Performance of the raised floor heating system with natural ventilation

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ABSTRACT

In Korea, all residential buildings adopt floor- heating systems for indoor thermal comfort. And the ratio of multi-family houses exceeds 50%. Lately problems of noise between floors and contamination of indoor air in multi-family houses emerged as social issue, and performance standards related to noise between floors and ventilation were established in Housing Regulation. Concerning these issues, many systems have been developed. Recently developed raised floor heating system is not only capable of basic function to reduce noise between floors, but also is a multi-functional floor heating system enabling natural ventilation. The procedure of this system for natural ventilation is to import outdoor air through bottom space of the floor heating system, circulate indoor space and discharge it out of ceiling. In winter, powerless natural ventilation is possible with buoyancy effect caused by temperature difference between outdoor and indoor. And it also allows saving of energy by importing pre-heated air in bottom space of the floor heating system.
To evaluate ventilation performance of this system, onsite measurement was conducted in 2 test laboratories, and the nominal air change rate was satisfied as 0.4~0.8 h^{-1} under the condition of outdoor temperature 5~-5, which was evaluated as highly possible to be applied as a natural ventilation system in multi-family houses.

1. INTRODUCTION

In Korea, all residential buildings adopt floor-heating systems for indoor thermal comfort. And the ratio of multi-family houses exceeds 50%. Lately problems of noise between floors and indoor air contamination in multi-family houses emerged seriously as social issue, and performance standards related to noise between-floors and minimum ventilation were established in Housing Regulation. One of the systems developed for this is the raised floor heating system with double-layer. The raised floor heating system is capable of basic function to reduce noise between floors using rubber supports and arrangement of space under the system. This system not only has function to heat up indoor space basically and but also is a multi-functional floor heating system providing natural ventilation by utilizing air stream of bottom space. The procedure of this system for natural ventilation is to import outdoor air through bottom space of the floor heating system, circulate indoor space and discharge it out of ceiling.
In this study, possibility of practical application is reviewed by evaluation of ventilation performance of this system for a mock-up room and an experimental house.

2. RAISED FLOOR HEATING SYSTEM

This is a multi-functional system basically developed to enable natural ventilation in winter using buoyancy effect caused by temperature difference between indoor and outdoor and it is equipped also so as to enable mechanical ventilation for spring, autumn and summer season.

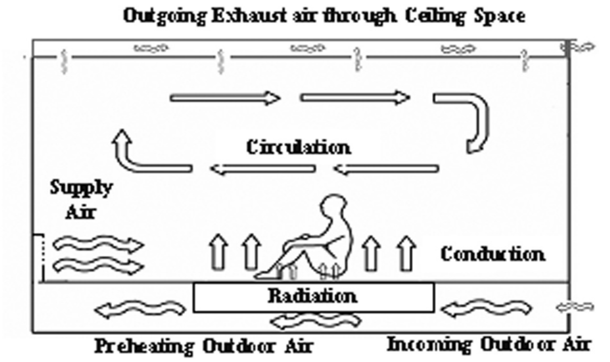
2.1 System with natural ventilation

This floor heating system has a function not requiring additional power owing to buoyancy effect arising from temperature difference between indoor and outdoor in winter. As Figures 1 and 2, initially cold outdoor air flows into the bottom space (height of 20mm) of the floor heating system through air channel owing to temperature difference between indoor and outdoor. Fresh cold air imported in bottom space is heated by the floor heating system, and thereafter, it flows into indoor space through supply grille. Warm air imported through supply grille is circulated in indoor space, and then flows out through exhaust grille in ceiling.
The air circulation is formed by pressure difference caused by temperature difference between indoor temperature, outdoor temperature and air heated in the bottom space of the floor heating system. This is energy-saving ventilation system because it is not necessary for additional power. In addition, as cold outdoor air flows into indoor space after being heated by excessive heat in the bottom of the floor heating system. So, cold draft problem at the time of importing directly outdoor air can be prevented.

