American University of Beirut

Maroun Semaan Faculty of Engineering and Architecture

MECH 674

CC-DV vs MIXED HVAC Systems

Spring 2023

Team Member	Name	ID
1	Jihad Jundi	202000646
2	Ahmad El Haj	202001178
3	Mohammad El Masri	202000044

I. Problem:

An office space has the following information:

- 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 office is located close to available seawater at a temperature of $15°C$.

The occupancy and weather conditions of the office are as follows:

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 owners of this office space are looking to provide air conditioning and ventilation in order to insure the productivity of the workers in this office.

II. Different Options:

One option is to use Mixed-Mode ventilation as discussed in **[1]**. This ventilation and cooling strategy involves manual mechanical systems like windows and refrigeration systems in order to provide adjustable ventilation. The size of the place discussed in the previous case study is 28 times our current space, but with similar occupancy; therefore, we will assume the consumption of this system to be that found in the study divided by 28 reaching an average of 125 kWh per month and consequently 1.5 MWh per year **[2]**.

Furthermore, Mixed-Mode ventilation provides enough air change to decrease CO2 levels and increase the productivity of the workers. The manual change optimizes energy consumption using the fact that the office is not always full.

It is shown that the usual mixed-mode system costs about 4$ per sq. ft equivalent to $43\$/m^2$. Therefore, the initial investment in order to install a mixed mode system in our office space is approximately 1080$ **[3]**.

The installation costs of the system are assumed 5% of the initial investment: 54$

We will assume the annual maintenance costs to be 1% of the initial investment; hence, 10.8$ per year.

The system is assumed to be inoperable at the end of its use and to be sold for 5% of its initial price: 54$ at the end of the lifecycle.

Another option is to use a chilled ceiling (CC) -displacement ventilation (DV) system.

- We will use seawater in order to cool our chilled ceiling.
- A fan of efficiency 0.5 and a pressure difference of 1.5 kPa will be used in order to supply air in the DV system.
- A pump of efficiency 0.8 and a pump head of 4 m will be used in order to circulate seawater to the chilled ceiling.

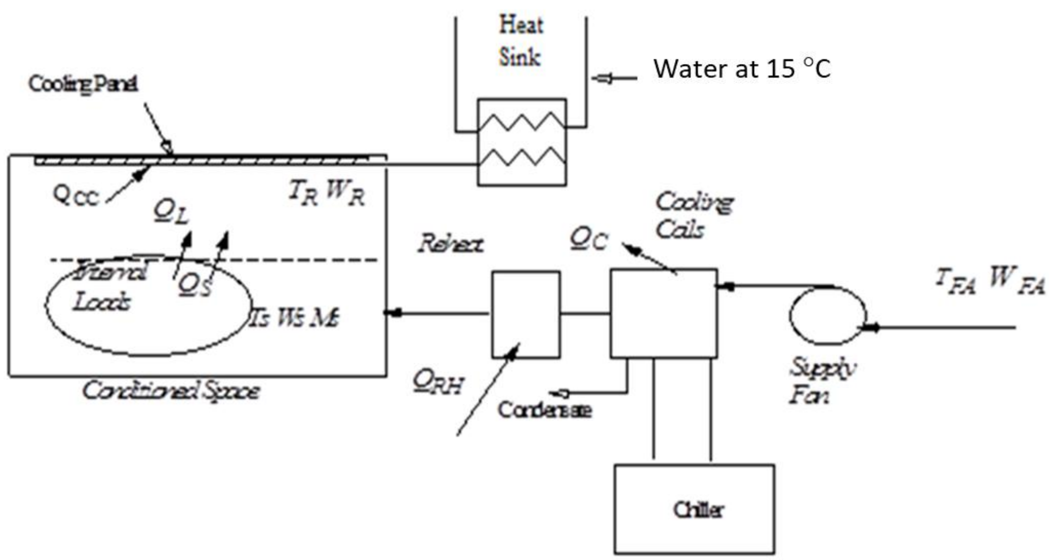

Figure 1: CCDV System Schematic

We will also take the following assumptions in the design of the system:

- 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. Design of CC-DV system:

