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# Investigation on the thermal comfort and energy efficiency of stratified air distribution systems

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ABSTRACT

A stratified air distribution (STRAD) system is a feasible air conditioning design for large space buildings that satisfies the energy conservation and thermal comfort requirements. In this paper, a novel energy efficiency index for STRAD systems is developed using theoretical analysis to properly evaluate the system's energy saving potential. Using a validated numerical model, two typical stratified air distribution designs in a hypothetical terminal building are evaluated based on thermal comfort and energy savings. The influence of supply and return diffuser distributions on the ventilation performance of these two ventilation designs is studied. When the air is supplied at mid-height, the local thermal comfort is greatly improved without sacrificing the energy efficiency due to additional return grilles located at exterior walls. When the air is supplied at floor level, installing additional return grilles at exterior walls slightly alleviates the local draft risk, but doing so largely impairs the energy saving capacity of the ventilation system. To achieve better thermal comfort and higher energy efficiency, a more uniform distribution of supply diffusers surrounding the occupied zone is suggested. When the air is supplied at the floor level, increases in solar radiation intensity can be mitigated by utilizing external shading designs, which are particularly important in preventing too large of a temperature gradient in the region exposed directly to the solar radiation.



## Introduction

Modern buildings with high ceilings and large spaces, such as airport terminal buildings and international conference halls, are usually enveloped with large areas of transparent glass curtain walls, which serve the purpose of providing architectural esthetics and natural lighting (Gordon, 2008). However, large areas of fenestration admit extensive solar heat gains and significantly increase the space-cooling load. Furthermore, in these types of buildings, the internal heat gains are always substantial due to high occupant densities and illumination intensities. As a result, considerable energy consumption is spent on air conditioning to maintain a satisfactory thermal environment for the occupants (Parker et al., 2011). A survey study revealed that the average annual energy consumption of air conditioning systems in 29 Greek airport terminals was approximately 120 kWh/$m^2$, which accounted for a large proportion of the total building energy consumption and was highly energy-intensive (Balaras et al., 2003). Nevertheless, thermal comfort issues still exist in these large space buildings. For example, occupants may frequently feel overheated due to a large amount of direct solar radiation that is caused by a large ratio of window area to wall area (Kim et al., 2001). On the other hand, in large space buildings with traditional mixing ventilation system design, the air cooling for thermal comfort in the occupied zone is quite a small proportion of total load, most of the cold load is wasted in the un-occupied zone. Thus, increasing the energy-efficiency of air conditioning while also improving thermal comfort in large space buildings is attracting considerable attention in building research.

The stratified air distribution system (STRAD) is an applicable air distribution design for large space buildings. In recent years, STRAD systems have developed rapidly because of their better ventilation efficiency and energy saving capacity compared with traditional mixing ventilation systems (Bauman, 2003). In a numerical study of the air conditioning system in the International Airport of Bangkok, Simmonds (1996) demonstrated that the STRAD system was a feasible energy-conserving design for terminal buildings. In a following study, they clearly revealed that the application of a hybrid conditioning system consisting of a variable-volume displacement ventilation system and a radiant cooled floor to a terminal building was able to reduce the energy consumption of the air conditioning system (Simmonds et al., 1999). Han and Gu (Han and Gu, 2008) numerically studied the thermal environment in the third terminal building of the Beijing Capital International Airport. They found that by using a stratified air distribution in the terminal building, a satisfactory thermal environment was achieved in the occupied zone. Similarly, Li et al. (Li et al., 2009) investigated different air distribution designs to optimize the thermal environment in a

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train station building with a high ceiling level. The numerical results indicated that satisfactory thermal comfort in the occupied region was realized using a stratified air distribution design and supplying air at mid-height horizontally.

However, such limited research conducted on the application of STRAD systems in large space buildings offers little insight into their aspects of energy conservation and thermal environment optimization. Further in-depth research is required to solve the problems that continue to confuse design engineers, including energy saving principles, the organization of air supply and return flow, the occupied zone cooling load calculations and the size determinations of refrigeration equipment. In this paper, the energy saving potentials for STRAD systems adopted in large space buildings are clearly illustrated using theoretical analysis. Based on this analysis, a novel energy efficiency index that is able to quantify the energy saving capacity of different stratified air distribution designs in large space buildings is also developed. Then, using a validated computational fluid dynamic (CFD) model, two typical and practical air distribution designs that can realize thermal stratification in a hypothetical terminal building are numerically evaluated based on thermal comfort and energy saving. The influence of diffuser location and solar radiation on the indoor thermal environment of the building and the energy consumption of the air conditioning in both designs are studied. Most importantly, we aim to harness the CFD results that provide sufficient evidence in order to make a recommendation for an effectively designed STRAD system in large space buildings.

## Methodologies

### *Energy conservation of STRAD systems*

In a STRAD system, cold air is delivered directly to the occupied zone and is normally returned at or near the ceiling level. Thermal plumes that are generated by heat sources within the room draw air from the surrounding space and direct it upward. The upward movement of air in the room takes advantage of its natural buoyancy and produces a vertical temperature gradient. Because only the lower occupied zone needs to be cooled, the STRAD system is inherently energy saving, and this benefit is even more likely to be realized in buildings with high ceilings (Lin et al., 2006; Hashimoto and Yoneda, 2009). A significant number of studies have been conducted on the energy conservation of STRAD systems and have achieved considerable success. One of the most important findings is that the energy efficiency of a STRAD system highly depends on the stratification height (Yuan et al., 1998), which refers to the height in the room where the combined airflow rate of the thermal plumes is equal to the total air supply volume entering the room. Previous researchers have demonstrated that for a certain building ventilated by a STRAD system, the stratification height is mainly determined by several design parameters, including the heat load distribution in space (Nielsen, 1996; Gladstone and Woods, 2001; Kaye and Hunt, 2010), the supply diffusion characteristics (Lin and Linden, 2005; Lee et al., 2009), the designed air flow rate and the organization of indoor air flow (Mundt, 1995; Webster et al., 2002). A schematic diagram showing the primary relationship between these key design parameters and the stratification height is presented in Fig. 1, which is based on a review of the studies mentioned above.

