

# The cooling performance of a building integrated evaporative cooling system driven by solar energy

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## ABSTRACT

The solar chimney is a passive cooling technique to enhance the natural ventilation of buildings. The effect is, however, limited under hot and humid climatic conditions. In the study, the solar chimney was accompanied by a dew-point evaporative cooler. The dew-point evaporative cooler was integrated with the ceiling of a building. The air flow induced by the solar chimney was predicted by simulation, and the cooling effect of the dew-point evaporative cooler was also analyzed by heat and mass transfer simulation. The results showed that the system was capable of coping with internal heat gains of an ordinary office building. In addition, the optimal geometry of the evaporative cooling channel was revealed.

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## 1. Introduction

Buildings are large energy consumers in urban areas. In Japanese case, for instance, approximately one-third of primary energy is consumed by residential and commercial buildings [1]. The heating of space is dominant for energy consumption in residential buildings because the traditional Japanese houses are made of wood, and kerosene or gas heaters are used in those houses. On the other hand, the cooling demand is dominant in commercial buildings, especially in office buildings, because of high-insulated and air-tight construction as well as increased office automation equipment. In the context of energy and environmental conservation, the solar heating/cooling is one of the most desirable schemes, one of technological challenges of which is the passive cooling technique in hot and humid climate.

The passive cooling can be defined as the cooling method without active mechanical devices such as pumps and fans, and the provision of cooling relies mainly on the air flow caused by the buoyancy [2]. Solar chimney, Trombe walls, and double facades are devices to induce the air flow by the stack effect, and the enhanced natural ventilation by those devices will deliver the cooling effect [3–7]. The natural ventilation may, however, harm the thermal comfort of buildings if it is implemented in hot and humid climate because the temperature and humidity of outdoor air are much higher than those of the air-conditioned room during daytime. Even

at night, it is not rare in Japan that the outdoor air temperature exceeds 25 °C throughout the night during the summer season.

One way to cool the outdoor air before it is supplied to the space is the evaporative cooling. The evaporative cooling technique can be used either with active systems or with passive systems. The studies of evaporative cooling for passive systems are not many, however, and the technology to integrate it with building structure is still under development [2]. Raman et al. [8] tested passive solar systems, which provided heating, cooling and ventilation, in composite climate including hot-dry, hot-humid and cold climate. It was presented that the passive system, which consisted of a south wall collector and a roof duct with evaporative cooling surface, moderated the temperature variation of the room, and the room temperature was kept around 30 °C when the ambient temperature was 42 °C at maximum during the summer period. Chungloo and Limmeechokchai [9] conducted performance measurement of a solar chimney combined with a water sprayed roof. The results showed that the combination of the two passive measures decreased the room air temperature by 2.0–6.2 °C compared with the ambient temperature. Maerefat and Haghighi [10] investigated a system with a solar chimney and an evaporative cooling cavity by using a mathematical model. The effects of the geometrical dimensions as well as of the ambient conditions on the cooling performance were analyzed, and the results showed that the proposed system was capable of improving the indoor conditions even with low solar intensity of 200 W/m<sup>2</sup> and high ambient temperature of 40 °C. The room air temperature strongly depended on the relative humidity of the ambient air, however. They concluded that the system provided comfortable indoor air conditions at high ambient temperature with the relative humidity lower than 50%.

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## Nomenclature

$A$	area ( $\text{m}^2$ )
$a$	channel height (m)
$C_c$	contraction coefficient
$C_d$	discharge coefficient
$c_p$	specific heat ( $\text{J kg}^{-1} \text{K}^{-1}$ )
$d$	hydraulic diameter (m)
$g$	gravitational acceleration ( $\text{m s}^{-2}$ )
$H$	height (m)
$h$	enthalpy ( $\text{J kg}^{-1}$ )
$hr$	hour from midnight (h)
$k$	thermal conductivity ( $\text{W m}^{-1} \text{K}^{-1}$ )
$L$	length (m)
$\dot{m}$	mass flow rate ( $\text{kg s}^{-1}$ )
$P$	pressure (Pa)
$\dot{Q}$	heat load (W)
$\dot{q}$	heat flux ( $\text{W m}^{-2}$ )
$\dot{q}_C$	convective cooling load ( $\text{W m}^{-2}$ )
$\dot{q}_R$	radiative cooling load ( $\text{W m}^{-2}$ )
$r$	latent heat of water ( $\text{J kg}^{-1}$ )
$S$	solar radiation ( $\text{W m}^{-2}$ )
$T$	temperature (K)
$t$	temperature ( $^{\circ}\text{C}$ )
$u$	velocity ( $\text{m s}^{-1}$ )
$\dot{V}$	volume flow rate ( $\text{m}^3 \text{s}^{-1}$ )
$W$	width (m)
$X$	humidity ratio ( $\text{kg kg}^{-1}$ )
$X^*$	humidity ratio of saturated air ( $\text{kg kg}^{-1}$ )
$z$	position (m)

