This is the published version of a paper published in *Indoor Air*.

Citation for the original published paper (version of record):

Lim, E., Sandberg, M., Ito, K. (2021) Returning characteristics of pollutants for a local domain in the presence of returning and recirculating airflow in indoor environments. *Indoor Air*, 31(4): 1267-1280  
https://doi.org/10.1111/ina.12803

Access to the published version may require subscription.

N.B. When citing this work, cite the original published paper.

Permanent link to this version:

http://urn.kb.se/resolve?urn=urn:nbn:se:hig:diva-35341
.Returning characteristics of pollutants for a local domain in the presence of returning and recirculating airflow in indoor environments

Eunsu Lim1 | Mats Sandberg2 | Kazuhide Ito3

1Faculty of Science and Engineering, Toyo University, Saitama, Japan
2University of Gävle, Gävle, Sweden
3Faculty of Engineering Sciences, Kyushu University, Kasuga, Japan

Abstract

Heating, ventilating, and air-conditioning (HVAC) systems usually supply air, which is a mixture of fresh air from the outdoor environment, and return air from rooms via the ventilation ductwork. This air reduces the heat load and cost impact of air conditioning using outdoor air. This recirculation of room air in air-conditioning systems is reasonable in terms of energy saving; however, the deterioration of air quality might be a concern because of the recirculation of contaminated room air. Here, we numerically investigate the effect of pollutant recirculation/return on the formation of concentration distributions of local pollutants in indoor environments when the mixing ratio of recirculated air in the HVAC system changes. We discuss the detailed structure of the formation mechanism of local pollutant concentration distributions using various indices for indoor ventilation efficiency in simplified room models. Among the indices, visitation frequency and net escape probability are the ones that directly assist in evaluating the recirculation/return characteristics of indoor pollutants. As a result, when the proportion of air that is recirculated becomes large, the number of pollutants returning to a target local domain, the visitation frequency, increases exponentially, and the net escape probability—which directly expresses the probability of pollutant discharged from the target domain—is close to zero.

KEYWORDS
local purging flow rate, net escape probability, net escape velocity, return air, ventilation efficiency, visitation frequency

INTRODUCTION

The rate of outdoor airflow required to maintain a certain level of indoor air quality is generally determined using two factors: the indoor threshold concentration, which is used as an index/guideline for ventilation design, and the amount of pollutants generated in the target indoor space. In general, to maintain indoor air quality, outdoor air will be introduced and will be mixed with temperature-controlled “return air” which is air returned to the air-conditioning system from the room via the ductwork.1,2 Most heating, ventilating, and air-conditioning (HVAC) systems at least partially recirculate air to increase the cooling and heating capacity to conditioned spaces while avoiding the energy and first cost impacts of conditioning outdoor air.3,4 Therefore, in such HVAC systems, if the pollutants generated in the room cannot be removed properly by filtration or removal devices in the HVAC systems, there is a possibility that some of the pollutants in the air-conditioned
air will be repeatedly mixed with the air in the supply inlet. Previous studies have reported calculation methods for pollutant concentration characteristics in ventilation systems with return air.\textsuperscript{5-9} Li et al. proposed an algorithm for calculating indoor contaminant distribution in complex different ventilation systems with recirculation.\textsuperscript{5} Hiyama et al. proposed a calculation method using the concentration response factor as the time serial to calculate the three-dimensional concentration transport in an air-conditioned room.\textsuperscript{6} Li et al. proposed a versatile numerical method to determine the pollutant distribution of a ventilation system with recirculation at steady state based on ventilation systems with supply air and a contaminant source.\textsuperscript{7} Waters and Simons studied the effect of ventilation effectiveness parameters, for example, the mean age of air and contaminant removal effectiveness, on the recirculation of air in a room with recirculation by the multizonal method.\textsuperscript{8} Shao and Li et al. derived a concise linear expression to establish the correlation between steady contaminant distribution and independent supply air and contaminant sources with air recirculation and evaluated the effects of supply air and contaminant sources based on an index of accessibility proposed by them.\textsuperscript{9}

When considering indoor ventilation efficiency, particularly in local domains such as an occupied zone or a breathing zone, the presence of return air means that old-aged air or pollutants are recirculated.\textsuperscript{10} A part of the conditioned air that is exhausted out of the room through the exhaust outlets (containing a part of the pollutants generated indoors) is exhausted outside the HVAC system, and the remaining part, a certain percentage, is returned to the room as "return air." When the air-conditioning airflow rate and the exhaust airflow rate are constant, that is, mass-balanced, and therefore, the flow field can be assumed to be in a steady state, the pollutants contained in the air-conditioning airflow from the supply inlet must be uniformly distributed in the room according to the law of mass conservation and linear superposition. When the proportion of return air is zero, i.e., all fresh air/outdoor air is being conditioned, a non-uniform concentration distribution is formed by pollutants that are heterogeneously distributed, and a non-uniform flow pattern is formed in the room. In case of a steady-state flow field, the pollutant transport equation becomes a linear equation, and then, the pollutant concentration distributions containing multiple pollutant sources can be calculated by linear superposition. In other words, the indoor pollutant concentration field in the presence of "return air" in the HVAC system is determined by the uniform pollutant concentration owing to the existence of return air and the non-uniform concentration distribution formed in the room by heterogeneous pollutant sources and the non-linear flow pattern produced when there is zero return air. When considering indoor ventilation efficiency, especially when discussing the formation of a concentration field focusing on the recirculation of pollutants by an HVAC system with return air, comprehensive evaluations of the formation structure of the pollutant concentration distribution and transport mechanism are essential because the proportion of return air has a dominant impact on indoor pollutant concentration levels.

Studies on indoor ventilation efficiency have been conducted for many decades, beginning with that by Sandberg et al, and various fruitful indices for evaluating ventilation efficiency in indoor environments have been proposed and used in the practical ventilation designs.\textsuperscript{11-13} For example, local purging flow rate (L-PFR), visitation frequency (VF), local purging flow rate (L-PFR), net escape velocity (NEV), and net escape probability (NEP), allow for quantitative evaluation of an exponential increase of pollutant concentration at a local domain or a local point that correlates with an increase in the proportion of air recirculated through the HVAC system.

