PERFORMANCE ASSESSMENT FOR LOCALLY BALANCED AND WALL-RETURN TURBULENT CLEAN ROOMS BY THE STOCHASTIC PARTICLE TRACKING MODEL

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ABSTRACT

This study analyzes, simulates, and compares two types of turbulent clean rooms, by computational fluid dynamics analysis. The dimensions of the clean room models are 7.8 m (L) × 7.8 m (W) × 2.7 m (H). The FLUENT flow solver with the k-ε two-equation turbulence model was used to calculate the turbulent flows. In addition to solving transport equations for the continuous phase (air), a discrete phase model in a Lagrangian frame of reference was used to evaluate and statistically analyze trajectories of sub-micro particles in the clean rooms. The stochastic tracking model was applied to predict the dispersion of particles due to turbulence of airflow. The wall-return type turbulent clean room is less able to exhaust particles than the locally balanced type. Particles generated close to an exhaust air opening are more effectively exhausted from the clean room.

1. INTRODUCTION

Turbulent type clean rooms are adopted in many areas, such as high-tech production environment, operation room in hospitals, precision machine assembly lines and so on. When designing the ventilation system for turbulent type clean rooms, understanding the flow field and the best means of controlling the contaminants are very important. A clean room with supply air opening on the ceiling and exhaust air opening on the wall, hereafter referred as a wall-return type clean room, is traditionally assumed to facilitate particle exhaust, because the direction of airflow is consistent with the movement of particle settling. Installing supply as well as exhaust openings on the ceiling is common in office buildings. However, from the view point of ordinary contamination control in a clean room such an arrangement is not recommended usually for the following reasons.

First, contaminants generated near the floor are carried toward the ceiling by the rising stream of air, and are thus exhausted only after rising to the ceiling, in a manner inconsistent with the principle of immediate exhaust of contaminants.

Second, a “short circulation” may be formed between the supply and the exhaust flows when the inlet and outlet are set very close to each other, possibly creating a stagnant flow region in which the contaminant is not exhausted effectively.

However, following the extensive application of the fan-filter units (FFU) as the primary mover of the clean room recirculation air, installing supply as well as exhaust air openings on the ceiling is getting accepted because such an arrangement applies the characteristic of negative pressure in a supply air plenum, facilitating the return air from the interior to the ceiling. Other than in a commercial building, the supply and exhaust air openings in a clean room with FFU can be arranged to match for each other. Therefore, it is possible to confine the diffusion of contaminants in the supply/exhaust territory, as indicated in Fig. 1(a).

A clean room with such an arrangement is referred as a “locally balanced” clean room, in which the supply and exhaust airflow rates are balanced locally within a flow unit. Fig. 1(b) schematically shows the flow unit created by the downward-upward streams with supply/exhaust opening installed on the ceiling. In practice, installing both supply and exhaust air openings on the ceiling is considered very effective in arranging the return air path for industrial clean rooms in which production lines and layout of the production tools are often changed (i.e. the return air is not influenced by the relocation of the production line).

Fig. 1(c) depicts a wall-return type clean room with FFU. For this type of clean room, return air shaft adjacent to the wall through which return air passes is required. This return air shaft occupies space in a clean room. Moreover, when exhaust air openings are installed next to the floor in walls, the return airflow is significantly influenced by production tools or devices arranged near the wall and by the movement of an operator.

This study aims to clarify the airflow characteristics and particle exhaust performance of
a wall-return clean room and a locally balanced clean room, by analyzing influential factors including obstacles, particle diameters, and particle injection positions from the viewpoint of particle trajectories.

2. PREVIOUS STUDIES

Murakami et al. [1] indicated that a change in arrangement or in the number of exhaust openings hardly affects the entire flow field. However, such change often has a large influence on the contaminant diffusion field since the path of contaminant transportation is changed greatly by the position of the exhaust openings. Murakami et al. [2] numerically studied the diffusion field of contamination in a turbulent flow type clean room, which has a locally balanced supply-exhaust airflow rate system.