## Greek symbols

$\alpha$	convective heat transfer coefficient ( $\text{W m}^{-2} \text{K}^{-1}$ )
$\alpha_m$	mass transfer coefficient ( $\text{m s}^{-1}$ )
$\beta$	radiative heat transfer coefficient ( $\text{W m}^{-2} \text{K}^{-1}$ )
$\epsilon$	emissivity
$\lambda$	friction loss coefficient
$\rho$	density ( $\text{kg m}^{-3}$ )
$\sigma$	the Stefan-Boltzmann constant ( $\text{W m}^{-2} \text{K}^{-4}$ )
$\zeta$	loss coefficient

## Subscripts

a	air or outdoor air
C	solar chimney
c	ceiling
DA	dry air
d	dry channel
dp	dew point
f	air in the solar chimney
g	glass
i	inlet
M	the M-Cycle
o	outlet
p	heat transfer plate
r	room
rev	reverse section
s	sky
s1, s2	water absorbing sheet
w	wall or wet channel
wind	wind

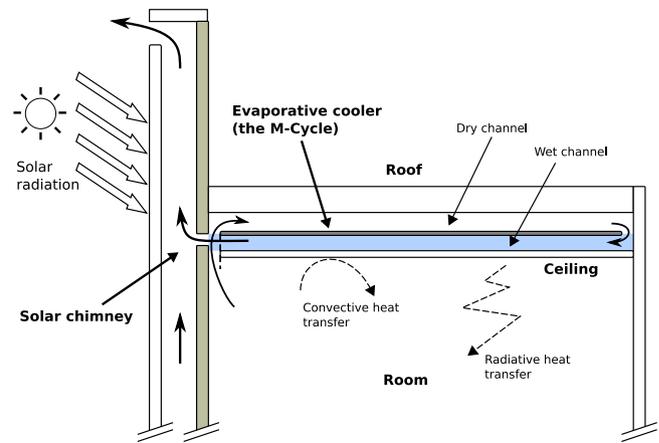


Fig. 1. The solar chimney driven evaporative cooling system integrated with the ceiling.

The purpose of the study is to develop a passive cooling device that will effectively reduce the cooling load of main air conditioners under hot and humid climatic conditions. The authors have proposed the building integration of a passive cooling system that consisted of a solar chimney and a dew point evaporative cooler [11]. This article reports the simulation of the cooling performance of the system with a modified design.

## 2. The system description

### 2.1. The building integrated solar cooling system

Our attempt is the integration of the solar cooling system into architecture. Fig. 1 depicts an overview of the building integrated evaporative cooling system and the direction of the air movement. The building is featured with a solar chimney that is attached to the south-facing wall. The evaporative cooler is integrated with the ceiling panel so that the wet channel is adjacent to the room side. The evaporative cooler is called the M-Cycle [12]. In our previous design, the dry channel of the M-Cycle was adjacent to the room to avoid problems that may arise from water leakage and penetration. The design was, however, inferior to the present design in the cooling effect.

The air inside the solar chimney is heated by the solar radiation, and the upward flow is generated due to the stack effect. The upward flow inside the chimney induces the air movement in the room. By connecting the air exit of the evaporative cooler with the inlet of the solar chimney, the air travels through the evaporative cooler before it flows the solar chimney. The inlet air duct of the evaporative cooler can be connected either to the room or to the outdoor. In hot humid climate, the inlet air of the evaporative cooler should be the conditioned room air so that the temperature of the ceiling panel is lower than the room air temperature. In this case, the cooling effect is owing to the dehumidified air by the room air conditioner, and the system assists the room air conditioner by reducing the cooling load. If the outdoor air is moderate in terms of temperature and humidity, the inlet air to the evaporative cooler can be the outdoor air. The system will provide cooling without any dehumidification work.

### 2.2. The dew point evaporative cooler

The evaporative cooler is a device to lower the air temperature by the latent heat of water. The water as a refrigerant evaporates by taking heat from the working air, and hence the working air is cooled and humidified. The working air has a potential to contain



The convective heat transfer coefficients,  $\alpha_g$  and  $\alpha_w$ , were found by the empirical correlations for the free convection flow on the vertical plane, which is given in Eq. (7)[15].

$$Nu = \begin{cases} \left\{ 0.825 + \frac{0.387Ra^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2 & (Ra > 10^9) \\ 0.68 + \frac{0.670Ra^{1/4}}{[1 + (0.492/Pr)^{9/16}]^{4/9}} & (Ra \leq 10^9) \end{cases} \quad (7)$$

On the other hand, the heat transfer coefficient by wind,  $\alpha_{wind}$  was given as a function of the wind velocity,  $u_{wind}$ , by Eq. (8)[16].

