Review Article

Experimental and Numerical Study of Air Flow Diffusion and Contaminants Circulation in Room Ventilation Related with Iraqi Climate

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To cite this article:

Alaa Abbas Mahdi, Zahraa Hassan Jasim. Experimental and Numerical Study of Air Flow Diffusion and Contaminants Circulation in Room Ventilation Related with Iraqi Climate. *International Journal of Mechanical Engineering and Applications*. Vol. 4, No. 2, 2016, pp. 24-42. doi: 10.11648/j.ijmea.20160402.11

Abstract: Illnesses of many indoor air quality problems occur in office room. Ventilation is one way to control the contaminant transport and to provide better indoor air quality with in the office. In the evaluation of indoor air quality, CO₂ concentration is regarded as a good indicator to estimate the air quality level and to assess the performance of a mechanical ventilation system used by many designers, So the CO₂ concentration was used as the tracer gas in this study, also the humans respiration taken into account as CO₂ sources were the rate of production of carbon dioxide (CO₂) by human respiration. Experimental measurement and computational fluid dynamics (CFD) simulation methods were applied. The results from this study show that the floor-supply displacement ventilation can improve indoor air quality because the pollutant concentration in the breathing zone is lower than that of mixing system and the risk of cross contamination can be effectively reduced. Nevertheless, the indoor spaces with floor-supply displacement ventilation might have a higher risk of discomfort, because of high temperature stratification between the ankle and head levels when compared to traditional mixing ventilation. The results indicated that the contaminant distribution in a mechanically ventilated office room need to be studied individually according to different cases.

Keywords: Mixing Ventilation (MV), Displacement Ventilation (DV), Computational Fluid- Dynamics, Thermal Comfort, Indoor Air Quality

1. Introduction

The quality of indoor environment and energy performance of ventilation highly depends on airflow patterns generated within a room and airflow distributions from air supply devices. Generally, room air distribution methods can be classified into a few classes mixed systems, stratified systems partially mixed systems (both mixing and stratification); and task/ambient (conditioning only for a certain portion of the space) conditioning systems, [1]. The mixed system or so called mixing ventilation (MV) assumes that fresh air delivered from the HVAC systems will completely mix with the indoors contaminants to reduce the concentration level of the pollutants to an acceptable level. However, a complete mixing is difficult to achieve and as a result, the concentration level in some

parts of an indoor space may exceed the permitted level. In addition, the complete mixing could enhance cross contamination between occupants due to the re-circulation inside the room. Displacement ventilation (DV) is the most widely used variant of the fully stratified systems in which room air flows provides fresh air directly to the occupied zone. Heated objects, such as the occupants and equipment, will bring the contaminants to the upper zone through the thermal plumes generated by the heat. Return exhausts in displacement ventilation are located at or close to the ceiling through which the warm air with higher pollutant concentrations is removed. The most common configuration for displacement ventilation supplies air from a diffuser from a low side-wall. Unfortunately, the airflow in such displacement ventilation is not one-dimensional in the occupied zone, [2]. These recirculation present the risk

of cross contamination between the occupants, thus it is better to use ventilation systems without the risk of cross contamination. this investigation is to examine the performance of displacement ventilation systems and mixing ventilation systems under high cooling load. More specifically, the investigation is to:

- To investigate air diffusion and contaminate concentration, indoor air quality and comfort issues by CFD (computational fluid dynamics) modeling in ventilation room related with Iraqi climate.
- To compare the performance of mixing and displacement ventilation system in removal of pollutants concentration inside Iraqi offices rooms.

