Effect of supply air temperature on air distribution in a room with radiant heating and mechanical ventilation

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INTRODUCTION

Radiant heating systems, such as floor heating systems (FH) and ceiling heating systems (CH) are regarded as energy efficient and comfortable heating systems; so, they have been extensively used in the modern buildings (Watson and Chapman 2002; Olesen 2002; Babiak et al. 2009; Bean et al. 2010). Many energy-efficient building technologies involving increased thermal insulation and airtightness have also been applied in the modern buildings with radiant heating systems. Unfortunately, these technologies may cause insufficient fresh air to be supplied by infiltration and may thus lead to poor indoor air quality and increased health symptoms (Wargocki et al. 2000; Daisey et al, 2003). To avoid this problem, a mechanical ventilation system, such as a mixing ventilation system (MV) or a displacement ventilation system (DV) for fresh air supply must be integrated with the radiant heating systems (Olesen et al. 2011; Krajcik et al. 2013; Wu et al. 2014).

A hybrid system with a radiant heating system and a mechanical ventilation system, which is regarded as an advanced HVAC system, has been applied in many modern buildings worldwide (Skistad 2003; Ouazia et al. 2011; Liu et al. 2013). The hybrid system normally includes at least one of the following cases: (1) a floor heating system combined with a mixing ventilation system (FH+MV), (2) a floor heating system combined with a displacement ventilation system (FH+DV), (3) a ceiling heating system combined with a mixing ventilation system (CH+MV), or (4) a ceiling heating system combined with a displacement ventilation system (CH+DV).

The mechanical ventilation systems, which are used in heating season, are generally integrated with a heat recovery system to save energy (Feist et al. 2005). The supply air temperature of the ventilation system depends on the indoor air temperature as well as the inlet air temperature and efficiency of heat recovery system (CEN 2003). Few studies focus on the effect of supply air temperature on air distribution with all combinations of radiant heating systems with mechanical ventilation systems. Therefore, in this paper, the effects of supply air temperature on air distribution in a room with floor heating or ceiling heating and mixing ventilation or displacement ventilation were studied. The results in this paper are relevant to the design and control of the hybrid systems with radiant heating systems and mechanical ventilation systems.

METHODOLOGIES

Test Room

Measurements were performed in a simulated multi-occupant room, as shown in Figure 1. The room has an approximate floor area of 72 [m.sub.2] (775 [ft.sub.2]) and a ceiling height of 2.7 m (8.9 ft). Eight heated cylinders equipped with 80 W (273 Btu/h) light bulbs were used to simulate office workers. Other internal heat sources, such as computers and lights, were also considered in the study, resulting in a total power of 1650 W (5630 Btu/h) when including the heated cylinders.

The exterior wall has a total area of 32.4 [m.sup.2] (348.8 [ft.sup.2]), including a total window area of 13.2 [m.sup.2] (142.1 [ft.sup.2]) (see Figure 2). The total heat transfer coefficients of windows and exterior wall were approximately 1.5 W * [m.sup.-2] [K.sup.-1] (0.26 Btu/[ft.sup.2]*h*[degrees]F) and 0.64 W * [m.sup.-2] [K.sup.-1] (0.11 Btu/[ft.sup.2]*h*[degrees]F), respectively (Gatis 2004). The mean outdoor air temperature during the measurements was from 2.3[degrees]C to 6.0[degrees]C (36.1[degrees]F to 42.8[degrees]F) with an average of 4.0[degrees]C (39.2[degrees]F), which is normal for a mild winter, and the resulting mean heat loss from windows, roof, and exterior wall was 1455 W (4965 Btu/h).

As shown in Figures 1 and 2, the distributions of internal heat sources as well as supply terminals and exhaust terminals are symmetrical in the left and right side of test room.

Test Systems and Conditions

The room was equipped with four hybrid systems: floor heating system combined with mixing ventilation system (FH+MV), floor heating system combined with displacement ventilation system (FH+DV), ceiling heating system combined with mixing ventilation system (CH+MV), and ceiling heating system combined with displacement ventilation system (CH+DV), as shown in Figure 3.

Fresh air was supplied through four supply terminals in the ceiling for mixing ventilation and through five supply terminals at the base of the interior wall for displacement ventilation, as shown in Figure 1(b) and Figure 2(b). The exhaust air was extracted through four exhaust terminals at the top of the interior wall (see Figure 2 [b]).