$$\alpha_{wind} = 10.21u_{wind} + 4.47 \quad (8)$$

The radiative heat transfer coefficients,  $\beta_{gw}$  and  $\beta_{gs}$ , were defined as follows:

$$\beta_{gw} = \frac{\sigma(T_g + T_w)(T_g^2 + T_w^2)}{1/\epsilon_g + 1/\epsilon_w - 1} \quad (9)$$

$$\beta_{gs} = \sigma\epsilon_g(T_s + T_g)(T_s^2 + T_g^2) \frac{(T_s - T_g)}{(T_a - T_g)} \quad (10)$$

where  $T_s$  is the sky temperature, which is calculated by Eq. (11)[17].

$$T_s = T_a[0.711 + 0.0056t_{dp} + 0.000073t_{dp}^2 + 0.013 \cos(15hr)]^{1/4} \quad (11)$$

where  $t_{dp}$  represents the dew point temperature of the outdoor air in degree Celsius, and  $hr$  denotes the hour from midnight.  $\dot{q}_f$  in Eq. (5) represents the heat flux to the fluid, which is given by the following expression.

$$\dot{q}_f = \frac{\rho_a c_{pa} \dot{V}_f}{WH} (T_{f,o} - T_{f,i}) \quad (12)$$

where  $W$  denotes the chimney width, and  $H$  denotes the chimney height.  $T_{f,i}$  and  $T_{f,o}$  are the temperatures of the fluid at the inlet and at the outlet of the chimney, respectively, and the mean fluid temperature can be estimated by Eq. (13).

$$T_f = \gamma T_{f,o} + (1 - \gamma) T_{f,i} \quad (13)$$

The value of  $\gamma$  will vary depending on the shape of the chimney because the bulk mean temperature changes in accordance with the temperature, density and velocity distributions. We found that  $\gamma = 0.5$  was a good approximation to represent the mean fluid temperature under our configuration of the solar chimney based on the CFD analysis [14].

By substituting Eq. (13) for Eq. (12), we obtain Eq. (14).

$$\dot{q}_f = \frac{\rho_a c_{pa} \dot{V}_f}{\gamma WH} (T_f - T_{f,i}) \quad (14)$$

The formula to predict the air flow rate due to buoyancy was presented by Andersen [18]. The air flow rate could be expressed based on the temperature difference between the fluid and the ambient as Eq. (15).

$$\dot{V}_f = C_{di} A_i \left( \frac{2gH_i \Delta T}{T_f} \right)^{1/2} \quad (15)$$

where  $A_i$  is the inlet aperture area, and  $C_{di}$  is the discharge coefficient at the inlet.  $H_i$  is the height from the center of the inlet aperture to the neutral pressure plane.

### 3.2. The mathematical model of the M-Cycle

The temperature profiles of the air in the M-Cycle and the ceiling surface were predicted by the mathematical model. The model was based on the heat and mass balances of a small control volume, which is illustrated in Fig. 4. The model used the following assumptions:

- The enthalpy balance of the wet air can be expressed independently by the sensible heat balance and by the mass balance (latent heat balance).
- The water absorbing sheet can maintain water on the surfaces of heat transfer plate and of the ceiling, and the air is in saturation at the interface between the water absorbing sheet layer and the air layer.
- The effects of the conductive heat transfer in horizontal direction inside the heat transfer plate, the water absorbing sheet, and the ceiling are negligible.
- The upper side of the dry channel is well insulated so that the heat gain through the roof is minimal.

The sensible heat balances of the air in the dry channel and in the wet channel were expressed by Eqs. (16) and (17).

$$\rho_d c_{pd} u_d a_d \frac{dT_d}{dz} = \alpha_d (T_p - T_d) \quad (16)$$

$$\rho_w c_{pw} u_w a_w \frac{dT_w}{dz} = \alpha_{w1} (T_{s1} - T_w) + \alpha_{w2} (T_{s2} - T_w) \quad (17)$$

The subscripts d, w, and s represent the dry channel, the wet channel, and the water absorbing sheet, respectively, and w1 and w2 represent the heat transfer plate side (upper side) of the wet channel and the ceiling side (lower side) of the wet channel, respectively. The subscripts s1 and s2 are in a similar definition.

The mass balance of the wet channel was expressed by Eq. (18).

$$\rho_{DA} u_w a_w \frac{dX_w}{dz} = \rho_{DA} (\alpha_{m1} + \alpha_{m2}) (X_w^* - X_w) \quad (18)$$

where  $X_w^*$  denotes the humidity ratio of the saturated air at temperature  $T_w$ .

