Active supply diffuser application in all-air heating systems

MEHRDAD RABANI
PhD research fellow at Oslo Metropolitan University
email: Mehrdad.Rabani@oslomet.no

HABTAMU BAYERA MADESSA
Associate professor at Oslo Metropolitan University
email: Bayera.Madessa@oslomet.no

NATASA NORD
Associate professor at Norwegian University of Science and Technology
email: natasa.nord@ntnu.no

Substantial reductions in energy use can be achieved by promoting low energy building technologies such as all-air systems. There is an idea that integrating active supply diffuser with all-air systems can potentially improve their performance. This paper numerically investigated the performance of an all-air heating system equipped with an active supply diffuser in a cell office, constructed based on Norwegian passive house, in terms of indoor air quality and thermal comfort of occupants. Simulations were performed using commercial Star-CCM+ software. The numerical results were validated using the available experimental data on the active supply diffuser from an office. The results showed that adopting active supply diffuser can avoid temperature stratification, which is the main reported issue in heating application of conventional all-air systems, and provide the thermal comfort with PPD ≤ 7% in most part of occupancy zone.

Keywords: All-air heating; active supply diffuser; temperature stratification; Indoor air quality; Archimedes number

Nowadays, low energy building technologies have drawn many attentions due their potential in substantial reductions of building energy use. The low energy HVAC systems can be considered as a practicable solution when the building space heating demands, especially in cold climate countries are really low. The passive house (PH) concept is an example in this point that aims at promoting the energy efficiency of buildings by significant reduction of space heating needs using a strict insulation level of building envelope. Consequently, it is reasonable to simplify the space heating system by covering both ventilation and space heating needs using a so-called all-air heating (AAH) system. The all-air heating means to supply warm air, usually at ceiling level, at a hygienic air flow rate. However, the system performance, depending on supply air temperature and flow rate, thermal load, and outdoor conditions, might not be desirable due to the presence of vertical air temperature gradient (temperature stratification) and poor indoor air quality (IAQ) due to ineffective mixing of supply air flow with the convective plum of occupants and equipment in the zone of occupancy.

In this regard, the AAH system performance was investigated by several research studies. Fisk et al. [1]

conducted several experiments with a ceiling supply/exhaust configuration and the results supported a significant short-circuiting of ventilation air between the supply air diffuser and return air. The same phenomenon was also reported by Offermann and Int-Hout [2]. A significant stratification of contaminants in the lower part of the occupancy zone [3], poor air distribution and temperature stratification [4], a stationary region in the zone of occupancy [5], and non-uniform distribution of air velocity and temperature [6] were other reported issues associated with the application of AAH. Therefore, the AAH system performance needs to be improved in order to be considered as a functional solution for HVAC system in cold climate countries.

