

Article

CFD-Based Optimization of Air Conditioning Airflow Organization and Thermal Environment of Atrium–Corridor Spaces in an Office Building

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Abstract

To improve the indoor thermal comfort of embedded atriums and corridors in office buildings during summer, this study aims to optimize air conditioning airflow organization in atriums using computational fluid dynamics (CFD) simulations. Field measurements were carried out to collect air parameters, which were subsequently used to validate the established CFD model. Taking a six-story office building in Xi'an as the research subject and stratified air conditioning as the baseline case, this study investigated the effects of air inlet layout, air inlet type, and air volume distribution on the indoor thermal environment. The results revealed significant vertical temperature stratification within the atrium, with average temperatures ranging from 23.5 °C to 46.1 °C. Based on comparative analysis of multiple optimization scenarios, the following conclusions are drawn: adopting swirl diffusers in the corridors with an air inlet quantity ratio of 1:1:1:1:2 from the first to fifth floors, combined with uniform air supply volume across the first to fourth floors, can maintain the average Predicted Mean Vote (PMV) of each floor within the range of −0.1 to 0.3. Conversely, excessive air supply volume on upper floors and insufficient air supply volume on lower floors significantly degrade the corridor thermal comfort.

Keywords: embedded atrium; airflow organization; computational fluid dynamics; indoor thermal comfort

1. Introduction

The architecture layout, with the atrium providing natural lighting, ventilation and inter-floor connectivity, features good practicality and aesthetic performance. As a multifunctional public activity space, the atrium, together with its surrounding corridors, generally enjoys a high utilization rate. However, atrium air-conditioning systems face prominent challenge: significant vertical temperature gradients and overheating at the upper floors in summer, which are induced by intense solar radiation through the skylight and thermal buoyancy effect. A reasonable airflow organization is therefore crucial for maintaining indoor thermal comfort in atriums.

Existing studies have carried out field measurements and simulations on the thermal environment of atriums involving different regions and building types, and widely confirmed the common problems of thermal stratification and thermal discomfort in such spaces. Takemasa et al. [1] established a simplified unsteady model to predict the vertical temperature distribution in atriums with skylights and proposed an occupied-zone cooling



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strategy. Laouadi et al. [2] measured indoor temperature of an enclosed three-story building with a pyramidal skylight in Canada and found that temperature stratification was particularly remarkable on sunny days when the air-conditioning system was switched off. For atriums in severely cold regions, Yan [3] pointed out that obvious vertical temperature gradients and considerable horizontal thermal non-uniformity occur in both winter and summer, and the temperature field is more uniform in winter than in summer. Hu et al. [4] addressed the vertical temperature gradient problem of shopping mall atriums and formed a full-cycle solution covering design to operation, which realizes a balanced temperature field among all atrium floors. Ge et al. [5] performed a field investigation on semi-enclosed shopping mall atriums in China's hot summer–cold winter climate zone, combining thermal parameter monitoring with 418 valid subjective questionnaires. They found that glass ceiling atriums presented a vertical temperature gradient of $0.20\text{ }^{\circ}\text{C}/\text{m}$ in summer, 50% higher than that of non-glass atriums, and 73% of occupants on upper floors reported warm or hot. Hou [6] conducted field tests on the thermal environment of the atrium in a six-story shopping mall in Changzhou, Jiangsu Province; they found that, on typical summer days, the average temperature on the top floor was $6.3\text{ }^{\circ}\text{C}$ higher than that on the first floor. Xu et al. [7] investigated the winter thermal environment of a teaching building atrium in Tianjin via field measurements and 70 valid thermal comfort questionnaires. The measured results showed a vertical temperature gradient of $0.49\text{ }^{\circ}\text{C}/\text{m}$. Shi et al. [8] established a year-round experimental database including on-site monitoring data and 2220 valid questionnaires for a library atrium and confirmed that obvious seasonal vertical temperature stratification caused by the greenhouse effect and stack effect.

Another research direction concerning the thermal environment of atriums lies in passive optimization design. Focusing on natural ventilation and shading regulation, Abdullah et al. [9] conducted field measurements in a hotel atrium in southern China in summer. The results indicated that high-performance integrated shading can reduce direct solar radiation at the atrium top, and the evaporative spray system can cool the glass surface, thereby effectively mitigating the overheating issue at the upper atrium. Pan et al. [10] verified via computational fluid dynamics (CFD) simulation that atrium height significantly affects vertical temperature distribution for atriums with the same floor area and geometric shape, which enriches the research on atrium geometric parameters. In terms of shading and geometric optimization, Pichatwatana et al. [11] adopted Thermal Analysis Simulation (TAS) simulation and found that ceiling-level shading devices achieve better thermal performance than curved-roof atriums without such facilities. Berardi et al. [12] simulated and evaluated the shading system design of residential daylighting atriums built in 1960s under different daylight indices, providing a reference for the optimal design of window shading and glass transmittance. Tabesh et al. [13] reviewed various atrium forms and suggested that reasonable design should fully consider energy performance, ventilation effect and daylight factor. Amani [14] used ECOTECT software and found four-sided atriums achieve the highest energy efficiency for residential buildings in summer under hot and dry climatic conditions. Fini et al. [15] investigated the effects of different wall angles on the atrium thermal environment through numerical simulation, further supplementing the research on atrium geometric design. Targeting low-energy office buildings in Malaysia, Moosavi et al. [16] found that cross ventilation can effectively improve the indoor thermal environment of atriums. Meanwhile, the combined passive cooling strategy integrating water walls, thermal flues and cross ventilation can remarkably alleviate atrium overheating.

Focusing on energy-efficient integrated passive strategies, references [17–20] comprehensively adopted EnergyPlus, Ecotect, and CFD-based tools to explore the effects of seasonal operation modes, architectural design, skylight configuration and spatial form

on atrium thermal performance, energy saving and natural ventilation, and further established evaluation frameworks and provided practical design optimization references. Hou [6] conducted CFD simulations and demonstrated that thermal stratification within daylighting atriums is significantly affected by outdoor temperature and solar radiation intensity. Both natural ventilation and roof greening are effective measures to reduce the temperature in the upper zone of the atrium. Mirzabeigi et al. [21] proposed a heat stress index for the thermal comfort evaluation of atrium spaces exposed to solar radiation.

Regarding active regulation strategies, existing studies have adopted various simulation models to analyze airflow characteristics and atrium air-conditioning systems. Li et al. [22] compared three air-supply modes, and identified the “middle supply, lower return, upper exhaust” scheme as the optimal airflow organization for atriums, while completing the optimization of stratified air-conditioning design. Wang et al. [23] coupled dynamic thermal models with CFD methods to investigate the effects of opening area ratio and air inlet arrangement (location and configuration) on the thermal performance and airflow organization of a three-story atrium under pressurized ventilation. Hussain et al. [24] numerically studied the thermal performance of a three-story atrium in a Canadian building equipped with a hybrid solar-assisted natural ventilation system. They also validated multiple Reynolds-Averaged Navier–Stokes (RANS) turbulence models coupled with the Discrete Transfer Radiation Model (DTRM) for the simulation of radiative heat transfer. Zhang et al. [25] adopted the PHOENICS software to simulate the air temperature distribution inside the 72 m high atrium and corridors of a hotel in Shenzhen. Better indoor thermal performance was achieved by optimizing the layout and air supply parameters of air inlets. Huang [26] investigated the atrium of a shopping mall located in the Pearl River Delta using PHOENICS, aiming to optimize its summer stratified air-conditioning scheme. The results showed that ventilation corridors and roof greening can effectively reduce the top-floor temperature of the atrium, offering practical guidelines for the thermal design of shopping mall atriums. Yan [3] found that mechanical ventilation performs better than natural ventilation in improving the indoor thermal environment and airflow distribution of atriums. Ferrucci et al. [27] pointed out that integrating CFD-derived airflow and temperature fields with thermal comfort indices is essential for the comprehensive performance evaluation of large-scale atriums. Shi et al. [8] established a TRNSYS-CONTAM co-simulation model for a library atrium to predict vertical temperature distribution and cooling load demands of each floor. On this basis, they adjusted the floor-by-floor air supply rate using a correction coefficient to narrow the vertical temperature difference, and further proposed a proportional air supply regulation strategy to improve the overall thermal environment of atrium buildings. Su et al. [28] investigated the thermal environment of a large glass-roofed atrium in a high-rise office building in Xi’an, China, by combining transient CFD simulations and full-scale field measurements. They put forward comprehensive improvement measures, including dynamic shading, stratified HVAC parameter optimization, intelligent thermal system regulation, top exhaust systems and low-emissivity glass application, to simultaneously enhance indoor thermal comfort and energy efficiency.

