DYNAMIC INSULATION APPLIED TO THE RESIDENTIAL BUILDING (PART 2)
Numerical Evaluation of Thermal Insulation Effect on Air Supply Window System

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ABSTRACT

We designed a new air supply window system to ventilate indoors through the air space of a double pane window and evaluated its insulation efficiency and probability of moisture condensation in order to confirm its feasibility and applicability. Then, to verify its thermal insulation efficiency, we evaluated its temperature contribution with different air space widths using computational fluid dynamics. In addition, to verify the probability of moisture condensation, we calculated the surface temperature of the windowpane with different outdoor temperatures based on fixed indoor conditions such as air temperature and humidity ratio. The calculated results show that the heat loss/gain in the room was reduced with increasing air space width of a double pane window. Moisture condensation on the surface of an indoor windowpane depends on the outdoor temperature and it occurred when the outdoor temperature was 0 ºC in all calculated cases of the proposed system. Moreover, low-temperature outdoor air comes into the indoors through the air space of the double pane window, the relationship of the temperature difference between the indoor air/outdoor air and supply air/outdoor air was calculated as a linear approximation.

INTRODUCTION

It is essential to reduce the inordinate amount of energy used for climate control in buildings. To insulate buildings more efficiently, many insulation methods have been proposed and successfully applied to the building envelope. One technical solution involves dynamic insulation, which blocks heat transport by making the incoming airflow pass through a porous material. As dynamic insulation not only reduces heat loss but also helps maintain indoor air quality, several different dynamic insulation systems have been proposed to improve the performance of specific building elements and these systems have been successfully applied to the building envelope. Furthermore, as the Building Standard Law of Japan has been in force since July 1st, 2003 in Japan requiring minimum indoor ventilation rates of 0.5 air changes per hour for the entire 24 hours, many studies have shown that it is possible to use dynamic insulation efficiently at the building envelope in dwelling houses.

In spite of their efforts, heat loss remains through windows because they have relatively poor insulating qualities and usually contribute the greatest heat loss by heat conduction. Moreover, they also have a high risk of moisture condensation occurring because they have a lower surface temperature than the rest of the building envelope. Although many studies have shown that it is possible to use low-emissivity glazing, gas-filled glazing, or vacuum glazing to solve this problem, these solutions are expensive. This paper presents a new insulation system to insulate the glass of windows efficiently by using a supply airflow window system.

OBJECTIVE

This paper proposes a new insulation system to insulate windows efficiently in residential buildings, because they usually exhibit the greatest heat loss. We evaluated the thermal insulation efficiency of the insulation material in the proposed insulation system in order to confirm its feasibility. We also evaluated whether it produces excessive moisture condensation depending on the outdoor temperature.

PROPOSED NEW SYSTEM

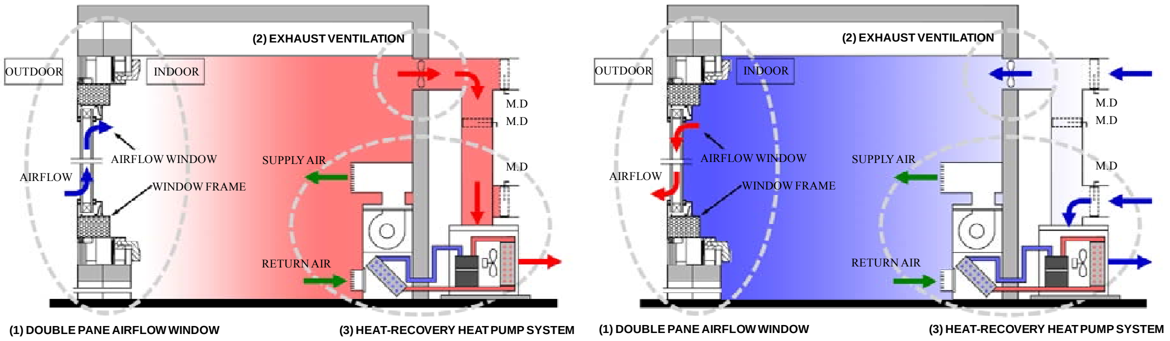
Figure 1 shows the concept of the proposed system to increase the thermal insulation and air-tightness of glass windows in residential buildings. This system is composed of three parts: an airflow window system applied to glass, a mechanical ventilation system, and a heat-recovery heat pump system.

