Study on Fresh Air Load Reduction System by Air-to-Earth Heat Exchange
Using Underground Double Floor Space

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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 fresh air is introduced into the double slab space and pass the opening bored through footing beam. The air is cooled by the heat exchange with inside surface of the double slab space in summer, and heated in winter. Then the heat is transferred to the adjacent soil through foundation slab and wall and to the room just above the mat through floor slab. 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. The object of this paper is examination of performance of fresh air load reduction system by using underground double floor space. Performance index are reduction rate of fresh air load, HVAC energy consumption, running cost and environmental load such as CO2 emission. In this paper, the system assumed to be applied for office building and these indexes are examined comparing with total enthalpy heat exchange system.

Keywords: fresh air load reduction system; soil capacity; underground double floor space; total enthalpy heat exchanger

1. Introduction
Currently 60% of energy consumption for commercial sector is used for air-conditioning and domestic hot water supply 1). Furthermore, energy consumed to treat fresh air load accounts for 10-30% of total energy consumption for HVAC system 2). Therefore utilization of natural energy to reduce fresh air load is useful for energy conservation.

River/sea water, ground water and earth are considered as natural heat source/sink for heating/cooling outdoor air (fresh air), because temperature is higher in winter and lower in summer than outdoor air. Among them earth has great potential since it exists everywhere and has large thermal capacity.

There are many kinds of method to utilize heat capacity of soil such as basement, water thermal storage tank under the ground, piping varied in soil, underground double slab space and cool-tube in which air or water is heat transfer medium.

In Japan various studies on this matter have been conducted. Tanaka et al. conducted numerical analysis seasonal thermal storage tank installed under ground 3). Nagai et al. performed experimental and numerical analysis on effectiveness of earth as long term thermal storage medium 4). Yosida et al. conducted field measurements of fresh air duct under the ground 5). Ishihara et al. analyzed numerically performance of cool-tube 6). Kimura et al. examined indoor air condition (temperature and humidity) by field measurements and numerical analysis 7). Hokoi et al. conducted numerical calculation for cooling effect of cool-tube by simultaneous equation of heat and mass transfer 8).

Leon Katz reported results of field measurements and numerical analysis on indoor temperature and humidity of basement and surrounding soil at Israel 9). M. Santamouris reported the results of experiments and numerical analysis of cool-tube in Ireland 10). These studied examined only amount of heating/cooling of medium for specific application or experimental setup.

The object of this paper is examination of performance of fresh air load reduction system by the air-to-earth heat exchanger using underground double floor space. On this system, the fresh air is introduced into the double slab space and cooled/heated by the heat exchange with inside surface of double slab space. Performance index are reduction rate of fresh air load, HVAC energy consumption, running cost and environmental load such as CO2 emission. In this paper, the system assumed to be applied for office building and these indexes are examined compare with traditional total enthalpy heat exchange system using numerical analysis.
2. Outline of Earth Tube System

2.1 Concept of earth tube system

On the earth tube system, the fresh air for air conditioning is introduced into the double slab space and passed the opening bored through footing beam. The air is cooled by the heat exchange with inside surface of the double slab space in summer and heated in winter, because of the effect of heat exchange based on the temperature difference between outdoor air and soil under the building basement (see Fig.1).

Initial cost of the system can be reduced, because it doesn’t need pipes or ducts in the ground as does the cool-tube system. However, geometry condition is restricted according to use and height of building since this space is designed for earthquake resistant. Therefore careful consideration should be given for designing of this system.

2.2 Composition and operation of system

Figure 2 shows the system examined in this paper. In this system single duct VAV is adopted and electric driven turbo chiller and gas-fired boiler are installed. Figure 3 shows diagram of heat source system and Table 1 and Figure 4 shows specification and characteristics of equipments.

The system is operated by the following two modes (Earth tube mode and Direct mode). When temperature of outlet air of earth tube is higher/lower than outdoor air for heating/cooling season, fresh air is introduce to air handling unit through earth tube (Earth tube mode). In opposite case, fresh air introduced to air handling unit directly (Direct mode).

