CFD Modeling for Ventilation System of a Hospital Room

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Abstract Efficient ventilation can contribute to reducing the cooling energy consumption of buildings, increasing comfort level of residence, and minimizing the risk of airborne infection in hospital rooms. In this paper, we investigate ventilation in a hospital room considering forced and natural ventilation, and the flow of bacteria particles originating from a sick patient.

Keywords: CFD, Ventilation, Hospital, Thermal Comfort

1. Introduction

More than two million people in Europe are infected due to Health-care Associated Infection (HAI) (Pittet et al., 2005). Although it is believed that transfer of infection via contact is the main cause for HAI, there are evidences that airborne bacteria may also cause infection due to inhalation of infectious bacteria (Hathway, 2008, Brachman, 1970). Therefore it is essential to understand dynamics of infectious particles due to respiratory diseases such as SARS and Tuberculosis, TB.

Indoor ventilation with good air quality control prevents infection by minimizing the spread of airborne respiratory and other infections in hospitals. Computational Fluid Dynamics (CFD) can be utilized to optimize airflow pattern in clean rooms such as hospital clean rooms. With CFD one can obtain a better insight into the aerosol contamination dispersion characteristics. CFD can also be used to optimize airflow pattern and temperature distribution to achieve a better thermal comfort level.

This paper analyses airflow pattern and aerosol transport in a single-bed hospital room. The following topics are discussed: thermal comfort level of a hospital room and respiratory transmission emanating from a patient mouth.

2. Problem Description

The hospital room in our model contains a patient, a doctor, a bed, a wardrobe, a lamp, medical equipment, an inlet and exhaust. We accounted for both forced and natural ventilation, and the flow of bacteria particles originating from a sick patient. Layout of the room is shown in Figure 1. Ventilation rate is 6 ACH (Air Change per Hour) for health care facilities per ASHRAE standard 170. Air enters the room from a ceiling diffuser (inlet) with temperature of 20°C and leaves the room through a ceiling mounted grill (outlet) as shown in Figure 1. Inlet and outlet are extended with small domains to avoid numerical instability. Here heat sources are a lamp, which is installed between the diffuser and the grill, and the medical equipment located at the corner of the room. In addition the doctor and the patient generate heat with a constant heat flux. Table 1 list the heat sources and their temperature or heat flux.

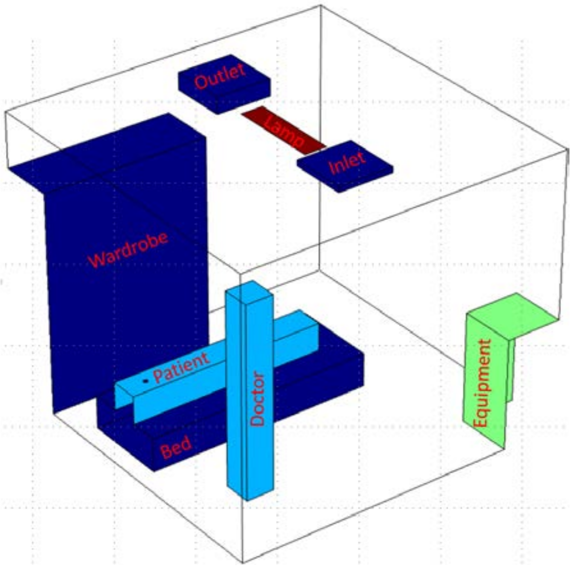

Figure 1. Room layout

It is assumed that the room is thermally isolated on three sides and its base, while heat exchange between the room and outside occurs through the ceiling and the fourth side of the room, which is the wall on the opposite side of the room from wardrobe.

Table 1: Heat sources

Body heat flux for doctor and patient	60 W/m^2
Equipment heat flux	100 W/m^2
Lamp heat Flux	200 W/m^2
Inlet temperature	20 °C
Outside temperature	5 °C

3. Use of COMSOL Multiphysics

For this study, we used “Turbulent Flow” and “Heat Transfer in Fluids” physics in COMSOL 5.1 in order to obtain airflow due to the natural and forced convection in the hospital room. Theses physics solve coupled Navier–Stokes and continuity equations for air as well as the convection-diffusion equation for heat. The k–omega (k–ω) turbulence model with consistent stabilization is used to solve turbulent airflow.

Variation of density with temperature, which is the driving force for natural convection is captured with “Multiphysics/Non-Isothermal Flow” physic in COMSOL. Therefore we did not use Boussineq approximation to account for variation of density. It should be noted that although Boussineq approximation was popular method for solving non isothermal flow it only slightly reduces the nonlinearity of the system and with today’s computational power the reduction in computational costs is negligible.