1) Airflow window system applied to glass

Although the existing airflow window system has been used to eliminate heat from sunlight in summer, this system was designed to ventilate indoors through the air space of double pane windows to reduce the heat loss of the conduction heat transfer from glass of windows in winter. It is useful to reduce the energy transfer of indoor/outdoor because it has a minimal temperature difference of the outdoor and air space between the glass panels of the window.

2) Mechanical ventilation system

A mechanical ventilation system is used to maintain an adequate supply of fresh air and to maintain the thermal insulation efficiency. This means that indoor air leaves the room through the ventilation system,



(a) operation mode in winter *(b) operation mode in summer*

Figure 1 Composition of new proposal system

while fresh air enters the room through the air space of the double pane window. The mechanical ventilation system works as exhaust ventilation functionally in winter to prevent moisture condensation by entering low humidity air from outdoors at the building envelope as shown in Fig. 1(a). On the contrary, it works as supply ventilation functionally in summer to prevent high humidity air entering at the building envelope as shown in Fig. 1(b).

3) Heat-recovery heat pump system

This system combines the exhaust mechanical ventilation controlling negative indoor pressure to solve sick house syndrome. In addition, it has expected the heat gain effect by exhausting evaporated lower air temperature than the supply air temperature adopting a heat-recovery heat pump system.

PRELIMINARY SIMULATION

Computational fluid dynamics (CFD) has been widely used to simulate air movement, heat transfer, mass transfer, and the interaction between indoor and outdoor environments. In this preliminary simulation, CFD was used to model fluid, heat flow, and condensation to validate the theoretical results of glazing.

1) Thermal transmittance of double glazing

An analytical solution to heat transfer through a multiple glazing system can be derived from double glazing. Figure 2 shows the heat transfer through double glazing under winter conditions. The thermal transmittance of double glazing is given by Eqn. [1].

$$U = \frac{1}{\frac{1}{h_e} + R_s + \frac{1}{h_i}} \qquad \ldots[1]$$

where U is the thermal transmittance of double glazing [W/(m^2·K)], h_e is the external heat transfer coefficient [W/(m^2·K)], h_i is the internal heat transfer coefficient [W/(m^2·K)], R_s is the thermal resistance of glass [m^2·K/W].

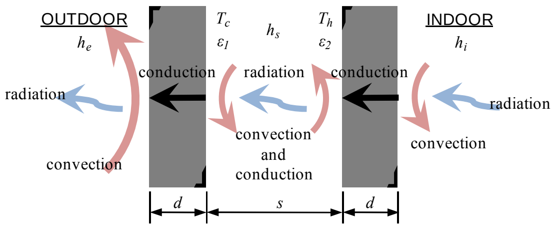


Figure 2 Schematic of heat transfer through double glazing unit in winter

The thermal resistance of multiple glazing is given by Eqn. [2].

$$R_s = \sum^{N} \frac{1}{h_s} + \sum^{M} \frac{d_m}{\lambda_m} \qquad \ldots[2]$$

where h_s is the overall heat transfer coefficient between an air space [W/(m^2·K)], N is the number of air space, M is the number of glass, d_m is the thickness of glass [m], λ_m is the thermal conductivity of glass [W/(m·K)].

The overall heat transfer coefficient between an air space is given by Eqn. [3].

$$h_s = h_c + h_r \qquad \ldots[3]$$

where h_c is the convective heat transfer coefficient between air [W/(m^2·K)], h_r is the radiative heat transfer coefficient between air [W/(m^2·K)].

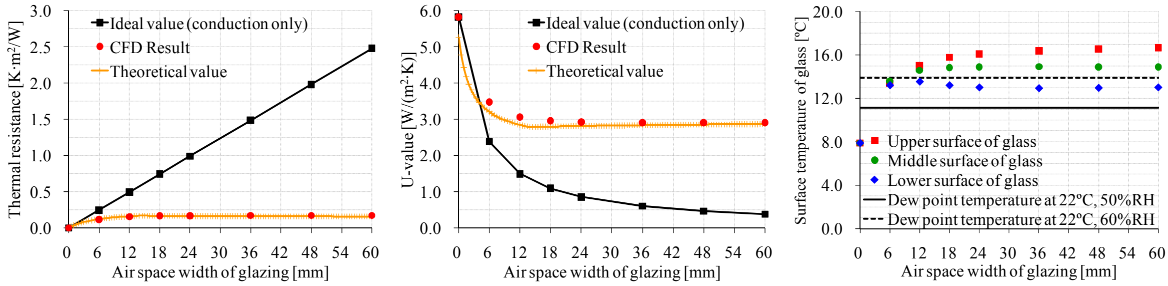
The radiative heat transfer coefficient for a tall continuous vertical air space is given by Eqn. [4]