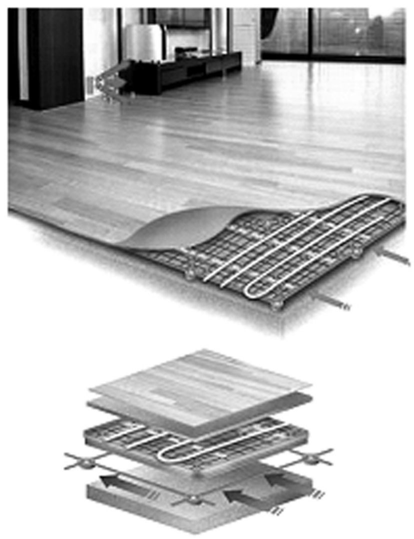
2.2 System with exhaust fan

The mechanical exhaust mode means a selective method to ventilate in the period when temperature difference between indoor and outdoor is not big and floor heating is not operated like summer.
As for the mechanical exhaust mode of this system, the route of air stream into the indoor space through air sup-

ply grille after passing through bottom space of the floor heating system is same as natural ventilation mode, but it is to exhaust compulsorily by operating fan installed in the back of grille on the ceiling as shown in Figure 2.



(a) Concept for air flow



(b) Air flow into the bottom space of the floor heating system
Figure 1 : Conceptual diagram for natural ventilation of the raised floor heating system

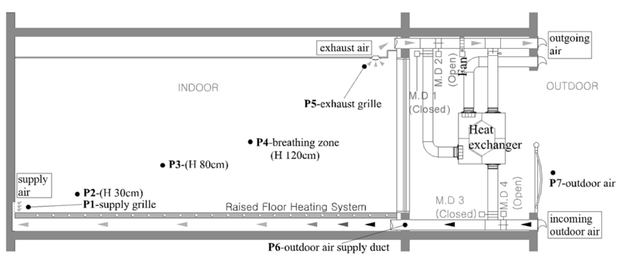


Figure 2 : the raised floor heating system with various ventilation conditions

2.3 System with supply & exhaust fan

The mechanical supply & exhaust mode of this system means a selective method to operate the fan for heat-exchange. It has an advantage to reduce cold draft problem and energy loss by increasing supply air temperature after recovering heat of exhaust air with heat exchanger in winter as shown in Figure 2.

3. EXPERIMENT OF VENTILATION PERFORMANCE

3.1 Experiment summary

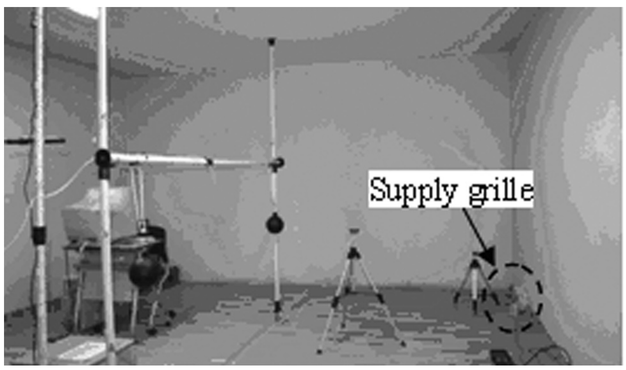
Ventilation performance of this system was evaluated in a mock-up room and an experimental house in winter. In case of the mock-up room, the system was applied as the composition of Figure 3.

In case of the experimental house, an exhaust fan is installed in front balcony as Figure 4b, the heat exchanger with a supply fan & an exhaust fan is installed in back balcony as Figure 4c and two supply grilles are installed in indoor space.

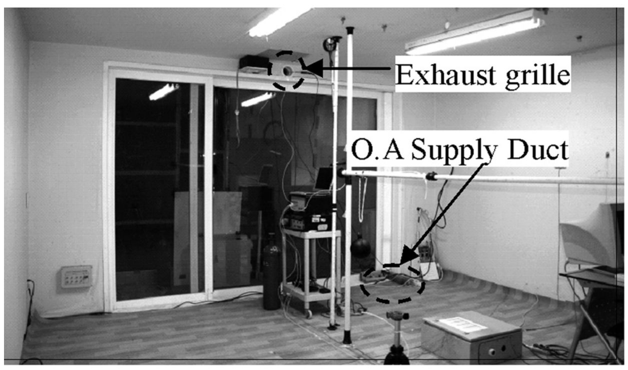
For tracer gas experiment, the positions of CO_2 gas sampler measurement are explained in Table 1 and they are shown in Figure 2. Air velocity and temperature measurement are explained in Table 2 and they are shown in Figures 2.

