and lighting is fixed at 1000 W.
- The internal load per person is 100 W
- Maximum Occupancy in the office space is 12 persons.
- The seawater used to cool the chilled ceiling comes in at a temperature of $15°C$.
- The fan efficiency is 0.5 and the pressure difference is 1.5 kPa.
- The pump efficiency is 0.8 and the pump head is equivalent to 4 m.

II. Assumptions:

- Assume air leakages and thermal losses through the walls are negligible.
- Assume the latent load to be negligible compared to the sensible load.
- Quasi-steady operation of the CC/DV system in order to use the design charts for the hourly load state.

III. Analysis:

The peak load is found by using Tables 1 and 2:

Hour	1	2	3	4	5	6	7	8	9	10	11	12
Number of people	0	0	0	0	0	0	1	4	5	10	10	12
load due to people (w/m2)	0	0	0	0	0	0	4	16	20	40	40	48
load due to equipment and lighting (w/m2)	40	40	40	40	40	40	40	40	40	40	40	40
total load (w/m2)	40	40	40	40	40	40	44	56	60	80	80	88
Hour	13	14	15	16	17	18	19	20	21	22	23	24
Number of people	12	10	10	10	8	7	7	6	4	2	0	0
load due to people (w/m2)	48	40	40	40	32	28	28	24	16	8	0	0
load due to equipment and lighting (w/m2)	40	40	40	40	40	40	40	40	40	40	40	40
total load (w/m2)	88	80	80	80	72	68	68	64	56	48	40	40

Table 3: Load calculation for every hour.

The load due to equipment per unit area is calculated as follows:

$$q''_{equipment} = \frac{Q_{equipment}}{A_{room}} = \frac{1000}{5 * 5} = 40 \ W/m^2$$

The load due to people is calculated as follows:

$$q''_{persons} = N_p * \frac{Q_{person\,load}}{A_{room}} = N_p * \frac{100}{5*5} = N_p * 4 \; (W/m^2)$$

Thus, the total load is:

$$q''_{total} = q''_{equipment} + q''_{persons} = 40 + 4N_p \; (W/m^2)$$

As shown in Table 3, the maximum load is between 11 pm and 1 pm and is equal to **88 W/m^2**
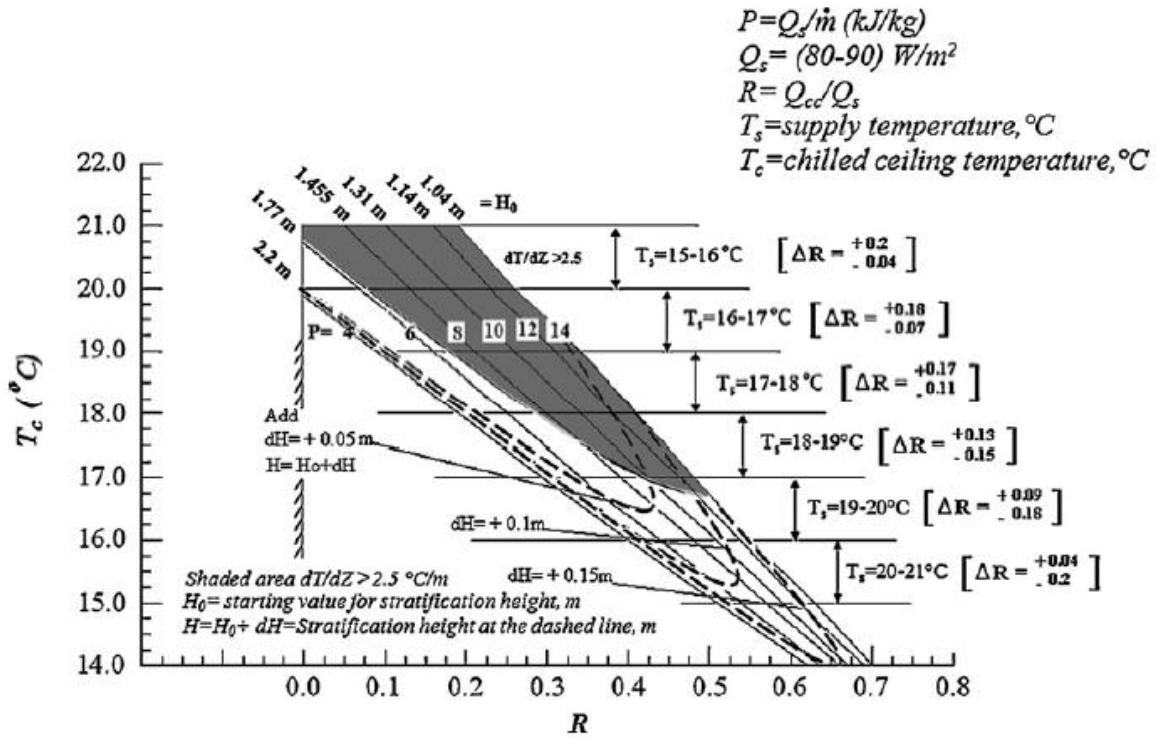
Fig. 5. Design chart of CC/DV system parameters for sensible load range of 80–90 W/m².