2. The Experimental Equipment and Procedure

Figures (1 a, b) shows the experimental office room (non-isothermal condition) to mimic displacement and mixing ventilation system. The test chamber has the inner dimension (length x width x height): (4 x 3.5 x 3.75 m), which is equivalent to a typical two-persons office, two computers, and two lights, a full description of tested room shown in table (1). The chamber walls for this office consists of multi layers (gypsum, cement plaster, common brick) depending on the Iraqi code of building, [3]. The experiments have been carried out at ambient laboratory condition of approximately (25-28)°C as temperature and 1 bar as pressure which done in the department of mechanical engineering at university of AL- Kufa. It was carried out during three weeks in May 2015, each week on five days and each day for 5hrs from 8: 00 am to 2: 00 pm. The air cooling system was operated until reach the temperature inside the tested rooms to operating limit of temperature as (23°C). Appropriate points have been chosen in the domain of rooms dimension in order to measure the thermal comfort inside the tested rooms. A hot wire thermo-anemometer, carbon dioxide "model GCH-2018" devices which used to measure the air velocity, temperature and CO₂ concentration of indoor air respectively. They were accurately calibrated according to manufacturer specifications. The sample unit consists of five divisions located at different height (0.4, 0.8, 1, 1.4, 1.8) m used to measure the parameters which located at (x = 1.5 m, 3 m) from south wall and located at (x = 1.5 m, 3 m) from south wall and (z = 2.3 m) from west wall as shown in figure (1), all doors and windows were close during experiment. According to Iraqi standard for cooling and to ASHRAE Standard [4], the indoor design temperature, velocities supplied and contaminant concentration air conditioning systems at AL-Najaf city listed in tables (2, 3).

(a)

(b)

(a) Displacement ventilation (DV), (b) Mixing ventilation (MV)

Fig. 1. Schematic drawing of the office room.**Table 1.** Room configuration.

Item	Location (m)			Size (m)			Heat (W)
	x	y	Z	Δx	Δy	Δz	
Room	0	0	0	4	3.75	3.5	-
DV	3.9	0.1	2	0.1	0.6	0.4	-
MV	3.9	2	2.5	0.1	0.2	0.2	-
Exit air	0	1.6	0.65	0	0.4	0.4	-
Person 1	0.5	0	0.4	0.4	1.1	0.35	75
Person 2	3.25	0	1.2	0.4	1.1	0.35	75
Computer 1	0.55	0.8	1.2	0.31	0.37	0.46	45
Computer 2	3.25	0.8	0.1	0.31	0.37	0.46	45
Light 1	1	3.7	1.25	0.1	0.05	1	140
Light 2	2.9	3.7	1.25	0.1	0.05	1	140

Table 2. Outdoor data for Iraq, [3].

Region	DBT	RH	DR	Altitude	Latitude	Longitude
AL_Najaf	48°C	14%	17°C	50	32.0°	44.32°

Table 3. Indoor data for Iraq, [3].

	DBT (°C)	RH%	Air velocity (m/s)
Human comfort	19 - 24	40 - 60	1.8 - 2
Recommended conditions inside the office	23 - 26	40 - 50	0.13 - 0.23
The maximum allowable CO ₂ in a room is 1000 parts per million by volume (ppm)			

3. Numerical Simulation

The numerical simulations were done using the FLUENT version (3.2.26) and GAMBIT with RNG turbulence model to create and grid the rooms geometries and then to simulate the room air distribution and concentration in enclosures to solve the governing equations for the conservation of mass, momentum, heat, contaminant concentrations, and turbulence quantities. The CFD program used the Re- Normalization Group (RNG) k-ε model because it is slightly more accurate than the standard k-ε model for indoor airflow simulations, [2]. The airflow transport can be described by the following time-averaged Navier– Stokes equation:

$$\text{div}(\rho V\phi - \Gamma_{\phi, \text{eff}} \text{grad } \phi) = S_{\phi} \quad (1)$$

where ρ is the air density (kg/m^3), $\Gamma_{\phi, \text{eff}}$ is the effective diffusion coefficient (kg/m.s), V is the air velocity vectors (m/s), S is the source term of the general flow property, and ϕ is any one of the components shown in table 4. When $\phi = 1$, the general equation becomes the continuity equation. The effective diffusion coefficients and the source terms for different ϕ are listed in table 4. In the table, the effective viscosity, μ_{eff} , is the sum of molecular viscosity and turbulent viscosity.

Table 4. Terms, coefficients and constants in Eq. (1).