Table 1 Measurement position of sampler

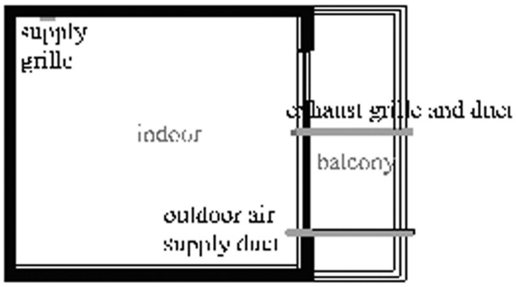
No.	Position of sampler
P1	supply grille
P2	30cm height from the floor
P3	80cm height from the floor
P4	120cm height from the floor (breathing zone)
P5	return grille



(a) Indoor air supply grille



(b) Air incoming part and exhaust grille



(c) Floor plan

Fig. 3 : Test in the mock-up room

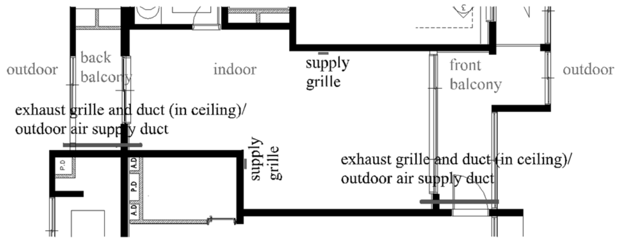
Table 2 Measurement position of air velocity and temperature

No.	Measurement position
P1	supply grille
P4	120cm height from the floor (breathing zone)
P5	return grille
P6	outdoor air incoming duct
P7	outdoor air

(a) Indoor air supply grille

(b) Air incoming part and exhaust grille on front side

(c) Air incoming part and exhaust grille on back side



(d) Floor plan

Figure 4 : Experimental House

3.2 Measurement of Age of air, air change effectiveness and calculation of air change rate

For evaluation of this system's ventilation performance, local age of air(A_i) and air change effectiveness(E_i) of each position was calculated by tracer gas density experiment and measurement of air current speed and temperature in major positions. In addition, ventilation performance of this system viewed through concerned index was evaluated by calculating the air change rate(I_n) through the nominal time constant(τ_n).

Formula for calculating air age(A_i) at location i in area is as following formula(1)(ASHRAE 1997(1)).

$$A_i = (1/C_o)\int_0^{\infty} C_i \delta t \qquad (1)$$

The age of air from tracer gas decay is calculated from the expression.

$$A_i = (t_{stop} - t_{start})C_{i,avg} / C_i(t_{start})$$

where

A_i = the age of air at location *i*

t_{stop} = the time of the final tracer gas measurement at location *i* during the tracer gas decay or, with time-integrated sampling at location *i*, the time when sampling is terminated

t_{start} = the time when outdoor airflow is started or tracer injection is stopped at the beginning of a tracer gas decay

$C_{i,avg}$ = the time-averaged tracer gas concentration at location *i* between t_{start} and t_{stop}

$C_i(t_{start})$ = the tracer gas concentration *i* at time t_{start}

$$C_{i,avg} = \frac{\sum_{n=first}^{n=last-1} [\frac{C_{i,n} + C_{i,n+1}}{2}(t_{n+1} - t_n)]}{(t_{last} - t_{first})}$$

The nominal air change rate(I_n), an index evaluating ventilation effectiveness, is calculated by the reciprocal number of the nominal time constant(τ_n) representing the time outdoor air stays in concerned zone or space (ASHREA 1997(2)), which is same as formula(2)(ASHRAE 1997(1)) and it is calculated by gross exhaust volume of imported air at exhaust gate at the time of final exhaust and air age.

$$\tau_n = \frac{\sum_m (Q_{ex,m} A_{ex,m})}{\sum_m Q_{ex,m}} \quad (2)$$

where,

τ_n = the nominal time constant

m = an identification number unique for each exhaust air stream

= the rate of air flow in exhaust airstream *m*

= the age of airflow in exhaust airstream *m*

= a summation for all m exhaust airstreams.