The energy balances concerning the heat transfer plate, the water absorbing sheets, and the ceiling were expressed by Eqs. (19)–(22), respectively.

$$\alpha_d (T_d - T_p) + \frac{k_{e1}}{a_{e1}} (T_{s1} - T_p) = 0 \quad (19)$$

$$\frac{k_{e1}}{a_{e1}} (T_p - T_{s1}) + \alpha_{w1} (T_w - T_{s1}) + \rho_{DA} \alpha_{m1} r (X_w - X_w^*) = 0 \quad (20)$$

$$\frac{k_{e2}}{a_{e2}} (T_c - T_{s2}) + \alpha_{w2} (T_w - T_{s2}) + \rho_{DA} \alpha_{m2} r (X_w - X_w^*) = 0 \quad (21)$$

$$\frac{k_{e2}}{a_{e2}} (T_{s2} - T_c) + \alpha_r (T_r - T_c) + \dot{q}_R = 0 \quad (22)$$

where  $a_{e1}$  and  $a_{e2}$  are effective thickness of the heat transfer plate and that of the ceiling, respectively, and they are defined as  $a_{e1} = a_p + a_{s1}$  and  $a_{e2} = a_c + a_{s2}$ , respectively.  $k_{e1}$  and  $k_{e2}$  are the effective thermal conductivity of the heat transfer plate and that of the ceiling and they are defined as  $k_{e1} = a_{e1} / (a_p/k_p + a_{s1}/k_{s1})$  and  $k_{e2} = a_{e2} / (a_c/k_c + a_{s2}/k_{s2})$ , respectively.  $\alpha_d$ ,  $\alpha_{w1}$  and  $\alpha_{w2}$  are the heat transfer coefficients between the air and the wall surfaces in the dry and wet channels.  $\alpha_r$  is the heat transfer coefficient between the room air and the ceiling surface, and given by  $\alpha_r = 1.52(\Delta T)^{0.33}$  [19]. The heat input to the ceiling by radiative heat exchange is given by  $\dot{q}_R$ .

The heat transfer coefficients inside the dry and wet channels were approximated by the Nusselt number of the developed laminar flow in rectangular shape channels under the condition of uniform heat flux along the flow direction and the uniform wall temperature at a cross section. The three side walls of the dry channel were insulated, while the two side walls facing each other were insulated in case of the wet channel. Thus, the Nusselt numbers were 5.38 and 8.23 for the dry channel and for the wet channel, respectively [20]. The effect of the entrance region was ignored because the heat transfer coefficient at the entrance region was

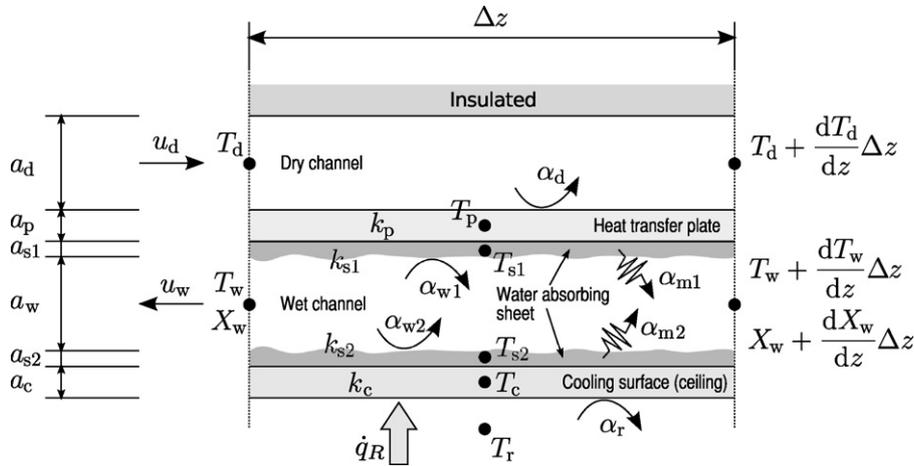


Fig. 4. The control volume of the heat and mass transfer model of the M-Cycle.

higher than that of the fully developed region. Therefore, the performance would not be over-estimated due to the neglect of the entrance region. To find the mass transfer coefficient inside the wet channel, a simple assumption where the Lewis number equals one was used. Then, the mass transfer coefficient was given as  $\alpha_m = \alpha_w / (\rho_w c_{pw})$ .

### 3.3. The integration of the solar chimney and the evaporative cooler

The air flow rate of the solar chimney expressed by Eq. (15) was derived from the pressure difference between the inside and the outside at the inlet of the solar chimney. In the case of the M-Cycle integrated solar chimney, the dry and wet channels of the M-Cycle will cause a large resistance to the air flow of the solar chimney. Because the pressure drop by the M-Cycle would be dominant for the air flow rate, the air flow rate induced by the solar chimney was estimated by taking into account the pressure drop from the inlet of the M-Cycle to inside the solar chimney.