$$h_r = \frac{4\sigma T_m^{\ 3}}{\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2} + \frac{1}{F_{12}}} \qquad \ldots[4]$$

where σ is Stefan-Boltsmann constant (5.67 × 10^{-8} W/(m^2·K^4)), ε_1 and ε_2 are the corrected emissivities of two panes of glass at mean temperature T_m [K], F_{12} is the view factor between two panes of glass and can be calculated.

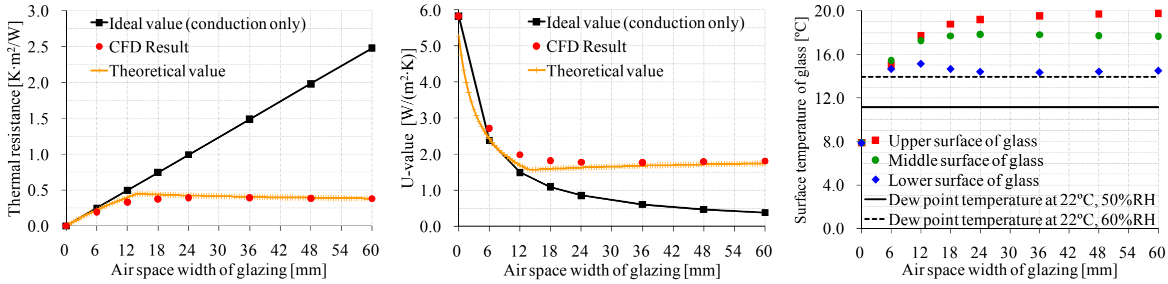
The convective heat transfer coefficient for a vertical air space is given by Eqn. [5]

$$h_c = Nu \frac{\lambda_a}{s} \qquad \ldots[5]$$



(a) Thermal resistance *(b) Thermal transmittance* *(c) Surface temperature of glass*

Figure 3 Validation of CFD results and theoretical results in uncoated glazing



(a) Thermal resistance *(b) Thermal transmittance* *(c) Surface temperature of glass*

Figure 4 Validation of CFD results and theoretical results of in Low-E film coated glazing

where Nu is the average Nussent number, λ_a is the thermal conductivity of air [W/(m·K)] and s is the mean width of air space [m].

For experimental determination of the thermal transmittance of vertical glazing the following empirical equation has been used to calculate the Nussent number.

$$Nu = 0.035 \cdot (Gr \cdot \mathrm{Pr})^{0.38} \qquad \ldots[6]$$

where Gr is the Grashof number, which is defined as Eqn. [7], Pr is the Prandtl number, which is defined in Eqn. [8].

$$Gr = \frac{9.81 \cdot s^3 \cdot \Delta T \cdot \rho^2}{T_m \cdot \mu^2} \qquad \ldots[7]$$

$$\mathrm{Pr} = \frac{\mu \cdot c_p}{\lambda_a} \qquad \ldots[8]$$

where ΔT is the temperature difference of two panes of glass [K], ρ is the density of air [kg/m^3], μ is the viscosity of air [kg/(m·s)], c_p is the heat capacity of air [J/(kg·K)].