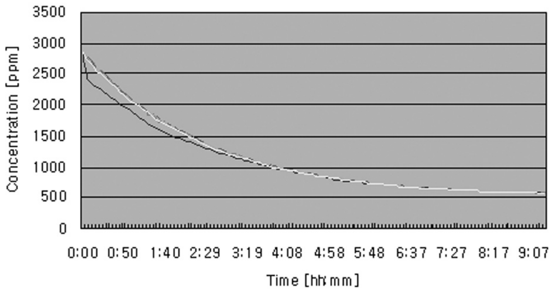
Ventilation efficiency is calculated from following formula(3).

$$E = \tau_n / A_{avg} \quad (3)$$

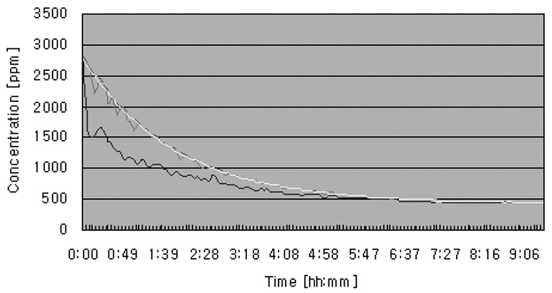
where,

E = the air-change effectiveness

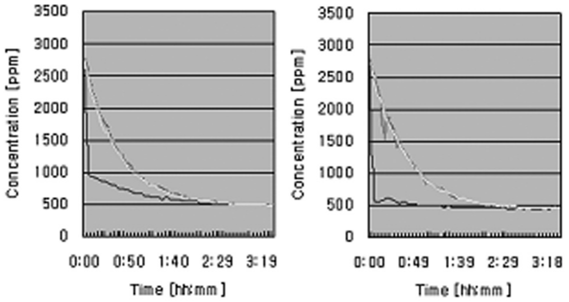
A_{avg} = the arithmetic average of the ages of air measured at breathing level within the test space



(a) Natural ventilation 1



(b) Natural ventilation2



(c) exhaust fan (d) supply & exhaust fan

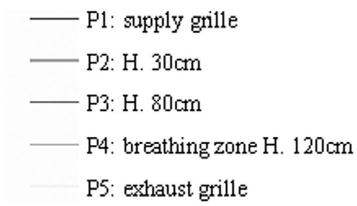


Figure 5 : The result of tracer gas test

4. RESULT AND EVALUATION OF EXPERIMENTS

4.1 The result of test in mock-up room

The conditions of test in experimental house are as same as Table 3 and the result of ventilation performance in the mock-up room is evaluated as Table 4 and Figures 5-6.

At the first natural ventilation experiment under the condition of indoor and outdoor temperature difference of 18.8K, supply air velocity at indoor supply grille was average 0.45m/s and at this time, supply air temperature was average 28.1, which was supplied to indoor space at temperature higher than indoor temperature after being heated.

At the second natural ventilation experiment under the condition of indoor and outdoor temperature difference of 27.5K, supply air velocity at indoor supply grille was average 0.62m/s, and at this time, supply air temperature was 30.0.

At the test with exhaust fan, supply air velocity at grille was average 0.52m/s close to that of natural ventilation. At this time, supply air velocity at indoor supply grille was average 29.2, which was supplied hotter than indoor temperature.

At the test with supply & exhaust fan, supply air velocity at indoor supply grille was average 1.12m/s, and at this time, supply air temperature was average 31.9.