Previous studies have made substantial progress in the multi-objective optimization of atrium thermal environment, daylight performance and energy efficiency through field measurements, numerical simulations, multi-objective optimization algorithms and machine learning methods. Chen et al. [29] developed a data-driven workflow combining the NSGA-II algorithm and machine learning to conduct multi-objective optimization of daylight performance, thermal comfort and energy consumption in teaching building atriums. They identified skylight and window area ratios as the dominant influencing factors and verified the high accuracy of the Light Gradient-Boosting Model. Xu et al. [30] established a climate-adaptive sustainable design framework for single-core commercial

atriums using parametric modeling and the NSGA-II algorithm, focusing on passive morphological optimization to reduce energy consumption and improve photothermal comfort. Xu et al. [31] proposed an efficient machine-learning-coupled multi-objective optimization framework for library atriums in hot summer–cold winter climate zones. With random forest models and the NSGA-II algorithm adopted, they achieved considerable improvements in energy efficiency, thermal comfort and daylight performance, and further classified the optimal design solutions via cluster analysis. Li et al. [32] carried out year-round field measurements on atrium buildings in cold climate regions and proposed integrated shading and ventilation retrofitting strategies. Such strategies effectively reduce cooling load and improved the indoor thermal environment, and a systematic retrofitting framework was accordingly established for existing atrium buildings.

Research directly focusing on embedded atriums and their adjoining corridors remains limited. In the simulation study by Zhang et al. [25], the problem of excessively high temperatures in the upper corridors of the atrium was mitigated by increasing the total air supply volume of fan coil units. Meanwhile, the non-uniform temperature distribution among corridors on each floor was improved by increasing the number of fan coil units and air supply inlets. Hou [6] carried out numerical simulations on summer air-conditioning thermal environment parameters of the atrium and corridors in a commercial building in Changzhou and explored the effects of various air supply combinations under different return air outlet layouts. Their results indicated that the scheme with return air outlets arranged on the side walls of each floor’s air conditioning—side-air supply on the first floor and an overhead air supply from the second to sixth floors—can reduce the vertical temperature gradient and maintain a reasonable air velocity distribution in both the atrium and corridors.

Despite abundant studies on atrium thermal environments, passive design optimization, airflow characteristics, energy simulation, and multi-objective optimization, most researchers mainly focus on independent atrium spaces, with insufficient attention paid to embedded atrium–corridor coupled spaces. Most existing studies mitigate atrium thermal stratification merely through passive strategies such as shading, natural ventilation, or simple atrium air distribution design, while neglecting the airflow organization optimization of multi-story corridors surrounding the atrium. Furthermore, few studies have systematically evaluated the influence of different air supply layout schemes on the overall thermal environment of atrium–corridor spaces. This constitutes a research gap, and there is still a lack of targeted design guidance for air-conditioning system optimization of such multifunctional building zones.

To fill these gaps, this study presents a CFD-based optimization of air conditioning airflow organization and thermal environment for atrium–corridor spaces in an office building. The research adopts a four-step workflow: (1) developing and validating a CFD model for a multi-story office building with an embedded atrium–corridor configuration based on on-site measurements; (2) evaluating the thermal performance of the conventional stratified air-conditioning system in office atriums using thermal comfort indicator; (3) quantifying the effects of air inlet layouts, inlet type, and inter-floor air volume distribution on the thermal environment of atrium–corridor spaces; and (4) proposing an optimized airflow organization strategy to achieve satisfactory thermal comfort in such embedded atrium–corridor office buildings.

2. Methods

In this study, a numerical investigation of air distribution in an embedded atrium was conducted using CFD. This section first introduces the geometry of the atrium and the design of air-conditioning system, followed by the definition of the study cases. Next,

the CFD model, including grid generation, boundary conditions, solar radiation model, and solver settings, is presented. The model was validated through field measurement to ensure its reliability, and thermal comfort was adopted as the key performance indicator for evaluating air distribution.

2.1. Description of Embedded Atrium and Stratified Air Conditioning Design

According to the Design Standard for Energy Efficiency of Public Buildings (GB 50189-2024) [33], a simplified geometric model of an embedded atrium in a typical six-story office building was established, as shown in Figure 1. The atrium dimensions are 25 m (length) $\times$ 15 m (width) $\times$ 16 m (height), with a story height of 4 m and a corridor width of 3 m. There are 24 windows on the south facade, each measuring 4 m $\times$ 1.92 m. The skylight has a size of 12.5 m $\times$ 7.5 m, the glass baffle of the corridors is 0.8 m in height, and the fixed smoke barrier is 1 m in height.

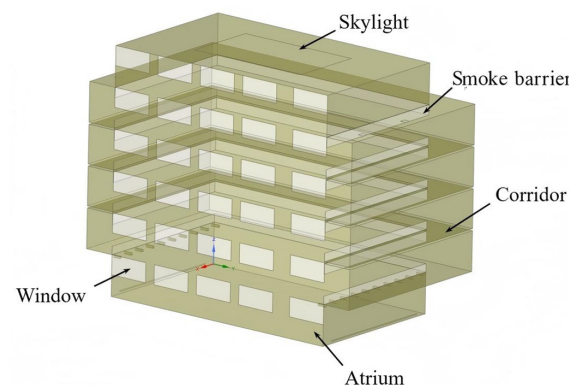


Figure 1. Geometry model of the embedded atrium in a typical six-story office building.

Based on the stratified cooling load calculation method [34], in the cooling load calculation of air-conditioned zones, besides the conventional items adopted in full-room air conditioning load computation, it is necessary to determine the additional cooling load induced by heat transfer from non-air-conditioned zones to air-conditioned zones, including both convective and radiative heat exchange. The cooling load of each air-conditioned zone is calculated by the following equation:

$$q_{c1} = q_{1w} + q_{1n} + q_x + q_f + q_d \quad (1)$$

where q_{c1} denotes the stratified cooling load of the air-conditioned zone, W; q_{1w} denotes the cooling load induced by envelope heat gain of the air-conditioned zone, W; q_{1n} denotes the cooling load from internal heat sources inside the air-conditioned zone, W; q_x denotes the cooling load contributed by outdoor fresh air supply to the air-conditioned zone, W; q_f denotes the cooling load caused by radiative heat transfer from non-air-conditioned zones to the air-conditioned zone, W; q_d denotes the cooling load caused by convective heat transfer from non-air-conditioned zones to the air-conditioned zone, W. Taking Xi'an as the typical representative city for the climatic condition, the summer air-conditioning outdoor design dry-bulb and wet-bulb temperatures are set to 35.1 °C and 25.8 °C, respectively. The indoor design parameters of the air-conditioned area are specified as a dry-bulb temperature of 26 °C and a relative humidity of 60%. The resultant cooling load of the air-conditioned zone is calculated to be 70.13 kW.

The air distribution scheme of the stratified air conditioning is configured as follows: side air supply is adopted in the air-conditioned zone; return air outlets are arranged at the lower part of the conditioned zone, and exhaust outlets are installed at the top of the non-conditioned zone. Rotating spherical nozzles are used for horizontal air supply

in the air-conditioned zone, while single-layer grilles are adopted as return air outlets. According to the calculation method for stratified air distribution, the stratification height is determined to be 3.81 m. A total of 24 air supply inlets are arranged with a spacing of 1.1 m. The spherical nozzles are simplified as square inlets with a dimension of 0.2 m $\times$ 0.2 m.