3. Simulation Composition and Models

System simulation program has been developed to estimate heat transfer properties of earth tube system and system energy performance. The program consists of three analysis blocks, the air-to-earth heat exchanger block to estimate outlet temperature of earth tube, the HVAC block to calculate energy consumptions of system elements and the evaluation block.

3.1 Earth tube model

The performance of earth tube is calculated following simple 2-dimensional thermal diffusion model. Validity of this model is warranted by comparison with field measurements 11).

The equations of the flow of heat in and surrounding the under floor is described by the following set of equations 12).

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

\[ c_p \frac{\partial \theta}{\partial t} = \lambda \left( \frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} \right) \]  \hspace{1cm} (1)

- The equation of energy conservation of the air in the under floor space is:
The heat transfer between the wall surface and the air is:
\[ q = \alpha_w (\theta_w - \theta_a) = -\lambda \frac{\partial \theta_w}{\partial y} \]  
(3)

- The heat transfer between the ground surface and the outdoor air is:
\[ q = \alpha_{so} (\theta_{so} - \theta_a) = -\lambda \frac{\partial \theta_{so}}{\partial y} \]  
(4)

3.2 Total enthalpy exchanger model

Characteristics of total enthalpy heat exchanger are modeled by using performance table of manufacture and are expressed as following equations.
- Enthalpy of outdoor air \( (E_{out}) \) is smaller than enthalpy of return air \( (E_{rtn}) \) in heating:
\[ E_{ex} = E_{out} + \frac{\eta \times (E_{rtn} - E_{out})}{100} \]  
(5)

- Enthalpy of outdoor air \( (E_{out}) \) is bigger than enthalpy of return air \( (E_{rtn}) \) in cooling:
\[ E_{ex} = E_{out} - \frac{\eta \times (E_{out} - E_{rtn})}{100} \]  
(6)

ON/OFF control based on enthalpy is adopted for the control of total enthalpy exchanger. When enthalpy of outdoor air is smaller than that of return air in cooling, total enthalpy exchanger will be turned off and turned on in opposite case. Table 2 shows performance of total enthalpy exchanger and its consumption of electric power.

3.3 Electric turbo chiller/gas fired boiler

Performances of electric turbo chiller and gas-fired boiler are defined on the basis of characteristic equations of the air conditioning system simulation program (HASP/ACSS/8502).

It is assumed that the rated COP of electric turbo chiller is 3 and efficiency of gas-fired boiler is 0.8 (Input fuel is 13A: total heat generation rate is 12.8kW/Nm³).

The multiple units and throttling control are applied for the control of heat source system depended on building hourly heat load.

3.4 Fan/Pump

Secondary water circulation pumps apply variable water volume by the two-way valve control. Air VAV handlers are installed for the airside HVAC system, these fans controlled using inverter motor. Pumps and fans of cooling tower are controlled constant during the operation. Characteristics of fan and pump gave

| Table 1. The Scheme of Heat Source and Transfer System |
|------------------------------------------------------|
| Turbo Refrigeration: 1 capacity 116.3 kW(3), 232.6 kW (1) 2 input 38.8kW(3), 77.6kW(1) 3 COP 3  |
| Boiler: 1 capacity 151.2kW(3), 302.4kW(1) 2 input of gas 14.8m³/h(3), 29.6 m³/h (1) 3 input of electric power 0.75kW(3), 1.5kW(1) 4 efficiency 0.8  |
| Air conditioner: 1 air flow rate supply fan 20500 m³/h(1), 30000 m³/h(3), 39000 m³/h (1) 2 air flow rate return fan 1300 m³/h, 2600 m³/h, 3900 m³/h 3 input of supply fan 11kW(1), 15kW(4) input of supply fan 1.5kW, 2.2kW, 3.5kW  |
| Cooling tower: input 5kW  |
| Hot water pump: 1 flow rate 24 m³/h (3), 48 m³/h (1) 2 input 3.7kW(1), 5.5kW(3)  |
| Chilled water pump: 1 flow rate 20 m³/h (3), 40 m³/h (1) 2 input 3.7kW(1), 5.5kW(1)  |
| Cooling water pump: 1 flow rate 100m/h 2 input 5.5kW  |