As a result of test, natural ventilation performance is evaluated to be enough with the nominal air change rate 0.42 h^{-1}, 0.79 h^{-1}. In case of the second natural ventilation test, due to decrease of outdoor temperature, the nominal air change rate is increased from 0.42 h^{-1} to 0.79h^{-1}. So, natural ventilation is shown to be affected by temperature difference between indoor and outdoor very largely. In addition, in cases with exhaust fan and with supply & exhaust fan, the nominal air change rate is more than 1.5 h^{-1}, which is enough.

The result of tracer gas test in the mock-up room is same as Figure 5 and the air change effectiveness of each case and the local age of air of each position are same as Table 5.

Table 3 : The conditions of ventilation performance test

		Natural ventilation 1	Natural ventilation 2	Exhaust fan	Supply & exhaust fan
Out-door air	Temperature	4.3	-4.7	8.5	-4.4
	Air velocity (m/s)	1.0	4.4	0.7	2.1
	Air direction	SW	NW	SW	W

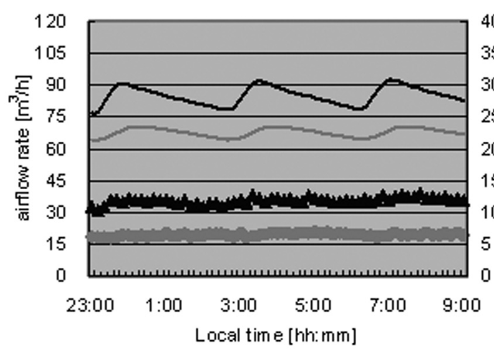
Indoor air temperature	22.6	22.8	22.5	22.9
ΔT between indoor/ outdoor[K]	18.8	27.5	14.1	27.3
Supply Air velocity(m/s)	0.45	0.62	0.52	1.16
Supply Air Temperature	28.1	30.0	29.2	31.9

Table 4 : The result of ventilation performance test

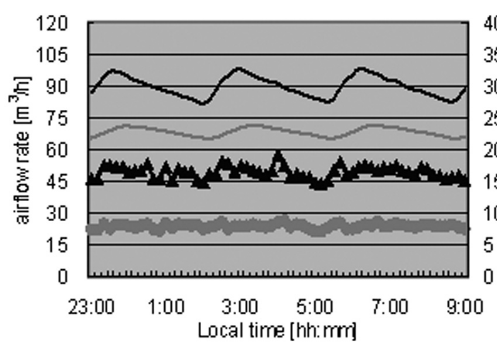
	Natural ventilation1	Natural ventilation2	Exhaust fan	Supply & exhaust fan
The Air change rate[I_n] (h^{-1})	0.42	0.79	1.56	1.79

Table 5 : The result of tracer gas test

	Natural ventilation1		Natural ventilation 2		Exhaust fan		Supply &exhaust fan	
	Ai (min)	Ei	Ai (min)	Ei	Ai (min)	Ei	Ai (min)	Ei
P1	128.7	1.11	43.0	1.76	14.8	2.44	5.3	6.22
P2	146.5	0.98	72.1	1.06	39.3	0.94	36.0	0.93
P3	146.5	0.98	75.9	1.01	38.7	0.95	32.9	1.02
P4	145.4	0.98	76.0	1.00	39.3	0.94	33.8	1.00
P5	142.9	1.00	76.3	1.00	36.8	1.00	33.8	1.00