The test conditions for the four hybrid systems are shown in Table 1, where the nominal air temperature and supply air flow rate refer to Category I for the non-low-polluting building in EN Standard 15251 (CEN 2007).

Supply terminal areas are different between mixing and displacement ventilation systems, which are 0.046 [m.sup.2] (0.5 [ft.sup.2]) for mixing ventilation system and 0.14 [m.sup.2] (1.5 [ft.sup.2]) for displacement ventilation system. The according supply air velocities are 4.8 m/s (945 fpm) for mixing ventilation system and 1.6 m/s (315 fpm) for displacement ventilation system. Besides, floor or ceiling surface temperature was adjusted to keep the nominal air temperature (dry-bulb temperature) at 22.0[degrees]C (71.6[degrees]F). The value of floor and ceiling surface temperature in Table 1 were obtained by solving the heat balance equation of the test room. During the experiment, the floor and ceiling surface temperatures were measured, which are the average of temperature measured along the heated surface.

Generally, the mechanical ventilation systems are integrated with a heat recovery system to save energy (Feist et al. 2005).The supply air temperature of the mechanical ventilation system depends on the indoor air temperature as well as the inlet air temperature and efficiency of heat recovery system (CEN 2003). In order to avoid frosting, the inlet air temperature should be more than -10[degrees]C (14[degrees]F), and thus the inlet air temperature is normally ranged from -10.0[degrees]C (14[degrees]F) to 5.0[degrees]C (51[degrees]F) (GB 50736 2012). Besides, until now the effectiveness of heat recovery system can be up to 80%. Hence, the supply air temperature of mechanical ventilation system with a heat recovery system is between 15.0[degrees]C and 19.0[degrees]C (59[degrees]F and 66.2[degrees]F), as shown in Table 1.

Measurement Parameters and Instruments

During the measurement, air temperature and velocity and pollutant concentration were collected by using spherical probes and the Innova system, as shown in Table 2. The measuring instruments were placed at four different heights (0.1 m [0.33 ft], 0.6 m [2.0 ft], 1.1m [3.6 ft], and 1.7 m [5.6 ft]) in each of the locations P1 through P4, as shown in Figure 1.

Due to the symmetrical distributions of temperature and velocity in the left and right side of test room, only distributions of temperature and velocity in the left side of test room were measured.

Freon 134a (tetrafluoroethane) was used as a tracer gas or contaminant to measure the air distribution effectiveness. Freon 134a dosing and sampling in the room were performed using the Innova 1303. One dosing point and one sampling point were separately placed in the supply duct and the exhaust duct of the ventilation system. Four sampling points (C1-C4) were located close to four of the dummies at a height of 1.1 m (3.6 ft) above floor level (corresponding to the breathing zone), as shown in Figure 1.

Due to no thermostat being installed in the room, the floor or ceiling surface temperature was adjusted by the author. The temperature measurements tests can be arrived at steady-state conditions after 5:00 p.m., then the measurement were taken.

Evaluation Indexes for Indoor Air Distribution

The vertical air temperature difference and local turbulence intensity was used in this study to evaluate the vertical distribution of air temperature and velocity. The air-distribution effectiveness was used to evaluate the horizontal distribution of contaminant concentration.

According to ISO Standard 7730 and ASHRAE 55, the vertical air temperature difference is the temperature difference between room air at a height of 0.1 m (0.33 ft) and room air at a height of 1.1 m (3.6 ft), as shown in Equation 1:

[DELTA] [t.sup.a0.1-1.1] = [t.sup.a1.1] - [t.sup.a0.1] (1)

where

[DELTA] [t.sup.a0.1-11] = vertical air temperature difference

[t.sub.a1.1] = room air temperature at a height of 1.1m (3.6 ft)

[t.sub.a0.1] = room air temperature at a height of 0.1m (0.33 ft)

Turbulence intensity can be calculated using Equation 2:

Tu = [SD.sup.v]/[bar.v] x 100% (2)

where

Tu = local turbulence intensity near the neck of occupants at a height of 1.1 m (3.6 ft)

[SD.sup.v] = standard deviation of room air velocity at a height of 1.1 m (3.6 ft)

[bar.v] = mean room air velocity at a height of 1.1 m (3.6 ft)

Air-distribution effectiveness (ASHRE 1997) (or ventilation effectiveness) includes contaminant removal effectiveness (CRE) and air change efficiency (ACE) (Mundt et al. 2004). The former indicates the ability of a ventilation system to remove airborne contaminants, and the latter indicates the ability of a ventilation system to exchange the air in the room. As the focus was on the fresh air supply, ACE is used in this paper. The tracer step-down method (or the decay method) were adopted in this study, so ACE can be calculated using Equations 3 through 5 (Mundt et al. 2004).