To evaluate the energy conservation of a STRAD system, the ventilation effectiveness for heat distribution $\varepsilon_t$ was defined by Awbi (Awbi, 1998), as:

$$\varepsilon_t = \frac{t_e - t_s}{tave - t_s} \tag{1}$$

where $t_s$is the supply air temperature, $t_e$ is exhaust air temperature, and *tave* is the mean value for the occupied zone. This index is effective in the comparison of the energy efficiency of different STRAD systems with combined locations of return and exhaust grilles. However, this index may underestimate the energy saving potential of STRAD systems with separated locations of return and exhaust grilles because there is an additional energy saving potential (Lau and Niu, 2003; Xu et al., 2009). Taking this into account, efforts towards developing a new energy efficiency index for STRAD systems are conducted here. A cooling coil load $Q_{coil}$ calculation method for a typical STRAD system has been developed (Cheng et al., 2013). The required $Q_{coil}$ is calculated by the following equation:

$$Q_{coil} = Q_{Space} + Q_{vent} - C_p \times \dot{m}_e \times (t_e - t_{set}) \tag{2}$$

where $Q_{Space}$ and $Q_{vent}$ are, respectively, the space cooling load and the ventilation load in the classical sense, $\dot{m}_e$ is the exhaust air flow rate, $t_e$ is the exhaust air temperature, and $t_{set}$ is the room set-point temperature. It is obvious that the cooling coil load in a STRAD system is smaller than that in a mixing ventilation system by the quality of $C_p\dot{m}_e(t_e - t_{set})$, identified as the cooling coil load reduction $\Delta Q_{coil}$. The larger the exhaust air temperature, the more cooling coil load reduction is achieved and the more energy is saved by the refrigerating unit. By installing the exhaust grille at ceiling level and the return grille at a middle height, as shown in Fig. 2, the convective heat generated in the upper zone can be discharged by exhaust air directly and excluded from the cooling coil load. Correspondingly, the exhaust air temperature is able to be further increased and extra energy is saved for the cooling coil. To evaluate the energy saving potential of a STRAD system, an energy efficiency index $\varepsilon$ is defined as:

$$\varepsilon = \Delta Q_{Coil} / Q_{Space}. \tag{3}$$

When the return and exhaust grilles are combined, the return air temperature is equivalent to the exhaust air temperature and the space cooling load can be described as:

$$Q_{Space} = m_s(t_e - t_s). \tag{4}$$

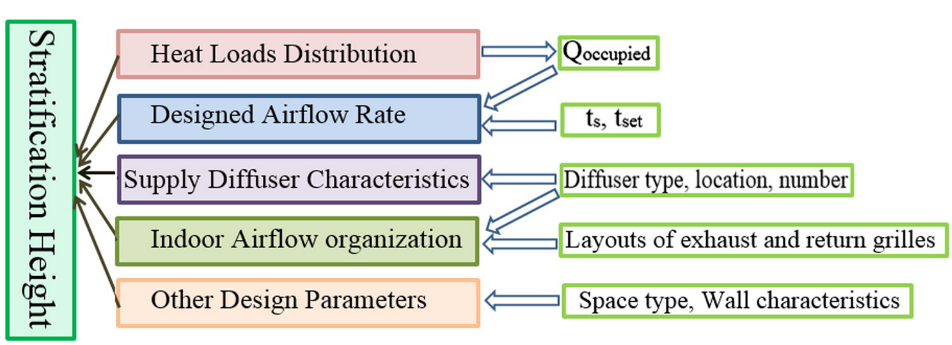



**Fig. 1.** Primary relationship between key design parameters and the stratification height.

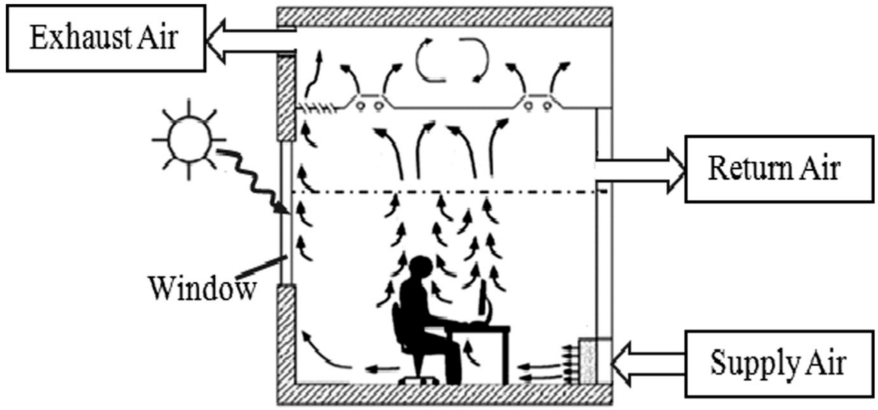



**Fig. 2.** A stratified air distribution system with separate locations of return and exhaust grilles.

Substituting $Q_{Space}$ in Eq. (3) with Eq. (4), we have:

$$\varepsilon = \theta \frac{t_e - t_{set}}{t_e - t_s} \tag{5}$$

where $\theta$ is the mass ratio of exhaust air to the total supply air. A larger value of $\varepsilon$ correlates to more energy being saved for the cooling coil and a higher energy efficiency obtained by the STRAD system. What should be noted is that in this study only sensible heat is considered.

