# Some Studies on Effect of Particle Source Locations on Ventilation Effectiveness in Displacement Ventilated Room

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## ABSTRACT

Ventilation effectiveness is the ability of the ventilation system to fulfill the design goals such as provision of fresh, healthy air to the occupants, dilution and removal of contaminants from the occupied zone of the mechanically ventilated room. Computational fluid dynamics (CFD) has become an efficient tool for determining the airflow, heat transfer and contaminant transport in the enclosed room environments and for wide range of other HVAC&R applications.

However, these CFD studies remain less relevant and useful unless and until these are backed up by proper experimental studies. Factors such as outdoor air quality, airflow rate, air supply method, moisture content, thermal environment and contaminant source location have greater impact on ventilation effectiveness.

In this paper, the effect of contaminant source location on ventilation effectiveness in a displacement ventilated room is investigated. Both experimental and numerical approaches are used. The results indicate that the contaminant removal effectiveness in occupied zone is higher if the contaminant source location is in the room as against the source location in the supply air and the contaminant removal effectiveness in breathing zone is higher if the contaminant source location is in the supply air as compared to contaminant source location in the room

**Keywords:** Computational Fluid Dynamics, Particle Size, Source Location, Air Distribution, Ventilation Effectiveness.

## INTRODUCTION

In this present era, more time (90% of their lifetime) is spent by people indoors [1]. Artificial climates are preferred by most of the people whether it is at work, home or in transport vehicles [2][3][4]. Clean room air is very much essential for good health. Polluted air is the one containing some harmful substances in high concentrations. It becomes even more harmful when this polluted environment lasts longer. Human health is adversely affected by various particle pollutants [5]. Indoor air pollution is the second highest killer in India [6]. According to OCED, air pollution costed India US$ 0.5 trillion [6]. Therefore maintaining good air quality indoors is very essential.

Generally, inferior quality of indoor air is attributed to prevalent contaminant sources in indoor environment. Small amount of airborne contaminants affect health of the people, comfort levels and performance at the workplace [2][3][7]. ISHRAE holds the view that most common indoor air quality problems are caused by inadequate ventilation and filtration, and improper air distribution [6].

For providing good indoor air quality and comfortable thermal conditions, HVAC systems preferably use Mechanical or forced ventilation. In this system air is circulated within the building and also, there is effective air exchange with outside air. Mechanical ventilation in combination with other HVAC system elements has many advantages over natural ventilation such as, better humidity and temperature control, optimal control over ventilation rate and air quality in extreme climatic conditions. More energy consumption is a major disadvantage. However, mechanical ventilation is particularly suitable for enclosed spaces, underground areas because of less practicality of natural ventilation.

In HVAC systems with mechanical ventilation, inlets and exhaust outlets are installed on walls, floor, or ceilings of indoor enclosed spaces. Mechanical ventilation systems are classified further based on the locations of inlet and exhaust/outlet as displacement, mixing, under floor air distribution and stratum ventilation systems [9].

In displacement ventilation, inlets are located on the side wall, near the floor, and exhaust outlets are installed near ceiling. In this system, the inlet air diffuses in the room, rises in the upper portion of the room from where it is exhausted. Displacement ventilation provides better indoor air quality. Few disadvantages are, limitations on layout, requirements of larger air flows and larger temperature gradients [10].

In commercial buildings and industrial facilities, displacement ventilation systems are less common compared to mixing ventilation systems. Displacement ventilation systems were initially applied in some industrial spaces in Scandinavian countries in late seventies, and since then it became popular in Scandinavian countries [11]. By 1989, within Nordic countries, more than 50% of industrial establishments and 25% of offices used displacement ventilation [12]. Some research studies were carried out to evaluate practicality of displacement ventilation systems for humid and hot climatic conditions [13].

In the present work, experimental test room was developed with proper air supply, contaminant supply, velocity, temperature and concentration measurement facilities. Measurement uncertainties were considered while interpreting the results. Numerical simulation studies were carried out considering all the practical details of the test room. Both experimental and numerical results were plotted graphically and the results were interpreted. The results were compared with similar studies carried out earlier [18][19] and after validation and agreement conclusions were drawn.

## METHODS

### 1.1 EXPERIMENTAL SET-UP FOR DISPLACEMENT VENTILATION

For displacement ventilation configuration the supply air should be provided from the lower side wall near floor and the exhaust should be on the upper side of the opposite side wall near ceiling. This arrangement was simulated in the experimental test room by providing air supply diffuser and exhaust diffuser as shown in the figure 2.1.