We used boundary layer meshes at all boundaries. The model has 1.4 million degrees of freedom and the computational time is not trivial. In order to speed up the simulation, the model is solved in three steps. In the first step we used a coarse mesh and higher viscosity for air. After the solution converges we use the developed flow from the first step as an initial condition for the second step, which has a fine mesh. In the third step the air viscosity is gradually reduced to its true value using a parametric sweep in COMSOL.

In order to simulate release of bacteria due to coughing, we used “Particle Tracing for Fluid Flow” physic. This physic traces particles after the airflow in the room is solved. Since masses of bacteria are negligible, we used the massless particle formulation for this study.

Two hundred and fifty bacteria particles are released during coughing time. The goal of this study is to quantify the percentage of bacteria that are leaving the hospital room through the exhaust of the ventilation system.

4. Results

4.1 Flow Field and Temperature Distribution

Figures 2 show temperature distributions and velocity vectors adjacent to the doctor and to the patient.

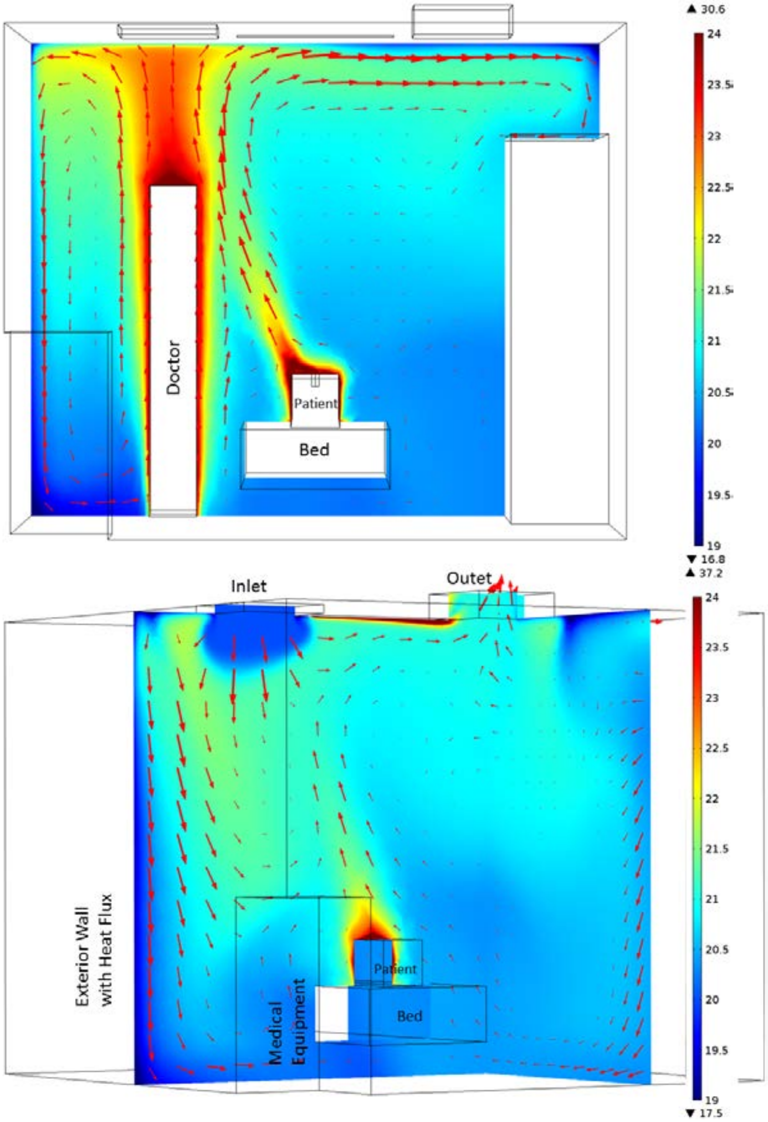

Figure 2. Temperature distribution and velocity vector in the room (Top shows a section that passes through the doctor and the patient, Bottom shows two perpendicular sections that passes through the inlet and the outlet). The color legend corresponds to temperature (°C)

Average temperature of the room is predicted to be 21°C. Upward air movement next to the doctor and patient, and downward air movement next to the wall are seen in Figure 2, which are due to natural convection. Figure 2 also shows flow of air from the patient toward the doctor. There are two recirculation zones above the wardrobe and next to the doctor, where bacteria can be trapped in and stay longer in the room.