2) Assumptions

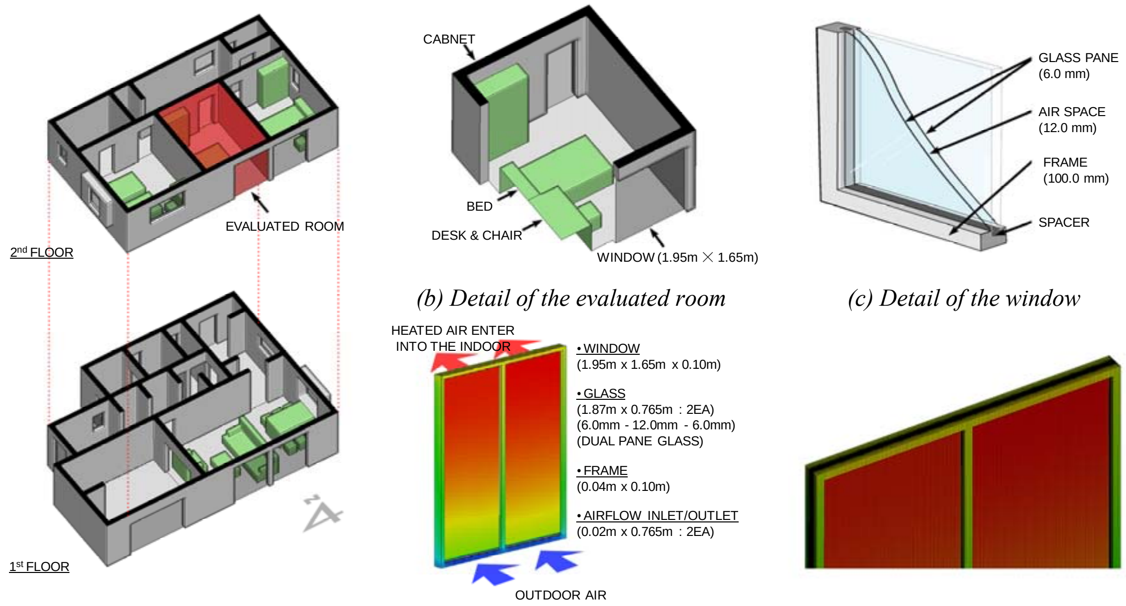
To evaluate the thermal insulation efficiency of the glazing unit, temperature contributions were calculated using 3-D steady-state CFD simulation including detailed ray-tracing-based radiation modelling as well as direct modelling of convective heat transfer under different air space widths of a glazing unit. A low-Reynolds number k-epsilon turbulence model was used to compute the turbulent viscosity and diffusivity.

The boundary conditions and material properties for the simulation are summarized in Table 1 and Table 2, respectively. The numerical method was demonstrated for a 0.765 × 1.870 × (0.006 + air space + 0.006) m^3 double glazing unit. The glazing was assumed to be made of uncoated glass with an emissivity of 0.90 and filled with air. The thermal boundary conditions were the internal surface temperatures for the unit T_h=22.0 ºC and T_c=0.0 ºC, so the mean temperature difference between the surface ΔT=22.0 ºC for the thermal performance of glazing in winter. For calculating the thermal transmittance under winter conditions, the external heat transfer coefficient for normal conditions of exposure was taken to be 23.26 W/(m^2·K) and the internal heat transfer coefficient was 9.09 W/(m^2·K). The edges of the glazing were insulated.

Consequently, the resulting thermal transmittance would be comparable with data for the central area of the glazing excluding the effects of the edges and frame. In addition, it was assumed that the effect of temperature on the air space dimensions of the glazing unit would be negligible.

Table 1 Boundary conditions

Item	Conditions
Calculation target	Double pane window (Winter operation mode)
Calculation room	0.765 m × 1.870 m × (0.006+air space+0.006) m
Turbulence model	Abe-Kondoh-Nagano Low Re k-ε model
Calculation Mesh	About 800,000 meshes (near wall y+ < 1)
Surface of glass	Velocity : No slip $k\|_{wall}$: No slip, $\varepsilon\|_{wall} = 2\nu\left(\frac{\partial\sqrt{k}}{\partial y}\right)^2$



(b) Detail of the evaluated room *(c) Detail of the window*

(a) Applied dwelling house *(d) Detail of the calculation model* *(e) Calculation mesh*

Figure 5 Schematic of calculation model

Table 2 Material properties

Item	Air	Glass
Specific heat (C_P)	1006.43 [J/(kg·K)]	753.00 [J/(kg·K)]
Conductivity (λ)	0.0242 [W/(m·K)]	0.65 [W/(m·K)]
Viscosity (μ)	1.79×10^{-5} [kg/(m·s)]	-
Molecule weight (M)	28.97 [kg/kgmol]	-
Emissivity (ε_1)	-	0.90 [-]
Emissivity of Low-E (ε_2)	-	0.10 [-]

3) Calculation cases

The calculation cases by different air space widths of a glazing unit are summarized in Table 3. The indoor and outdoor pressure difference was assumed to be 10 Pa for calculation.