### Practical implications

- Returning of pollutants at a local point in a room results from circulation of indoor airflow and returning air from the room via the ventilation ductwork of the HVAC system. Assuming a steady-state flow field, the pollutant transport equation becomes a linear system, and the pollutant concentration value of the local point is determined by superimposing the concentration distributions.

- The ventilation efficiency indices, for example, visitation frequency (VF), local purging flow rate (L-PFR), net escape velocity (NEV), and net escape probability (NEP), are indicators that can evaluate ventilation efficiency at a local point in the room. Using these indices, it is possible to obtain the distribution of ventilation efficiency at a local area in the room.\textsuperscript{18-24} We have proposed new ventilation efficiency concepts, NEV and NEP, which are indicators that can evaluate ventilation efficiency at a local point in the room. Using these indices, it is possible to obtain the distribution of ventilation efficiency at a local point in the room.\textsuperscript{25-28} In this context, Lim et al.\textsuperscript{25,26} and Chung et al.\textsuperscript{27} proposed using more constructive concepts, such as net escape velocity (NEV) and net escape probability (NEP), as indices to define the ventilation efficiency at a certain point in an enclosed space and investigated the basic properties of pollutant mixture via numerical analysis. Lim et al. also reported the results of evaluating the concentration field formation in indoor air using the concept of NEV considering air-phase chemical reactions.\textsuperscript{28}

Against this background, the purpose of this study is to investigate the applicability of ventilation efficiency indices, for example, NEV, NEP, VF, and L-PFR, to a heterogeneous pollutant mixing field where "return air" from an HVAC system exists and to discuss quantitatively the formation mechanism of pollutant concentration distributions in the local domain in indoor environments. We discuss the impact of the proportion of return air on indoor ventilation performance based on computational fluid dynamics (CFD) analysis, the prediction accuracy of which is discussed and verified.

### 2 | VENTILATION EFFICIENCY INDICES

To understand the ventilation process, it is necessary to elucidate the transport and formation mechanism of pollutant concentrations by distinguishing between the action caused by the two populations: a
population ventilating the space, that is, leaving the space and never returning to the domain, and a population returning (recirculating) to the domain, thereby spreading the contaminants. The following information is essential: (a) the number of times the pollutants generated in the target local domain return to the domain after leaving. This is quantified by the visitation frequency (VF), which is related to the population of air/contaminant returning, and the net escape probability (NEP), which denotes the probability that the contaminant is transported directly to the exhaust outlet without re-visiting the target point along the flow path. They reflect the state of the flow, for example, whether the flow is unidirectional or not. (b) The amount of fresh air used to dilute and discharge the pollutants in the local domain. This is quantified by the local purging flow rate (L-PFR) and the corresponding velocity, which is the net escaper velocity (NEV). These concepts are related to the population leaving and never returning. In this study, we performed CFD analysis with Reynolds averaged Navier-Stokes (RANS) modeling and solved the transport equation of an ensemble-averaged scalar quantity numerically.

\[
\frac{\partial \phi_p}{\partial t} + \nabla \cdot (\rho U \phi_p) = \nabla \cdot (D_{\text{eff}} \nabla \phi_p) + S_p
\]

\[
D_{\text{eff}} = D_p + \frac{v_t}{Sc_t}
\]

Here, \(\phi_p\) represents the volume-averaged pollutant concentration (kg m\(^{-3}\)) in the target control volume (CV), \(U_j\) represents the ensemble-averaged scalar quantity of advection velocity in the \(j\) direction, and \(S_p\) represents the generation rate of the pollutant. The effective diffusivity constant \(D_{\text{eff}}\) is given by Equation (2), where \(Sc_t\) represents the turbulent Schmidt number, \(v_t\) is the turbulent viscosity, and \(D_p\) represents the molecular diffusivity.

### 2.2 Net escape probability

According to the context of NEV, we also defined two probabilistic indices for determining the recirculation of pollutants, that is, the returning probability \(\alpha\), which represents the probability that a pollutant returns to the target point (here, CV) as a local domain under the condition that a contaminant is generated in the CV, and the NEP, which denotes the probability that the contaminant is transported directly to the exhaust outlet without re-visiting the target CV along the flow path. By definition, the sum of NEP and \(\alpha\) is equal to 1.0. They can be defined as follows:

\[
\alpha = \frac{F_{\text{pf}} A_{\text{if}}}{F_{\text{pf}} A_{\text{if}} + q V_{CV}} = \frac{F_{\text{pf}} A_{\text{if}} - F_{\text{pf}} A_{\text{af}}}{F_{\text{pf}} A_{\text{if}}}
\]

\[
\text{NEP} = 1 - \alpha = \frac{q V_{CV}}{F_{\text{pf}} A_{\text{if}} + q V_{CV}} = \frac{F_{\text{pf}} A_{\text{if}} - F_{\text{pf}} A_{\text{af}}}{F_{\text{pf}} A_{\text{if}}}
\]

Here, \(F_{\text{pf}}\) and \(A_{\text{if}}\) denote the total outflow flux (convective and diffusive flux) from the target CV (kg m\(^{-2}\) s\(^{-1}\)) and the area through the outflow flux in the target CV (m\(^2\)), respectively. \(q V_{CV}\) is the total amount of contaminant generated in the CV (kg s\(^{-1}\)).

### 2.3 Local purging flow rate

Local purging flow rate (L-PFR) is an index of ventilation efficiency in a local domain, such as an occupied zone or confined space in a room. It was originally defined as the effective airflow rate to remove/ purge contaminants from the local domain. In addition to the original definition by Sandberg, Kato et al. proposed an additional definition based on VF and the average-staying time at target local domain (\(T_p\)).

\[
L - \text{PFR} = \frac{q_p}{C_p} = \frac{V_p}{VF \cdot T_p}
\]

where L-PFR represents the local purging flow rate (m\(^3\) s\(^{-1}\)), \(V_p\) is the volume of the local domain \(p\) (m\(^3\)), VF is the visitation frequency of fluid parcels or pollutant (\(\cdot\)), \(T_p\) is the average-staying time of parcels/
pollutant in the local domain (s one staying \( ^{\pm} \), \( q_p \) is the parcels/pollutant generation rate per unit time (kg s\(^{-3}\)), and \( C_q \) represents the local domain-averaged concentration (kg m\(^{-3}\)).