The pressure drop due to the M-Cycle was expressed by Eq. (23).

$$\Delta P_M = \frac{1}{2} \rho_a \zeta_{M,i} u_{M,i}^2 + \frac{1}{2} \rho_a \left( \lambda_M \frac{2L}{d_M} + \zeta_{rev} + \zeta_{C,i} \right) u_M^2 \quad (23)$$

where  $u'_{M,i}$  and  $u_M$  denote the velocity of vena contracta at the inlet of the M-Cycle and the velocity of the air inside the M-Cycle, respectively. The friction coefficient is given by  $\lambda_M$  and the loss coefficients at the inlet of the M-Cycle, at the reverse section and at the inlet to the solar chimney (the exit of the M-Cycle) are represented by  $\zeta_{M,i}$ ,  $\zeta_{rev}$  and  $\zeta_{C,i}$ , respectively.  $L$  is the length of the channel, and  $d_M$  is the hydraulic diameter of the channel.

Then, the pressure difference between the inlet of the M-Cycle ( $P_r$ ) and the inside of the solar chimney ( $P_{C,i}$ ) could be given by:

$$\Delta P_i = P_r - P_{C,i} = \frac{1}{2} \rho_a u_{C,i}^2 + \Delta P_M \quad (24)$$

Using the mass balance,  $\rho_a C_{cM,i} A_{M,i} u'_{M,i} = \rho_a C_{cC,i} A_{C,i} u_{C,i}$ , Eq. (24) could be rewritten as Eq. (25).

$$\Delta P_i = \frac{1}{2} \rho_a \psi u_{M,i}^2 \quad (25)$$

where  $\psi$  was defined by Eq. (26).

$$\psi = \left( \frac{C_{cM,i} A_{M,i}}{C_{cC,i} A_{C,i}} \right)^2 + \zeta_{M,i} + \left( \lambda_M \frac{2L}{d_M} + \zeta_{rev} + \zeta_{C,i} \right) \left( \frac{C_{cM,i} A_{M,i}}{A_M} \right)^2 \quad (26)$$

where  $C_{cC,i}$  and  $C_{cM,i}$  are the contraction coefficients at the inlet of the M-Cycle and at the inlet of the solar chimney, respectively.

$A_{C,i}$  denotes the inlet aperture area of the solar chimney.  $A_{M,i}$  and  $A_M$  represent the inlet aperture area and the sectional area of the M-Cycle, respectively.

Similar to the case of a single solar chimney, the air flow rate could be predicted by Eq. (27).

$$V_M = C_{dM} A_{M,i} \sqrt{\frac{2 \Delta T g H_M}{\bar{T}_f}} \quad (27)$$

where  $C_{dM}$  is the discharge coefficient including the M-Cycle, and it is defined as  $C_{dM} = C_{cM,i} / \sqrt{\psi}$ .

### 3.4. The cooling performance of the system

The system provides radiative cooling by direct heat exchange between the ceiling surface and the objects in the room such as people, furniture and appliances. The radiative cooling effect was expressed by  $\dot{q}_R$  in the model. On the other hand, the convective heat transfer between the ceiling surface and the room air results in the convective cooling effect. The convective cooling effect at local position was defined as  $\dot{q}'_c = \alpha_r (T_r - T_c)$ . Then, the global average of the convective cooling effect was defined by:

$$\dot{q}_C = \int_0^L \frac{\dot{q}'_c dz}{L} \quad (28)$$

## 4. Results and discussion

### 4.1. The conditions of the simulation

The weather data were taken from Ref. [21]. The outdoor air temperature and the humidity ratio are shown in Fig. 5, and the

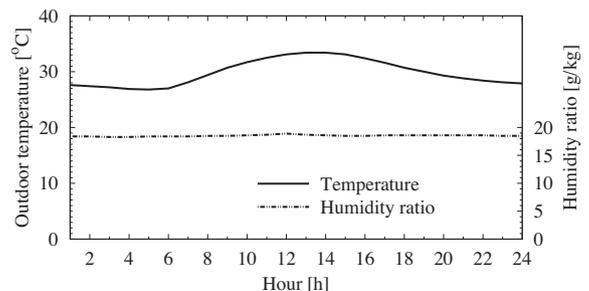


Fig. 5. The outdoor temperature and humidity ratio on a representative summer day in Tokyo, Japan.

**Table 1**  
The radiation properties of the solar chimney materials.

Property	Glass	Wall
Transmissivity	0.84	0.00
Absorptivity	0.06	0.95
Emissivity	0.9	0.9

**Table 2**  
The materials and properties of the M-Cycle.

Component	Materials	Thickness (m)	Thermal conductivity ( $\text{W m}^{-1} \text{K}^{-1}$ )
Heat transfer plate	Aluminum	0.002	237
Water absorbing sheet	Fiber	0.002	0.4 (including water)
Ceiling	Plasterboard	0.01	0.5

solar radiation on the horizontal and vertical surfaces are given in Fig. 6. A constant wind velocity of 1 m/s was used for the simulation of the solar chimney.

