Effect of Supply and Return Locations of a Floor-Supply Cooling System on Thermal Comfort

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Key words: Floor-supply cooling, Thermal comfort, Supply, Return, PMV, PPD

Abstract

This study numerically investigates thermal comfort of a space cooled by a floor-supply air-conditioning system, in which three different combinations of supply and return locations, one floor-supply/ceiling-return and two floor-supply/floor-return, are treated. A complementary experiment is performed to validate the present numerical analysis, and the prediction agrees favorably with the measured data. In the numerical procedure, a simplified model mimicking the inlet flow through a diffuser is developed for efficient simulations. The calculated results show that the ceiling-return type is far better in terms of thermal comfort than the floor-return ones within the extent of this study, which seems to be caused by effective vertical penetration of the supply air against natural convection. It is also revealed that the arrangement of port locations in the floor-supply/floor-return system has insignificant effect on the cooling performance. For selecting a proper system, other characteristics including the heating performance should be accounted for simultaneously with the present considerations.

Nomenclature

$A, \cdots, H$: locations
F_{p-i} : radiation configuration factor
PMV : predicted mean vote
PPD : predicted percentage of dissatisfied [%]
T : temperature [℃]
T_{rad} : mean radiant temperature [℃]
V_m : air velocity [m/s]
x, y, z : coordinate system, Fig. 1
$\langle \rangle$: volumetric average

Subscripts

b, f, l, r : back, front, left and right side of the test space
i : representative surface of each wall

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1. Introduction

Recent development in industry and living standard has increased the importance of indoor thermal environment. Precise control of thermal environmental factors such as temperature, humidity, air velocity and radiant temperature increases not only productivity in industry but also thermal comfort in living spaces. When developing a HVAC system, thermal comfort, environmental friendliness, and energy saving should be evenly considered besides the system performance. In Korea, energy consumption for cooling and heating buildings totaled about 15 billion dollars in 1997.

Traditional air-conditioning system focused on controlling thermal factors such as temperature, humidity, air velocity and radiant temperature. But nowadays indoor air quality that deals with particles is becoming very important too. Modern buildings are becoming more and more airtight to minimize uncontrolled infiltration of external air. This can reduce the consumption of energy for cooling and heating, but can cause serious problems to indoor air quality, since natural ventilation which replaces the contaminated indoor air with outdoor fresh air is no longer sufficient. There is a large variation in the fresh air requirements for people depending on the usage and purpose. ASHRAE Standard 62-1989 recommends $30\sim54\,m^3/(h \cdot person)$ of outdoor air for commercial facilities. Ventilation which provides fresh air and removes contaminants can occur by natural means or can be promoted by mechanical means using a fan. Performance of ventilation depends not only on the nominal air change rate but also on the air flow distribution which is related to the method of supplying and extracting air. For a poorly designed ventilation system, large air change rate may not guarantee good air-flow distribution in the room. One example is an exhaust being too close to an inlet, which makes the supply air short circuit to the exhaust. Thus, in order to achieve good ventilation performance with less fan power consumption, factors affecting the air-flow distribution should be carefully studied and implemented in the design and operation of the ventilation system.

The floor supply air-conditioning (FSAC) system is an air distribution system delivering conditioned air from floor level. This system has many advantages over traditional ceiling-supply system because diffusers are placed closer to the occupied zone.[1-3] In the past, FSAC system has been mostly applied to computer rooms for cooling purposes. Computers generate a large amount of heat and cool air supplied from the floor can efficiently absorb and exhaust heat. The return outlets are usually placed in the ceilings so that the flow can be assisted by buoyancy. As air absorbs heat it gets lighter and tends to move upward.

FSAC system of the past was mainly for cooling equipment and did not consider thermal comfort of human as an important factor. In these days, HVAC & R (heating, ventilating, air-conditioning and refrigerating) engineers are studying to adopt FSAC system for human occupant spaces. Thermal comfort and energy saving became very important in modern buildings and FSAC is an ideal system that can satisfy both these requirements.

Indoor thermal factors depends on the air-flow distribution which is related to the supply and exhaust locations, wall conditions and loads.[4] Among these factors, supply and exhaust locations are most significant and there have been numerous studies related to this topic.[5,6] For FSAC system, however, effect of supply and exhaust locations has not been extensively studied.

