Effect on the Performance of Underfloor Air Distribution (UFAD) system under different supply air parameters

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Abstract. Underfloor Air Distribution (UFAD) systems have received increasing attention during the past decades due to thermal stratification and energy saving potential over conventional mixed air conditioning system. Different supply air parameters have significant impacts on the UFAD system performance. In this study, experimental study and energy simulation software EnergyPlus were used to investigate the UFAD performance in a full size experimental room. The distance between diffusers and human, supply air temperature and velocity were discussed. The orthogonal experiments were employed to discuss the effect of supply air parameters on the temperature distribution, thermal comfort and indoor air quality (IAQ). By the analysis, the most important factor is the distance between diffusers and human. Furthermore, the preferable supply air parameters are suggested by means of controlling parameters. The results show that better thermal stratification and thermal comfort can be achieved when the distance between the swirl diffusers and human is 0.7m, the Supply Air Temperature (SAT) is 18-20°C and the Supply Air Velocity (SAV) is 1.2-1.5m/s. A further contribution of this paper is that the energy consumption of UFAD system is calculated by EnergyPlus. The calculation shows that the satisfied thermal stratification and prominent energy saving can be achieved simultaneously by using reasonable supply air parameters, such as the SAT is 18°C and the SAV is 1.2m/s.

1. Introduction
The (Underfloor Air Distribution) UFAD system can be well combined with weak electronic wiring, and the diffuser position can be flexibly arranged according to the function of office building. A well-designed UFAD system can not only improve the ventilation efficiency and IAQ, but also make it easy for person to control the local thermal environment[1]. Therefore, the applications of UFAD systems have become more and more popular in office building both at home and abroad[2]. At present, the UFAD system is applied in Germany, Switzerland, Netherlands, Japan and Canada especially in North America[3-6]. The UFAD system accounts for 40% of office building air conditioning system[7]. In China, more and more building owners, HVAC designers, architects and architectural consultants prefer to consider using UFAD systems in future office buildings.

Many scholars at home and abroad performed in-depth research on the temperature distribution, thermal comfort and the energy consumption of UFAD system. Webster T et al studied the influence...
of supply air volume and diffuser characteristics on indoor vertical temperature distribution. Results showed that when keep the heat source constant, smaller supply air volume can contribute to larger temperature stratification[8]. Haiying Wang et al concluded that the four determinants of air distribution respectively were the diffuser type, the distance between diffusers and human, Supply Air Temperature (SAT) and Supply Air Velocity (SAV) through the human thermal comfort experiment[9]. Gon Kim et al simulated the thermal comfort of large space using UFAD system with constant SAT and different SAV and diffusers position by CFD method, and drew the conclusion that, the UFAD system was able to provide a satisfactory thermal comfort when the indoor heat source was 38.3W/m², SAT was 18°C, SAV was 0.5-0.8 m/s[10]. Ali F. Alajmi et al compared the energy consumption of office building in hot area with UFAD system and Overhead (OH) system. Comparison showed that the UFAD system provided a good indoor thermal stratification when the indoor setting temperature was 26°C and the SAT was 18°C. At this point, compared to OH system, the UFAD system achieved that the indoor air conditioning cooling load reduced by 10%-17%, energy consumption reduced by 37-39%[11]. Tom Webste et al studied the effect of SAT on the UFAD system performance with EnergyPlus. The research showed that with the SAT rising, not only the chiller coefficient of performance (COP) improved, but also the free cooling time extended, as well as the chiller energy consumption reduced. However, delivery system energy consumption increased simultaneously, which lead to the increase of the total energy consumption[12].

Therefore, in this paper, the orthogonal experiments are used to study the influence of multiple factors on temperature distribution and thermal comfort, the optimal distance between diffusers and human of 0.7m is determined. Under this condition, the effect of SAT and SAV on UFAD system performance is studied using the method of control parameters, expecting to get good temperature stratification, better thermal comfort and ideal supply air parameters. Furthermore, the UFAD system model is built with EnergyPlus to calculate the energy consumption with ideal supply air parameters, finally the satisfied thermal stratification, better thermal comfort and prominent energy saving can be achieved by a reasonable supply parameter.