Table 3. Calculation cases with air space width

Item	Glass
Air space width	0, 6, 12, 18, 24, 36, 48, 60 [mm]
Low emissive film	uncoated, coated
Indoor/Outdoor pressure	-10 [Pa] / 0 [Pa]
Indoor temperature	22 [ºC]
Outdoor temperature	0 [ºC]

4) Results and discussion

CFD calculation results by the air space width of the low emissive film uncoated glazing and low emissive film coated glazing are shown in Fig. 3, Fig. 4, respectively. Figure 3(a), Fig. 4(a) is the thermal resistance, Fig. 3(b), Fig. 4(b) is the thermal transmittance with the air space of glazing.

Calculation results show that the thermal resistance increased with increasing air space width. In case of low emissive film uncoated glazing, the thermal resistance and thermal transmittance was calculated 0.17 K·m^2/W, 2.91 W/(m^2·K) with more than approximately 24 mm of air space width. It was shown that the insulation performance of glazing is unchangeable with increasing air space width.

Similarly, in case of low emissive film coated glazing, the thermal resistance and thermal transmittance was calculated to be 0.39 K·m^2/W, 1.77 W/(m^2·K). It is approximately 164.9% of the insulation performance enhancement in comparison with low emissive film uncoated glazing.

In addition, to confirm the validity of the numerical method, the predicted thermal resistance of air space was compared with theoretical calculated data using Eqn. [1] and [2] The error of the CFD calculation results was small and the reliability of CFD calculation was confirmed by comparison of theoretical thermal transmittance value.

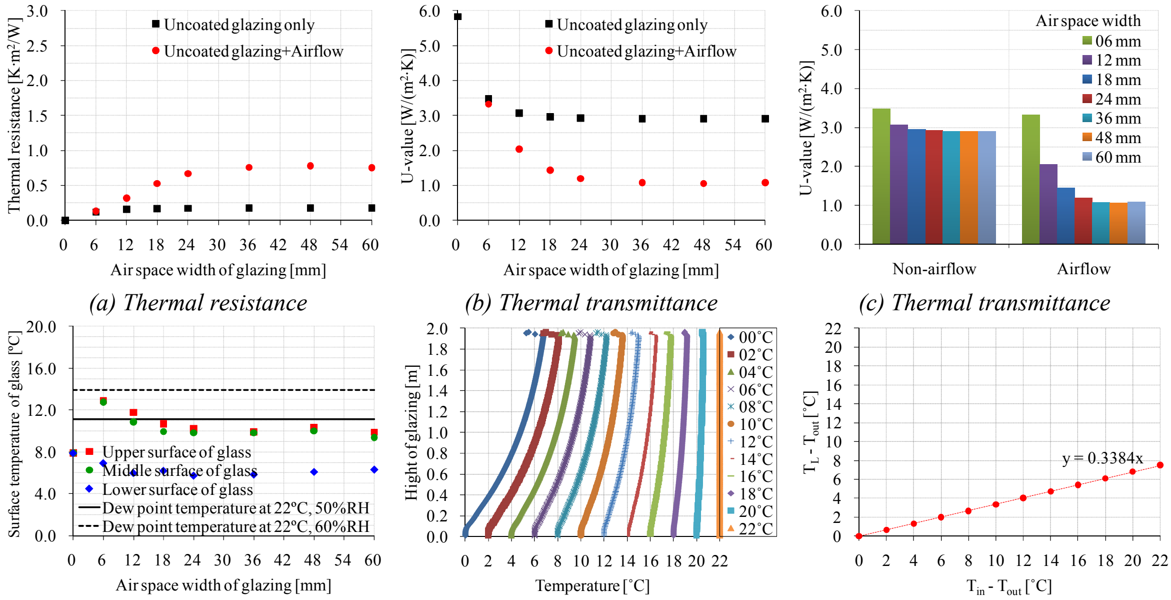
Figure 3(c) and Fig. 4(c) show the calculated results of the surface temperature of the glass to confirm the occurrence of moisture condensation. The calculated results show that moisture condensation occurred at the lower surface of the glass when indoor temperature and relative humidity is above 22.0 ºC, 60 %RH on low emissivity film uncoated glazing. However, condensation does not occur on low emissivity film coated glazing.