2.2. Case Description

Nine simulation cases of air distribution for the air-conditioning system were designed in this study to investigate the effects of supply inlet quantity, air supply angles, and floor-by-floor air volume distribution on the thermal environment of embedded atriums and surrounding corridors in office buildings in Xi'an. As listed in Table 1, the base case represents the conventional stratified air-conditioning scenario, with no air supply arranged in the surrounding corridors. Cases D-N1 to D-N3 correspond to different layout quantities of air supply inlets on each corridor floor, while Cases S-N3-V1 to S-N3-V5 are set with different air supply volume distributions across floors.

Table 1. Case setup.

Cases	Quantities of Air Supply Inlets for First to Fifth Floor	Air supply Volume for First to Fifth Floor, m ³ /s	Air Supply Angle of the Corridors
Base case	24, 0, 0, 0, 0	7.73, 0, 0, 0, 0	--
D-N1	4, 0, 0, 4, 16	1.29, 0, 0, 1.29, 5.15	D
D-N2	4, 0, 4, 4, 12	1.29, 0, 1.29, 1.29, 3.86	D
D-N3	4, 4, 4, 4, 8	1.29, 1.29, 1.29, 1.29, 2.58	D
S-N3-V1	4, 4, 4, 4, 8	1.29, 1.29, 1.29, 1.29, 2.58	S
S-N3-V2	4, 4, 4, 4, 8	0.77, 1.16, 1.55, 1.94, 2.32	S
S-N3-V3	4, 4, 4, 4, 8	0.66, 1.11, 1.55, 1.99, 2.43	S
S-N3-V4	4, 4, 4, 4, 8	0.48, 0.97, 1.45, 1.94, 2.90	S
S-N3-V5	4, 4, 4, 4, 8	0.60, 0.95, 1.43, 1.79, 2.98	S

For all cases, the atrium air-conditioning system on the first floor adopts a side-supply and lower-return pattern, whereas the surrounding corridors adopts an up-supply and up-return mode. Figure 2 presents the air-conditioning layout of Case S-N3-V1, where blue arrows denote air supply inlets and red arrows indicate return air outlets on each floor. Air supply inlets are arranged at equal intervals along the corridor length, and all air inlets on the same floor operate at the same flow rate. Both side-air supply inlets and top-air supply inlets adopt the same dimension of 0.2 m $\times$ 0.2 m. For the air supply angle in corridor zones, symbol D represents vertical downward air supply, and S denotes swirl air supply at an angle of 45°.

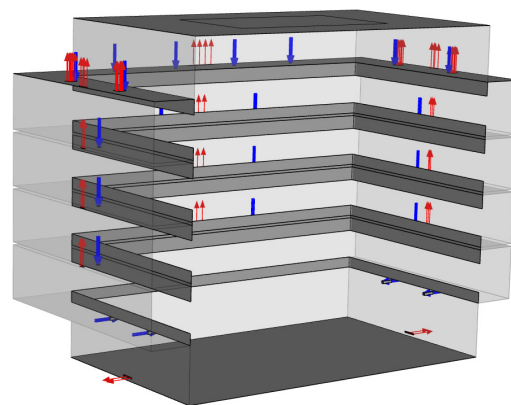


Figure 2. Layout of air inlets and outlets in Case S-N3-V1. (blue arrows represent air supply inlets; red arrows indicate return air outlets.).

2.3. CFD Model and Settings

2.3.1. Grid Generation

The established geometric model was meshed using hexahedral structured grids in ANSYS ICEM CFD 2021 R1. Local refinement was applied at the air supply inlets, return air outlets, and near-wall regions to capture large velocity gradients and temperature gradients accurately. The average skewness of grid was below 0.25, and the minimum orthogonal quality was greater than 0.75, which meets the CFD simulation requirements. Figure 3 shows a medium grid with 2.7 million cells for Case S-N3-V1.

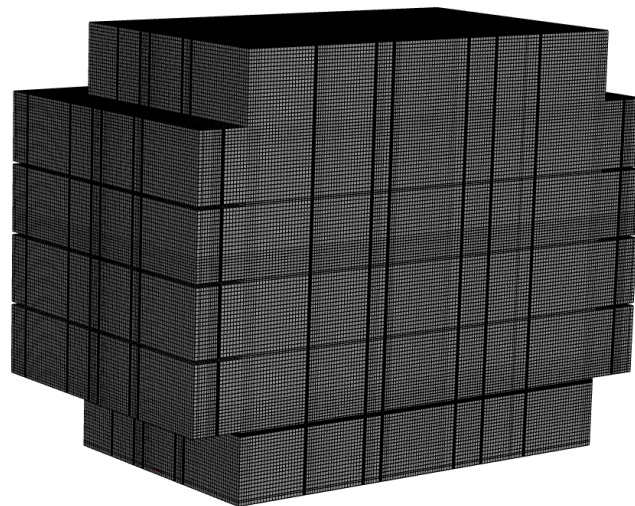


Figure 3. Medium grid for Case S-N3-V1.

2.3.2. Boundary Conditions and Model Setup

The boundary conditions are specified as follows: (a) The ground is set as a wall boundary with a first-type (constant temperature) condition at 20 °C. (b) The south exterior wall is defined as a wall boundary considering combined convective and radiative heat transfer. The south glazing windows are set as transparent wall boundaries that also account for combined convective and radiative heat transfer. (c) The east, west and north corridor walls and floor slabs are assigned as adiabatic wall boundaries. (d) Air supply inlets are defined as velocity-inlet boundaries. (e) Corridor return air outlets are set as mass-flow-outlet boundaries, with mass flow rates identical to those of the corresponding supply inlets. Return air outlets on the first floor are defined as a pressure-outlet boundary.

The numerical simulation is carried out in ANSYS Fluent with the standard $k-\epsilon$ turbulence model. Thermal buoyancy is treated via the Boussinesq approximation. The species transport model is adopted to calculate indoor water vapor concentration, which is further converted into the indoor relative humidity. For the building envelope, external walls are insulated with rock wool sandwich panels. The roof, excluding the skylight area, is thermally insulated with centrifugal glass wool, while insulated glass units are used for south-facing windows. The thermal properties of the building envelope materials adopted in this study are summarized in Table 2.

Table 2. Thermal properties of building envelope materials.

Material	Density (kg/m ³)	Specific Heat Capacity (J/(kg·K))	Thermal Conductivity (W/(m·K))	Thickness (mm)
Rock wool sandwich panel	80	750	0.043	100
Centrifugal glass wool	16	700	0.04	100
Insulating glass unit	2500	790	0.9	21

The Solar Ray Tracing model is adopted to simulate solar radiation, as this approach enables the evaluation of the impact of solar radiation penetrating the building atrium on the indoor thermal environment and human thermal comfort. The specifications of the boundary convective and radiative heat transfer parameters are presented in Table 3.

Table 3. Boundary convective and radiative heat transfer parameters.

Surface		Roof Surface	External Wall	Window and Skylight	Ground
External emissivity		0.9	0.9	0.49	--
Heat transfer coefficient ($W/(m^2 \cdot K)$)		0.5	0.5	2.7	--
Spectral absorption rate	Direct visible	0.25	0.25	0.49	0.91
	Direct IR	0.9	0.9	0.49	0.92
	Diffuse	--	--	0.49	--
	Hemispherical	--	--	0.49	--
Spectral transmittance	Direct visible	--	--	0.3	--
	Direct IR	--	--	0.3	--
	Diffuse hemispherical	--	--	0.32	--

A second-order upwind scheme is adopted for the discretization of pressure, temperature, momentum, turbulent kinetic energy k and dissipation rate ϵ . The SIMPLEC algorithm is employed for pressure–velocity coupling.

2.3.3. Grid Independence Verification

A grid independence verification was performed using three grid schemes (coarse, medium, and refined) to balance computational accuracy and time cost. For Case S-N3-V1, three grid layouts were studied: a coarse grid with 1.55 million cells, a medium grid with 2.7 million cells, and a refined grid with 3.5 million cells. As shown in Figure 4, obvious velocity deviation can be observed between the coarse and medium grids, while the relative difference in velocity between the medium and refined grids is less than 3%. Therefore, the medium grid was finally selected for the subsequent CFD numerical simulations.

