AN ANALYSIS OF THE FRESH AIR LOAD REDUCTION SYSTEM BY USING UNDERGROUND DOUBLE FLOOR SPACE FOR AIR CONDITIONING

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ABSTRACT

This paper presents a feasibility study of a fresh air load reduction system by using an underground double floor space. The system was introduced into a real building (Aichi Children's Center in Japan) and was examined by the field measurement. Judging from the measurements during two years (summer, 1996 ~ summer, 1998), the state of the system operation was very stable through this period and it was clear that the system contributes to reduction of energy consumption for air-conditioning.

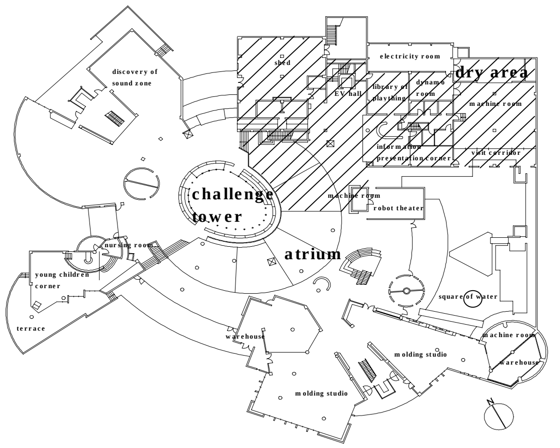
Furthermore, a simulation model used the simple heat diffusion equation was developed to simulate its thermal characteristics and performances. The simulations resulted in air temperature in good agreement with the measurements. Also, from the result of numerical analysis, it is clear that the amount of heat supply by using this system is more than the amount of energy loss to the room above it. Therefore, it is concluded that this system is very useful and the proposed numerical model can be used for the prediction of system thermal performance.

INTRODUCTION

In order to solve global environment issues and avoid exhaustion of natural resources, many technologies using various natural and unused energy have been proposed[1][2].

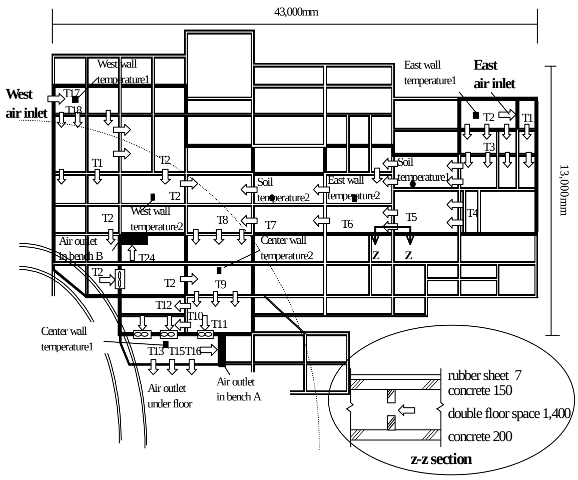
In this study, the feasibility of a fresh air load reduction system using underground double floor space is investigated experimentally and numerically. The fresh air is introduced into the double slab space and pass the opening bored through walls in it. And it is cooled through the heat exchange with its surface in summer, and heated in winter. Then the heat is conducted between a double slab and the adjacent soil and also transferred to the room just above the mat. The initial cost of the system can be reduced, because it doesn't need a pipe or duct in the ground as does the cool-tube system[3].

However, it has some problems to be solved. The first one is an increase in the heating/cooling load due to the heat transfer between the fresh air which is introduced into the double slab space and the room air. The second one is the deterioration of system



*Hatched area represents double slab used for this system.

Fig. 1. The floor plan of the building applied this system.



*Enclosed area with heavy line represents double slab used for this system.
*A number in section is a thickness of material.(unit : mm)

Fig. 2. The diagram of the under floor.

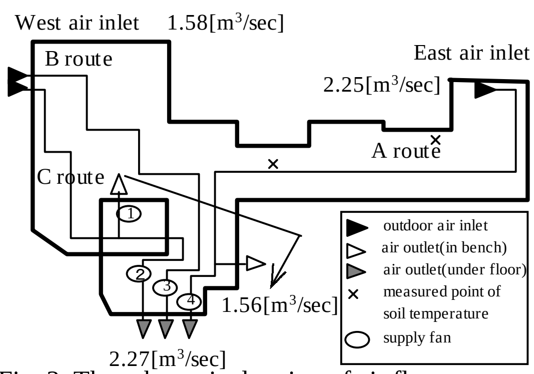


Fig. 3. The schematic drawing of air flow routes.