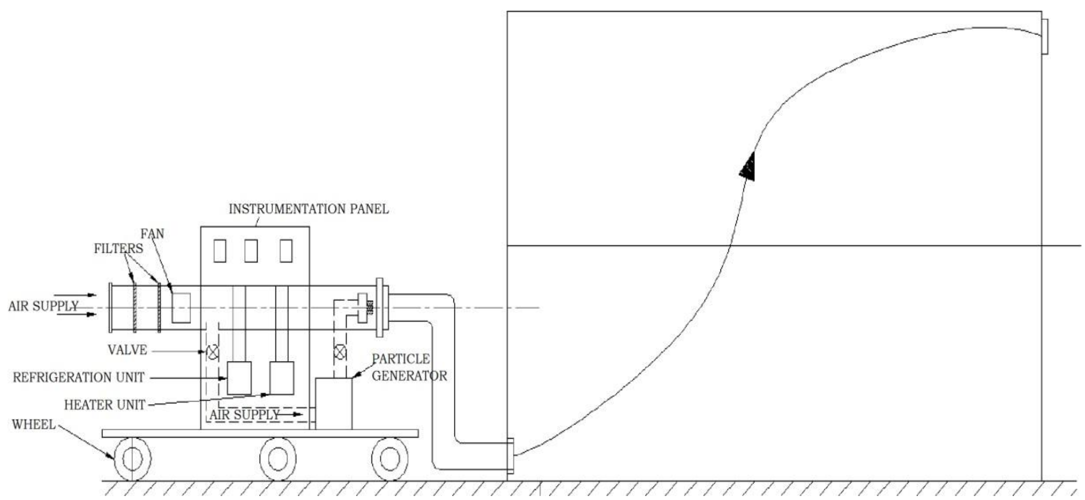


**Figure 2.1 Experimental Set-up for Displacement Ventilated Room**

In displacement ventilated room the supply air was provided from the lowest portion of side wall (0.2m from floor). The exhaust outlet was provided at a location (0.2m below ceiling) on the opposite side wall.

The test room dimensions were 4.4 m (L) x 4.0 m (W) x 3.2 m (H). The inlet and outlet openings were both of size 0.2 m x 0.2 m, symmetrical with center plane at Y =2.0 m (Figure 2.2). In this investigation, measurements of airflow, temperature and concentration of particles were carried out in Experimental Test Room at measurement locations described earlier.

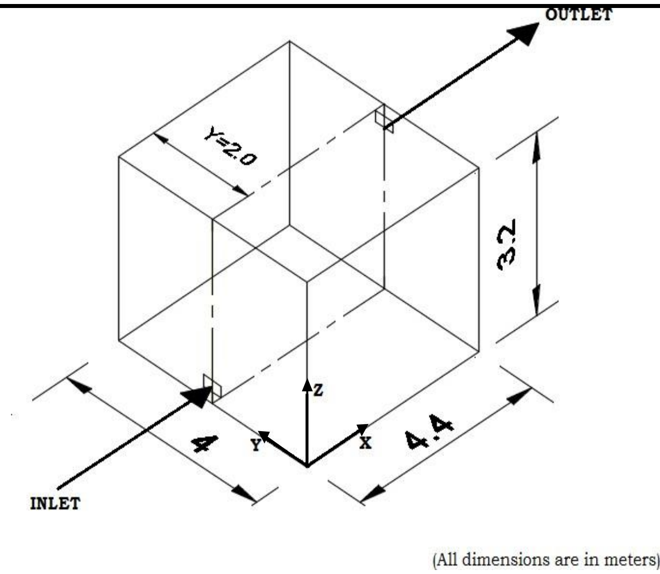


**Figure 2.2 Geometry of model displacement ventilated room**

The density of the particles was 1150 kg/m$^3$. The inlet contaminant particle mass concentration was maintained at 620 µg/m$^3$. Air velocity, temperature and particle concentration measurements at various measurement points in the Experimental Test Room were carried out using appropriate instruments.

### 1.2 NUMERICAL SIMULATION MODELS

#### 1.2.1 RNG K-Є MODEL FOR AIR FLOW SIMULATION

The RNG k-ϵ model was derived using a rigorous statistical technique called re- normalization group theory. It is similar in form to the standard k-ϵ model, but includes few refinements which improves accuracy significantly [14]. More detailed discussion of RNG theory with turbulence applications can be found in Fluent manual [14].