SIMULATION

This chapter presents with the evaluation of thermal performance and the occurrence of moisture condensation in the proposed system using CFD.

1) Assumptions

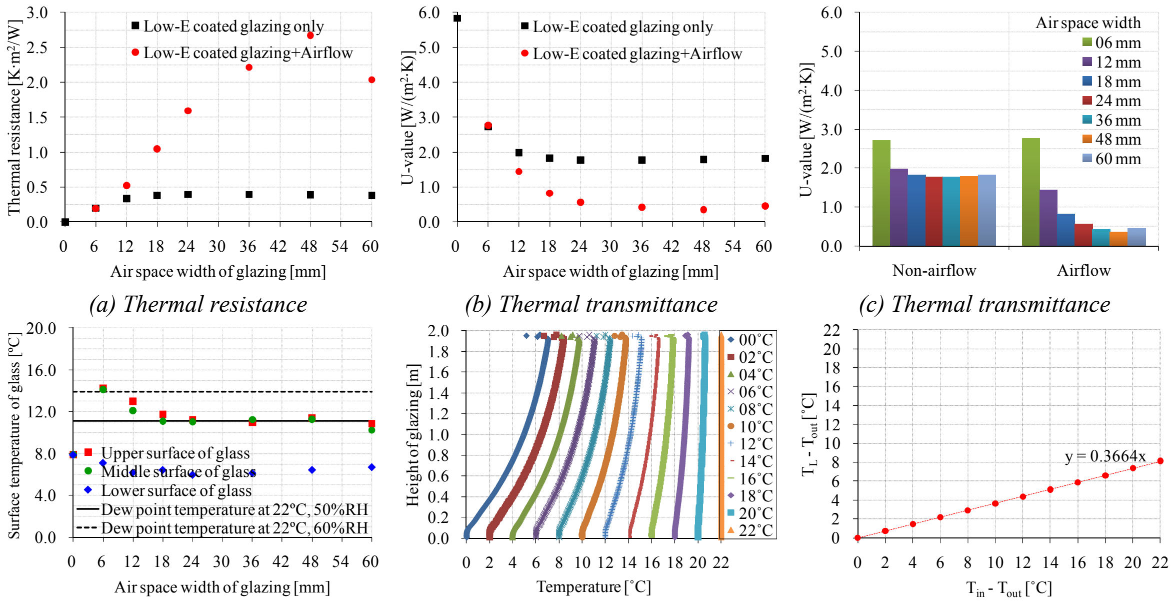
To evaluate the thermal insulation efficiency of the proposed system, the temperature contributions were



(a) Thermal resistance *(b) Thermal transmittance* *(c) Thermal transmittance*

(d) Surface temperature of glass *(e) Temperature in air space* *(f) Supply temperature*

Figure 6 CFD results in uncoated glazing with airflow



(a) Thermal resistance *(b) Thermal transmittance* *(c) Thermal transmittance*

(d) Surface temperature of glass *(e) Temperature in air space* *(f) Supply temperature*

Figure 7 CFD results in Low-E film coated glazing with airflow

calculated using 3-D steady-state CFD simulation including detailed ray-tracing-based radiation modelling as well as direct modelling of convective heat transfer under different air space width of the glazing unit and different outdoor temperature. Figure 5(a) shows a plan of the building model used for the calculation, which is proposed as a standard dwelling house model in Japan by the Institute for Building Environment and Energy Conservation. In this study, a children's room of the 2nd floor (2.40 m (H) × 2.90 m (W) × 3.50 m (D) = 24.36 m^3) is only used for calculations as shown in Fig. 5(b). Figure 5(c) shows the details of the window (1.95 m (H) × 1.65 m (W) × 0.10 m (D)) installed according to the calculation model with Fig. 5(d) and Fig. 5(e). The model included an indoor zone, an outdoor zone, a window, and a window frame. The airflow window system applied to the window glass allowing fresh air into the room and the frame area is about 10% of the whole window area. The opening size (0.02 m (H) × 0.0765 m (W) : 2 EA) of the airflow window system was assumed to satisfy room required ventilation rates at the indoor/outdoor pressure difference of 10 Pa. In Japan, indoor ventilation rates need to provide at least one half of air changes per hour (0.5 ACH) for the entire 24 hours. The effect of the heat-recovery heat pump system is not considered, in order to keep the calculation model simple in this study.

