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Study on the impact of parallel jet spacing on the performance of multi-jet stratum ventilation

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**HIGHLIGHTS**

- The optimal PJS takes into account both tuyere location and climate factors.
- The influence of PJS is explored based on multi-parameter performance evaluation.
- The combined effects of PJS on summer and winter are analyzed emphatically.
- Optimal PJS can help improve thermal comfort and reduces energy consumption.

**ARTICLE INFO**

**Keywords:**
Multi-jet stratum ventilation
Parallel jet spacing
Jets flow interaction
Ventilation performance
Thermal comfort

**ABSTRACT**

With the wide spread of novel coronavirus SARS-CoV-2 pandemic around the world, high quality indoor environment and more efficient mechanical ventilation become the new focus of scholars’ attention. Stratum ventilation refers to the ventilation mode that the air supply port on the side wall slightly higher than the height of the working area directly sends fresh air into the working breathing area. As an efficient mechanical ventilation mode, it can create a more healthy and comfortable indoor environment. However, the impact caused by airflow characteristic under stratum ventilation on the thermal performance and indoor comfort is noteworthy due to its supply air outlets are close to the occupied zone. It is widely known that parallel turbulent jets are important for the flow structure and air distribution. Hence, an optimum parallel jet spacing (PJS) between two jet centerlines can obviously enhance the fluid interaction and indoor thermal comfort with low energy consumption. Therefore, this study aims to investigate the impact of the PJS on the performance of multi-jet stratum ventilation. A validated Computational Fluid Dynamics (CFD) model was used to conduct the year-round multivariate analysis. A total of eight PJSs, four inlet locations and five climate zones were discussed synthetically. Air distribution performance index (ADPI), ventilation effectiveness (Et) and economic comfort coefficient were employed as the evaluation indicators to assess the thermal comfort and energy efficiency in various scenarios. Research results indicated that the PJS showed different influences on the indoor thermal comfort and energy utilization efficiency as a result of cooperative effect including energy dissipation, air short-circuit probability, air distribution uniformity and airflow path. Combining with building energy simulation method, the optimum PJSs of stratum ventilation with different air inlet positions in five climate zones were obtained, which can help provide a comfortable indoor thermal environment and improve energy efficiency in a low-cost way. The data and conclusions presented in this study can supplement the theoretical basis for the actual applications of multiple-jet stratum ventilation used in an office.

1. Introduction

Recently, the indoor thermal environment grows awareness due to that people usually spend around 90% of their time indoors [1]. Poor indoor environment can not only seriously affect occupants’ health and work efficiency [2,3], but also increase the mortality caused by lung cancer, respiratory diseases and cardiovascular diseases [4,5]. More seriously, with the wide spread of novel coronavirus SARS-CoV-2...
pandemic around the world, high quality indoor environment and more efficient mechanical ventilation become the new focus of scholars’ attention [6]. Until now, building energy consumption has accounted for 20–40% of the total energy consumption in China [7,8]. And heating and cooling operation as the main sections make up 40% of the building energy consumption [9,10]. Therefore, exploring an efficient air conditioning system to build a comfortable and healthy indoor environment is significant to the building energy conservation.

Existing studies indicate that improving the air flow turbulence is treated as one of the widely used methods to seek a more suitable air conditioning mode at present [11]. Hence, stratum ventilation as a sustainable air distribution for low-energy buildings is proposed, this mode is characterized by the formation of a fresh air layer through the side wall vents and directly to the working breathing area. Related studies demonstrate that stratum ventilation can build a comfortable environment in an efficient way due to its working principle [12,13]. The fresh air can be forced to flow directly through the occupied zone under stratum ventilation by creating a layer of fresher air in the under stratum ventilation condition [16]. As for heating applications, stratum ventilation can reduce energy consumption by 25% and increase the occupied zone temperature by 1.8 °C relative to mixing ventilation [17]. Meanwhile, the indoor comfort can also be enhanced by 10% under stratum ventilation [17]. Hence, stratum ventilation has the advantage in improving the indoor thermal comfort and energy saving for space heating and cooling simultaneously.

To further enhance the performance of stratum ventilation, scholars conduct a series of parameter studies, including supply air temperature, airflow rate, supply air angle, supply air outlet type, and inlet position and so on [18–20]. Research has found that draft complaint can be obviously reduced with the supply air over 20 °C served by stratum ventilation [21]. Zhao H et al. [22] point out that the average predicted percentage of dissatisfied (PPD) reaches the peak with the supply air angle of 90° under stratum ventilation. Fabric diffusers as the most suitable air terminals of stratum ventilation are reported to provide a clean, healthy and comfortable indoor environment by an experimental study conducted by Yao T et al [23]. Zhang S et al [24] comprehensively investigate the effects of different operation parameters on heating performance of stratum ventilation, using experimentally validated Computational Fluid Dynamics (CFD) simulations.