Fig. 1(a): Empty clean room with locally balanced supply-exhaust arrangement (SA = supply air)

Fig. 1(b): Flow unit created by the downward-upward streams with supply/exhaust opening on the ceiling

Fig. 1(c): Empty clean room with wall-return supply-exhaust arrangement (RA = return air)
Lu and Howarth [3] used a numerical model predicting the air movement and aerosol particle deposition and migration in two interconnected ventilated zones. The particle deposition and migration are mainly influenced by the particle properties, the ventilation conditions and the airflow patterns in the two zones. Particle migrations at high ventilation rates are more severe and faster than those at low ventilation rates. Large particles \( d_p \geq 4 \ \mu m \) deposit much faster than small particles \( d_p \leq 2 \ \mu m \).

Shimada et al. [4] studied experimentally and theoretically the change in concentration distribution of particulate contaminants emitted in a uni-directional flow type clean room.

3. MATHEMATICAL MODEL

- Mean flow field simulation

The air movement in the steady flows is governed by the Eulerian conservation equation:

\[
\text{div} \ (\rho V \phi) = \text{div} \ (\Gamma \phi \nabla \phi) + S \phi
\]  

where \( \phi \) represents each of 1, the three velocity components \( u, v, \) and \( w, \) the turbulence kinetic energy \( k, \) and the dissipation rate of the turbulence kinetic energy \( \varepsilon. \) \( \Gamma \phi \) is the effective exchange coefficient for the dependent variable \( \phi. \) \( S \phi \) the source term, is given in detail in Table 1.

In Table 1, \( G_k \) is the kinetic energy generation rate and expressed as below:

\[
G_k = 2\mu_t \left[ \left( \frac{\partial u}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial y} \right)^2 + \left( \frac{\partial w}{\partial z} \right)^2 \right]
+ \mu_t \left[ \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + \left( \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right)^2 \right]
\]  

The standard k-\( \varepsilon \) two-equation model of turbulence is applied in this study. \( \mu_e \) is the effective viscosity, defined by:

\[
\mu_e = \mu + \mu_t
\]  

\( \mu \) is the molecular viscosity, \( \mu_t \) is the turbulent viscosity and is defined as follows:

\[
\mu_t = C_\mu \rho \frac{k^2}{\varepsilon}
\]  

The constants in the k-\( \varepsilon \) two-equation model are assigned the following values, which were recommended by Launder and Spalding [5]:

\[
C_1 = 1.44, \ C_2 = 1.92, \ C_\varepsilon = 0.09, \ \sigma_k = 1.0, \ \sigma_\varepsilon = 1.3
\]

- Fluctuating velocities simulation

Assuming homogeneous turbulence, the fluctuating velocities of air are calculated from kinetic energy of turbulence:

\[
u'_i = \zeta \sqrt{2k/3}
\]  

where \( \zeta \) is a normally distributed random number ranging from 0 to 1, and \( k \) is the turbulent kinetic energy. For small particles that move with the fluid, the integral time scale becomes the fluid Lagrangian integral time scale, \( T_L. \) This time scale can be approximated as:

\[T_L = C_L k / \varepsilon\]

where \( C_L \approx 0.15 \) for k-\( \varepsilon \) two-equation turbulence model. The characteristic lifetime of the eddy (\( \tau_e \)) is defined as:

\[
\tau_e = 2 T_L
\]

<table>
<thead>
<tr>
<th>Table 1: Source terms in the governing equations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equation</td>
</tr>
<tr>
<td>Continuity</td>
</tr>
<tr>
<td>u momentum</td>
</tr>
<tr>
<td>v momentum</td>
</tr>
<tr>
<td>w momentum</td>
</tr>
<tr>
<td>Kinetic energy</td>
</tr>
<tr>
<td>Dissipation rate</td>
</tr>
</tbody>
</table>
Particle equation of motion

The trajectory of a discrete phase particle is predicted by integrating the force balance on the particle, which is expressed in a Lagrangian reference frame. It is noted that the dispersion of submicron particles is strongly affected by the instantaneous fluctuation velocity. These turbulence fluctuations are random functions of space and time.