### 2.4 Visitation Frequency

\( VF \) represents the number of times a particle enters and passes through the local domain. \( VF = 1 \) means that after being generated, a particle appears in the local domain only once, and then, after leaving the local domain, never returns.\(^{12} \) \( VF = 2 \) means a particle enters the local domain for the first time, is transported to the outside, and then returns again to the local domain because of the recirculating flow only one time. \( VF \) can be calculated using Equation (9). 

\[
VF = 1 + \frac{\Delta q_p}{q_p} = 1 + \frac{J_p}{M_p} 
\]

where \( J_p \) is the amount of parcels/pollutants, visiting (returning to) the target local domain per unit time (Particle s\(^{-1}\)), \( M_p \) is the amount of parcels/pollutants generated in the local domain (Particle s\(^{-1}\)), \( \Delta q_p \) is the inflow rate of parcels/pollutants into the local domain per unit time (kg s\(^{-3}\)), and \( q_p \) denotes the particle generation rate per unit time (kg s\(^{-3}\)).

### 2.5 Normalized concentration at local domain

The average pollutant concentration can provide fundamental and essential information regarding the ventilation efficiency, and then, the normalized pollutant concentration at a local domain becomes dimensionless with the representative concentration, that is, perfect mixing concentration defined simply by pollutant generation rate and ventilation rate, usually adopted for ventilation design.

\[
C_n = \frac{C_p}{C_e} = \frac{C_p \cdot Q_m}{q_p \cdot L - \frac{Q_m}{PPR}} 
\]

Here, \( C_p \) is the mean concentration at local domain \( P \) (kg m\(^{-3}\)), \( C_e \) is the mean concentration at the exhaust outlet (kg m\(^{-3}\)), \( C_n \) is the normalized concentration at local domain \( P \), and \( Q_m \) is the airflow rate from the supply inlet opening (kg m\(^{-3}\)).

In this study, these ventilation efficiency indices are numerically analyzed in the simplified room models described next.

### 3 NUMERICAL METHODS

We used two simplified room cases forming a large internal circulating flow in spaces with different aspect ratios and different air mixing conditions: a simple square room model (two-dimensional analysis) and a simple rectangular room model (three-dimensional analysis), which is the IEA Annex benchmark test case with detailed measurement data. The geometries of the two room model cases were set in order to investigate the effect of the differences in the aspect ratio of the room on the change in ventilation efficiency distribution at local domains. The temperature conditions were assumed to be isothermal. To avoid discussing the dependence of the influence of Reynolds number (Re) of the flow field, a constant and fixed supply airflow rate through the supply inlet of the HVAC system was assumed, and the proportion of return air in the supply airflow rate was gradually increased from zero (0%) to 90%. The 0% return air condition represents an all fresh air supply. In the case of 100% return air, fresh outdoor air is not supplied in the room; consequently, the pollutant concentration in the room becomes infinity because the pollutants generated in the enclosed space are not diluted. To analyze the ventilation efficiency, room models were divided into upper and lower parts, and the lower part of each room model was assumed to be a target local domain, for example, occupied space. We assumed the area below \( y = 5.64L_p \) in the square case and \( y = 0.5H \) in the rectangular case as local domains for including strong airflow along the wall surface and the stagnant area. These values represent the height through the area where the velocity is almost zero in the center of recirculating flow.

The amount of supply air is set constantly, and when there was return air in the HVAC system, the ratio of outdoor fresh air included in the supply air was gradually changed, and the scalar (pollutant) concentration included in the supply air was determined by the ratio of return air. The returning pollutant rate was set by the ratio of return air. If the ratio of return air to supply air in the room is assumed to be \( \gamma \), the ratio of outdoor air to supply air will be \( 1-\gamma \). Here, the returning pollutant rate included in supply air \( q' \) (kg s\(^{-3}\)) is written as follows:

\[
q' = q \cdot \frac{r}{1-\gamma} 
\]

where \( q \) represents the pollutant generation rate per unit time in a target local domain (kg s\(^{-3}\)) and \( q' \) represents the returning pollutant rate included in supply air per unit time (kg s\(^{-3}\)).

Assuming \( n \) is infinite, \( q' \) is given by the following equation via an infinite series.

\[
q' = q \cdot \frac{1}{1-\gamma} 
\]

If the returning pollutant rate \( q' \) is assumed to represent a pollutant generation source in the target room, the total pollutant generation rate will be \( q + q' \) as follows:

\[
q + q' = q \cdot \frac{1}{1-\gamma} 
\]

Flow fields were analyzed using the standard \( k-c \) model with a generalized log law for wall function. In the scalar advection and
diffusion analysis performed in this study, passive contaminants, for example, hypothetical pollutants with exactly the same physical properties as air, were assumed. To obtain the L-PFR and VF distributions of the target area, a pollutant was generated constantly in the target local domain for advection and diffusion analysis. On the other hand, to obtain the NEV and NEP distributions of a target area or a room, the pollutant was set to be generated in each CV for NEV and NEP calculations, and the NEV and NEP distributions of the target area or the room were calculated by overlapping each value for each CV. This means that the NEV distribution was obtained by overlapping the results of separate, independent NEV analyses. Scalar advection and diffusion field analyses were performed using the same number of calculations with the number of total CVs of the target area in the case of NEV and NEP analyses. For example, in the case of a target area with 1000 total CVs, 1000 analyzed cases were required to obtain the NEV and NEP distributions of the target area.\textsuperscript{25-27} Figure 1 shows the schematic diagram of the room model and HVAC system with return air.

3.1 Numerical analysis conditions for the square room model (two-dimensional case)

An enclosed square room model with a spatial size of 10 $L_{in} \times 10 L_{in}$ (where $L_{in}$ is the size of the supply inlet opening; 4 meshes in this study) is illustrated in Figure 2. The room was divided into equal intervals of 40 $\times$ 40 CVs at the same size as the 1/4 width of the supply inlet $L_{in}$. This grid design was determined as a result of a careful grid independence check. The numerical calculation conditions and boundary conditions for the square room model are summarized in Table 1. The supply inlet openings were located on the wall near the floor, the non-dimensional air velocity $U_{in} = 1.0$ (-) was set, and the turbulence intensity was set to 30%.