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 $\mathbf{88\ W/m^2}$

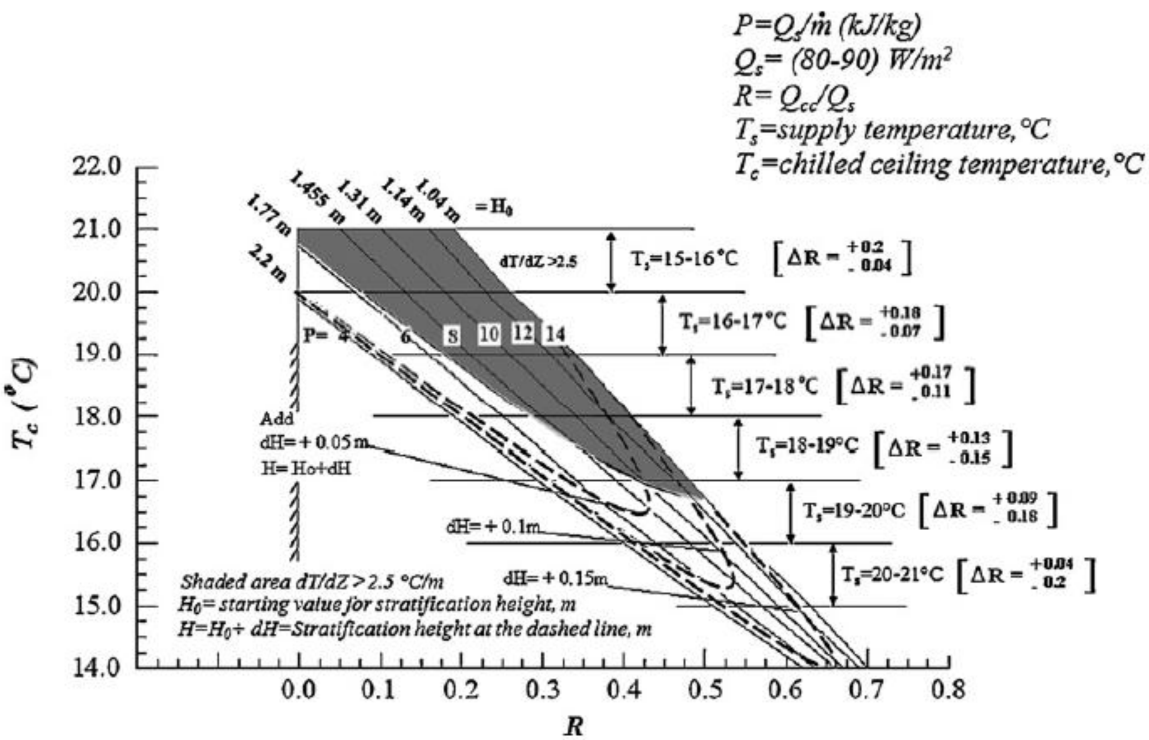

Fig. 5. Design chart of CC/DV system parameters for sensible load range of 80–90 W/m^2.

Figure 2: Design Chart of CC/DV system parameters for a sensible load range of 80-90 W/m2 **[4]**

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 the 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:

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

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

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

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(kg/s)	Tc($\circ C$)	Ts($\circ C$)	R	H(m)	dT/dZ ($\circ C/m$)
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 $T_r(°C)$:

$$T_r = T_s + (\frac{dT}{dZ})(\frac{3}{4H_r})$$

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

Design	P	Mdot(kg/s)	Ts($\circ C$)	dT/dZ ($\circ C/m$)	Tr($\circ C$)
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 (kg/kg)	Tdp ($\circ C$)
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.

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

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:

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

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:

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

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(kg/s)	Tc ($\circ C$)	Ts($\circ C$)	R	H(m)	dT/dZ ($\circ C/m$)
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(W)	Qsdv(W)	Qrh(W)	Wv(kg/kg)	Tdp,min ($\circ C$)	HR (wa, max) (kg/kg)
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 ($\circ C$)	Ws. Max (kg/kg)	m cond (kg/kg)	Qc (W)	fan power P (W)	E DV(W)
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(kg/s)	Tw_out ($\circ C$)	Ppump(W)	E total(W)
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.

Relative Humidity Calculation:

Designs	Tr ($\circ C$)	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 psychometric chart shown in Figure 3. The comfort zone range is between 30% and 70% relative humidity values.

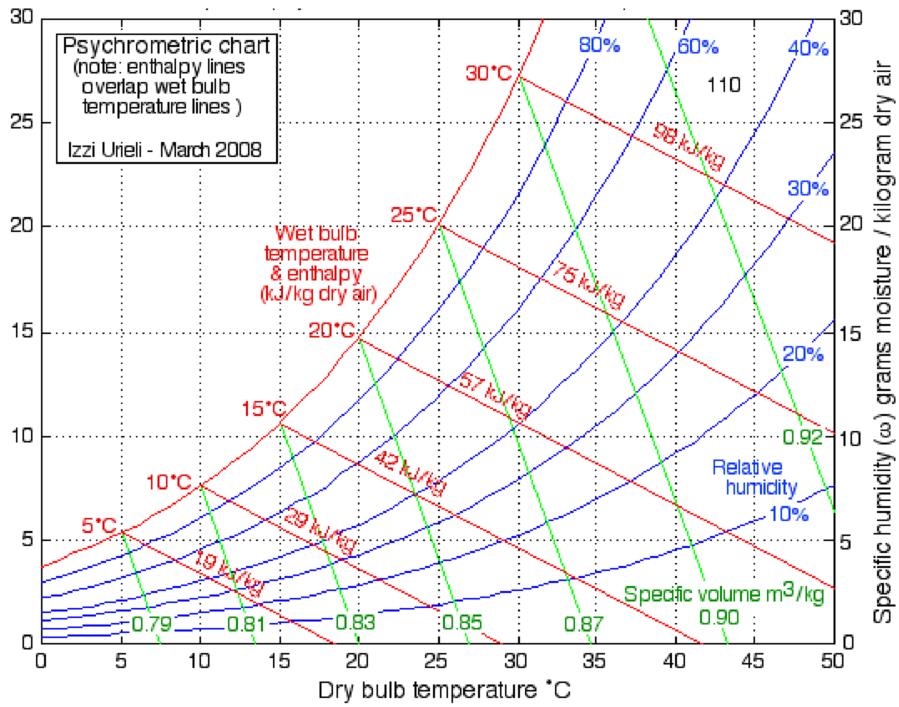

Figure 3: Psychometric Chart

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($\circ$ C)	RH	Etot(W)
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 the entire thermal and air quality requirements is Design 6.