The equation of motion of a small aerosol particle including the electrical force is given by:

\[
\frac{du_i^p}{dt} = \frac{3\nu C_D}{4d^2} \left( \frac{u_i - u_p}{\nu} \right) + F_l^i + n_i(t) + \frac{F_e}{m} + g_i \tag{8}
\]

and

\[
\frac{dx_i}{dt} = u_i^p \tag{9}
\]

Here, \( u_i^p \) is the velocity of the particle, \( x_i \) is its position, \( d \) is the particle diameter, \( S \) is the ratio of particle density to fluid density, \( F_l^i \) is the lift force per unit mass, \( n_i(t) \) is the Brownian force per unit mass, \( F_e \) is the electrical force, \( g_i \) is the gravity, and \( m \) is the mass of the particle.

The first term on the right hand side (RHS) of equation (8) is the drag force due to the relative slip between the particle and the fluid. The drag force is, generally, a dominating force. The drag coefficient, \( C_D \), is given as:

\[
C_D = \frac{24}{Re_p^2} \quad \text{for} \quad Re_p < 1 \tag{10}
\]

and

\[
C_D = \frac{24}{Re_p^2} \left(1 + \frac{1}{6 Re_p^2} \right) \quad \text{for} \quad 1 < Re_p < 400 \tag{11}
\]

where \( Re_p \) is the particle Reynolds number defined as:

\[
Re_p = \frac{d|u_j - u_j^f|}{\nu} \tag{12}
\]

In equation (12), \( C_c \) is the Stokes-Cunningham slip correction given as:

\[
C_c = 1 + \frac{2\lambda}{d} \left[ 1.275 + 0.4e^{-1.1d/2\lambda} \right] \tag{13}
\]

where \( \lambda \) is the molecular mean free path of the gas.

The second term on the RHS of equation (8) is the Saffman lift force per unit mass given by:

\[
F_l^i = \frac{2K\nu^3}{Sd(d_{ij}d_{kj})} \left( u_j - u_j^f \right) \tag{14}
\]

Here, \( K = 2.594 \) is the constant coefficient of Saffman’s lift force, and \( d_{ij} \) is the deformation rate tensor defined as:

\[
d_{ij} = \frac{1}{2} \left( \frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right) \tag{15}
\]

The third term on the RHS of equation (8) is the Brownian force per unit mass, which is very important for submicron particles. The Brownian force is modeled as a Gaussian white noise random process.

The last term on the RHS of equation (8) is the electrical force acting on a charged particle.

It is noted that both the Brownian force and electrical force are neglected. The Brownian force is not considered because the flow is assumed to be turbulent.

4. SIMULATION ANALYSIS

Basic assumptions

Several assumptions are made to simplify the simulation. They include, (1) incompressible, iso-thermal, and 3D steady-state turbulent flow, (2) spherical solid particles, and (3) no interaction between the continuous and discrete phases.

Boundary conditions

The boundary conditions specified for continuous and discrete phases in this study are defined as follows. The inlet flow is defined as entering the room with a uniform velocity, \( w_i = 0.35 \, \text{m/s} \) (25 air changes per hour). The inlet air turbulence kinetic energy is assumed as \( k = 0.002 w_i^2 \), which corresponds to a turbulence intensity level of 4%. The turbulence dissipation rate is given by \( \varepsilon = C_{\mu} \frac{k^{1.5}}{l} \), with the mixing length \( l = 0.07 \, \text{L} \), where \( L \) is the hydraulic diameter of the supply inlet. Neumann boundary conditions are applied at the outlet to satisfy the mass conservation law. A non-slip condition at the solid wall is applied, such that the velocity at its surface is set to zero. The standard wall function is applied to describe the turbulent flow properties near walls.
For discrete phase calculations, the diameters of particles are divided into three groups - 0.1 µm, 0.5 µm, and 1.0 µm. The “escape” boundary condition is imposed at the outlet boundaries. The “escape” condition defines the particles as having “escaped” when they encounter the boundaries. When particles encounter any wall, they are assumed to be “trapped” on the wall, indicating that the calculation is terminated and the particles remain “attached” to the walls. All particles are assumed to have the same density of 1000 kgm⁻³.