In this paper, effect of supply and exhaust locations for cooling in a FSAC system has been conducted using numerical analysis. Numerical results are verified by comparing with experimental results for a representative case. Studies were done for 3 cases in which the exhaust is at the floor or ceiling. Supply is located at the floor for all cases. For effective computation analysis, a simplified diffuser model is developed. Thermal environment is analyzed in terms of PMV and PPD. Thermal comfort and energy saving are also discussed.

2. Experiment

Experiment is done to study the characteristics of floor-supply air-conditioning system and also supply experimental data for verifying the numerical analysis. Schematics of a thermal chamber used in the experiment is shown is Fig. 1. Size of 4.5 m, 5.5 m and height 2.4 m were selected to simulate an ordinary office space. Four supply and 4 exhaust diffusers are located as shown in the figure. Diameter of a diffuser is 0.24 m and detailed geometry is described later. Exhaust is square shaped with 0.3 m on each side. Floor and ceiling are insulated and 4 walls are covered with 18 mm thick aluminum plates. Copper tubes with a diameter of 7.94 mm were attached to the plates so that the wall can be controlled to the required temperature by circulating temperature-controlled water in the tubes.

Vertical temperature distribution was measured using T-type thermocouples which were installed on a traverse system. Ten vertical points (z=0.0, 0.1, 0.3, 0.6, 0.9, 1.1, 1.3, 1.7, 2.1, 2.4 m) and 8 horizontal locations designated as A-H, as shown in Fig. 2, were the measurement points. Locations C and F coincide with the supply diffuser locations.

Supply temperature was 18℃, and air flow rate was 720 cmh which is equal to the air change rate of 12 times per hour. Though all wall temperatures were controlled to be at the same temperature, there were some deviations. Actual measured average temperatures were T_r=35.6℃, T_l=36.7℃, T_f=36.8℃ and T_b=36.9℃.

3. Diffuser

Supply diffuser structure has a great effect on the distributions of indoor temperature and velocity.[7] Supply diffusers are the most important factor of a FSAC system since it directly affects the thermal distribution of indoor

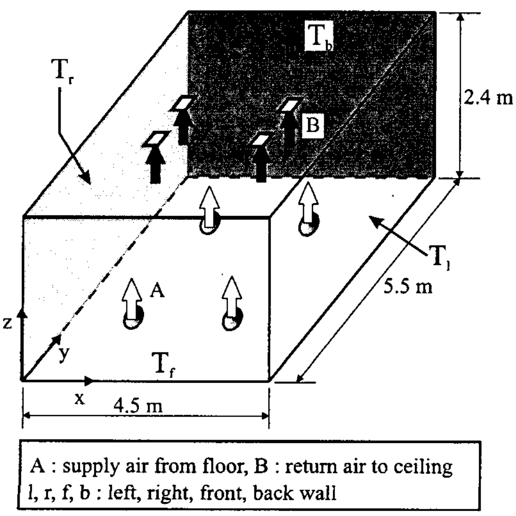

Fig. 1 Schematic of the experimental chamber.

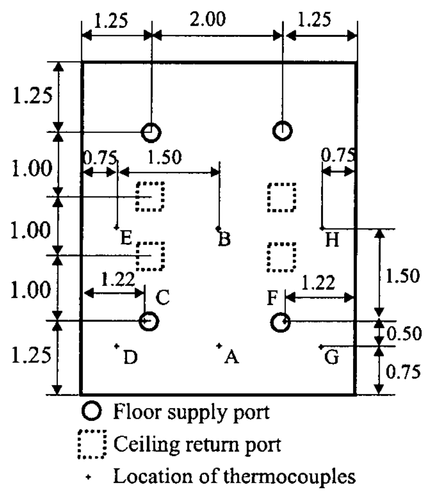

Fig. 2 Locations of supply/return ports and temperature measurement points.

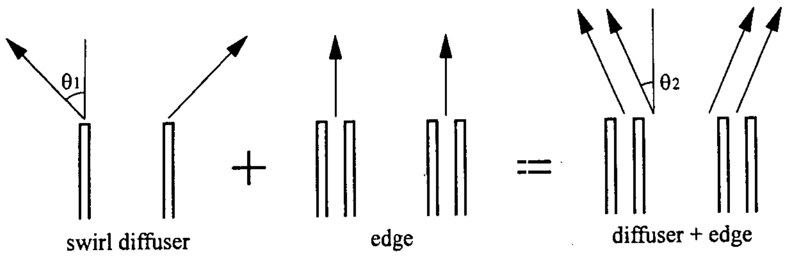

Fig. 3 Diffuser with swirl intensity and diffuse angle control features.