3.2 Numerical analysis conditions for the rectangular room model (three-dimensional case)

The IEA Annex 20 test case, which was used as the rectangular room model, is shown in Figure 3.\textsuperscript{30-36} The numerical and boundary conditions are summarized in Table 2. The supply inlet opening was located on the wall near the ceiling, and the supply velocity $U_{in} = 0.455$ m s$^{-1}$ was adopted. The turbulence intensity was set to 4% in accordance with experimental results. We conducted a preliminary CFD analysis to determine the independent grid design and numerical analysis conditions for reproducing experimental results with sufficient accuracy.

3.3 Validation and quality control of CFD

In the CFD analyses structured grid designs were adopted to ensure the prediction accuracy of the flow fields. Grid dependence checks were carefully conducted prior to the final calculation, and the cell quality, for example, expansion ratio and aspect ratio in the rectangular room model (3D), was carefully controlled. The total quality of the CFD simulations was carefully controlled in accordance with the guidelines and the previous reported benchmark test results.\textsuperscript{30}

| TABLE 1  | Numerical and boundary conditions for square room model (2D Case) |
|-----------------|-----------------------------------------------|
| Geometry: 10 $L_{in} \times 10 L_{in}$ ($L_{in}$: size of the inlet opening) | Meshes: Structured grid with total of 1600 (=40 (X)$\times$40 (Y)) meshes |
| Turbulence model: Standard $k$-$\varepsilon$ model | Algorithm: SIMPLE |
| Discretization scheme: QUICK scheme for the advection term | Outflow boundary: \( \nabla U_{out} = \nabla k_{out} = \nabla \varepsilon_{out} = 0 \) |
| Inflow boundary: $U_{in} = 1.0$ [-], $k_{in} = 3/2 \times (U_{in} \times 0.3)^2$ [m$^2$/s$^2$], $\varepsilon_{in} = C_{\mu} \times k^{3/4} / L_{in}$ [m$^2$/s$^3$] | Wall treatment: (Velocity) generalized log law, (Scalar) $\nabla U_{in} = 0$ |

\textsuperscript{25-27} Figure 1 shows the schematic diagram of the room model and HVAC system with return air.
4 | RESULTS

4.1 | Airflow distribution

The airflow distributions for the square room model (2D case) are shown in Figure 4. Figure 4A shows the streamline for the square room case, and Figure 4B shows the distribution of air velocity normalized by the velocity of the supply air for the square room case. The airflow supplied along the floor formed a large circulation flow in a counterclockwise direction. Relatively, high air velocity values were observed in the vicinity of the floor and on the wall opposite the supply inlet. In contrast, a stagnant flow area formed in the center of the room model.

The airflow distributions for the rectangular room model (3D case) used in the Annex benchmark test case are shown in Figure 5. Figure 5A shows the airflow vector distribution in the room, Figure 5B shows the airflow scalar distribution, and Figure 5C shows a comparison of the experimental and numerical results at two horizontal positions (y = H-h/2 and y = h/2) and two vertical positions (x = H and x=2H) at the center of the z-direction (z = 0.5H) in the rectangular room model (3D case). In Figure 5, the air velocity values were normalized by the supply air velocity (=0.455 m/s). The numerical results agreed with well the experimental results. The ventilation airflow was supplied along the ceiling and formed a large circulation flow in the clockwise direction. Relatively, high air velocity values were observed in the vicinity of the floor and on the wall opposite the supply inlet and the floor. In contrast, a stagnant flow was formed in the center area of the room model.

4.2 | Diffusion field results

4.2.1 | Pollutant concentration distributions

Figure 6 shows the pollutant concentration distributions normalized by the concentration at the exhaust outlet under the condition of 0% return air (all supplied air is fresh air). The pollutants were generated uniformly in the local domain (the zone lower than y = 5.6L in for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case). The maximum values of the pollutant concentration normalized by the concentration at the exhaust outlet were observed in the stagnant area, which is at the center area in the square room case and the vicinity of the left-side wall including the supply opening in the rectangular room case. The maximum values compared with the perfect mixing conditions in each case were 3.1 times and 2.2 times higher for the square room and rectangular room cases, respectively.

Figure 7 shows the distributions of the normalized concentration in the local domain p using the concentration at the exhaust opening, C_p/C_e, as function of the percentage of return air (%). The horizontal axis depicts the percentage of return air (%) from the supply inlet, and the vertical axis indicates the mean normalized concentration to the local domain (the areas below the y = 5.6L_in line for the square room case and below the y = 0.5H section for the rectangular room case).

The normalized concentration of pollutant increased exponentially when the return air ratio exceeds 50%. The normalized concentrations of pollutant under 80% return air condition compared with 0% return air condition were 3.1 times and 3.5 times higher than those for the square room and rectangular room cases, respectively.

Figure 8 shows the average pollutant concentration distributions as function of the percentage of return air (%). Figure 8A denotes the changes in the average concentration of the whole line (y = 5.6L_in for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case) according to the change in the percentage...
of return air (%). The horizontal axis shows the percentage of return air (%) from the supply inlet, and the vertical axis indicates the mean concentration (averaged value along the $y = 5.6L_m$ line for the square room case and $y = 0.5H$ in the $z = 0.5H$ section for the rectangular room case), which was normalized using the concentration at the exhaust outlet in the 0% return air condition. Figure 8B and C shows the average concentration distributions along the line $y = 5.6L_m$ for the square room case and $y = 0.5H$ in the $z = 0.5H$ section for the rectangular room case according to the change in the percentage of return air (%).
When the percentage of return air exceeded 50%, the pollutant concentration increased rapidly. When there was 0% return air, that is, all fresh air, the average concentrations were 0.5 in the square room case and 0.4 in the rectangular room case (Figure 8A). These results indicate that indoor air environments are 50% and 60% better, respectively, compared with those of rooms assumed to have the perfect mixing condition.