- Room models

Figs. 2(a) and 2(b) depict the wall-return clean room and locally balanced clean room, respectively. The wall-return clean room has an exhaust mode in which four exhaust openings are located in the wall near the floor. The locally balanced clean room has supply and exhaust in a diamond pattern to catch the rising streams at the ceiling. A box-shaped obstacle is placed at three different positions, including the center of the room, between the center of the room and the wall, and near the wall.
Particles are released at a height of 80 cm above the floor, (positions A, C, and E), and 150 cm above the floor, (positions B, D, and F), as shown in Fig. 2. For a locally balanced clean room, positions A and B are under the supply air openings in the center of room; positions E and F are under the exhaust air openings, and positions C and D are between the supply and exhaust air openings. One hundred particles with each diameter, 0.1 µm, 0.5 µm, and 1.0 µm, are released from each position from A to F, therefore six hundred particles per diameter are injected.

3. **MESH SYSTEM**

A wall-return model with an obstacle in the center of the room is selected to execute the grid-independent test. Fig. 3 compares the results for seven monitoring points in the x direction, named x1, x2, x3, x4, x5, x6, and x7. These points are located in the center of room in the y direction, referring to line A, as shown in Fig. 2(a).

Three meshes each have different numbers of cells $65(x) \times 65(y) \times 22(z)$ (with 92,950 control
volumes), 52(x) × 52(y) × 18(z) cells (with 48,672 control volumes), and 39(x) × 39(y) × 14(z) cells (with 21,294 control volumes). Fig. 4 shows the results of the grid-independent test. The velocities at the monitoring points in the case of 52(x) × 52(y) × 18(z) cells are close to those in the case of 65(x) × 65(y) × 22(z) cells, while those in the case of 39(x) × 39(y) × 14(z) cells are very different from the velocities in the other two cases, especially at positions x1 and x7. The case of 52(x) × 52(y) × 18(z) cells in a uniform mesh is adopted to ensure reasonable computational effort in terms of CPU time and memory. The space discretization methods applied for the various transportation equations are depicted in Table 2. The values of cell Peclet number (Pe) of the three different mesh systems examined are much greater than 1. To a strongly convective flow in a turbulent type clean room, the second order unwind scheme used in this study is appropriate.

### Table 2: The discretization methods

<table>
<thead>
<tr>
<th>Items</th>
<th>Discretization methods</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>Standard</td>
</tr>
<tr>
<td>Momentum</td>
<td>Second order upwind</td>
</tr>
<tr>
<td>Pressure-velocity coupling</td>
<td>SIMPLE</td>
</tr>
<tr>
<td>Turbulence kinetic energy</td>
<td>Second order upwind</td>
</tr>
<tr>
<td>Turbulence dissipation rate</td>
<td>Second order upwind</td>
</tr>
</tbody>
</table>

Fig. 3: Monitoring position for grid-independent test

Fig. 4: Results of grid-independent test
• Case setup

Table 3 describes all the simulated cases and their associated case codes. First, two types of exhaust opening designs for an empty room, cases W30 and C30, were compared. Next, the effect of flow obstacles is examined using room models of wall-return and locally balanced clean rooms, and the best ventilation efficiency case for wall-return clean room and the worst for locally balanced clean room were evaluated. Finally, the effect of changing the position of injection of particles was studied using a typical case, in which an obstacle was placed between the supply jets, for each type.

5. RESULTS AND DISCUSSION

• Verification of predicted airflow patterns

The airflow patterns are depicted in terms of velocity vectors. The predicted velocity vectors in the middle section (Y = 4.9 m) were compared with the experimental data of Murakami et al. [1], as shown in Figs. 5, 6, and 7 respectively. The predicted and measured velocity vectors exhibit the same trend of flow patterns, as discussed here.

In Fig. 5, the rising airstreams between the two supply air openings cannot reach the ceiling and become stationary between one half to one-third of the room height. In Fig. 6, the rising airstreams along the wall and between the two supply air openings reach the ceiling. In Fig. 7, the supply jet in the center of the room diffuses sideways and forms recirculating flows next to the obstacle, after the jet strikes the top of the obstacle.