Equations	ϕ	$\Gamma_{\phi, \text{eff}}$	S_{ϕ}
Continuity	1	0	0
Momentum	U_i	μ_{eff}	$-\frac{\partial p}{\partial x_i} + g_i(\rho - \rho_o)$
Turbulence kinetic energy	k	$\mu_{\text{eff}}/\sigma_k$	$P_k + G_b - \rho\varepsilon + G_k$
Turbulence kinetic energy dissipation rate	ε	$\mu_{\text{eff}}/\sigma_{\varepsilon}$	$\frac{\varepsilon(C_1 p_k - C_2 \varepsilon)}{k} + \frac{C_3 G_k \varepsilon}{k} - R_{\varepsilon} + S_{\varepsilon}$
Thermal energy	T	$\mu_{\text{eff}}/\sigma_T$	S_T
Concentration	C	$\mu_{\text{eff}}/\sigma_C$	S_C

$$P_k = \mu_t(U_{i,j} + U_{j,i})U_{i,j}, \mu_{\text{eff}} = \mu_t + \mu = \mu \left[1 + \sqrt{\frac{C_{\mu}}{\mu}} \frac{k}{\sqrt{\varepsilon}} \right]^2, (\sigma_k, \sigma_{\varepsilon}, \sigma_T, \sigma_C, C_{\mu}, C_1, C_2, C_3) = (1.0, 1.314, 0.9, 1.0, 0.0845, 1.42, 1.68, 1.233).$$

The environmental chamber can provide controlled air supply and inlet temperatures depending on the total cooling load.

Step one: Determination of heat transfer from outside to inside, [3]:

$$Q = U * A * CLTDc \quad (2)$$

$$U = 1/Rt \quad (3)$$

$$R_t = \frac{1}{h_i} + \frac{x_1}{k_1} + \dots + \frac{1}{h_o} + \frac{x_n}{k_n} \quad (4)$$

CLTDc: Correct cooling load temperature difference, its calculate as:

For walls:

$$CLTDc = (CLTD + LM) \times K + (25.5 - T_i) + (T_m - 29.4) \quad (5)$$

Where:

LM: Corrector latitude and month,

T_i : inside room temperature and

T_o : outlet design temperature.

For doors and windows:

$$CLTDc = CLTD + (25.5 - T_i) + (T_m - 29.4) \quad (6)$$

$$T_m = T_o - D_R/2$$

The heating load transition through the window generated by solar radiation is evaluated as:

$$Q_r = A * SC * SHG * CLF \quad (7)$$

Step two: Determine the supply air temperature (T_s) and air flow rate. There are some steps to find ventilation rate and supply air temperature (T_s), [5]:

$$Q = (0.295q_{oe} + 0.132q_l + 0.185q_{ex}) / (\rho cp \Delta T_{hf}) \quad (8)$$

$$Ts = T_{sp} - \Delta T_{hf} - [(A_z \cdot q_t) / (0.584Q^2 + 1.208A \cdot Q_s)] \quad (9)$$

$$q_t = q_{oe} + q_l + q_{ex} \quad (10)$$

Then assume ($\Delta T_{hf}=3^\circ\text{C}$, [6], $T_{sp}=24^\circ\text{C}$, $q_{oe}=75\text{W}$, $T_o=47^\circ\text{C}$, [3]), the values of flow rate (Q) and air temperature supply (T_s) can be determine as list in table 5. The area of diffusers was changing depending on required inlet velocity. DFIC series type [7, 8], gives limit air supply velocity for each area. Table 6 shows the air diffusers which have been used in experiment work.

Table 5. Values of air flow rate(Q_s), supply air temperature(T_s) and air change per hour (ACH).

Parameters			Heat transfer (W)				
Q L/s	T_s °C	ACH	west wall	north wall	south wall	window conduction	Radiation
681.4	17	5.8	225.96	401.24	188	758.622	1008.87

Table 6. Different air distribution systems.

