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Passive Ventilation System Analysis using Solar Chimney in South of Algeria

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Abstract**—**The work presented in this study is related to an energy system analysis based on passive cooling system for dwellings. It consists to solar chimney energy performances determination versus geometrical and environmental considerations as the size and inlet width conditions of the chimney. Adrar site located in the southern region of Algeria is chosen for this study according to ambient temperature and solar irradiance technical data availability. Obtained results are related to the glazing temperature distributions, the chimney air flow and internal wall temperatures. The air room change per hour (ACH) parameter, the outlet air velocity and mass air flow rate are also determined. It is shown that the chimney width has a significant effect on energy performances compared to its entry size. A good agreement is observed between these results and those obtained by others from the literature.

Keywords**—**Solar chimney, Energy performances, Passive ventilation, Numerical simulation

I. Introduction

BUILDINGS consume energy specifically for heating, ventilating and air conditioning (HVAC) systems. The hot countries such as the Middle East regions and south of Algeria are very greedy of energy for air conditioning, despite their significant solar energy potential. The household electricity bill is very high in summer because of air conditioning uses where expenses are not within reach of every budget. The idea is to promote natural ventilation in dwellings by free convection. This passive cooling technique can be used to reduce mechanical air conditioning requirements in areas where cooling is a dominant problem. Solar chimney technology is used to produce hot air throughout by greenhouse effect. This air may be used for drying, power generation and other uses like air ventilation for dwellings.Climate change and global warming concerns by greenhouse effects due to the excessive use of fossil fuel, renewable energy systems can offer clean and interesting opportunities. Solar chimney technology is one of these systems. It is a natural power generator that uses solar radiation to increase the internal energy of flowing air through the system. The gain of useful solar energy is converted in kinetic flow that can be transformed in electrical energy through a turbine or an appropriate device for another use, such as ventilation by exhausting the warm humid air outside. A typical solar chimney device consists to a glazing system to raise the energy level of the air by greenhouse effect and a tower that circulates air by density gradient.Thermal comfort concern in dwellings is not novel. It has accompanied human civilisations throughout [1]. Bahadori [2] was the first author to introduce the idea of capturing the wind within a wind tower and then passing it through wetted conduit walls. Bansal et al. [3] can be considered as the pioneers to investigate solar chimney configurations and performances for ventilation in buildings. The authors presented a steady state mathematical model for a solar chimney which is used to enhance the effect of thermally induced ventilation in buildings. The model takes into consideration different sizes of the openings of a solar chimney with varying values of the discharge coefficients.Significant research works have been done on solar chimneys these decades both theoretically and experimentally [4]- [11]. These researches have been conducted on using natural ventilation for different applications including passive solar heating and cooling of buildings, ventilation, and power generation.The purpose of this present work is related to the study of energy performance of solar chimneys versus geometrical parameters and environmental sites. Obtained results are related to the glazing temperature distributions, the chimney air flow and internal wall temperature distribution along the chimney, the mass flow rate and air change per hour (ACH).

II. Physical and Mathematical Models

A. Physical Model

Figure 1 shows the schematic representation of the passive system using solar chimney device. An opening at the bottom of the wall allows room air. The inner surface of the wall is painted black to increase solar radiation absorption. The solar radiation effect on the glazing and absorber generates the air flow by driving force in the chimney. This driving force which controls the airflow rate is the thermal buoyancy effect. Figure 2 gives the physical model used for the solar chimney.

Air enters the chimney at the inlet temperature, $T_{f,i}$, which is assumed equal to the room air, T_r. Warm air exits at the outlet temperature, $T_{f,o}$, from the top of the chimney. Temperatures at the surfaces of the glass, T_g , and wall, T_w , and mean air temperature in the flow channel, T_f, are all assumed to be uniform.