Although the measured velocity vector at each grid cell is not as precise as that obtained from numerical simulation, the predicted airflow patterns are quite similar to those obtained from experiment. Figs. 8 to 10 quantitatively compare the velocity magnitudes at the working level (the third row) and the ceiling level (the sixth row) for cases 1 to 3. The velocity vectors (Figs. 5 to 7) are divided into six rows (from z1 near the floor, to z6 close to the ceiling) and 11 columns (from x1 near the wall, to x11 in the center of the room). In Figs. 8(a) and 8(b), the predicted velocity magnitude in the downward flow region (x10 and x11) is lower than the measured velocity. However, the predicted velocity magnitudes in other positions were close to the measured values. Figs. 9 and 10 show good agreement between the predicted and measured velocity magnitudes in the ceiling level (sixth row) and a predicted velocity magnitude that is lower than the measured value in the working level (third row). These discrepancies may be due to the over-predicting the kinetic energy of turbulence of the k-ε turbulence model, resulting in under-predicted velocity magnitudes in the jet and recirculation regions. In general, the predicted velocity magnitudes are quite close to the measured ones.

| Table 3: Specifications of analyzed cases |

<table>
<thead>
<tr>
<th>Item</th>
<th>Case</th>
<th>Particle starting position</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exhaust openings</td>
<td>W30</td>
<td>Line A ~ F (598 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C30</td>
<td>Line A ~ F (586 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>W30</td>
<td>Line A ~ F (598 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>W31</td>
<td>Line A ~ F (598 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>W32</td>
<td>Line A ~ F (598 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>W33</td>
<td>Line A ~ F (598 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C30</td>
<td>Line A ~ F (586 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C31</td>
<td>Line A ~ F (587 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C32</td>
<td>Line A ~ F (586 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C33</td>
<td>Line A ~ F (586 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>W31</td>
<td>Line A ~ F (598 particles tracked)</td>
</tr>
<tr>
<td>Flow obstacle</td>
<td></td>
<td>Line A ~ F (587 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C31</td>
<td>Line A ~ F (587 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C32</td>
<td>Line A ~ F (586 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>C33</td>
<td>Line A ~ F (586 particles tracked)</td>
</tr>
<tr>
<td></td>
<td>W31</td>
<td>Line A ~ F (598 particles tracked)</td>
</tr>
<tr>
<td>Injection position</td>
<td>W32</td>
<td>A(100)/B(98)/C(100)/D(100)/E(100)/F(100)</td>
</tr>
<tr>
<td></td>
<td>C32</td>
<td>A(90)/B(100)/C(99)/D(100)/E(97)/F(100)</td>
</tr>
</tbody>
</table>

Notes:
1. The complete case code is case-xxx in which the first character indicates room type (C = Locally balanced and W = Wall-return), the second indicates supply air velocity (3 = 0.35 ms⁻¹), and the third indicates the position of an obstacle (0 = no obstacles, 1 = an obstacle in the center of the room, 2 = an obstacle between the center of the room and the wall, and 3 = an obstacle near the wall).
2. In Cartesian coordinates (x, y, z), the injection positions of the particles are, A = (0.2~7.6,3.9,0.8), B = (0.2~7.6,3.9,1.5), C = (0.2~7.6,4.5,0.8), D = (0.2~7.6,4.5,1.5), E = (0.2~7.6,2.7,0.8), and F = (0.2~7.6,2.7,1.5), in units of meters.
Comparison of the number of particles released

From 600 to 600,000 particles with diameters of 0.1 µm are released to determine whether the number of particles injected into the clean rooms significantly impacts the results of the numerical simulation. As shown in Table 4, for cases W30 and C30, when 0.1 µm particles are released from 600 to 600,000, the number of particles tracked is increased by about 1,000 times, and so is the number of particles escaped. Accordingly, the number of particles injected into a clean room has only a minor effect on the results of the numerical simulation. To reduce the required computational effort in terms of CPU time and memory, therefore, six hundred particles are injected into the clean room in this study to simulate and analyze their trajectories.