*Numerical study*

To the best of our knowledge, there are usually two methods for investigating the ventilation performance in buildings. One is through experimental investigation, including small-scale experimental studies according to the similarity theory and full-scale field test research. The other is numerical model analysis based on computational fluid dynamics (CFD). With rapid advance in computing power and the development of user-friendly CFD program interfaces, numerical simulations are more and more popular for predicting ventilation performance in buildings (Chen, 2009). Consequently, the CFD technology is adopted in this paper and numerical studies are conducted.

*Description of physical model*

A hypothetical terminal building with a rectangular shape serves as the focus of this research. The facades of the terminal building are constructed with single-layer clear glass. Considering the computational resource limitations, only the middle section of an elongated waiting hall in the terminal building, with a dimension of 24.9 m (W) × 11.2 m (L) × 8 m (H), is investigated, as shown in Fig. 3. Two air conditioning units are located at the center of the space. There are 128 seats provided for occupants waiting for departure. All of the occupants are assumed to be in a sedentary condition with a sensible heat loss of 70 W/person. Twenty-four lamps are simplified as panels and located at ceiling level, and four skylights are installed on the roof 1 m above the ceiling level to take advantage of natural lighting. The thermal loads of these lamps are set to 16 W/m$^2$, according to the ASHRAE Standard (ASHRAE, 2013) for a terminal building. The heat gain from the large glazed facades is set to 29 W/m$^2$, and that from the skylights and the ceiling is set to 19 W/m$^2$, according to the Hong Kong guide for overall thermal transfer value (OTTV) (OTTV criteria and Code of Practice for Overall Thermal Value in Buildings, 1995). The thermal load of each facility, for example the electronic screen provided for flight information, is set to 200 W, and the power of the moving walkway is assumed to be 800 W. The total heat gains of the building as a function of floor area is 85.4 W/m$^2$.

*Mathematical model and boundary conditions*

The computational domain is divided into several sub-regions and meshed with high-quality structured grids. The total mesh number is approximately 1.4 million. The conservation of mass, momentum and energy at each point of the flow field was governed by using the Navier–Stokes equations, which were solved with a commercial code — Fluent based on a finite-volume method. The renormalization group (RNG) $k - \varepsilon$ model contained differential viscosity models to account for the

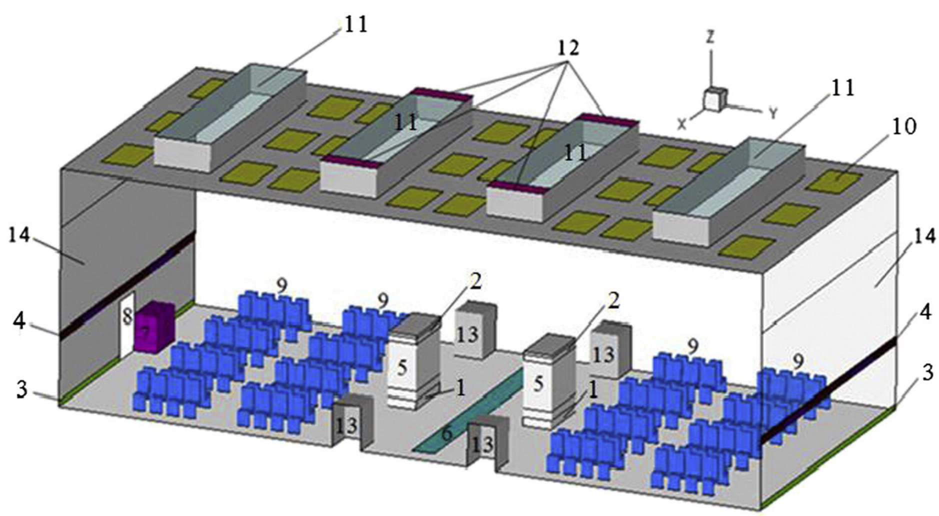



**Fig. 3.** Geometric model of the terminal building.

effect of a low Reynolds number and is adopted to simulate a turbulent effect. This model is reasonably accurate for mixed convection flows in many engineering applications (Posner et al., 2003; Zhang and Chen, 2007). All thermo physical properties are assumed to be constant except for density, which is treated with a Boussinesq model. The convergence criteria are such that the residuals are set to $10^{-4}$ for the continuity and momentum equations and $10^{-7}$ for the energy equation, which are rigorous qualitative measurements for convergence (Gilani et al., 2013). Details of the numerical methods and the boundary conditions are listed in Table 1. It is worthwhile to note that the radiant heat transfer is calculated by the discrete ordinates (Do) radiation model. This model solves the radiative transfer equation (RTE) for a finite number of discrete solid angles, each associated with a vector direction fixed in the global Cartesian system.

*Model validation*

Results from measurements in a climate chamber are used to validate the computational model (Cheng et al., 2013). The climate chamber, with dimensions of 4.0 m (L) × 2.7 m (w) × 2.4 m (H), is ventilated by an independent displacement ventilation system, as shown in Fig. 4. The arrangements in the climate chamber are presented in Fig. 5. The numerical validation procedures were designed to fully follow the experimental work. Fig. 6 shows the comparisons between measured and simulated air temperature distributions at three positions with three heights in the test chamber. The agreement between the measured and modeled temperature profiles in the chamber was acceptable.