The simulations of the solar chimney and the M-Cycle were based on the conditions as follows.

#### 4.1.1. The solar chimney

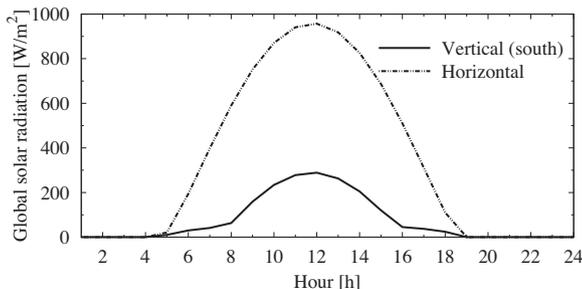
The chimney was consisted of the outer glass layer and the inner wall layer. The radiation properties of the materials are given in Table 1. The geometry of the solar chimney was 3 m in height, 1 m in width. It was assumed that the air gap in the chimney was 0.3 m, which was near the optimum to maximized the air flow rate [14]. The inlet and outlet apertures were the same size and it was  $0.1 \text{ m} \times 1 \text{ m}$ . The contraction coefficient was assumed as 0.6 for both apertures. The value of  $\gamma$  in Eq. (13) was 0.5.

#### 4.1.2. The M-Cycle

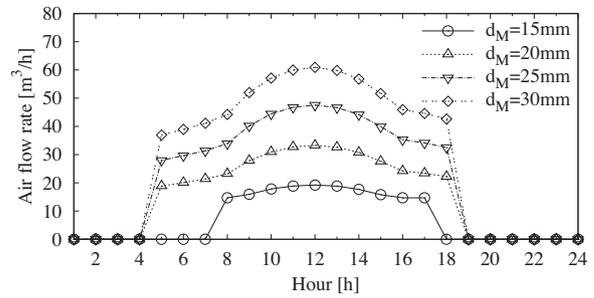
The length of the channel was 5 m and the width was 1 m. The gap width of both the air flow channels was 25 mm under a nominal standard condition. The value was changed for parametric analysis. The aperture size was  $0.1 \text{ m} \times 1 \text{ m}$  for both the inlet and outlet apertures. The material, thickness and thermal conductivity of the heat transfer plate, the water absorbing sheet and the ceiling are given in Table 2. The inlet air to the M-Cycle was the conditioned room air. It was assumed that the temperature and the relative humidity of the room were maintained at  $26^\circ\text{C}$  and 50%, respectively, by a room air-conditioner.

#### 4.2. The airflow induced by the solar chimney

The hourly variations of the air flow induced by the solar chimney are shown in Fig. 7. The effect of the channel width of the



**Fig. 6.** The solar radiation on the horizontal and vertical surfaces on a representative summer day in Tokyo, Japan.



**Fig. 7.** The air flow rate of the solar chimney with the channel width of 15 mm, 20 mm, 25 mm and 30 mm.

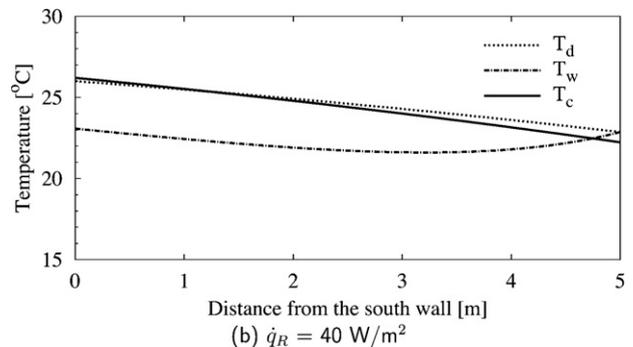
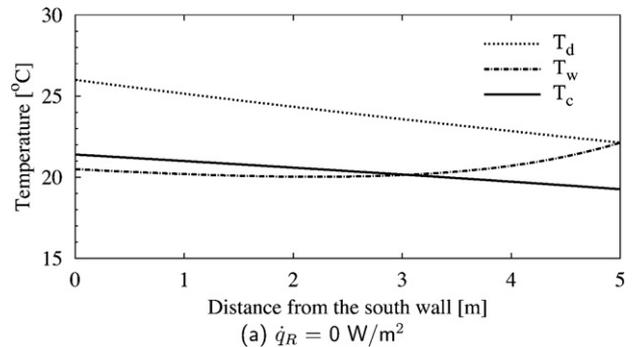
M-Cycle on the air flow rate was also presented in the figure. The results showed that the sufficient air flow could be obtained when solar radiation was available.

The channel width of the M-Cycle was influential on the air flow rate. The air flow rate was amplified by the wider channel width because the pressure drop was mitigated. On the other hand, the wider channel width will result in a lower cooling performance because the heat transfer area per channel volume will decrease. As the system contributes to energy savings in terms of reductions of mechanical ventilation work as well as of air-conditioning load, the optimal geometry of the M-Cycle has to be chosen by the consideration of both the air flow rate and the cooling performance.