Due to the supply air outlets of stratum ventilation are close to the occupied zone, the impact caused by airflow characteristic on the thermal performance and indoor comfort is more obvious than other ventilation methods. Parallel turbulent jets are important for the flow structure and air distribution [25]. It is well known that the interaction of parallel jets with adjacent jets or with their surroundings plays a crucial role in the effectiveness of the interacting flow. Hence, optimum jet spacing between two jet centerlines can notably enhance fluid interaction and indoor environment comfort with low energy and cost consumption [26]. However, current studies mainly focus on the research about supply air temperature, supply airflow rate, supply air angle, air terminal types and air outlet position. The research about the influence of PJS on the performance of multi-jet stratum ventilation is scarce. To fill the knowledge gap and enhance the energy efficiency and thermal comfort of stratum ventilation in a low-cost way, the optimum PJS for the multiple-jet stratum ventilation in an office was investigated by an experimentally validated CFD model in this study. Ventilation effectiveness, air distribution performance index (ADPI) and economic comfort coefficient were introduced as the evaluation indicators and optimization objectives. Combining with building energy simulation, the effect of various PJSs on the thermal performance and indoor comfort in different climate zones with four widely used inlet positions was analyzed. The purpose of this study is to determine the optimal PJS by exploring the effects of different PJSs by investigating the effects of different PJSs on the performance of stratum ventilation under heating and cooling conditions. The optimal PJS can help the stratum ventilation to create a more comfortable and energy-saving indoor environment in a more economical way, so as to promote the promotion of the stratum ventilation energy-saving technology.

2. Methodology

The methods used in this study were summarized in Fig. 1. The corresponding steps were to:

(a) CFD simulation was first introduced to carry out univariate and multivariate collaborative research, which can obtain the effect of various PJSs on ADPI and $E_m$ under different inlet positions.

(b) Building energy simulation was secondly used to calculate the building load weight coefficient ($\lambda$) of an office located in different climate zones, which was subsequently employed to derive the year-round ventilation correction factor ($\beta$) based on economic comfort coefficient. Due to that heating is not required in hot summer and warm winter climate zone (HSWW), only severe cold A/B climate zone (SC-A/B), severe cold C climate zone (SC-C), cold A/B climate zone (C-A/B) and hot summer and cold winter climate zone (HSCW) participated in the energy consumption calculation in this section.

(c) Finally, the curve of the year-round ventilation correction factor changing with PJSs can be formed based on CFD simulation and building energy simulation. By smoothing spline fitting, the minimum value of the curve can be obtained. The optimum PJSs for stratum ventilation applied in different climate zones were recommended by considering the energy saving and indoor thermal comfort simultaneously.

Each of these steps was explained in the following sections (corresponding to the numbers in Fig. 1).

2.1. CFD simulation

Computational Fluid Dynamics (CFD) has been widely applied in the field of indoor thermal environment simulation [27]. It can model indoor airflow distribution, indoor temperature distribution and indoor air quality simultaneously. Moreover, compared with the experimental measurement, CFD simulation is less expensive and less time consuming.
The accuracy of the model has been verified in another study [36]. Standard-Gird independence test for selection of the convenient mesh. Standard-Gird independence test for selection of the convenient mesh. Table 2-Gird independence test for selection of the convenient mesh.

| Mesh nodes | Simulated temperature (°C) | Maximum relative error in computing values of temperature compared to the previous mesh values (%) |
|------------|----------------------------|--------------------------------------------------------------------------------------------------|
|            | 0.1 m                     | 0.6 m                                | 1.1 m                   | 1.4 m                   | 2.1 m                   | 2.8 m                   |
| 3          | 844,865                   | 19.36                                | 20.39                   | 21.03                   | 21.05                   | 20.45                   | 19.93                   | –                        |
| 3          | 1,952,834                 | 18.36                                | 18.85                   | 19.21                   | 19.20                   | 18.75                   | 18.34                   | 8.79                     |
| 3          | 2,949,594                 | 18.42                                | 18.90                   | 19.21                   | 19.18                   | 18.72                   | 18.31                   | 0.33                     |

Fig. 2. 3D CFD model geometry.

Table 1-Boundary conditions of CFD model.

| Boundary condition | Winter condition | Summer condition |
|--------------------|------------------|------------------|
| Room air inlet     | Uniform velocity outlet | 27.0 °C, 12 ACH | 20.0 °C, 10 ACH |
| Room air inlet     | Neumann boundary condition | – | – |
| Outer Wall         | Constant wall temperature, Constant wall temperature, | –3.2 °C | 33.1 °C |
| Ceiling            | Constant wall temperature, Constant wall temperature, | 14.5 °C | 27.6 °C |
| Floor              | Constant wall temperature, Constant wall temperature, | 10.3 °C | 24.1 °C |
| Interior wall (Left/Right) | Constant wall temperature, Constant wall temperature, | 16.4 °C | 27.1 °C |
| Interior wall (near the corridor) | Constant wall temperature, Constant wall temperature, | 12.5 °C | 27.4 °C |
| Occupant           | Constant heat flux solid boundary, Constant heat flux solid boundary, | 100 W | 100 W |
| Light              | Constant heat flux solid boundary, Constant heat flux solid boundary, | 75 W | 75 W |