ACE = [[tau].sup.n]/<[bar.[[tau].sup.p]]> (3)

[MATHEMATICAL EXPRESSION NOT REPRODUCIBLE IN ASCII] (4)

[MATHEMATICAL EXPRESSION NOT REPRODUCIBLE IN ASCII] (5)

where

ACE = air distribution effectiveness

[[tau].sup.n] = time constant

<[bar.[[tau].sup.p]]> = local mean age of room air

[c.sup.e](0)/[c.sup.e](t) = initial/instantaneous contaminant concentrations in the exhaust duct

[c.sup.p](t) = instantaneous local contaminant concentration in room

RESULTS AND DISCUSSIONS

The vertical distribution of air temperature and velocity in the occupied zone and horizontal distribution of contaminant concentration in the breathing zone for the four hybrid systems were measured in this study.

Vertical Air Temperature Profiles in the Occupied Zone

Vertical air temperature profiles in the occupied zone for the four hybrid systems are shown in Figure 4, where mean air temperature is the mean value of local air temperatures at different measurement locations and the local air temperature is the average of the air temperature at one location over the measurement period.

As shown in Figure 4, when supply air temperature was between 15.0[degrees]C and 19.0[degrees]C (59[degrees]F and 66.2[degrees]F), vertical distribution of air temperature in the occupied zone was more uniform with mixing ventilation compared to displacement ventilation when it was with floor or ceiling heating. Substituting the mean air temperatures at a height of 0.1 m (0.33 ft) and at a height of 1.1 m (3.6 ft) into Equation 1, we can get the vertical air temperature difference for the four hybrid systems, as shown in Table 3.

Table 3 shows that the vertical air temperature differences were less than 0.3[degrees]C (32.5[degrees]F) in the room with FH+MV and CH+MV when supply air temperature was between 15.0[degrees]C and 19.0[degrees]C (59[degrees]F and 66.2[degrees]F). The supply air temperature has nearly no impact on the vertical air velocity profiles for mixing ventilation with floor or ceiling heating. This is mainly due to the fact that the air in the room with mixing ventilation was almost completely mixed.

Table 3 also shows that the vertical air temperature differences were between 1.9[degrees]C and 4.0[degrees]C (35.4[degrees]F and 39.2[degrees]F) in the room with FH+DV and between 2.5[degrees]C and 4.2[degrees]C (36.5[degrees]F and 39.6[degrees]F) in the room with CH+DV. The vertical air velocity profiles for displacement ventilation with floor or ceiling heating increased as the supply air temperature reduced. This is mainly due to the fact that the vertical air temperature difference with displacement ventilation was mainly determined by the temperature difference between indoor air and supply air when the supply air flow rate was kept constant (Causone et al. 2010; Wu et al. 2013). The reduced supply air temperature will increase the temperature difference between indoor air and supply air, as well as the vertical air temperature difference. The supply air temperature should be more than 17[degrees]C (62.6[degrees]F) to avoid the local thermal discomfort regarding the vertical air temperature difference (<3[degrees]C [37.4[degrees]F]) according to ISO Standard 7730 (2005) and ASHRAE Standard 55 (2010), as shown in Table 3.

Vertical Air Velocity Profiles in the Occupied Zone

Vertical air velocity profiles in the occupied zone for four hybrid systems are shown in Figure 5, where mean air velocity is the mean value of local air velocities at different measurement locations and the local air velocity is the average of the air velocity at one location over the measurement period.

As shown in Figure 5, when supply air temperature was between 15.0[degrees]C and 19.0[degrees]C (59[degrees]F and 66.2[degrees]F), vertical distribution of air velocity in the occupied zone was more uniform with mixing ventilation compared to displacement ventilation when it was with floor or ceiling heating. Besides, supply air temperature has nearly no impact on the vertical air velocity profiles for the four hybrid systems.