4.2 Thermal Comfort

Thermal comfort is the state of mind that expresses satisfaction with the thermal environment. Thermal comfort is not related to the equations for heat and mass transfer and energy balance; it is assessed by subjective evaluation (ASHRAE Standard 55). ASHRAE thermal sensation scale given in Table 2 is often used to characterize thermal comfort level.

Table 2: ASHRAE Thermal sensation scale

Value	Sensation
+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
-1	Slightly cold
-2	Cool
-3	Cold

The main factors that influence thermal comfort are those that affect heat and mass transfer in energy balance models, namely metabolic rate, clothing insulation, air temperature, mean radiant temperature, air speed and relative humidity. The common approach to characterizing thermal comfort is to correlate the results of psychological experiments to thermal analysis variables. The average thermal sensation response of a large number of subjects using ASHRAE thermal sensation scale is called the Predicted Mean Vote (PMV). ASHRAE 55 recommends the acceptable PMV range for thermal comfort to be between -0.5 and +0.5 for an interior space.

Fanger's Comfort model (Fanger 1967), which is derived from experimental data, is a mathematical relationship between all the aforementioned environmental and physiological factors and PMV. Fanger also developed Predicted Percentage of Dissatisfied (PPD) model, which is a quantitative measure of the thermal comfort of a group of people at a particular thermal environment. PPD predicts percentage of occupants that will be dissatisfied with thermal conditions as PMV moves from 0 or neutral. ASHRAE 55 uses the PMV model to set the requirements for indoor thermal conditions. It requires that at least 80% of the occupants be satisfied.

Based on our simulation, average air temperature and air speed experienced by the patient is 20.8°C and 0.06 m/s respectively. These data along with metabolic rate, humidity, clothing level and mean radiant temperature are needed to estimate PMV. We calculated PMV and PPD per ASHRAE Standard 55-2013. The metabolic rate of a patient who is sleeping is 1 Met. We assumed that the patient is wearing trousers and a long-sleeve shirt and therefore his clothing level is 0.61 clo. If humidity is assumed to be 50% and mean radiant temperature 21°C, then PMV is -1.59 and PPD is 56%. In that case, the probability of the patient being dissatisfied with the room temperature is 56%, with a sensation of being cool.

4.3 Bacteria Distribution due to Coughing

Coughing is a transient phenomenon with a flow rate similar to a skewed triangular pulse for the duration of coughing. Coughing characteristics are obtained from "Flow dynamics and characterization of a cough" by (Gupta et al, 2009). Computational time of a transient analysis for coughing in a hospital room is extensive. Since the airflow rate from coughing and its momentum are negligible compared to the airflow rate inside the room, we assumed that coughing does not affect the general airflow pattern of the room. Therefore for the steady state model of airflow and temperature, we used a constant airflow rate at the mouth of the patient equivalent to the coughing flow rate. In the second analysis of particle tracing, bacteria particles are released for a typical coughing duration, which is 0.5 seconds.

Figure 3 shows transmission probability of bacteria at the exhaust versus time; none of the bacteria leaves the room in less than 30 seconds after coughing. After 300 seconds 8% of the bacteria are still remaining in the room, which can be the source of infection.

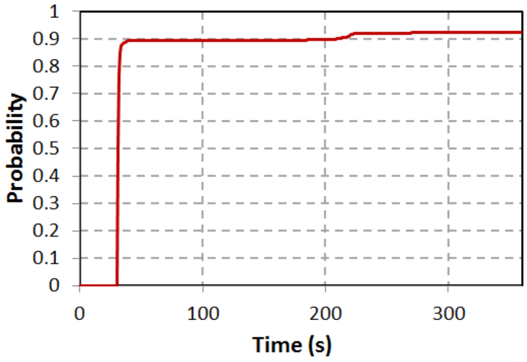

Figure 3. Transmission probability of bacteria at the exhaust

Figure 4 shows the motion of bacteria at times 30, 60, 180, and 230 seconds after coughing. The majority of bacteria leave the room through exhaust 30 seconds after coughing. However about 10 percent of them stay in the room and move to the bottom corner of the room over the next 30 seconds. Figure 4c shows that during the next 120 seconds they move slowly parallel to the floor and toward the patient again. They rotate and move up next to the patient. About 2 percent of the remaining bacteria leave the room 220 seconds after coughing. However the last 8% stay in the room as shown in Figure 4(d). The remaining bacteria may contaminate the entire room and increase the risk of airborne infections.