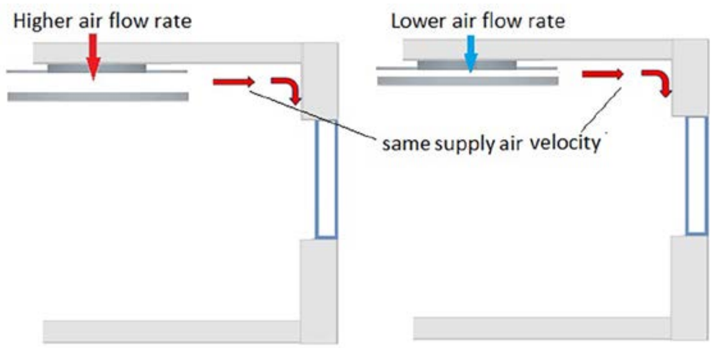


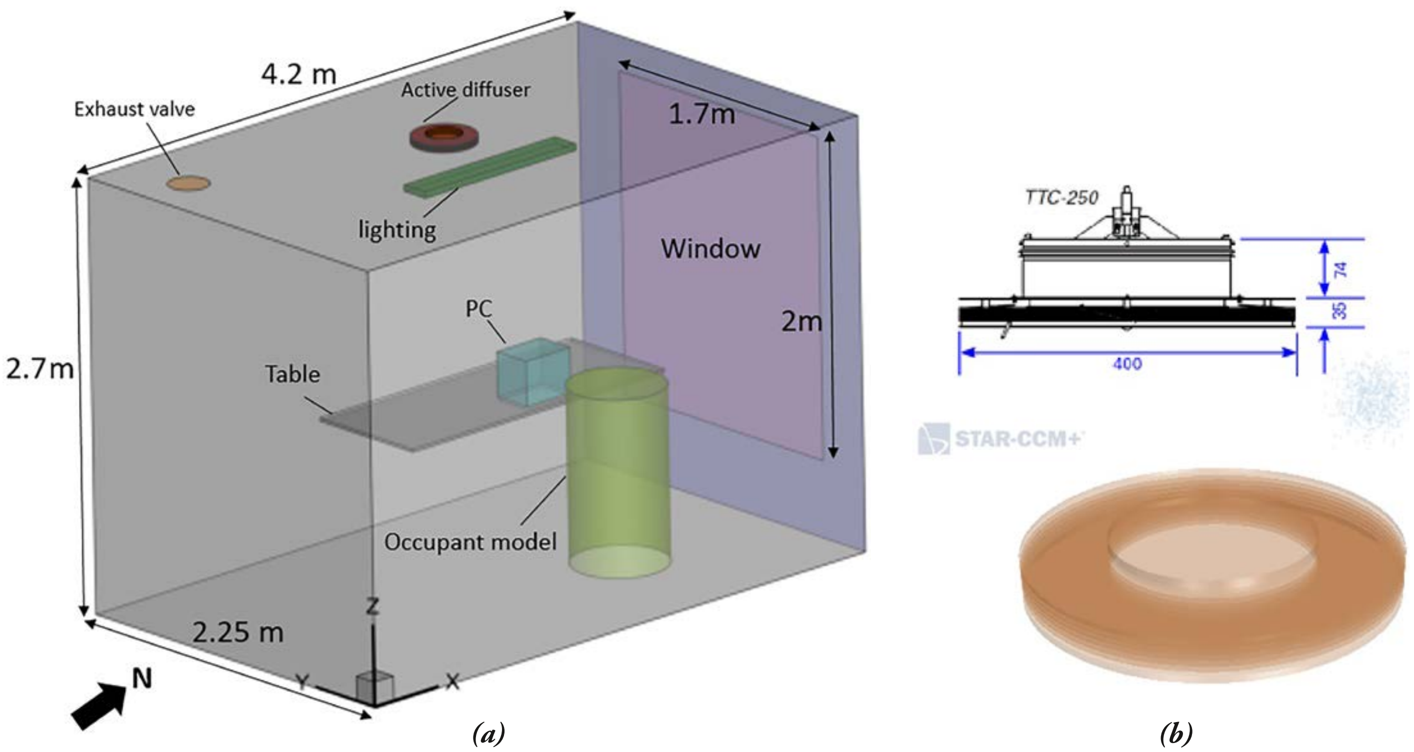
Figure 1. Conceptual schematic of active supply diffuser application in AAH system.

The aim of this study was to remedy the aforementioned issues in AAH systems using an active supply diffuser. The existing constant air volume (CAV) system could be converted to a variable air volume (VAV) system by installing active supply diffuser and isolating existing duct systems. **Figure 1** illustrates how the active supply diffuser functions. In typical VAV systems without active supply diffuser, the supply air velocity changes as the supply air flow rate changes due to constant supply area. This may increase the risk of draft in AAH systems, especially at low air flow rates. To solve this problem, the active supply diffuser will change the supply area proportional to the air flow rate so that a constant supply velocity is achieved for different air flow rates.

Methodology

Case study and boundary conditions

This study used Star-CCM+ software to analyze the performance of the AAH system equipped with the active supply diffuser in a cell office, constructed based on the Norwegian PH standard [7] and located in a Nordic climate. **Figure 2** shows the cell office configuration dimensions, and type of active supply diffuser. The active supply diffuser in this system was a TTC-250 active supply diffuser comprised of several moving plates [8]. For the boundary condition the inlet of the active supply diffuser was modelled using



(a) *(b)*

Figure 2. (a) Cell office configuration and dimensions and (b) TTC-250 active supply diffuser.

constant air velocity profile. Pressure outlet with zero gage pressure was considered for the exhaust. Lighting, PC, and occupant were modelled using constant heat source boundary condition 136W, 120W, and 30W, respectively. The external wall, internal wall, window, floor, and ceiling were modelled using heat transfer boundary conditions calculated from the energy balance with overall heat transfer coefficients taken from the experimental study [9].

