Experimental Study for Assessment Air Age and Occupant Comfort by Adopting Combined Displacement Ventilation with Chilled Ceiling System in Iraq-Hilla City

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ABSTRACT :

This paper presents an experimental study of the displacement ventilation combined with chilled ceiling system in the office room and predicting air age, air temperature distribution and decay of carbon dioxide concentration with time. This study aims to present experimental results for air age and thermal comfort parameters to gain an understanding of the thermal performance of displacement ventilation with the chilled ceiling system under a range of operating conditions in a hot and dry climate (as Iraq-Hilla city climate). Two cases are studies depending on the shape of the supply diffuser. Case-I for the rectangular diffuser and Case-II for the semicircle diffuser. Four tests are performed for each case to observe the effects of parameters such as the shape of the diffuser and portion of the cooling load treated by the chilled ceiling. The four tests for each case based on the portion of cooling load treated by the chilled ceiling as (0%, 25%, 50% and 80%) respect to total cooling load in an office room and represented (0, 12.3, 24.6 and 39.5W/m2 per floor area). For each test, the air supply flow rate is fixed at (0.035m3/s). Air supply temperature varied depending on the cooling load treated by displacement ventilation as (49.3W/m^2, 37W/m2, 24.6 and 9.8W/m^2) for each test. Experimental work results show that the local air age increase with increase cooling load treat by the chilled ceiling. The temperature distribution effectiveness decreases by an average 0.05 with an increased portion of the cooling load treated by the chilled ceiling and used semicircular diffuser gives better effectiveness compared with the rectangular diffuser. The decay of CO_2 with time by used semicircle diffuser is faster compared with rectangle diffuser at any value of portion treated by the chilled ceiling.

KEYWORDS :

chilled ceiling; displacement ventilation; age of air; temperature effectiveness, thermal comfort.

1. INTRODUCTION :

One of the methods that used to provide improved indoor air quality and remove pollutants emitted by occupants is the displacement ventilation system (DV). The ventilation performance by used displacement ventilation was better compared to the poor ventilation provided by the mixing system Awbi (1998). The simple definition of a (DV) system, the air is supplied at a low speed and temperature through air supply devices located near floor level, and the exhaust air is located near ceiling level (Awbi, 1998, Schiavon et al., 2012). The methods developed by Chen and Glicksman (2012) and Skistad et al. (2002) were the most used reference for (DV) systems operation and design. One of the distinguishing characteristics of the displacement ventilation system is the formation of stratified layers due to supply cold air near the floor level, which may cause thermal inconvenience to the occupants due to excessive temperature layers according to ASHRAE standard (2010).

Chilled ceiling with displacement ventilation system used to reduce the different temperature between air layers in the occupied zone (Schiavon et al., 2012). The displacement ventilation system with chilled ceiling has been a system that has gained wide popularity in recent years, it combines thermal comfort and energy saving (Novoselac snd Srebric, 2002). The efficiency of the chilled ceiling combines with the energy efficiency of the displacement ventilation gave the best advantages compared with other ventilation systems (Xing et al., 2011). Air age in a closed room was studied at the beginning of 1980, a new procedure has been introduced as "The ability of the ventilation process to replace the old air in a room with fresh air" (Cao et al., 2014). The local mean air age was regarded as the average time for the air to move from a supply unit to any point in a ventilated space room. It could be used to evaluate the overall ventilation performance in a room (Sandberg, 1983). In most time the air being exhausted from the room before it's full of indoor contaminants because the supply air does not mix perfectly with the room air. As a result, there will be different concentration rates in the occupied zone and to achieve the threshold limit value a larger air supply rate will be required (Awbi, 2008). Knowing the age of air inside a closed room can provide us with many data, for example, evaluating air quality in an indoor environment, refers to the effectiveness of replacing the air in the room with fresh air from the ventilation system and assessing air pollution control (Awbi, 2017, Meiss et al., 2017). Ayoub et al. (2006) and Gheddar et al. (2008) developed general design charts for combined CC/DV system. Design diagrams have been developed for a 100% roof coverage factor. A sensitivity analysis has been performed for 80% ceiling coverage factor. Novoselac and Srebic (2002) undertook a comprehensive literature review about the performance and design of chilled ceiling with displacement ventilation system. They mentioned that designing of combined system was more difficult than designing a chilled ceiling and displacement ventilation system operating independently. Chicoteet et al. (2012) studied heat transfer coefficients and cooling capacity for the cooled radiant ceiling. Tests were made in a climate tested room. Walls are made from 30 mm thick sandwich type with polystyrene insulation puts between two layers of steel sheet. The main conclusions were, the operative temperature for cooled ceiling was not unique reference to predict the output cooling of cooled radiant ceiling, both convective and radiant phenomena separately was needed and provided approximate average values of (4.2W/m^{2}k) and (5.4W/m^{2}k) were found for convective and radiant heat transfer coefficients, respectively.

Hao. et. al. (2007), Kim and Leibundguta (2014), Muslmani. et. Al. (2016), Tian. et. al. (2019) and Jin. et. al. (2020) studied the condensation problem on the chilled ceiling and found many ways to solve this problem as used desiccant dehumidification succeeded, dehumidification in the Airbox and running the DV system for 300s before running chilled ceiling. Guo et. al. (2020) found that the condensation on chilled ceiling was prevented by operating chilled ceiling after air room temperature reached to 2°C less than dew point. Amini. et. al. (2020) studied the condensation problem also and found that the dehumidification coil up the chilled ceiling was succeeded to decrease the condensation on cooling panels. Saving energy by used the CC/DV combined system was studied by Bahman et. al. (2009), Chakrounet al. (2011), Itani et. al. (2015), and Seblany et. al. (2018) proved that the combined system was saving more energy compared with conventional system. Air temperature and velocity distribution for combined system were studied by Ayoub. et. al. (2006), Rees and Haves. (2013), Mateus and Grac. (2015), and Krajcik. et. al. (2016) and found that the chilled ceiling has influences the air distribution characteristics of displacement ventilation. Yang. et. al. (2017) studied the effect of chilled ceiling location on the thermal comfort in an office room and found that the acceptable location at ceil.

Air supply diffuser shape and age of air under(CC/DV) system had not been studied in depth in previous studies, especially in hot and dry climate, although the study of air age can give us many data about air pollution control (Awbi, 2017, Meiss, 2013) .There are not enough extensive studies on the performance of the combined CC/DV system under hot and dry climate, especially in the Middle East, Some of indoor experimental data related with DV/CC system were studied by some researchers in Arab countries as (Lebanon and Kuwait), but a very little of these studies focused directly on the effects of portion of cooling load treated by chilled ceiling and shape of air supply diffuser on age of air under CC/DV system, addition to that the combined ventilation system is not used in Iraqi buildings yet; then, it's a good starting point for study the thermal comfort parameters and indoor air quality by used this system in hot and dry climate (as Iraq-Hilla climate).

This work conducted a set of full-scale room experiments to study the age of air and thermal comfort parameters to gain understanding air quality and thermal performance of the displacement ventilation with chilled ceiling system in a range of operating conditions under hot and dry climate as Iraq-Hilla city climate in summer. The experimental procedures depend on calculating the total cooling load and estimating the actually magnitude of the inlet air flow rate and temperature needed for best ventilation at different ratio of cooling load treated by chilled ceiling, set up and instill chilled ceiling to notes the air age and temperature distribution behavior around occupants due to multiple heat sources and different air supply diffuser shape.

2. EXPERIMENTAL FACILITIES AND ROOM DESCRIPTION :

The experiments in this study consisted mainly of a test chamber with a full scale of displacement ventilation and chilled ceiling system to obtain data under steady state conditions over a range of operating conditions are varied. A set of full-scale office room experiments are conducted to study the local mean air age and thermal comfort with the DV/CC system. The facility, an isolated office room is located within a large test hall in the laboratory building for the Mechanical Engineering Department in Babylon University Fig.1. The tested room was delivered and setup with all the necessary equipment for DV/CC system.