*Description of simulation cases*

Two typical ventilation schemes that are able to create a stratified thermal environment in the terminal building are designed and investigated. In the first design, the air is horizontally supplied at floor level at a low velocity and returned at mid-height, a setup that simulates a typical stratified air distribution. In the second design, the air is horizontally supplied at mid-height at a high velocity and returned at floor level, which is frequently adopted in practice for large spaces. Diffuser 1 and Diffuser 2, located at the air conditioning units as indicated in Fig. 3, are the primary air diffusers used as either supply or return grilles for the different cases. Diffuser 3 and Diffuser 4 are located on the side walls, which is rare in practical applications. However, to investigate the impact of diffuser locations on ventilation performance, these two diffusers are optionally used as additional return inlets or supply outlets. Diffuser 1 and Diffuser 3 are located at floor level, while Diffuser 2 and Diffuser 4 are located at a height approximately 2 m above floor level. It should be emphasized here that the exhaust air grilles are installed near the skylights and completely separated from the return grilles, for the purpose of achieving additional energy savings (Cheng et al., 2012). There are eight simulation cases in total, which are listed in Table 2. In Case 1 and 3, the two typical air distribution designs with general solar radiation intensity are studied and compared from viewpoints of thermal comfort and energy saving. Based on that, optimization studies are conducted and the impacts of diffuser locations on the ventilation performances of the two typical air distribution designs are investigated in Case 2, 4, 5 and 6, respectively. In Cases 7 and 8, the air distributions are the same as those in Cases 5 and 6, respectively. However, the solar radiation intensity in these two cases is heightened. A higher thermal load is set for the large glazed facades at 150 W/m$^2$ for Cases 7 and Case 8, as presented in Table 1. Correspondingly, the total heat gain of the space as a function of floor area is increased to 137.5 W/m$^2$ in Case 7 and 8. In the first six cases, the air supply temperature is constant at 18 °C with a total supply airflow rate of 2.75 kg/s, based on the assumption that 80% of the space cooling load is dispersed into the occupied zone. The set-point air temperature in the terminal building at head level is 25 °C. To maintain the same set-point indoor air temperature, an increase of the air supply volume up to 3.96 kg/s is required for Case 7 and 8.

**Table 1**
Details of the numerical methods and boundary conditions.

| Turbulence model | RNG $k-\varepsilon$ model |
|---|---|
| Numerical schemes | Staggered third order PRESTO scheme for pressure; upwind second order difference for other terms; SIMPLE algorithm |
| Heat sources | Wall boundary with constant heat flux |
| Floor surface | Adiabatic wall boundary |
| Supply air outlet | Velocity inlet |
| Return air inlet | Velocity inlet with a negative direction |
| Exhaust air inlet | Pressure-outlet |
| Radiation heat | Discrete ordinates (Do) radiation model |

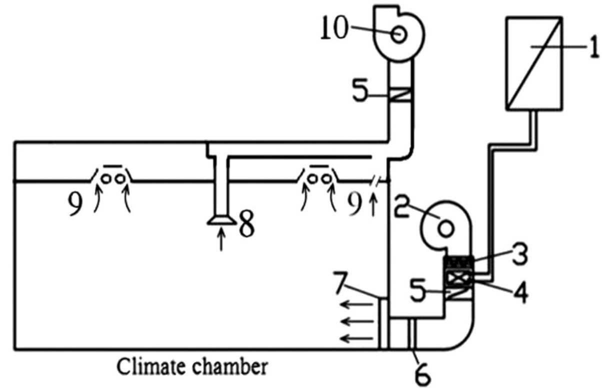



**Fig. 4.** Ventilation system of the climate chamber.

## Results

*Thermal comfort*

Fig. 7 compares the temperature and velocity distributions for these two typical air distribution designs at Plane X = 5.6 m, which runs through the passageway and the air conditioning units. The shapes of occupants are also given in the figure for better perspective. In Case 1, with floor level air supply, the air flow in the occupied zone is dominated by the thermal buoyancy of heat sources. In Case 3, with mid-height air supply, the momentum flux from the supply diffusers is the main driving force of the air flow in the occupied zone. Thermal stratification occurs throughout the entire space in Case 1 and is especially evident in

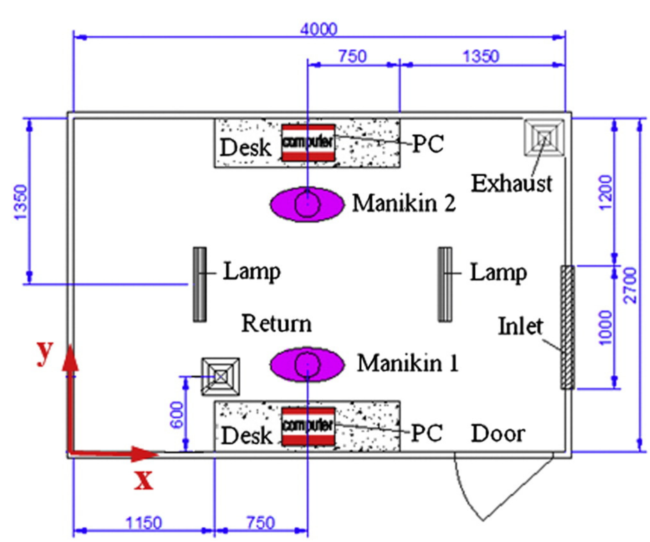
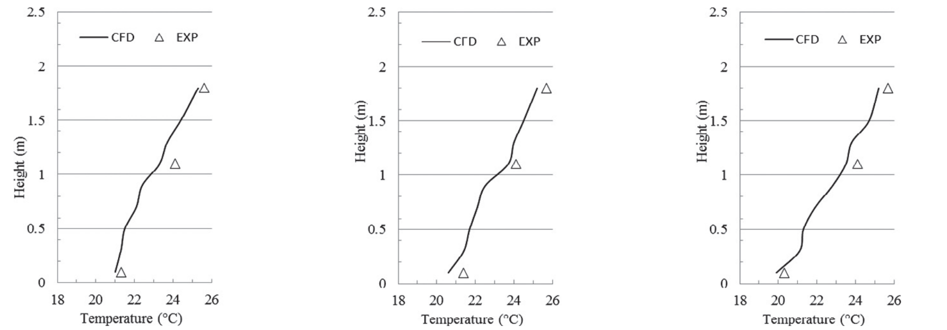