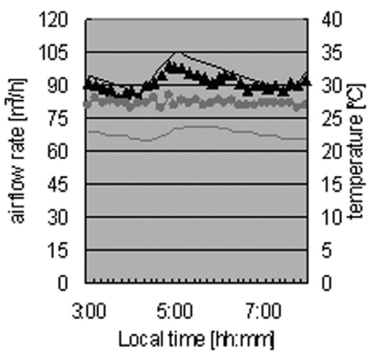


(a) Natural ventilation 1



(b) Natural ventilation 2

(c) Exhaust fan



(d) Supply & exhaust fan

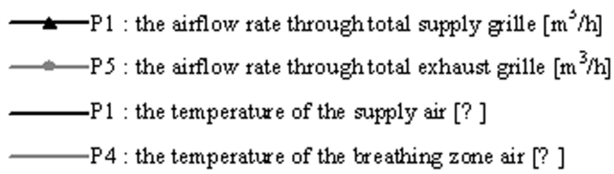


Figure 6 : Distribution of air flow rate and temperature

For all cases as shown in Figure 6, it was found that airflow rates in supply grille are similar with pattern of indoor temperature. That is the reason why air temperature in the bottom of floor-heating system changes according to pattern of heating-on/off. When air in the bottom of floor heating system is heated by hot water supply at the time of heating-on, pressure difference increases due to temperature difference, and then airflow rate also increases. When air temperature in the bottom of floor-heating system decreases at the time of heating-off, airflow rate decreases as well.

4.2 The result of test in experimental house

The conditions of test in experimental house are as same as Table 6 and the result of ventilation performance in the experimental house is as same as Table 7.

At the natural ventilation test under the condition of temperature difference of indoor and outdoor of 17.3K, supply air velocity at indoor supply grille was average 0.26m/s, and at this time, supply air temperature was heated up to average 29.7.

At the test with exhaust fan under the condition of temperature difference of indoor and outdoor of 17.9K, supply air velocity at indoor supply grille was average 0.35m/s close to that of natural ventilation and at this time, temperature of supply air was average 29.1, which was heated and supplied hotter than indoor temperature by about 6K.

At the test with supply & exhaust fan under the condition of temperature difference of indoor and outdoor of 18.5K, supply air velocity at indoor supply grille was

average 0.56m/s, and at this time, supply air temperature was average 32.6, which was heated and supplied hotter than indoor temperature by 9K.
The result of test in the experimental house showed the nominal air change rate per hour as $0.52h^{-1}$ under outdoor temperature of 5.6 as Table 6. Considering the basis of the result of actual object test, it was evaluated as showing enough ventilation performance. In addition, in cases with exhaust fan and with supply & exhaust fan, the result of tracer gas test shows the nominal air change rate as $1.91\ h^{-1}$ and $3.08\ h^{-1}$, which proves enough ventilation.

Table 6 : The conditions of ventilation performance test

		Natural ventilation	Exhaust fan	Supply & exhaust fan
Outdoor air	Temperature	5.6	4.9	4.6
	Air velocity(m/s)	0.8	1.0	0.7
	Air direction	NW	NW	NW
Indoor air temperature		22.9	22.8	23.1
ΔT between indoor/ outdoor[K]		17.3	17.9	18.5
Supply air velocity(m/s)		0.26	0.35	0.56
Supply air temperature		29.7	29.1	32.6

Table 7 : The result of ventilation performance test

	Natural ventilation	Exhaust fan	Supply & exhaust fan
The air change rate [I_n] (h^{-1})	0.52	1.91	3.08

5. CONCLUSION

In this study, after installing the raised floor-heating system was installed in 2 actual object laboratories, evaluation of ventilation performance of system using bottom space was conducted in winter.
As a result of on-site measurement, when outdoor air was imported under the condition of outdoor temperature of around -5, indoor supply air temperature is increased to about 30 forming temperature difference of 33K. This is driven as a condition to make indoor natural ventilation, and this temperature difference can sufficiently be used for passive means to induce natural ventilation. When this system was applied, in overall, it satisfied the nominal air change rate of $0.4\sim0.8h^{-1}$ under the condition of outdoor temperature of 5~-5, and as outdoor temperature is -6 7 in January when is the coldest season in Korea. So, this system as hybrid ventilation system using natural ventilation is evaluated as highly possible for application. It also allows saving of energy by importing pre-heated air in bottom space of the floor heating system. Moreover, it was proven that air temperature at supply grille was maintained higher than indoor temperature, by which it was proven to enable blocking of cold draft problem.

In future, it is needed to study the performance of natural ventilation using a floor-heating system in actual residence and plan study on hybrid ventilation system applicable in summer as well.

ACKNOWLEDGMENTS

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