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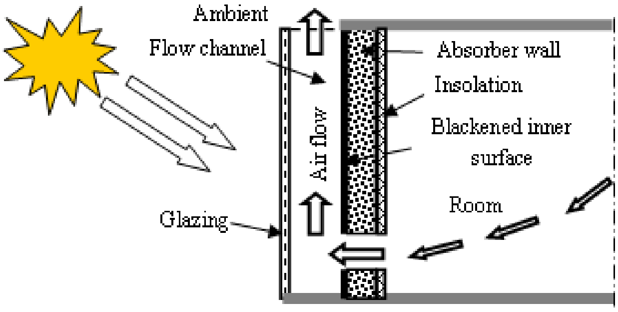

Fig. 1 Schematic representation of the passive ventilation using solar chimney

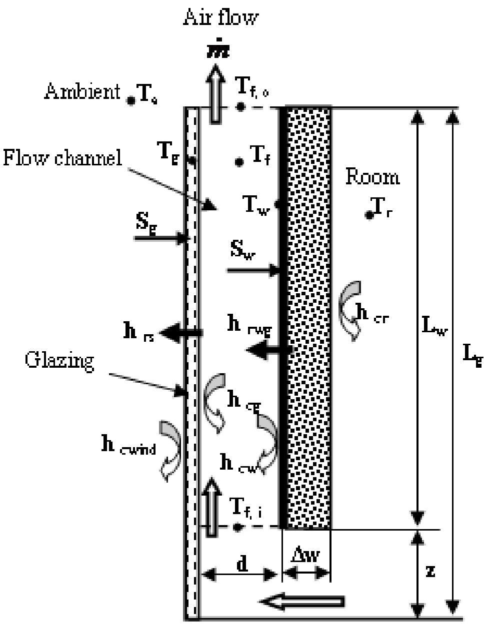

Fig. 2 Physical model used for the solar chimney

B. Mathematical Model

The mathematical modelling of the problem is based on heat balance equations at the glazing, the absorber and along the flow channel. Heat balance at the glazing is given by [5]:

$$S_g A_g + U_t A_g (T_a - T_g) = h_{cg} A_g (T_g - T_f) + h_{rwg} A_w (T_g - T_w) \tag{1}$$

Where: U_t is the overall heat transfer coefficient between air and glazing given by

$$U_t = h_{cwind} + h_{rs} \tag{2}$$

Convective heat transfer coefficient, h_{cwind}, is defined by [1]

$$h_{cwind} = 2.8 + 3.V_a \tag{3}$$

Radiative heat transfer coefficient, h_{rs}, between glazing and sky is given by:

$$h_{rs} = \frac{\sigma \varepsilon_g (T_g + T_s)(T_g^2 + T_s^2)(T_g - T_s)}{(T_g - T_a)} \tag{4}$$

With : $T_s = 0.0552\, T_a^{1.5}$

Solar radiation heat flux absorbed by glazing is

$$Sg = \alpha_g H \tag{5}$$

Radiative heat transfer coefficient, h_{rws}, between glazing and wall is given by:

$$h_{rwg} = \frac{\sigma (T_g^2 + T_w^2)(T_g - T_w)}{\left(\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_w} - 1\right)} \tag{6}$$

Convective heat transfer coefficient, h_{cg}, is defined by [1]

$$h_{cg} = \frac{Nu.K_f}{Lg} \tag{7}$$

With : $Nu = 0.60(Gr.\Pr.\cos\theta)^{0.2}$, $Gr = g\beta_f \Delta T L_g^3 / \nu_f^2$

$K_f = 0.00263 + 0.000074(T_f - 300)$

$\mu_f = [1.846 + 0.00472.(T_f - 300)].10^{-5}$

$\rho_f = 1.1614 - 0.00353(T_f - 300)$, $\beta_f = 1/T_f$

Heat balance on the air flow channel is given by [5]:

$$q'' = h_{cw} A_w (T_w - T_f) + h_{cg} A_g (T_g - T_f) \tag{8}$$

With:
$q'' = \dot{m} C_{p,a} (T_{f,0} - T_{f,i})$, $C_{p,a} = [1.007 + 0.00004(T_f - 300)].10^3$.