Table 1
Equipment schema

Supply fan	air volume : 4,000CMH, dynamic pressure : 5mmAq, input : 0.4kW
	operating number : four in summer, 1996 except : two
Operating condition	operating schedule : AM8:00~PM5:00
	on-off :
	on : outdoor air temperature is more than 25°C in summer
	outdoor air temperature is lower than 10°C in winter

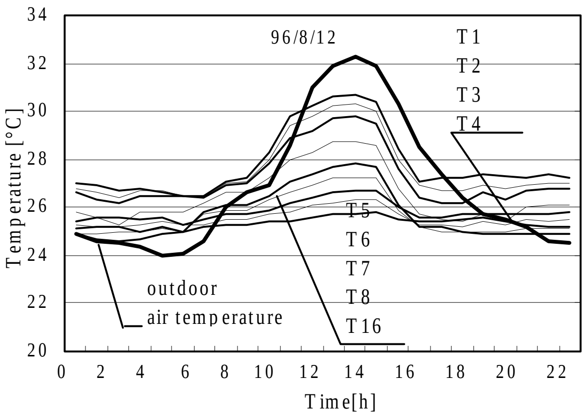


Fig. 4. The daily variations of air temperature in the A route (in summer)

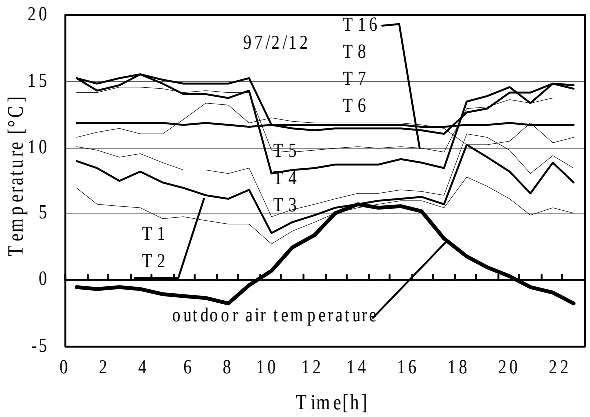


Fig. 5. The daily variations of air temperature in the A route (in winter)

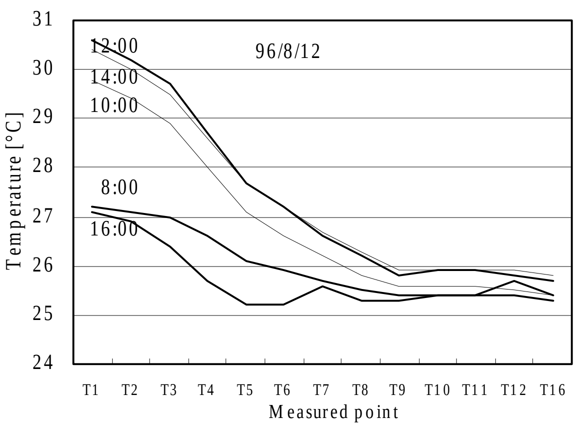


Fig. 6. The variation of air temperature at each times. (in summer)

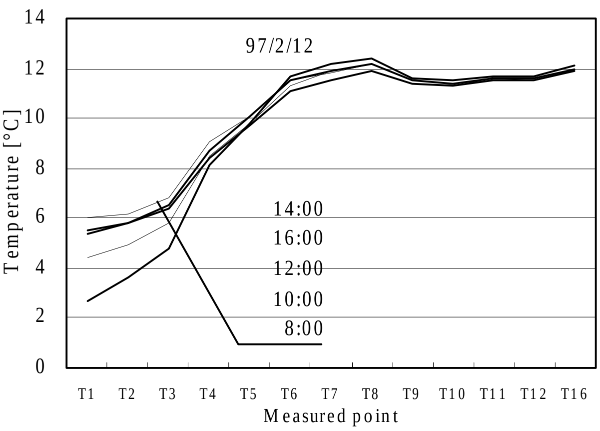


Fig. 7. The variation of air temperature at each times. (in winter)

performance caused by annual imbalance between the amount of heat extracted from its surrounding soil in winter and released to it in summer. In order to solve these problems, performances and thermal characteristics of the system were investigated for a real building. Additionally, a numerical model is developed to predict thermal characteristics and performances of the system, and the validity of model is verified by the comparison between results of calculations and measurements. Finally, the problem of the deterioration of system performance (as is mentioned above) is discussed.

