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Experimental and numerical investigation of micro-environmental conditions in public transportation buses

Shengwei Zhu\textsuperscript{a,b,*}, Philip Demokritou\textsuperscript{a}, John Spengler\textsuperscript{a}

\textsuperscript{a}Department of Environment Health, School of Public Health, Harvard University, Boston, MA, USA
\textsuperscript{b}Harvard University Center for the Environment, Cambridge, MA, USA

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\section*{ABSTRACT}
This study examines both numerically and experimentally the micro-environmental conditions in public transportation buses. A Computational Fluid Dynamics (CFD) model was developed and experimentally validated. The developed CFD model was used to calculate the spatial distributions of the mean age and mean residual lifetime of air in the bus environment and evaluate the efficiency of the bus ventilation system. Additionally, the passengers’ exposures to a variety of environmental conditions were experimentally monitored in “real world” field campaigns using the Harvard University shuttle bus system. Real time continuous monitoring systems were used to assess indoor environmental quality in the buses. It was found that CO levels were very low, while the levels of particulate matter varied and were influenced by the ambient air penetrated into the bus through the operation of the doors and the ventilation system. The CO\textsubscript{2} level was found elevated and greatly affected by occupancy conditions. The elevated CO\textsubscript{2} level indicates that the current bus ventilation is insufficient to dilute air pollutants in the bus especially under heavy occupancy conditions. This lack of sufficient ventilation indicates an elevated risk for airborne transmitted diseases in such a popular public transportation system.

\section*{1. Introduction}
Commuting occupies around 7\% of the daily time of people who live in urban areas \cite{1} with the Americans spending approximately 5.5\% of their typical day inside a vehicle \cite{2}. And the bus transportation system is considered to be one of the most available and popular transportation modes around the world \cite{3}.

There are a number of published field studies focusing on air quality and thermal comfort in the bus microenvironment. In a study performed in Munich, Germany showed that the concentrations of PM\textsubscript{10} in buses and trams depended on the ambient sources and the road traffic conditions \cite{4}. In a similar study performed in another urban setting, Guangzhou, China, the parameters such as poor vehicle emission controls, poor vehicle maintenance and high-density traffic conditions were linked to poor indoor air quality in buses. Furthermore, it was found that the concentration levels of carbon monoxide (CO) and PM\textsubscript{10} in buses could be lowered substantially by using a HVAC system \cite{5}. This is due to the “protective” effect of mechanically ventilated and air-conditioned buses which results to a lower indoor penetration factor for a variety of air pollutants (CO, PM\textsubscript{2.5}, PM\textsubscript{10}, etc) \cite{6}. Chan \cite{3} also reported that CO\textsubscript{2} concentration could be up to 10 times higher than outdoor concentrations under overcrowded conditions, which is usually the case during rush hours. However, there are a limited number of studies focusing on the effect of ventilation strategies on diluting or removing air pollutants generated inside the buses.

In addition, in terms of health effects linked with the unique microenvironment in the buses, nausea, dizziness and respiratory allergies have been identified in previous studies \cite{3,7}. It also became apparent in recent years, that airborne transmitted diseases, such as TB, Severe Acute Respiratory Syndrome (SARS), Avian Influenza and Swine Influenza (H1N1), may impose a serious global health and economic burden. Therefore, there is a need to further study and understand the transmission of airborne infectious diseases in the indoor environment, and the bus environment might be one of the most challenging ones because of the very close proximity of the people.

In recent years, with the rapid development of computer technology and advanced numerical methods, investigators were also able to successfully use Computational Fluid Dynamics (CFD) models to study the microenvironments in a various of public transportation systems, including aircraft \cite{8-10}, trucks \cite{11} and
cars [12,13]. Those advanced numerical methods have been proved an effective tool to obtain the detailed spatial and temporal distributions of temperature, velocity and contaminants in micro-environmental settings, which are usually very difficult and expensive to be obtained in experimental measurements.