Figure 2: Design Chart of CC/DV system parameters for sensible load range of 80-90 W/m2

Now, the design load is known to be 88 W/m^2 allowing us to use the design chart of CC/DV systems for a sensible load range between 80 and 90 W/m^2 shown in Figure 2.

The requirements of our design are:

1. $\frac{dT}{dZ} < 2.5°C/m$, avoid discomfort due to difference in temperature between feet and head
2. $H > 1\,m$, Ensure good air quality for occupants sitting down.
3. $T_c > T_{dp}$, avoid condensate forming on the chilled ceiling.

The following equations will be used in the calculations to assess the systems that will be proposed in this project in order to satisfy the thermal comfort and indoor air quality requirements:

$$\dot{m}_s \left(\frac{kg}{s}\right) = \dot{m}_{FA} = \frac{Q_s \left(\frac{kW}{m^2}\right) * A_{room}(m^2)}{P\left(\frac{kJ}{Kg}\right)}$$

$$H = 0.017T_c + 0.034T_s - 0.036P + 0.007Q_s - 0.084R + 0.374$$

$$\frac{dT}{dz} = -0.136T_c - 0.008T_s + 0.052P + 0.006Q_s - 0.213R + 2.946$$

The temperature of the chilled ceiling cannot be selected to be under $15°C$, because the temperature of the seawater cooling it is at that temperature. For our proposed designs, we need to select complementary T_c and P values where the intersection is outside the shaded region in Figure 2 in order to satisfy Requirement 1. This limits us to choosing P values at $T_c > 15°C$ and a P value where the intersection is not in the shaded region.

Thus, the following selection shall be studied:

Design	P	mdot	Tc	Ts	R	H	dT/dZ
1	4	0.55	17	18	0.3	1.7218	1.1621
2	6	0.37	17	18	0.35	1.6456	1.25545
3	8	0.28	16	19	0.475	1.5801	1.460825
4	10	0.22	16	19	0.525	1.5039	1.554175
5	12	0.18	15.5	20.5	0.575	1.4702	1.703525
6	14	0.16	15.5	20.5	0.6	1.3961	1.8022

Table 4: Calculated values for multiple designs.

The values of Ts and R are taken from Figure 2.

As we can see in Table 4 the requirements of comfort (dT/dZ and H) are satisfied. The energy consumption of each design can now be calculated through the following steps:

A. Finding the exhaust air temperature $T_r(°C)$:
$$T_r = T_s + \left(\frac{dT}{dZ}\right)\left(\frac{3}{4H_r}\right)$$

$H_r = 3m$, which is the room height.

Design	P	mdot	Ts	dT/dZ	Tr

1	4	0.55	18	1.1621	20.61473
2	6	0.37	18	1.25545	20.82476
3	8	0.28	19	1.460825	22.28686
4	10	0.22	19	1.554175	22.49689
5	12	0.18	20.5	1.703525	24.33293
6	14	0.16	20.5	1.8022	24.55495

Table 5: Calculated return air temperature for multiple designs.

B. Finding the amount of energy removed by the chilled ceiling and that by displacement ventilation system using the following energy balance:

$$Q_S A_{room} - Q_{SDV} - Q_{cc} = 0$$

We note that according to the definition:

$$R = \frac{Qcc}{Qcc + Q_{SDV}} \rightarrow Q_{SDV} = Q_{cc}(\frac{1}{R} - 1)$$

$$Q_{SDV} + Q_{cc} = \frac{Q_{cc}}{R} = Q_S A_{room}$$

$$\rightarrow Q_{cc} = R * Q_s * A_{room}$$

It should be noted that the highest ambient humidity at peak load is 0.01622 $\frac{kg\ of\ H2O}{kg\ of\ air}$ and the ambient temperature is 30.7 °C.

C. Finding the amount of energy needed to reheat from $T_{dp,min}$ to T_s:

$$Q_{RH} = \dot{m}_s c_{pa}(T_s - T_{dp})$$

Note that T_{dp} is after the condensation is done. We find it based on the supply humidity ratio $W_{S,max}$ and relative humidity 100% through the psychometric chart.