BUILDING AND SYSTEM SCHEMA

Fig. 1 shows the floor plan of the building applied this system. This building is a facility for children and there is an atrium on the center of it. Strict control of the room temperature isn't required, because it is a playing hall for children. Therefore, the spot heating/cooling system is introduced.

The diagram of airflow in the underground double floor space is shown in Fig. 2. There are two inlet openings which are on the areaways (dry area) at east/west side of the building to introduce outdoor air

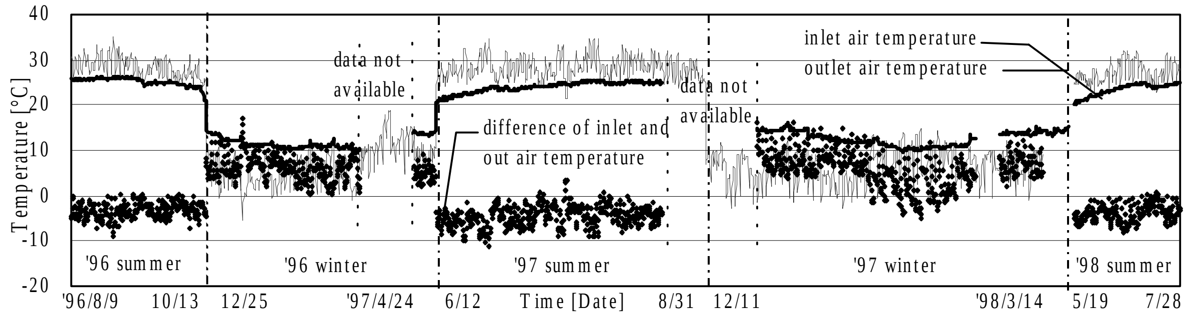


Fig. 8. The variations of inlet/outlet air temperature and its difference when supply fan is operating.

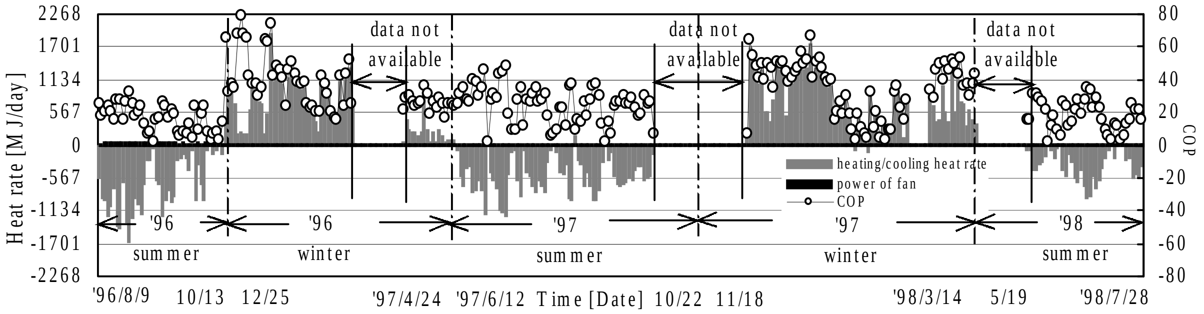


Fig.9. The variations of heating/cooling heat rate to introduced outdoor air temperature and coefficient of performance.

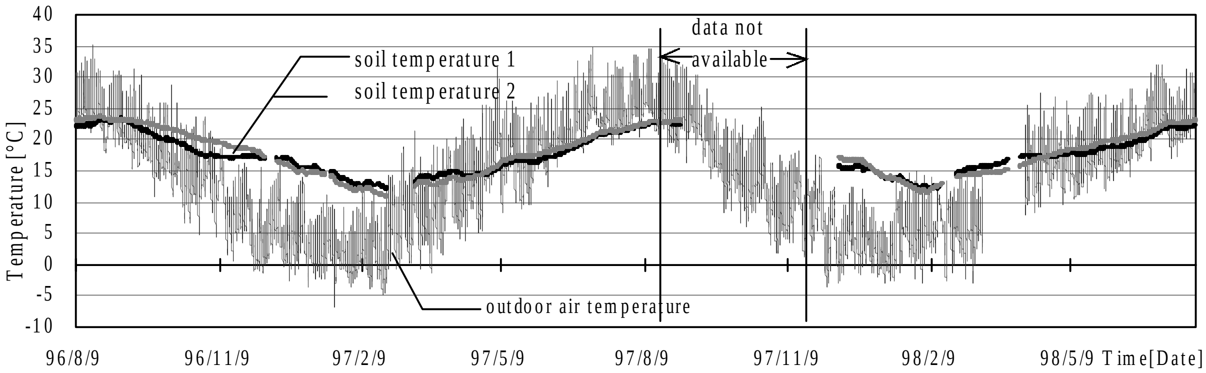


Fig. 10. The variations of outdoor air temperature and soil temperature.