**Fig. 5.** The arrangements of the climate chamber.




Point A (x=0.8 m, y=1.32 m) Point B (x=1.9 m, y=1.32 m) Point C (x=3 m, y=1.32 m)

**Fig. 6.** Comparison of simulated and measured temperature profiles.

the occupied zone. The temperature difference between head level and ankle level is more than 3 °C, which may result in undesired thermal comfort problems. In comparison, the air is well mixed in the occupied zone in Case 3. The thermal environment in Case 3 is much more homogeneous than that in Case 1. Therefore, thermal comfort issues caused by an excessively large temperature difference between head and ankle levels would be avoided in Case 3. However, the air mixing in the occupied zone in Case 3 is not as complete as it would be in a smaller space, and horizontal temperature variations still exist. As shown in Fig. 7(b), the occupants seated close to the air conditioning units and below the supply jets may not obtain enough cooling and will experience a temperature that is too warm. Increasing the supply air velocity is an optional choice to enhance the air mixing in the occupied zone and alleviate the local thermal island phenomenon, but this option also holds the possibility of increasing the local draft risk for the occupants exposed directly to the cold supply jets.

Local thermal comfort issues in Case 1 and 3 are revealed in Fig. 8 by using the thermal index of draft rate (DR), which is defined as the percentage of people predicted to be dissatisfied due to draft (ASHRAE: Thermal environmental conditions for human occupancy, 2004). The draft rate contains the two main factors affecting the thermal comfort indoors, the air temperature and velocity, and can be calculated according to the following equation:

$$DR = \left([34-t_a] * [\nu-0.05]^{0.62} * (0.37 * \nu * Tu + 3.14)\right. \quad (6)$$

where, $t_a$ is the local air temperature, °C; $v$ is the local mean air speed, m/s; $Tu$ is the local turbulence intensity, %. The DR distribution in Fig. 8(a) indicates that local thermal discomfort resulted from undesired local cooling exists in Case 1 in the regions nearest to the supply diffusers. Similar with that, there is serious local draft risk in Case 3 for occupants exposed directly to the cold supply airstreams, as shown in Fig. 8(b). The DR values throughout the test region are even as high as 30, which are greatly exceed the maximum value and may lead to severe thermal comfort problem. Therefore, the unwanted local cooling draft was considered as the main reason that deteriorating the thermal environment in the terminal building with the two typical air distribution designs.

Fig. 9 shows the DR values at Plane X = 5.6 m for Cases 2 and 5. In Case 2, the air is supplied at floor level, the same as in Case 1, but additional supply diffusers are located on the exterior walls. There is back flow of air caused by the collision of the two supply airstreams, which enhances the local turbulence intensity and the air mixing in the bottom of the occupied zone. Hence, it was assumed that the temperature gradient in the occupied zone would decrease. The DR values in the collision region show a slight increase, as is indicated in Fig. 9(a) by the red rectangular frame, and approach their limit. Consequently, the local draft risk in this region for Case 2 is marginally higher than it is for Case 1. Placing additional return grilles at an exterior wall moves the collision region closer to the exterior wall due to the induced function of the return grilles but slightly alleviates the local draft risk, as shown in Fig. 9(b) for Case 5.

Fig. 10 displays the DR values at Plane X = 5.6 m in Cases 4 and 6, in which the air distribution is similar to that in Case 3 and pertains to the up-inlet and down-outlet modes. By locating additional supply diffusers on exterior walls in Case 4, occupants exposed directly to the supply airstreams still suffered from local draft and the draft risk region expanded due to the collision of supply airstreams, as indicated in Fig. 10(a). Accordingly, there is a possibility that more occupants may complain about local thermal discomfort in Case 4 than in Case 3. However, back flow formed by the collision of the two supply airstreams greatly enhanced the air flow turbulence in the occupied zone and eliminated most of the local thermal island regions. In addition, with additional supply air diffusers, the momentum of each supply airstream is reduced, and the throw of each inlet jet is shortened, as the total supply air flow rate is constant. This is beneficial to alleviating the local draft for occupants exposed directly to supply airstreams. The local thermal

**Table 2**
Different simulation cases.

| Cases | Supply diffuser locations | Supply air velocity (m/s) | Return grille locations | Heat gain for the right side wall (W/m$^2$) | Essence of each case |
|---|---|---|---|---|---|
| Case 1 | 1 | 0.32 | 2 | 29 | B-S-M-R* |
| Case 2 | 1 & 3 | 0.31 | 2 | 29 | B-S-M-R* with additional supply diffuser |
| Case 3 | 2 | 2.88 | 1 | 29 | M-S-B-R** |
| Case 4 | 2 & 4 | 2.1 | 1 | 29 | M-S-B-R** with additional supply diffuser |
| Case 5 | 1 & 3 | 0.31 | 2 & 4 | 29 | B-S-M-R* with additional supply and return diffusers |
| Case 6 | 2 & 4 | 2.1 | 1 & 3 | 29 | M-S-B-R** with additional supply and return diffusers |
| Case 7 | 1 & 3 | 0.44 | 2 & 4 | 150 | B-S-M-R* with additional supply and return diffusers; Enhanced solar radiation |
| Case 8 | 2 & 4 | 2.94 | 1 & 3 | 150 | M-S-B-R** with additional supply and return diffusers; Enhanced solar radiation |

* B-S-M-R: Typical air distribution with bottom supply and mid-height return.
** M-S-B-R: Typical air distribution with mid-height supply and bottom return.