The thrust of this investigation is two-fold: (a) develop and use a CFD model to understand the bus micro-environmental conditions, especially the ventilation effectiveness and air distribution system, and how they may affect the dilution and removal of any internally generated air pollutant; (b) perform an extensive field campaign using the Harvard University shuttle bus system and monitor continuously the indoor environmental quality as well as the personal exposure levels of passengers for a variety of air pollutants including CO, CO₂, and particulate matter in its various respirable size fractions such as PM₂.₅, PM₁₀ and ultrafine particles (UFPs).

2. Numerical methods

2.1. Bus geometrical characteristics used in CFD model

Fig. 1 shows the geometrical characteristics of the bus used in the CFD model. The bus used in the simulations was a Daimler SLF 200 from the Harvard University Shuttle Service, with a total volume of 33.49 m³. The cabin space was divided into three compartments as shown in Fig. 1(a): the driver’s space, the front compartment and the rear compartment.

This bus uses the widely used mixing ventilation method. Six linear air supply diffusers are located near the ceiling. On each side of the bus, there is a two-slot diffuser supplying air downwards and a one-slot diffuser supplying air horizontally (Fig. 1). The total area of the air supply openings is 0.21 m². The exhaust opening with an area of 0.544 m² (0.34 m × 1.6 m) is placed in the ceiling close to the rear door.

2.2. Model assumptions

In the CFD simulation, the bus was modeled with all of the doors and windows assumed to be closed. The air infiltration through leakage pathways was also omitted. Fig. 2 shows the geometry of the air diffusers. The total air supply rate was 0.54 m³/s, which accounts for an air exchange rate of 57.6 ACH assuming no air recirculation in the bus. In addition, solar and long wave radiation as well as occupancy was not considered in the developed model.

2.3. CFD method

A 3D CFD model based on the Reynolds-averaged Navier–Stokes (RANS) equations [14] was used to simulate the bus airflow, temperature and scalar contaminant fields. The CFD model is based on the conservation equations of mass, momentum, energy, scalar concentration, turbulent kinetic energy, and dissipation rate of turbulent kinetic energy, that govern the transport phenomena in the bus.

2.4. Boundary conditions

The aforementioned governing equations can be closed with appropriate boundary conditions at all of the boundaries such as air supply openings, air exhaust opening, and wall surfaces. The detailed boundary conditions are summarized in Table 1. For air supply openings, the actual values of air temperature, velocity and turbulence intensity were measured in the field experiments, and used as boundary conditions (see following sections). The turbulence scale was calculated as one-half of the width of the linear opening. Fig. 1(c) shows the location and direction of the supplied airflows in cross section B–B'. At the surfaces of floor, windows and lights, the temperatures were fixed based on the data obtained in the field experiment. The rest of wall surfaces were assumed adiabatic. Moreover, the standard wall function [15] was used for the near wall boundary layer.

2.5. Ventilation efficiency indices: mean age and mean residual lifetime of air

In this study, the spatial distributions of the mean age and mean residual lifetime of air [16] were used as the indices to estimate the ventilation efficiency in the bus. The mean age of air is defined as the mean time that it takes for the air from the supply openings to be transported at a specified location; the mean residual lifetime of air is defined as the mean time that it takes for the air at certain location to be exhausted. Numerically, these distributions can be obtained using the mathematical methods of SVE3 and SVE6 (Scale for Ventilation Efficiency 3 and 6, see Table 2) [17–19], under the assumption that the air pollutant (“tracer gas”) is generated uniformly and continuously in the bus.

Based on the so-called SVE3 method, the mean age of air can be calculated by solving the following transport equation [20],

\[
\frac{DC}{Dt} = \frac{\partial C}{\partial t} + \frac{\partial (u_j C)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \nu \frac{\partial C}{S_c \partial x_j} \right) + q_0
\]  

(1)

where: C: concentration [kg/kg]; u_j: velocity [m/s]; \( \nu \): turbulent kinematic viscosity [m²/s]; Sc: turbulent Schmidt number [–]; q_0: generation term of C and is uniform at any point indoors [kg/s].