The mean air temperature in the flow channel is given by

$$T_f = \gamma T_{f,0} + (1 - \gamma) T_{f,0} \tag{9}$$

The coefficient, γ, is determined experimentally [5] and the temperature $T_{f,i}$ is assumed equal to the room air, T_r , as it is specified previously. Then

$$q'' = \frac{\dot{m} C_{p,a} (T_f - T_r)}{\gamma} \tag{10}$$

By substituting equation (10) into (8), then

$$h_{cg}A_gT_g - \left(\frac{h_{cg}A_g + h_{cw}A_w + \dot{m}C_{p,a}}{\gamma}\right)T_f + h_{cw}A_wT_w = -\left(\frac{\dot{m}C_{p,a}}{\gamma}\right)T_r \quad (11)$$

Convective heat transfer coefficient, h_{cw}, is defined by [1]

$$h_{cw} = \frac{Nu.K_f}{L_W} \quad (12)$$

The mass flow rate is given by

$$\dot{m} = V_{f,0}\rho_{f,0}A_0 = C_d \frac{\rho_{f,0}A_0}{\sqrt{\left(1+\frac{A_0}{A_i}\right)}}\sqrt{\frac{2gL_w(T_f - T_r)}{T_r}} \quad (13)$$

The air change per hour (ACH) is defined by

$$ACH = \frac{Q_V * 3600}{Room\ \ Total\ \ volume} = \frac{\frac{\dot{m}}{\rho_{f,0}}.3600}{Room\ \ Total\ \ volume} \quad (14)$$

The instantaneous efficiency of the chimney, η_i, can then be deduced as

$$\eta_i = \frac{\dot{m}C_{p,a}(T_{f,0} - T_{f,i})}{WL_gH} \times 100\% \quad (15)$$

Heat balance on the wall absorber is given by [5]:

$$-h_{rwg}A_wT_g - h_{cw}A_wT_f + (h_{cw}A_w + h_{rwg}A_w + U_bA_w)T_w = S_wA_w + U_bA_wT_r \quad (16)$$

Where: U_b is the overall heat transfer coefficient between insulation panel and room given by

$$U_b = \frac{K_{ins}}{\Delta w_{ins}} \quad (17)$$

Solar radiation heat flux absorbed by the absorber wall is

$$S_w = \tau\alpha_w H \quad (18)$$

III. Results And Discussion

Heat balance equations at the glazing, the absorber and along the flow channel, given by equations (1), (11) and (16) were solved numerically using Gauss- Seidel iterative method with a relaxation factor in Fortran [12]. Temperature distributions, T_g, T_f and T_w related to glazing, fluid flow and absorber wall are then determined.

Technical data related to the geometrical prototype of the chimney used for the numerical simulation are those of [5].

Figures 3 and 4 show the comparison between the results obtained by the program developed in this study and those of [5] at the same operating conditions. As it is shown, good agreement is observed between these results which validate the numerical results obtained by the program.

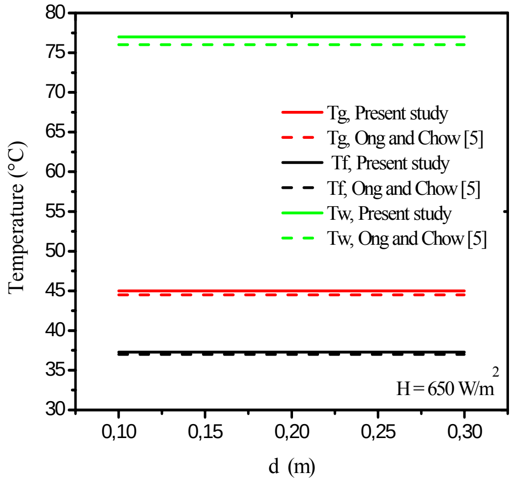

Fig. 3 Glazing, fluid flow and absorber wall mean temperature distributions versus chimney width. Results comparison

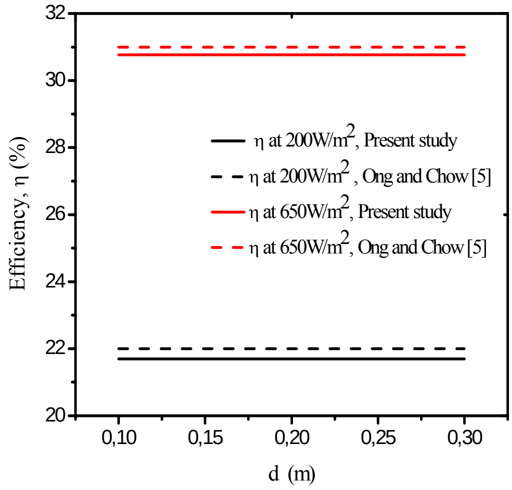

Fig. 4 Instantaneous solar chimney efficiency distributions versus chimney width. Results comparison

Figure 5 gives the monthly evolution of average solar radiation and instantaneous solar chimney efficiency of Adrar site. We can note that the minimum of radiation is in December month with a value of approximately 219 W/m² and the best solar radiation is in June with a value of approximately 588 W/m². The corresponding instantaneous solar chimney efficiency varies between 15 and approximately 40%.