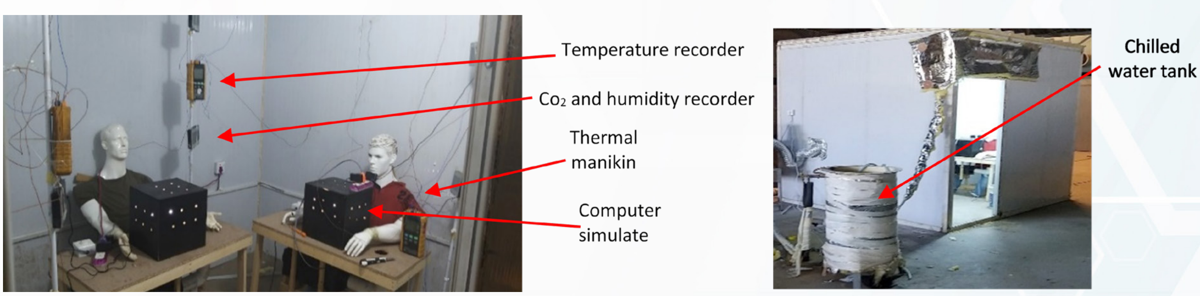


Fig.1photograph of the experimental tested room

The dimensions of the tested room for study cases is (3m x 2.5m and 2.5m height). Fig.2 shows the schematic of the model tested room. The tested office room has no windows and the walls and ceil made from isolated material (Sandwich banal, thermal conductivity is K=0.14W/m.K).The inlet air diffuser was located at the north wall. The outlet grill was placed at the center of ceil. Two occupants, two computers and one lump were placed as heat sources equal to 50W/m^2 of floor area. Chilled ceiling locate at height 2.5m from floor and represented 80% of floor area.

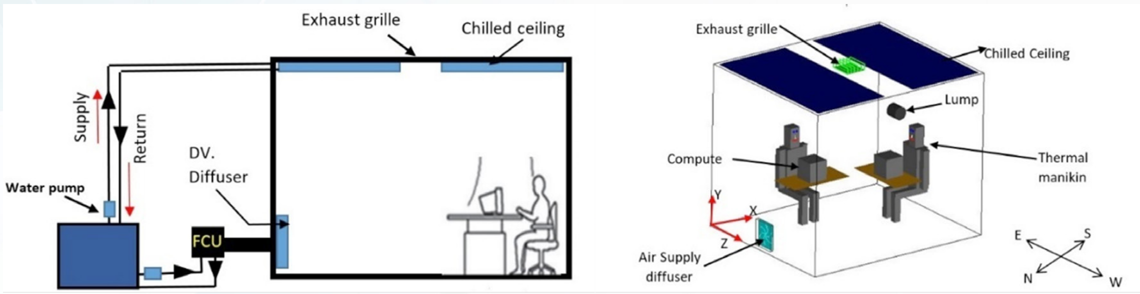


Fig.2 schematic of the model tested room.

The cooled fresh air delivered by air-fan coil that directly connected to the supply chilled water to treat the ventilation load. Chilled water delivered by chiller. Chilled water moved inside isolated plastic pipe to the chilled ceiling by used water pump.

Table (1) gives a description of the configuration and location of the different objects in tested office room. Human body properties represented by using a seated manikin at breathing height 1.1m (ASHRAE. Standard 55, 2010), Fig.3. The thermal manikinheat generate about 75W. (ASHRAE Fundamentals Handbook, 2001, Skistad et al., 2002).

To design and operation of DV systems, the method developed by (Chen and Glicksman, 2003) and (Skistad et. al., 2002) were the most commonly references used. ASHRAE standard 55 (2010) showed that the high different temperature between indoor air temperature and air supply temperature caused local thermal discomfort due to draft, then the different temperature between design indoor air temperature and air supply temperature should not exceed 5.5°C (ASHRAE handbook, 2012).

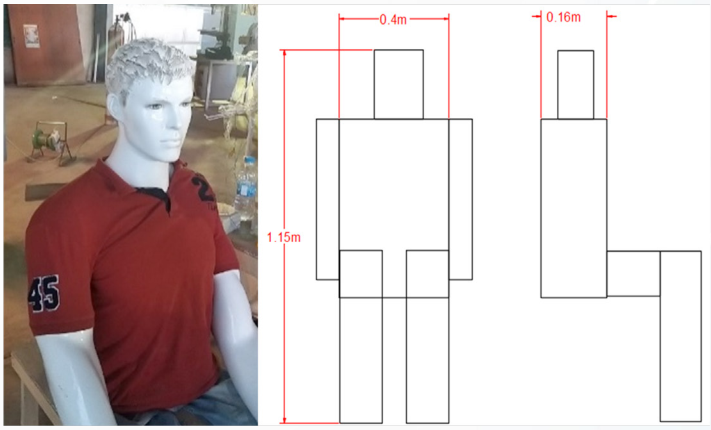
For displacement ventilation in this study and because there is no return air, the supply air flow rate equals 100% fresh air. The supply flow rate (Q_{DV}) is calculated by used Eqn. (1) (Skistad et. al., 2002).

$$Q_{DV}=\frac{0.295q_{oe}+0.132q_{l}+0.185q_{ex}}{ñC_{p}ÄT_{hf}} \quad \text{......................... } 1$$

Item	Location, m			Size, m			Heat, W
	X	Y	Z	Δx	Δy	Δz	
Room	0	0	0	3	2.5	2.5	-
Rectangular Diffuser	0.05	0.05	1.1	0.1	0.42	0.325	-

Item	Location, m			Size, m			Heat, W
	X	Y	Z	Δx	Δy	Δz	
Sime Circular Diffuser	0.05	0.05	1.1	Diameter 20 cm at height 42cm			-
Exhaust grille	1.35	2.5	1.1	0.3	0	0.35	-
Person1	1.25	0	0.25	0.4	1.15	0.125	75
Person2	2.75	0	1.25	0.125	1.15	0.4	75
Compute 1	1.1	0.7	0.6	0.3	0.3	0.3	60
Compute 2	2.1	0.7	1.1	0.3	0.3	0.3	60
Lump	1.5	2.2	2.3	0.2	0.2	0.2	100
Table1	0.75	0.7	0.5	1	0.05	0.5	0
Table2	2	0.7	0.75	0.5	0.05	1	0
					Total load		370W

Table.1 Test room configuration



a- Photograph b- Front view c- Side view

Fig.3 thermal manikin in the experimental work

The supply air temperature for displacement ventilation diffuser can be calculated by using Eqn. (2) (Skistad et. al., 2002).

$$T_s = T_{dr} - \Delta T_{hf} - \frac{A_f C L_{DV}}{0.584 Q_{DV}^{2} + 1.2 A_f Q_{DV}} \qquad \text{......................... } 2$$

$$CL_{DV} = q_{oe} + q_l + q_{ex} \quad \text{......................... } 3$$

$$CL_{DV} = \tilde{n} Q_{DV} C_p (T_e - T_s) \quad \text{......................... } 4$$

The indoor air conditions for office room design as dry bulb temperature (T_{dr}) of 25°C (ASHRAE handbook., 2012). The supply air temperature varied depend on the amount of cooling load cover by displacement ventilation (CL_{DV}) at constant flowrate. For displacement ventilation the different temperature between head and foot for occupant (ΔT_{hf}) is 2°C for seated person and 3°C for standing person as in ASHRAE Standard 55 (2010).

2.1 AIR SUPPLY DIFFUSER :

In this study for experimental tests, two types of supply diffusers are used, Fig. 4

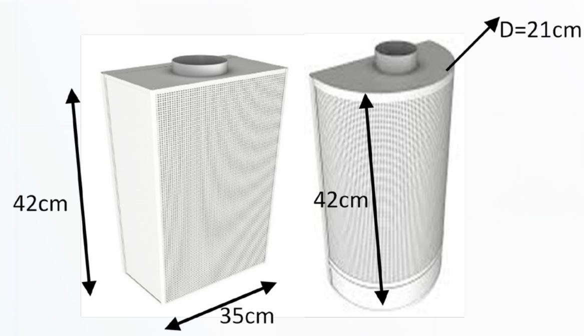
1- One-way rectangular diffuser.

2- Sime circular diffuser.

In displacement ventilation there are some goals must be satisfied such as thermal comfort, clam operation, low velocity.

Area of diffuser can be calculated by used Eqn. (5), assuming the supply air velocity is 0.25 m/s (Chen, 2003, Awbi, 2008), who specified the supply air velocity between 0.2-0.35m/s), at constant air flowrate (0.035 m^3/s), then the effective area of supply air diffuser (A_s) is (0.14m^2).

$$A_s = Q_{DV} * u_s \quad \text{......................... } 5$$



a-one way rectangle diffuser b- semicircle diffuser

Fig.4 supply air diffuser

2.2 CHILLED CEILING DESIGN :

In this study, the type of chilled ceiling used is (Radiant chilled ceilings). A chilled ceiling surface that provides space cooling by both radiation and convection. The cooling load removed by chilled ceiling (CLCC) (convection and radiation heat transfer) can be calculated by measuring the water mass flowrate, supply and return water temperature as show in Eqn. (8). (Schiavon et al., 2012).

$$CL_{cc}=\dot{m}_{ws}C_{P,w}(T_{wr}-T_{ws}) \quad \text{......................... 6}$$

temperature of chilled ceiling surface is important factor affecting on condensation problems, then chilled ceiling surface temperature (T_{CC}) should be at least (1°C) higher than the dew-point temperature of indoor air to avoid condensation on chilled ceiling surface.