If q_0 is assumed to be generated uniformly and continuously throughout the bus, the air mass from a supply opening is gradually contaminated as it is regarded proportional to the time elapsed from the time the air mass leaves the supply opening until it reaches the point. The total amount of tracer gas generated in the room divided by the airflow rate of ventilation is termed a “perfectly mixed concentration C₀ (kg/kg)”. Under a steady state condition, this is equivalent to the concentration averaged over the exhaust opening. Because the age of air averaged over the exhaust opening is equivalent to the inverse of the air exchange rate, the distribution of trace gas C in the room can be converted to the distribution of the mean age of air.

Similarly based on the so-called SVE6 method, the mean residual lifetime of air is computed by applying the same computational procedure as SVE3, except that the time sequence of the flow field is reversed.

2.6. Numerical procedure

The CFD numerical grid system used in the model consists of 126,838 triangular surface meshes and 1,062,098 tetrahedral fluid cells, which were created using Gridgen V15.10 at a growth rate of 1.1. A good grid was ensured with over 99.9% of the meshes of less than 0.8 in EquiAngle Skewness, which represented the maximum ratio of the cell’s inner angle to the angle of an equilateral element.
The commercial CFD software, Star-CD V4.08, was used to solve the governing equations together with the standard $k-e$ model and implicit SIMPLE algorithm [21]. The finite volume method with the first-order upwind scheme (UD) was adopted for discretizing the governing equations [22].

3. Experimental methods

3.1. Field experiments for CFD model ventilation (steady state conditions)

In order to validate the developed CFD method, a field experiment was also conducted under stationary conditions using a bus (Model: Daimler SLF 200, Maker: DaimlerChrysler Commercial Buses; Engine type: Mercedes Benz MBE904; Year: 2005) from the Harvard University shuttle bus fleet. Boundary conditions and other data for validating the developed CFD method were collected during this experiment. The bus engine was kept in idle condition and there were no people inside. During the measurements, all of the windows and doors were fully closed, and the air-conditioning system operated as usual, with the cabin thermostat set to 20.0°C. Before the measurement began, the bus engine had been left running for about 2 h to have the bus environment reach the steady state conditions.

The surface temperatures of the floor, light and windows (front windshield, side windows and windows fixed in the doors) were
measured and used as boundary conditions in the CFD model. Furthermore, air temperatures at the heights of 0.1 m, 0.6 m, 1.1 m and 1.7 m from the floor in the midlines of the cross sections B–B’, C–C’ and D–D’ were measured for validating the CFD model. These measurements were repeated at an interval of 30 s using thermocouples (HOB0, see Table 3). The averages of the total 140 data points (for 70 min) were recorded.

The velocities and temperatures of the supply airflow at the linear diffusers were also measured using a hot-wire anemometer (TSI 964, see Table 3) at an interval of 1 s for the duration of 3 min in order to obtain the boundary conditions for the air supply openings. The measurements were performed at four bus locations (in the front and rear compartment at each side of the bus). The average of the 180 data points was used as the value for that location. In addition, turbulence intensity was calculated automatically as the percent of the standard deviation of the turbulent velocity fluctuations to the mean velocity over the measuring period at each location [23]. The averages of those results were also used as the boundary conditions for air supply opening.

It should be addressed that while the average values for temperature and velocity were used at the walls and supply inlets, in real life these parameters vary as a function of time based on weather and other operating conditions.

### 3.2. Experimental investigation of IEQ in buses

The Harvard University shuttle bus system was used for our field investigation study. The Harvard University Shuttle Service is the main transport mode for the students, staff and faculty commuting throughout the Cambridge and Allston campuses of Harvard University. It is estimated that over 25,000 passengers per week use the Harvard University Shuttle Service. Two different routes were selected. Both routes have similar traffic conditions of primary and secondary streets. A typical route consists of four stops and takes approximately 15 min. At each stop, it usually takes approximately 30 s for boarding/unboarding passengers. The age profile of passengers is also unique since 85% of the passengers are under 25 years old on our survey investigation.