2. UFAD Experimental system

2.1. UFAD Laboratory Bench

This test room is a 4.8m square with an area of 23m² and a height of 3.6m. Underfloor plenum a 0.35m high and the ceiling is 0.45m high. The composite panels are adopted for walls, filled with 0.1-metre-thick polyurethane foam in the middle. The floor is steel cement raised floor; each floor board is 0.6m in length and width. Laying carpet on the floor can reduce air leakage from the gap between two floors and the 0.03-metre-thick extruded polystyrene foam under the plenum also can reduce heat transfer from the floor to the plenum. There are two 0.6m×0.6m return air outlets evenly distributed on the ceiling; the diffusers are set on the raised floor, their quantity, position and type can be changed any time when needed. The configuration of the UFAD experimental system and layout of measuring points are presented in Figure 1.
differential pressure transmitter: AA,BB,CC- electrically adjustable damper; A,B- switch damper

*Figure 1*. UFAD experimental systems and layout of measuring points

Air-cooled heat pump units whose rated refrigerating capacity is 9.9kW were employed to provide cold source. The supply water pipe was equipped with vortex flowmeter to measure the chilled water flowing into the cooling section. An electric control valve was fitted between the supply and return water pipes to regulate the amount of return water. The return water pipe was equipped with an electric heater which was used to heat the return water to ensure that the chiller units can operate in a stable condition. At the end of the chiller units, there was an air-handing unit with a centrifugal fan whose rated supply air volume was 1500m³/h. Adjustable electric heater was used behind the cooling section to achieve the SAT, and an inverter was employed to control the air volume.

2.2. Interior Layout of Room

As shown in Figure 2, this experiment used six circular swirl diffusers with the diameter of 20cm, which were evenly distributed on the raised floor. Two mannequins entwined with heating wires were employed as body heat source, the calorific value was 134W. Similarly, the computer hosted 288W and the monitors hosted 74W. Office equipments were substituted by two cubes covered with heating wires, one of 144W near the table and the other of 288W far away from the table. All above were calibrated by multimeter and powered by regulated power supply. In addition, there were four 36W fluorescent lamps. So the total heating power of the indoor heat source was 1.568kW.

There were 3 mobile stands in the room and 7 measured points on each stand in the vertical direction, their heights were 0.1m, 0.6m, 1.0m, 1.4m, 1.8m, 2.2m and 2.6m, respectively. In addition, there are two temperature sensor points at each diffuser along the direction of supply air that can attain the temperature rise of the underfloor plenum. Also two temperature sensor points were set on the upper and lower underfloor surfaces to measure the heat transfer between the conditioned room and the raised floor.

*Figure 2*. Real view of UFAD system heat source and measured points

3. Experimental study on UFAD system performance

Two main environmental factors affecting thermal comfort are the average temperature and wind speed in the occupied zone[13], which depends on diffuser type and quantity, the distance between diffusers and human, SAV and SAT of UFAD system. There are several influential factors on the stratification and thermal comfort of the UFAD system such as velocity of supply air, temperature of supply air, distance between human and air diffuser etc. The orthogonal experiment can determine the primary and secondary relationship of different factors. Therefore, the orthogonal experiments were used to discuss the effect of SAT and SAV on the room temperature distribution and thermal comfort.

3.1. Orthogonal Experiment Design

In this experiment, three factors were selected to study, the distance between diffusers and human, SAV and SAT. Nine kinds of experiments were conducted according to the L9(3^3) orthogonal array, as shown in Table 1. Three experiments were performed for the three levels of each factor, and the level configuration is uniform and reasonable. Experiments will give the optimal level combination. The code 1, 2, 3 in this column represent the column factors.
### 3.2. Experimental Testing and Methods

In order to gain the ideal supply air parameters, the relative humidity, temperature and air velocity in the room were set to 50%±10%, 25±2℃ and less than 0.25m/s, which were regarded as an indicator of thermal comfort judgment based on the ASHRAE[14]. Any program shown in Table 1 that made the room parameter deviate from the indicator above will be considered unreasonable. Considering the nine kinds of experimental conditions different from each other, an inverter was employed to control the supply air volume, and also an adjustable electric heater was used to achieve the experimental SAT. Throughout the experiment, the supply air volume should be maintained during 800-1200m³/h where fresh air requirement was 120m³/h, diffusers outlet air velocity was 1.2-1.5m/s, and the air changes 10-15/h. Due to the heat stored in the underfloor plenum seriously affects the result, the data collection system should start 6-10h after the operation of the entire system for eliminating the heat. Then record the operating data after the SAT and indoor temperature maintain stable (the temperature fluctuation is at ±0.1℃ within half an hour).