#### 4.3. The temperature distribution on the cooled surface

The prediction of the M-Cycle performance was carried out under the condition where the air flow rate was  $45 \text{ m}^3/\text{h}$ . Fig. 8 shows the temperature profiles of the air and the ceiling for the case of no radiative cooling load and the radiative cooling load of  $40 \text{ W}/\text{m}^2$ .

At the M-Cycle part of the system, the air first enters the dry channel from the inlet located nearby the south wall, and flows



**Fig. 8.** The temperature profiles of the air in the dry and wet channels and the ceiling: (a) no radiative cooling load, (b) the radiative cooling load of  $40 \text{ W}/\text{m}^2$ .

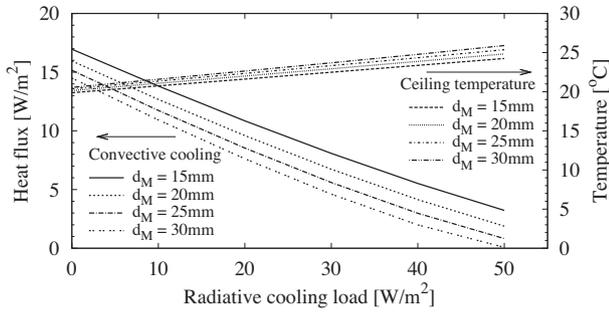


Fig. 9. The effects of the channel width as well as the radiative cooling load on the convective cooling capacity and on the average ceiling temperature.

toward the direction of interior of the room. The air flow direction is reversed at the end of the dry channel, and then the air flows back in the wet channel to the solar chimney attached to the south wall. It was shown that the air temperature was gradually decreased in the dry channel and it was the minimum at the farthest side. The ceiling temperature was also varied correspondingly.

The ceiling temperature should be lower than the room temperature to provide cooling effect, but it was strongly affected by the radiative cooling load  $\dot{q}_R$ . In the case of the no radiative cooling load, the ceiling temperature was under 22 °C throughout the surface. In the case of  $\dot{q}_R = 40 \text{ W/m}^2$ , however, the ceiling temperature was close to the room temperature around the inlet of the M-Cycle. The results implied that 40 W/m<sup>2</sup> of the cooling load was close to the maximum cooling capacity of the system.

4.4. The cooling capacity of the system

The effects of the radiative cooling load on the convective cooling capacity and the average ceiling temperature are shown in Figs. 9 and 10. Fig. 9 was based on the constant air flow rate of 45 m<sup>3</sup>/h with the variations of the channel width 15–30 mm. While, Fig. 10 was under the fixed channel width at 25 mm and the air flow rate was varied from 15 to 60 m<sup>3</sup>/h.

Both the results showed the decreasing convective cooling capacity and the increasing average ceiling temperature by the magnification of the radiative cooling load. The overall cooling capacity of the system was the sum of the radiative and convective cooling loads, and the overall cooling capacity was amplified with the increase of the radiative cooling load. The larger radiative cooling load would, however, have a negative effect on the thermal comfort of the room because of the higher ceiling temperature.

As a result, the overall cooling capacity of the system would depend on the internal conditions of the room, such as the heat emission from people and equipment as well as the temperature of them. Compared with internal heat gains from people and equipment, which would be around 50 W/m<sup>2</sup> for an ordinary office

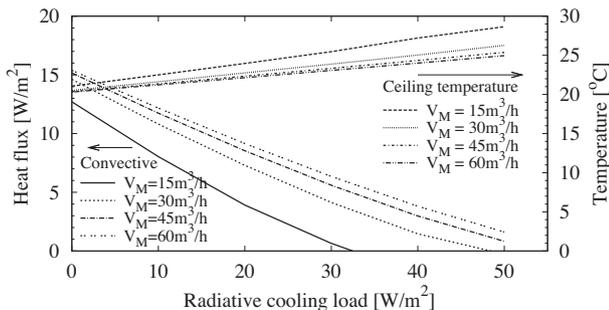


Fig. 10. The effects of the air flow rate as well as the radiative cooling load on the convective cooling capacity and on the average ceiling temperature.