Air diffusers	Air distribution System	Supply pattern	Return pattern
Wall grille	Mixing ventilation	End wall mounted	Return opening below or beside supply terminal
Low speed air diffuser	Displacement ventilation	End wall mounted and ground mounted in the middle of the space	End wall mounted below ceiling space

3.1. Assumptions

To study the distribution of an additional chemical species, an extra variable is introduced in the CFD model. Both air as the main fluid and the additional chemical species representing CO_2 are treated identically as air at normal atmospheric pressure. In the present study, the flow characteristics are assumed to be steady, three-dimensional flow, Newtonian and incompressible fluid, no chemical reaction and turbulent flow. normal atmospheric pressure. In the present, the flow characteristics are assumed to be steady, three-dimensional flow, Newtonian and incompressible fluid, no chemical reaction and turbulent flow.

3.2. Mesh Generation

Mesh has been carried out in GAMBIT as triangle elements pave type for Quad square and Tet/Hybrid elements for volumes. After applied this mesh the number of meshes for the model was about (883439) cells.

3.3. Boundary Conditions

Correct simulation of airflow in a room by CFD depends on proper specification of the boundary conditions by the user. The surface temperatures of walls and window were listed in table 5 used as the boundary conditions in CFD simulations. In the evaluation of indoor air quality, CO_2 concentration is regarded as a good indicator to estimate the air quality level and to assess the performance of a mechanical ventilation

system used by many designers [9], so the CO_2 concentration being used as the tracer gas in this study as shown in figure (2) moreover, the humans respiration is taken into account as CO_2 sources where the rate of production of carbon dioxide (CO_2) is considered by human respiration.

Fig. 2. Photograph of CO_2 Concentration Source.

Velocity inlet boundary conditions for both air inlet and contaminant source were considered uniform and the flow was normal to inlet section. Contaminant inlet was assumed to be a gas-phase contaminant source. One advantage of taking the density of both constituents to be equal and constant is the possibility of making use of the Boussinesq approximation to model buoyancy effects. The standard FLUENT wall function (no slip, smooth and no diffusive flux of the species) was used. Under relaxation factors were manipulated to get quick convergence and solution was assumed to converge when the residuals for all scalars were less than or equal to 10^{-6} .

4. Results and Discussion

4.1. Experimental Results

Figure (3) show an experimental results of the isothermal contours for x-y plane in the office room with pollutant contaminant source. In which concentration of the contaminant from the panel source put on the table in front of the second person. In this figure, it can be seen that the temperature at lower part near the inlet is relatively low, also the temperature increases from (24°C) near the supply diffuser to reach about (32°C) near the heater, while it

reaches (35.5°C) in the exist regions of the plane. Figure (4) show the relation of air temperature with the room height in case there sources of pollutant, through y-axis at $x = 1$ and at $x = 3$ m. This figure shows that the temperature directly proportional with the height. Since in case of displacement ventilation system, the upward air movement due to the buoyancy force carried the heat to the upper part of the room. Figures (5, 6) show the turbulent which has been located in air distribution near and above the heater where the velocity increases to 0.5m/s in this region. Figure (7) displays the contour map of carbon dioxide concentration (ppm) for x- y plane in the office room. It can be seen that the values of carbon dioxide increases at the upper part of the plane. This is due to that the CO₂ releasing during the breathing of the persons and from the machines. The released amount of carbon dioxide were carried by the supply air stream (which contain low value of carbon dioxide) inside the domain towards the exhaust grilles. Figure (8) illustrates the relation of the carbon dioxide with the vertical distance through y-axis at $x = 1$ and $x = 3$ m. In this figure, the air temperature and pollutant concentration increases almost linearly with the room height for a floor surface source, Skistad 1988, Mundt 1990 proved that.

Fig. 3. Contours of experimental air temperate distribution through x-y plane.

Fig. 4. Measured air temperature distributions.

Fig. 5. Vector map of experimental air velocity results through x-y plane.

Fig. 6. Measured air velocity distributions.

Fig. 7. Contour map of carbon dioxide concentration(ppm) through x-y plane.

Fig. 8. Measured CO_2 concentration.

Fig. 9. Contour of experimental air through x-y plane.