From psychrometric chart, the dew-point temperature of indoor air for office room at (25°C) dry pulp temperature and 50% relative humidity is (14°C), then the supply chilled water must be large than 15°C.

$$\eta=\frac{CL_{cc}}{CL_{DV}+CL_{cc}} \quad \text{......................... 7}$$

η (eta) represented the ratio of cooling load removed by chilled ceiling (CL_{CC}) to the total cooling load as show in Eqn. (9) (Schiavon et al., 2012).

The (η) value in experimental tests change by 0-80% as (0, 25, 50 and 80%), η =0 means that a full displacement ventilation system is used.

2.2.1 CHILLED CEILING PLATES :

The chilled ceiling consists of copper coil secured to the aluminum metal plate (Chen and Glicksman., 2003). The diameter of copper pipe is (1/2 in). The chilled ceiling plates cover about 80% of the ceiling area (Ayoub et al., 2006). Aluminum Plates and copper pipe have high thermal conductivity are (237W/m.°C for aluminum and 390W/m.°C for copper (ASHRAE Handbook, 2012)). There are four plates fixed at ceiling, the dimensions of each plate are 1.25m(length) × 1.2m(width). The back of each plate is covered with aluminum sheet and insulation made of glass wool (thermal conductivity is 0.04 W/m.°C) as shown in Fig.5, to prevent the heat loss and reduce the noise produces during the water passing through pipes according to ASHRAE standards (2001).

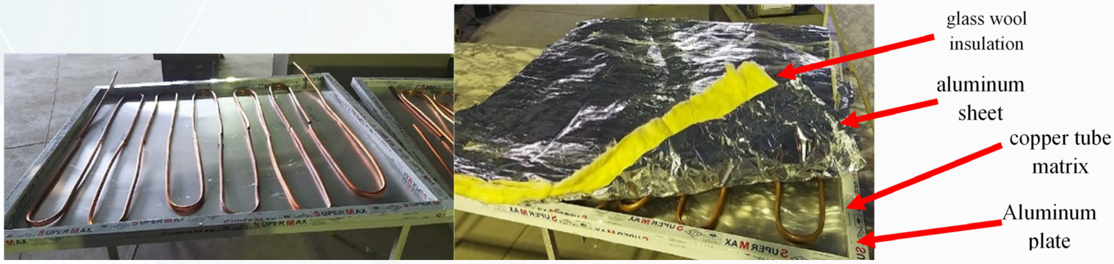


Fig.5 chilled ceiling panel construction showing the chilled ceiling materials.

2.2 CHILLED CEILING DESIGN :

The main header pipe that supplies child water for each chilled ceiling plate as alone to avoid irregular in plats temperature, and as the same method for collect and return main pipe. The copper pipe at diameter (½ in) and (15m) long for each plate used as thermal pipe in chilled ceiling plate. Fig.6 show the dimensions and specification of chilled ceiling plate with copper pipe.

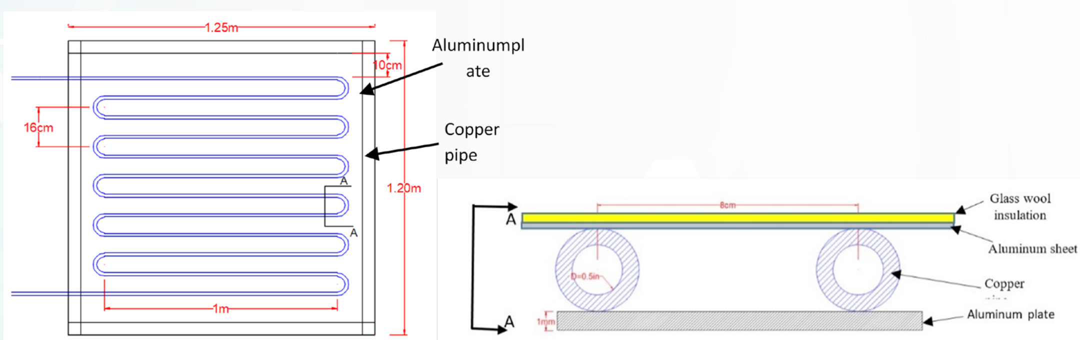


Fig.5 chilled ceiling panel construction showing the chilled ceiling materials.

3. DESCRIBE OF CASES STUDY :

The steady state experiments aim to investigate the displacement ventilation combine with chilled ceiling system at difference of operating conditions in a steady state and reported sets of air age, carbon dioxide concentration and thermal comfort parameters. The operating conditions of the DV/CC system in the test office room are:

1- The air flow rate is fixed at 35 l/s.

2- The flow rate of chilled water supply is (0.144 to 0.5M^3/h) depending on CL_{CC} value at constant water temperature different.

3- The air velocity of displacement diffuser is 0.25m/s (constant for all the tests).

4- The supply temperature of the chilled water and air are varied depend on the ratio of cooling load removed by chilled ceiling (η).

2.2 CHILLED CEILING DESIGN :

A series of eight tests were completed with the DV/CC system operating divided in to two cases. The firs case (Case-I) with one way rectangular diffuse. It's has four tests (from Test-1 to Test-4), one of these tests with the displacement ventilation system operating alone and other three tests with DV/CC system at different ratio of the cooling load removed by chilled ceiling (η) (as 25%, 50% and 80%). The second case (Case-II) have same procedure for (case-I) but with semicircle air supply diffuser (from Test-5 to Test-8). Table (2) shows the operating conditions for each test in case-I and case-II, where (RH_s) are measured at experimental day for each test.

	Test	η %	CL_{DV} W	CL_{CC} W	ACH 1/h	Q_{DV} 1/s	T_s°C W	RH_s %
Case-I	Test - 1	0	370	0	6.7	35	20	72
	Test - 2	25	277.5	92.5	6.7	35	21.5	64
	Test - 3	50	185	185	6.7	35	23	57
	Test - 4	80	74	296	6.7	35	24	57
Case-II	Test - 5	0	370	0	6.7	35	20	70
	Test - 6	25	277.5	92.5	6.7	35	21.5	66
	Test - 7	50	185	185	6.7	35	23	57
	Test - 8	80	74	296	6.7	35	24	54

Table (2) operating conditions for two cases

4. INITIAL CONDITIONS :

Initial value of Co2 concentration inside test room is (1000 ppm) to estimate the time required to reach to the normal concentration which recorded outside the room at tested day. Ambient temperature and relative humidity (RH) depend on the experimental test day under Iraq-Hilla climate as shown in Table (3).

Case	Test	Experimental day	T_{amb}°C	RH%	CO_2 (ppm) concentration
Case-I	Test - 1	2020/8/10	41.5	32	488
	Test - 2	2020/8/12	41	31	486
	Test - 3	2020/8/17	42.5	29	484
	Test - 4	2020/8/19	42	31	486

Case	Test	Experimental day	T_{amb}°C	RH%	CO_2 (ppm) concentration
Case-II	Test - 5	2020/8/24	39	34	484
	Test - 6	2020/8/26	40	34	488
	Test - 7	2020/9/2	38	33	486
	Test - 8	2020/9/7	38	34	487

Table. 3 ambient temperature, Co2 concentration and relative humidity at test days

5. MEASUREMENT DEVICES :

The Air, walls, and chilled ceiling surface temperature measurement system is based on a set of thermocouples are fixed in test room. There are 34 thermocouples (K-Type) which connected to three data temperature recorder (BTM4208 SD) systems Fig.8a, each supporting 12 channels.