### 4.2.2 Ventilation efficiency distributions

Figure 9 shows the normalized VF distributions using the VF under the condition of 0% return air to the local domain (the areas below the $y = 5.6L_{in}$ line for the square room case and below the $y = 0.5H$ section for the rectangular room case) as function of the percentage of return air (%). The horizontal axis depicts the percentage of return air (%) from the supply inlet, and the vertical axis indicates the mean VF to the local area (the areas below the $y = 5.6L_{in}$ line for the square room case and below the $y = 0.5H$ section for the rectangular room case).

The VF values gradually increased as the proportion of return air increased and exhibited a trend similar to the results of average pollutant concentration shown in Figure 7A. When the return air exceeded 50% in both the square room and rectangular room cases, the VF values exponentially increased. In the 0% return air condition, the VF values before normalization were 15.0 in the square room case and 3.7 in the rectangular room case. In the 80% return air condition, the values were 2.54 in the square room case and 3.36 in the rectangular room case. It means that the number of times a particle enters and passes through the local domain after leaving it increased 2.54 times and 3.36 times in the square room case and in the rectangular room case, respectively.

Figure 10 shows the normalized $L$-PFR to the local domain (the areas below the $y = 5.6L_{in}$ line for the square room case and below the $y = 0.5H$ section for the rectangular room case) as function of the percentage of return air (%).
the y = 0.5H section for the rectangular room case) distributions as function of the percentage of return air (%). Figure 10A denotes the normalized L-PFR using the amount of supply air, L-PFR/Qin. Figure 10B denotes the normalized L-PFR using the amount of outdoor air included in supply air, that is, the ventilation airflow rate, L-PFR/Qsa. The horizontal axis shows the percentage of return air (%) from the supply inlet, and the vertical axis indicates the normalized L-PFR.

When the percentage of return air is increased, the L-PFR/Qin gradually decreased and approached zero, because the VF represents returning time of the pollutant to the target local domain exponentially increased, as shown in Figure 9. The L-PFR/Qsa values in the 80% return air condition compared with those in the 0% return air condition decreased by 68% and 72% in the square room and rectangular room cases, respectively. On the other hand, the L-PFR/Qin gradually value increased and approached 1.0 as the percentage of return air increased because the pollutant concentration distributions were close to the perfect mixing condition. In the 0% return air condition, the normalized L-PFR/Qsa value was 0.52 in the square room case and 0.63 in the rectangular room case, as shown in Figure 9B. The L-PFR/Qin values in 80% return air condition compared to the 0% return air condition increased by 62.0% and 42.0% in the square room and rectangular room cases, respectively.

Figure 11 shows the distributions of normalized NEV using the supply air velocity (here, NEV) as function of the percentage of return air (%). Figure 11A denotes the change in the average NEV of the whole line (y = 5.6Lin for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case) according to the change in the percentage of return air (%). The horizontal axis shows the percentage of return air (%) from the supply inlet opening, and the vertical axis indicates the mean NEV (averaged value along with the y = 5.6Lin line for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case). Figure 11B and C shows the NEV (calculated at each CV) distributions along the line y = 5.6Lin for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case according to the change in the percentage of return air (%). Under the 0% return air condition, that is, all fresh air, NEV values were 0.22 and 0.5 in the square room and rectangular room cases, respectively (Figure 11A). NEV gradually decreased and approached zero as the ratio of return air increased. NEV values under the 80% return air condition decreased by approximately 90% in both the square room and rectangular room cases, which means that the ventilated speed of the pollutant became very slow.

Figure 12 shows the distributions of normalized NEV* using the supply air velocity (NEV*/U; here, NEV*) as function of the percentage of return air (%). Figure 12A denotes the changes in average NEV* of the whole line (y = 5.6Lin for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case) according to the changes in the percentage of return air (%). The horizontal axis shows the percentage of return air (%) from the supply inlet opening, and the vertical axis indicates the mean NEV* (averaged value along the y = 5.6Lin line for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case).
C shows the $NEV^*$ (calculated at each CV) distributions along the line $y = 5.6L_{in}$ for the square room case and $y = 0.5H$ in the $z = 0.5H$ section for the rectangular room case according to the changes in the percentage of return air (%). In the 0% return air condition, $NEV^*$ values representing the transportation speed of a pollutant at a CV by the effects of advection flux and diffusion flux of outflow flux on the target CV were 0.52 in the square room case and 0.34 in the rectangular room case (Figure 12A). The value of $NEV^*$ gradually decreased and approached a constant value as the ratio of return air increased. This is because the effect of diffusion on the transport of pollutants was reduced and the pollutants’ transportation was governed by advection flux. The $NEV^*$ values under the 80% return air condition decreased by approximately 30% and 55% in the square room and rectangular room cases, respectively.
Figure 13 shows the distributions for the normalized NEV*, NEV*/U_{in}, by advection velocity, NEV*/U, at each CV as function of the percentage of return air (%). Figure 13A shows the changes in the average normalized NEV*/U of the whole line (y = 5.6L_{in}) according to the changes in the percentage of return air (%). The horizontal axis denotes the percentage of return air (%) from the supply inlet opening, and the vertical axis indicates the mean normalized NEV*/U (averaged value along the y = 5.6L_{in} line for the square room case and the y = 0.5H in the z = 0.5H section for the rectangular room case). Figure 13 (b) and (c) shows the normalized NEV*/U (which is calculated at each CV) distributions along the line y = 5.6L_{in} for the square room case and the y = 0.5H in the z = 0.5H section for the rectangular room case according to the changes in the percentage of return air (%). In the 0% return air condition, the NEV*/U values representing the diffusion effect on the transport of pollutants at a CV were 2.3 in the square room case and 34.5 in the rectangular room case (Figure 13A). The value of NEV*/U represents the ratio of the integrated effect of advection and diffusion to the advection effect on the transport of pollutants. When the ratio of return air is increased, the pollutant concentration distributions are close to the distribution under the perfect mixing condition, and the diffusion effect will be close to zero. Therefore, the NEV*/U value gradually changed and approached 1.0 as the ratio of return air increased. The normalized NEV*/U values have their maximum values at the center (x = 4.6L_{in} in the square room case and x = 2.1H in the rectangular room case) of the recirculating flow (Figure 13B and C). The maximum values of NEV*/U under the 0% return air condition were 18.3 in the square room case and 49.0 in the rectangular room case. The values under the 80% return air condition decreased 52% in the square room case and 75% in the rectangular room case.