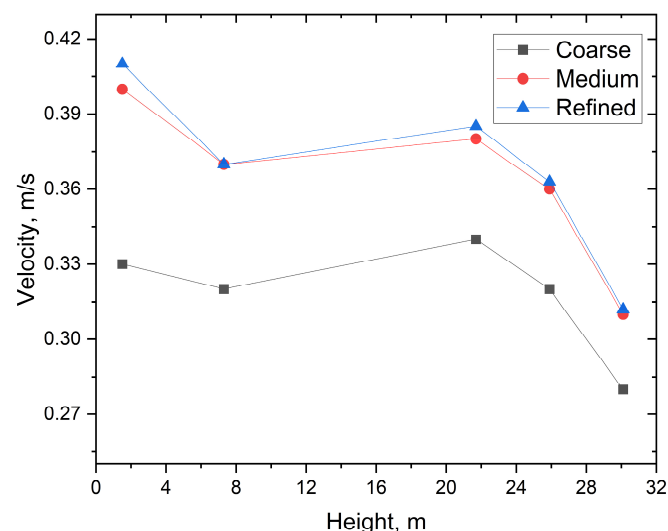


Figure 4. Grid independence verification via velocity profiles.

2.4. Validation of CFD Model

To verify the reliability of the CFD model in simulating the flow field and thermal-humidity environment, the measured data were compared with the CFD simulation results. The field measurements were carried out in the atrium and corridors of an eleven-story

office building at a university in Xi'an. As shown in Figure 5, the 33.7 m high atrium vertically connects floors 1 to 7 of the building. The corridor surrounding the atrium is 52.2 m long (east–west) and 18 m wide (north–south). The first floor has a height of 5.8 m, while the second and third floors are 5.1 m high, and the floor height from the fourth to the seventh floors is 4.2 m. Smoke baffles are installed between the surrounding corridors and the atrium. The top of the atrium is fitted with a tempered laminated glass skylight.

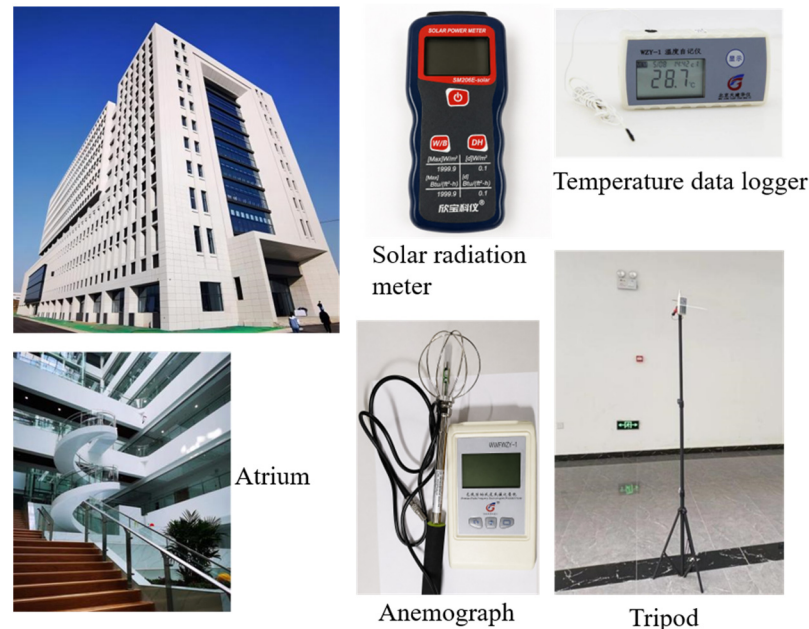


Figure 5. Layout of the test building and photos of loggers. (The non-English terms shown in the figure are original labels and nameplates on the instrument.).

The measurements were performed from 14:00 to 15:00 on 24 September 2024, under non-air-conditioned conditions. To analyze the thermal environment of the surrounding corridors, four temperature measuring points were arranged on the east, west, south, and north sides of the corridors on the first to sixth floors, as shown in Figure 6. The temperature data loggers (Model: WZY-11; Accuracy: ± 0.2 °C; Origin: Beijing, China) were mounted on tripods at a height of 1.5 m above the floor on each level. The corresponding vertical heights of the measuring points were 1.5 m (first floor), 7.3 m (second floor), 12.4 m (third floor), 17.5 m (fourth floor), 21.7 m (fifth floor), and 25.9 m (sixth floor). In addition, indoor air velocity measuring points were placed on the east side of the corridors on each floor at 1.5 m above the floor, using an anemograph (Model: WWFWZY-1; Accuracy: ± 0.05 m/s; Origin: Changchun, China). Weather monitoring points were set outside the building to record ambient dry-bulb temperature, air velocity and solar radiation intensity. Solar radiation was measured using a solar radiation meter (Model: SM206E-solar; accuracy: ± 0.1 W/m²; origin: Xi'an, China).

The CFD simulation was conducted for 14:00 on 24 September 2024, using measured outdoor meteorological parameters: an ambient dry-bulb temperature of 31.8 °C, total solar irradiance of 412.4 W/m², and a air supply velocity at the hall inlet of 3.97 m/s. As shown in Figures 7 and 8, the results indicate that both the simulated and measured temperatures exhibit a positive vertical temperature gradient in the atrium and corridors, and the variation trend of air velocity at each floor is consistent in the vertical direction. Since the air flow and heat transfer between the atrium, corridors and surrounding office areas are neglected in the numerical model, the simulated air velocities in the corridors are lower than the measured values, and heat accumulation within the atrium space is

more pronounced. Consequently, the simulated temperature gradient is greater than the measured one.

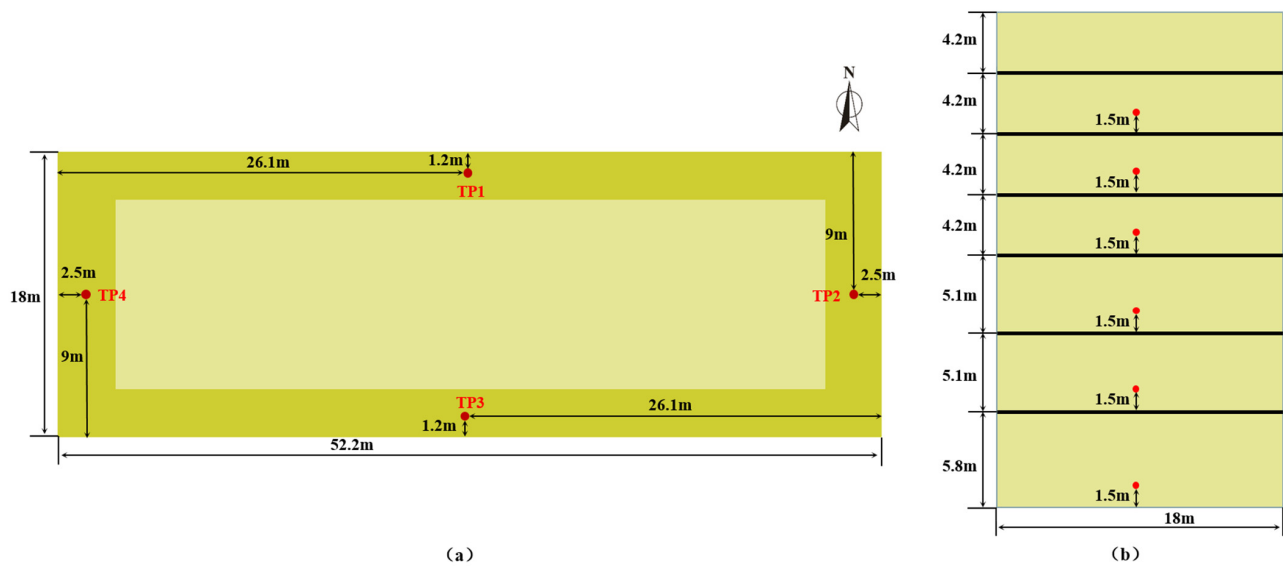


Figure 6. Location of measuring points: (a) floor plan; (b) vertical profile of TP2.