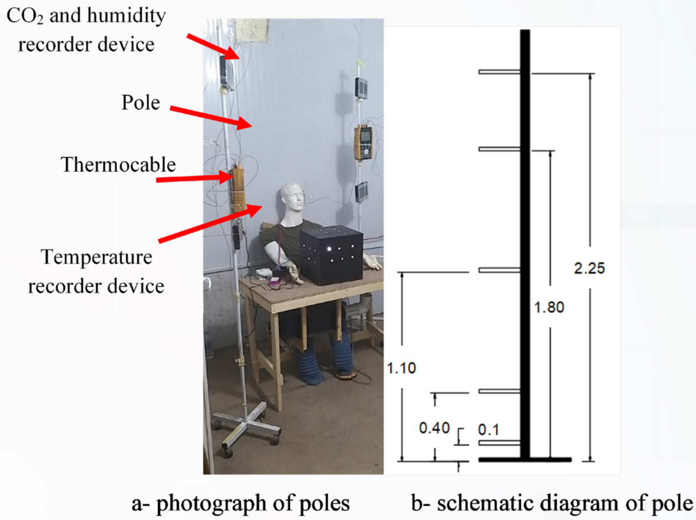
During experiment tests, 15 thermocouples were placed on side walls and floor (three thermocouples at each wall). Four thermocouples were placed on chilled ceiling and 15 thermocouples fixed on the three poles to measure air temperature distribution with high. CO_2 concentration and humidity measurement systems are based on a set of CO_2 and humidity recorder devices Fig.8b. Fixed at six points in different locations of test room at breathing zoon for the sitting and standing person as (0.5, 1.1, 1.25), (0.5, 1.8, 1.25), (1.85, 1.1, 0.4), (1.85, 1.8, 0.4), (2.1, 1.1, 1.85) and (2.1, 1.8, 1.85). Supply water flowrate was measured by Platon flowmeter device as show in Fig.8c and air supply temperature and velocity was measured by thermos-anemometer device, Fig.8d.

a b c d

Fig.8 measurement devices.

5.1 DESCRIPTION AND ARRANGEMENT MEASUREMENT DEVICES :

Three vertical poles are placed in test room. Each pole has five levels from 0.1m to 2.25m as show in Fig.9. These poles carried thermocouples, CO2 measurement and humidity devices to cover environment in occupied zone.



a- photograph of poles b- schematic diagram of pole

Fig.9 distribution of the measurement devices on the poles

The Poles inside the tested room puts at three locations as show in Fig.10. The first pole (Pole-1) is located near the supply diffuser and other poles (Pole-2 and Pole-3) puts near the thermal manikin. Decay method will be used to measure the local mean age of air at study state (ASHRAE Fundamentals Handbook., 2001, Yang et al., 2017, Bartak et al., 2001). The air entering the room is marked with a gas (the tracer gas), and the concentration of that tracer gas is recorded at the location of interest, the time needed to record the concentration is the air age. This assumes that the tracer gas behaviors the same as the air. The concentration of carbon dioxide is measured every 100 second, starting from (1000 ppm) at the moment of the DV/CC system is turned on, in order to observe the unique time period to reach the natural concentration (ambient concentration). This method gives a good idea about the system's efficiency for removing pollutants. Temperature distribution with height is measured at study state condition.

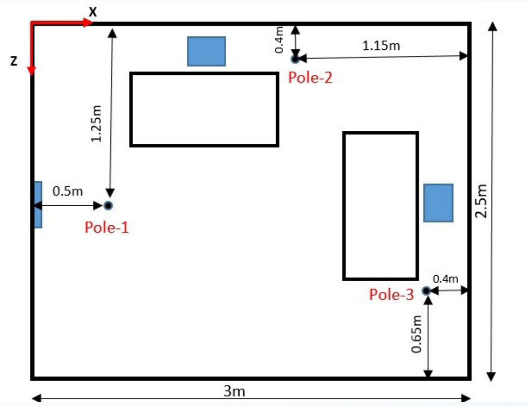


Fig.10 locations of the measurement poles.

6. RESULTS AND DISCUSSION :

Displacement ventilation with chilled ceiling system was studied experimentally in an office room at summer conditions under Iraqi- Hilla city climate. The study was divided in to two cases depending on the shape of air supply diffuser by used rectangular diffuser for case-I and semicircular diffuser for case-II. Each case has four tests based on the portion of cooling load treated by chilled ceiling (η) as (0%, 25%, 50%, and 80%). The temperature for displacement ventilation air supply is varied as (20°C, 21.5°C and 23°C and 24°C) at constant air flow rate for two cases which cover (100%, 75%, 50% and 20%) of cooling load respectively. Table (4) lists the main measuring parameters obtained in the experimental tests.

Fig.11 shows the local air age at study steady conditions for three poles at different value of (η) for two cases (at 1.1m and 1.8m height in each pole). Notes that the local air age at 1.8m level is higher compared with level at 1.1m in two cases for different value of (η). that's mean the local air age increase with height regardless of the shape of diffuser and value of cooling load treated by chilled ceiling. Also notes that the air age increasing with increase portion of cooling load treat by chilled ceiling due to mixing between hot air rise and cold air down after it cold by chilled ceiling by convection and tend to reduce the air speed. For each measured point the local air age when used rectangular diffuser is higher than of air age when used semicircle diffuser at six local points and at each value of (η) although the airflow was constant in each test. This result shows that the semicircle diffuser is more efficient in air exchange from rectangle diffuser. At pole-2 which located between two occupants (between heat sources) as shown in Fig.10, notes that the local air age at 1.8m height is minimum compared with the air age value at other points regardless of value of (η) due to the air in this zone is hotter than other zones in office room, that cause base of the plumes leads to increase in air velocity (as proved by Rees and Haves., 2015), increase in air velocity lead to decrease air age.

					Displacement ventilation			Chilled ceiling				
Case	Test	η %	T_{av} °C	RH_{av}%	CL_{DV} W	T_s °C	T_e °C	CL_{CC} W	$T_{w,s}$ °C	$T_{w,r}$ °C	Q_{ws} m^3/h	T_{cc} °C
Case-I	Test - 1	0	25.2	55.4	370	20	27.4	0	-	-	0	-
	Test - 2	25	24.9	55.9	277.5	21.5	26.2	92.5	19.7	20.25	0.144	22.5
	Test - 3	50	26	54	185	23	27.1	185	18.4	18.93	0.288	21.2
	Test - 4	80	26.27	56.9	74	24	26.8	296	17	17.56	0.5	19.4

					Displacement ventilation			Chilled ceiling				
Case	Test	η %	T_{av} °C	RH_{av}%	CL_{DV} W	T_s °C	T_e °C	CL_{CC} W	$T_{w,s}$ °C	$T_{w,r}$ °C	Q_{ws} m³/h	T_{cc} °C
Case-II	Test - 5	0	25.27	55.2	370	20	27.8	0	-	-	0	-
	Test - 6	25	25.3	59.2	277.5	21.5	26.85	92.5	19.2	19.8	0.144	22.3
	Test - 7	50	25.6	54.7	185	23	26.56	185	18.4	19.1	0.288	21.4
	Test - 8	80	26.1	53.5	74	24	26.75	296	16.8	17.27	0.5	19.5

Table. 4 the main measuring parameters obtained in the experimental tests.

Figures.12 and 13 show the temperature distribution with height for each pole at different value of (η). Layer stratified was one of the hallmarks of displacement ventilation. Supply air at low temperature flows along the floor, after that the air is heated when basses through the heat sources and move upward due to effects of buoyancy. For two cases and for three poles notes that the temperature rise with height at each value of (η), also notes that the temperature stratification decreases with increasing the portion of load treated by chilled ceiling due to effect of low chilled ceiling surface temperature. The air temperature at the upper zone (above 1.8m height) will be decreasing due to convection with chilled ceiling and then the air being move to the lower zone led to minimize the air temperature in the zone below 1.8m height and cause decrease in temperature different with height. These results are similar to the founding by Schiavon. et.al., (2012) they studied room air temperature stratification in the CC/DV system by used a different numbers of ceiling panels as shown in Fig.14.

Now if you compared air temperature distribution with height between two diffuser types notes that the air temperature distribution with height by used semicircle diffuser (case-II) was lower compared with rectangular diffuser (case-I) when used chilled ceiling regardless of (η) value at three measuring poles as shown in Fig.15. The exhaust temperature by used semicircle temperature (as shown in table (4)) is higher compared with rectangular diffuser at each value of (η), these results give preference to semicircle diffuser in term of temperature distribution efficiency and explain effect of supply airflow spread method by 180° with semicircle diffuser while the air supply is flow in one way direction with rectangular diffuser.

