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Numerical study on the dispersion of airborne contaminants from an isolation room in the case of door opening

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A negative pressure isolation room is built to accommodate and cure patients with highly infectious diseases. An absolutely airtight space effectively prevents infectious diseases from leaking out of the isolation room. Opening the door leads to a breakdown in isolation conditions and causes the dispersion of infectious air out of the isolation room. Extensively employed to manage smoke in cases of fires at subway and highway tunnels, a concept of controlling airflow is applied to the study. This study proposes a design of ventilation system to control air flow rate for containing airborne contaminant and preventing its spread to the adjacent rooms when the door to the isolation room is opened and closed. This paper employs computational fluid dynamics (CFD) as a more effective approach to examine the concentration maps of airborne contaminants and the airflow patterns of room air and discuss the influence of temperature differences between two rooms on airborne dispersion. Results show that an air velocity above 0.2 m/s via a doorway effectively prevents the spread of airborne contaminants out of the isolation room in the state of door opening.

1. Introduction

Effective isolation measures in the isolation room are important for controlling the airborne spread of infectious diseases. Several infectious agents such as severe acute respiratory syndrome (SARS) [1] and tuberculosis (TB) [2] are transmitted by aerosols. The experience with SARS in 2003 has triggered increased attention to the problem of aerosol transmission, short range between healthcare workers and their patients [3–5]. Airborne transmission of contaminants may be restricted by the use of a negative pressure isolation room [6]. However, when the door of an isolation room is opened, the ventilation system fails to maintain the required negative pressure difference within the room, and this failure results in the dispersion of airborne contaminants out of the room [7]. An experimental model indicates that opening the door leads to a breakdown in isolation conditions and causes the transmission of infectious air out of the isolation room [8]. How to prevent the airborne transmission in the case of door opening is therefore an issue of grave importance. Therefore, this study proposes an arrangement of an anteroom to separate the isolation room and the corridor. The anteroom acts as an airlock to prevent direct dispersion of airborne contaminants from the room to the corridor. However, when the door of an isolation room is opened, the airborne contaminants in the isolation room will directly disperse to the anteroom instead of the corridor; healthcare workers leaving the isolation room and entering the anteroom are therefore at a higher risk of being infected.

Hence, it is crucial to understand the transmission of contaminated air through the doorway into the anteroom and take proper measures to reduce the spread of airborne contaminants from the isolation room. Air changes per hour (ACH) of ventilation system and airflow pattern of room air movement exercise considerable impacts on the control of airborne contaminant diffusion [9]. It is noted that airflow control is extensively employed to manage smokes from fires at subway, railroad and highway tunnels. Accordingly, controlling the airflow to dominate the room air movement will be evaluated in the present work on the prevention of airborne contaminant transmission. In general, the computational fluid dynamics (CFD) method is effectively employed to evaluate the effect of different air supply systems on indoor air distribution and ventilation effectiveness [10]. The numerical results from CFD software Fluent adopted to predict particle transport and distribution in ventilated rooms agree well with associated experimental data [11]. Using CFD to evaluate the performance of ventilation systems, our study aims to propose a strategy of preventing the dispersion of airborne contaminants from the isolation room when the door to the isolation room is opened and closed.

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2. Methods

2.1. Computational model

Fig. 1(a) illustrates the configuration of the studied model, including an anteroom of 2.5 m (width) × 2.5 m (height) × 1 m (length) in the X, Y, and Z directions respectively, and an isolation room of 2.5 m (width) × 2.5 m (height) × 3.7 m (length) in the X, Y, and Z directions respectively. In the isolation room illuminated by a set of light (80 W), the study had a patient lying on a bed at the height of 0.6 m from the floor. An exhaust air vent (2.5 m × 0.2 m) was located on the lower wall behind the head of the bed. In the anteroom illuminated by a set of light (40 W), a ceiling (1 m × 1 m) supply air vent was installed at the height of 2.5 m. As revealed in Fig. 1(b), a healthcare worker and medical equipment with a heat dissipation of 100 W stood by both sides of the bed. The sizes of the patient and healthcare worker were listed in Table 1. Except the location and the facing direction of the healthcare worker were different from the one of the patient, as shown in Fig. 1(a) and (b), the healthcare worker had the same size as the patient. The distance between the healthcare worker and the patient reported 0.75 m. In order to simulate the transmission of airborne contaminants, certain conditions were assumed regarding the patient, including an activity of 1 met unit (1 met = 58 W per square meter) corresponding to a resting state, a CO2 generation rate of 0.043. The temperatures of the supplied outdoor air and the patient, including an anteroom of 2.5 m (width) × 2.5 m (height) × 3.7 m (length) in the X, Y, and Z directions respectively, and CO2 as an indicator of airborne contaminants.