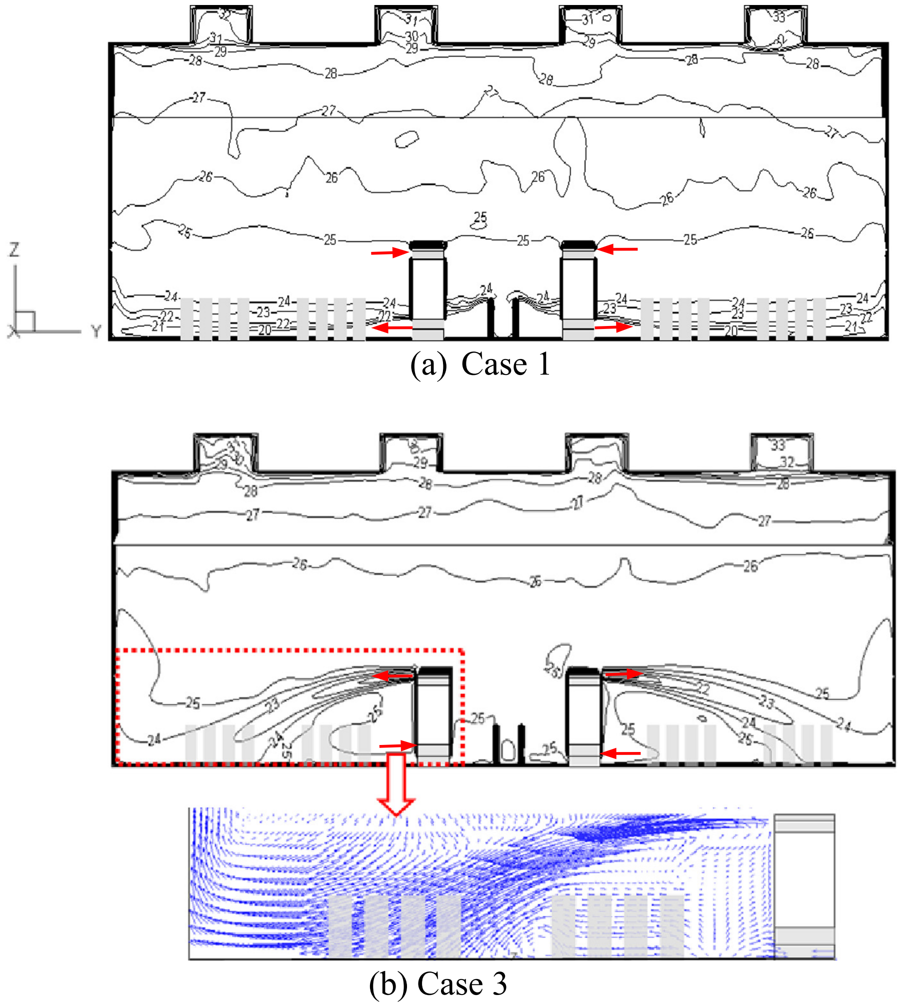



**Fig. 7.** The temperature and velocity distributions at Plane X = 5.6 m in Case 1 and Case 3.

environment is improved to a large extent by locating additional return grilles at the exterior wall, as shown in Fig. 10(b) for Case 6. The induced function of the additional return grilles largely reduced the local draft risk for the collision region of supply airstreams, which is especially obvious on the right side of the occupied zone. On the left side of the occupied zone, because of the arrangement of the information desk no additional return grilles are installed at the position X = 5.6 m on the exterior wall. As a result, the effects of additional return grilles on improving the local thermal comfort are not as prominent as those on the right side of the occupied zone.

*Energy saving*

In this study, the proposed evaluation criterion is adopted to assess the energy saving potential of different air distribution designs in the terminal building. The mass-weighted average exhaust air temperature in each case is obtained through post-processing of the simulation results. The calculated $\varepsilon$ values in different cases are summarized in Table 3. These results show that the cooling coil load reduction and the energy efficiency of STRAD systems are directly related to the exhaust air temperature. By locating additional supply diffusers at the