Therefore, assuming this system is operated weekdays for 5 hours under these conditions throughout the hot months. We can assume the average energy consumption per year is 30% of this value.

$$Etot, annual = 4259.26 * (365 - 102) * 0.3 * 5 = 1.68\ MWh$$

Furthermore, we will install Photovoltaic solar panels in order to generate 50% of the energy needed to sustain the system. The solar panels cost 900$ and their installation 100$. This will cut annual energy cost by half to a value of 0.84 MWh.

Using the following study on pricing and sizing chilled ceiling and displacement ventilation systems **[5]** and taking into account the size of the space, we can take a realistic estimate of 1000$ to be the initial price for our system.

Installation price is assumed 10% of the initial investment: 100$

Maintenance costs are assumed 1% of the initial investment: 10$

The salvage value is assumed 5% of the initial investment: 50$

<u>IV- Simulation:</u>

First, we drew the geometries of the 2 systems. The occupants are represented by 12 cylinders with height 1.1m distributed evenly in the room. For the CCDV system, we created 2 inlets at the base of the wall and the outlet in the ceiling, while for mixed ventilation system, the 2 inlets and 1 outlet are in the ceiling. This can be illustrated by the figures below.

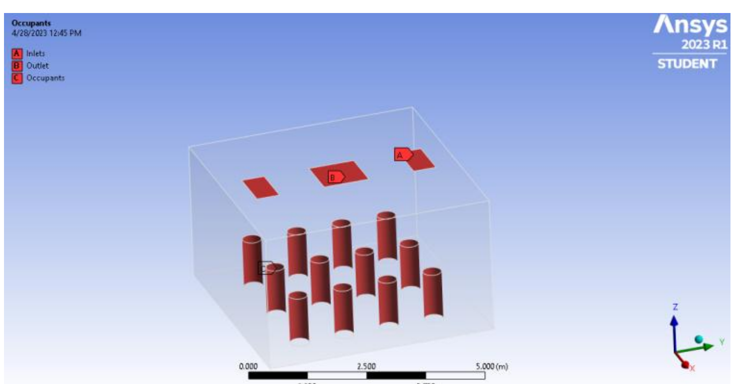

Figure 4: Mixed Mode Ventilation Geometry

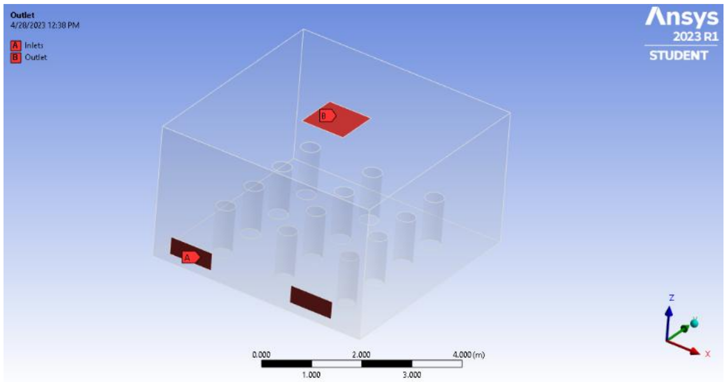

Figure 6: CCDV Geometry

The meshing of both systems is done in a manner to obtain around 150000 elements with an average orthogonality of 0.76 and a maximum skewness of 0.8. Inflation is applied in the space surrounding the occupants to allow for better resolution since this volume is the most important.

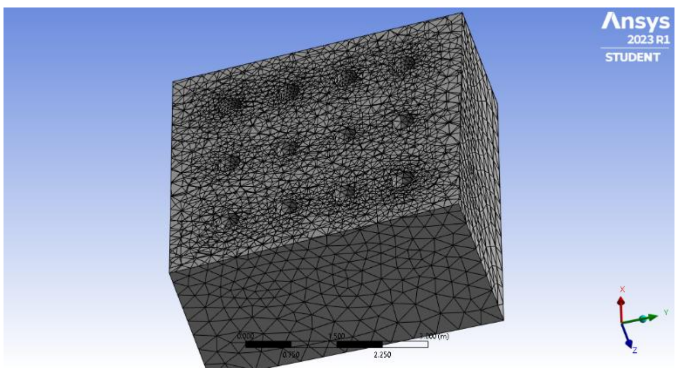

Figure 5: Mesh of the system

For thermal comfort, Figure 6 presents a plane cut of the mesh.

We imposed a 0.16kg/s flow rate of air at the inlets of both systems (Figures 6, 7, 8). The temperature of the inlet flow is taken as that of the ambient 20.5°C. These values were taken from design 6 above. For the CCDV system, we imposed a ceiling temperature 15.5 °C, the temperature of the chilled water going through the ceiling.