### 4. Experimental results and analysis

#### 4.1. Determine the Primary and Secondary Relationship of Factors Affecting Thermal Comfort

According to the thermal comfort judgment indicator offered by ASHRAE, take the difference of the occupied zone temperature and 25℃ as the analysis index. Five temperature sensors which were installed to measure the room air temperatures along the vertical direction of the occupied zone at the height of 1.8m. The measured data were recorded and averaged, and how the average value was varied over time was observed.

The thermal comfort index in this paper mainly refers to the predicted mean vote (PMV) and predicted percentage of dissatisfaction (PPD) which characterize the human thermal response proposed by P.O.Fanger, PMV and PPD can be calculated as shown in Equation 1 and Equation 2[15], based on the established PMV thermal sensation index which is shown in Table 2.

**Equation 1:**

**PMV proposed by P.O.Fanger.**

\[
PMV = [0.303 \exp(-0.036(M-W))] + 0.028 \{ (M-W) - 3.05 \times 10^{-1}[5377-6.99(M-W)-p_w] \\
-0.42[(M-W)-58.15]-1.7 \times 10^{-5} M(5867-p_w)-0.0014 M(34-T_a) \\
-3.96 \times 10^{-8} f_w[(T_\phi+273)^4-(T_\phi+273)^4]- f_w h_f(T_a-T_\phi) \}
\]

Where:

- \( M = \) human energy metabolism rate (W/m²)
- \( W = \) mechanical work done by human, take 0 when sitting
- \( p_w = \) partial vapor pressure (Pa)
- \( T_a = \) indoor temperature (℃)

#### Table 1. Experimental program

| experiment serial number | A-distance between diffusers and human (m) | B-SAV (m/s) | C-SAT (℃) |
|--------------------------|------------------------------------------|-------------|------------|
| 1                        | 0.4 (1)                                  | 1.2 (1)     | 16 (1)     |
| 2                        | 0.4 (1)                                  | 1.5 (2)     | 18 (2)     |
| 3                        | 0.4 (1)                                  | 1.8 (3)     | 20 (3)     |
| 4                        | 0.7 (2)                                  | 1.5 (2)     | 20 (3)     |
| 5                        | 0.7 (2)                                  | 1.8 (3)     | 16 (1)     |
| 6                        | 0.7 (2)                                  | 1.2 (1)     | 18 (2)     |
| 7                        | 1.0 (3)                                  | 1.8 (3)     | 18 (2)     |
| 8                        | 1.0 (3)                                  | 1.2 (1)     | 20 (3)     |
| 9                        | 1.0 (3)                                  | 1.5 (2)     | 16 (1)     |
— $T_r$= mean radiant temperature (°C)
— $f_d$= dressing area coefficient
— $T_c$= temperature of clothes outside surface
— $h_c$= convective heat transfer coefficient, a function of air velocity W/(m²·K)

**Equation 2:** PMV proposed by P.O. Fanger:  
$$PMV = 100 - 9\exp\left[-\left(0.0335PMV + 0.2179PMV^2\right)\right]$$  

Experimental results are presented in Table 2. The clothes of human were assumed to be short-sleeved and pants, the clothing thermal resistance is 1.1, the metabolic rate is 1.2 and the mechanical work is 0 when calculating the value of PMV, PPD.

**Table 2. Analysis of experimental results**

| experiment serial number | A   | B   | C   | analysis index Y | PMV index | PPD/% |
|--------------------------|-----|-----|-----|------------------|-----------|-------|
| 1                        | 1   | 1   | 1   | 3.92             | -2.59     | 95.1  |
| 2                        | 1   | 2   | 2   | 2.97             | -2.18     | 84.1  |
| 3                        | 1   | 3   | 3   | 2.39             | -1.41     | 46.1  |
| 4                        | 2   | 2   | 3   | 1.17             | -0.91     | 22.5  |
| 5                        | 2   | 3   | 1   | 2.83             | -1.25     | 37.3  |
| 6                        | 2   | 1   | 2   | 1.96             | -0.74     | 16.5  |
| 7                        | 3   | 3   | 2   | 2.5              | -1.45     | 48.2  |
| 8                        | 3   | 1   | 3   | 1.92             | 0.94      | 23.7  |
| 9                        | 3   | 2   | 1   | 1.66             | -1.44     | 47.7  |
| I                        | 9.28| 7.8 | 8.41|                  |           |       |
| II                       | 8.96| 5.8 | 7.43|                  |           |       |
| III                      | 6.08| 8.72| 5.48|                  |           |       |
| P                        | 203.37| 170.52| 155.96|                  |           |       |
| M                        | 67.79| 56.84| 51.99|                  |           |       |
| R                        | 1.08| 0.97| 0.98|                  |           |       |
| S                        | 17.28| 6.34| 1.49|                  |           |       |