Performance index parameters

In order to evaluate the performance of AAH system with active supply diffuser the following parameters and index were defined:

■ Archimedes number (*Ar*): describes the characteristics of supply jet and was defined according to the Eq. 1

$$Ar = \frac{g\beta a_0^{\frac{5}{2}}(T_e - T_s)}{Q_s^2} \tag{1}$$

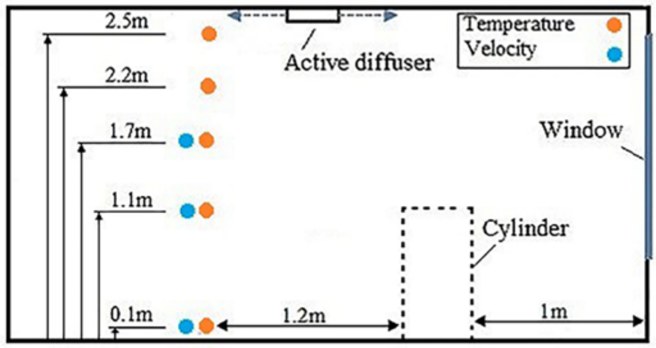
for mixing ventilation system.

where T_s was the supply air temperature, T_e was the exhaust air temperature, u_0 was the supply air velocity, and β was the coefficient of thermal expansion, a_0 was the net opening area of the supply, and Q_s was the ventilation air flow rate.

■ Predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) – show the thermal comfort of occupants. PMV examines the occupant thermal sensation according to seven scale points (from –3, cold, to 3, hot) with regard to six factors the air temperature, the mean radiant temperature, the air velocity, the air humidity, physical activity and clothing isolation level. PPD gives the thermal dissatisfaction predicted by PMV quantitatively. In order to have a favorable environment for a building with low energy use, the PPD should be less than 10% associated with $-0.5 < PMV < 0.5$ [10].

Numerical method validation

The numerical method was validated using the available experimental data. The details of experimental setup can be found in [11]. **Figure 3** indicates the location of experimental sensors and the comparison of air temperature and velocity obtained from simulations with the experimental data. It is observed that the results of simulations were in the uncertainty range of experimental data.



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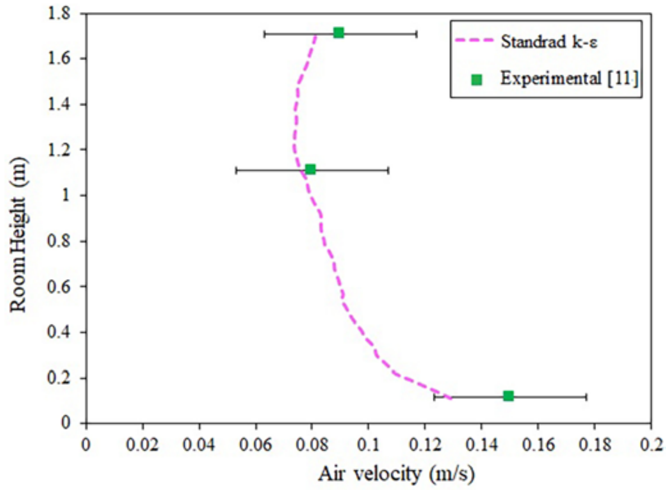

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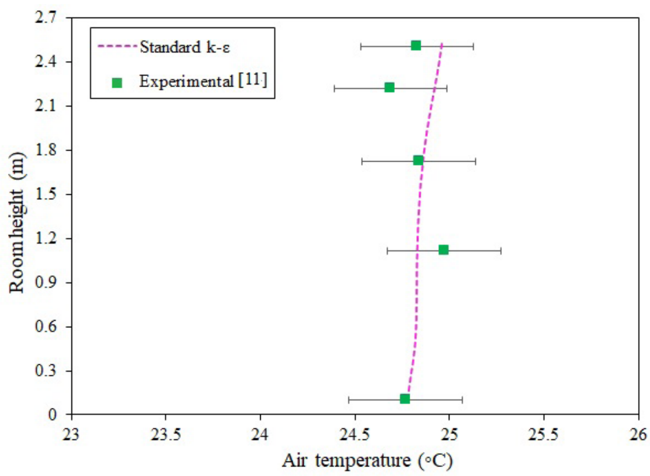


(c)

Figure 3. (a) location of experimental sensors, (b) air velocity and (c) air temperature variations at the measurement points.

Results and discussion

The performance of the system was analyzed for the most critical condition with the outdoor temperature −20 °C, which was the design outdoor temperature for Oslo, Norway, in the winter season [12]. The active supply diffuser was also operated under two different air flow rates: the minimum air flow rates required by the Norwegian PH standards and the maximum air flow rates for which the active diffuser could operate. **Table 1** describes the details of two scenarios. The negative Archimedes number shows that a negatively buoyant air was supplied.