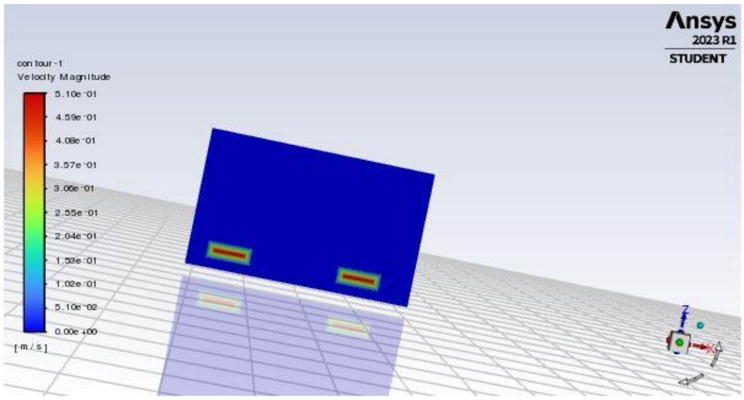

Figure 6: Flow rate simulation at the inlet of CCDV system

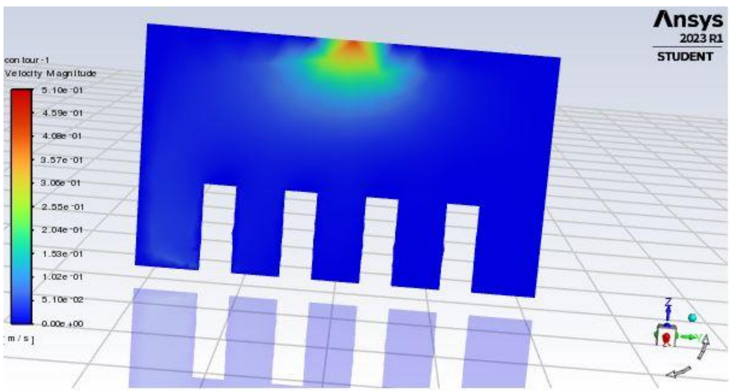

Figure 7: Flow rate simulation at the outlet of CCDV system

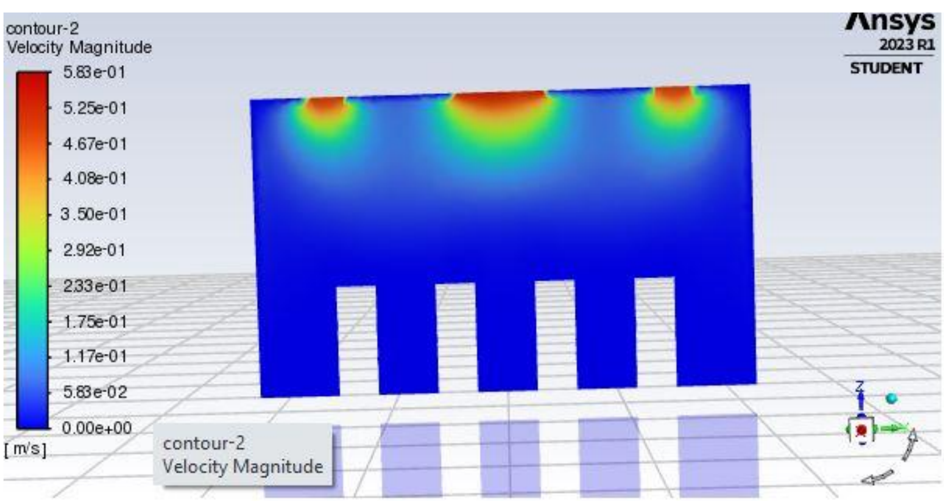

Figure 8: Flow rate simulation at the inlet and outlet of mixed ventilation system

From the simulation, the temperature head and chest zones of the occupants in the mixed ventilation is around 30°C while in the case of CCDV system, it is 25°C. The simulations are illustrated below in contour and vector forms in figures 9,10,11,12. In addition, according to ASHRAE, thermal comfort is attained mainly in the head and chest zones. That is why we are seeing the temperature in those zones are below the temperatures at the feet.