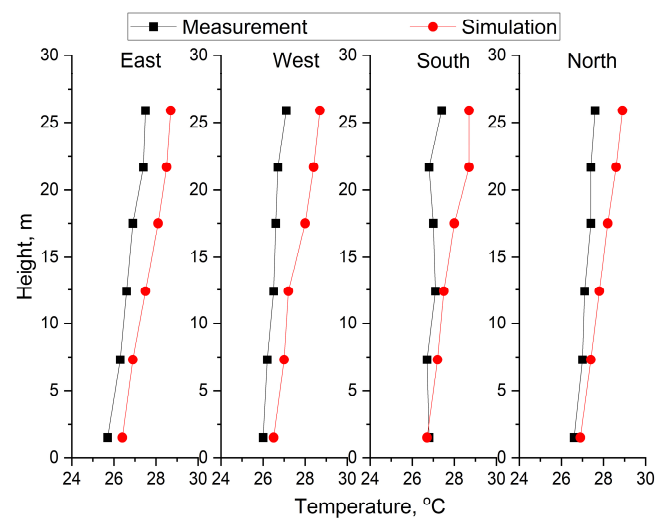


Figure 7. Comparison of measured and simulated temperatures.

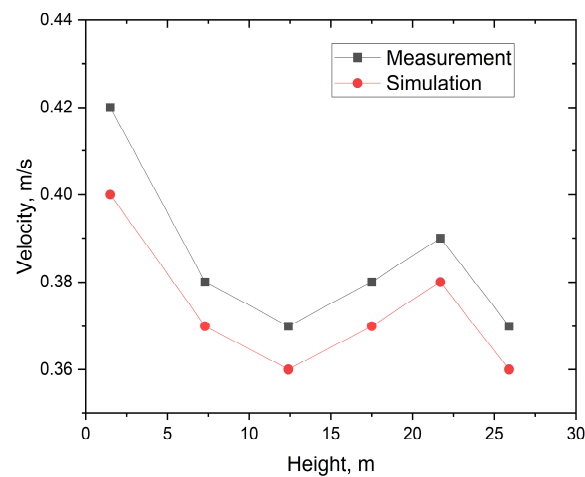


Figure 8. Comparison of measured and simulated velocities.

2.5. Thermal Comfort Evaluation

Predicted Mean Vote (PMV) and Predicted Percentage of Dissatisfied (PPD) indices were adopted to quantify thermal comfort at all monitoring points. Developed by Professor Fanger [35], the PMV criterion is the most prevalent thermal comfort evaluation model based on the human body heat balance mechanism. Referencing the adaptive model from ASHRAE 55-2020 [36], this study assessed the thermal performance of the target cases. The core influencing factors of PMV include metabolic rate, clothing insulation, and mean radiant temperature; the corresponding mathematical formula is presented below:

$$\begin{aligned} \text{PMV} = & [0.303e^{-0.036M} + 0.028] \times \{ (M - W) - 3.05 \times 10^{-3} [5733 - 6.99(M - W) - P_a] \\ & - 0.42(M - W - 58.15) - 1.7 \times 10^{-5} M(5867 - P_a) - 0.0014M(34 - t_a) \\ & - 3.96 \times 10^{-8} f_{cl} [(t_{cl} + 273.15)^4 - (t_r + 273.15)^4] - f_{cl} h_c (t_{cl} - t_a) \} \end{aligned} \quad (2)$$

where M is the metabolic heat production per unit body area, assigned as 60 W/m^2 ; W is the external mechanical work, W/m^2 ; P_a is the ambient water vapor partial pressure, Pa; t_a is the dry-bulb air temperature, $^{\circ}\text{C}$; f_{cl} is the clothing area factor; t_{cl} is the clothing surface temperature, specified as 29°C ; t_r is the mean radiant temperature, $^{\circ}\text{C}$.

Additionally, the calculation formulas for f_{cl} and t_{cl} are as follows:

$$f_{cl} = \begin{cases} 1.05 + 0.645I_{cl} & (I_{cl} \geq 0.078) \\ 1.00 + 1.290I_{cl} & (I_{cl} < 0.078) \end{cases} \quad (3)$$

$$t_{cl} = 35.7 - 0.028(M - W) - I_{cl} \{ 3.96 \times 10^{-8} f_{cl} [(t_{cl} + 273)^4 - (t_r + 273)^4] + f_{cl} h_c (t_{cl} - t) \} \quad (4)$$

where I_{cl} denotes the clothing thermal resistance, specified as $0.08525 \text{ m}^2 \cdot \text{K/W}$.

The convective heat transfer coefficient between clothing and ambient air in Equations (2)–(4) is defined as:

$$h_c = \begin{cases} 2.38(t_{cl} - t)0.25(h_c \geq 12.1\bar{u}^{0.5}) \\ 12.1\bar{u}^{0.5} & (h_c < 12.1\bar{u}^{0.5}) \end{cases} \quad (5)$$

where $\bar{u}$ is the air velocity, m/s .

The PPD index characterizes the proportion of occupants dissatisfied with the indoor thermal environment. It is derived from the PMV according to the correlation presented below:

$$\text{PPD} = 100 - 95e^{(-0.3353\text{PMV}^4 - 0.2179\text{PMV}^2)} \quad (6)$$

3. Results and Discussion

Based on the numerical model and thermal comfort evaluation method established in Section 2, this section presents and discusses the simulation results. The baseline performance of the base case, which has no air supply to the corridors, is first evaluated to quantify the problems of thermal stratification and thermal discomfort in the atrium and corridor spaces. Subsequently, the effects of air supply inlet layout and air volume distribution strategy on the above issues are investigated, with the aim of providing practical design guidance for optimizing airflow organization in atrium–corridor spaces.

3.1. Distribution of Air Parameters in Base Case with No Air Supply in Corridors

For the base case of a conventional stratified air-conditioning system in office atriums, air supply is only provided on the ground-floor atrium, while the corridors above the second floor are non-air-conditioned. A characteristic section at $Y = 26.9 \text{ m}$ is selected for the result analysis. This vertical section passes through the east corridors from the second floor to the fifth floor. Figure 9 presents the contour distributions of air temperature,

relative humidity, air velocity, and thermal comfort indices. The simulated temperature contours (Figure 9a) of the corridors from the second to the fifth floor show good horizontal uniformity and distinct vertical thermal stratification driven by thermal buoyancy, which is consistent with the stack effect in atrium buildings reported in previous studies. On the fifth floor, the corridor area suffers severe overheating, with temperatures ranging from 32 °C to 36 °C, far exceeding the acceptable thermal comfort range. Hot air plume, induced by solar radiation and internal heat gains, accumulates in the upper zone of the atrium. In contrast, the second floor has the lowest temperature distribution, mainly between 24 °C and 26 °C, with a localized cold zone observed on its right side. For the fourth floor, the temperature drops remarkably compared with the fifth floor, ranging from 26 °C to 30 °C, without obvious extreme temperature regions. Relative humidity (Figure 9b) is negatively correlated with temperature, reaching 54–62% on the second floor and only 38–50% on the fifth floor. Air velocity (Figure 9c) remains below 0.4 m/s in most regions, satisfying thermal comfort requirements; only locally high velocity (approximately 1 m/s) appears at the top of the fifth floor due to entrainment of thermal plume. The PMV distribution (Figure 9d) is consistent with the temperature field: the fifth floor yields PMV values of 1.5–2 (uncomfortably hot), the third–fourth floors fall in the slightly warm range (0.5–1), and the second floor maintains an optimal comfort level ($PMV \leq 0.5$). Such PMV levels results in a considerably higher PPD on the fifth floor, failing to meet high-quality indoor environmental requirements.