Four full day field campaigns were performed in the fall of 2008 (November 17th, 19th, 25th and 26th). All buses were Daimler SLF 200, same as that used in the field experiments for CFD model validation. The buses were air-conditioned, and all of their windows were fully closed when in operation. Outdoor air infiltrated through the doors and the ventilation system. The field campaigns started early in the morning before 9:00 AM, with a lunch break around noon and continued until 16:30 PM in the afternoon. It is worth pointing out that typical traffic patterns in the Cambridge campus area were reflected in our field campaigns. The drivers were also asked to operate as usual without intervening in the temperature setup and the ventilation settings in the bus.

Continuous instruments were used to monitor indoor environmental conditions in the bus. Air temperature, relative humidity and the concentration levels of CO₂, CO and particulate matters in three size fractions, PM₁₀ and PM₂.₅, and ultrafine particles (UFPs) were continuously measured in the bus. Real time measurements for the above IEQ parameters were recorded at an interval of 10 s. The instruments were placed in two mesh boxes made by coarse wire, which were hanged at the shoulders of two of our field personnel at a height between 0.6 m and 1.1 m from the floor. The field personnel remained standing in the middle of the front compartment for the whole trip as shown in Fig. 1(a).

Moreover, since there was only one particle counter used in the

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**Table 1**

| Boundary conditions used in CFD model. |
|---------------------------------------|
| **Air supply opening**                |
| Velocity: 2.54 m/s, Airflow rate: 0.54 m³/s, |
| Turbulence intensity: 2.5%, Turbulence Scale: 0.005 m, |
| Temperature: 20.2 °C                   |
| **Air exhaust opening**               |
| Free-slip                             |
| **Lights**                            |
| No-slip, Temperature: 25.0 °C         |
| **Floor**                             |
| No-slip, Temperature: 15.5 °C (in front compartment) |
| 19.8 °C (in rear compartment)         |
| **Windows**                           |
| Front windshield                      |
| In front door                         |
| In middle door                        |
| On the side of doors                  |
| **Other walls**                       |
| On the side of driver                 |
| No-slip, adiabatic conditions         |

**Table 2**

| Scales for ventilation efficiency 3 and 6. |
|--------------------------------------------|
| SVE(X) – C(X)/Cs (1)                      |
| SVE6(X) – C₆(X)/Cs (2)                    |
| Cₙ = q/Q (3)                              |
| Cₙ(X) Age of air at position X            |
| Cₙ(X) Residual lifetime at position X     |
| Cₙ(X) Concentration at position X where contaminant is uniformly generated throughout a room in total generation rate q [Kg/m³] |
| Cₙ(X) Virtual concentration at position X where contaminant is uniformly generated throughout a room in total generation rate q [Kg/m³] and time passes inversely |
| q Total generation rate of contaminant in exhaust opening [Kg/m³] |
| Q Total airflow rate [m³/s]              |

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**Fig. 2.** Real shape and CFD model of linear air supply diffuser.
experiments for the measurement of PM$_{2.5}$ and PM$_{10}$. PM$_{2.5}$ concentrations were measured on Nov. 17th and 25th, while PM$_{10}$ concentrations were measured on Nov. 19th and 26th. Table 3 summarizes the instruments used in the continuous measurements of the aforementioned IEQ parameters.

### Table 3

| Sensor             | Equipment                  | Range            | Accuracy                        | Resolution   |
|--------------------|----------------------------|------------------|---------------------------------|--------------|
| Velocity           | TSI 964 with Q-Trak Model 7565 | 0–50 m/s         | ±3% of the reading              | 0.01 m/s     |
| Temperature        | TSI 964 with Q-Trak Model 7565 | -10–60 °C        | ±0.3 °C                         | 0.1 °C       |
| Relative humidity  | HOBO                       | -20–70 °C        | ±0.4 °C from 0 to 50 °C         | 0.03 °C at 25 °C |
| Carbon monoxide    | TSI 982 with Q-Trak Model 7565 | 0–500 ppm        | ±2.5% from 10% to 90%          | 0.03%        |
| Carbon dioxide     | GE Telaire 7001            | 0–4000 ppm       | ±3% of reading                  | 0.1 ppm      |
| PM$_{2.5}$ & PM$_{10}$ | SidePak AMS10         | 1–20,000 μg/m$^3$ | ±50 ppm or ±5% of reading      | 1 μg/m$^3$   |
| Ultrafine particles| TSI Model 3007             | 0–1×10$^5$ pt/cm$^3$ | +0.5 μg/m$^3$ per °C          | 1 pt/cm$^3$  |