*The number in a parenthesis shows the number of unit.

| Table 2. The Performance of Total Enthalpy Heat Exchanger |
|----------------------------------------------------------|
| Air flow rate [m³/h] | 1300 | 2600 | 3900 |
| Temperature exchange efficiency | 75% | 73% | 72% |
| Enthalpy exchange efficiency | 66% | 64% | 62% |
| Cooling | | | |
| Heating | | | |
| Input [kW] | 0.86 | 1.7 | 2.4 |

Fig.4. Characteristics of Heat Source
characteristics equation made by HASP/ACSS like heat source. Secondary pump use both throttling control and multiple units operation. The other side, the increment of fan caused to power friction loss and pressure loss by using earth tube system is calculated, few fan power increase. Pressure drop and thermal loss on ducts and piping are not considered.

4. Simulation Summary
4.1 Building condition
Model office building proposed in the dynamic thermal load calculation program HASP/ACLD/8001 is chosen for the system simulation. The typical floor plan of the object building is shown in Figure 5 and specifications of the building are shown in Table 3. It is supposed the building sites at Nagoya in Japan.

The building has five stories and 5000m² total floor area. It has 650m² air conditioning area per floor. 130 persons occupy the air conditioning area per floor under the assumption of the occupancy density is 5m² per person.

4.2 Description of compared systems
In order to examine the performances of earth tube system as fresh air load reduction system, three systems are supposed as follows.

① Air conditioning system without fresh air load reduction system (BAS).
② Air conditioning system with earth tube system as fresh air load reduction system (ETS).
③ Air conditioning system with total enthalpy heat exchanger as fresh air load reduction system (TES).

4.3 Simulation summary and condition
In this examination, length and section of double slab space assumed to be 50m and 7m x 1.4m based on design of general application. Three different cases of amount of the fresh air supply are given as 10m³/h, 20 m³/h and 30 m³/h. Simulation conditions and its number are shown Table 4.

Simulations of room load and room temperature are performed using the air conditioning system simulation program HASP/ACLD/8501 and HASP/ACSS/8502.

Air conditioning is performed winter (December ~ March) and summer (June ~ September), intermediate season (April ~ May, October ~ November) applies no air conditioning and only operates the ventilation system. And air conditioning operates on a weekday (08:00 to 18:00), and Saturday (08:00 to 15:00). Indoor temperature settings are 26°C in cooling season, 22°C in heating season and relative humidity setting is 50% in either season. The weather data used for the simulation is Nagoya standard weather data supported in the HASP.

Table 3.
The Condition of Simulation

| Total floor area | 5000m²   |
|------------------|----------|
| Typical floor area | 1000m² per floor |
| Air conditioned floor area | 650m² per floor |
| Floor height | 4.4m  |
| Ceiling height | 3.0m |
| Outside wall area | 285m² per floor |
| Inside wall area | 165m² per floor |
| Window area | 135m² per floor |
| Outside wall thermal insulation | 3.5kW/m²°C |
| Roof thermal insulation | 1.8kW/m²°C |
| Window | 6.4 kW/m²°C |
| A resident area per person | 0.2m² per person |
| Air requirement for air ventilation | 10, 20, 30m³/h per person |
| Room temperature and humidity | 26°C, 50% in summer, 22°C, 50% in winter |

Table 4.
The Compared System

| Air requirement for air ventilation | CASE | Fresh air load Reduction system |
|------------------------------------|------|---------------------------------|
| 10m³/h per person | BAS10 | - |
| Air flow: 6500 m³/h | ETS10 | ETS (0.15m³/s) |
| 20m³/h per person | BAS20 | - |
| Air flow: 13000 m³/h | ETS20 | ETS (0.35m³/s) |
| 30m³/h per person | BAS30 | - |
| Air flow: 19500 m³/h | ETS30 | ETS (0.55m³/s) |