On the other hand, two kinds of outlet openings are on inner wall of the atrium. One openings are small louvers under benches at the perimeter of the atrium. The other ones are normal openings on inner wall. The fresh air introduced into the under floor space is heated or cooled during passing through it and supplied to the atrium. At the location shown in Fig. 2 four fans are installed in the under floor space. They are usually run from 8:00 to 17:00 except days when the facility is closed. These fans are controlled by central monitoring system, and they are run when outdoor air temperature is higher than 25°C in summer and lower than 10°C in winter (table 1). There is no mixing of the atrium air and the under floor air when the fans are not running, because shutters at outlet openings are closed when the fans are not running. However, a little bit of fresh air enters into the under floor space by the temperature difference between inside air and outside air, and the wind force, because inlet openings on the areaway always open.

MEASUREMENT PROCEDURE

Air temperatures, inner wall surface temperatures in the under floor space and soil temperatures at the center of it (at depth of 1m) are measured as shown in Fig.2. All temperatures are measured by Coper-Constantan thermocouples at an hour interval. The

Table 2
The condition of simulation

Indoor temperature of atrium	summer : 26°C uniformity (June ~ September) winter : non-air conditioning period : 15°C air conditioning period : 18°C (December ~ March) inter-phase : linear interpolation of control room temperature of the summer / the winter season
Air velocity in double slab space	non-air conditioning period : 0.09 m/s, air conditioning period : 0.18 m/s
The corresponding length	50 m (A route)
Disturbance	outside-air temperature, long wave radiation, solar radiation
The bottom department of soil temperature	15.1°C (annual mean outside-air temperature)
heat transfer coefficient in interior	9.3 ($W/m^2 \cdot K$)
heat transfer coefficient in double slab space	α_I = 6.16 + 4.19V
Finite-difference schema	convection : the upwind schema, diffusion term : central-difference schema time term : time-forward

outdoor air temperature is also measured by the central monitoring system.

RESULT OF MEASUREMENT

The field measurement for the actual system has been started from Aug. 1996. This paper presents the measurement results over two years (Aug. 1996 - Jul. 1998).

As shown in Fig. 3, the air is supplied into the atrium through the three routes (A,B and C routes). The daily variations of air temperature in the A route, which is the longest (about 50m) among three routes, in summer (a hottest day) and winter (a coldest day) are shown in Fig. 4 and 5, respectively. From Fig. 4 and 5, temperature of the point close to the outlet (T16) is kept around 26℃ in the afternoon of the hottest day and 12℃ at the coldest day. In addition, the temperature difference between near the inlet and the outlet during out of air conditioning period in winter is lager (about 10℃) than that in summer (3℃). The main reason is that difference in temperature between outdoor air and the air through double slab space in winter is lager than that in summer.

The air temperature distribution along the route-A during air conditioned period are shown in Fig. 6 (in a summer day) and 7 (in a winter day), respectively. From Fig. 6 and 7, only a little variation of temperature of air along the double slab is observed beyond the point of around T7 all day. This result indicates that flesh air is cooled/heated sufficiently before reaching around T7. Therefore, it is concluded that the system had enough area of heat transfer surface.

The air temperature at the inlet / outlet and temperature difference between them are shown in Fig. 8. The outlet air temperature is 20℃ at the beginning of summer and it is kept about 26℃ even at the end of summer. In winter, it is 15℃ at the beginning, 10℃ at the end. In addition, mean temperature difference of the air between inlet and outlet are 5℃ in summer and 7℃ in winter, respectively. Thus, it is found that the system can supply fresh air to the room with nearly stabilized temperature all though the heating/cooling season.

Fig. 9 shows the variation of exchanged heat rate, energy consumption of fans. Also the coefficient of performance (COP= exchanged heat rate / energy consumption) over two years and the annual average COP is about 25. Therefore, this result leads to the conclusion that the system can achieve high energy-performance.

