

Design Method and Experiment Research for UFAD with Swirl Diffusers in Office Building

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Abstract: According to the principle similarity between underfloor air distribution (UFAD) and displacement ventilation (DV) systems, a design method was presented for UFAD with swirl diffusers in office building, which was on the basis of design method for displacement ventilation and characteristics of swirl diffusers. Depending on the design method, a standard office configuration was set up for measurement. Based on the measurement results of indoor air, the design method was verified and the performance of the system was evaluated.

Keywords: UFAD, design method, dimensionless temperature, vertical temperature gradient, thermal comfort

1. INTRODUCTION

Conventional centralized HVAC systems are to use the standardized approach of providing a single uniform thermal and ventilation environment. Recent trends in today's office environment make it increasingly difficult for them to satisfy the environment preferences of individual office workers. An increasing amount of attention has been paid to air distribution systems that individually condition the immediate environments of office workers within their workstations. Underfloor air distribution (UFAD) systems are innovative methods are also important in the cooling performance of displacement ventilation systems. UFAD and DV systems deliver cool air into the conditioned space at or near floor level and return it at or near ceiling level. Thermal plumes that develop over heat sources in the room play a major role in driving the overall floor-to-ceiling air motion by entraining air from the surrounding space and drawing it upward. UFAD systems differ from the DV systems primarily in the way that air is delivered to the space: (1) air is supplied at higher velocity with correspondingly greater mixing, through smaller-sized supply outlets. By introducing supply air with greater momentum, UFAD systems with swirl diffusers alter the behaviour in the lower region of the space compared with displacement ventilation systems, by increasing the temperature near the floor, by increasing the temperature gradient, all other conditions being equal. (2) local air supply conditions are generally under the control of the occupant, allowing comfort conditions to be optimized. Based on the analysis presented above, the purpose of this paper is to present a design method for UFAD with swirl diffusers in office building, which is on the basis of design method for displacement ventilation and characteristics of swirl diffusers. Depending on the design method, a standard office configuration was set up for measurement. Measurement results were used to verify the method and evaluate the performance of the system.

for delivering conditioning in offices and other commercial building, which meet the demand presented above. An underfloor air distribution (UFAD) uses the open space (underfloor plenum) between a structural slab and the underside of a raised floor system to deliver conditioned air to supply outlets located at or near floor level within the occupied zone (up to 1.8 m height) of the space. Underfloor air distribution systems with swirl diffusers are based on many of the same principles that

2. DESIGN METHODOLOGY FOR UFAD WITH SWIRL DIFFUSERS

2.1 Application condition of the design method

Based on the temperature and flow rate of supply air, the conditioning room should abide by the following requirements: 1) the height of the room should be more than 2.5m, 2) cooling load distribution should

have such relations, $\frac{Q_{OC}}{Q_T} \leq 0.68$, $\frac{Q_L}{Q_T} \leq 0.43$, $\frac{Q_{EX}}{Q_T} \leq$

0.92, Where Q_{OC} stands for the cooling load of the occupant, desk lamp and equipment, Q_L , Q_{EX} and Q_T respectively stand for the cooling load of overhead lighting, heat conduction (through the room envelope and transmitted solar radiation) and total cooling load.

2.2 Design procedure

Step (1): Determine the room air t_h in summer.

There is vertical temperature gradient in the conditioning room with UFAD systems. Since most of the occupants in the office are sedentary, the air

temperature at the head level of sedentary occupant (1.1 m) is used as room air temperature.

Step (2): Calculate the cooling load in summer. Similar to conventional overhead systems, the total cooling load Q_T is itemized into: 1) the occupant, desk lamp and equipment, Q_{OC} (W); overhead lighting, Q_L (W); and heat conduction through the room envelope and transmitted solar radiation, Q_{EX} (W).

Step (3): Determine the required flow rate for summer cooling, V_C . UFAD creates a thermal stratification. To maintain a comfort level, the design air temperature difference between the head and foot level of a sedentary occupant Δt_{hf} should be less than 2°C. The required flow rate, V_C , can be determined according to equation (1):

$$V_C = \frac{\alpha_{OC} Q_{OC} + \alpha_L Q_L + \alpha_{EX} Q_{EX}}{\rho C_p \Delta t_{hf}} \quad (1)$$

Where

C_p = specific heat of air (J/kg·K)

ρ = air density (kg/m³)

α_{OC} , α_L , and α_{EX} = 0.295, 0.132, and 0.185, respectively; the coefficients stand for the fractions of the cooling loads entering the space between the head and feet of a sedentary occupant (0.1m-1.1m).