*The number in a parenthesis shows air velocity of ETS or efficiency of total enthalpy heat exchanger.
5. Heating and Cooling Load

Figure 6 shows monthly room load. Annual heating load per air conditioning area is about 237.3 MJ/year/m² and 93.7 MJ/year/m², respectively. Fresh air load rate to the room load becomes 10~30% on the base case.

Figure 7 shows annual fresh air load in each case. In BAS case, the annual fresh air load is 33.6~100.8 MJ/m² for each air requirement for air ventilation. The system which is the largest reduction rate of fresh air load is TES system, it can reduce about 72~75%.

The fresh air reduction rate of ETS system is 36~62%. In this system, the different of air volume greatly influence on reduction rate of fresh air load. The reduction rate of fresh air load of TES10 system was the largest.

Figure 8 shows calculation results for outlet air temperature through the earth tube, room and outdoor air temperature. Average temperature difference between room and outdoor air becomes 2.3°C in summer and 14.2°C in winter. And then, average temperature difference between outlet air of earth tube and outdoor air are 4°C in summer and 6°C in winter, respectively.

6. Energy Performance

6.1 Performance index

The energy performance of the whole system was evaluated using SCOP defined as the following equation. The conversion factor of electric power and gas into primary energy is shown in Table 5.

\[
\text{SCOP} = \frac{\text{Air conditioning load (room load + fresh air load)}}{\text{Energy consumption for air conditioning}}
\]

6.2 Energy performance

Figure 9 shows air conditioning load, annual energy consumption and SCOP. Table 6 shows annual consumption of air conditioner, heat source and conveyer machine, and total energy performance.

Annual energy consumption per total floor area was about 116 kW/m². For TES30 system, reduction rate that is annual primary energy consumption of TES30 and ETS30 is about 6.69% and 5.54% respectively.

In no cases, the big difference was seen for electric power consumption. It is because the curtailment effect of the electric power consumption by the fresh air load reduction system didn’t appear remarkably since the total electric power consumption used for air conditioning is large. The TES system had the largest power consumption. In this case, although the electric power consumption of a heat source and conveyer machine became less than other systems by reduction of fresh air load, the electric power consumption of total enthalpy heat exchanger became large. Consequently, it is because total electric power consumption became larger than other systems.

![Fig. 6. The Room Load of Object Building](image)

![Fig. 7. The Comparison of Annual Fresh Air Load](image)

![Fig. 8. The Comparison of Outlet Temperature TES (10, 20, 30) and ETS (10, 20, 30)](image)

![Fig. 9. The Comparison of Annual Fresh Air Load](image)

Table 5. The Conversion Factor of Electric Power into the Primary Energy

| Energy Source | Conversion Factor |
|---------------|-------------------|
| Electric power| 2.85kW/kWh        |
| Gas           | 12.8 kW/m² (13A)  |
On the other hand, the BAS system had the largest gas consumption. Air conditioning system with ETS or TES can reduce gas consumption of heat source by reducing fresh air load. BAS system consumed more gas 0.46~1Nm³/m² in comparison with other system.

The primary energy consumption summed electric power and gas consumption is reduced 8kW/m² in ETS system case, and reduced 11kW/m² in TES system case, in comparison with BAS system.

Compared with reduction of the heat source energy consumption by reducing fresh air load, the difference of SCOP didn’t appeared remarkably, since the energy consumption of an air conditioner and a conveyer machine was large. However, SCOP of ETS system is the highest, except the case of ETS30 system that fresh air volume is large.