Furthermore, the variation of outdoor air temperature and soil temperature (at the depth of 1 m from the under surface of the tube) is shown in Fig. 10. Judging from the soil temperature change, it is found that the system can operate cyclically in a year.

Table 3
Thermal property used for simulation[4] [5].

	Volumetric specific heat ($J/m^3 \cdot K$)	Thermal conductivity ($W/m \cdot K$)
Concrete	1887.91	1.40
Soil	2800.47 (saturation)	2.88 (saturation)
	1753.95 (12.5vol.%)	2.17 (12.5vol.%)
Air	1.21	

NUMERICAL MODEL

The shape in the under floor space is considerably complicated. Therefore, it is simplified as the shown in Fig. 11. For the sake of simplicity we shall additionally assume that:

(1) The velocity of the air flow is constant at average measured value and it is one way without buoyant flow.

(2) The heat transfer in the soil/wall is simple heat

diffusion.

(3) The horizontal right-angled way heat flow to the air flow is negligible, so the heat flows in soil/wall are two-dimensional ways.(see Fig. 11)

Under these assumptions, the equations of the flow of heat in and surrounding the under floor is described by the following set of equations[6][7].

-The equation of energy conservation of the solid (soil and wall) is:

$$c\rho\frac{\partial\theta}{\partial t}=\lambda\left(\frac{\partial^2\theta}{\partial x^2}+\frac{\partial^2\theta}{\partial y^2}\right) \qquad (1)$$

-The equation of energy conservation of the air in the under floor space is :

$$c_a\rho_a\frac{\partial\theta_a}{\partial t}=-c_a\rho_a V\frac{\partial\theta_a}{\partial x}+\frac{\alpha_i}{D}\left(\theta_{cu}+\theta_{cd}-2\theta_a\right) \qquad (2)$$

-The heat transfer between the wall surface and the air is:

$$q=\alpha_i\left(\theta_a-\theta_w\right)=-\lambda\frac{\partial\theta_w}{\partial y} \qquad (3)$$

-The heat transfer between the ground surface and the outdoor air is:

$$q=\alpha_o\left(\theta_o-\theta_s\right)=-\lambda\frac{\partial\theta_s}{\partial y} \qquad (4)$$

CALCULATED CONDITIONS

Numerical analysis of the under floor space and its surrounding area is treated as a two-dimensional coordinate system and performed by the finite difference method.

Outdoor air temperature used in this analysis is measured value. The room temperatures above the under floor space are the seasonal average of measured value, because their variations during each seasons are very small. The velocity of air flow in the under floor space is also the average of measurements in winter. Summary of calculation is shown in Table 2.

VALIDITY OF NUMERICAL MODEL

Fig. 12 shows daily average of the temperature difference between inlet and outlet air. As in figure, calculation results almost agree with measured values through the year. The comparisons of the variations of outlet temperature in summer and winter are shown in Fig. 13 and 14, respectively. As shown in both figures, the differences between the calculations and measurements are small and the mean error is approximately 1°C. From these results, it is confirmed that the proposed calculation model is valid for predicting thermal characteristics of the system.

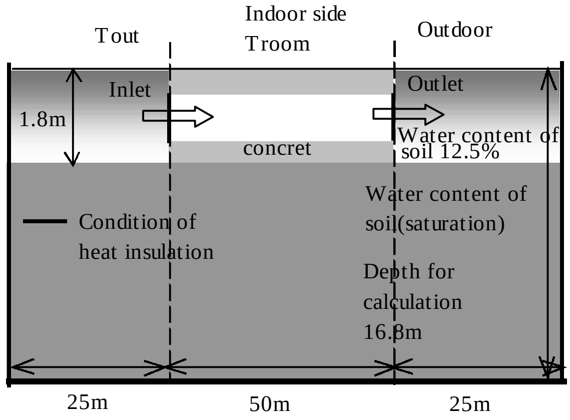


Fig. 11. The schematic of the under floor space and surrounding ground area.

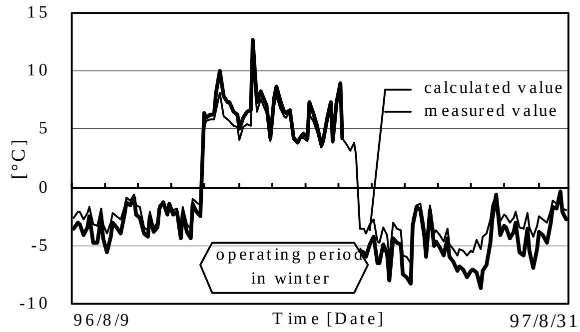


Fig. 12. Comparison of daily average of the temperature difference between inlet and outlet air.