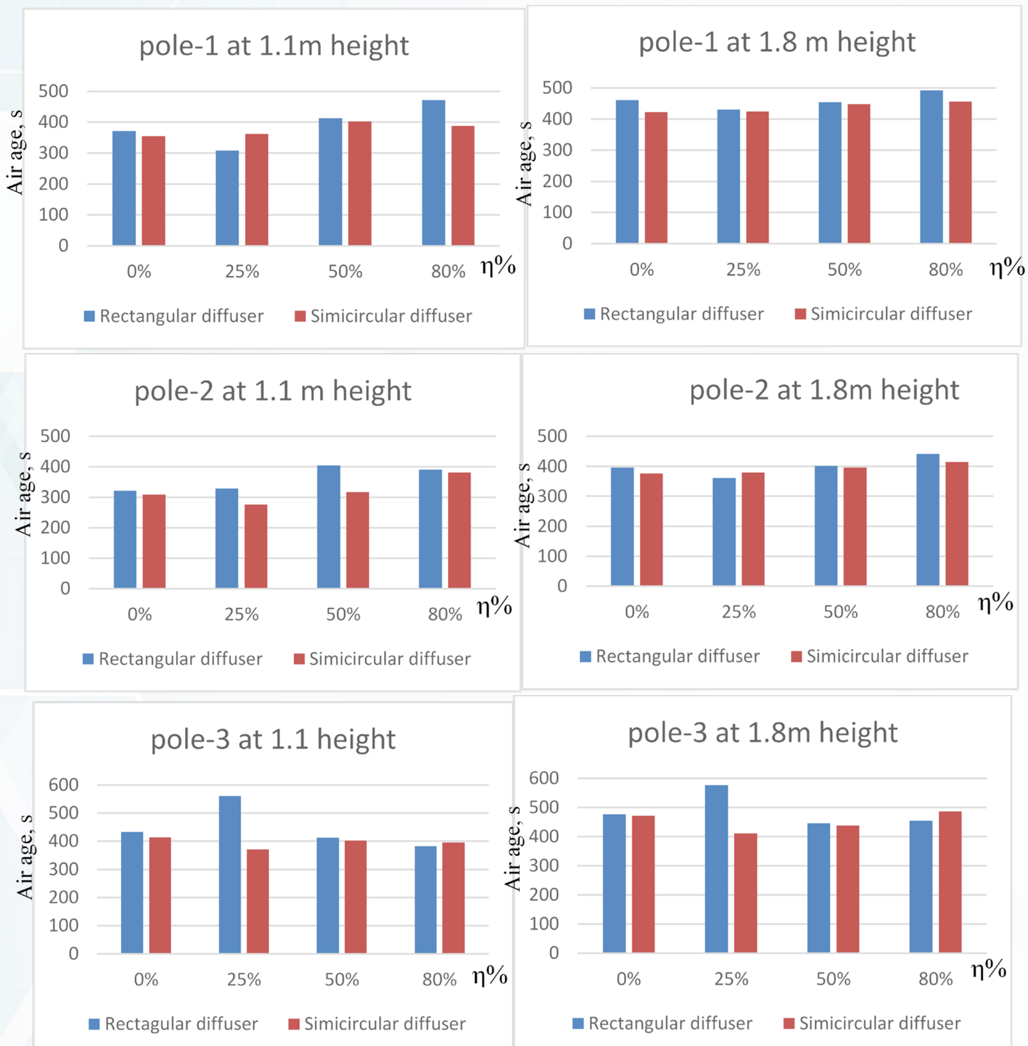


Fig.11 local air age at 1.1m and 1.8m height for three poles at different value of (η) for two cases

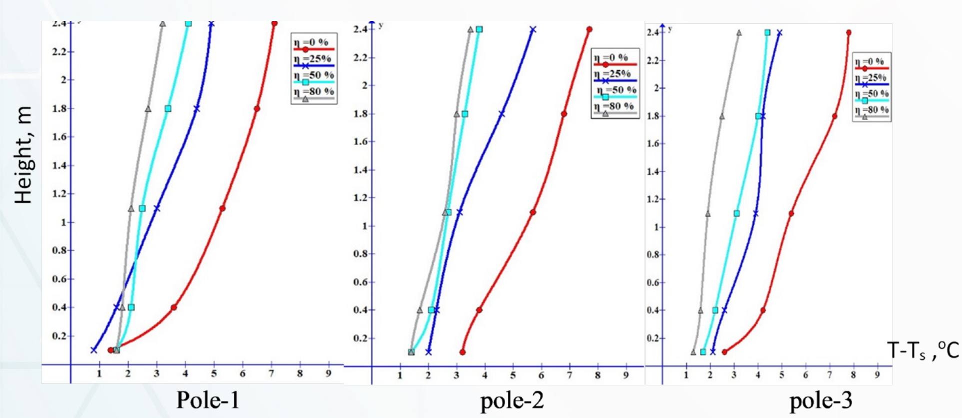


Fig.12. temperature distribution with height at different value of (η) for three poles (case-I)

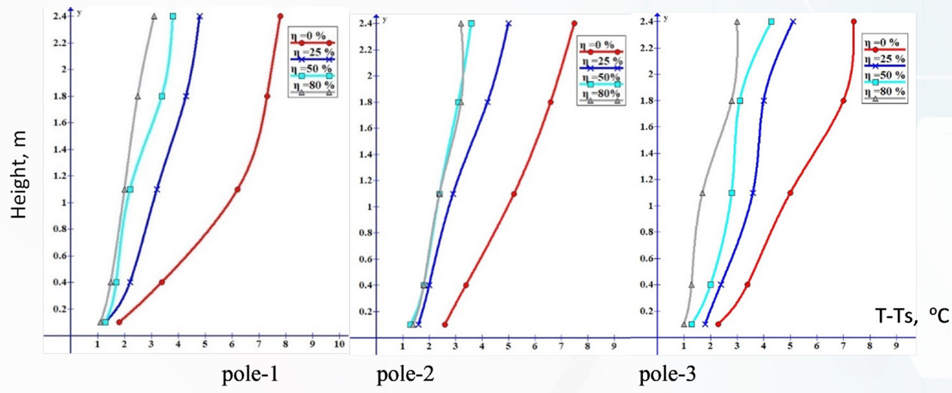


Fig.13. temperature distribution with height at different value of (η) for three poles (case-II)

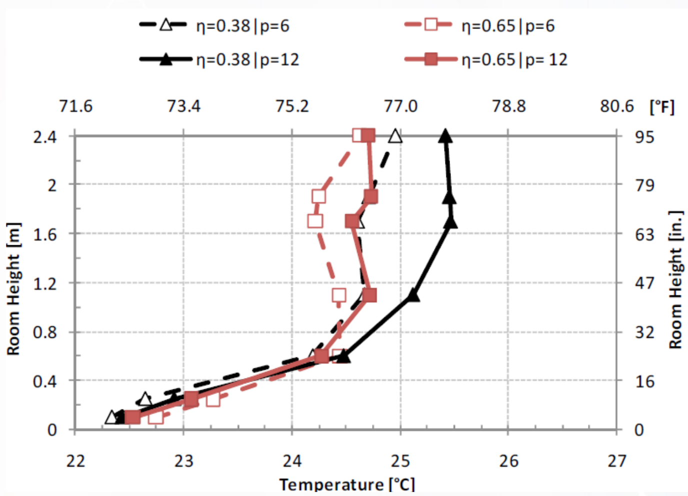
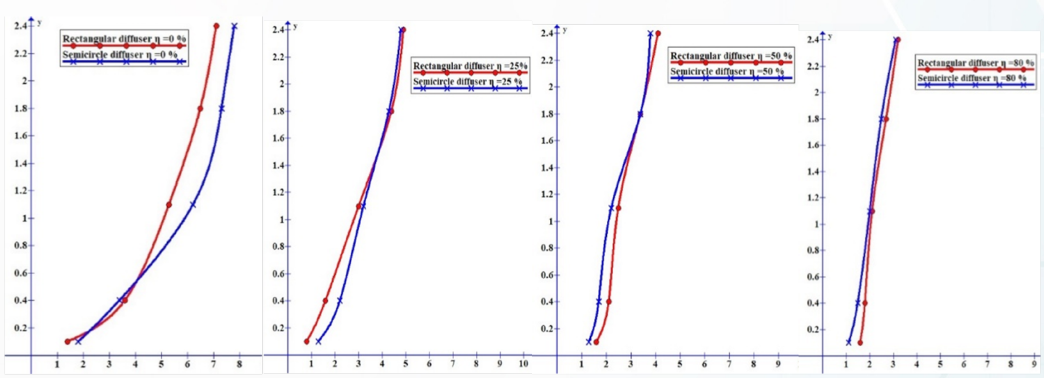


Fig.14. Temperature profiles for test η=0.38 and η=0.65. (Schiavon et al., 2012)