The gases exhaled by the patient were assumed to be composed of air and CO2 only and they were released from the mouth of the patient, with the temperature of 37 °C and the velocity magnitude of 0.17 m/s. The exhaled gases consisted of CO2 with a mass fraction of 0.043. All boundary walls of the model room were assumed to be adiabatic. Moreover, the fluid was considered incompressible with constant properties, and the flow was assumed to be steady and three-dimensional.

2.2. Numerical methodology

A reliable turbulence model, numerical scheme, and numerical algorithm could increase the accuracy of simulation. In this study, the CFD software Airpak [13] was utilized to predict room airflow distribution and concentration fields. Airpak is used as a tool for modeling, meshing, and post processing the computational model.

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### Table 1

| Name | Patient | Healthcare worker |
|------|---------|-------------------|
| Location, (X, Y, Z) in meter | (1.45, 0.7, 2.825) | (0.7, 3.8) |
| Facing direction | Y | X |
| Total height, (m) | 1.75 | 1.75 |
| Body, width (m) × depth (m) | 0.3 × 0.2 | 0.3 × 0.2 |
| Mouth, width (m) × height (m) | 0.04 × 0.02 | 0.04 × 0.02 |
| Head fraction | 0.18 | 0.18 |
| Body fraction | 0.4 | 0.4 |
| Lap fraction | 0.15 | 0.15 |

---

### Table 2

| Terms, coefficients and constants used in the governing equations |
|---------------------------------------------------------------|
| Equations | Constants |
| $H = C_l(T - T_{ref})$ | $C_l = 1.44$ |
| $P_r = \mu_l(u_l + u_j)u_j$ | $C_r = 1.92$ |
| $G_3 = \mu_3 u_i u_j / l_3^2$ | $C_3 = C_l \tanh(l_3/u_i)$ |
| $\mu_3 = \mu_3 \tanh(l_3/u_i)$ | $C_3 = 0.09$ |
| $\mu_l = \mu_l \tanh(l_3/u_i)$ | $C_3 = 0.9$ |
| $\nu = \nu + \nu_l$ | $C_3 = 1.0$ |
| $l_3 = C_3 l_3 / l_3$ | $\sigma_1 = 1.3$ |
| $l_3 = C_3 l_3 / l_3$ | $\sigma_1 = 0.85$ |
| $D_0 = D_0 + D_0$ | $S_0 = 0.7$ |

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Fig. 1. Isometric view of an anteroom and an isolation room: (a) heat generated from a patient and lightings, and (b) additional heat generated from a healthcare worker and medical equipment.
Fluent [14] solver engine solves the governing equations for the conservation of mass and momentum as well as for energy and other scalars, such as turbulence and chemical species. Turbulence models (Reynolds Averaged Navier-Stokes equations), especially two-equation model, already become the universal used methods in the computation of indoor flow field [15]. The popular $k$-$\varepsilon$ model has been successfully applied to simulate indoor airflow fields [16,17], and the performance of the standard $k$-$\varepsilon$ turbulence model is good for the prediction of natural and forced convection in a room [16]. The standard $k$-$\varepsilon$ turbulence model [18] developed earlier, gained the broadest application in the indoor airflow calculation [7,11,19–22], and was used to predict the airflow field in our study. The related fundamentals of CFD modeling of fluid flow and turbulence are available in References [23].

A control-volume-based technique [14] is used to convert the governing equation into an algebraic equation that can be solved numerically. This control volume technique consists of integrating the governing equations about each control volume, yielding discrete equations that conserve each quantity on a control-volume basis. The governing equation complies with the principle of conservation and is expressed in general form as follows:

$$
\frac{\partial}{\partial t} (\rho \phi) + \nabla \cdot (\rho \mathbf{V} \phi) = \nabla \cdot (\Gamma \nabla \phi) + S_{\phi}.
$$

(1)

Here $\rho$ represents the density, $\phi$ the dependent variables, $\Gamma \phi$ the effective exchange coefficient for the dependent variable $\phi$, and $S_{\phi}$ the source term. Table 2 tabulates the parameters $\phi$, $\Gamma \phi$, and $S_{\phi}$ employed in continuity, momentum, energy, contaminant, and turbulence equations.