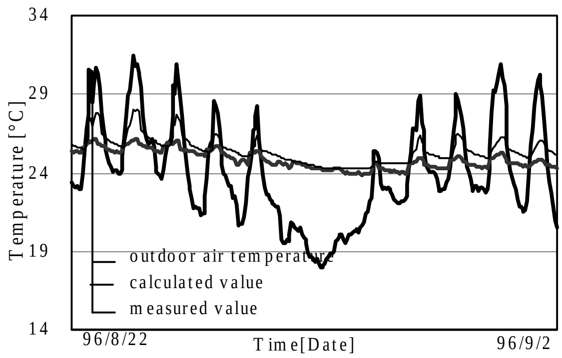


Fig. 13. Comparison of measured and calculated outlet air temperature in summer.

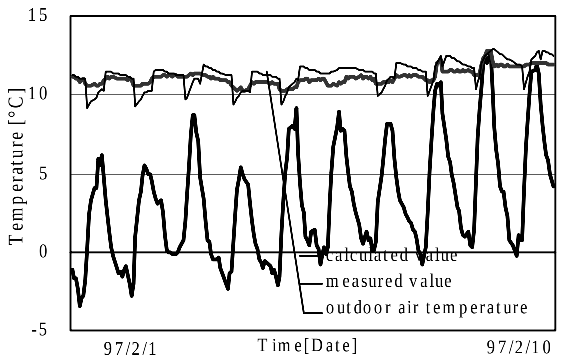


Fig. 14. Comparison of measured and calculated outlet air temperature in winter.

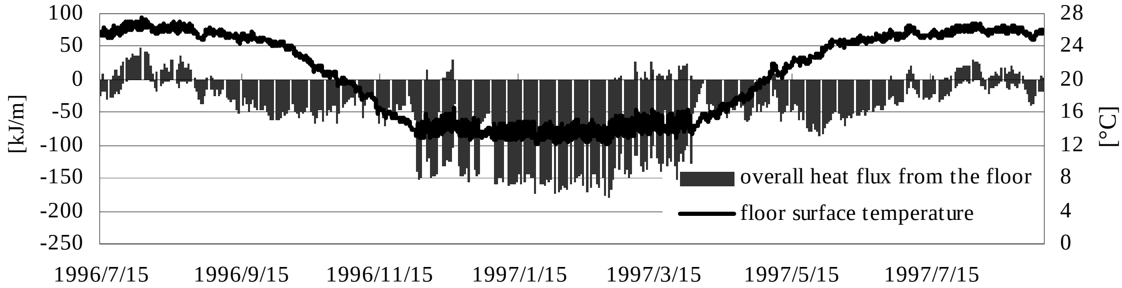


Fig. 15. The variations of floor surface temperature on the room side and overall heat flux from the floor when this system is in operation.

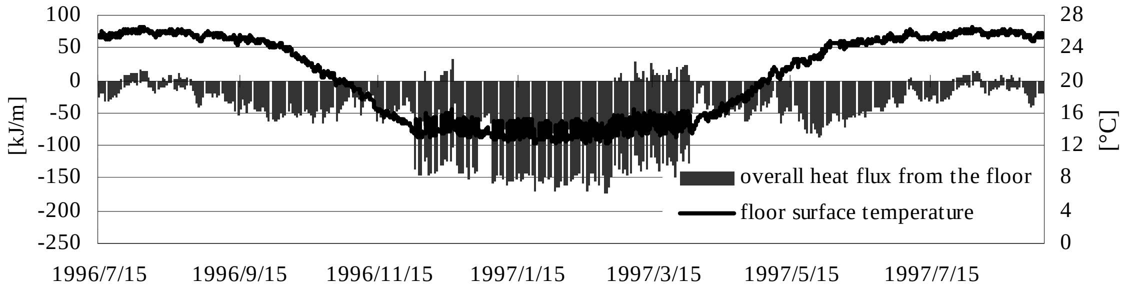


Fig. 16. The variations of floor surface temperature on the room side and overall heat flux from the floor when this system is out of operation.