Pole-1: η =0 % η =25 % η =50 % η =80 %

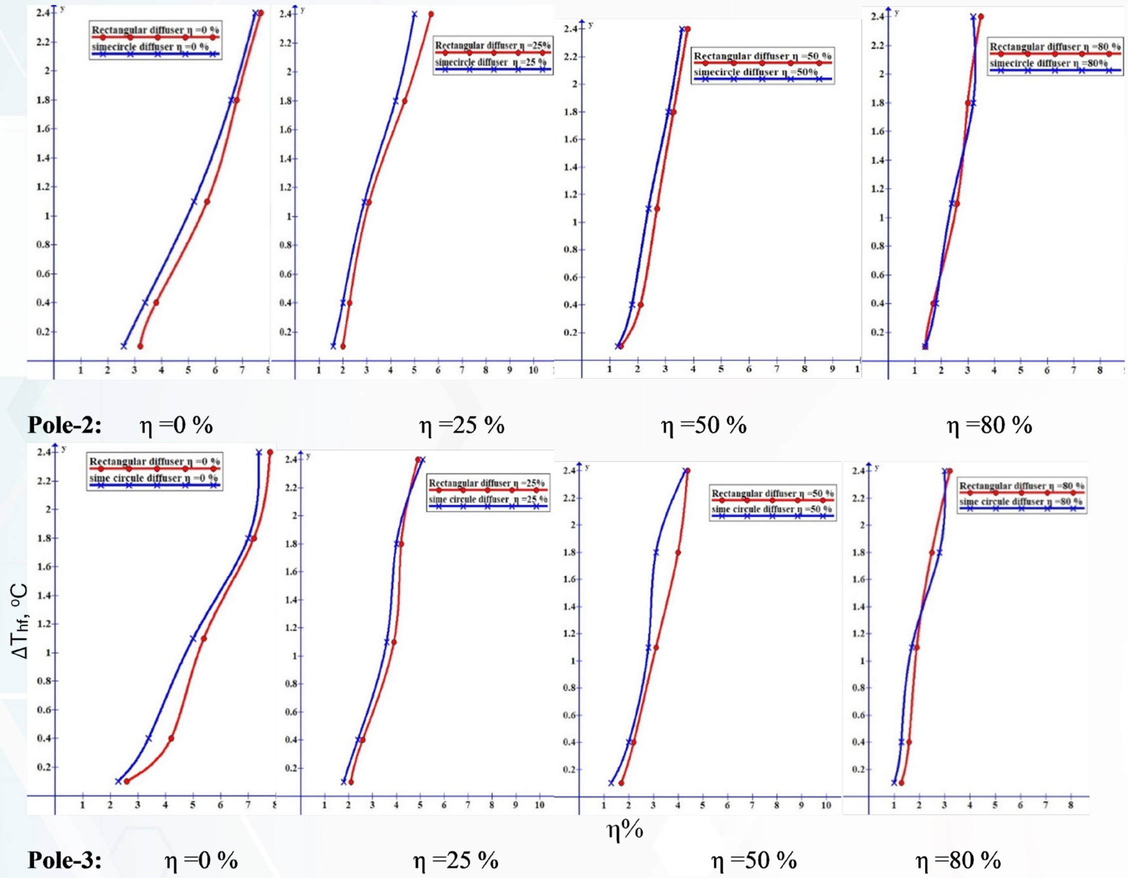


Fig.15temperature gradient with height for two cases at different (η) for three poles

The temperature difference between head and foot for a seated person (between 0.1m-1.1m height) was specified by ASHRAE Standard (2010) about 2°C. The study finds that the value of temperature difference between head and foot for a seated person decreasing with increased portion of cooling load treated by the chilled ceiling as shown in Fig.16. The decrease in air temperature gradient with (η) between 25-80% lead to decreasing in temperature difference between head and foot of a seated by (1.14°C) when used rectangular diffuser while its decrease by (1°C) when used semicircle diffuser. Notes the temperature difference between head and foot by used rectangular diffuser is lower compared with semicircle diffuser at test without chilled ceiling (η =0) while at other tests (η value equal 25-80%) notes that the value of (ΔT_{fh}) by used semicircle diffuser is lower compared with the rectangular diffuser. This means that increasing the value of (η) tends to reduce the temperature stratification for two cases. This result explains clearly the effect of chilled ceiling temperature on thermal comfort. The cooling load factor (η) cannot be counted as a unique factor to predict stratification prediction, because the chilled ceiling area respect to ceil room area and temperature of the chilled ceiling are important factors (Schiavon et al., 2012). Fig.17 shows the temperature difference between head and foot as a function of chilled ceiling temperature for two cases. Note that the (ΔT_{hf}) directly proportional with chilled ceiling temperature (increase chilled ceiling temperature mean decrease in the portion of cooling load treat by chilled ceiling), then it agrees with the results from Fig.16.

These results are consistent with Tan et. al. (1998) and Ghaddar. (2008), they found that stratification was reduced with increase cooling load treated by the chilled ceiling or by reducing the displacement air flow rate at a constant supplyair temperature.

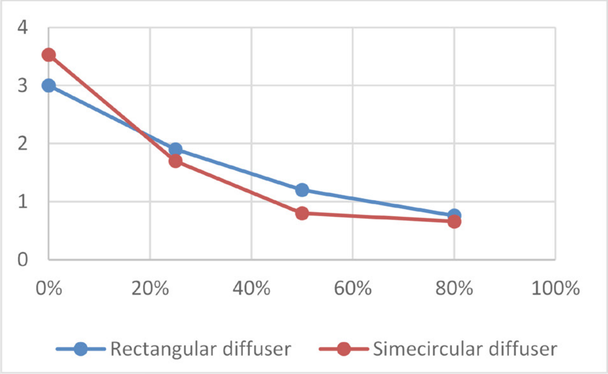


Fig.16. temperature difference calculated between head and foot for seated occupant (1.1 - 0.1 m) as function of the chilled ceiling cooling load ratio (η)

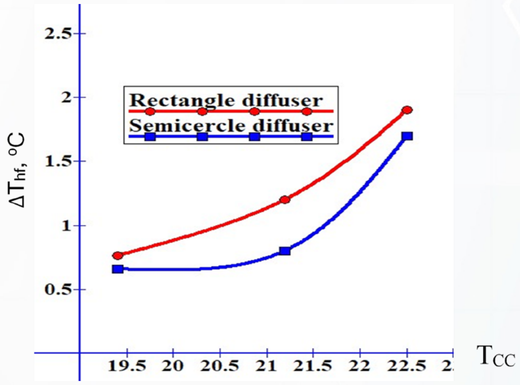


Fig.17. temperature different between head and foot for set person as a function of chilled ceiling temperature.

Fig.18 shown the relation between cooling load ratio (η) and thermal efficiency for two cases. In general, the low temperature of the chilled ceiling means increases cooling load treated by chilled ceiling. A decrease in the chilled ceiling temperature leads to decrease air temperature which contact with it, so it is noted that the exhaust air temperature decreases with the increased value of cooling load ratio (η) and lead to decrease in the temperature distribution effectiveness (ε_t) depend on the Eqn.8, (Awbi., 2008).

$$\varepsilon_t = \frac{T_e - T_s}{T_{av} - T_s} \qquad \text{.......................... 8}$$

Fig.18 shows clearly that the temperature distribution effectiveness (ε_t) by used semicircular diffuser is high compared with rectangular diffuser and gives a good idea of the advantage of the semicircle diffuser.

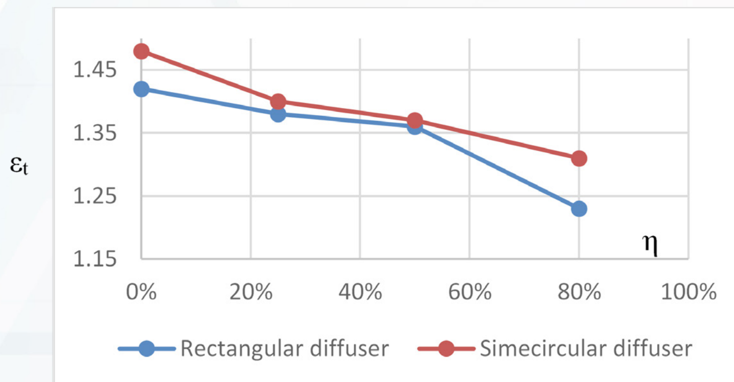
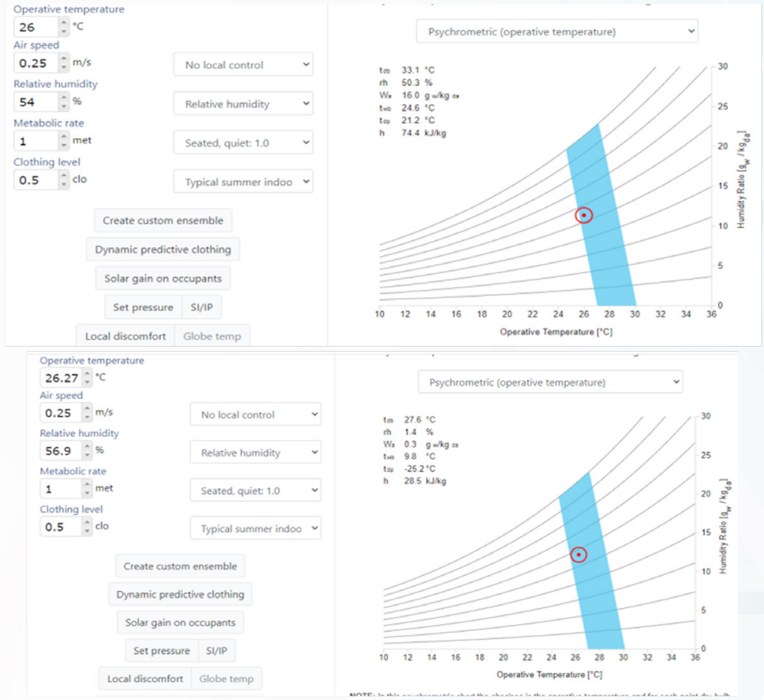