2) Calculation cases

The calculation cases by different air space width (6, 12, 18, 24, 36, 48, 60 mm) of the glazing unit of uncoated glazing and Low-E film coated glazing are summarized in Table 4. In this case, the indoor and outdoor temperature is assumed to be 22.0 ºC, 0.0 ºC for calculation the indoor/outdoor pressure difference is fixed at 10 Pa. Moreover, the calculation cases by outdoor temperature (0, 2, 4, 6, 8, 10, 12, 14, 16, 18, 20, 22 ºC) of uncoated glazing and Low-E film coated glazing are summarized in Table 5. In this case, the supply air temperature is calculated with a fixed air space width of 12 mm and a fixed indoor/outdoor pressure difference.

Table 4. Calculation cases with air space width

Item	Glass
Air space width	6, 12, 18, 24, 36, 48, 60 [mm]
Air supply inlet	0.02 m × 0.765 m : 2EA
Low emissive film	uncoated, coated
Indoor/Outdoor pressure	-10 [Pa] / 0 [Pa]
Indoor temperature	22 [ºC]
Outdoor temperature	0 [ºC]

Table 5. Calculation cases with outdoor temperature

Item	Glass
Air space width	12 [mm]
Low emissive film	coated, uncoated
Indoor/Outdoor pressure	-10 [Pa] / 0 [Pa]
Indoor temperature	22 [ºC]
Outdoor temperature	0, 2, 4, 6, 8, 10, 12, 14, 16, 18, 20, 22 [ºC]

RESULTS AND DISCUSSION

1) Calculation results with air space width

The thermal resistance and thermal transmittance was predicted for the proposed airflow window system with a range of air space widths. Figures 6(a), 6(b), 6(c) show the calculation results of thermal resistance and thermal transmittance with the air space width on uncoated glazing with airflow and Figs. 7(a), 7(b), 7(c) show that on Low-E film coated glazing with airflow.

The calculation results show that thermal resistance increases with increasing air space width. In case of low emissive film uncoated glazing combined airflow, the thermal resistance and thermal transmittance was calculated to be 0.67 K·m^2/W, 1.19 W/(m^2·K) with more than approximately 24 mm of air space width. It was shown that the insulation performance of glazing is unchangeable with increasing air space width above 24 mm. It is approximately 245.7% of insulation performance enhancement in comparison with the dual pane window.

Similarly, in case of low emissive film coated glazing combined airflow, the thermal resistance and thermal transmittance was calculated to be 1.59 K·m^2/W, 0.57 W/(m^2·K). It is approximately 208.8% of insulation performance enhancement in comparison with low emissive film uncoated glazing combined airflow.

2) Calculation results of moisture condensation

Figures 6(d), 7(d) show the calculated results of the surface temperature of glass to confirm the occurrence of moisture condensation.

The calculated results show that moisture condensation occurred on the lower surface of glass when the indoor temperature and relative humidity is above 22.0 ºC, 50 %RH. It was not only the low emissivity film uncoated glazing combined airflow but also the low emissivity film coated glazing combined airflow.

3) Calculation results of inlet air temperature

Figures 6(e), 6(f), 7(e), 7(f) show the calculated results for the airflow window of the proposed system by the outdoor temperature at the fixed indoor/outdoor pressure difference of 10 Pa. Figures 6(e), 6(f) show the air temperature in air space in uncoated glazing with airflow and Fig. 7(e), 7(f) show air temperature in air space in Low-E film coated glazing with airflow.

The calculated results show that the variation of the supply air temperature increased with a decrease in the indoor/outdoor temperature difference at a fixed pressure difference. In winter, low-temperature outdoor air comes indoors through the air space of glazing. The relationship of the temperature difference between indoor air/outdoor air and supply air/outdoor air is calculated and leads to Eqn. [9] and [10] as a linear approximation:

$$T_L = 3.384 \times 10^{-1} \times (T_{in} - T_{out}) + T_{out} \qquad \ldots[9]$$

$$T_L = 3.664 \times 10^{-1} \times (T_{in} - T_{out}) + T_{out} \qquad \ldots[10]$$

where, T_L is the supply air temperature [ºC], T_{in} is the indoor air temperature [ºC], T_{out} is the outdoor air temperature [ºC].