4. Results

4.1. Simulation results

4.1.1. Spatial distribution of air velocity

Fig. 3 shows the spatial distribution of air velocity in the bus. As shown in cross section A–A’, the air close to the front windshield was heated and moved upward, an indication of a “drafty” condition in the proximity of the driver. Moreover, an eddy was generated in the mid upper region as shown in cross section B–B’. In cross section D–D’, a large eddy was shown at each side of the bus. These eddies are indicative of the highly turbulent nature of the flow in the bus environment. However, overall, the air velocities were mostly distributed under 0.3 m/s in the bus. This is almost satisfactory based on the ASHRAE Standard 55-2004 [24], which has a limit of 0.2 m/s for air velocity.

4.1.2. Spatial distribution of air temperature

Fig. 4 shows the spatial distribution of air temperature in the bus environment. Air temperatures were within 6°C–14°C of the set point (20.2 °C) in the passenger compartment with the exception of the area close to the windows and floor. As shown in cross sections B–B’ and C–C’, large air temperature gradients were present close to the floor, primarily because of the high thermal losses and the very poor air circulation under the seats. The same large temperature gradients were also shown close to the front windshield and the floor in the driver’s space.

Fig. 5 shows the comparison of the measured vs. simulated air temperatures in the bus. Each simulated air temperature was calculated as the average value of the air temperatures of the cells located around the position. The simulation results agreed reasonably well with the experimental data in cross section D–D’. At floor level in the driver’s space and the front compartment where there was a very low air circulation and Reynolds number,
there was not a great agreement between measured and simulated results. This is expected since it is a known limitation of the standard wall function used in the model for the wall surfaces covered by a low Reynolds number turbulent flow. This might be also attributed to a cold air infiltration close to the floor, which was not considered in the CFD model.

4.1.3. Ventilation efficiency: mean age and residual lifetime of air

Fig. 6 shows the spatial distribution of the mean age of air in the bus. For the driver’s breathing zone, the mean age of air was calculated to be generally in 62–86.8 s. In the front compartment, it was mainly in the range of 49.6–62 s. In the rear compartment, it was mostly less than 62 s (with the exception of the area involved in the developed eddy as shown in Fig. 3(d)). It is worth to point out that in general, fresh air can reach most of the area covering passengers’ breathing zone within 74.4 s.

Similarly, Fig. 7 shows the spatial distributions of the mean residual lifetime of air. It was found that the mean residual lifetime of air varied significantly from 0 to over 111.6 s in the bus. As expected, it was minimum in the area directly under the exhaust opening (less than 12.4 s), and increased very quickly with the distance from the air exhaust opening. In conclusion,
Fig. 6. Spatial distribution of mean age of air [s]. Breathing level is within the black lines in each picture. (a) In cross section A–A’. (b) In cross section B–B’. (c) In cross section C–C’. (d) In cross section D–D’.

Fig. 7. Spatial distribution of mean residual lifetime of air [s]. Breathing level is within the black lines in each picture. (a) In cross section A–A’. (b) In cross section B–B’. (c) In cross section C–C’. (d) In cross section D–D’.
the polluted air in passengers’ breathing zone will be exhausted within 2 min.

4.2. Experimental results

4.2.1. Thermal comfort conditions

Table 4 shows the weather conditions occurred during the experimental campaigns. Table 5 summarizes the measured air temperature and relative humidity in the buses for each day of the field campaigns.