air. Since diffusers are located close to the occupant area, it is important that the air velocity is reduced to a certain level (less than 0.2 m/s) before reaching occupant space. Human bodies are known to feel uncomfortable draft if the air velocity around the body is too high. One technique for introducing a large volumetric flow of air with small velocity is swirl. Swirl will induce air from vicinity and enhance mixing. For human comfort, the temperature difference between 0.1 m (ankle location) and 1.7 m (head location) should be less than 3℃. Well-designed diffusers will minimize the vertical temperature difference. To prevent small objects – especially heels of ladies' shoes – from falling into the diffuser, the opening width of a floor diffuser is designed to be less than 7 mm.

Same diffusers are used for heating and cooling. During heating seasons, supply air is lighter than the air in the room and it tends to move upward due to buoyancy. To enhance mixing with the room air, it is advantageous to increase the diffuse angle during heating. On the contrary, it is advantageous to supply air vertically during cooling period. In this case supply air is heavier than the room air and it tends to move downward. Therefore a diffuser that has the feature of controlling the diffuse angle is suitable for a FSAC system. Fig. 3 shows the principle of a diffuser that has the feature of controlling the swirl intensity and diffuse angle. The diffuser consists of two parts which are center and edge sections. Swirl is generated in the center section and vertical flow is generated in the edge section. By controlling the air flow rate through the center and the edge, the swirl intensity and diffuse angle can be adjusted. If the edge section is fully closed, swirl intensity and diffuse angle will be maximum. If the swirl section is closed, swirl intensity and diffuse angle will be zero.

Fig. 4 shows the dimension of a diffuser that has been newly developed by adopting the principles explained earlier. Overall diameter is 0.24 m and the slope of the path in the internal section is 37.5 degree. To simplify the numerical computation, however, only the internal section which generates swirl is open. The edge section is assumed to be fully closed in this study.

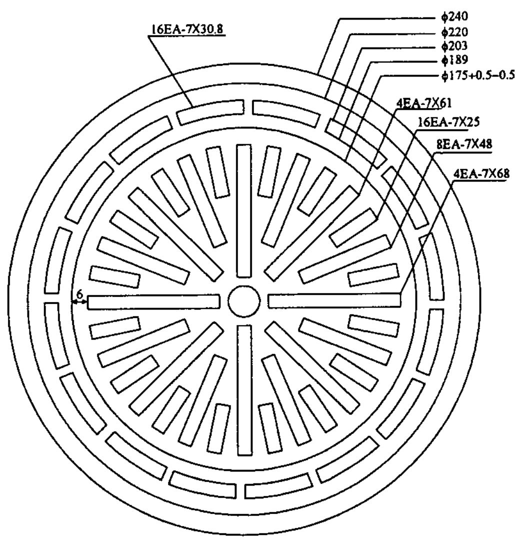

Fig. 4 Dimensions of the diffuser.

4. Locations of the supply and exhaust diffusers

Four combinations of supply and exhaust diffuser locations are considered. Experiment was carried out only for floor-supply/ceiling-return as shown in Figs. 1 and 2. These results were then compared with the computational analysis. Each case of computational analysis is denoted as I, II and III as shown in Fig. 5. In case I, supply and exhaust locations coincide from a top view. In case II, supply and exhaust diffusers are arranged in a straight line. To study the effect of diffuser locations in a floor-supply/floor-return, return locations are placed at the center while supply diffusers at the perimeter in case III. For all cases, number of supply and return diffusers are 4 each. Thus, total areas of supply and return diffusers are equal for all cases.

5. Thermal index

To quantify the effect of thermal environment on human comfort, various thermal indices were introduced. Among these indices, PMV (predicted mean vote) introduced by Fanger[11] is most widely used for describing indoor thermal comfort. This index is derived based on an energy balance between human body and surrounding environment. About 1,300 western subjects were studied in the experiment and the resulting thermal sensation was categorized into 7 levels, from -3 (cold) to $+3$ (hot). Discomfort level increases as the value deviates from neutral value 0. PMV is based on temperature, humidity, mean radiant temperature, air velocity, clothing and metabolism. It is known that PMV can be used for Korean people if applied under certain constraints.[12] Calculation of PMV with predicted velocity and temperature distribution is done using ISO 7730 program.[13] Mean radiant temperature is calculated as Eq. (1) according to ISO 7726.[14] All temperatures are in Kelvin and F_{p-i} is a radiation shape factor between control surface and wall i.