Table 1. *The main parameters for the scenarios investigated in this study.*

Scenario	T_{out} (°C)	T_s (°C)	$\dot{V}_s$ (l/s)	*Ar*
S1	−20	24.8	49.4	-1.56×10^{-3}
S2	−20	26.4	16	-1.73×10^{-3}

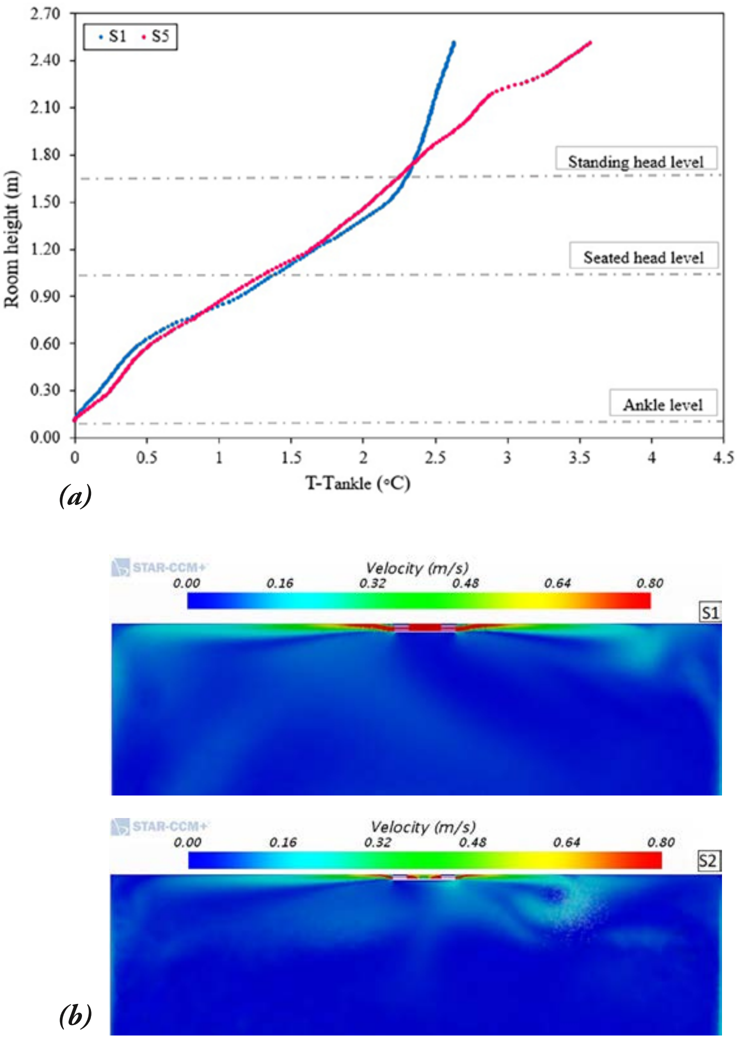


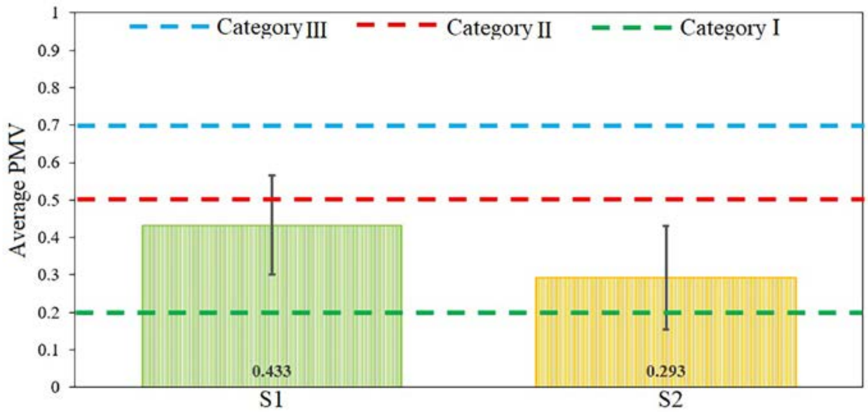
Figure 4. (a) Temperature stratification and (b) air velocity distribution for two scenarios in a plane passing through the window

Figure 4 illustrates the temperature stratification and air distribution for two cases. Maximum temperature stratification between seated head level and ankle level was around 1.5 K (**Figure 4a**), which was within the maximum recommended range by the EN ISO 7730 standard for the IAQ category II, indicating that active supply diffuser could avoid temperature stratification at both maximum and minimum air flow rates. This can be also seen in **Figure 4b** where the throw length towards internal wall (left side) was preserved by the active supply diffuser. However, the throw length towards window side (right side) was different due to interaction between cold current from the window and the supply jet.