Field variables (stored at cell centers) must be interpolated to the faces of the control volumes in the finite volume method. Accounting for the buoyancy flow within the model room, the Boussinesq approximation was employed. The diffusion terms were central-differenced and always second-order accurate. Using the second-order discretization will generally obtain more accurate results. Thus, the second order upwind scheme was used for the treatment of the convection and diffusion-convection terms in the governing equation. After discretization, the transport equation contains the unknown scalar variable $\phi$ at the cell center as well as the unknown values in surrounding neighbor cells. This equation, in general, is non-linear with respect to these variables. A linearized conservation equation for a general variable $\phi$ at a cell $P$ can be written as follows [14]:

$$
a_{cb} \phi_{c} = \sum_{nb} a_{cb} \phi_{nb} + b,
$$

(2)

where the subscript $nb$ refers to neighbor cells, $a_{cb}$ the center coefficient for $\phi$, $a_{cb}$ the influence coefficients for the neighboring cells $\phi_{nb}$, and $b$ the source term. This results in a set of algebraic equations with a sparse coefficient matrix. For scalar equations, Fluent solves this linear system using a point implicit (Gauss-Seidel) linear equation solver in conjunction with an algebraic multi-grid (AMG) method. The algorithm of semi-implicit method for pressure linked equations (SIMPLE) [24] was used to solve iteratively the pressure-velocity coupling equations [11,25–29].

Moreover, this study judged the convergence not only by examining residual levels, but also by monitoring relevant integrated quantities. All discrete conservation equations are obeyed in all cells to a specified tolerance and solution no longer changes with more iterations. The convergence criterion in this study requires that the scaled residuals decrease to $10^{-4}$ for all equations in the governing equations except the energy, for which the criterion is $10^{-6}$.

Four grid densities, 83,274 (Case 1), 153,618 (Case 2), 220,714 (Case 3), and 310,284 (Case 4) cells, were investigated to perform the grid sensitivity study. The grid is structured, and cells are hexahedral. Fig. 2 presents the results of grid independent test; here $V$ refers to the velocity magnitude of room air monitored at a specific location close to the patient, $V_{o}$ the maximum velocity magnitude of room air at the boundary condition of velocity inlet, $H$ the height of room air monitored at the specific location close to the patient, and $H_{o}$ the height of room. The dimensionless velocity at the monitored points in the case of 220,714 cells was quite close to the one in the case of 310,284 cells; moreover, the relative error of the average dimensionless velocity of room air between Cases 1 and 2, Cases 2 and 3, and Cases 3 and 4 reported 0.89%, 1.06%, and 0.42%, respectively. As the difference between Cases 3 and 4 was insignificant, it could be concluded that the grid system reached an independent solution. Therefore, the grid density with 220,714 cells was found to be sufficient and applied into the continue study.

The standard wall functions for the $k$-$\varepsilon$ turbulence model were adopted to link the solution variables at the near-wall cells and the corresponding quantities near the wall. The wall standard wall functions employed in our study is based on the proposal of Launder and Spalding that have been extensively used for industrial flows. To resolve reliable turbulence phenomena near the wall, the grid nearest to the wall fell into the logarithmic layer (that is, $y^+ >30–60$). Moreover, the turbulence kinetic energy $k$ and the turbulence kinetic energy dissipation rate $\varepsilon$ employed in air supply and air outlet are calculated by the following equations:

$$
k = \frac{3}{2} (u_{avg})^2,
$$

(3)

$$
\varepsilon = C_{\mu} \left( \frac{l}{k^{3/2}} \right),
$$

(4)

$$
l = 0.07D_{h},
$$

(5)

where $u_{avg}$ refers to the mean flow velocity, $l$ the turbulence intensity, $D_{h}$ the hydraulic diameter, and $C_{\mu} = 0.09$.

3. Results and discussion

3.1. Airflow patterns

In order to provide controlled airflow patterns, the air supply and exhaust are located so that clean air flows first to parts of the room where healthcare workers probably work and then across the infectious source and into the exhaust [30]. The air, therefore, flows from a cleaner area (the anteroom) to a contaminated area (the isolation room) to prevent the spread of airborne contaminants. In case of mechanical ventilation failure as in Fig. 3(a), the...
Fig. 3. Stream traces of airflow of (a) 0, (b) 12, (c) 24, and (d) 48 mechanical ACH at the y-z plane of $x = 0.5$ m (cuts across the center of doorway).