Where, I, II, III respectively refer to the sum of analysis index corresponding to the data in column A, B and C. R is the difference of the maximum and minimum average of experimental index. 

$P=I2+I1+I12; M=P/3; G=Y1+Y2+...+Y9; K=G2/9; S=M-K.$

It can be seen from the variance S that the primary and secondary relationship of the three factors that affect the indoor temperature distribution and thermal comfort are the distance between diffusers and human body, SAV and SAT. Also we can see from the range R that the influence of the SAV and SAT is small, 0.97°C and 0.98°C respectively, while that of the distance is 1.07°C.

In the 9 experiments, the 6th experimental result represents the best thermal comfort, whose PMV and PPD are both within the ideal range, the experimental condition is swirl diffuser, the distance between the diffusers and human is 0.7m, the SAT is 18 °C and the SAV is 1.2m/s. The 4th experimental result seems better, the experimental condition is swirl diffuser, the distance between the diffusers and human is 0.7m, the SAT is 20 °C and the SAV is 1.5m/s. Furthermore, when the distance reaches 0.4m, human will feel uncomfortable, the air-conditioning demand cannot be satisfied; when the distance between diffusers and human is 1.0m, human will feel too cold or too hot, it is difficult to obtain suitable supply air parameters. In addition, human feel poorer when SAT is 16 °C. So the SAT set at 16 °C is not recommended. Consequently, if the distance between the diffusers and human reaches 0.7m, the distance will be ideal to achieve a better thermal comfort.

### 4.2. Effect of SAT and SAV on Temperature Distribution, Thermal Comfort and IAQ

To exclude the effect of other factors, when the distance between swirl diffusers and human is 0.7m, we kept the heat source and diffuser type, position and quantity unchanged, only adjusted the fan frequency and SAT by the control system. By doing this, the unrelated variables can be effectively controlled, which make it possible to obtain the effect of SAT and SAV on temperature distribution, IAQ and thermal comfort, and ultimately get the desired UFAD system supply air parameters.

Ventilation efficiency is the main metric of air quality, which consists of ventilation thermal
efficiency and ventilation decontaminated efficiency, reflecting the airflow energy efficiency and pollutant exclusion rate. Since the heat source is also the pollutant source in the experimental room, ventilation efficiency can be reflected by the dimensionless temperature coefficient $\eta$. It shows the ability for the system to take full advantage of the supply air. Large $\eta$ indicates small load of occupied zone and stronger heat remove ability of supply air[16].

Equation 3: The dimensionless temperature coefficient.

$$\eta = \frac{T_{at} - T_{st}}{T_{de} - T_{st}} = \frac{C_{rai} - C_{st}}{C_{de} - C_{st}} \quad (3)$$

Where:
- $C_{rai}$ - return air concentration
- $C_{oz}$ - occupied zone concentration
- $C_{st}$ - supply air concentration

Experimental data and analytical results are shown in Table 3. Indoor vertical temperature profiles under different operating conditions are shown in Figure 3.

**Table 3.** Different conditions of supply and return air parameters, thermal comfort index and ventilation efficiency ($\eta$)