Figure 5 illustrates the variation of thermal comfort indices for both scenarios. Using active supply diffuser could almost satisfy PMV requirement for both scenarios (**Figure 5a**) according to comfort category II [10]. The spatial distribution of PPD for both scenarios at three cross sections are shown in **Figure 5b**. The thermal comfort for both cases were always satisfied at the standard seated head level in the occupancy zone (1.1 m above the floor) with PPD ≤ 7% implying the decent performance of active supply diffuser in providing uniform distribution of supply air in the occupancy zone. However, some small region above that level located in the convective plume of occupant had higher PPD.

Conclusions

This study dealt with numerical simulation of IAQ in a standard cell office equipped with active supply diffuse and located in a Nordic climate country. Two different scenarios for maximum allowable and required minimum air flow rates were analyzed and the results showed that applying active diffuser at ceiling level could avoid temperature stratification, with maximum temperature stratification around 1.5 K, even at minimum air flow rates by changing the slot opening of diffuser. The thermal comfort analysis showed that both scenarios almost satisfied the average PMV requirements according to the comfort category II. Furthermore, spatial PPD values at the seated head level was less than 7% satisfying the requirements for the comfort category II. It is worth mentioning that the cooling performance of active supply diffuser would be interesting to be evaluated as the all-air system should also cover the building cooling load. ■



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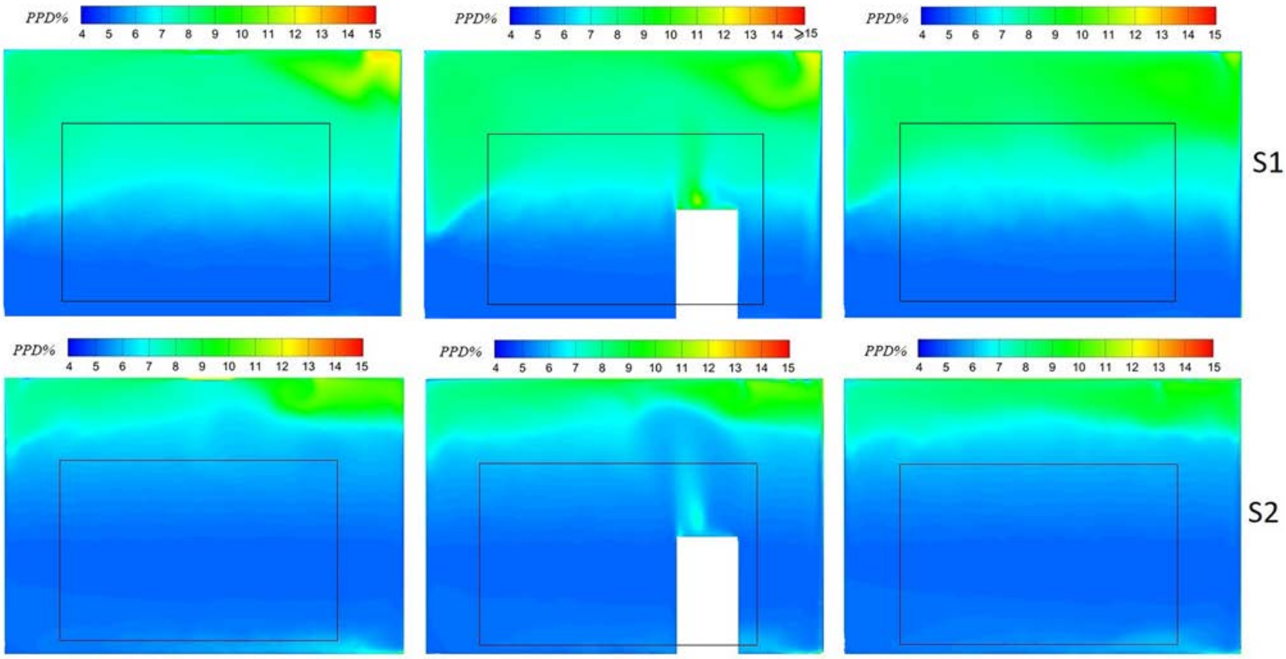

(b) Y = 1.4 m Y = 1.0 m Y = 0.6 m

Figure 5. (a) Average PMV in the zone of occupancy and (b) PPD distribution in three cross sections

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