#### 1.2.2 MIXTURE MULTIPHASE MODEL FOR CONTAMINANT SIMULATION

For modeling of contaminant particles, the mixture model with single fluid approach is used. This mixture model allows interpretation of phases i.e., air plus particles. Using the concept of slip velocity, which is often called as drift flux velocity, this model permits the movement of phases at different velocities. Drift – flux model is advantageous as particles with different sizes can easily be incorporated in this model. For indoor environment studies, drift flux model results do not differ much from the conventional mixture model [15]. For dispersion of particles, the drift-flux model is given by equation 2.1 as-

$$\nabla.\left(\rho\left(\vec{u}_a+\vec{v}_s\right)c\right)=\nabla.\left(\frac{\mu_{eff}}{\rho_c}\nabla c\right)+s_c \qquad (2.1)$$

Where, $v_s$ = Particle settling velocity, c = Concentration of particles, $\rho_c$ = Turbulent diffusivity, $S_c$ = Generating rate of particle source, $\mu_{eff}$ = Sum of the molecular ($\mu_m$) and turbulent dynamic viscosity ($\mu_t$).

Major assumptions that are used in drift flux model are :

(1)The particle effect on turbulence of air flow is not considered because of small particle loadings and lower particle settling velocities [15][16].

(2)The coagulation effects are neglected in this room ventilation studies as the size distribution of particles is not affected and changed by coagulation. Also, 200 days are required for halving the number of particles because of coagulation [17].

(3)The body force is neglected, as for the given body sizes the difference in densities of particle and fluid is very small.

For this model, particle setting velocity is derived by equating the gravity force to drag force exerted by fluid on the particle and can be given by equation 2.2.

$$|v_s| = \left[\frac{4}{3}\frac{g.d_p}{f_{drag}}\frac{(\rho_p-\rho_a)}{\rho_a}\right]^{\frac{1}{2}} \qquad (2.2)$$

Where,

$\rho_p$ = Density of particle in kg/m$^3$

$\rho_a$ = Density of air in kg/m$^3$

$g$ = Gravitational acceleration in m/s$^2$

$d_p$ = Particle aerodynamic diameter, m.

$f_{drag}$ = Drag force of fluid on the particle in N.

$V_s$ = Settling velocity in m/s.

Grid independence checks and validation studies were carried out. For solving time averaged Navier-Stokes equations, a finite volume approach using SIMPLE algorithm was used. Momentum discretization was done using QUICK, second order finite volume scheme for non- uniform grid. The second order upwind scheme was used for volume fractions of particles. For volume fractions of particles second order upwind scheme was used. The governing equation for turbulent kinetic energy and its dissipation rate were discretized by first order upwind scheme. Convergence was considered achieved for the residual value of the order of 1 X $10^{-5}$.

## 3. RESULTS AND DISCUSSIONS

### 2.1 PARTICLE SOURCE AT S1

For particle source in supply airflow (S1), the contaminant removal effectiveness in occupied zone ($\varepsilon_{oz}$) is 1.130 and the contaminant removal effectiveness in breathing zone ($\varepsilon_{bz}$) is 1.182. This indicates that for particle source in supply airflow (S1), at ACH of 3.6, for 10.0 μm particles, contaminant removal effectiveness in breathing zone ($\varepsilon_{bz}$) is higher than the contaminant removal effectiveness in occupied zone ($\varepsilon_{oz}$ ).

### 2.2 PARTICLE SOURCE AT S2

For particle source in room (S2), the contaminant removal effectiveness in occupied zone ($\varepsilon_{oz}$) is 1.135 and the contaminant removal effectiveness in breathing zone ($\varepsilon_{bz}$) is 1.173. This indicates that for particle source in room (S2), at ACH of 3.6, for 10.0 μm particles, contaminant removal effectiveness in breathing zone($\varepsilon_{bz}$) is higher than the contaminant removal effectiveness in occupied zone($\varepsilon_{oz}$ ).

### 2.3 SUMMARY

As seen from the figure 3.1(a), 3.1(b), 3.1(c), 3.1(d), 3.2(a) and figure 3.2(b), it is observed that for particle source in supply airflow (S1), the average contaminant concentration in occupied zone, is higher than the average contaminant concentration for particle source in room (S2) and average contaminant concentration in breathing zone for particle source location (S1) is lower than for particle source location (S2). The average contaminant concentration in lower zone for particle source location (S1), is almost similar to the average contaminant concentration for particle source in room (S2) and the average contaminant concentration in upper zone for particle source location (S1), is higher than the average contaminant concentration for particle source in room (S2).

Also, it can be observed that the contaminant removal effectiveness in occupied zone ($\varepsilon_{oz}$) for particle source location (S2) is higher than for particle source location (S1)(1.135> 1.130) and the contaminant removal effectiveness in breathing zone ($\varepsilon_{bz}$) for particle source location (S1) is higher than particle source location (S2) (1.182>1.173).