Based on the above characteristic, Airpak 3.0 was introduced to simulate the indoor temperature and airflow distributions in this study. Fig. 2 presents the physical CFD model used in this research, whose dimensions are 6 m (length) × 4 m (width) × 3.5 m (height). This is a typical office model, which sets up lighting, equipment, occupants, desks and chairs to simulate the real scene of the office. Similar structures have been used in stratrum ventilation studies for many times, and the accuracy of the model has been verified [17,20]. Standard k-ε model was employed to simulate the air temperature and airflow on account of its accuracy and slightly higher computational efficiency [31]. The buoyancy effect was considered by the Boussinesq model [32]. The air in the test room was assumed to be an incompressible Newtonian fluid, and the fluid acted as a continuous flow. The air outlets were set as outflow. The size of the supply air outlets was changed to ensure the supply air velocity remain the same, thus to rule out the effect of velocity on the conclusions. Occupants and lights were defined as constant heat flux as the solid boundary condition. The power of occupants and lights were 100 W and 75 W respectively, which was accepted by the similar studies [24]. The walls were modeled to be constant temperature according to the measured values observed in the experiment to consider the influence of the outdoor environment [33]. The detailed boundary conditions are provided in Table 1.

A grid independence test was performed to come up with the number of elements sufficient to generate accurate results. The average temperature at the height of 1.1 m calculated by coarse grid, moderate grid and fine grid was compared and presented in Table 2. Finally, the Mesh 2 was selected since it resulted in maximum relative error in computing values of temperature lower than 5% compared to Mesh 3 (Table 2).

2.2. Building energy simulation

Building energy modeling (BEM) has come a long way in its integration with other tools for HVAC system simulation and sizing. BEM software can be used to design building HVAC systems to satisfy loads within a building and also predict building energy use. Many energy performance analysis models are available, such as Trane TRACE, DOE-2, eQuest, EnergyPlus, and TRNSYS etc. Due to the long development history supported by DOE and the global acceptance by engineers and researchers, EnergyPlus [34] in conjunction with OpenStudio was selected in this study. In this paper, twenty typical cities located in four climate zones with heating requirement, namely severe cold A/B, severe cold C, cold A/B, and hot summer and cold winter A/B (shorten as HSCW A/B) were selected. Due to that heating is not required in the hot summer and warm winter climate zone (HSWW), it was not involved in the energy consumption calculation. The computer model of an office in each selected city was simulated as a ten-story building with a gross floor area of 23230 m² (2323 m² each floor) in EnergyPlus. The detailed locations of the selected cities and corresponding 3D model are provided in Fig. 3. The related envelope thermal parameters in different climate zones were set according to Chinese Building Design Standard [35]. The detailed input parameters are presented in Table 3. The calculation accuracy of the model has been verified in another study [36].

2.3. Study cases

Typically, parallel jets include three distinct regions as presented in Fig. 4, i.e. the converging region, merging region, and combined region. The converging region corresponds to the initial stage of interaction between two jets, and recirculation exists. At the end of converging region, the mean velocity along the axis of symmetry is zero and is known as the merging point. Two jets start to merge after the merging point. The point at which the flow behaves in a manner similar to that of a single jet is termed as the combining point, and the mean velocity at the symmetry line reaches its maximum at this point. The flow between merging point and combining point is defined as the merging region.

The turbulent diffusion basic equation of constant velocity flow can be expressed as:
\[
\rho u_0 \frac{\partial \theta}{\partial y} = G_{\theta T} - \delta \frac{\partial}{\partial y} \left[ \delta \left( \rho v' \theta' \right) \right]
\]

(1)

\[
G_{\theta T} = \rho v' \theta'
\]

(2)

\[
\theta \propto L \frac{\partial \theta}{\partial y}
\]

(3)

where \(\theta\) is temperature, \(K\); \(\theta'\) is temperature fluctuation, \(K\); \(\rho\) is density, \(kg/m^3\); \(u_0\) is centerline velocity, \(m/s\); \(v'\) is transverse velocity pulsation, \(m/s\); \(G_{\theta T}\) is the turbulent diffusion flow through the fluid along the jet direction. \(\delta\) equals to 0 corresponds to the plane case while \(\delta\) equals to 1 corresponds to the axial symmetry case. Eq. (3) can be derived from Eq. (1) and Eq. (2) and \(L\) is the mixing length, m.

It is widely known that dense supply air outlets can realize the uniform flow distribution more easily and bring higher thermal comfort. However, dense supply air outlets can also cause the serious energy dissipation problems, and then lead to the waste of energy consumption and indoor environment disorder. Besides, the short circuit probability will also increase when the outlets are located directly below the inlets. Hence, a reasonable PJS is worth investigating for achieving thermal performance and energy efficiency simultaneously in a low-cost way, especially for stratum ventilation whose supply air outlets are close to the occupied zones.

Considering the inlet location can obviously affect the determination of the optimum PJS, four air inlet positions were discussed as presented in Fig. 5. The air inlets are respectively on the ceiling (shorten as CE), at the height of 0.5 m on the same wall as the air outlets (shorten as SA-0.5), at the height of 0.5 m on the opposite wall of the air outlets (shorten as OP-0.5) and at the same level opposite to the air outlets (shorten as OP-1.2).