Table 4

| Date    | Weather condition | Air temperature [°C] | Relative humidity [%] | Wind [m/s] |
|---------|-------------------|----------------------|-----------------------|------------|
| Nov. 17th | Sunny             | 5.6                  | 49                    | 4.9        |
| Nov. 19th | Sunny             | -2.2                 | 46                    | 6.3        |
| Nov. 25th | Rainy             | 7.2                  | 87                    | 7.6        |
| Nov. 26th | Sunny             | 3.9                  | 71                    | 3.8        |

On the contrary, as shown in Tables 4 and 5, relative humidity in the bus was affected greatly by the weather conditions with its daily mean value ranged from 49.8% when it was raining to 12.9% when it was sunny. The high relative humidity on the rainy day was caused by the water drops taken into the bus by the passengers or entering the bus when the doors were opened to load/unload the passengers. In addition, as indicated in Fig. 9, relative humidity was greatly affected by the opening of bus doors during loading/unloading of the passengers. Also, it was as expected that relative humidity depended on the occupancy load, primary because of the latent heat generation from the human occupancy.

The thermal comfort range for winter as it is recommended by ASHRAE standard 55-2004 [24] was presented in Fig. 10. This was made possible by plotting the experimental data points of the hourly mean air temperature and relative humidity measured in the bus throughout the field campaigns. It is necessary to point out that this is under the assumption that the measured air temperature in the bus is equal to the operative temperature and the clothing conditions of the passengers and drivers is 1 Clu as stated in the ASHRAE standard. Operative temperature was not measured in the field experiments and it can be higher than the air temperature if thermal radiation is considered. Therefore, Fig. 10 is considered just an approximation of the thermal comfort conditions in the bus. As shown in Fig. 10, most of the results were distributed at the “warmer side” of the thermal comfort zone but still in the comfort zone based on the ASHRAE standard 55-2004.

4.2.2. Air quality conditions

Table 6 summarizes the measured concentration levels of all the air pollutants monitored continuously in the bus (CO, CO2, PM2.5, PM10 and UFPs). Similarly, Fig. 11 also illustrates the CO2 concentration levels and the occupancy load as a function of time. Additionally, Figs. 12 and 15 show the concentration levels of CO, PM2.5, PM10 and UFPs as a function of time, respectively. The door open/close status was also indicated on those figures. In addition, also only the data for a representative campaign day were plotted in the figures since the results were similar on other days.

As it was shown in the aforementioned figures, the CO2 concentration level in the bus varied widely during the field campaigns. It usually varied from around 1000 ppm, and sometimes reached the levels of over 2000 ppm. It is worth to point out that based on the guidelines of the National Institute for Occupational Safety and Health (NIOSH) for indoor environments, CO2 concentration levels higher than 1000 ppm is an indication of inadequate ventilation [25]. Furthermore, Fig. 11 reveals that CO2 concentration levels in the bus were linked with the passenger occupancy. As it was expected, the CO2 concentration level increased with higher occupancy load. Moreover, the CO2 concentration level decreased sharply when the doors were open and

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![Fig. 8. Air temperature and occupancy condition in the bus as a function of time (Nov. 19th).](image-url)
increased soon after doors were closed in most cases due to the dilution effect from infiltrating air.

The CO concentration levels were below 1.9 ppm, much lower than the 8-hour average limit of 9 ppm, which was recommended by the US National Ambient Air Quality Standards (NAAQS). Additionally, as it is shown in Fig. 12, the CO concentration level generally reached its peak values when the doors were open to load and unload passengers. This is a clear indication of the influence of the traffic conditions in the street on the CO concentration level.

The mass concentrations of PM2.5 and PM10 were always below 220 μg/m³, with the daily mean values to be much lower than the recommendation by the NAAQS (35 μg/m³ for PM2.5 and 150 μg/m³ for PM10). As it is shown in Fig. 13, PM2.5 concentration generally did not exceed 20 μg/m³ except for the “heavy traffic hours” in the morning and afternoon (before 9:30 AM and after 14:00 PM). It is worth pointing out that PM2.5 concentrations during peak hours were found to be an order of magnitude higher. This is due to the high traffic conditions during peak hours and PM2.5 particles are well correlated with mobile sources (traffic conditions). Similar trend for the PM10 concentration level is shown in Fig. 14.