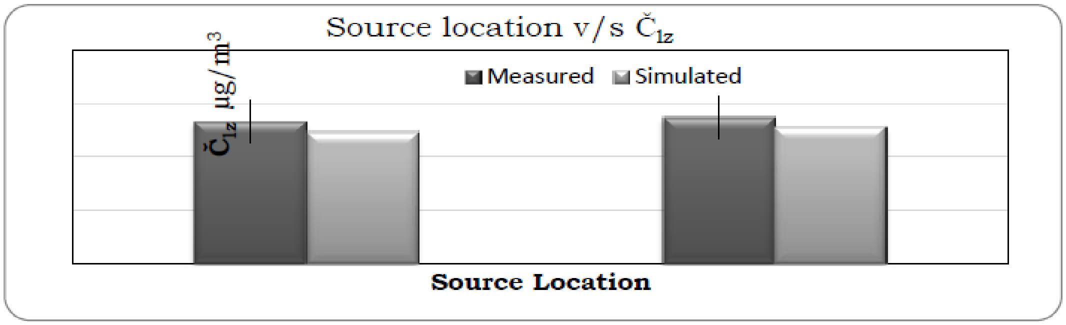


**Figure 3.1(a) Effect of particle source location on $\check{C}_{lz}$**

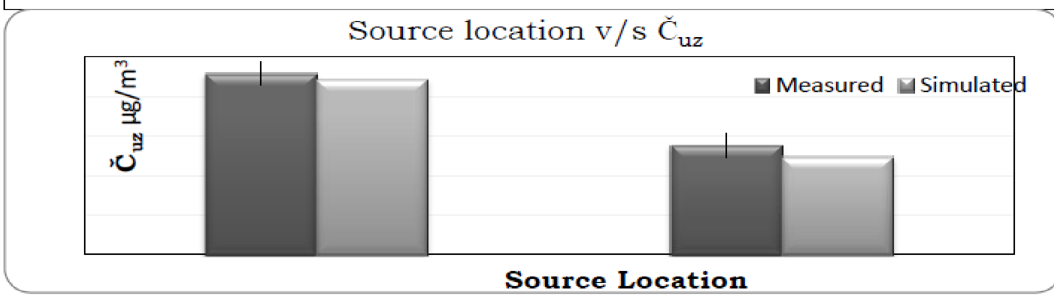


**Figure 3.1(b) Effect of particle source location on $\check{C}_{uz}$**

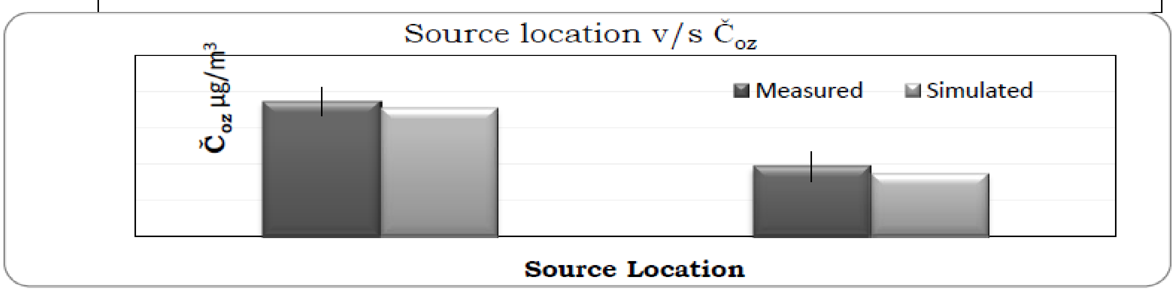


**Figure 3.1(c) Effect of particle source location on $\check{C}_{oz}$**

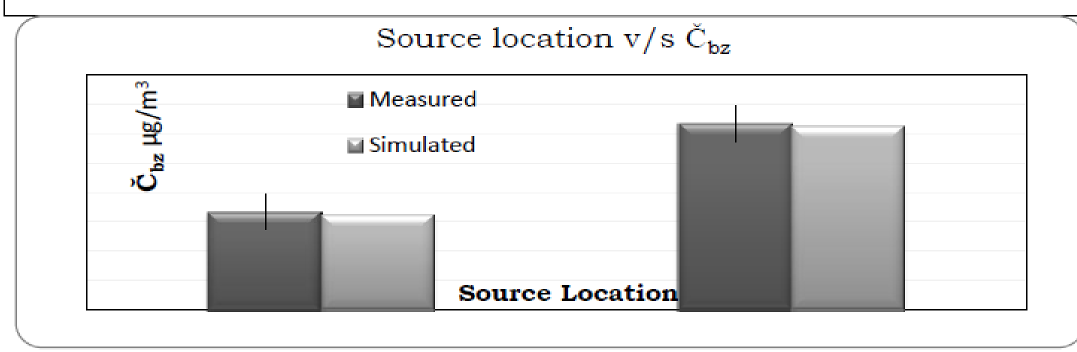


**Figure 3.1(d) Effect of particle source location on $\check{C}_{bz}$**

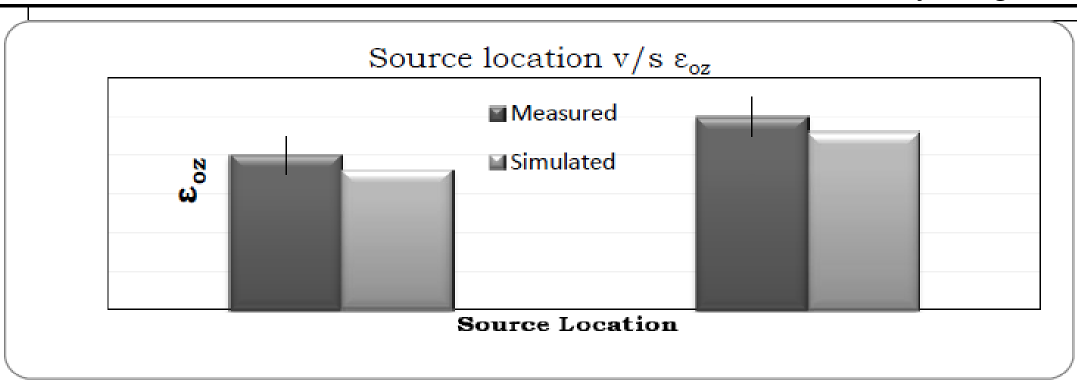


**Figure 3.2(a) Effect of particle source location on $\varepsilon_{oz}$**

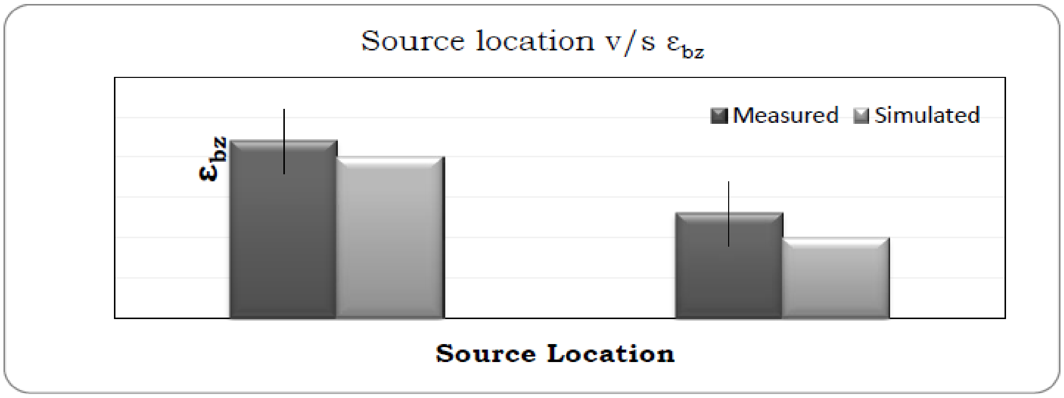


**Figure 3.2(b) Effect of particle source location on $\varepsilon_{bz}$**

## 3 CONCLUSIONS

a. The average contaminant concentration in occupied zone is higher for the contaminant source location in the supply air as compared to contaminant source location in the room.
b. The average contaminant concentration in breathing zone is higher for the contaminant source location in the room as compared to contaminant source location in the supply air.
c. The average contaminant concentration in lower zone is marginally lower for the contaminant source location in the room as compared to contaminant source location in the supply air.
d. The average contaminant concentration in upper zone is higher for the contaminant source location in the room as compared to contaminant source location in the supply air.
e. The contaminant removal effectiveness in occupied zone is higher if the contaminant source location is in the room as against the source location in the supply air.
f. The contaminant removal effectiveness in breathing zone is higher if the contaminant source location is in the supply air as compared to contaminant source location in the room.
g. For 10 µm contaminant particles at lower ACH the displacement ventilation system provides highest contaminant removal effectiveness in the breathing zone if the contaminant source is in the supply air.

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