Moreover, the results show that it is warmed the outdoor air more Low-E film coated glazing than uncoated glazing.

CONCLUSIONS

This paper proposes a new airflow window system to reduce energy consumption, and the results of a feasibility study on its effectiveness using CFD simulation. As a result of this study, we found that the thermal insulation efficiency of the proposed system increased with increasing air space width but it is unchanged with above 24 mm of air space width. Moreover, the calculation results show that it is effective to use low emissive film in the airflow system of the proposed system.

A summary of the general findings of this study is as follows.

- The error of CFD calculation results was small and the reliability of CFD calculation was confirmed by comparison of the theoretical thermal transmittance value on the dual pane window.
- In preliminary simulation, the results of the low emissive film coated glazing showed the thermal resistance and thermal transmittance to be 0.39 K·m²/W, 1.77 W/(m²·K). It is approximately 164.9% of the insulation performance enhancement in comparison with low emissive film uncoated glazing.
- The thermal resistance and thermal transmittance was predicted for the airflow window of the proposed system with a range of air space widths. Although, calculation results show that thermal resistance increased with increasing the air space width, insulation performance of glazing is unchangeable with increasing the air space width above 24 mm. The low emissive film uncoated glazing combined airflow, the thermal resistance and thermal transmittance was calculated to be 0.67 K·m²/W, 1.19 W/(m²·K).
- The moisture condensation occurs at the lower surface of glass when the indoor temperature and relative humidity are above 22.0 ºC, 50 %RH. This occurrs not only with the low emissivity film uncoated glazing combined airflow but also with the low emissivity film coated glazing combined airflow.
- The variation of the supply air temperature increased with a decrease in the indoor/outdoor temperature difference at a fixed pressure difference. The relationship of the temperature difference between indoor air/outdoor air and supply air/outdoor air is calculated as a linear approximation:

FUTURE PERSPECTIVE

A futher study is required to evaluate various design models such as double dual pane, triple glass, low-e glass etc., because it is important to prevent moisture condensation on an indoor glass surface. Moreover, thermal comfort should be calculated for a realizable room model by examining the effects of any cold drafts and the energy-saving effects of the heat pumps included in the proposed system, and this will be evaluated in future investigations. First, the following themes will be studied in order not to cause backflow, to ventilate stable air supply in the near future.

- The effect of indoor/outdoor pressure caused by outside wind conditions.
- The ventilation due to buoyancy caused by differences in indoor/outdoor temperature.
- In case of two-side-openings (when openings are installed in different directions).
- The effect of thermal environments caused by drafts.

ACKNOWLEDGEMENTS

This study was made possible by the financial support received from the Ministry of the Environment, Japan. Moreover, we would like to thank the staff of Sankyo Tateyama Aluminium Inc., J Architecture System Inc., and Tsuchiya Hometopia Inc. for their generous cooperation.

NOMENCLATURE

U : Thermal transmittance [W/(m²·K)]
R_s : Thermal resistance of glass [m²·K/W]
h_c : Convective heat transfer coefficient between an air [W/(m²·K)]
h_e : External heat transfer coefficient [W/(m²·K)]
h_i : Internal heat transfer coefficient [W/(m²·K)]
h_r : Radiative heat transfer coefficient between an air [W/(m²·K)]
h_s : Overall heat transfer coefficient between an air space [W/(m²·K)]
N : Number of air space [-]
M : Number of glass [-]
d_m : Thickness of glass [m]
λ_m : Thermal conductivity of glass [W/(m·K)]
T_L : Supply air temperature [ºC]
T_{in} : Indoor air temperature [ºC]
T_{out} : Outdoor air temperature [ºC]
T_m : Mean temperature at air space of glass [K]
σ : Stefan-Boltsmann constant [W/(m²·K⁴)]
ε_1 : Emissivities of one panes of glass [-]
ε_2 : Emissivities of another one panes of glass [-]
F_{12} : View factor between two panes of glass [-]
Nu : Average Nusselt number [-]
λ_a : Thermal conductivity of air [W/(m·K)]
s : Mean width of air space [m]
Gr : Grashof number [-]
Pr : Prandtl number [-]
ρ : Density of air [kg/m³]
μ : Viscosity of air [kg/(m·s)]
c_p : Heat capacity of air [J/(kg·K)].

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