Fig. 5: Velocity vectors in an empty wall-return clean room

Fig. 6: Velocity vectors in an empty locally balanced clean room

Fig. 7: Velocity vectors in a locally balanced clean room with an obstacle
Fig. 8: Velocity magnitude in an empty wall-return clean room

Fig. 9: Velocity magnitude in an empty locally balanced clean room
Fig. 10: Velocity magnitude in an empty locally balanced clean room with an obstacle

Table 4: Particles released in an empty clean room for cases W30 and C30

<table>
<thead>
<tr>
<th>Fates of 0.1 µm particles</th>
<th>Case W30</th>
<th>Case C30</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Number of particles</td>
<td>Times</td>
</tr>
<tr>
<td>Released</td>
<td>600</td>
<td>600,000</td>
</tr>
<tr>
<td>Tracked</td>
<td>598</td>
<td>597,973</td>
</tr>
<tr>
<td>Escaped</td>
<td>110</td>
<td>107,229</td>
</tr>
</tbody>
</table>

- Effect of changing layout of exhaust openings

For a wall-return type clean room (Fig. 11(a)), the rising streams become stationary at about half of the height of the room above the floor. In a locally balanced type clean room (Fig. 11(b)), the rising streams move along the wall or between the two closest jets and reach the ceiling. The general airflow patterns of these two types with supply openings in the ceiling resemble each other, and include one supply jet and the rising stream around it.

Fig. 12 shows the path lines of velocity vectors at the diagonal section of supply-exhaust openings in a locally balanced type clean room. No supply streams directly return to exhaust air openings from supply air openings and a “short circulation” does not occur between the supply and the exhaust flows.

The R value, which is the ratio of number of particle escaped (refer Table 4) to number of particle tracked from the clean room, is applied to quantify the extract particle performance of the clean rooms. Fig. 13 shows the fates of particles in cases W30 and C30. Fewer 0.1 µm particles (R = 18.4%) escaped from exhaust openings in case W30 than those in case C30 (R = 38.9%), and so is
for 0.5 µm, 1.0 µm and big particles, such as 5 µm, and 1.0 µm. Almost no particle with diameter greater than 50 and 100 µm escapes from exhaust openings in both cases. More escaped particles correspond to more effective ventilation; therefore, the effectiveness of the ventilation in case C30 is greater than that in case W30. Whatever the size of particles is, a locally balanced type room has more escaping particles than a wall-return type room.

• Effect of an obstacle

In the case W31 (Fig. 14(a)), in which an obstacle is located under the supply jets in the center of room, streams rise along the wall toward the ceiling, other streams slant toward the exhaust openings after the supply jets collide with the obstacle, and other streams between the supply jets fall from the ceiling toward the floor. Fig. 14(b) illustrates case W32, in which an obstacle is placed between two supply jets. In case W32, the downward streams move toward the obstacle and the streams that rise along the wall reach the ceiling. Fig. 14(c) shows that case W33 is similar to case W30 except in that the rising streams between the supply jet and the wall reach the ceiling because of the presence of an obstacle near the wall in the former case.

Fig. 15 shows the effect of changing obstacle locations in the wall-return clean room in case W30 (R = 18.4% for particles with 0.1 µm diameter, R = 15.2% for particles with 0.5 µm diameter, and R = 21.6% for particles with 1.0 µm diameter). In case W31 most particles (R = 24.9% for 0.1 µm particles, R = 25.8% for 0.5 µm particles, and R = 24.9% for 1.0 µm particles) escaped from the exhaust openings and in the other cases, W32 and W33, which differ only slightly, fewer particles escaped. The supply jet and the rising streams around the jet limit the diffusion of particles and the falling streams dominate the particles that move toward the exhaust openings. Consequently, the efficiency of ventilation is slightly reduced when an obstacle is placed near the wall, but is increased when an obstacle is placed in the center of the room, under the supply openings.

For a locally balanced clean room, Fig. 16(a) shows the flow fields in case C31, when a box-shaped obstacle is placed in the center of room just under the supply openings. The supply jet diffuses sideways after colliding with the obstacle, forming large recirculating flows beside the obstacle. In case C32 (Fig. 16(b)), in which an obstacle is placed between supply jets, strong slanting rising streams are formed and move toward the ceiling along the wall and between supply jets. The motion of particles generated in the room is dominated by the motion of these strong rising streams toward the ceiling, and are exhausted effectively. As shown in Fig. 16(c), case C33, in which an obstacle is located near the wall beside the supply jets, does not differ much from case C32.