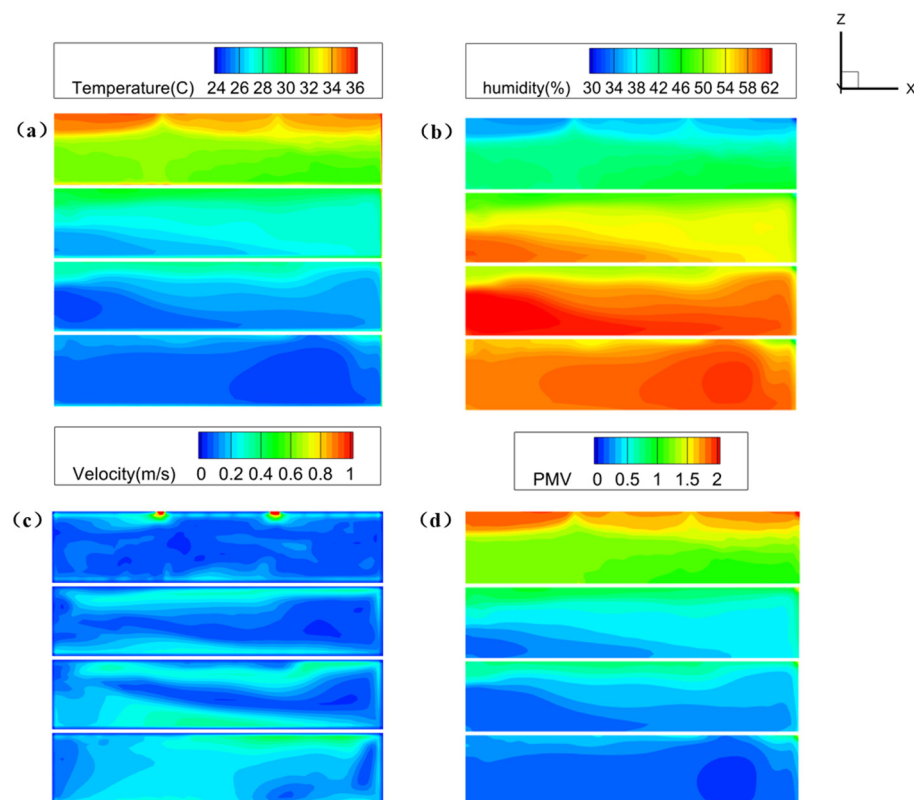


Figure 9. Air parameters of corridors on the second to fifth floors at $Y = 26.9$ m in base case. (a) temperature; (b) relative humidity; (c) velocity; (d) PMV.

According to the simulation results, the average air parameters within the occupied zone at a height of 1.5 m above each floor are statistically analyzed and summarized in Table 4. The CFD results reveal obvious vertical thermal stratification inside the atrium. The average atrium temperature increases moderately from the first floor to the fourth floor but rises sharply on the fifth and sixth floors due to the stack effect. From the second to

the fourth floor, the average temperature of the surrounding corridors is slightly higher than that of the atrium, owing to weak air movement and heat accumulation within the corridor spaces. By contrast, the average corridor temperature is lower than the atrium temperature on the fifth floor, because hot air ascends along the atrium and accumulates at the upper atrium level. The average relative humidity of the surrounding corridors decreases with increasing floor height. Humidity levels on the second to fourth floors comply with the thermal comfort requirements specified in ASHRAE 55, while the fifth floor presents a relatively dry indoor environment. In terms of the average PMV and PPD values of the corridors, thermal comfort deteriorates as the floor height increases. The second and third floors meet the comfort criteria; the fourth floor is close to the acceptable limit; the fifth floor shows excessively high PMV and PPD values, indicating severe thermal discomfort.

Table 4. Air parameters of the base case simulation.

	Average Temperature of Atrium, °C	Corridors			
		Average Temperature, °C	Average Relative Humidity, %	Average PMV	Average PPD
first floor	23.5	--	--	--	--
second floor	24.2	24.9	61.6	0.2	5.1
third floor	24.4	25.3	59.8	0.4	9.3
fourth floor	26.1	26.4	55.3	0.5	15.2
fifth floor	32.1	31.0	43.9	1.2	34.5
sixth floor	46.1	--	--	--	--

The results demonstrate that the stack effect and airflow patterns are the core factors contributing to inter-floor thermal comfort differences and the unsatisfactory thermal environment in the upper corridors. Therefore, an optimized air supply strategy is necessary to improve the overall thermal environment of the atrium and corridors.

3.2. Air Inlet Layouts and Types: Effects on Atrium and Corridor Thermal Comfort

Figure 10 illustrates the air velocity contours at the vertical section $Y = 26.9$ m for four representative cases with different air supply inlet layouts and types. For the vertical downward air supply cases (D-N1, D-N2, D-N3), the airflow distribution is notably influenced by the inlet configuration. Case D-N1, which has no inlets on the second–third floors and 16 inlets on the fifth floor, exhibits strong downward jets from the fifth floor, accompanied by severe airflow stagnation (velocity < 0.5 m/s) on the second–third floors. This leads to local overheating and degraded thermal comfort. Case D-N2, with supplementary inlets on the third–fourth floors, effectively improves air distribution on the intermediate floors. Case D-N3, with uniformly arranged inlets across the second–fifth floors, eliminates airflow stagnation on the second floor and achieves balanced air supply on all floors. However, local high-velocity regions (> 2.5 m/s) persist near the air supply inlets, introducing potential draft risks. Case S-N3-V1, which adopts 45° swirling air supply based on the uniform inlet layout of Case D-N3, achieves a fundamental improvement in indoor air distribution. The vertical downward jets are effectively diffused, with the peak velocity reduced to approximately 2.5 m/s and the jet coverage area expanded. The swirling airflow enhances horizontal air mixing between the corridors and the atrium, thereby mitigating the local high-velocity zones.

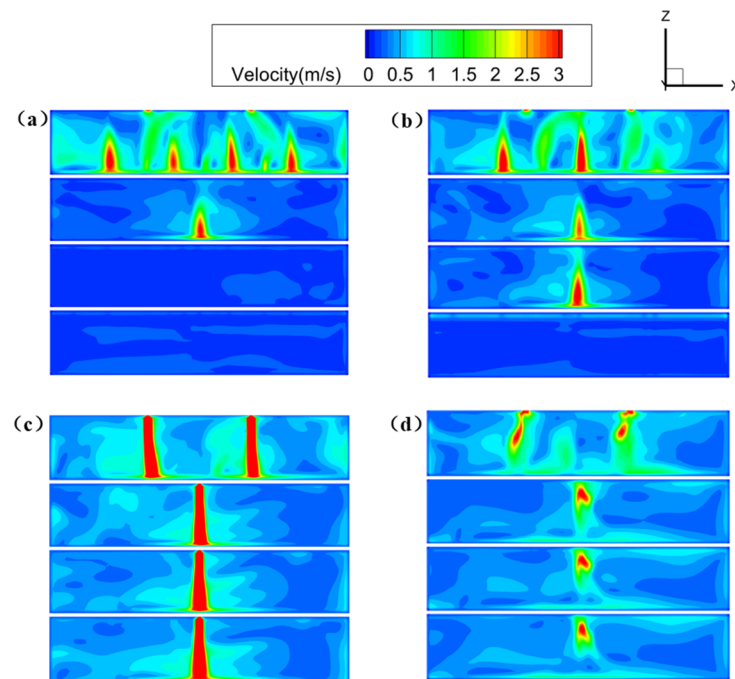


Figure 10. Air velocity distribution in corridors on floor 2–5 at $Y = 26.9$ m for cases with different air inlet layouts and types. (a) D-N1; (b) D-N2; (c) D-N3; (d) S-N3-V1.

Figures 11 and 12 depict the average temperatures in the atrium and corridors at different floors under various cases, respectively. The base case only supplies air to the ground-floor atrium, leading to pronounced vertical thermal stratification in the atrium and severe overheating in the upper corridors, where the average temperature of the fifth-floor corridors reaches 31°C . With the introduction of air supply to the corridors on the second to fifth floors, the vertical thermal stratification in the atrium is gradually mitigated, accompanied by a reduction in corridor temperatures. In Case D-N1, the 16 concentrated air inlets on the fifth floor effectively cool the fifth-floor corridors to 25°C . However, the reduced air supply on the first floor and the absence of air supply on the second–third floors result in elevated temperatures in the intermediate corridors. The strong downward jets induced by the fifth-floor air inlets cause cold air to overflow into the lower atrium, resulting in lower temperatures on the fifth and sixth floors, yet a substantial temperature rise still occurs in the upper atrium. In Case D-N2, supplementary air supply on the third floor lowers the temperature of the intermediate corridors, although the temperature in the second-floor corridor remains relatively high. Case D-N3 adopts uniform air supply across the second–fifth floors, which further weakens the vertical thermal stratification of the atrium and makes corridor temperatures on each floor more uniform, ranging from 25°C to 26°C . For Case S-N3-V1, replacing vertical downward air supply with swirling diffusers in the corridors further reduces the atrium temperature, owing to the enhanced horizontal and vertical airflow mixing promoted by the swirling air distribution.