We obtain the following values:

Design	Ws. Max	Tdp
1	0.006946	8.572
2	0.0067	8.001
3	0.005527	5.291
4	0.005296	4.682
5	0.004335	1.871
6	0.004113	1.142

Table 6: Temperature after condensation for multiple designs.

D. Finding the new humidity ratio due to the water vapor generated by the occupants.

$$W_v = \frac{N_p \dot{w}_p}{\dot{m}_s}$$

Note that $\dot{w}_p = 1.4 * 10^{-5} \frac{kg\ of\ H2O}{kg\ of\ air}$ per second

E. Finding the maximum allowable humidity ratio of air near the ceiling to avoid condensation at the dew point temperature.

$$HR\ (W_{a,max}\ ) = \%\ \frac{\exp\left(\frac{17.625 * T_{dp,min}}{243.04 + T_{dp,min}}\right)}{\exp\left(\frac{17.625 * T_r}{243.04 + T_r}\right)}$$

Where,

$$T_{dp,min} = T_c - 1.5°$$

F. Finding the maximum allowed supply air humidity ratio ($W_s\ \frac{kg\ of\ H2O}{kg\ of\ air}$) based on the new humidity ratio, maximum allowable humidity ratio of the air near the ceiling and the supply one.

$$W_{s,max} = W_{a,max} - W_v$$

G. We find the mass of water condensate by applying mass balance for water

$$\dot{m}_v - \dot{m}_{condensate} + \dot{m}_{FA}(W_{FA} - W_R)$$
$$\dot{m}_{condensate} = \dot{m}_{FA}(W_{FA} - W_R) + \dot{m}_v$$

H. Finding the cooling coil load using the following energy balance on the entire DV system:

$$Q_{RH} - Q_C + \dot{m}_{condensate}h_{fg} + \dot{m}_{FA}c_{pa}(T_{FA} - T_s) = 0$$
$$\rightarrow Q_c = Q_{RH} + \dot{m}_{condensate}h_{fg} + \dot{m}_{FA}c_{pa}(T_{FA} - T_s)$$

I. Finding the fan power using the following equation:

$$P_{fan}(kW) = \frac{\dot{m}_s(\frac{kg}{s})\Delta p(kPa)}{\eta_{fan}\rho_a(\frac{kg}{m^3})}$$

J. Finding the DV system's total energy as its electrical equivalent:

$$E_{DV}(W) = 0.5Q_c + 0.33Q_{RH} + P_{fan}(W)$$

K. Finding the temperature of the water exiting the chilled ceiling $T_{w,out}$ using the following equation:

$$T_c = \frac{T_{w,out} + T_{w,in}}{2} \rightarrow T_{w,out} = 2T_c - T_{w,im}$$

L. Finding the pump's mass flow rate(kg/s) using the equation of chilled ceiling heat removal:

$$Q_{cc} = \dot{m}_w c_{pw}(T_{w,out} - T_{w,in}) \rightarrow \dot{m}_w = \frac{Q_{cc}}{c_{pw}(T_{w,out} - T_{w,in})}$$

M. Finding the input power of the pump P_{pump}:

$$P_{pump} = \frac{\dot{m}_w g H_p}{\eta_p}$$

N. Finding the total energy used:

$$E_{total} = P_{pump} + E_{DV}$$

Below are the calculated values for each design:

Design	P	mdot	Tc	Ts	R	H	dT/dZ
1	4	0.55	17	18	0.3	1.7218	1.1621
2	6	0.37	17	18	0.35	1.6456	1.25545
3	8	0.28	16	19	0.475	1.5801	1.460825
4	10	0.22	16	19	0.525	1.5039	1.554175
5	12	0.18	15.5	20.5	0.575	1.4702	1.703525
6	14	0.16	15.5	20.5	0.6	1.3961	1.8022

Table 7: Design Data 1.

Design	Qcc	Qsdv	Qrh	Wv	Tdp,min	HR (wa, max)
1	660	1540	5185.4	0.000305	15.5	0.007251
2	770	1430	3666.3	0.000458	15.5	0.007158
3	1045	1155	3769.975	0.000611	14.5	0.006138
4	1155	1045	3149.96	0.000764	14.5	0.00606
5	1265	935	3415.317	0.000916	14	0.005252
6	1320	880	3041.971	0.001069	14	0.005182

Table 8: Design Data 2.