Figure 14 shows the NEP distributions as function of the percentage of return air (%). Figure 14A denotes the change in the average NEP of the whole line (y = 5.6L_{in} for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case) according to the changes in the percentage of return air (%). The horizontal axis shows the percentage of return air (%) from the supply inlet, and the vertical axis indicates the mean NEP (averaged value along the y = 5.6L_{in} line for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case). Figure 14B and C shows the NEP (calculated at each CV) distributions along the line (y = 5.6L_{in} for the square room case and y = 0.5H in the z = 0.5H section for the rectangular room case) according to the changes in the percentage of return air (%). In the 0% return air condition, the NEP value is 0.91 in the square room case and 0.73 in the rectangular room case (Figure 14A). The values of NEP gradually decreased as the ratio of return air increased. The values under the 80% return air condition decreased 43% in the square room case and 67% in the rectangular room case. This means that the probabilities of pollutant discharged from the rooms by the proportion of return air decreased 43% in the square room case and 67% in the rectangular room case.

5 | DISCUSSION

In this study, we considered the return air of an HVAC system and assumed that the pollutants contained in the air supplied by the supply inlet were not newly generated pollutants but just recirculated pollutants. The pollutants were generated uniformly in a local domain for VF and L-PFR analyses and were generated in each CV for NEV and NEP analyses. When the percentage of return air increased, the average pollutant concentration also increased. In particular, when the conditions exceeded 50% return air, the pollutant concentration increased rapidly. The value of NEP decreased and approached 1.0 as the ratio of return air increased. This means that the transport of pollutants is largely dominated by the convection effect as the ratio of return air increases.

Assuming that the target local domain is completely mixed, the average pollutant concentration in the local domain becomes 1.0 everywhere (here, the concentration is normalized by the concentration at the exhaust outlet under the 0% return air condition). For example, under the 20% return air condition, considering that the supply inlet airflow rate is constant, the average concentration at the local domain increases to 1.25 because the outdoor clean airflow rate decreases to 80% of that in the case of all fresh air. Similarly, the average concentration in the local domain increased to 2.0 for the 50% return air condition and 5.0 for the 80% return air condition. This is a universal trend of linear superposition of pollutant

![Figure 13](image-url)

![Figure 14](image-url)
concentration under steady-state flow conditions. According to this increase in the average concentration, the value of VF against this local domain also increased.

The PFR standardized by the supply airflow rate through the supply inlet, with the outdoor airflow rate included in the supply air, showed an inverse relationship with the average pollutant concentration standardized by the perfect mixing concentration at the exhaust outlet in the presence of return air. Under this condition, when the proportion of return air increased, the normalized PFR approached 1.0 because the average concentration at the local domain became similar to the perfect mixing condition and hence was close to 1.0. Because the pollutants in the return air supplied from the supply inlet are diffused uniformly in the room, the concentration gradient at each CV in the room is formed by the pollutant generation in the room. Therefore, the concentration gradient at each CV in the room did not change even if the percentage of return air changed. In other words, even when the ratio of return air changes, the value of diffusion flux at each CV, that is, \(-\Delta F_{\text{diff}}|_{\text{CV}}\), does not change and is always constant. In other words, the increase in VF when the proportion of return air increases corresponds to an increase in the advection flux, \(U_{\text{CV}}\). Assuming a steady-state flow field, the factor that determines the advection flux is the air velocity and concentration value of each CV, and hence, it is proportional to the rate of increase in the concentration when the pollutants in the return air supplied from the supply inlet opening are uniformly distributed in the indoor environment.

In contrast, the percentage of return air with respect to the total supply airflow rate was controlled to be constant in this study, and the changes in the returning rate of the pollutant had a significant impact on the changes in the indoor pollutant concentration levels. For example, if the emission rate of a pollutant accidentally doubles, the room-averaged concentration also doubles under the condition of 0% return air. Similarly, the room-average concentration becomes 4 times higher for the 50% return air condition and 10 times higher for the 80% return air condition.

When ventilation design is conducted assuming a constant pollutant emission rate, changes in the pollutant emission rate in the presence of return air can easily deteriorate indoor air quality, and hence, pollutant concentration monitoring and demand-controlled ventilation systems are important.\(^{37-42}\) When designing indoor ventilation for a room such as a classroom in a university where the number of residents/students is variable, the setting of the ventilation volume often depends on the maximum number of users, and the cooling or heating load increases because a larger than necessary ventilation volume is introduced. Hence, optimal design of an HVAC system and demand control that balances indoor air quality with energy consumption is important.

### 6 | LIMITATIONS OF THIS STUDY

The nature of pollutants in returning and recirculating air has been well recognized in both the academic and industrial communities, because returning and recirculating airflows are deterministic factors of pollutant concentration distributions in indoor environments. In addressing this issue, there have not been many examples of quantitative evaluation of the impact of indoor recirculation and returning via HVAC ductwork system on the formation of pollutant concentration distributions using various ventilation efficiency indices. In order to extract the essence of the formation mechanism of pollutant concentration distributions in an enclosed space, very simple boundary conditions were adopted in this study and the analyses assumed an isothermal condition. Therefore, the effects of the temperature distribution or the thermal plume on the pollutant transport were ignored.

The target model rooms in this study were very simple rectangular shapes used to obtain universal results; they were not intended to be an exact reproduction of the actual buildings and there was no reproduction of furniture and residents. The HVAC system was also simplified and the removal of pollutants by filters and deposition of pollutants in ducts was not included in this study. Pollutants used in the ventilation efficiency analysis were also assumed to be passive scalars and the specific characteristic of pollutants, for example, gas and particle phase, particle size, and Schmidt number, were not taken into account. These simplifications indicate that this study was conducted under very limited conditions; however, we believe that these simplifications make the formation mechanism of the pollutant concentration distributions clearer. An application-focused study targeting actual building spaces and ventilation design will be an importance issue to address in the future.