| operating conditions | SAT / °C | RH /% | TOZ / °C | RRAH / °C | RH /% | SAV (m/s) | PMV | PPD /% | $\eta$ |
|----------------------|----------|-------|----------|-----------|-------|-----------|-----|--------|-------|
| a                    | 16       | 61.1  | 21.62    | 23.4      | 38.2  | 1.2       | -1.4| 45.6   | 1.32  |
| b                    | 16       | 62.5  | 20.89    | 22.2      | 42.6  | 1.5       | -1.95| 74.5   | 1.27  |
| c                    | 16       | 64.5  | 20.65    | 21.6      | 40.7  | 1.8       | -1.25| 37.3   | 1.18  |
| d                    | 18       | 62.9  | 23.05    | 25.1      | 40.1  | 1.2       | -0.74| 16.5   | 1.38  |
| e                    | 18       | 65.0  | 22.31    | 23.7      | 44.8  | 1.5       | -1.32| 41.3   | 1.32  |
| f                    | 18       | 66.8  | 22.55    | 23.1      | 48.8  | 1.8       | -1.93| 73.5   | 1.12  |
| g                    | 20       | 61.2  | 24.47    | 26.2      | 41.4  | 1.2       | 0.85 | 20.3   | 1.37  |
| h                    | 20       | 62.5  | 24.06    | 25.2      | 44.9  | 1.5       | -0.91| 22.5   | 1.28  |
| i                    | 20       | 63.9  | 24.23    | 24.6      | 47.9  | 1.8       | -1.19| 34.7   | 1.08  |

It is evident from the Table 3 that the thermal comfort index PMV and PPD are able to meet the air conditioning requirements and provide higher ventilation efficiency when the SAT is 18-20°C and the SAV is 1.2-1.5m/s. When the SAV maintains constant and lower the SAT, the large difference of supply air and heat source promotes the heat exchange effect between them which enhances the thermal plume entrainment, therefore, the waste heat rises to the upper layer of the room with the thermal plume and finally is exhausted to outdoor, along with ventilation efficiency increasing. When the SAT maintains constant, with the growing SAV, thermal plume rises to the higher layer of room, indoor thermal stratification is destroyed or even disappear, the air is more evenly mixed, the temperature of the return air is closer to the occupied zone temperature, with the ventilation efficiency lowering.

Consequently, in order to take full advantages of the UFAD system, and establish good temperature stratification and IAQ, the SAT should be maintained lower. In addition, lower SAV contributes to lower fan pressure that reduces fan energy consumption.
In Figure 3, when the SAT maintains constant, if the SAV increases, the thermal stratification is not obvious; if the SAV reduces, the indoor vertical temperature stratification is obvious, such as condition a, b and c; when the SAV maintains constant, the indoor temperature distribution curve will change with SAT (condition a, d, and g). When the air temperature is 16°C, the SAV is 1.2-1.5m/s, it demonstrates that lower indoor temperature results in worse thermal comfort.

Keeping the SAT at 18°C, when the SAV is 1.2m/s, the temperature difference between head and feet is 1.9°C, the indoor temperature stratification meets the thermal comfort. When the SAV is higher than 1.2m/s, the indoor thermal stratification is destroyed. Keeping the SAT at 20°C, when the SAV is 1.2-1.5m/s, the indoor temperature stratification is better, and the temperature difference is lower than 3°C[14], which satisfies the demand of human thermal comfort. Therefore, this condition can be regarded as an ideal operating condition.

5. Comparison of energy consumption and ideal air supply parameters

Optimal thermal stratification and thermal comfort are achieved through the above experimental study. Furthermore, good artificial environment can not be established with huge energy consumption, therefore, it is necessary to compare the energy consumption of different operating conditions, and then find one that attributes to good temperature distribution, better thermal comfort and lower energy consumption.

5.1. UFAD System Building Model Characteristics

Yifu office building is located in the campus of Southeast University in Nanjing. Architectural models have been built in DesignBuilder[17] in accordance with the actual building structure. The height of the office building room is 3.6m with the air-conditioned area of 4670m². Adjacent to the test room are two same rooms. One side of the room is interior walls which connects to the inner aisles, the other side is exterior wall on which several single glass windows are fitted. The internal layout of the room is: the height of the underfloor plenum is 0.35m and the height of the return air box at the ceiling is 0.45m. Internal and external partitions in the room are not taken into account. Heat transfer coefficient of the building envelope and the ratio of window to wall are determined according to the literature[18]. Exterior walls are made of 24 brick wall, and the ratio of window to wall is 35%.

It can be known that, the indoor personnel density is 0.1persons/m², the heat emission intensity of human activity is 134W/person, the indoor lighting power density is 11W/m², the internal office equipment heat is 18W/m², and each personal computer heat is 135W. Office staff time is from 8:00 to 18:00 in working days with the equipment occupancy rate of 100% and the lighting utilization rate of 90%, and from 9:00 to 18:00 in non-working days with the equipment occupancy rate of 18% and the lighting utilization rate of 30%. The ideal air-conditioning system supply parameters are used, in order to obtain the influence law of SAT and SAV on energy consumption, two operating conditions of SAT 17°C and 19°C are added, as shown in Table 4.