Figure (9) illustrates the contour distribution of temperature values for x-y plane in the experimental office room domain. it can be seen that the temperature at upper part near the inlet is relatively low where the upper part has lower temperature than the other regions in the domain. This is attributed to the effect of cold air stream from the air inlet, which leads to reduce the temperature at region near the supply diffuser. The temperature increases from 24.9°C near the supply diffuser to reach about 26.45°C near the south wall. Figure (10) shows the relation of temperature with the vertical distance through y-axis. It can be seen that the temperature increases at $y = 1\text{m}$ especially at pole-1 due to presence of the heat load. Figure (11) displays the vector map of air velocity for x-y plane in the office room. It can be seen that the air velocity decreases from (1.5m/s) near the inlet supply air to reach (0.35m/s). In this figure, the air velocity at upper region is relatively high due to the effect of entering air to increase the velocity then, the velocity decreases through the domain of the room. Air

velocity decreases through the domain is due to the friction of air layers and the impact of air with items. Figure (12) illustrates the relation of air velocity with the vertical distance through y-axis at $x = 1$ and at $x = 3\text{m}$. It can be seen that the velocity reduces with increasing the distance from the center. This is because of increasing the distances from the inlet and the losses of air energy. Figure (13) expresses the contour map of carbon dioxide values in the experimental office room. From this figure, it can be seen that the values of the carbon dioxide increasing through the experimental room. This is due to the carbon dioxide that is released from the persons during the breathing and the machines through the experimental office room. Figure (14) displays the relation of the carbon dioxide values with the vertical distance through y-axis. The value of carbon dioxide decreases with increasing the vertical distance due to the air stream carries the carbon dioxide that releases from the machines and persons towards the exhaust grilles.

Fig. 10. Measured air temperature distribution. temperature distribution.

Fig. 11. Vector map of experimental velocity through x-y plane.

Fig. 12. Measured air velocity distributions air.

Fig. 13. Contour of experimental air distribution through x-y plane.

Fig. 14. Measured air temperature distribution.

4.2. Numerical Simulation

Figure (15) show the contour line of the temperature distribution for same case with pollutant source. The vertical air temperature increases and would not be acceptable in terms of thermal comfort, figure (16) show the increment in temperature for all whole room where maximum temperature reach to (40°C), it is clear that the temperature increases started just above 0.7m (at heat source level) with increases of the load. Figures (17 a, b) illustrates the air velocity vectors distribution patterns with displacement ventilation for a: center and floor planes, b: at $y = 0.4\text{m}$. These results were modeling for the tested room with pollutant source. The supply air velocity is 0.25m/s from the inlet at the floor level. The flow is circulated with symmetric eddies on both sides of the domain. This circulation is due to the impact of air stream by the ground and the obstruction of the items inside the tested room. Therefore, the stream lines will be accelerated in these regions. Accelerated stream lines will increase the velocity of adjacent lines. At the same time, these lines will be decelerated by the impact with the nearby stream lines, therefore velocities difference will be generated between

adjacent stream lines.

Figure (18) display the contours of air temperature distribution for mixing ventilation with CO_2 concentration at mid plane. The maximum air temperature just above the floor and the minimum air temperature at a height of (1.8 m) were increased. The maximum air temperature (33°C) occurs in the zone near the heat source. From the histogram of total temperature to the whole room the maximum temperature do not exceeding (38°C), figure (19) show that. Figures (20 a, b) shows the air flow pattern in the tested room with mixing ventilation and the containment source. Velocity of the air at inlet (2 m/sec) is reduced and reaches to the minimum value at the stagnation point in the center of recirculation zone at the floor level. In this figure, it can be seen that the values of air velocity distributed through the tested room from (0.37m/s to 0m/s) at $y = 0.4\text{m}$. Air velocity decreases with increasing the distances through the domain of the tested room. The air velocity in occupied zone is generally below (0.5 m/s) for DV and (1 m/s) for MV. The air velocity of DV is generally lower than that of MV in the occupied zones for all types of indoor space.

Fig. 15. Distribution of air temperature contours for DV with contaminate concentration.

Fig. 16. Histogram of total temperature for DV.

(a)

(b)

Fig. 17. Air velocity vectors distribution patterns for DV with contaminate concentration, a: center and floor planes, b: at 0.4m plane.

Fig. 18. Distribution of air temperature contours for MV with contaminate concentration.