UFPs concentration level also varied widely over time, and reached its peak values when the doors were open, a clear indication of the influence of the traffic conditions. This is expected since UFPs are primarily emitted from combustion engines related to mobile sources such as cars, buses, etc. The daily mean UFPs concentration levels were found to be in the range of $4.0 \times 10^4$–$5.7 \times 10^5$ pt/cm³. This is in line with the findings from the similar studies in other cities in the US (9.5 $\times 10^3$–$5.3 \times 10^4$ pt/cm³ in Ann Arbor, MI, 3.0 $\times 10^4$–$7.5 \times 10^4$ pt/cm³ in Chicago, and 7.0 $\times 10^3$–$5.0 \times 10^4$ pt/cm³ in Atlanta [26]).

5. Discussion
5.1. Thermal comfort conditions

As it was shown in both the simulation results and the experimental results previously presented, there was overall thermal
satisfaction based on the ASHRAE 55-2004. However, the low relative humidity levels below 25% on sunny days documented in the field campaigns raise a health concern in terms of the airborne transmission of certain viruses. For example, according to the previous animals studies [27,28], the most favorable relative humidity levels for the airborne transmission of influenza virus was in the range of 17–35%. Our field campaign results imply that the low relative humidity found in this microenvironment may elevate the risk for influenza transmission.

5.2. Air quality conditions

As outlined in the previous result section, the concentration levels of CO, PM$_{2.5}$ and PM$_{10}$ were found in compliance with the existing standards for indoor air quality. Their concentration levels were always very small when the doors were closed and generally reached their peak values when the doors were open. Since there are no indoor sources of CO, the only reasonably explanation for this change is that the use of mechanical ventilation in the bus provided a “shielding effect” in terms of particulate matter and CO. This agrees with the results from the previous studies on air-conditioned buses [6]. However, the CO$_2$ concentration levels were found to be usually high and not in compliance with indoor environmental guidelines and standards. This is an indication of inadequate ventilation that is linked to an increased risk for airborne transmitted diseases. The simulation results also confirmed the lack of ventilation efficiency and found the mean residual lifetime of air in the bus for the breathing zone of passengers to exceed 112 s in some areas of the bus.
6. Conclusions

The developed CFD model can be used to predict the ventilation efficiency in the passenger’s breathing zone and evaluate alternate operational scenarios related to the ventilation system. The simulation results of mean age and mean residual lifetime of air indicate very good ventilation efficiency in an empty bus with 100% fresh air supply (no air recirculation), under the conditions different from those in the “real world” field campaigns on IEQ in the buses.

Based on both our numerical and field investigation on the bus environment, there is a need to further study and identify ventilation strategies and alternate air distribution methods in order to improve the ventilation efficiency in this important microenvironment. Additionally, due to the very close proximity of people in the bus environment, supplementary air filtration and air disinfection methods such as upper room UV irradiation might be necessary to be explored as a mean to minimize the risk of airborne infectious diseases. We plan to use the developed CFD model to parametrically analyze the impact of the operational conditions of ventilation system, such as air recirculation, air distribution method, etc.

The field experiments indicated that the thermal comfort conditions were found on the warmer side of the ASHRAE recommended comfort zone for winter conditions. It was also found that the ventilation system provided a “shielding effect” and protected the passengers from the harmful exposures to ambient air pollutants infiltrating indoors, such as CO and particulate matter. The high CO2 concentration levels found in the field campaigns is a good indicator of insufficient ventilation in the buses. These results reveal an increased risk for the airborne transmitted diseases. Therefore, alternate ventilation strategies and possible air filtration/purification systems may be needed in order to safeguard the health of people using this public transportation system.

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