Step (4): Find the required flow rate of fresh air for acceptable indoor air quality, V_f . Use ANSI/ASHRAE Standard 62 (ASHRAE 2001) to find the required flow rate for acceptable indoor air quality, V_r . This flow rate is based on mixing ventilation with ventilation effectiveness of one.

$$V_r = R_p \cdot P_D + R_B \cdot A_B \quad (2)$$

Where

R_p — the minimum ventilation rate per person, L/(person·S).

P_D — the number of occupants, (person).

R_B — the minimum ventilation rate per area, L/(S·m²)

A_B — the building area, (m²)

The required flow rate of fresh air for UFAD, V_f , can be calculated by

$$V_f = V_r / \eta \quad (3)$$

Where

η — the ventilation effectiveness at the breathing level.

The η can be determined by

$$\eta = 3.4 (1 - e^{-0.28n}) (Q_{OC} + 0.4 Q_L + 0.5 Q_{EX}) / Q_T \quad (4)$$

Where n — ventilation rate.

Step (5): Determine the supply air temperature. Choose the greater of the required flow rate for summer cooling, V_C , and that for indoor air quality, V_f , as the design flow rate of the supply air, V .

$$V = \max\{V_f, V_C\} \quad (5)$$

Step (6): Calculate the supply air temperature, t_s .

$$t_s = t_f - \theta_f \Delta t_{ex} \quad (6)$$

Where

Δt_{ex} — temperature difference between supply air and exhaust air,

$$\Delta t_{ex} = \frac{Q_T}{\rho C_p V} \quad (7)$$

t_f — temperature at the height of 0.1m,

determined by, $t_f = t_h - \Delta t_{hf}$.

θ_f — the dimensionless temperature. The

result for UFAD show that the θ_f remain close to a constant level of 0.7 over a fairly wide range of air flow rate, which is typically different from DV system.

Based on the design procedure, the design parameters of the conditioning room can be determined.

3. EXPERIMENTAL METHODS

Designing parameters were calculated with the design method in three typical cases of cooling load environments. Cooling load of overhead lighting, Q_L , were changed by altering the number of daylight lamps. Cooling load of the occupant, desk lamp and equipment, Q_{OC} were changed by altering the number of thermal manikins and radiant heat panels. Cooling load of heat conduction through the room envelope and transmitted solar radiation, Q_{EX} , were changed by controlling the temperature of the wall. The room air t_h was 26.0 °C. The designing results are shown in Table 1 and Table 2.

Table 1: Space cooling load calculation

case	Q_L (W)	Q_{OC} (W)	Q_{EX} (W)	Q_T (W)	$\sum a_i x_i$ Q_i (W)
1	320	300	897	1517	297
2	640	300	895	2015	392
3	772	520	1098	2390	458

Table 2: Designing parameters calculation

case	$\Delta t_{hf} = t_h - t_f = 2^\circ\text{C}$				
	V (m^3/h)	Δt_{ex} ($^\circ\text{C}$)	θ_f	t_s ($^\circ\text{C}$)	t_e ($^\circ\text{C}$)
1	466	9.7	0.7	19.2	28.9
2	583	10.3	0.7	18.8	29.3
3	681	10.4	0.7	18.7	29.3

3.1 Controlled experiment chamber

The floor-based system was tested in a controlled environment chamber measuring $6.7\text{m} \times 4\text{m} \times 3\text{m}$. The arrangement of the furniture, including one desk, two thermal manikins (60W per manikin), two chairs, eighteen daylight lamps (40W per lamp), four radiant heat panels (120W per panel) and one controlled wall, is shown in Figure 1. The arrangement of air inlet and outlet is shown in Figure 2. In Figure 3 a photograph is shown of the model office room in the climate chamber configuration.

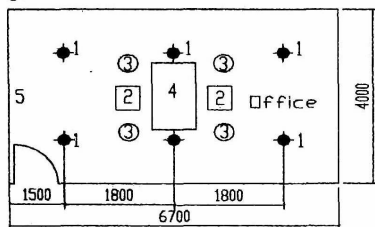


Figure1: Chamber plan: 1=swirl diffuser, 2=thermal manikin, 3=radiant heat panel, 4= desk, 5= controlled wall.