For each air inlet position, eight cases with different PJSs were carried out (Cases 1–8) under both heating and cooling conditions. The PJS was changed from 0.6 m to 2.0 m, which is commonly used in the small and medium spaces. To ensure the supply air velocity remain unchanged to exclude its impact on the conclusions, the size of the supply air outlets was varied to adapt to the change of PJS. The detailed parameters of the studied cases are listed in Table 4.

Table 3

| Description        | Parameter | Severe cold A/B | Severe cold C | Cold A/B | HSCW A/B |
|--------------------|-----------|-----------------|---------------|----------|----------|
| Physical details   | No. of stories | 10              | 10            | 10       | 10       |
|                    | Area per floor (m²) | 2323            | 2323          | 2323     | 2323     |
|                    | Conditioned area per floor (m²) | 2323            | 2323          | 2323     | 2323     |
|                    | Floor to floor height (m) | 2.4             | 2.4           | 2.4      | 2.4      |
|                    | Heat transfer | 0.38            | 0.43          | 0.5      | 0.6      |
|                    | Wall coefficient | 0.28            | 0.35          | 0.45     | 0.4      |
|                    | Roof coefficient | 2.20            | 2.30          | 2.40     | 2.50     |
| Envelope details   | Shading coefficient (SC) | 0.45            | 0.45          | 0.45     | 0.45     |
|                    | Window to wall ratio | 0.40            | 0.40          | 0.40     | 0.40     |
| Internal load      | Occupancy density (m²/person) | 10              | 10            | 10       | 10       |
|                    | Lighting power density (W/m²) | 9               | 9             | 9        | 9        |
|                    | Equipment power density (W/m²) | 15              | 15            | 15       | 15       |

Fig. 3. The locations of the selected cities and the 3D model.

Fig. 4. Flow camber along the jets’ axis.
2.4. Assessment criteria

In this study, the following assessment indexes were used to explore the influence of PJS on the stratum ventilation performance.

(1) Air distribution performance index (ADPI);
(2) Ventilation effectiveness (Et);
(3) Economic comfort coefficient.

To describe the uniformity of the synergistic effect of the velocity and temperature for stratum ventilation, Lin proposed a new formula of the EDT for stratum ventilation recently, which is found to be reliable and straightforward in evaluation of the performance in thermal comfort [37,38]. Just like PMV, EDT can also consider the effect of temperature, velocity, radiant temperature, relative humidity and human activity etc. on the thermal comfort. The calculation formula is written as [39]:

$$\text{EDT} = (T_x - T_o) - (v_x - 1.1)$$  \(\text{(4)}\)

where EDT is the effective draft temperature, K; $T_x$ is the dry-bulb temperature of evaluation point, °C; $T_o$ is the occupied zone average temperature, °C and $v_x$ is the centerline speed of evaluation point, m/s. The thermal environment can be considered to be satisfactory if the EDT is between −1.2 K and +1.2 K.

Air distribution performance index (ADPI) as presented in Eq. (5) was explicitly modified to apply to the stratum ventilation and introduced to assess the indoor thermal comfort in this study [23].

$$\text{ADPI} = \frac{N_c}{N} \times 100\%$$  \(\text{(5)}\)

where $N$ is the total number of calculated nodes in the occupied zone; $N_c$ is the number of calculated nodes that EDT and $v$ are both within the acceptable range. A larger ADPI indicates that the indoor environment is more comfortable and acceptable.

Ventilation effectiveness was introduced to estimate the energy efficiency. A higher $E_t$ demonstrates that the ventilation method is more energy saving. Its definition is shown as follows [40]:

$$E_{tw} = \frac{(T_s - T_e) - (T_o - T_s)}{N}$$  \(\text{(6)}\)

$$E_{ts} = \frac{(T_e - T_s) - (T_o - T_s)}{N}$$  \(\text{(7)}\)

where $E_{tw}$ is the ventilation effectiveness under heating condition and $E_{ts}$ is the ventilation effectiveness under cooling condition; $T_s$ is the supply air temperature, °C and $T_e$ is the exhaust air temperature, °C.

Economic comfort coefficient is used to evaluate the energy consumption and indoor comfort comprehensively, which is proposed and proved by Kong et al [32]. The physical meaning of economic comfort coefficient is the energy consumption used to realize the thermal comfort for one person served by the studied ventilation method. The calculation process is presented as below:

The number of the comfortable nodes ($N_c$) is defined as:

\begin{table}[h]
\centering
\begin{tabular}{cccc}
\hline
Case & PJS (m) & The number of the air outlets & The size of the air outlets (m × m) \\
\hline
1 & 0.600 & 9 & 0.141 × 0.141 \\
2 & 0.667 & 8 & 0.150 × 0.150 \\
3 & 0.750 & 7 & 0.160 × 0.160 \\
4 & 0.857 & 6 & 0.173 × 0.173 \\
5 & 1.000 & 5 & 0.190 × 0.190 \\
6 & 1.200 & 4 & 0.212 × 0.212 \\
7 & 1.500 & 3 & 0.245 × 0.245 \\
8 & 2.000 & 2 & 0.300 × 0.300 \\
\hline
\end{tabular}
\caption{The detailed parameters of the studied cases.}
\end{table}