Design	Tdp	Ws. Max	m cond	Qc	fan power P (W)	E DV
1	8.572	0.006946	0.005101	13756.84	1,650.00	10,239.60
2	8.001	0.0067	0.003491	9784.129	1,100.00	7,201.94
3	5.291	0.005527	0.002941	8165.2	825.00	6,151.69
4	4.682	0.005296	0.002403	6787.254	660.00	5,093.11
5	1.871	0.004335	0.002179	6234.584	550.00	4,794.35
6	1.142	0.004113	0.001902	5537.137	471.43	4,243.85

Table 9: Design Data 3.

Design	mw	Tw_out	Ppump	E total
1	0.039286	19	1.926964	10,241.53
2	0.045833	19	2.248125	7,204.19
3	0.124405	17	6.102054	6,157.79
4	0.1375	17	6.744375	5,099.86
5	0.30119	16	14.77339	4,809.12
6	0.314286	16	15.41571	4,259.26

Table 10: Design Data 4.

CO2 Calculation:

The CO2 concentration in the room is calculated for a room volume of $75\ m^3$ during the peak where there are 12 people present in the room from 11:00 pm until 1:00 pm. Thus, the duration used for the calculation is 2 hours. The initial CO2 concentration in the room is assumed to be $700\ ppm$, since there are people present in the room before 11:00 pm as shown in Table 1. The outdoor concentration is assumed equal to 350 ppm. The CO2 product of each person is assumed to be $18 \times 10^{-3}\ m^3/h$. Thus, the CO2 concentration after 2 hours is given by the following equation:

$$C = C_o + \frac{S}{V \times N} - [C_o + \frac{S}{V \times N} - C_{io}]e^{-N \times t}$$

Where:

- C_o: outer concentration $= 350\ ppm$.
- S: total production of $CO2 = 12 \times 18 \times 10^{-3} = 0.216\ ^{m^3}/_h$
- t: is the time $= 2\ hours$.
- C_{io}: initial CO2 generation $= 700\ ppm$.
- V: is the volume of the room $= 75\ m^3$
- N: is the air change per hour and can be calculated using the below equation:

$$N = (\dot{m}_s/(\rho_{air} \times V_{room})) \times 3600$$

msdot	Density of air	Volume of room	Air change per hour	Concentration of CO2 (ppm)
0.55	1.204	75	21.9	481.5
0.37	1.204	75	14.75	545.25
0.28	1.204	75	11.162	608.0182
0.22	1.204	75	8.77	678.39
0.18	1.204	75	7.176	751.337
0.16	1.204	75	6.378	801.55

Table 11: Carbon Dioxide concentration calculation for peak load.

Sample calculation for the first design:

$$C = 350 + \frac{0.216 \times 10^6}{75 \times 21.9} - [350 + \frac{0.216 \times 10^6}{75 \times 21.9} - 700]e^{-21.9 \times 2}$$

⇨ C= 481.5ppm

Therefore, the CO2 concentrations of the 6 designs are below the recommended level, which is 1000 ppm. Hence, this room is following the CO2 maximum concentration standards.

Relative Humidity Calculation:

Designs	Tr (degrees Celsius)	Specific Humidity (HR) (Kg of H2O/Kg of air	Relative Humidity (%)
1	20.61473	0.007251224	47%
2	20.82476	0.007158051	45%
3	22.28686	0.006137678	40%
4	22.49689	0.006059797	38%
5	24.33293	0.005251669	35%
6	24.55495	0.005182324	31%

Table 12: Design Parameter Comparison.

These values are taken from the psychrometric chart shown in figure 3. The relative humidity values are between 30% and 70%, the comfort zone range.
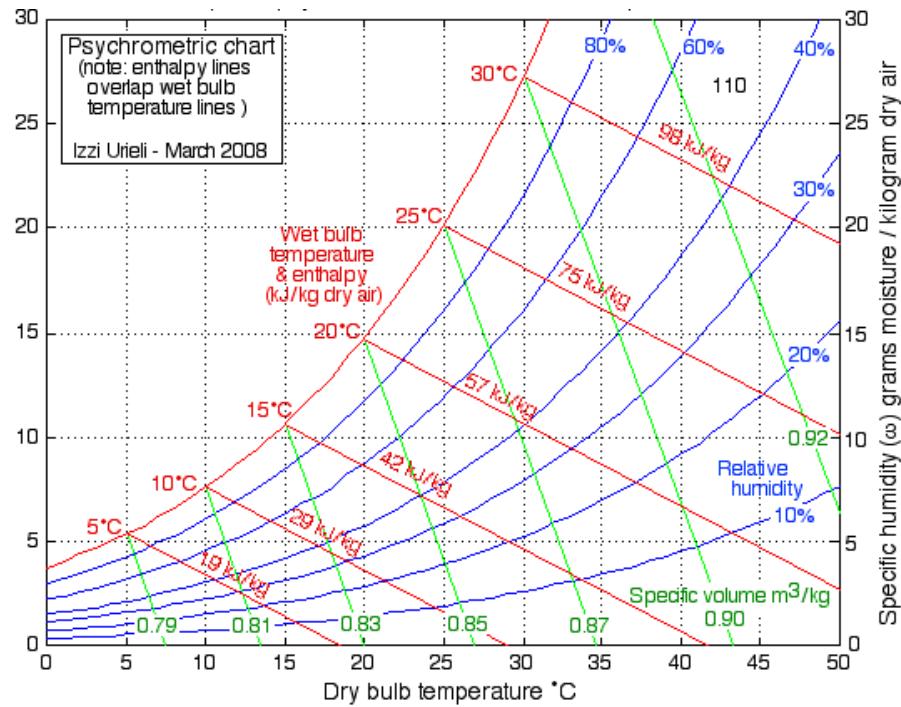