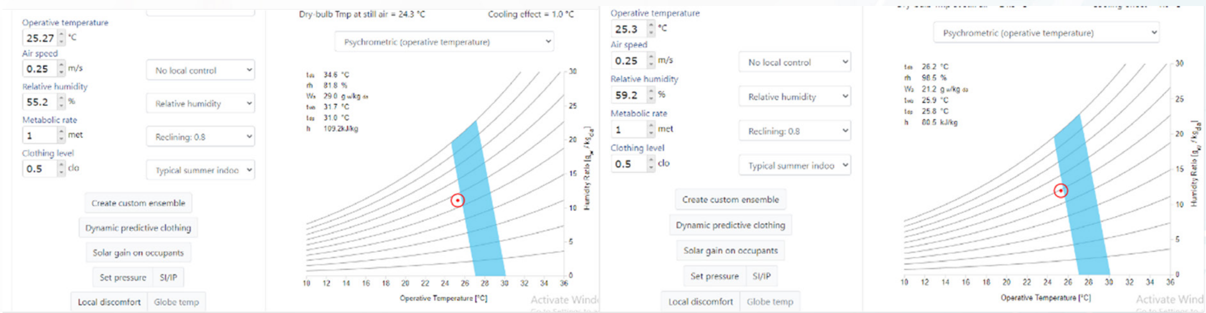
Fig.18 temperature distribution effectiveness with (η) for two cases

η= 0% η=25%

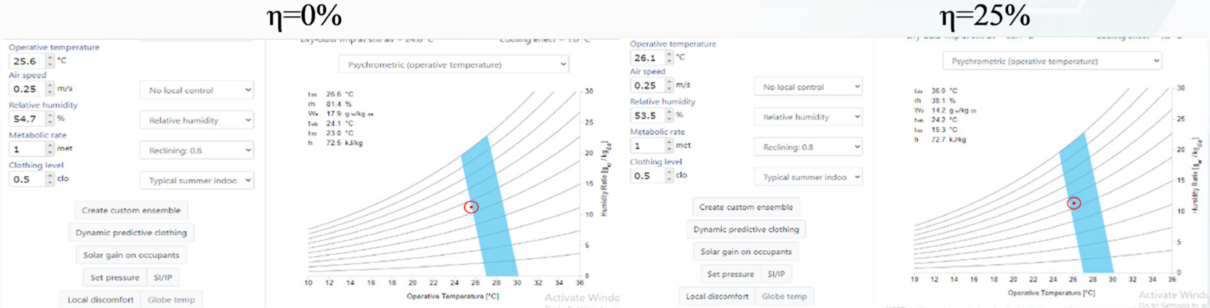


η=50% η=80%

Fig.19 thermal comfort conditions using CBE tool for four tests (Case-I)



η=0% η=25%



η=50% η=80%

Fig.20 thermal comfort conditions using CBE tool for four tests (Case-II)

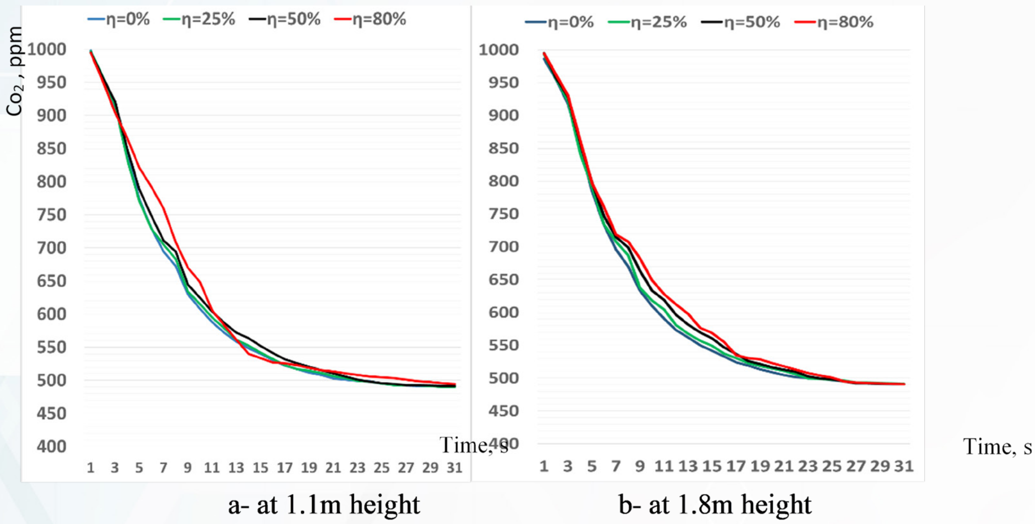


Fig.21CO_2 concentration with time per one hundred at different value of (η) for case-I

To predict thermal comfort for eight tests under Iraq-Hilla climate, CBE tool (Center for the Built Environment) is used. This tool produced by building energy center to calculation of thermal comfort conditions according to the ASHRAE standard 55, (2017). The tool depends on the average room dry bulb temperature, air speed and average relative humidity.

Figs.19 and 20 show the thermal comfort conditions by using CBE tool for case-I and case-II respectively at steady state conditions. By observing the location of the red circle point with respect to thermal comfort zoon (blue color zoon), notes that the thermal comfort does not achieve at η=0% (test without chilled ceiling) and(η=25%), while notes the thermal comfort satisfy at (η=50% and 80%) and complies with ASHRAE standard 55, (2017) for two cases. These results mean that increase in the portion of the cooling load treated by the chilled ceiling leads to more thermal comfort in the occupied zoon.

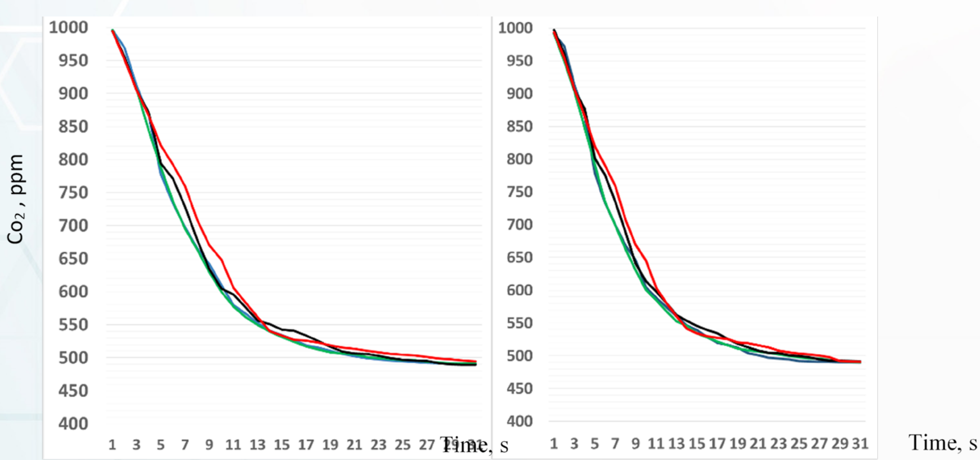
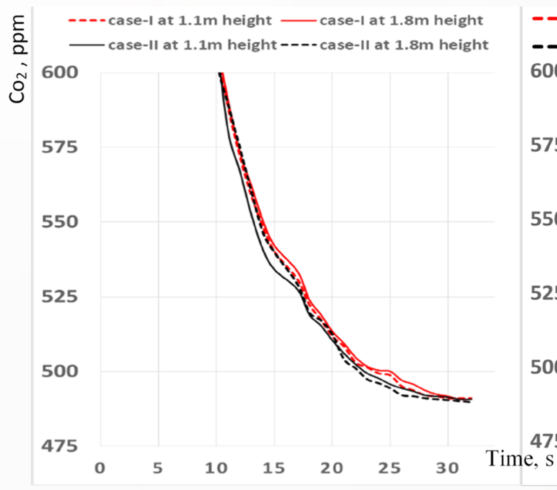


Fig.22 CO2 concentration with time per one hundred at different value of (η) for case-II