Substituting the standard deviation of room air velocity and mean room air velocity at a height of 1.1 m (3.6 ft) into Equation 2, we can get the turbulence intensity for the four hybrid systems, as shown in Table 4.

Table 4 shows that the turbulence intensities were from 12.5% to 15.5% with FH+MV or CH+MV and from 6.0% to 10.8% with FH+DV or CH+DV when supply air temperature was ranged from 15.0[degrees]C to 19.0[degrees]C (59[degrees]F to 66.2[degrees]F). The supply air temperature had slight influence on the turbulence intensity for the four hybrid systems. Besides, the turbulence intensity values were all larger for mixing ventilation than for displacement ventilation, which agreed with the results of Hanzawa and Melikov's studies (Hanzawa et al. 1987; Melikov et al. 1990). This is probably because cold air supplied by mixing ventilation enters the occupied zone directly, so that the air speed at the neck level is determined by the inertial force of the supply jets with a high discharging air velocity and, thus, has large fluctuation. For displacement ventilation, indoor air flow is determined by the thermal buoyancy, and it is free to vary and has low air velocity and small fluctuation.

Contaminant Concentration in the Breathing Zone

Contaminant (Freon 134a) concentrations in the breathing zone for four hybrid systems are shown in Figure 6, where mean contaminant concentration is the mean value of local contaminant concentration at different measurement locations, and the local contaminant concentration is the average of the contaminant concentration at one location over the measurement period.

Figure 6 shows that supply air temperature has nearly no impact on the distribution of mean contaminant concentration in the breathing zone for the four hybrid systems when supply air temperature was between 15.0[degrees]C and 19.0[degrees]C (59[degrees]F and 66.2[degrees]F). Substituting the instantaneous local contaminant concentration in room and the exhaust duct into Equations 3 through 5, we can get the air-distribution effectiveness values for four hybrid systems, as shown in Table 5.

As shown in Table 5, the air-distribution effectiveness for FH+MV and CH+MV were close to 1.0, which is the recommended value for mixing ventilation with a ceiling supply of cool air in ASHRAE Standard 62.1 (2007). The air-distribution effectiveness for FH+DV and CH+DV was from 1.06 to 1.16, which is slightly less than the recommended value (1.2) for displacement ventilation with floor supply of cool air and ceiling return in ASHRAE Standard 62.1 (2007). This may be due to the effect of the downdraft caused by cold windows and external wall and the small temperature different between supply air and room air. Table 5 also shows that the supply air temperature had nearly no influence on the air distribution effectiveness for FH+MV and CH+MV but slight impact on the air distribution effectiveness for FH+DV and CH+DV.

CONCLUSIONS

The effects of supply air temperature on air distribution in a room with floor or ceiling and mixing or displacement ventilation were studied when supply air temperature were ranged from 15.0[degrees]C to 19.0[degrees]C (59[degrees]F to 66.2[degrees]F).The following conclusions can be drawn:

1. The vertical air temperature difference did not change for FH+MV or CH+MV, but greatly increased for FH+DV or CH+DV as the supply air temperature was reduced.

2. The supply air temperature had slight influence on the turbulence intensity for the four hybrid systems.

3. The supply air temperature had nearly no influence on the air-distribution effectiveness for FH+MV or CH+MV, but had a slight impact on the air-distribution effectiveness for FH+DV or CH+DV.

4. The air distribution effectiveness for FH+DV and CH+DV was slightly less than the recommended value (1.2) for displacement ventilation with the floor supply of cool air and ceiling return in ASHRAE Standard 62.1 (2007).

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Caption: Figure 1 Setup of the test room and location of measurement sensors (C are the measurement points for contaminant concentrations, and P are the measurement points for air temperatures and air velocities).

Caption: Figure 2 Exterior and interior walls.

Caption: Figure 3 Schematic diagram of test systems.

Caption: Figure 4 Vertical air temperature profiles in the occupied zone for the four hybrid systems.

Caption: Figure 5 Vertical air velocity profiles in the occupied zone for the four hybrid systems.

Caption: Figure 6 Contaminant concentrations in the breathing zone for four hybrid systems.