### 7 | CONCLUSIONS

In general, the air in a ventilated space consists of two populations.

- A population ventilating the space, that is, leaving the space and never returning to the domain.
- A population returning (recirculating) and thereby spreading the contaminants.

The two populations are generated by the air distribution system itself because air is entrained into the airflow supplied to the room. The final result is that the air distribution system generates a flow rate within the room that is larger than the supplied airflow rate. However, air is extracted from the room at the same rate as it is supplied. This implies that the surplus air must return (recirculate) to the room. It recirculates several times within the room before it finally leaves. To understand the formation mechanism of pollutant concentration distributions under the existence of “return air” of an HVAC system and large recirculating flow indoors, the returning characteristics of pollutants were investigated by ventilation efficiency indices, for example, \(\text{NEV, NEP, VF, and L-PFR} \), under conditions in which the proportion of the return air in the HVAC system was changed from 0% to 90%.
The numerical investigation findings can be summarized as follows:

- The pollutant concentration and VF increased and NEV, NEP, and L-PFR decreased as the proportion of return air increased. When the proportion of return air exceeded 50%, the pollutant concentration and VF increased rapidly.
- The value of NEV* approached the advection air velocity as the percentage of return air increased. NEV* gradually decreased and approached 1.0 as the percentage of return air increased because the transport of pollutants is largely dominated by the convection effect as the ratio of return air increases.
- When the proportion of return air increased, the L-PFR approached 0 and the L-PFR normalized by the ventilation airflow rate approached 1.0 because the average concentration at the local domain became a perfect mixing condition.

ACKNOWLEDGMENTS

This study was partially supported by JSPS (Japan Society for the Promotion of Science) Fund for the Promotion of Joint International Research (KAKENHI), Category (A) of Scientific Research (Grant Number JP 18H03807).

NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| $\phi_p$ | volume-averaged pollutant concentration (kg m$^{-3}$) in the target control volume (CV) |
| $U_j$ | ensemble-averaged scalar quantity of advection velocity in the j direction |
| $S_p$ | generation rate of the pollutant |
| $D_{eff}$ | effective diffusivity constant |
| $Sc_t$ | turbulent Schmidt number |
| $V_t$ | turbulent viscosity |
| $D_b$ | molecular diffusivity |
| $\alpha$ | returning probability to the target CV of the target pollutant which is generated in the target CV (-) |
| $\gamma$ | ratio of the return air in a HVAC system (-) |
| $F_{in}$ | total inflow flux (convective and diffusive flux) from the target CV (kg m$^{-2}$ s$^{-1}$) |
| $F_{out}$ | total outflow flux (convective and diffusive flux) from the target CV (kg m$^{-2}$ s$^{-1}$) |
| $A_{in}$ | area through the inflow flux in the target CV (m$^2$) |
| $A_{out}$ | area through the outflow flux in the target CV (m$^2$) |
| $q_{CV}$ | total amount of contaminant generated in the CV (kg s$^{-1}$) |
| $V_p$ | volume of the local domain (m$^3$) |
| $VF$ | visitation frequency of fluid parcels or pollutant (-) |
| $T_p$ | average-staying time of parcels/pollutant in the local domain (s one staying$^{-1}$) |
| $q_p$ | parcels/pollutant generation rate per unit time in a local domain (kg s$^{-1}$) |
| $C_p$ | local domain-averaged concentration (kg m$^{-3}$) |
| $J_p$ | amount of parcels/pollutants, visiting (returning to) the target local domain per unit time (Particle s$^{-1}$) |
| $M_p$ | amount of parcels/pollutants generated in the local domain (Particle s$^{-1}$) |
| $\Delta q_p$ | inflow rate of parcels/pollutants into the local domain per unit time (kg s$^{-1}$) |
| $Q$ | pollutant generation rate per unit time in a target local domain (kg s$^{-1}$) |
| $q'$ | pollutant rate included in supply air per unit time (kg s$^{-1}$) |
| $C_p$ | mean concentration at local domain (kg m$^{-3}$) |
| $C_e$ | mean concentration at exhaust outlet (kg m$^{-3}$) |
| $C_n$ | normalized concentration at local domain (-) |
| $Q_{in}$ | airflow rate from the supply inlet opening, the amount of fresh outdoor air (m$^3$ s$^{-1}$) |
| $Q_{sa}$ | amount of supply air summed the amount of return air and the amount of outdoor fresh air (m$^3$ s$^{-1}$) |

REFERENCES

1. Taylor ST. Return air systems. ASHRAE J. 2015;3:44-47.
2. Taylor ST. Controlling return air fans in VAV systems. ASHRAE J. 2014;10:54-58.
3. Brightman HS, Moss N. Sick building syndrome studies and the compilation of normative and comparative values. In Spengler JD, McCarthy JF, Samet JM, eds. Indoor Air Quality Handbook. New York, NY: McGraw-Hill; 2001:7.1-7.18.
4. ASHRAE. ASHRAE Standard 62.1-2019. Ventilation for Acceptable Indoor Air Quality.
5. Li D, Li X, Guo Y, Yang J, Yang X. A generalized algorithm for simulating contaminant distribution in complex ventilation systems with recirculation. Numerical Heat Transfer A Appl. 2004;45:583-599.
6. Hiyama K, Ishida Y, Kato S. Coupling 3d transient pollutant transport in a room into a flow network model with concentration response factor method. ASHRAE Trans. 2008;114(2):119-129.
7. Li X, Shao X, Ma X, Zhang Y, Cai H. A numerical method to determine the steady state distribution of passive contaminant in generic ventilation systems. J Hazard Mater. 2011;192:139-149.
8. Waters JR, Simon MW. The effect of recirculation on ventilation effectiveness parameters. Proc ROOMVENT. 2000;2:913-918.
9. Shao X, Liang S, Li X, Liang C, Yan S. Quantitative effects of supply air and contaminant sources on steady contaminant distribution in ventilated space with air recirculation. Build Environ. 2020;171:106672.
10. Skåret E, Mathisen HM. Ventilation efficiency. Environ Int. 1982;8(1–6):473-481.
11. Sandberg M. What is ventilation efficiency? Build Environ. 1981;16:123-135.
12. Sandberg M, Sjöberg M. The use of moments for assessing air quality. Build Environ. 1983;18:181-197.
13. Etheridge DW, Sandberg M. Building Ventilation: Theory and Measurement, 1st ed. Wiley; 1996:241-281.
14. Sandberg M. Ventilation effectiveness and purging flow rate — A review, International Symposium on Room Air Convection and Ventilation Effectiveness. University of Tokyo. 1992:17-27.
15. Csanady GT. Dispersal by randomly varying currents. J Fluid Mech. 1983;132:375-394.
16. Kato S, Ito K, Murakami S. Analysis of visitation frequency through particle tracking method based on LES and model experiment. Indoor Air. 2003;13:182-193.