Fig. 5. Different air inlet positions (a: SA-0.5; b: OP-0.5; c: OP-1.2; d: CE).
Ventilation is used to eliminate the residual heat and humidity in the room, thereby generating energy consumption. Ventilation effectiveness is also called the energy utilization efficiency. When the same energy is sent into the room through different ventilation types, the energy that ends up being used efficiently is different. For the non-uniform indoor environment, the commonly used lumped parameter method is no longer applicable. Hence, the energy consumption introduced in this study is used to evaluate the energy utilization of the treated air sent into the room from the outlets, which is independent of the HVAC system. Thereby, the total energy consumption of the studied ventilation method can be calculated as:

\[ W = \sum_{n=1}^{N} Q_e \times E_t \times ADPI \times N \quad (9) \]

where \( W \) is the energy consumption used to maintain a comfortable indoor environment under the studied ventilation method, kWh; \( Q_e \) is the hourly cooling (heating) demand of the buildings, kW; \( t \) is the cooling (heating) time, h, \( t = 1, 2, 3, \ldots, n \).

According to the physical meaning of the economic comfort coefficient (\( \eta \)), it is defined by the following equations:

\[ \eta = \frac{W}{N_e} = \frac{\sum_{n=1}^{N} \frac{Q_e}{ADPI} \times N}{\sum_{n=1}^{N} Q_e} \times \frac{1}{Et \times ADPI} \quad (10) \]

\[ \alpha = \frac{\sum_{n=1}^{N} Q_e}{N} \quad (11) \]

\[ \beta = \frac{1}{Et \times ADPI} \quad (12) \]

For one building, \( \alpha \) is only concerned with the building thermal characteristics, and has nothing to do with the ventilation methods. Hence, it can be defined as a constant in this study. \( \beta \) is the ventilation
correction factor, which depends on the ventilation performance. Hence, Economic comfort coefficient can be divided into two parts, one part is related to the building thermal characteristics (α) and the other part is connected with ventilation method (β). It can be used to comprehensively assess the energy consumption and indoor thermal comfort, which are two aims that contradict each other in practice.

3. Results

3.1. Model validation

3.1.1. Test procedure

In this paper, experiments were used to validate the CFD model. The experiment chamber is located in Tianjin, China. The test was conducted in an environmental chamber with a size of 6 m (length) × 4 m (width) × 3.5 m (height). Fig. 6 provides the schematic diagram of the air treatment system. Dry bulb temperature sensor and flow nozzle were used to measure the supply air temperature and velocity with the accuracy of ±0.1 °C and ±2%, respectively.

The treated air was supplied through the air outlets (S1-S2) with a size of 0.30 m × 0.30 m. S1 and S2 were installed at the height of 1.2 m on the side wall. The treated air was returned via the air inlet (R1) located at 0.5 m above the floor, and its size was 1.05 m × 0.30 m. Six measurement lines (L1-L6) were evenly distributed throughout the test chamber. The positions of the measurement lines are presented in Fig. 7.

Every measurement line was equipped with temperature and velocity probes at the height of 0.1 m, 0.6 m, 1.1 m, 1.4 m, 2.1 m and 2.8 m, respectively. Six occupants and two incandescent lamps were placed in the test chamber for simulating the internal heat sources. A cuboid thermal manikin of 0.4 m × 0.25 m × 1.2 m was employed to simulate the sedentary occupant, which is widely used in the indoor thermal environment experiments [41]. The method has been proved that it is accurate enough to reflect the occupant influence on the thermal environment. A light bulb with a power of 100 W was used to simulate the thermal manikin heat release, which can consider the thermal radiation effect and heat dissipation simultaneously. Two incandescent lamps with the power of 75 W were installed on one of the ceiling.

3.1.1.1. Measurement instrumentation. In this study, dry-bulb temperature and air velocity were measured and collected by Agilent Data acquisition. An error analysis method proposed by Kline and McClintock was introduced to assess the measurement instrumentation accuracy for the directly measured variants [42]. The relative uncertainty is described by Equation (13):

\[ E_r = \frac{(T_o - T_i)}{(T_o - T_r)} \]  

where \( R \) is the variant and \( X \) is the independent variant. By calculating, the measurement accuracy of temperature and velocity in this study are 0.25 % and 0.08 % respectively, which is acceptable for the similar research.

3.1.1.2. Model validation. The boundary conditions used in the CFD model were consistent with the experiment (c.f. Table 1). To validate the accuracy of the numerical models, the simulated airflow and temperature were compared with the measured values obtained in the experiment. Fig. 8 presents the comparisons of the temperature and velocity at the height of 1.1 m between the simulation results and experimental values. To quantitatively assess the performance of the model, four commonly used statistical methods were employed. They are the mean absolute error (MAE), the root-mean-square error (RMSE) [43], and the index of agreement (IA) [44].