### Table 6.
**Results of Simulation**

| ARV (m³/h) | CASE  | FAN (MW) | TEH (MW) | CHP (MW) | HTP (MW) | B (MW) | RT (MW) | CT (MW) | CLP (MW) | E (Km³/h) | G (MW) | P (MW) | SCOP       |
|------------|-------|----------|----------|----------|----------|--------|---------|---------|----------|-----------|--------|--------|------------|
| 10         | BAS10 | 36.3     | 4.4      | 2.0      | 0.29     | 71.4   | 5.89    | 8.84    | 367.98   | 14.08     | 548.14 | 0.5969 |
|            | ETS10 | 36.3     | 4.3      | 1.7      | 0.25     | 70.2   | 5.89    | 8.84    | 363.28   | 12.91     | 528.39 | 0.6193 |
|            | TES10 | 36.3     | 7.0      | 4.0      | 1.6      | 66.5   | 5.89    | 8.84    | 371.62   | 12.59     | 532.68 | 0.6143 |
| 20         | BAS20 | 43.7     | 5.0      | 2.6      | 0.36     | 77.9   | 5.47    | 8.20    | 407.88   | 16.56     | 619.76 | 0.5754 |
|            | ETS20 | 43.7     | 4.8      | 2.2     | 0.31     | 75.5   | 5.46    | 8.18    | 399.35   | 14.71     | 587.64 | 0.6071 |
|            | TES20 | 43.7     | 14.5     | 4.2      | 1.8      | 68.4   | 5.47    | 8.21    | 417.74   | 13.32     | 588.16 | 0.6064 |
| 30         | BAS30 | 57.6     | 5.6      | 3.2      | 0.44     | 84.7   | 5.28    | 7.91    | 469.45   | 19.40     | 717.57 | 0.5380 |
|            | ETS30 | 57.6     | 5.4      | 2.7      | 0.38     | 82.0   | 5.25    | 7.88    | 459.33   | 17.08     | 677.80 | 0.5696 |
|            | TES30 | 57.6     | 20.5     | 4.5      | 2.1      | 71.4   | 5.29    | 7.94    | 483.22   | 14.40     | 667.37 | 0.5785 |

**Notes:**
- FAN: Annual electric power consumption of fan for air conditioner.
- TEH: Annual electric power consumption of total enthalpy heat exchanger.
- CHP: Annual electric power consumption of chilled water pump.
- HTP: Annual electric power consumption of hot water pump.
- B: Annual electric power consumption of boiler.

### Table 7.
**The Conversion Factor of CO2, NOx, SOx Emission**

| Type                  | Electric Power | Gas  |
|-----------------------|----------------|------|
| Unit [**]             | kWh            | MJ   |
| CO2 emission [kg-C**] | 0.143          | 0.0186 |
| NOx emission [g-NOx**]| 0.231          | 0.026 |
| SOx emission [g-SOx**]| 0.209          | -    |

### 7. Emission of Environmental Pollution Gas

Based on the amount of energy consumption obtained by the simulation, the amount of carbon dioxide emission (CO2), nitrogen oxide emission (NOx), and SOx (SOx)
emission was calculated. The equivalent value used for calculation is shown in table 7. Fig 11 shows CO2, NOx and SOx emission of all system in operation.

For environmental load, difference between BAS system and TES system isn’t observed. The tendency of environmental load differs from primary energy. CO2, NOx and SOx emission of TES system is the largest in comparison with other system, because the electric power consumption of TES system is the largest.

The influence which uses natural energy for reduction of fresh air load had appeared on environmental load.

8. Evaluation of the Operation Cost

Based on the consumption of electric power and gas, energy charge is calculated. According to maximum operating condition, demand charge is decided. The value used for calculation is shown table 8.

The running cost of all systems is shown in Table 9. When a fresh air requirement volume is 10m³/h and 20m³/h per a person, running costs of ETS system is lowest. It can reduce 16 and 26JPY per floor area in compared with BAS for a fresh air requirement volume of 10m³/h and 20m³/h.