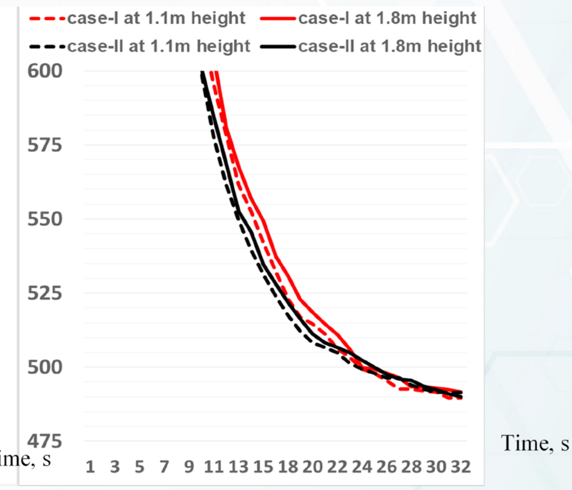
Figs.21 and 22present average measured of carbon dioxide concentrations vs. time at two height levels for breathing zone at seating and standing persons for two cases respectively at a different value of (η) where (a) at 1.1m height level and (b) at 1.8m height level. All eight tests show variations in carbon dioxide concentration with a value of (η). At the time between (0-500s) at CO_2 concentration (1000- 800 ppm) respectively, notes that the value of (η) have been little effect on the CO_2 concentration removing, after that the CO_2 concentration removing with time decrease with increase portion of cooling load treated by chilled ceiling until reach to the concentration for air supply. The remove of CO2 concentration associated with air movement in space room, decrease chilled ceiling temperature leads to decrease air temperature by convection and move down, cause's obstruction of air removal from the exhaust and decreases Co_2 remove.

Variations in CO_2 concentration with office room height is shown in Fig.23. At the value of (η) between (0-50%) for two cases notes that the Co_2 concentration at 1.8m height is more by about (10-20ppm) than the concentration at 1.1m height, while at (η=80) notes that difference between concentration at two levels (1.1m and 1.8m) be fading due to cooled air in the upper zone. These results compared with the results found by Jimenez H. et.al. (2018), it's showed acceptable converge. He studied decay of Sulfur hexafluoride (SF6) with time at different room levels as shown in Fig.24.

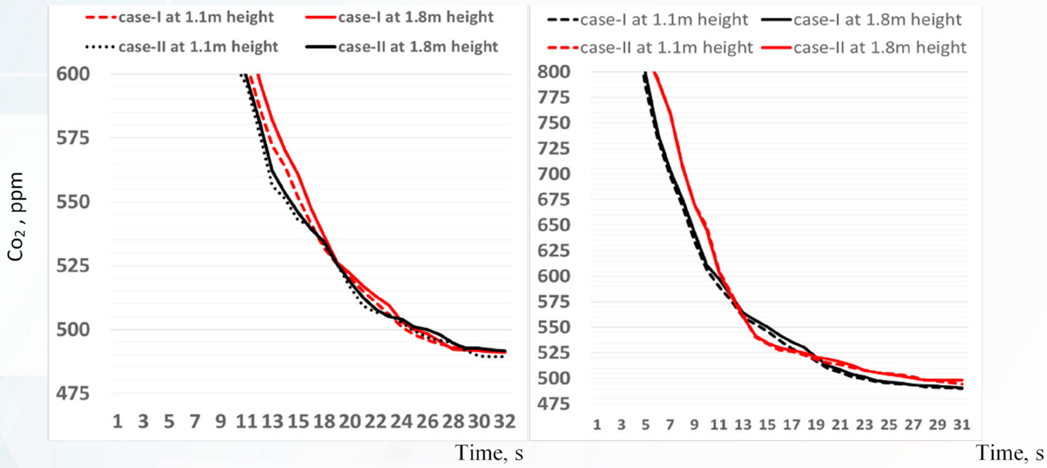
The shape of the air supply diffuser has a direct effect on the remove CO_2 from space, which clearly in Fig.23, notes that the CO_2 removed by the used semicircle diffuser is faster than rectangle diffuser at any value of (η).



η = 0%



η = 25%



η = 50% η = 80%

Fig.23 CO2 concentration with time one hundred at two levels for two cases and different (η)

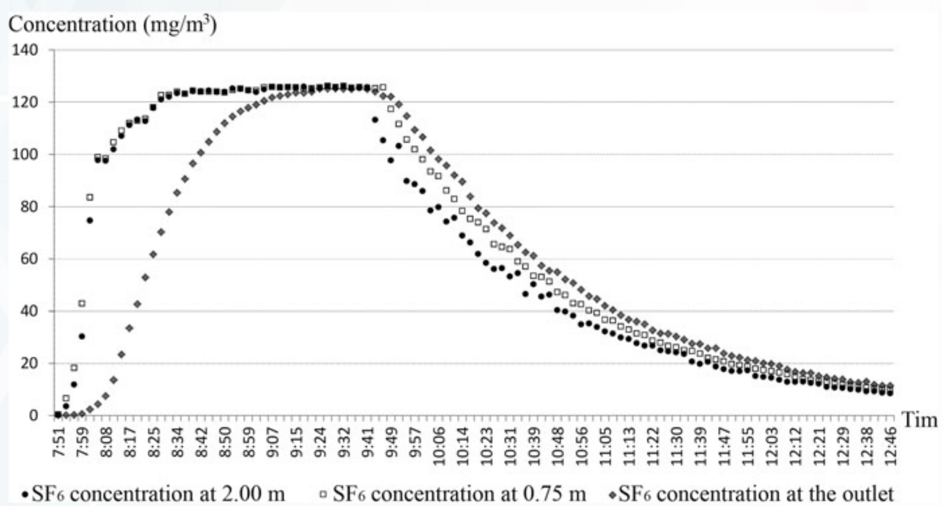


Fig.24 decay SF6 concentration with time at different height levels. (Hormigos-Jimenez, 2018)

7. CONCLUSIONS :

The study is based on the portion of cooling load treated by chilled ceiling respect to the total cooling load in the office room at a constant air flow rate supplied by two types of air supply diffusers by used CC/DV system. The cooling load ratio treated by chilled ceiling varies from (0% to 80%).

Eight experimental tests presented effects of supply diffuser shape and portion of cooling load treated by the chilled ceiling on indoor air age and comfort level in the office room under displacement ventilation with chilled ceiling system in the hot and dry climate (as Iraqi-Hilla city climate).

the novelty of this work is used displacement ventilation with chilled ceiling system under a range of operating conditions in a hot and dry climate (as Iraq-Hilla city climate) to understanding the indoor air age and analyses thermal comfort parameters.

The experimental main conclusions are found as follows:

1- The local air age increases with height regardless of the shape of the diffuser and the value of cooling load treated by chilled ceiling. The local air age increases with increase cooling load treat by the chilled ceiling with a used rectangle diffuser or semicircle diffuser.

2- The temperature stratification decreases with increased portion of load treated by chilled ceiling and the air temperature distribution with height by used semicircle diffuser is lower compared with rectangular diffuser regardless of (η) value. The temperature difference between head and foot for a seated person decreasing with increase portion of cooling load treated by the chilled ceiling and it's directly proportional with chilled ceiling temperature.

3- The temperature distribution effectiveness decreases with increase portion of cooling load treated by the chilled ceiling and used semicircular diffuser gives better effectiveness compared with the rectangular diffuser.

4- CO_2 concentration in DV/CC system increase with height but this difference is fading with increase portion of cooling load treat by the chilled ceiling and lead to decrease in CO_2. The CO_2 removed by the used semicircle diffuser is better compared with the used rectangle diffuser at any value of (η).

Nomenclature		Sub – Scripts	
A	.surface area, m2	av	average
Cp	specific heat, J/Kg.oC	cc	chilled ceiling .
.m	air mass flowrate, Kg/s	dr	design room
q	heat gain, W	e	exit
Q	.s/air supply flowrate. m3	ex	external
u	air supply velocity	f	floor
T	temperature, oC	hf	.head to foot level
x,y,z	coordinates direction in a cartesian	l	overhead light

Nomenclature		Sub – Scripts	
Greek letters		n	nominal
η	The percentage of the cooling load covered by chilled ceiling to the total cooling load	oe	occupants and equipment
		s	supply
ρ	air density, kg/m۳	w	water
ε	Efficiency	ws	water supply
TΔ	temperature difference, oC	wr	water return
Abbreviations			
ACH	h/۱ ,air change per hour		
amp	ambient temperature. oC		
CL	cooling load, W		
DV	displacement ventilation		
RH	.٪ ,relative humidity		

Table. 3 ambient temperature, Co2 concentration and relative humidity at test days

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