The definition of the MAE is as follows:

\[ MAE = \frac{1}{p} \sum_{i=1}^{p} |x_i - y_i| \]  

where \( x_i \) represents the experimental values, \( y_i \) represents the simulated values, and \( p \) is the number points in the experimental test. The definition of the MAPE is as follows:

\[ MAPE = \frac{1}{p} \sum_{i=1}^{p} \left( \frac{|x_i - y_i|}{|x_i|} \right) \times 100\% \]  

The RMSE is as follows:

\[ RMSE = \sqrt{\frac{1}{p} \sum_{i=1}^{p} |x_i - y_i|^2} \]  

The IA is as follows:

\[ IA = 1 - \frac{\sum_{i=1}^{p} |x_i - y_i|^2}{\sum_{i=1}^{p} (|x_i| + |y_i|)^2} \]
When the simulation results are in complete agreement with the experimental results, the MAE, MAPE and RMSE should be 0, and IA should be 1. Table 5 provides the calculation results of the temperature and velocity based on the four statistical indicators. The data indicates that the performance of the model used in this study is better than that employed in the similar research [24]. Hence, the numerical model is accurate enough for the further study.

### 3.2. Impact analysis of PJS on ADPI

A higher ADPI represents the airflow distribution is more reasonable, thus to build a more comfortable indoor environment. Dense supply air outlets can make the air distribution more uniform, but raise the energy dissipation between the parallel jets simultaneously. Meanwhile, the risk of short-circuiting also increases under certain inlet locations, such as SA-0.5, OP-1.2 and CE. Besides, the airflow path obviously affects the ADPI as well. Hence, the impact of PJS on ADPI is an integrated result of various actions: air distribution uniform, parallel jet interaction, short-circuiting probability and airflow path. The heating and cooling condition were both discussed in the following sections.

#### 3.2.1. Heating condition

Under heating condition, the airflow will float upwards with the influence of thermal buoyancy. Hence, short-circuiting problem is more likely to happen when the inlets are located on the ceilings (CE), followed by OP-1.2 and SA-0.5. Airflow path determines the environmental regulation effect and draft sensation in the occupied zone. As presented in Fig. 9, it is apparent that the ADPI fluctuates significantly along with PJS due to that the main influencing factors are various at different stages. For SA-0.5, the ADPI reaches the peak with the PJS being 0.75 m. It means that the synergy effect caused by the mentioned factors can strike a balance when the PJS is 0.75 m. For OP-0.5, the two peaks of ADPI appear when PJS is 0.54 m and 0.79 m respectively. It is due to that the short-circuiting is hard to happen under OP-0.5. Besides, the airflow path is more reasonable than the other inlet locations under heating condition. Similarly under SA-0.5, the peak of ADPI is obtained with PJS being 0.76 m under OP-1.2 and 0.78 m under CE. Only considering the air distribution rationality and indoor comfort, the optimum PJSs for SA-0.5, OP-0.5, OP-1.2 and CE is 0.75 m, 0.54 m, 0.76 m and 0.78 m, respectively. Meanwhile, the derivative curve indicates that the ADPI changes dramatically at certain PJSs. In the actual design, the following PJSs should be avoided. For SA-0.5, the sensitive PJSs are 0.6 m, 0.84 m, and 1.39 m; for OP-0.5, the sensitive PJSs are 0.5 m, 0.6 m, and 0.71 m; for OP-1.2, the sensitive PJSs are 0.6 m, 0.84 m, and 1.39 m; for CE, the sensitive PJSs are 0.5 m, 0.7 m, and 0.91 m.

### Table 5

The performance of the model built with Airpak.

| Metric | Temperature | Velocity |
|--------|-------------|----------|
|        | Winter      | Summer   | Winter | Summer |
| MAE    | 0.2 °C      | 0.2 °C   | 0.02 m/s | 0.01 m/s |
| MAPE   | 1.1%        | 0.8%     | 8.11% | 8.46% |
| RMSE   | 0.2 °C      | 0.2 °C   | 0.02 m/s | 0.01 m/s |
| IA     | 0.95        | 0.92     | 0.93  | 0.96   |

![Fig. 9. ADPI with different PJSs under winter condition.](image-url)
3.2.2. Cooling condition

Under cooling condition, the airflow will sink as a result of thermal buoyancy. Due to the supply air outlets are installed at the height of 1.2 m, the downdraft can go through the occupied zone by removing the adverse heat dissipation. A larger PJS can weaken the interaction between the parallel cold airflow, thus to increase the temperature difference between non-isothermal jets and surrounding air. Hence, as shown in Fig. 10, the ADPI reaches the peak when the PJS is 2.0 m under all the discussed inlet locations. Similarly, the ADPI derivative curve under cooling condition presents that the following PJSs should be avoided to reduce the ADPI sensitivity. For SA-0.5, the sensitive PJSs are 0.6 m, 0.76 m, and 0.95 m; for OP-0.5, the sensitive PJSs are 0.6 m, 0.71 m, and 0.83 m; for OP-1.2, the sensitive PJSs are 0.5 m, 0.7 m, and 0.95 m; and for CE, the sensitive PJSs are 0.5 m, 0.7 m, 0.8 m and 0.95 m. Compared with those under heating condition, most sensitive distances are shifted backwards under cooling condition.