Figure 3: Psychometric Chart

Recommendations:

In Table 7, it is shown that the stratification heights for all the given designs are greater than 1, and the vertical temperature variation is less than $2.5°C$; thus, satisfying the requirements.

All six designs have also been adjusted in order to avoid condensation on the chilled ceiling with a factor of safety of $1.5°C$. Therefore, no condensation will occur for any of the designs above.

The carbon dioxide concentration during the peak CO2 production has also been calculated as shown in Table 11. The CO2 concentration does not go above 1000 ppm for an initial concentration of 700 ppm for any of the designs after the duration of the peak load (2 hours). This implies that the Displacement Ventilation system is providing enough air at the respective flow rates for each design.

The room temperatures for each design—shown in Table 5— vary significantly from $20.61473\ °C$ for Design 1 and $24.55495\ °C$ for Design 2. We consider the optimum range of thermal comfort to be at an optimum temperature of $24°C$ with an acceptable range of $22 - 26°C$. Thus, Design 1 and 2 are not acceptable. In addition, it should be noted that Designs 1 and 2 have a higher mass flow rate than the other designs, which might cause a draft and discomfort for the occupants of the office space.

Furthermore, the values for the relative humidity of all the designs are acceptable for a desired range of 30-70%. However, we note that designs 5 and 6 have very low relative humidity, which might lead to dryness depending on the physiology of some occupants.

Therefore, designs 3 through 6 have met the thermal comfort and air quality standards.

Finally, we would like to choose the most energy-efficient design. After modeling the system and calculating the needed energy costs for the condenser, reheater, air fan, and water pump. In Table 10, it is shown that Design 6 with a P value of 14 and R of 0.6 is the most energy-efficient of all the designs.

We compare the most important parameters for designs 2 through 6 in Table 12:

	Tr	RH	Etot
Design 3	22.28685625	40	6,157.79
Design 4	22.49689375	38	5,099.86
Design 5	24.33293125	35	4,809.12
Design 6	24.55495	31	4,259.26

Table 13: Design Parameter Comparison.

We observe that energy cost slightly differs between designs 4, 5, and 6; however, the temperature of design 4 might be too cool. The relative humidity is ideal for designs 3, 4, and 6, while for design 6 it is at the edge of the acceptable range.

The most energy-efficient design that satisfies all of the thermal and air quality requirements is Design 6; however, design 5 is a viable option for improved RH and room temperature but greater energy consumption. Designs 3 and 4 consume more energy and have a significantly lower room temperature very close to the acceptable lower bound. Considering the given weather condition in Table 2, the clothes the occupants would be wearing would be summer clothes and such lower temperatures might cause thermal discomfort. Hence, it is not advisable to implement designs 3 and 4.

Potential Improvements:

Potential improvements to the systems discussed above include the use of a desiccant wheel rather than a condenser and reheater for the dehumidification process, this would not only decrease the energy costs associated with the reheater and condenser but would also make sure of the exhaust room air if we use that air to regenerate the desiccant. However, it would be advisable to take into account the price needed to do such changes and the size that such a system might need.

Considering that we designed for peak load, the efficiency of the fan used might not as high during other times of the day. It is recommended to have smaller fans work in sequence, and turn off some of them when a lower load is needed in order to keep a high-energy efficiency for the fans.

Alternative Approaches:

A different recommended approach to selecting the optimum design is to set the needed thermal and air quality constraints in the model, write the objective function for energy, and run an optimization algorithm for the optimum R and P values. However, we lack the needed correlation to implement such a method currently. Best-fit curves might be capable of giving us such correlations to some degree of accuracy.