Fig. 19. Histogram of total temperature.

(a)

(b)

Fig. 20. Air velocity vectors distribution patterns for MV. with contaminate concentration, a: center and floor planes, b: at 0.4m plane.

Fig. 21. CO₂ concentration for DV. at 0.4m and 2m planes.

Fig. 22. CO_2 concentration at mid planes for DV. at mid plane.

Fig. 23. CO_2 concentration for MV. at $y=0.4m$ & $y=2m$ planes.

Fig. 24. CO₂ concentration at mid planes for MV. at mid plane.

Figure (21) reveal to the CO₂ concentration distribution patterns for at $y = 0.4\text{m}$ and $y = 2\text{m}$ planes above the floor with displacement ventilation. From this figure, it is noticed that the values of the carbon dioxide increasing through the upper parts of room. It is observed that the high levels of concentration about 1040 ppm above the contamination source where the contaminant is flowing towards the ceiling, figure (22). show that. The concentration of CO₂ in the upper zone of the room is maximum due to lower ventilation rate in comparison with the other zones in the room and the clean air do not access to region near the north wall so, high levels of contaminated concentrated access in that region. Figures (23, 24) describes the CO₂ concentration distribution patterns with mixing ventilation. the middle of the room experienced good mixing of air. The back location of the room, particularly the area below the supply diffuser units, experienced poor mixing of air where air flow is inadequate. The result in a stagnant temperature and lower air velocity value may causes higher concentration. From the analysis above, when the supply inlet is located at the lower room zone, this region will be clean and the contaminant will disperse only in the upper zone. When the supply inlet is located at the upper zone, the contaminant will disperse all over the room regardless of the contaminant source location. It is widely believed that low energy displacement ventilation systems are better than traditional mixing systems at removing contaminants from a space. This is because there is a belief that these systems will use the same mechanism for

contaminant removal as they do for heat removal, where there are clearly more efficient, [10] proved that.

5. Conclusions

The major conclusions from this study are summarized as follows:

1. Local average concentration in breathing zone can be predicted based on the relative locations of the supply air, contaminant source, and the person.
2. Type and location of contaminant source can greatly affect local average contaminant concentration in breathing zone. However, even if under the worst condition, average concentration in breathing zone in DV has not much difference from a completely mixing ventilation system.
3. Increasing the velocity proves to increase air circulation within a space. In mixing ventilation this is the case required however, for displacement ventilation low mixing is required.
4. Mechanical ventilation should be employed in the offices room, especially, when there is large population density in office room.
5. From practical and economic terms, it cannot be applied displacement system for spaces with excessive cooling load because the higher cooling loads require re-air circulation greater than the mixing system to achieve conditions of thermal comfort in occupied zone.

Nomenclature

A	Surface area for wall. (m^2)	N	Total number of draft temperature points measured in occupied zone
C_p	Specific heat of the air at constant pressure. (KJ/Kg.K)	RH	Relative humidity%
DR	Daily Rang for outlet temperature. ($^{\circ}C$)	Q	Heat transfer through the wall. (W)
q_{ex}	Cooling load for the heat conduction through the walls and transmitted solar radiation. (W)	SC	Shadow coefficient.
q_l	Cooling load for the overhead lighting. (W)	SHG	Solar heat gain.
q_{oe}	Cooling load for occupants, desk lamps and equipment. (W)	ΔT_{hf}	Temperature difference from head to foot level. ($^{\circ}C$)
Q_r	Radiation heat transfer. (W)	T_r	Total mean temperature. ($^{\circ}C$)
h_o	Convection heat transfer coefficient for outside air. $W/m^2.k$	T_e	Exhaust air temperature ($^{\circ}C$)
h_i	Convection heat transfer coefficient for inside air. ($W/m^2.k$)	T_m	outlet design temperature. ($^{\circ}C$)
K	Color factor correct (Dark = 1.0, Med = 0.83, Light = 0.65)	T_{sp}	Setup (design)temperature. ($^{\circ}C$)
K_n	Conduction heat transfer coefficient for final layer of wall. (W/m.k)	U	Total heat transfer coefficient $W/m^2.K$

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