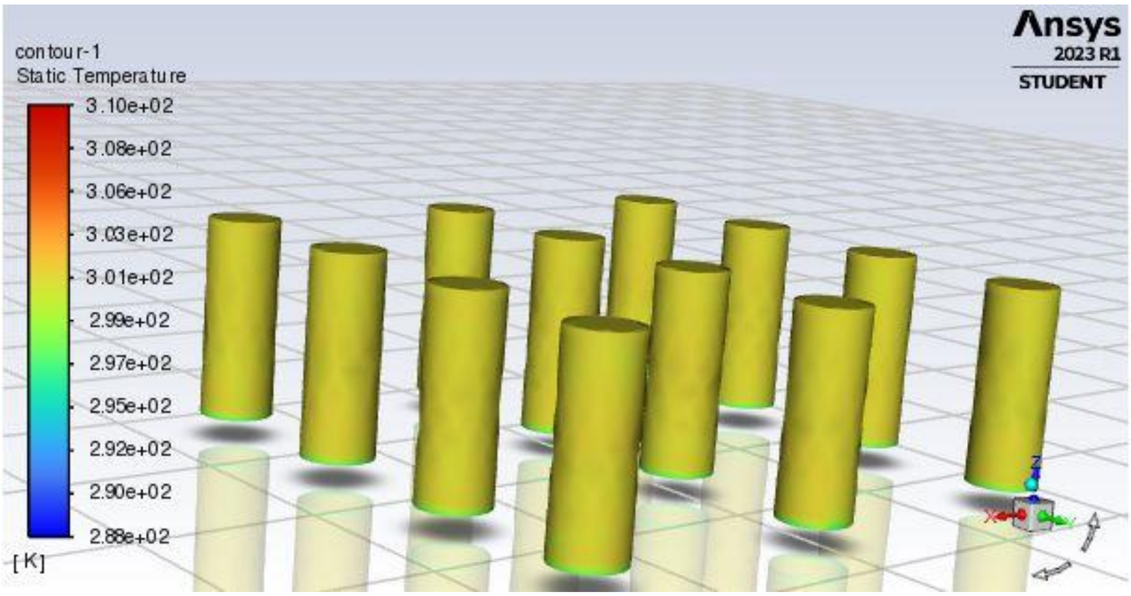

Figure 9: Contour temperature simulation of the occupant in CCDV system

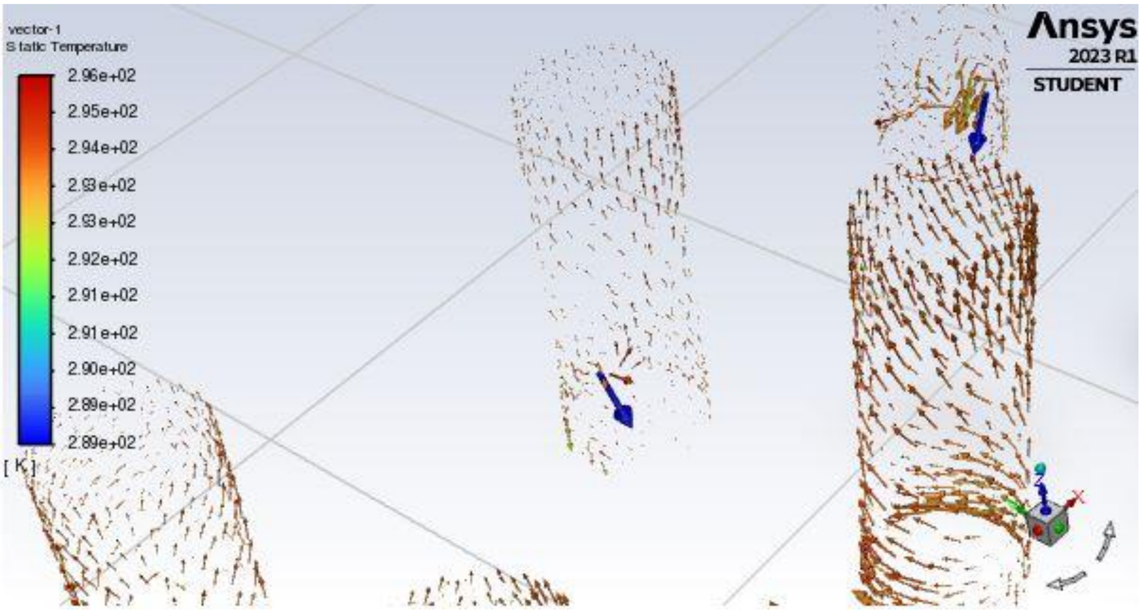

Figure 10: Vector temperature simulation of the occupant in CCDV system

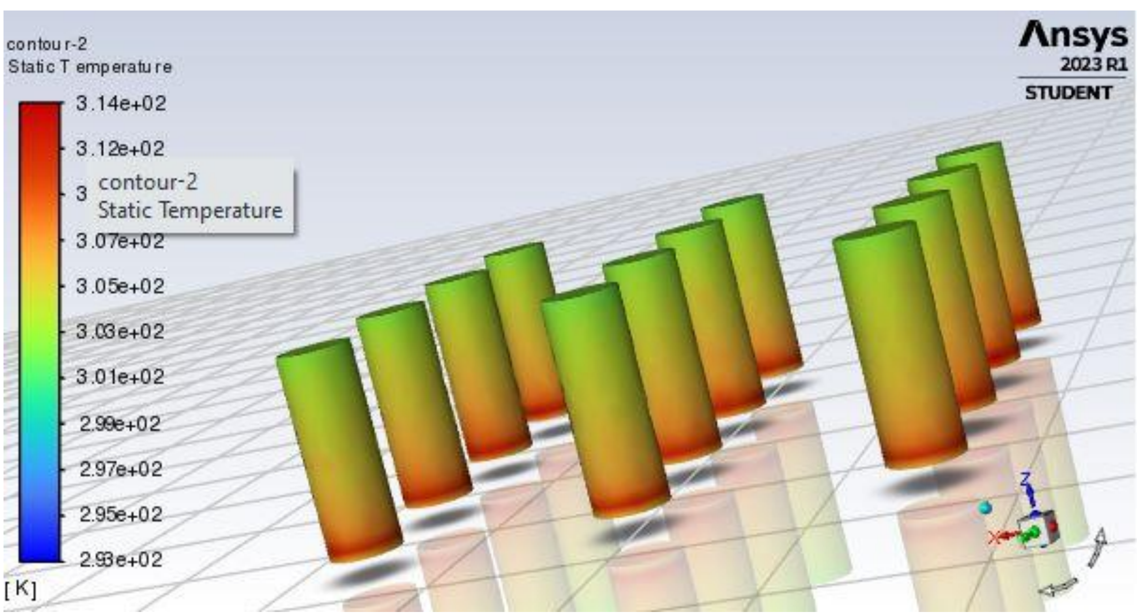

Figure 11: Contour temperature simulation of the occupant in mixed ventilation system

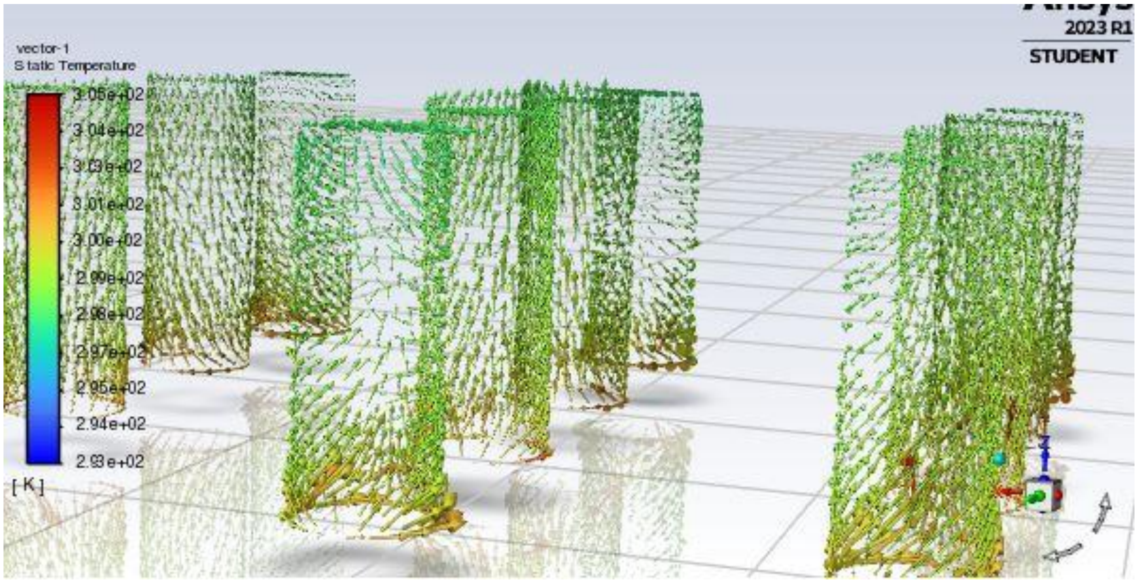

Figure 12: Vector temperature simulation of the occupant in mixed ventilation system

V. Economic Analysis:

A study period of 20 years will be taken.

System	Initial Investment	Installation Cost	Annual Energy Cost	Salvage Value	Annual Maintenance Costs
CC-DV	1000$+900$	100$+100$	0.84 MWh	54$	10$
Mixed Mod	1080$	54$	1.5 MWh	50$	10.8$

Table 14: System Economic Specifications

The electricity price is assumed to equal 0.168$/kWh with an annual increase of 2% **[6]**.

The electricity is assumed to be provided by a diesel power plant, which has a fuel-to-energy ratio of 0.4 L/KWh. So, 336 Liters of diesel are needed for the CC-DV system and 600 Liters for the Mixed Mode system **[7]**.

The annual nominal interest rate i is assumed to equal 4.75%. (https://countryeconomy.com/key-rates/usa)

The annual inflation rate f is assumed to equal 3.8% **[8]**.

Hence, the real interest rate r is equal to $r = \frac{1+i}{1+f} - 1 = 0.915\%$

We will assume that before this project, the owners were using 2 MWh per year for air conditioning and ventilation using split units.