Figure 13 quantifies the thermal comfort level in the corridor region via PMV, where positive values indicate a warm thermal sensation, and negative values correspond to a cold sensation. The introduction of vertical downward air supply considerably improves the PMV performance of the corridors, while uniform air distribution further refines the thermal comfort conditions. In Case D-N1, the PMV on the fifth floor decreases to 0.3 due to concentrated air supply; however, the PMV on the second floor rises to 1.3, leading to a substantial discrepancy in thermal comfort across intermediate floors. In Case D-N2, supplementary air supply applied on the third and fourth floors effectively lowers the PMV values of the intermediate floors while maintaining a low PMV on the fifth floor, thereby

improving the overall uniformity of thermal comfort. In Case D-N3, uniform air supply across the second to fifth floors restricts the PMV of each floor within the range of -0.1 to 0.3 , close to the neutral thermal comfort zone. Case S-N3-V1 yields a PMV distribution across all floors comparable to that of Case D-N3. Furthermore, the swirling air supply mitigates local high air velocities and the associated cold draft sensation, enabling further optimization of thermal comfort based on the uniformly arranged air inlet layout.

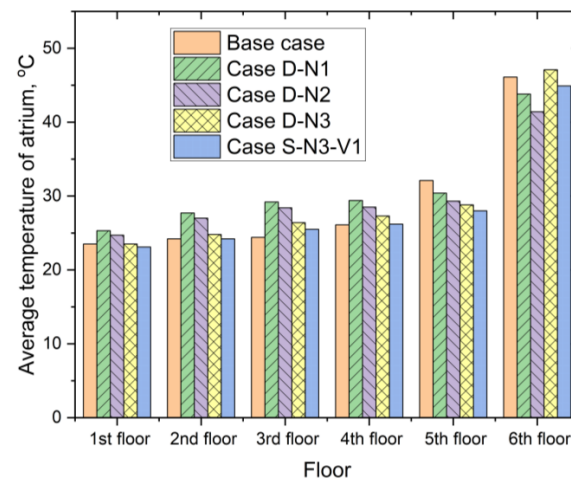


Figure 11. Average temperature of the atrium for cases with different air inlet layouts and types.

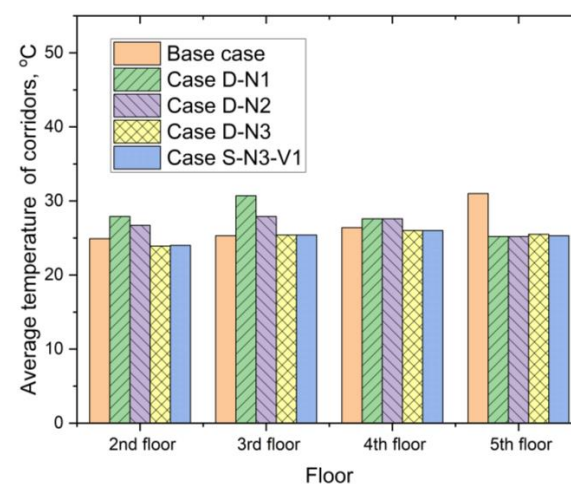


Figure 12. Average temperature of corridors for cases with different air inlet layouts and types.

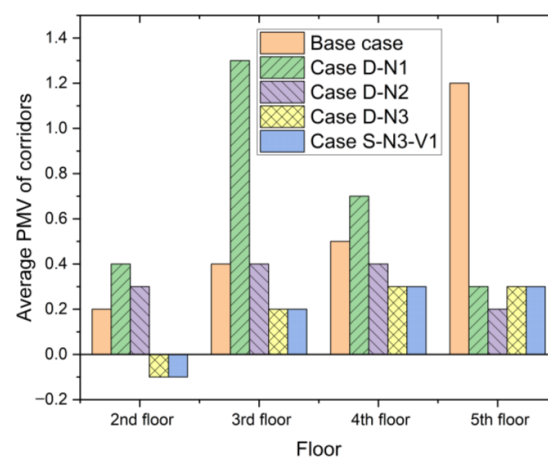


Figure 13. Average PMV of corridors for cases with different air inlet layouts and types.

3.3. Effect of Air Supply Volume Distribution on Thermal Environment of Atrium and Corridors

To further optimize the thermal environment of the atrium–corridor spaces, the influence of air supply volume distribution on the thermal performance of the atrium and corridors was investigated under the uniform inlet layout with swirling air supply (Case S-N3-V1). Figures 14–16 present, respectively, the average atrium temperature, average corridor temperature, and average corridor PMV within the occupied zone at 1.5 m above each floor for five cases with different air volume distributions.

As shown in Figure 14, the vertical temperature stratification of the atrium is significantly affected by the air supply volume distribution. Case S-N3-V1, which adopts a relatively uniform air supply scheme ($1.29 \text{ m}^3/\text{s}$ for the first–fourth floors and $2.58 \text{ m}^3/\text{s}$ for the fifth floor), achieves the lowest atrium temperature on the first floor (approximately 23°C). By contrast, cases with concentrated air supply to the upper floors (e.g., Case S-N3-V4, with $0.48 \text{ m}^3/\text{s}$ and $0.97 \text{ m}^3/\text{s}$ on the first–second floors, $1.45 \text{ m}^3/\text{s}$, $1.94 \text{ m}^3/\text{s}$ and $2.9 \text{ m}^3/\text{s}$ on the third–fifth floors) show a lower sixth-floor temperature (approximately 38°C) but higher temperatures in the lower atrium. Notably, excessive concentration of air supply volume on the upper floors (e.g., Cases S-N3-V3, S-N3-V4, and S-N3-V5) reduces the upper atrium temperature but leads to undesirable temperature rises in the occupied lower atrium.

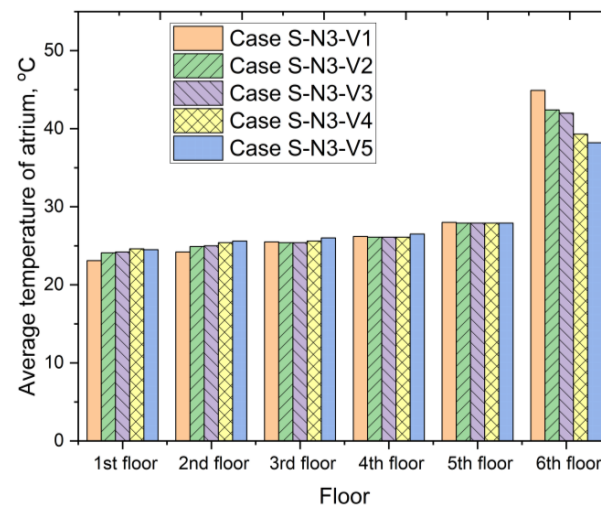


Figure 14. Average temperature of atrium for cases with different air supply volume distributions.

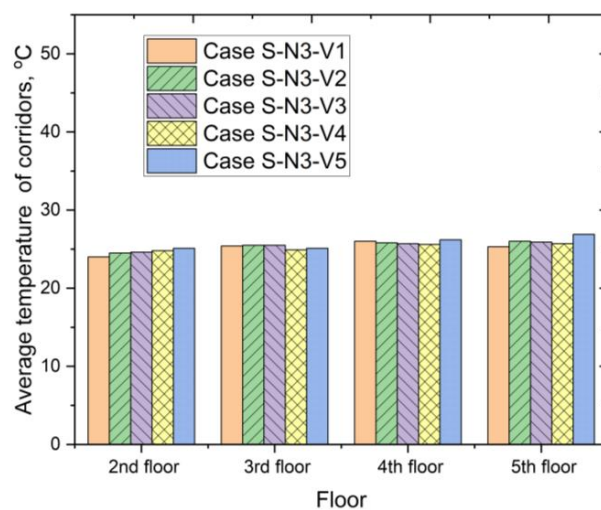


Figure 15. Average temperature of corridors for cases with different air supply volume distributions.