Fig. 4. Contaminant contours in a y-z Cross-Section (cuts across the center of doorway) at airflow of (a) 0, (b) 12, (c) 24, and (d) 48 mechanical ACH.
airflow direction was thermally driven. A clockwise swirl of airflow was generated in the isolation room; room air flowed from the isolation room to the anteroom; and the swirl carried the airborne contaminants from the contaminated area to the cleaner area. As revealed in Fig. 3(b), when the mechanical ventilation started to function at a rate of 12 ACH, the direction of its airflow was contrary to the one in Fig. 3(a). It was noted that the reversed airflow occurred at the doorway due to the effects of thermal buoyancy and an average temperature difference of 1.59 °C between the two rooms, causing the airborne contaminants to be spread into the anteroom. An arrangement of left-upper ceiling supply and right-lower wall exhaust plus a mechanical ventilation system resulted in airflow swirl rotating counter-clockwisely in the upper parts of the isolation room. However, increasing the flow rate to 24 or 48 ACH helped eliminate the phenomenon of reversed flow and restricted the swirls to the local area of the isolation room (Fig. 3(c) and (d)), as more air flowing at the rates of 24 and 48 ACH reduced the average temperature difference between the two rooms to 1.05 and 0.63 °C, respectively, and decreased thermal buoyancy. Additionally, the ASHRAE Standard limits the air velocity to a specified maximum of 0.8 m/s in summer for thermal comfort [31]. Based on the results of simulation, the maximum air

Fig. 5. Contaminant contours in a x-z Cross-Section at the height of breathing zone (y = 1.5 m) at airflow of (a) 0, (b) 12, (c) 24, and (d) 48 mechanical ACH.

Fig. 6. Contaminant contours in a y-z Cross-Section (cuts across the center of doorway) at airflow of (a) 12 and (b) 48 mechanical ACH in the case of higher heat source.
velocity in the isolation room read 0.17, 0.31, and 0.61 m/s for airflow rates of 12, 24, and 48 mechanical ACH, respectively, and these values were well below the ASHRAE Standard.

3.2. Concentration profile of airborne contaminants

A greater concentration of airborne contaminant occurred at the corner and sides of the isolation room. In Fig. 4(a) where the isolation room has zero mechanical ACH, the concentration of airborne contaminants in most areas was 111–113 ppm in both rooms. It was noted that in Fig. 4(b)–(d) the mechanical ventilation systems diluted and removed contaminated air and controlled airflow patterns in the isolation room. Directional airflow prevented the spread of airborne contaminants from the isolation room into the adjacent anteroom. Increasing air flow rate to 48 ACH effectively prevented the contaminated air from drifting out of the isolation room into the anteroom. As shown in Fig. 4(d), the contaminant level in the y-z cross-section within the anteroom was around 0–2 ppm, which was much lower than the concentration of 28,750 ppm of the source contaminants generated from the patient. Fig. 5 shows the concentration contours of airborne contaminants at a height of the breathing zone of a standing person. It was found that the Case of 48 ACH (Fig. 5(d)) effectively prevented the dispersion of airborne contaminants out of the isolation room as the contaminant level in the x-z cross-section at the height of 1.5 m in the anteroom revealed 0 ppm.

In the case of mechanical ventilation, fresh air came from the anteroom through the doorway and left from the isolation room. Accordingly, the average room concentration of airborne contaminants in the anteroom was lower than that in the isolation room. The calculations show that the average room concentration of airborne contaminants in both rooms decreased when ACH of the ventilation system increased. The average concentrations of airborne contaminants within the isolation room in the cases of 24 and 48 ACH were, respectively, 50.3% and 27% of that in the case of 12 ACH, indicating that the ability of removing airborne contaminants in the cases of 24 and 48 ACH was, respectively, 2.0 and 3.7 times greater than the one in the case of 12 ACH. The average concentrations of airborne contaminants within the anteroom in the cases of 24 and 48 ACH were, respectively, 16.3% and 1% of that in the case of 12 ACH, meaning that the ability of preventing the escape of airborne contaminants in the cases of 24 and 48 ACH was, respectively, 6.1 and 97.9 times greater than the one in the case of 12 ACH. However, in the case of mechanical ventilation failure, the thermal buoyancy controlled the airflow direction from the isolation room toward the anteroom. The result was an increased concentration of airborne contaminants in the anteroom. But the concentration of airborne contaminants in the anteroom remained slightly lower than that in the isolation room as higher concentration of airborne contaminants occurred in the isolation room.