$$T_{\mathrm{rad}}^4 = \sum_{i=1}^{6} T_i^4 F_{p-i} \tag{1}$$

Absolute humidity is assumed a constant value of 0.01 kg/kg. Clothing and metabolism are assumed 0.5 clo (0.08 m^2℃/W) and 1.0 met (58 W/m^2), respectively, according to ISO 7730[13] for a sedentary activity in an office during summer. The relation between PMV and PPD is given in Eq. (2).[13,14]

$$\mathrm{PPD} = 100 - 95 \times \exp(-0.03353\,\mathrm{PMV}^4 - 0.2179\,\mathrm{PMV}^2) \tag{2}$$

6. Numerical analysis

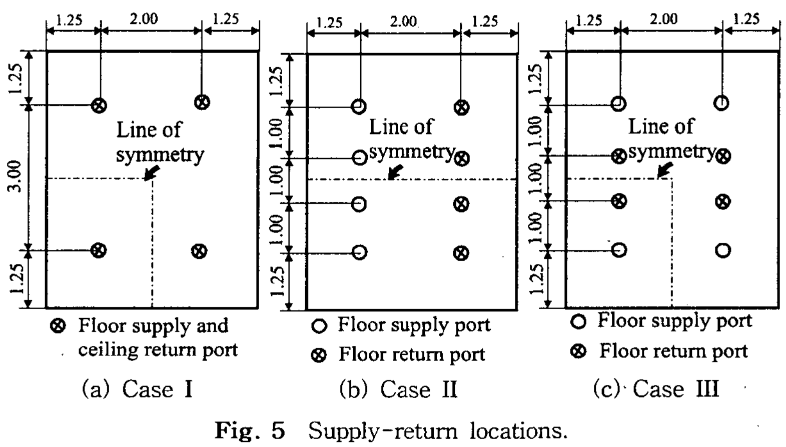

Fig. 5 Supply-return locations.

Computation was done using a commercial software. Since this program adopts an unstructured non-orthogonal grid structure, it is suitable for a complicated geometry such as the problem in this study. The grid structure with boundary condition for case I is given in Fig. 6 as a representative example. It represents 1/4 of the space shown in Fig. 5 (a). The grids are denser at the diffuser locations and near the walls. Though the well-known governing equations are omitted, some methodologies for the computation are as follows.[15-21] For turbulence model, $k-\varepsilon$, for coupling of pressure and velocity field, PISO (pressure implicit with splitting of operators) algorithm, for buoyancy to consider natural convection, ideal gas relation are used.

For analysis of indoor thermal-fluids system, wall boundary condition plays an important role. The results vary significantly depending on the types, locations and directions of walls. In reality, no wall conditions can be assumed

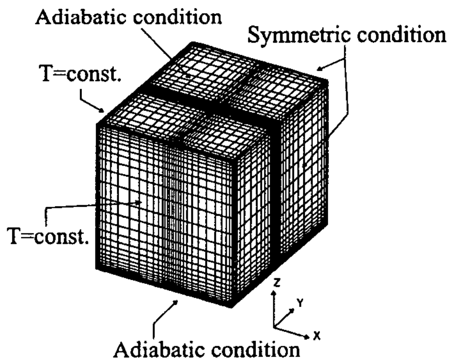

Fig. 6 An example of the grid system.

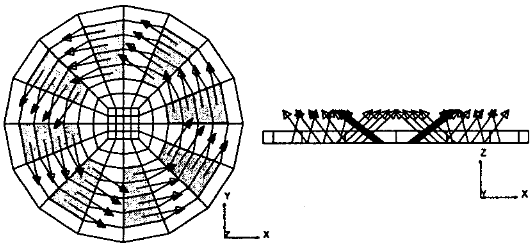

Fig. 7 Grid system and velocity vector for the diffuser.

as iso-thermal nor constant heat flux. In this study, however, to simplify the problem, wall conditions are assumed constant at a temperature of 37℃. Ceiling and floor are assumed insulated. A simplified diffuser model is developed as shown in Fig. 7. The velocity vectors at the diffuser outlet are assumed to be parallel to the path of the diffuser which is at an angle of 37.5 degree from the vertical line.

7. Results and Discussion

To check the validity of the numerical analysis, vertical temperature distributions are compared with the experiment as shown in Fig. 8. Both simulation and experiment are in good agreement except near the floor.