EFFECT OF ENERGY SAVING

This system would include an increase in the room cooling/heating load due to the heat flux through the floor of the room. If the quantity of energy saving achieved by this system were less than the quantity of this increase in room load, adoption of this system would not make sense. Therefore by numerical calculation, this problem is examined.

The main conditions of simulation are the following:
-Real meteorological conditions of NAGOYA, JAPAN is used.
-The room is assumed to locate only above route A.

The variations of floor surface temperature on the room side and overall heat flux from the floor when this system is in and out of operation are shown in Fig. 15 and 16, respectively. From these results, the increase of room heat load due to heat flux through the floor is 420.39MJ in summer and 798.76MJ in winter. On the other hand, energy saving achieved by this system (reduction of fresh air load) is 27.9GJ in summer and 36.8GJ in winter. Accordingly, it is clear that the increase in room heat load are very small and the system coefficient of performance(SCOP= [annual reduction of fresh air load – increase in room load] / fan power) is 13.8 (see Fig. 19).

Table 4
The condition of simulation

Indoor temperature of play atrium	summer : 26°C uniformity (June ~ September) winter : non-air conditioning period : 17°C air conditioning period : 22°C (December ~ March) inter-phase : linear interpolation of control room temperature of the summer / the winter season
Air velocity in pit	non-air conditioning period : 0.09 m/s, air conditioning period : 0.18 m/s
The corresponding length	50 m (A route)
Disturbance	outside-air temperature, long wave radiation, solar radiation
The bottom department of soil temperature	15.1°C (annual mean outside-air temperature)
heat transfer coefficient in interior	9.3 ($W/m^2 \cdot K$)
heat transfer coefficient in double slab space	$\alpha_I = 6.16 + 4.19V$
Finite-difference schema	convection : the upwind schema, diffusion term : central-difference schema time term : time-forward

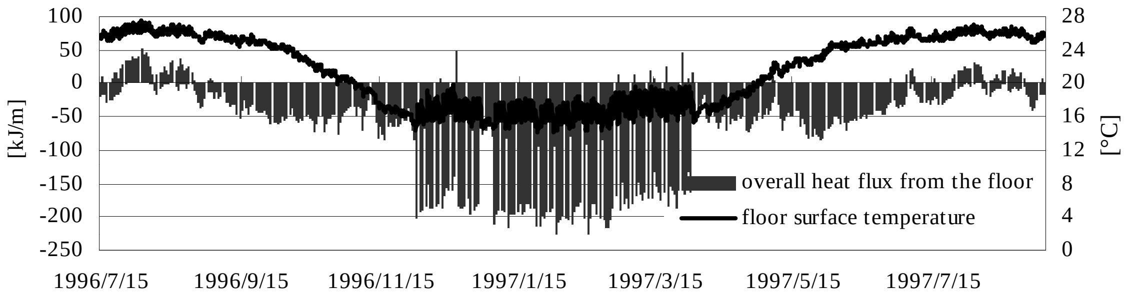


Fig. 17. The variation of floor surface temperature on the room side and overall heat flux from the floor in general building when this system is in operation.

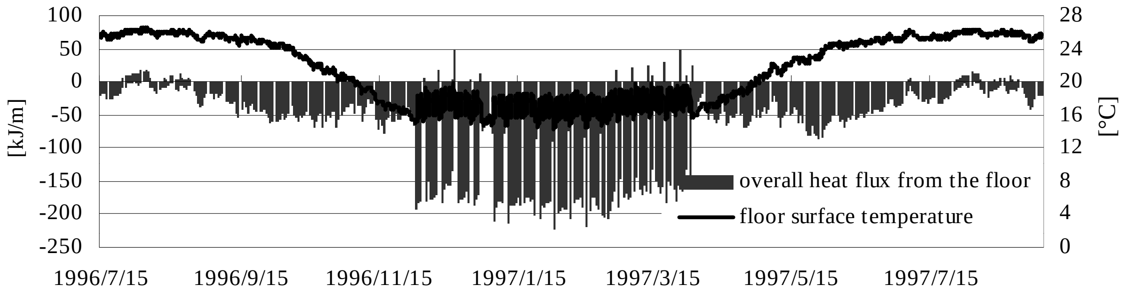


Fig. 18. The variation of floor surface temperature on the room side and overall heat flux from the floor in general building when this system is out of operation.