3.3. Impact analysis of PJS on ventilation effectiveness

3.3.1. Heating condition

For the ventilation effectiveness under multiple-jet stratum ventilation, it is mainly affected by the energy dissipation between the parallel jets, airflow path and short-circuiting probability. Reasonably designing the PJS can make more energy stay among the occupied zone, and then realizing the energy conservation.

3.3.1.1. Cooling condition

Contrary to the heating condition, the airflow sinks due to the higher density. The factors caused by PJS that mainly affect the ventilation efficiency under cooling condition are still: parallel jet interaction, non-isothermal jet temperature difference and short-circuiting probability. There is a large difference of the airflow characteristic between the cooling condition and heating condition. As provided in Fig. 12, the optimum PJS for SA-0.5, OP-0.5, OP-1.2 and CE is 2.0 m, 1.33 m, 1.50 m and 2.0 m under cooling condition. Similar to the results under heating condition, no significant differences can be

Fig. 10. ADPI with different PJSs under summer condition.
found in ventilation efficiency when PJSs are changing from 0.5 m to 2.0 m under OP-0.5, OP-1.2 and CE. Under SA-0.5, the ventilation efficiency with PJS being 0.5 m is the lowest (0.797) while reaches 0.927 with PJS being 2.0 m, enhanced by 16.4 % due to the “airflow short circuit”. Interestingly, PJS is also observed to show different effect degrees on ADPI and ventilation effectiveness under both heating and cooling modes. Thereby, a comprehensive analysis is necessary to determine the optimum PJS for the multiple-jet stratum ventilation in offices.

4. Discussions

4.1. Sensitivity analysis

According to the previous analysis, the results indicate that the impact of PJS on the indoor comfort and ventilation efficiency is different under heating and cooling conditions due to the various airflow characteristic. To explore the effect degree of PJS on the key parameters of the multiple-jet stratum ventilation, sensitivity analysis was introduced in the following. Sensitivity coefficient is used to quantize the impact degree. The related calculate equation is presented as below.

\[
S_c = \sum_{i=1}^{n} \frac{(y_i - \bar{y})}{\bar{x} - x_i} / n
\]

where \(S_c\) is the sensitivity coefficient; \(x_i\) is the independent variables; \(y_i\) is the dependent variables; \(n\) is the number of variables. In this study, PJS is the independent variable, ADPI and ventilation efficiency are the dependent variables. \(S_c\) can reflect the effect degree of independent variables on the dependent variables. When \(S_c\) of the evaluation factor is larger than 1.0, the factor can be seen as sensitive factor.

As shown in Fig. 13, it can be concluded that: a) the impact of PJS on the ventilation efficiency is less than that on the ADPI; b) the impact of PJS on the ventilation efficiency is largest with the air inlet position being SA-0.5, and least with the air inlet position being CE under both heating and cooling conditions; c) but there is an obvious difference existing in the impact of PJS on the indoor comfort between heating and cooling conditions. Under heating condition, the effect is largest when the air inlet position is CE. While the effect shows largest with the air inlet position being SA-0.5 under cooling condition.

4.2. Optimum PJS for multiple-jet stratum ventilation

By analyzing, the effect of PJS on the energy utilization and indoor comfort is conflicting. Higher indoor comfort and more uniform airflow distribution need more energy consumption. In addition, air flow characteristic difference due to the thermal buoyancy and air inlet positions also make the impacts various. To enhance the practicality and applicability of the research conclusions, economic comfort coefficient as the comprehensive indicator is employed to obtain the optimum PJS for multiple-jet stratum ventilation in the following section.

Economic comfort coefficient is proposed to balance the indoor comfort and energy conservation of stratum ventilation in this study (c.f. equations 8–12). For the same research building, ventilation correction factor \(\beta\) only depends on the performance of the served ventilation method. A low \(\beta\) demonstrates that less energy is consumed to realize the thermal comfort for one person under the studied ventilation. Taking the ventilation correction factor \(\beta\) as the optimization objective, the
optimum PJSs of four air inlet positions under heating and cooling conditions were obtained. As shown in Fig. 14, it can be seen that air distribution characteristic under heating condition is much more complex than that under cooling condition. Compared with hot air, the cold air is easier to be delivered to the occupied zone. Therefore, the optimum PJSs of SA-0.5, OP-0.5, OP-1.2 and CE under heating condition are 1.28 m, 1.38 m, 1.41 m and 1.46 m, while those under cooling condition are all 2.0 m. It means that the energy consumption used to realize the thermal comfort for one person is the least with the mentioned PJSs under the corresponding ventilation methods.