Representative calculation results are shown

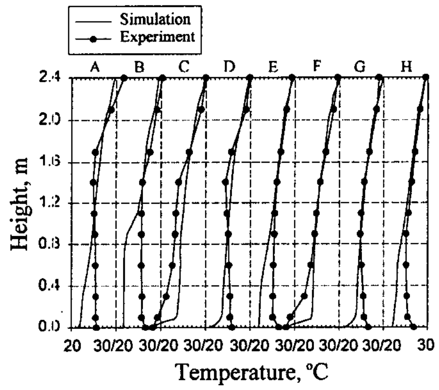

Fig. 8 Comparison of the vertical temperature distribution.

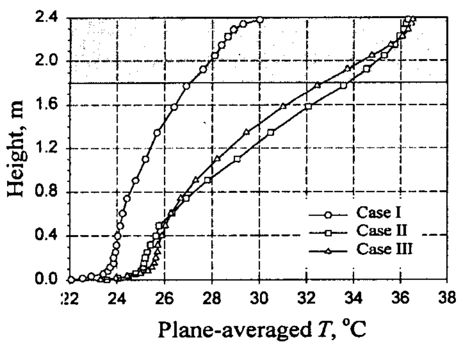

Fig. 9 Vertical variation of the plane-averaged air temperature.

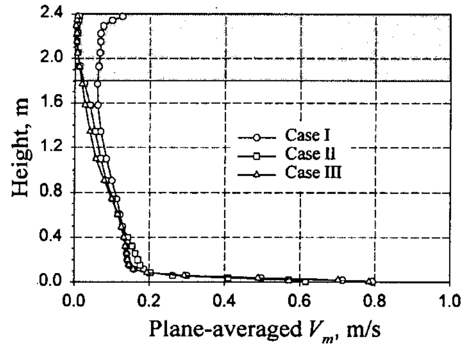

Fig. 10 Vertical variation of the plane-velocity magnitude.

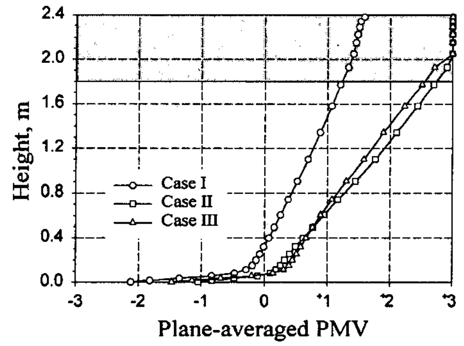

Fig. 12 Vertical variation of the plane-averaged PMV.

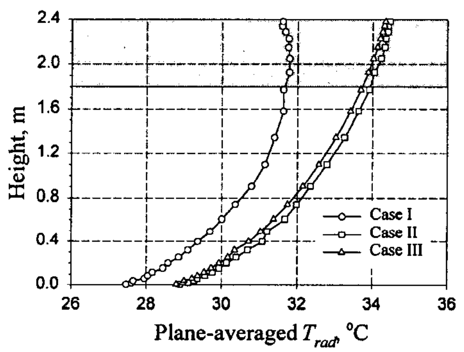

Fig. 11 Vertical variation of the plane-averaged mean radiant temperature.

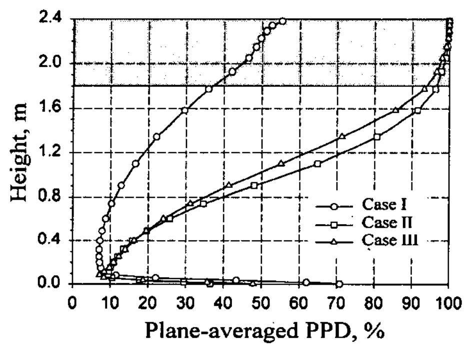

Fig. 13 Vertical variation of the plane-averaged PPD.

in Figs. 9~13. All results are plane-averaged and a non-occupied zone which is above 1.8 m is shown shaded. Though quantitatively different to some extent, thermal stratifications can be observed for all cases. Considering that ASHRAE[22] recommends a vertical temperature difference of 3℃ between 0.1 and 1.7 m height, cases II and III can not be considered as comfortable air-conditioning methods. As expected, low temperature supply air of floor-supply and floor-return can not penetrate effectively into the occupied zone before being exhausted due to buoyancy. It is also attributed to air supplied with a low swirl angle, since mixing is constricted to a region close to the floor only. Though similar, case III shows slightly better results than II, which indicates the effect of diffuser locations. Though based on numerical calculations with some constrained conditions, the results show that ceiling-return is superior to floor-return in terms of average temperature distribution.