APPLICATION TO GENERAL BUILDING

In the previous sections, performance of the system is examined for a real building which doesn't require strict control of indoor thermal environment, especially in winter. However, in this section performance of the fresh air load reduction system applied to a general building is investigated by the same method used in the last section. The condition of simulation is shown in Table 4. The variations of floor surface temperature on the room side and overall heat flux from the floor when this system is in and out of operation are shown in Fig. 17 and 18, respectively. As a result, the increase in overall heat transfer load through floor is 990.33MJ in summer and 3963.11MJ in winter. The energy saving obtained by the fresh air reduction system is 24941.7GJ in summer and 74798.8GJ in winter. Accordingly, annual increment of floor overall heat transfer load is 4953.45MJ and it is 4.73% of energy saving (fresh air load reduction). Furthermore, SCOP is 21.2 (see Fig. 19). Therefore, it is also confirmed that this system is effective to a building which require general indoor thermal condition.

LONG TERM SYSTEM PERFORMANCE

In this section, the stability of the system during the long-term system operation, that is "the problem of the deterioration of system performance", is discussed. Table 5 summarize the calculation results of the fresh air load reduction for ten years. Outdoor condition of the first year is assumed to continue periodically for subsequent years. As shown in Table 5, the ratio cooling and heating loads in the first year is different from those of following 9 year for the both buildings. However, these ratios are almost constant since second year in spite of the difference between cooling load and heating load (the ratio of cooling/heating for real building is approximately 2/3 and approximately 2/7 for general building). Judging from these calculated results, it is confirmed that this system has a stability for long-term operation, because surrounding soil has sufficient heat capacity.

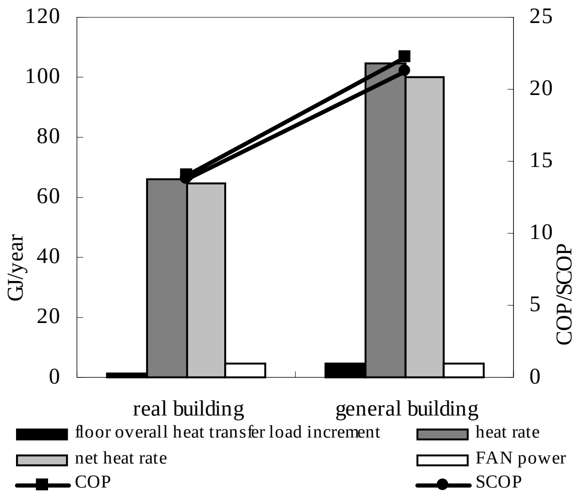


Fig. 19 The effect of system in real building and general building.

Table 5
Estimated heat rate for ten years (unit : MJ)

	real building		general building	
year	cooling heat rate	heating heat rate	cooling heat rate	heating heat rate
1	-28121.3	37351.1	-28945.38	87642.48
2	-22062.3	33803.4	-25932.03	78761.87
3	-22175.4	33710.3	-26037.41	78631.44
4	-22188.9	33675.3	-26057.42	78607.22
5	-22270.5	33657.2	-26062.23	78601.21
6	-22222.8	33642.0	-26063.26	78599.97
7	-22303.4	33634.1	-26063.48	78599.73
8	-22246.8	33632.1	-26063.52	78599.69
9	-22281.0	33631.6	-26063.53	78599.68
10	-22177.8	33631.5	-26063.53	78599.67

CONCLUSIONS

From the viewpoint of utilization of natural energy, the fresh air load reduction system using underground double floor space was investigated experimentally and numerically. This system has been introduced to a real building since July, 1996. From the measurements, it is concluded that this system is very effective to reduce fresh air load.

A simulation model used the simple heat diffusion equation was developed to examine thermal characteristics and performance of the system. The results of simulation agreed well with the measured values for thermal characteristics. From this verification, it was clarified that the proposed method can be used for the thermal design of this system.

Furthermore, the performance of it and the stability for long-term operation were investigated and it was made clear that this system is very useful and has an enough stability for long-term operation.

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NOMENCLATURE

θ	temperature [°C]
λ	thermal conductivity [W/m·K]
c	specific heat [J/m^3·K]
ρ	density [kg/m^3]
q	heat rate [W/m^2]
t	time [sec.]
V	air velocity [m/sec.]
α	heat transfer coefficient [W/m^2·K]
x , y	length, depth [m]
D	height of pit [m]

*subscript -- a : air, i : in the under floor space, o : out door air,
cu, cd : upper and lower concrete surface in the double slab space,
w : wall surface, s : ground surface