5. Conclusion

Computational fluid dynamics can be used to increase the comfort level and improve ventilation design and energy efficiency of buildings.

Indoor ventilation with good air quality control prevents infection with minimizing the spread of airborne respiratory and other infections in hospitals. With computational fluid dynamics we can obtain better insight into aerosol contamination dispersion characteristics to optimize airflow pattern in hospital clean rooms.

6. References

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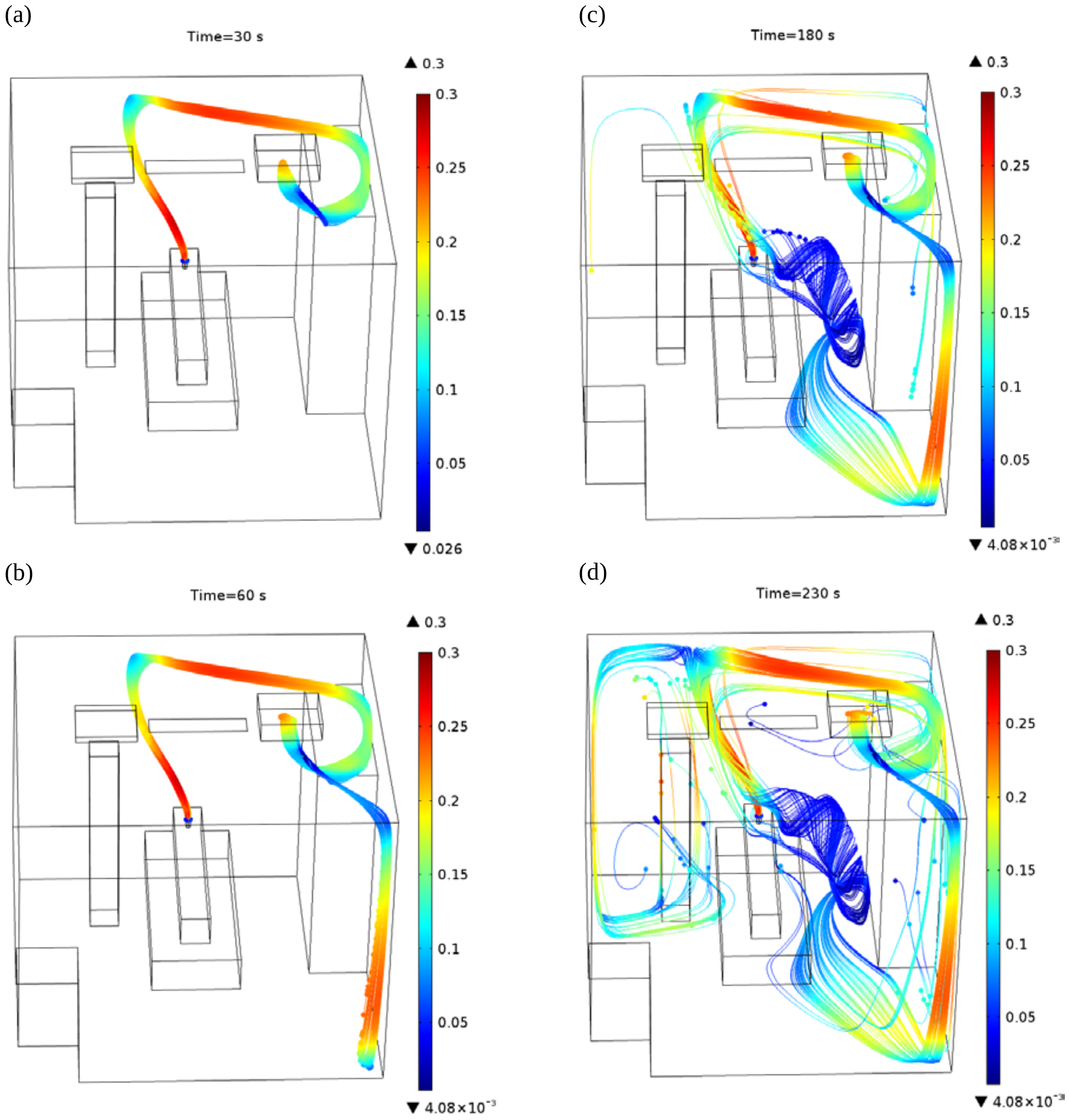

Figure 4. Particle tracing showing the motion of bacteria particles at 30, 60, 180 and 230 seconds after patient coughing. Particle color corresponds to velocity (m/s)