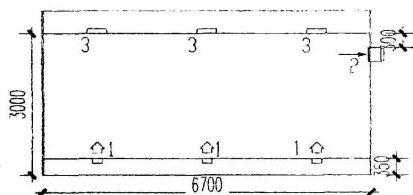


Figure 2: Arrangement of air inlet and outlet, 1=swirl diffuser, 2=air outlet, 3= daylight lamp.

3.2 Instrumentation

Detailed air temperature measurements within the test room were accomplished by using a lightweight sensor rig fabricated of aluminum tubing that allowed a vertical array of sensors to be positioned at desired measurement heights and move around the room to map out a grid of selected measurement locations. At each position air temperature were measured at five height: 0.1m; 0.5m; 1.1m; 1.7m; 2.5m. All sensors

were calibrated prior testing. The measurement error of the temperature sensors was $\pm 0.1^\circ\text{C}$. Temperature sensors were sampled 50 times over a 90 s measurement period.

Supply and return air volumes were monitored with a high-precision flow measurement set up consisting of a series of small-diameter straight-length pipes with pitot tubes mounted to measure the fully developed flow. Additional temperature measurements in the ducts and room were monitored and recorded through the personal-computer-based direct digital control system. Some instruments were shown in Figure 4 and Figure 5.



Figure3: Overview of the climate chamber Configuration.



Figure4: instrument for temperature measurement system.

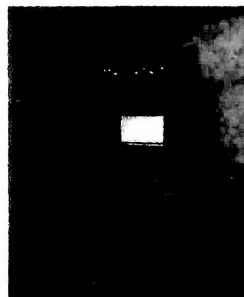


Figure5: computer with data acquisition system.

4. RESULTS AND DISCUSSION

4.1 Air parameters measurement

Table 3 shows the testing results of air parameters in the cooling room. Temperature of room air is between 25 °C and 26 °C. Temperature difference between the head and foot level in the experiment is smaller than that calculated in design procedure. The dimensionless temperature is close to 0.7. With the design procedure, the UFAD system produced a good thermal comfort environment.

Table 3: Testing results of air parameters in the cooling room

case	t_s (°C)	V (m ³ /h)	t_h (°C)	θ_f
1	19.1	456	25.6	0.64
2	18.8	580	25.1	0.63
3	18.7	680	25.3	0.67

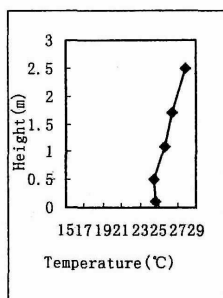


Figure 6: Vertical temperature gradient in case 1

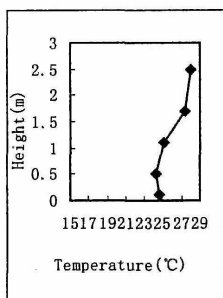


Figure 7: Vertical temperature gradient in case 2

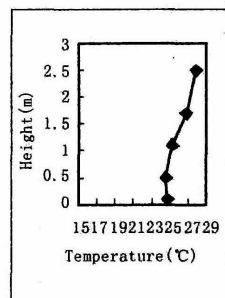


Figure 8: Vertical temperature gradient in case

case	Δt_{hf}	t_e (°C)	Δt_{ex} (°C)
1	0.9	27.8	8.7
2	0.5	28.0	9.2
3	0.5	27.8	9.1

4.2 Room temperature distribution

Figure 6, 7 and 8 show the vertical temperature gradient in the three cases. The major research findings are summarized below.

1. Under the relatively low supply volumes, the performance of the floor-based system resembles that of displacement ventilation systems. The horizontal temperature differences are small compared to the vertical ones. With the supply volume increasing, the vertical temperature gradient becomes smaller.

2. The lowest temperature is around the height of 0.5 m, which is determined by the characteristics of the swirl diffusers. Air temperature differences in the space between 0.1 m and 1.7 m are respectively 1.8°C, 2.7°C and 2°C, which accords with the requirement of ASHRAE Standard 55-1992.

5. CONCLUSIONS

A design method for UFAD systems with swirl diffusers was developed and tested. Based on the characteristics of swirl diffusers, the dimensionless temperature, θ_f , in the design procedure is around 0.7, which is different from that calculated in DV design procedure.

The experiments showed that design method was suitable for UFAD systems with swirl diffusers. Temperature of room air and temperature difference between the head and foot level in the experiment was even better than that calculated in design procedure.

6. ACKNOWLEDGEMENTS

The authors would like to acknowledge the support and technical assistance of The Trox Company.

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