ORCID

Eunsu Lim @ https://orcid.org/0000-0002-6038-9383
Kazuhide Ito @ https://orcid.org/0000-0002-7715-7896
17. Kato S, Murakami S. New ventilation efficiency scales based on spatial distribution of contaminant concentration aided by numerical simulation. *ASHRAE Trans.* 1988;94:309-330.
18. Federspiel CC. Air-change effectiveness: Theory and calculation methods. *Indoor Air.* 1999;9:47-56.
19. Lavidson D, Olsson E. Calculation of age and local purging flow rate in rooms. *Build Environ.* 1987;22:111-127.
20. Peng SH, Davidson L. Towards the determination of regional purging flow rate. *Build Environ.* 1997;32(6):513-525.
21. Zvirin Y, Shinnar R. Interpretation of internal tracer experiments and local sojourn time distributions. *Int J Multiphase Flow.* 1976;2:495-520.
22. Peng SH, Holmberg S, Davidson L. On the assessment of ventilation performance with the aid of numerical simulations. *Build Environ.* 1997;32(6):497-508.
23. Hang J, Sandberg M, Li Y. Age of air and air exchange efficiency in idealized city models. *Build Environ.* 2009;44(8):1714-1723.
24. Hang J, Wang Q, Chen X, et al. City breathability in medium density urban-like geometries evaluated through the pollutant transport rate and the net escape velocity. *Build Environ.* 2015;94:166-182.
25. Lim E, Ito K, Sandberg M. New ventilation index for evaluating imperfect mixing condition- analysis of Net Escape Velocity based on RANS approach. *Build Environ.* 2013;61:45-56.
26. Lim E, Ito K, Sandberg M. Performance evaluation of contaminant removal and air quality control for local ventilation systems using the ventilation index Net Escape Velocity. *Build Environ.* 2014;79:78-89.
27. Chung J, Lim E, Sandberg M, Ito K. Returning and net escape probabilities of contaminant at a local point in indoor environment. *Build Environ.* 2017;125:67-76.
28. Lim E, Chung J, Sandberg M, Ito K. Influence of chemical reactions and turbulent diffusion on the formation of local pollutant concentration distributions. *Build Environ.* 2020;168:106487.
29. Sandberg M, Kabanshi A, Wigö H. Is building ventilation a process of dilution of contaminants or delivering clean air? *Indoor Built Environ.* 2020;29(6):768-774.
30. Website of CFD Benchmarks. Aalborg University. https://www.cfd-benchmarks.com/benchmarkstest/
31. Lemaire AD, Chen Q, Ewert M, Heikkinen J, Inard C, Moser C, Nielsen PV, Whittle G. Room Air and Contaminant Flow, Evaluation of Computational Methods. Subtask-1 Summary Report 1993; International Energy Agency, Annex 20: TNO Building and Construction Research. 1993.
32. Nielsen PV. Specification of a Two-Dimensional Test Case. Department of Building Technology and Structural Engineering 1990; Aalborg University, IEA Annex 20: airflow Patterns within Buildings.
33. Ito K, Inthavong K, Kurabuchi T, et al. CFD Benchmark Tests for Indoor Environmental Problems: Part 1 Isothermal/non-isothermal flow in 2D and 3D room model. *Int J Archit Eng.* 2015;2:1-22.
34. Sorensen D, Nielsen PV. Quality control of computational fluid dynamics in indoor environments. *Indoor Air.* 2003;13:2-17.
35. Zhai Z, Zhang Z, Zhang W, Chen Q. Evaluation of various turbulence models in predicting air-flow and environments by CFD: Part 1 summary of prevalent turbulence models. *HVAC&R Res.* 2007;13:853-870.
36. Zhang Z, Zhang W, Zhai Z, Chen Q. Evaluation of various turbulence models in predicting air-flow and environments by CFD: Part 2 comparison with experimental data from literature. *HVAC&R Res.* 2007;13:871-886.
37. Fan Y, Ito K. Optimization of indoor environmental quality and ventilation load in office space by multilevel coupling with BES and CFD. *Build Simul.* 2014;7:649-659.
38. Fan Y, Kameishi K, Onishi S, Ito K. Field-based Study on Energy saving effects of CO2 demand controlled ventilation in office with application of energy recovery ventilators. *Energy Build.* 2014;68:412-422.
39. Fan Y, Ito K. Energy Consumption Analysis Intended for Real Office Space with Energy Recovery Ventilator by Integrating BES and CFD Approach. *Build Environ.* 2012;52:57-67.
40. Labeodan T, Zeiler W, Boxem G, Zhao Y. Occupancy measurement in commercial office buildings for demand-driven control applications—A survey and detection system evaluation. *Energy Build.* 2015;93(15):303-304.
41. Hong T, Fisk WJ. Assessment of energy savings potential from the use of demand controlled ventilation in general office spaces in California. *Build Simul.* 2010;3(2):117-124.
42. Schell M, Inthout D. Demand control ventilation using CO2. *ASHRAE J.* 2001;18:24.

How to cite this article: Lim E, Sandberg M, Ito K. Returning characteristics of pollutants for a local domain in the presence of returning and recirculating airflow in indoor environments. *Indoor Air.* 2021;31:1267-1280. https://doi.org/10.1111/ina.12803