As shown in Fig. 14, the optimum PJSs are different under cooling and heating conditions for the same air inlet position. In practice, most air conditioning system usually runs all the year round. Hence, the impact of PJS on the heating and cooling performance of multiple-jet stratum ventilation should be considered simultaneously. The year-round economic comfort coefficient ($\eta_y$) can be defined as follows:

$$\eta_y = \sum_{n=1}^{N} Q_{ew} \times \beta_w + \sum_{i=1}^{N} Q_{es} \times \beta_s$$

$$\lambda = \sum_{n=1}^{N} Q_{ew} / \sum_{i=1}^{N} Q_{ew}$$

$$\eta_y = \sum_{n=1}^{N} Q_{ew} \times (\beta_w + \lambda \times \beta_s)$$

$$\beta_{\lambda} = \beta_w + \lambda \times \beta_s$$

where $\sum_{n=1}^{N} Q_{ew}$ is the cumulative heating load of the research building; $t_e$ is the heating time, $t_e = 1, 2, 3...; \sum_{i=1}^{N} Q_{es}$ is the cumulative cooling load of the research building; $t_c$ is the cooling time, $t_c = 1, 2, 3...; \lambda$ is the building load weight coefficient, which is strongly related to the climatic conditions.
Fig. 14. Optimum PJS for multiple-jet stratum ventilation.
zones where the building is located and with the building passive thermal characteristics; $\beta_n$ is the ventilation correction factor under heating condition, $\beta_s$ is the ventilation correction factor under cooling condition, and $\beta_y$ is the year-round ventilation correction factor.

According to the meteorological parameter characteristics of the five climate zones, corresponding building load weight coefficients can be obtained by building energy simulations. The year-round ventilation correction factors for the multiple-jet stratum ventilation used in offices located in different climate zones can be calculated according to the following equation:

$$\beta_y = \begin{cases} 
\beta_n + \lambda_{SC-A/B} \times \beta_{forseverecoldA/\text{Bclimatezone}} \\
\beta_s + \lambda_{SC-C} \times \beta_{forseverecoldC/\text{climatezone}} \\
\beta_n + \lambda_{A/B} \times \beta_{forcoldA/\text{climatezone}} \\
\beta_s + \lambda_{HSCW} \times \beta_{forHSCWA/\text{climatezone}} \\
\beta_n \times \lambda_{forHSSWA/\text{climatezone}} 
\end{cases}$$

(23)

where $\lambda_{SC-A/B}$, $\lambda_{SC-C}$, $\lambda_{A/B}$, and $\lambda_{HSCW}$ are the building load weight coefficients for the severe cold A/B climate zone, severe cold C climate zone, cold A/B climate zone, and HSCW A/B climate zone, respectively. Since heating is not needed in the hot summer and warm winter (HSWW) climate zone, the year-round ventilation correction factor of HSSW is equal to the ventilation correction factor under cooling condition.

Based on the building load weight coefficients, the optimum PJSs for offices with different air inlet positions located in five climate zones can be calculated by nonlinear fitting technique. The results can be used to guide the practical design and application of the multiple-jet stratum ventilation, which is provided in Fig. 15. It is noted that the energy conservation measures are not considered in building load weight coefficients, therefore the energy saving line for cooling and heating is added in Fig. 15 to point to the change direction of the optimum PJSs.

5. Conclusions

This study focused on the impact of parallel jet spacing (PJS) on the thermal performance of multiple-jet stratum ventilation used in offices. A total of eight PJSs, four inlet locations and five climate zones were discussed and studied comprehensively. The research results indicated that the PJS can affect the indoor thermal comfort and ventilation efficiency with significant seasonal differences. PJS mainly affects the thermal comfort under heating condition while affects the energy utilization efficiency under cooling condition. To assess the indoor thermal comfort and energy utilization efficiency simultaneously, the economic comfort coefficient was introduced to obtain the optimum PJSs for spacing cooling and heating under various ventilation methods. Considering the air conditioning systems are usually whole year operation in practical applications, the year-round optimum PJSs for the multiple-jet stratum ventilation in different climate zones were discussed by introducing building energy simulation. Some meaningful data and conclusions were subsequently obtained. It is therefore a better choice for multiple-jet stratum ventilation design if the performance requirements are met. The related research results can provide a comprehensive guideline for multiple-jet stratum ventilation in the practical applications and enhance its performance in a low-cost way. In addition, there is room for further improvement in this study: indoor geometric structure and furniture placement will affect air flow organization, and more accurate geometric conditions will help to calculate more accurate optimal PJS. In this paper, the general applicability of typical buildings is obtained, and the optimal PJS of special climate regions (such as HSSW region) should be improved in the following research. The ultimate goal of all of these is to construct and improve the evaluation standard with PJS as the evaluation center, which is also the future direction of this study.

CRediT authorship contribution statement

Han Li: Conceptualization, Methodology, Formal analysis, Investigation, Data curation, Writing – original draft. Zheng Fu: Software, Validation, Writing – review & editing. Chang Xi: Investigation, Software, Resources. Nana Li: Investigation, Software, Resources. Wei Li: Investigation, Software, Resources. Xiangfei Kong: Conceptualization, Methodology, Funding acquisition, Writing – review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgment

The work presented in this paper is financially supported by Natural Science Foundation of China (Grant No. 52008147), Natural Science Foundation of Hebei Province (No. E2019202452), Fundamental Research Funds of Hebei University of Technology (Project No. JBYKTD2003), S&T Program of Hebei (Project No. 216Z4502G) and Hebei Province Funding Project for Returned Scholars, China (Project No. C20190507).

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