The average air velocity through the doorway was 0.02, 0.07, 0.11, and 0.22 m/s, respectively, for airflow rates of 0, 12, 24, and 48 mechanical ACH. A low pressure differential can effectively separate clean and less clean adjacent zones by means of displacement airflow, and its airflow velocity should be typically above 0.2 m/s [32]. In the case of 48 ACH flow rate, the average room concentra-
tion of airborne contaminants in the anteroom was 0.155 ppm, i.e. 0.000538 % of the concentration of airborne contaminants at the source generated by the patient. This value was much lower than the source concentration and approached zero ppm; therefore, the ventilation system at a rate of 48 ACH effectively separated the anteroom and the isolation room. The present study verified that an airflow velocity above 0.2 m/s as recommended in the ISO 14644, a standard for cleanrooms and associated subjects, could be adopted for application to isolation rooms and effectively controlled the movement of airborne contaminants.

3.3. Influence of the healthcare worker and medical equipment in the isolation room on the ventilation effect

Lightings and the patient were the original heat sources and resulted, respectively, in an average temperature difference of 1.59 and 0.63 °C between the isolation room and anteroom for the cases of 12 and 48 ACH. With the presence of a healthcare worker and medical equipment in the isolation room, these additional heat sources raised the average temperature difference to 3.5 and 1.8 °C between the two rooms for the cases of 12 and 48 ACH, respectively. The standing person caused a flow disturbance, and the heat generated thermal plumes that could bring the airborne contaminants to the upper zone. The convective flow around the patient brought the air at a lower zone to the breathing zone of the standing person. Therefore, the healthcare worker was exposed in an area with a higher concentration of airborne contaminants. However, a mechanical ventilation system above 48 ACH was able to extract airborne contaminant and reduced its concentration. The left-side upper corners of the isolation room as shown in Fig. 6 stayed as high contamination zones. The increase in the temperature difference caused airflow reversal at the doorway in the cases of 12 and 24 ACH, resulting in stronger diffusion of airborne contaminants from the isolation room to the anteroom and raising the contaminant concentration in the anteroom in the cases of 12 and 48 ACH from 15.16 to 21.25 ppm and from 0.16 to 0.55 ppm, respectively. The contaminant concentration in the isolation room was slightly increased as well in the cases of 12 and 48 ACH from 48.811 to 50.81 ppm and from 13.19 to 13.40 ppm, respectively. Fig. 7 illustrates that the highest contaminant concentration was found above the head of the patient and stronger airflow (Case 48 ACH) effectively prevented the diffusion of contaminant concentration.

3.4. Ventilation system recommended for a negative pressure isolation room

According to the above results, this study proposed a ventilation system as shown in Fig. 8 to prevent the spread of airborne contaminants from the negative pressure isolation room. Normally, the motorized airtight dampers M1-M4 were open, and manual volume dampers, VP, were modulated once in order to provide the required ACH and negative pressure differentials in the anteroom and the isolation room. In the case of door opening, while interlocking with the door, airtight dampers M2 and M3 were automatically closed to raise the flow rate of air into the anteroom. Accordingly, sufficient air flowed at a rate above 48 ACH from the ceiling supply, air vent in the anteroom through the doorway and resulted in an air velocity above 0.2 m/s to effectively prevent the spread of airborne contaminants.

4. Conclusions

The present study improves understanding on how ventilation air is distributed and how airborne contaminants are transported in the isolation room and anteroom and increases awareness of the potential health risk associated with the airborne contaminants in the case of mechanical ventilation system failure. Airflow patterns are analyzed to be affected by ventilation rates and air temperature differentials. In addition, a design of ventilation system is proposed to control air flow rate for containing airborne contaminant and preventing its spread to the adjacent anteroom when the door to the isolation room is opened and closed. It should be further noted that studies on clean rooms suggest that an airflow velocity above 0.2 m/s is capable of effectively controlling the movement of airborne contaminants. Our research verifies that the same finding can also be applied to regulate the movement of airborne contaminants in isolation rooms. Yet, how the movement of healthcare workers through the doorway of the isolation room may influence the ability of airflow to prevent the spread of airborne contaminants remains an issue that merits future studies.

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