Fig. 10 shows that similar trends exist for average velocity magnitude distribution for all cases. Velocity magnitudes are less than 0.2 m/s above 0.1 m height and thus become free of draft problem. Relatively larger velocity magnitudes of case I as shown in Fig. 10 is obviously due to the overall effect of induction of exhaust and natural buoyancy, which affects the temperature distribution shown in Fig. 9. Fig. 11 shows mean radiant temperature distribution calculated from imposed wall temperatures and calculated floor and ceiling surface temperatures. Since the mean radiant temperature is mainly affected by the wall temperatures, the difference between ceiling-return and floor-return is relatively small contrary to the temperature distribution.

To analyze the overall thermal environment, plane-averaged PMV and PPD distributions are calculated as shown in Figs. 11 and 12. As mentioned before, the effect of velocity magnitudes are negligible except for the region near the floor. Thus thermal indices are governed mainly by air and mean radiant temperatures. Increase of PMV due to height can be observed for all cases. Though PMV is in the comfortable range for the whole occupied region for case I, cases II and III show uncomfortable range for height over 0.7 m. This trend is also reflected in PPD. Since coolness or unsatisfaction are inherent features of floor-supply cooling system, it is necessary to put efforts in reducing these effects. Complete elimination of these problems, however, would be very difficult with FSAS system.

Calculated results of temperature, air velocity, mean radiant temperature, PMV and PPD for cooling are summarized in Table 1 for occupied zone (0~1.8 m), unoccupied zone (1.8~2.4 m) and the whole zone. Except for the slight difference in velocity magnitude, the results of ceiling-return are better than those of floor-return. For cases II and III which are similar except for the different return locations, the latter shows slightly better results.

Table 1 Summary of the calculated results

	Height [m]	Case I	Case II	Case III
$\langle T \rangle$ [℃]	0~1.8	25.0	28.5	28.0
	1.8~2.4	28.4	35.5	35.2
	0~2.4	25.8	30.0	29.7
$\langle V_m \rangle$ [m/s]	0~1.8	0.115	0.103	0.096
	1.8~2.4	0.074	0.008	0.006
	0~2.4	0.106	0.082	0.076
$\langle T_{rad} \rangle$ [℃]	0~1.8	30.5	32.2	32.0
	1.8~2.4	31.7	34.2	34.1
	0~2.4	30.8	32.7	32.5
⟨PMV⟩	0~1.8	+0.47	+1.43	+1.32
	1.8~2.4	+1.44	+3.00	+3.00
	0~2.4	+0.69	+1.87	+1.77
⟨PPD⟩ [%]	0~1.8	18.1	51.8	47.4
	1.8~2.4	47.6	99.0	98.5
	0~2.4	24.8	62.5	59.1

Though this study shows better thermal environment for ceiling-return in terms of cooling, it is necessary to consider other factors for a complete analysis. First, if the space requires heating, then it is natural to consider heating effect as well. Second, additional study is necessary for determining optimum diffuser configuration. For consistency, the same diffuser is used for all cases, but it is obvious that optimum diffuser configuration should exist depending on exhaust induction effect, natural buoyancy, and mixing of supply and indoor air. Diffuser favorable for ceiling-return may not be so for floor-return and vice versa. Third, for floor-return, air-conditioning space in the ceiling would not be required, which is of great advantage in terms of space. Fourth, it is important to impose actual boundary conditions. Simplified boundary conditions, such as iso-thermal or constant heat flux are not legitimate for accurate indoor thermal analysis.

8. Conclusions

The objective of this study is to compare and analyze thermal comfort of a floor-supply cooling system for different supply and return locations. When numerical calculations are compared with the experiment which is carried out for constant wall temperature for floor-supply and ceiling-return, both were in good agreement. Based on this validity, numerical calculations for 3 other cases with different supply and return diffuser locations have been carried out. The diffuser used in this study has been newly developed for floor-supply system. For efficient numerical analysis, a simplied diffuser model has been developed.

Calculation results are plane-averaged for comparison and analysis. Except for a little disadvantage in terms of velocity magnitude, ceiling-return is superior to floor-return for all factors compared. This is due to the fact that supply air penetrates more efficiently to the occupied zone due to induction and forced convection. For latter case, since forced and natural convections operate in the same direction, mixing is enhanced in the region near the floor and can not penetrate into the occupied zone. It is also found that locations of diffusers do not play a significant role for floor-return system.

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