Table 4. Air-conditioning system operating parameters
Nanjing city belongs to the hot summer and cold winter area, locating at 32 degrees north latitude and 118.80 degrees east longitude. Meteorological parameters of the typical year of Nanjing are selected to calculate the annual energy consumption of air conditioning systems. The summer atmospheric pressure is 101180Pa, the air-conditioning design day is 15th, July, the outdoor dry bulb temperature is 34.8°C and the wet bulb temperature is 27.9°C, the average wind speed is 2.4m/s.

5.2. Description of EnergyPlus UCSD-UFAD Model

In the Room Air module, indoor temperature is replaced by dimensionless temperature, namely, the module divides the vertical height and transfers into a dimensionless value. Then the dimensionless temperature difference is input according to the dimensionless height. Interpolation method is used to calculate the temperature between two adjacent heights, by which the temperature of the entire space is defined. In this way, temperature can be flexibly defined and divided in detail, which is especially suitable for the condition of temperature distribution already known. Dimensionless height is calculated as follows:

\[
\lambda_i = \left( H_i - H_f \right) / H
\]

Where:
- \( \lambda_i \) = dimensionless height at any position
- \( H_i \) = the vertical height of the position(m)
- \( H_f \) = the average height of the indoor floor(m)
- \( H_c \) = ceiling height(m)

The temperature distribution in the vertical direction under different conditions can be defined in Room Air module. As is shown in Figure 3, the indoor temperature distribution can be transferred into Table 5 which gives the UFAD system indoor temperature when the SAT is 20 °C and the SAV is 1.2m/s.

| Table 5. UFAD system indoor temperature setting |
|-----------------------------------------------|
| H/m  | \( \lambda_i \) | conditions g T\(_{oz} \)/ °C | \( \Delta T_{ai} \)/ °C | conditions h T\(_{oz} \)/ °C | \( \Delta T_{ai} \)/ °C | conditions i T\(_{oz} \)/ °C | \( \Delta T_{ai} \)/ °C | H/m |
|------|----------------|-------------------------------|----------------|-------------------------------|----------------|-------------------------------|----------------|------|
| 0.1  | 0.04           | -0.29                         | -0.3           | -0.51                         | 0              |                               |                |      |
| 1.5  | 0.54           | 24.47                         | 0.57           | 0.48                          | 24.23          | 0.16                          | 0              |      |
| 1.8  | 0.64           | 1.3                           | 24.06          | 1.03                          | -0.1           |                               |                |      |
| 2.4  | 0.86           | 1.59                          | 1.16           | 0.17                          |                |                               |                |      |

The UCSD-UFAD interior model is able to predict the temperature of the three representative points: (1) The temperature on the surface of the floor: the point of 0.1m in height that represents the area is 0.2m higher than that of the floor; (2) The temperature of the occupied zone: the point of 1.5m in height that represents the area between the underfloor and stratification height; (3) The temperature of the upper mixed zone: the area between the upper stratification height and the ceiling height.
The model simplifies the indoor temperature distribution, defines temperature of the occupied zone $T_{OZ}$, temperature of the mixed zone $T_{RA}$ and the stratification height $Z$. In order to work out the relationship of temperature distribution and external parameters, two dimensionless parameters are defined:

**Equation 5:** The dimensionless parameters. \[ \Phi = \frac{(T_{RA} - T_{OZ})}{(T_{RA} - T_{sa})} \] (5)

**Equation 6:** The dimensionless parameters. \[ \Gamma = Q^{3/2} \left[ \frac{m(n / mA_d)^{1/4}}{B^{3/2}} \right] \] (6)

Where:
- $\Gamma$=dimensionless temperature
- $B$=indoor heat buoyancy flow, $B=gQ/(\rho c_p T)$ (m$^4$/s$^3$)
- $T$= air temperature(K)
- $Q$=the heat source heating power(W)
- $A_d$=diffuser effective area(m$^2$)
- $n$=number of diffusers
- $m$= number of plumes.

Three equations can be derived to predict the relationship of $T_{RA}$, $T_{OZ}$, $Z$ and cooling load $W$, $A_d$ and $Q$.