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![Fig. 11: Distribution of velocity vectors in (a) case W30 and (b) case C30](image-url)
Fig. 17 shows the effect of an obstacle in terms of the number of escaped particles for the locally balanced clean room. In case C31, in which an obstacle is placed under the supply jets, fewer particles (R = 36.3% for 0.5 µm particles and R = 35.1% for 1.0 µm particles) escape due to the formation of recirculation flows. In cases C32 and C33, in which an obstacle is beside the supply jets, more particles (R = 46.8% and 46.6% for 0.5 µm particles, and R = 44.0% and 42.8% for 1.0 µm particles, respectively) escape than in case 31, because strong rising streams move toward the ceiling. Thus, in a locally balanced model, placing an obstacle just under the supply jets results in lower particle exhaust efficiency.

When an obstacle is placed under the supply openings in the center of the room, the number of particles that escape from the wall-return clean room increases, but that from the locally balanced clean room decreases. When the obstacle is located between the supply jets, the number of particles that escape from a locally balanced clean room increases, but that from a wall-return clean room declines. When the obstacle is placed near the wall, the number of particles that escaped from the locally balanced clean room increases, but that from a wall-return clean room declines.

Case W31 - the most effective ventilation case for wall-return clean room is compared with case C31 - the least effective ventilation case for locally balanced clean room. The number of 0.1 µm escaped particles in case W31 (R = 24.9%) is much smaller than that in case C31 (R = 38.8%), implying that the wall-return clean room exhausts particles less efficiently than the locally balanced clean room. Similarly, for 0.5 µm and 1.0 µm particles, case C31 has better ventilation efficiency than case W31. Thus, in whichever position an obstacle is located, the wall-return model has the worst ventilation efficiency.
Effect of changing position of particle injections

An obstacle located between the supply jets are considered here. In case W32 (shown in Fig. 18(a)), many particles (R = 29.0% with 0.1 μm diameter and R = 36.0% with 0.5 μm diameter) escape when released from line E and fewer particles (R = 10.2% with 0.1 μm and R = 10.2% with 1.0 μm diameter) are exhausted when injected from line B. Fig. 18(b) presents the number of particles that escaped in case C32. More particles (R = 58.0% for 0.1 μm particles, 65.0% for 0.5 μm particles, and 59.0% for 1.0 μm particles, respectively) injected from line F are effectively exhausted and fewer particles (R = 28.9% for 0.1 μm particles, 25.6% for 0.5 μm particles, and 38.9% for 1.0 μm particles, respectively) injected from line A escape than those injected from any other lines. Moreover, particles released from lines B, D, and F are much more easily exhausted than those introduced from lines A, C, and E. Therefore, more particles escape when more are introduced nearer the exhaust openings.

Fig. 14: Velocity vectors in case W31, W32, and W33
Fig. 15: R in cases W30 to W33

Fig. 17: R in cases C30 to C33

Fig. 16: Velocity vectors in case C31, C32 and C33
6. CONCLUSIONS

Flow fields and particle trajectories in types of wall-return and locally balanced of turbulent clean rooms were studied by numerical simulation based on the stochastic tracking model. This work clarifies the effect of exhaust opening layout, flow obstacles, and particle starting positions on the ventilation efficiency of clean rooms. Based on the results and discussion, the following conclusions are drawn:
The layout of exhaust openings greatly influences particle trajectories in an empty room. More particles escaped from a locally balanced type clean room than from a wall-return type clean room.

Obstacles display significant influences on the flow fields and particle trajectories. Even with an obstacle, more particles escaped from a locally balanced type clean room than from a wall-return type clean room.

Particles generated close to an exhaust opening are more effectively exhausted from a room. Consequently, arranging contaminant sources near exhaust openings is recommended to facilitate contaminant extraction. Larger particles (more than 50 µm diameter) are hard to be exhausted.

It is confirmed that setting the exhaust air openings in the ceiling does not reduce the efficiency of ventilation for exhausting contaminants and that a short circulation between the supply and the exhaust flows does not occur when the inlet and outlet are set close together.

REFERENCES