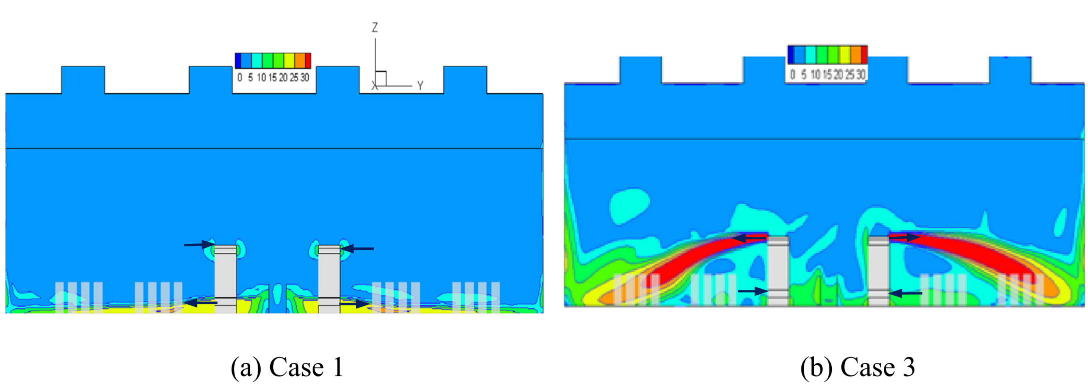



**Fig. 8.** DR values at Plane X = 5.6 m in Case 1 and Case 3.

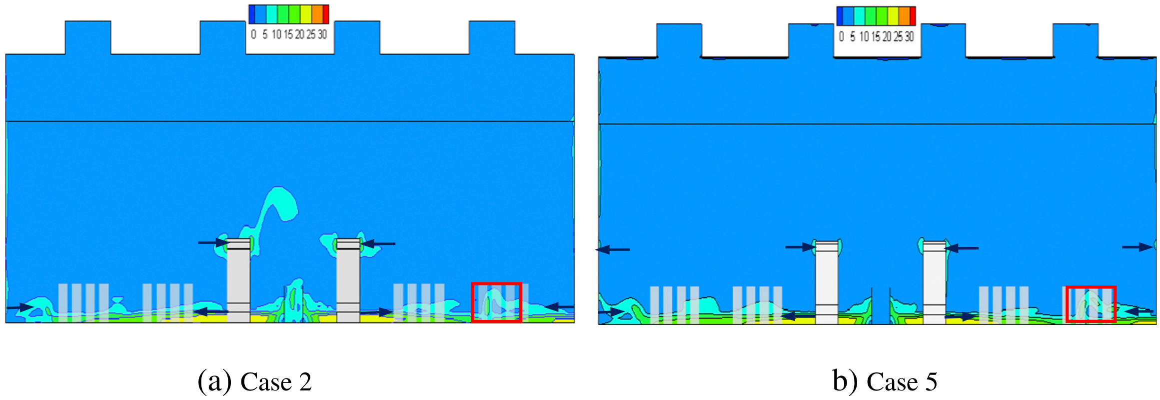



**Fig. 9.** DR values at Plane X = 5.6 m in Case 2 and Case 5.

exterior walls, the exhaust air temperature increased in both typical air distribution designs. Correspondingly, the energy efficiency $\varepsilon$ increased from 16.8% in Case 1 to 20.8% in Case 2 when the air is supplied at the floor level, while the value is improved from 15.4% in Case 3 to 16.6% in Case 4 when the air is supplied at mid-height.

When the air is supplied at floor level, installing the return grille on exterior walls impairs the energy-saving capacity of the ventilation system. The induced effect of the return grilles located at the exterior walls blocks the upward flow of warm air along the exterior wall. As a result, much more warm air moves into the occupied zone and increases the cooling coil load. With additional air returning location of Diffuser 4, the energy efficiency $\varepsilon$ is obviously reduced from 20.8% in Case 2 to 17.6% in Case 5. In comparison, when the air is supplied at mid-height installing the return grille on the exterior walls is beneficial for improving the energy efficiency of the ventilation system, but the effect is minimal. For example, the energy efficiency $\varepsilon$ is 16.6% in Case 4, but increases to 17.3% in Case 6. In the first six cases, the energy saving of the cooling coil is most prominent in Case 2, in which it is as high as 20.8%.

*Effect of solar radiation intensity on the ventilation performance*

Two additional cases, Cases 7 and 8, are conducted separately to investigate the impact of solar radiation intensity on the thermal environment and the energy consumption of the building for each of the two typical ventilation designs. The ventilation system in Case 7 is designed identically to that in Case 5, whereas in Case 8, it is the same as in Case 6. The space cooling load $Q_{Space}$ in Cases 7 and 8 are obtained according to the intensified solar radiant, as presented in Table 1. The energy efficiency $\varepsilon$ in Cases 7 and 8 are presented in Table 3. The table shows that as the solar radiation increases, the energy efficiency $\varepsilon$ is improved for both typical air distribution designs. One of the main reasons for this is that as the solar radiation increases, the thermal buoyancy flow along the exterior wall is correspondingly strengthened. According to Fig. 11, more warm air flows upward along the exterior wall to the un-occupied zone in Case 7 than it does in Case 5. Consequently, in Case 7, more heat accumulates at the region close to ceiling level and increases the exhaust air temperature more than it does in Case 5. This means that a much higher proportion of the space heat is discharged from the occupied zone and excluded from the cooling coil load. However, Fig. 11 also indicates that in Case 7 due to the increase in solar radiation, the temperature gradient in the occupied zone for the region exposed directly to solar radiation is apparently larger than that in Case 5, which may lead to undesirable thermal problems. In Case 8, the intensified thermal buoyancy flow along the exterior wall is also present. In addition, the streamlines of the supply air are much more linear in Case 8 than in Case 6 and are the result of the enhanced thermal plumes flowing from the floor level to the upper zone. Therefore, the supply air entrains less warm air from the upper zone to the occupied zone in Case 8; correspondingly, the energy efficiency $\varepsilon$ is improved from 17.3% in Case 6 to 19.5% in Case 8.

## Discussion and conclusions

This paper investigated two typical air distribution designs in a hypothetical terminal building, one that supplied air at floor level and the other at mid-height, and evaluated them based on thermal comfort and energy saving. The impacts of the diffuser location and solar radiation intensity on the ventilation performance are illustrated for both ventilation designs. When the air was supplied at floor level and returned at mid-height, a distinct thermal stratification was identified in the occupied zone. By locating additional supply diffusers on exterior