Therefore, we will have the following:

Costs without interest rate:($)										
System	0	1	2	3	4	5	6	7	8	9
CCDV	-2100	184.88	188.7776	192.7532	196.8082	200.9444	205.1633	209.4665	213.8559	218.333
Mixed Mode	-1134	73.2	74.88	76.5936	78.34147	80.1243	81.94279	83.79764	85.6896	87.61939

10	11	12	13	14	15	16	17	18	19	20
222.8996	227.5576	232.3088	237.155	242.0981	247.14	252.2828	257.5285	262.879	268.3366	327.9034
89.58778	91.59553	93.64344	95.73231	97.86296	100.0362	102.2529	104.514	106.8203	109.1727	161.5721

Table 15: Cash Flows

Costs with intest rate:($)										
System	0	1	2	3	4	5	6	7	8	9
CCDV	-2100.00	183.20	185.37	187.56	189.77	192.00	194.25	196.53	198.82	201.14
Mixed Mode	-1134.00	72.54	73.53	74.53	75.54	76.56	77.58	78.62	79.67	80.72

10	11	12	13	14	15	16	17	18	19	20
203.49	205.86	208.25	210.67	213.11	215.57	218.06	220.58	223.12	225.69	273.28
81.79	82.86	83.95	85.04	86.14	87.26	88.38	89.52	90.66	91.82	134.66

Table 16: NPV Values for each cash flow

The NPV is calculated using the following equation:

$$NPV = -Initial\ Investment + \sum_{k=1}^{20} \frac{R_k - C_k}{(1+r)^k} + \frac{S}{(1+r)^{20}}$$

The CER is obtained as follows:

$$CER = \frac{NPV}{Initial\ Investment}$$

The internal rate of return is calculated using the following equation:

$$0 = -Initial\ Investment + \sum_{k=1}^{20} \frac{R_k - C_k}{(1+IRR)^k} + \frac{S}{(1+IRR)^{20}}$$

System	NPV($)	IRR	CER
CCDV	2046.30	8%	0.974430167
Mixed Mode	557.36	5%	0.491496583

Table 17: CER and IRR of each system.

If we compare the CER of each system:

$$CER_{ccdv} > CER_{mixedmode}$$

Therefore, using this criterion the CCDV system is recommended.

If we compare the IRR of each system:

$$IRR_{ccdv} > IRR_{mixedmode}$$

Therefore, using the IRR as the criterion the CCDV system is recommended.

Hence, we recommend the CCDV system for the air conditioning and ventilation of the office space.

VI. Benefit-Cost Analysis:

$$\boldsymbol{B/C} = \frac{\Sigma NPV_{benefit}}{\Sigma NPV_{cost}}$$

$$\Sigma NPV_{benefit} = \sum_{k=1}^{20} \frac{R_k}{(1+r)^k} + \frac{S}{(1+r)^{20}}$$

$$\Sigma NPV_{cost} = Initial\ Investment + \sum_{k=1}^{20} \frac{C_k}{(1+r)^k}$$

First, the following are the benefits of each year without interest:

Benefits without interest rate: ($)										
System	0	1	2	3	4	5	6	7	8	9
CCDV	0.00	336.00	342.72	349.57	356.57	363.70	370.97	378.39	385.96	393.68
Mixed Mode	0.00	336.00	342.72	349.57	356.57	363.70	370.97	378.39	385.96	393.68

10	11	12	13	14	15	16	17	18	19	20
401.55	409.58	417.77	426.13	434.65	443.34	452.21	461.26	470.48	479.89	543.49
401.55	409.58	417.77	426.13	434.65	443.34	452.21	461.26	470.48	479.89	539.49

Table 28: The Benefits Cash flows without interest taken into account.

In addition, the following are the costs at each year without interest:

Costs without interest rate: ($)										
System	0	1	2	3	4	5	6	7	8	9
CCDV	-2100.00	-151.12	-153.94	-156.82	-159.76	-162.75	-165.81	-168.92	-172.10	-175.34
Mixed Mode	-1134.00	-262.80	-267.84	-272.98	-278.22	-283.57	-289.03	-294.59	-300.27	-306.06

10	11	12	13	14	15	16	17	18	19	20
-178.65	-182.02	-185.46	-188.97	-192.55	-196.20	-199.93	-203.73	-207.60	-211.55	-215.59
-311.96	-317.99	-324.13	-330.40	-336.79	-343.31	-349.96	-356.74	-363.66	-370.72	-377.92

Table 19: The Costs Cash flows without interest taken into account.

Finding the NPV of each cash flow results in the following values:

Benefits with Interest rate: ($)										
System	0	1	2	3	4	5	6	7	8	9
CCDV	0.00	332.95	336.53	340.15	343.81	347.50	351.24	355.01	358.83	362.69
Mixed Mode	0.00	332.95	336.53	340.15	343.81	347.50	351.24	355.01	358.83	362.69

10	11	12	13	14	15	16	17	18	19	20
366.58	370.53	374.51	378.53	382.60	386.72	390.87	395.07	399.32	403.61	452.96
366.58	370.53	374.51	378.53	382.60	386.72	390.87	395.07	399.32	403.61	449.62

Table 20: The Benefits Cash flows NPV.

Costs with interest rate: ($)										
System	0	1	2	3	4	5	6	7	8	9
CCDV	-2100.0	-149.7	-151.2	-152.6	-154.0	-155.5	-157.0	-158.5	-160.0	-161.5
Mixed Mode	-1134.0	-260.4	-263.0	-265.6	-268.3	-270.9	-273.7	-276.4	-279.2	-282.0

10	11	12	13	14	15	16	17	18	19	20
-163.1	-164.7	-166.3	-167.9	-169.5	-171.1	-172.8	-174.5	-176.2	-177.9	-179.7
-284.8	-287.7	-290.6	-293.5	-296.5	-299.5	-302.5	-305.6	-308.7	-311.8	-315.0

Table 21: The Costs Cash flows NPV.