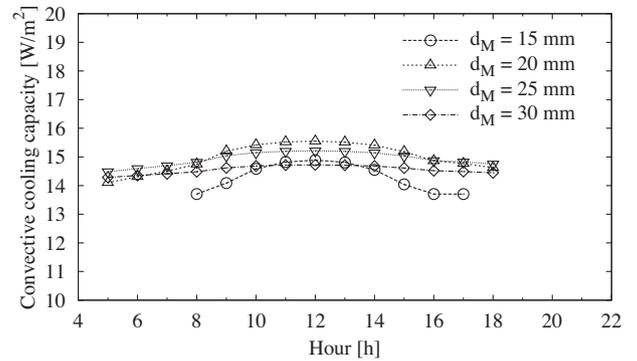


Fig. 11. The effects of the channel width on the hourly convective cooling capacity without the radiative cooling load.

building, for example, it could be expected that the proposed system was capable of removing those internal heat gains if all the ceiling surface was covered by the system. In addition, it was reported that the maximum cooling loads of existing office buildings in Japan were approximately 50–100 W/m<sup>2</sup> [22]. Compared with these ranges of maximum cooling load, the cooling capacity of 50 W/m<sup>2</sup> has a potential to cover 50–100% of the maximum cooling load of an office building. Even if the proposed system is used for a quarter of the ceiling area, the system can reduce the maximum cooling load by more than 10%.

Another finding from the results was that the cooling effect could be enhanced by the narrower channel width, but the negative effect of reduced air flow rate was strong in the case of the low air flow rate. As it was shown in Fig. 7, the channel width was dominant for the air flow rate induced by the solar chimney. Even though the M-Cycle performance was improved by the narrow channel width, the cooling effect of the system would be diminished due to the low air flow rate.

4.5. The hourly cooling performance

The convective cooling capacity of the system was predicted based on the hourly air flow rate shown in Fig. 7. Fig. 11 depicts the hourly variation of the convective cooling capacity in the case of no radiative cooling load, while Fig. 12 depicts that with the radiative cooling load of 30 W/m<sup>2</sup>.

When the radiative cooling load was not given to the system, the convective cooling capacity was the largest with the channel width of 20 mm during the daytime. While, the largest convective cooling capacity during the daytime was obtained by the channel width of 25 mm in the case of the radiative cooling load of 30 W/m<sup>2</sup>. The difference would be due to the effect of the enhanced air flow

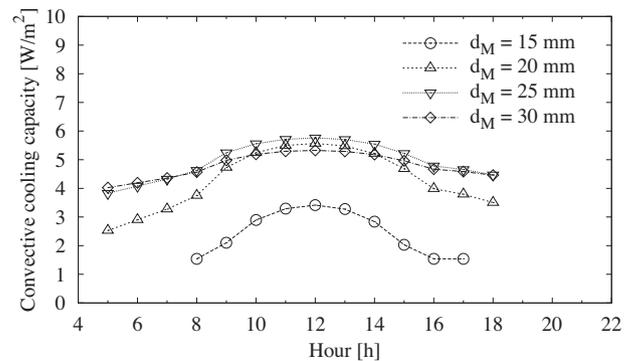


Fig. 12. The effects of the channel width on the hourly convective cooling capacity with the radiative cooling load of 30 W/m<sup>2</sup>.

rate on the improvement of the cooling capacity. As shown in Fig. 10, the effect of the air flow rate on the convective cooling capacity was more significant when the radiative cooling load was larger. Therefore, the optimum channel width was extended with larger radiative cooling load.

Assuming that the system has to cope with some radiative cooling load, the optimum configuration of the system would be the channel width of 25 mm. Up to the channel width of 25 mm, the increase of the air flow rate was dominant for the improvement of the cooling capacity. By the further increment of the channel width, the positive effect of the increased air flow rate would be overwhelmed by the deterioration of the cooling effect due to the wider channel width.

## 5. Conclusions

The study proposed the solar chimney driven evaporative cooling system that was integrated with the ceiling of a room. The cooling performance of the system was predicted by simulation of the solar chimney and the M-Cycle evaporative cooling channel. The main findings of the study are summarized as follows.

- The proposed system was feasible as a solar energy driven cooling system because the sufficient air flow to the M-Cycle evaporative cooling channel could be induced by the solar chimney when the solar radiation was available.
- The system was capable of coping with the radiative cooling load as much as 40–50 W/m<sup>2</sup> without a severe increase of the ceiling temperature. It was expected that the internal heat gains of buildings could be removed by the proposed system. Compared with the maximum cooling load of office buildings in Japan, it is expected that the system can reduce the maximum cooling load by more than 10% if a quarter of the ceiling area is replaced with the proposed system.
- The channel width of the M-Cycle was an influential parameter on the air flow rate of the solar chimney as well as the cooling effect of the M-Cycle. The optimum channel width to maximize the convective cooling capacity during the daytime would exist depending on the radiative cooling load, and it was 25 mm in the case of the radiative cooling load of 30 W/m<sup>2</sup>. The further increase of the channel width would result in the deterioration of the cooling capacity because the negative effect of the wider channel width would overwhelm the effect of the increased air flow rate.

The paper presented a concept of solar chimney driven evaporative cooling system, and the performance of the system was predicted by simulation. The verifications of the solar-induced air

flow rate and the cooling performance are necessary as a next step, which will be our future work.

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