The Equations are[19]:

**Equation 7:** The return air temperature. \[ T_{RA} = \frac{0.128W}{Qg} T_{sa} + T_{sa} \] (7)

**Equation 8:** The occupied zone temperature. \[ T_{OZ} = T_{RA} - 1.61^{-0.76} (T_{RA} - T_{sa}) \] (8)

**Equation 9:** The thermal stratification height. \[ Z = \sqrt{A_d n/m} (7.43 \ln(\Gamma) - 1.35) + \frac{Z_s}{2} \] (9)

Where:
- $Z_s$=vertical height of the heat source(m).
- $W$=cooling load (W).

### 5.3. Different Conditions Analysis of Cooling Energy Consumption

Figure 5 gives the variation of load hourly in the design day (15th, July). It can be seen from the Figure 6 that, the outdoor environment influences the air-conditioning load greatly, from 10:00 am to 17:00 pm; the building cooling load is maintained at a high level with the peak value appearing at 15:00. The total load is 818.65kW and the cooling load per unit area is 175.3W/m$^2$.
In order to compare the energy consumption under different operating conditions, the energy utilization index kWh/(m².a) is introduced to represent the annual energy consumption. Figure 6 shows the electric breakdown for variation under different SAT and SAV. It can be seen that the electric energy consumption increases with the rise of SAV when the SAT maintains constant, decreases with the fall of SAT when the SAV maintains constant.
SAT and SAV

Figure 7 shows the variation of the total energy consumption under different SAT and SAV. As shown in Figure 9, total energy consumption increases with the SAT grow when the SAV maintains constant, which varies linearly. Within the range of the ideal supply air parameters, when the SAT is 18°C, and the SAV is 1.2 m/s, 1.5 m/s and 1.8 m/s, the corresponding average temperatures of the occupied zone are 23.05°C, 22.31°C and 22.55°C, and the corresponding returning air temperatures are 25.1°C, 23.7°C and 23.1°C (See Figure 3) respectively. The total energy consumption appears minimal when the SAT is 1.2 m/s. When the SAT is 20°C, SAV is 1.2 m/s, 1.5 m/s and 1.8 m/s, the corresponding average temperature of occupied zone is 24.47°C, 24.06°C and 24.23°C, and the corresponding returning air temperatures are 26.2°C, 25.2°C and 24.6°C (See Figure 3), respectively. When the SAT is 18°C and SAV is 1.2 m/s, the average temperature of the whole room increases for 2°C, the room cooling load reduces, the refrigeration unit energy consumption reduces by 2.76%, the fan and water pump energy consumption increases, all of which make the total energy consumption increase by 14.8%.

As shown in Figure 8, the COP of the chiller improves and the cooling capacity increases as the SAT. When the SAT is 18°C, the operating COP of chiller is 4.86, with the SAT increasing to 20°C, the operating COP of the chiller improves by 5.8%, the energy consumption reduces by 9.23%, but the energy consumption of fans and pumps increase, and the amount of increase is greater than that of the reduced chiller energy consumption which increases the total energy consumption.

Therefore, it should compare the pros and cons for the SAT of UFAD system. Low SAT will reduce the air volume which affects the IAQ and thermal comfort, and the fan energy consumption can be decreased, but the chiller operating efficiency decreases simultaneously. The energy consumption analysis shows that, when the SAT is 18°C, the SAV is 1.2 m/s, the lowest energy consumption of UFAD system can be achieved, and the supply parameters are more conducive to building energy saving, which can be regarded as the optimum supply parameters of UFAD system.

6. Conclusion

Based on the experimental results of the UFAD system and the building energy simulation results of the properly operated UFAD systems, the following conclusions can be drawn:

—The orthogonal experiment determined the primary and secondary relationship of factors affecting thermal comfort; they are the distance between diffusers and human, SAV and SAT in turn. The most important factor is the distance between diffusers and the human. When the distance between diffusers and human is 0.7 m, better thermal comfort can be achieved.

—The distance between diffusers and human is 0.7 m, when the SAT is 18-20°C, and the SAV is 1.2-1.5 m/s, the indoor temperature stratification will be better, and the ASHRAE thermal comfort indices PMV and PPD are both in ideal range simultaneously.

—The energy consumption of the UFAD system has been compared with EnergyPlus, the result shows that the satisfied thermal stratification and prominent energy saving can be achieved simultaneously by reasonable supply air parameters that the SAT is 18°C and the velocity is 1.2 m/s, which can be regarded as the best cooling operation condition of the UFAD system.

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