When a fresh air requirement volume is 30m³/h per a person, running costs of TES system is lower than those of other systems, since the largest rate of primary energy was reduced. Curtailment of an about 36 JPY can be performed, in compared with a BAS system. In this case, ETS system can achieve curtailment of about 32JPY per floor area, in compared with a BAS system.

9. Conclusion

The fresh air load reduction using the underground double floor space system (earth tube system) was proposed, and the energy performance, environmental load, and the running cost were compared with the conventional system (BAS system) and the fresh air load reduction system using total enthalpy heat exchanger(ETS system).

Examination showed that to a BAS system, a proposed system can reduce the fresh air load of 36-62% annually, and can save primary energy consumption of about 40MW per year, CO2, NOx and SOx emissions in operation can be reduced about 1.2-2.3%, and an about 80000-120000 JPY of running cost can be saved.

In this paper length of earth tube is fixed to 50m, while the efficiency of total enthalpy heat exchanger is assumed to be optimal. The length of 50m is not optimal design especially for large volume of fresh air requirement. However under this condition, ETS system can reduce environmental load in every case.

Therefore, it can be concluded that ETS system is useful and viable system for reduction of fresh air load.
\[ x, y \quad \text{length, depth [m]} \]
\[ D \quad \text{height of pit [m]} \]
\[ E \quad \text{Enthalpy} \]
\[ \eta \quad \text{Efficiency of total enthalpy heat exchanger} \]

*subscript

\[ a \quad : \text{air} \]
\[ i \quad : \text{in the under floor space} \]
\[ \text{out} \quad : \text{outdoor air} \]
\[ \text{cu} \quad : \text{upper concrete surface in the double slab space} \]
\[ \text{cd} \quad : \text{lower concrete surface in the double slab space} \]
\[ w \quad : \text{wall surface} \]
\[ s \quad : \text{ground surface} \]
\[ \text{ex} \quad : \text{outlet} \]
\[ \text{rtn} \quad : \text{return air} \]

References
1) Division of general policy planning, Agency of natural resources and energy, 2002, About the energy supply-and-demand actual result in the 2000 fiscal year, 7.
2) Nakahara, N. (1983) The energy conservation of building/architecture equipment, 54-55.
3) Tanaka, H. and Okumiya, M. (1997) Energy performance of annual cycle energy system with seasonal water thermal storage. Journal of Architecture, Planning and Environmental Engineering, No. 500, 59-64.
4) Matsumoto, M. Nagai, H. and Ushio, T. (1995) Numerical analyses of thermal behavior in and around the thermal well. Journal of Architecture, Planning and Environmental Engineering, No. 470, 37-44.
5) Yoshida Toru et al. (1944) Field measurement of thermal the ground using underground concrete duct, Technical paper of 1994 SHASE annual technical meeting, SHASE, B-30.
6) Ishihara Osamu et al. (1995) Analysis of cooling ability of cooling tubes by experiments and simulations. Journal of Architecture, Planning and Environmental Engineering No.477, 11-18.
7) Kimura, K., K. Maeda and H. Onojima (1983) Study on the passive cooling effects of cool tube. Summaries of technical paper of annual meeting, Architectural institute of Japan, 675-678.
8) Ueda, S. and S. Hokoi. (1997) Cooling effect of cool tube -in the case of simultaneous heat and moisture transfer in the ground- Summaries of technical paper of annual meeting, Architectural institute of Japan, 411-412.
9) LEON KATZ and BARUCH GIVONI (1983) Earth Temperatures and Underground Building, Energy and Buildings, 922-932.
10) G. Mihalakakou, J.O. Lewis, M. Santamouris (1996) On the heating potential of buried pipes techniques - application in Ireland, Energy and Buildings, 19-25.
11) Son Wontug et al. (1999) An analysis of the fresh air load reduction system by using underground double floor space for air conditioning. Building and simulation in Kyoto
12) Hokoi, S, Uwda, S. and Yoshida, T. (1995) Study on the cooling effect of cool tube, Summaries of technical paper of annual meeting, Architectural institute of Japan, 495-496.