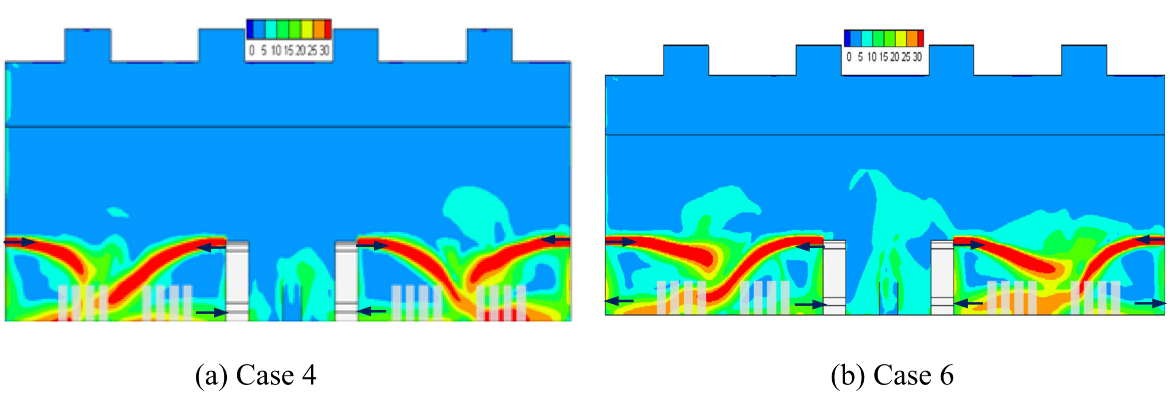



**Fig. 10.** DR values at Plane X = 5.6 m in Case 4 and Case 6.

**Table 3**
Energy-savings for cooling coil.

| Cases | $t_r$ (°C) | $t_e$ (°C) | $\Delta Q_{coil}$ (W) | $\varepsilon$ |
|---|---|---|---|---|
| Case 1 | 24.9 | 30.5 | 4094 | 16.8% |
| Case 2 | 24.7 | 31.8 | 5047 | 20.8% |
| Case 3 | 25.1 | 30 | 3737 | 15.4% |
| Case 4 | 24.8 | 30.4 | 4154 | 16.6% |
| Case 5 | 24.9 | 30.7 | 4273 | 17.6% |
| Case 6 | 24.7 | 30.6 | 4199 | 17.3% |
| Case 7 | 25.7 | 34.1 | 6774 | 19.3% |
| Case 8 | 25.4 | 34.2 | 6871 | 19.5% |

walls, the temperature gradient in the occupied zone was significantly reduced. However, special attention should be given to the airstream collision regions in order to avoid undesirable local drafts. The locations of return grilles slightly influenced the thermal environments of the tested conditions. When the air was supplied at mid-height and returned at floor level, a high local draft risk existed for the occupants exposed directly to the cold supply air jets. With additional supply diffusers on exterior walls, the air mixing in the occupied zone was enhanced and most of the local thermal island regions were eliminated. The local draft risk was alleviated to a large extent by the induced function of the return grilles by locating the additional return grilles on exterior walls.

By locating additional supply diffusers on the exterior walls, the energy efficiency $\varepsilon$ of the ventilation systems was increased significantly for both designs. However, when the air is supplied at the floor level, installing the return grilles on exterior walls also impairs the energy-saving capacity of the ventilation system. In comparison, locating the return grille on the exterior wall can enhance the energy efficiency of the ventilation systems when the air is supplied at mid-height (2 m), but the effect is still minimal. Therefore, to achieve a satisfactory overall performance of the air distribution system designed for large space buildings, it is recommended to uniformly install supply diffusers surrounding the occupied zone. When the air is supplied at mid-height, a more distributed spacing of return diffusers is also recommended. However, return diffusers should not be located along exterior walls when the air is supplied at floor level.

As the solar radiation increased, the upward thermal buoyancy flow along the exterior wall was strengthened, and the energy efficiency $\varepsilon$ of the ventilation system increased. However, when the air was supplied at floor level, the temperature gradient in the region exposed directly to the transmitted solar radiation also increased. Thus, external shading designs are particularly important in this configuration. The intensified thermal buoyancy flow in the occupied zone also uncurled the streamlines of the supply air when the air was supplied at mid-height. This was beneficial to reducing the entrainment of warm air from the upper zone to the occupied zone. Correspondingly, the energy efficiency of the ventilation system was further improved.

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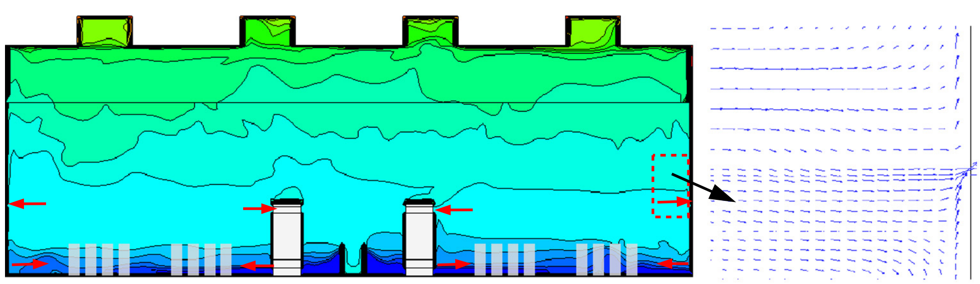
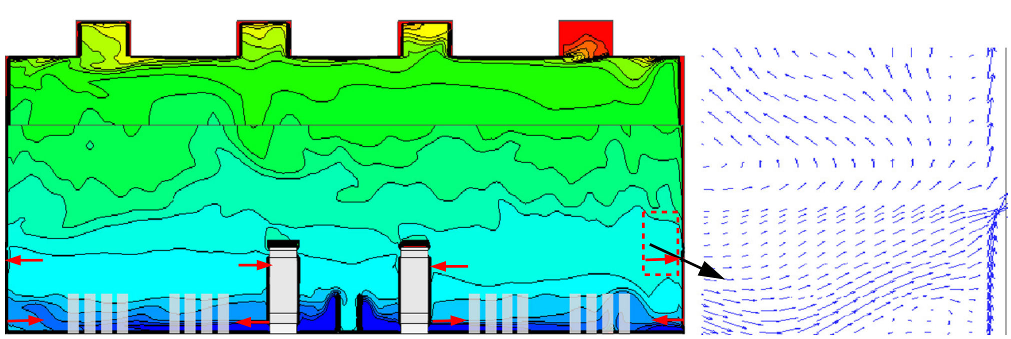


(a) Case 5



(b) Case 7

**Fig. 11.** Temperature contours and local velocity vectors at Plane X = 5.6 m in Case 5 and Case 7.

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