We calculate the Benefit-Cost (B/C) Ratio:

System	NPV Benefit	NPV Costs	B/C Ratio
CCDV	7430.01	-5383.71	1.380091819
Mixed Mode	7426.68	-6869.32	1.081137156

Table 22: Benefit-Cost Ratio of each system.

The benefit-cost ratio of the CCDV system is greater than that of the mixed-mode system.

Therefore, we are implored to use the CCDV system even more by this benefit-cost analysis.

VII. Electricity Generation Analysis:

We would like to study the efficiency of each system's power input. The CCDV system is powered 50% by PV cells and 50% by a diesel generator, while the mixed mode system is powered fully by a diesel generator.

We would like to compare them by using the levelised cost of electricity (LCOE), where:

$$LCOE = \frac{Initial\ Investment + \sum_{k=1}^{20} \frac{A_k}{(1+r)^k}}{\sum_{k=1}^{20} \frac{E_t}{(1+r)^k}}$$

System	Initial Invest +Installation	Annual Energy cost($)	Annual Energy Production(kWh)
Full Diesel	0	1500*electricity cost	1500
Half Diesel Half PV	1000	840*electricity cost	1680

Table 22: Energy Production of each system.

Cashflow costs for energy																					
System	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Full Diesel	0	252	252	252	252	252	252	252	252	252	252	252	252	252	252	252	252	252	252	252	252
Half Diesel Half PV	1000	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12	141.12

Table 23: Cash flow of the costs of each system.

NPV costs for energy																					
System	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Full Diesel	0	249.7146	252.3988	255.112	257.8543	260.6261	263.4276	266.2593	269.1215	272.0143	274.9383	277.8938	280.881	283.9003	286.952	290.0366	293.1543	296.3055	299.4906	302.71	305.9639
Half Diesel Half PV	1000	139.8402	141.3434	142.8627	144.3984	145.9506	147.5195	149.1052	150.708	152.328	153.9655	155.6205	157.2933	158.9841	160.6931	162.4205	164.1664	165.9311	167.7148	169.5176	171.3398

Table 24: NPVs of the costs of each system.

Energy with interest																					
System	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Full Diesel	0	1486.396	1472.916	1459.558	1446.321	1433.204	1420.206	1407.325	1394.562	1381.914	1369.382	1356.962	1344.656	1332.461	1320.376	1308.402	1296.535	1284.777	1273.125	1261.579	1250.137
Half Diesel Half PV	0	1664.764	1649.666	1634.704	1619.879	1605.188	1590.63	1576.204	1561.909	1547.744	1533.707	1519.798	1506.014	1492.356	1478.822	1465.41	1452.12	1438.95	1425.9	1412.968	1400.154

Table 25: The Energy produced by each system with interest included.

System	Sum Cost	Sum Energy	LCOE($/kWh)
Full Diesel	5538.754778	27300.79291	0.202878898
Half Diesel Half PV	4101.702676	30576.88806	0.134143889

Table 26: LCOE of each system.

Since, the 100% diesel energy production system for the mixed-mode system has a lower LCOE than that of the half diesel-half PV system used for the CCDV system. It is also recommended to use the CCDV system.

References:

1. Daaboul, J., Ghali, K., & Ghaddar, N. (2018). Mixed-mode ventilation and air conditioning as alternative for energy savings: a case study in Beirut current and future climate. Energy Efficiency, 11(1), 13-30.
2. Brager, G. (2011, August 5). Mixed-mode cooling. eScholarship, University of California. Retrieved April 24, 2023, from https://escholarship.org/uc/item/3bb8x7b8
3. 2020, 13 J. (n.d.). How your natural ventilation system can pay for itself. Indoor climate solutions. Retrieved April 24, 2023, from https://www.windowmaster.com/expertise/news-and-trends/how-your-natural-ventilation-system-can-pay-for-itself/#:~:text=Usually%2C%20a%20mixed%20mode%20ventilation,ft.
4. Ghaddar, N., & Ghali, K. (2009). Chilled-ceiling-displacement-ventilation-design-charts-correlations-to-employ-in-optimized-system-operation-for-feasible-load-ranges 2009 Energy-and-Buildings. Energy and Buildings, 41, 1155-1164.
5. Hu, S. (1999). Energy and first costs analysis of displacement and mixing ventilation systems for US buildings and climates (Doctoral dissertation, Massachusetts Institute of Technology).
6. Average price: Electricity per kilowatt-hour in U.S. city average. FRED. (2023, April 12). Retrieved April 24, 2023, from https://fred.stlouisfed.org/series/APU000072610
7. Diesel Generator. Diesel generator - Energy Education. (n.d.). Retrieved April 24, 2023, from https://energyeducation.ca/encyclopedia/Diesel_generator

8. Inflation rates in the United States of America. Worlddata.info. (n.d.). Retrieved April 24, 2023, from https://www.worlddata.info/america/usa/inflation-rates.php#:~:text=During%20the%20observation%20period%20from,year%20inflation%20rate%20was%206.0%25