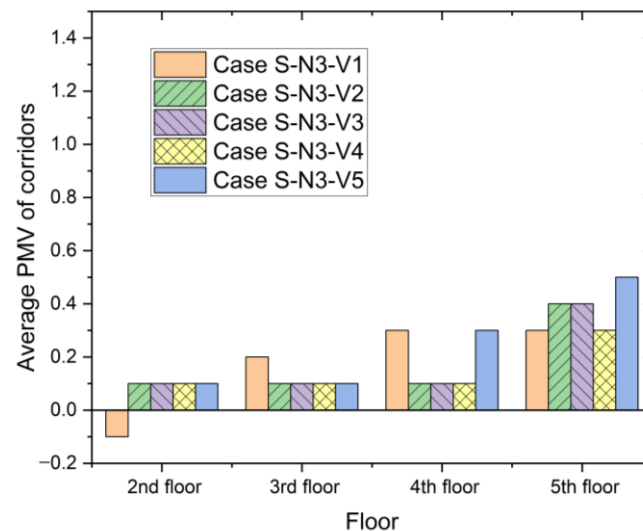


Figure 16. Average PMV of corridors for cases with different air supply volume distributions.

Figure 15 illustrates the average corridor temperatures on the second to fifth floors for five cases with different air supply volume distributions. The corridor temperatures remain within 24–27 °C for all scenarios, indicating that the uniform air supply layout effectively avoids severe overheating. Additionally, the air volume distribution exhibits only a slight effect on the corridor temperatures on the second to fifth floors. Case S-N3-V1, with balanced air supply volume to each floor, achieves the most uniform corridor temperatures, ranging from 24.2 °C to 25.5 °C across the second to fifth floors. Compared with Case S-N3-V1, cases with higher air volume allocated to the third to fifth floors (e.g., Cases S-N3-V2 to S-N3-V5) present slightly lower temperatures in the third and fourth floors corridors but higher temperatures in the second-floor corridor.

The average PMV values in the corridors on the second to fifth floors, as shown in Figure 16, further verify the superior thermal performance of balanced air volume distribution. Case S-N3-V1 demonstrates the optimal PMV distribution, with values ranging from −0.1 to 0.3, all near the neutral thermal comfort zone. Increasing the air supply volume to the third and fourth floors (e.g., Cases S-N3-V2 to S-N3-V4) lowers the PMV of the middle-floor corridors but increases that of the first and second-floor corridors. For Case S-N3-V5, concentrating the air supply volume on the fifth-floor corridor instead leads to a PMV rise to 0.5, indicating a warm sensation. This is because insufficient air supply on the lower floors causes a temperature rise in the first to fourth floor atrium, resulting in degraded thermal comfort on the upper floors.

In summary, a moderately uniform air volume distribution (Case S-N3-V1) not only effectively weakens the vertical thermal stratification of the atrium but also achieves acceptable thermal comfort in the corridors. Excessive concentration of air supply on the upper floors may improve the cooling effect in the upper atrium but compromises the thermal comfort of the corridors.

3.4. Recommendations

To address the thermal discomfort problems of the conventional stratified air-conditioning system, which only supplies air to the ground floor of embedded office atriums, a uniform inlet layout with 45° swirling air supply and balanced air volume distribution for corridors on the second to fifth floors is recommended as the optimal design scheme. Thermal stratification induced by the stack effect, which causes overheating in upper corridors, can be effectively suppressed by adopting air supply in the middle and upper corridors. The 45° swirling air supply can significantly enhance horizontal

air mixing between the corridors and the atrium, while restricting local air velocity near the inlets to avoid draft discomfort. In terms of the air volume distribution, excessive air supply concentration on upper floors should be avoided, and a balanced air volume allocation across lower corridors is suggested. To guarantee satisfactory thermal comfort in the ground-floor atrium and surrounding corridor spaces in practical office buildings, flexible adjustment of the air supply layout and air volume is recommended during design and operation.

4. Conclusions

This study adopts CFD simulation to investigate the thermal environment characteristics of atrium–corridor spaces in an office building and optimize the airflow organization. Taking the conventional stratified air-conditioning system as the base case, this paper explores the effects of different air supply inlet layouts, inlet types and air supply volume distributions. The main findings and conclusions are summarized as follows:

- (1) The CFD simulation results confirm that the base case, which only supplies air to the ground-floor atrium without air conditioning for the corridors on the second–fifth floors, induces significant vertical thermal stratification in the atrium and severe overheating in the upper corridors. The stack effect and unreasonable air-flow patterns are identified as the key factors responsible for the poor indoor thermal environment, with PMV values on the fifth floor reaching 1.5–2.
- (2) Comparative analysis of various air supply layouts shows that a uniform inlet layout arranged for corridors on the second–fifth floors effectively eliminates airflow stagnation on intermediate floors, weakens vertical thermal stratification, and improves the uniformity of corridor temperature distribution. Replacing vertical downward air supply with 45° swirling air supply enhances horizontal air mixing between the corridors and the atrium, and mitigates local high-velocity regions.
- (3) In terms of air supply volume distribution, the uniform air supply strategy (Case S-N3-V1: 1.29 m³/s for the first to fourth floors and 2.58 m³/s for the fifth floor) achieves the optimal overall thermal environment. Under this scheme, the average corridor temperatures are maintained within 24.2–25.5 °C, and PMV values are controlled within the neutral thermal comfort range of −0.1 to 0.3; meanwhile, the stack effect is effectively suppressed. By contrast, excessive air supply concentrated on upper floors and insufficient air supply on lower floors deteriorate the thermal comfort of corridor spaces.

In summary, the conventional stratified air-conditioning system is inappropriate for the atrium–corridor spaces of the studied office building. The comprehensive optimization strategy, integrating a uniform inlet layout, 45° swirling diffusers, and a moderately balanced air supply volume distribution, can effectively alleviate thermal discomfort, weaken vertical thermal stratification, and realize uniform and high-quality indoor thermal comfort. This study provides reliable theoretical reference and practical guidance for the design, optimization, and operation of air-conditioning systems in similar office buildings with atrium–corridor spaces.

5. Limitations and Future Perspectives

One limitation of this study lies in the simplification of the geometric model for the embedded atrium–corridor structure. The numerical model neglects the office rooms adjacent to the corridors and internal floor passageways and simplifies the corridor surrounding walls as adiabatic boundaries. This treatment causes certain discrepancies between the simulated physical field and the actual on-site conditions.

In addition, the steady-state CFD simulation adopted in this work cannot reflect the transient thermal evolution of atrium–corridor spaces under dynamic meteorological conditions. Furthermore, the current optimization only focuses on the thermal environment performance of the air-conditioning system, without evaluating the energy consumption of comprehensive strategies combined with passive design measures.

Future research will refine the embedded atrium–corridor building model by incorporating surrounding offices and internal passageways. Transient CFD simulations considering real-time solar radiation will be conducted, together with multi-objective optimization that integrates multiple regulation strategies, including shading design, ventilation, and air-conditioning operation.

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Nomenclature

f_{cl}	clothing area factor
h_c	convective heat transfer coefficient between clothing and ambient air, $W/(m^2 \cdot K)$
I_{cl}	clothing thermal resistance, $m^2 \cdot K/W$
M	metabolic heat production per unit body area, W/m^2
P_a	ambient water vapor partial pressure, Pa
PMV	Predicted Mean Vote
PPD	Predicted Percentage of Dissatisfied
q_{cl}	stratified cooling load of the air-conditioned zone, W
q_d	cooling load caused by convective heat transfer from non-air-conditioned zones to the air-conditioned zone, W
q_f	cooling load caused by radiative heat transfer from non-air-conditioned zones to the air-conditioned zone, W
q_{1n}	cooling load from internal heat sources inside the air-conditioned zone, W
q_{1w}	cooling load induced by envelope heat gain of the air-conditioned zone, W
q_x	cooling load contributed by outdoor fresh air supply to the air-conditioned zone, W
t_a	dry-bulb air temperature, $^{\circ}C$
t_{cl}	clothing surface temperature, $^{\circ}C$
t_r	mean radiant temperature, $^{\circ}C$
$\bar{u}$	air velocity, m/s
W	external mechanical work, W/m^2

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