Table 1. Test Conditions for the Four Hybrid Systems

Hybrid System	Supply Air Temperature, [degrees]C ([degrees]F)	Supply Airflow Rate, L/s (cfm)	Nominal Air Temperature, [degrees]C ([degrees]F)
	15.0 (59.0)	224 (475)	22.0 (71.6)
FH+MV	17.0 (62.6)	224 (475)	22.0 (71.6)
	19.0 (66.2)	224 (475)	22.0 (71.6)
	15.0 (59.0)	224 (475)	22.0 (71.6)
FH+DV	17.0 (62.6)	224 (475)	22.0 (71.6)
	19.0 (66.2)	224 (475)	22.0 (71.6)
	15.0 (59.0)	224 (475)	22.0 (71.6)
CH+MV	17.0 (62.6)	224 (475)	22.0 (71.6)
	19.0 (66.2)	224 (475)	22.0 (71.6)
	15.0 (59.0)	224 (475)	22.0 (71.6)
CH+DV	17.0 (62.6)	224 (475)	22.0 (71.6)
	19.0 (66.2)	224 (475)	22.0 (71.6)

Hybrid System	Ceiling Surface Temperature, [degrees]C ([degrees]F)	Floor Surface Temperature, [degrees]C ([degrees]F)	Internal Heat Load, [degrees]C ([degrees]F)
	--	~29.0 (84.2)	1650 (5630)
FH+MV	--	~29.0 (84.2)	1650 (5630)
	--	~29.0 (84.2)	1650 (5630)
	--	~29.0 (84.2)	1650 (5630)
FH+DV	--	~29.0 (84.2)	1650 (5630)
	--	~29.0 (84.2)	1650 (5630)
	~29.0 (84.2)	--	1650 (5630)
CH+MV	~29.0 (84.2)	--	1650 (5630)
	~29.0 (84.2)	--	1650 (5630)
	~29.0 (84.2)	--	1650 (5630)
CH+DV	~29.0 (84.2)	--	1650 (5630)
	~29.0 (84.2)	--	1650 (5630)

Table 2. Measuring Parameters and Instruments

Parameter	Instrument
Air velocity	Spherical probe
Air temperature	Spherical probe
Contaminant concentration	Innova 1412

Parameter	Range
Air velocity	0.05-1.00 m/s (10-200 fpm)
Air temperature	10[degrees]C-40[degrees]C (50[degrees]F-104[degrees]F)
Contaminant concentration	0-100 ppm (0-460 [micro]g/[m.sup.3])

Parameter	Accuracy
Air velocity	[+ or -] 0.02 m/s (4 fpm)
Air temperature	[+ or -] 0.2[degrees]C (0.4[degrees]F)
Contaminant concentration	[+ or -] 1 ppm (4.6 [micro]g/[m.sup.3])

Table 3. Vertical Air Temperature Difference for the Four Hybrid Systems, [degrees]C ([degrees]F)

Hybrid System	Test Conditions	
	[t.sub.s] = 15[degrees]C (59[degrees]F)	[t.sub.s] = 17[degrees]C (62.6[degrees]F)
FH+MV	0.2 (32.4)	0.0 (32.0)
FH+DV	4.0 (39.2)	2.7 (36.9)
CH+MV	0.3 (32.5)	0.3 (32.5)
CH+DV	4.2 (39.6)	3.3 (37.9)

Hybrid System	Test Conditions
	[t.sub.s] = 19[degrees]C (66.2[degrees]F)
FH+MV	0.1 (32.2)
FH+DV	1.9 (35.4)
CH+MV	0.2 (32.4)
CH+DV	2.5 (36.5)

Table 4. Turbulence Intensity for the Four Hybrid Systems, %

Hybrid System	Test Conditions		
	[t.sub.s] = 15.0[degrees]C (59[degrees]F)	[t.sub.s] = 17.0[degrees]C (62.6[degrees]F)	[t.sub.s] = 19.0[degrees]C (66.2[degrees]F)
FH+MV	12.8	15.5	13.40
FH+DV	10.8	9.8	7.7
CH+MV	12.5	14.1	14.1
CH+DV	6.5	6.0	7.5

Table 5. Air Distribution Effectiveness for the Four Hybrid Systems

Hybrid System	Test Conditions		
	[t.sub.s] = 15.0[degrees]C (59[degrees]F)	[t.sub.s] = 17.0[degrees]C (62.6[degrees]F)	[t.sub.s] = 19.0[degrees]C (66.2[degrees]F)

FH+MV	0.99	0.98	1.00
FH+DV	1.10	1.09	1.14
CH+MV	0.98	0.98	0.97
CH+DV	1.13	1.16	1.06