Figure 6 shows the monthly mean temperature evolution of the glazing, the fluid flow and the absorber wall. We must underline that the mean absorber wall temperature is higher than that of the fluid flow and glazing, which is predictable, because thermal radiation absorption induces an increase in absorber wall temperature which affect the air flow in contact with these walls.

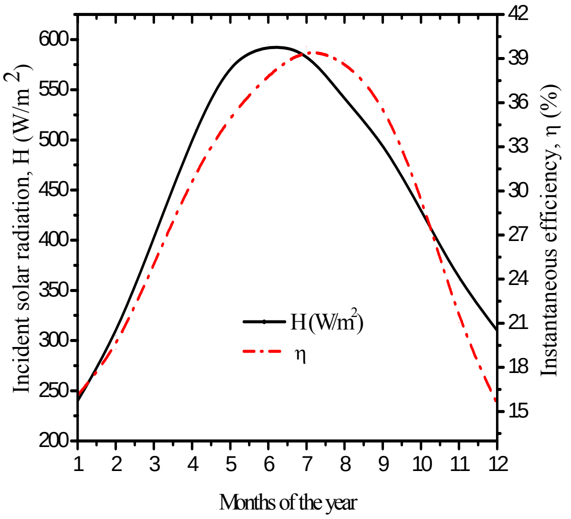

Fig. 5 Monthly evolution of average solar radiation and Instantaneous solar chimney efficiency

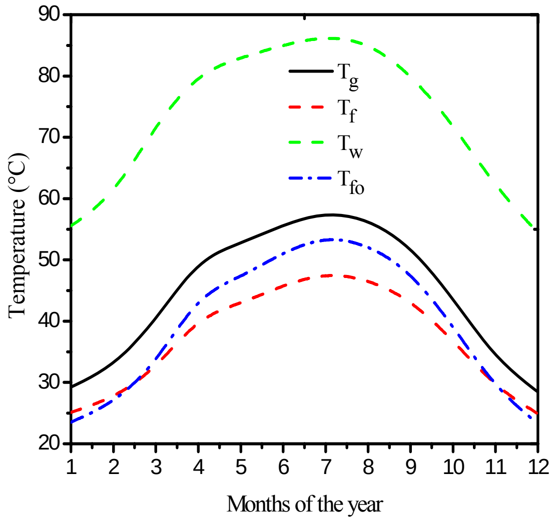

Fig. 6 Monthly mean temperature evolution of the glazing, the fluid flow and the absorber wall

Figures 7 and 8 illustrate the monthly evolution of the air change per hour (ACH) and mass flow rate parameters for different values of z/Lg and for a fixed value of d/W. As it is shown in these figures, their maximum corresponds to the maximum value of incident solar radiation. These parameters increase with the flow rate increasing following the absorber wall aperture size and no difference between curves can be observed when z/Lg trends to be equal to 0.3. This result is also confirmed by Bassiouny and Nader [8].

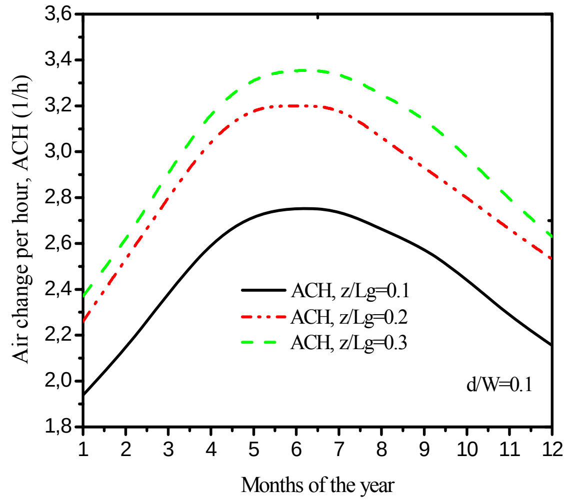

Fig. 7 Monthly evolution of air change per hour for different values of z/Lg and for : d/W= 0.1

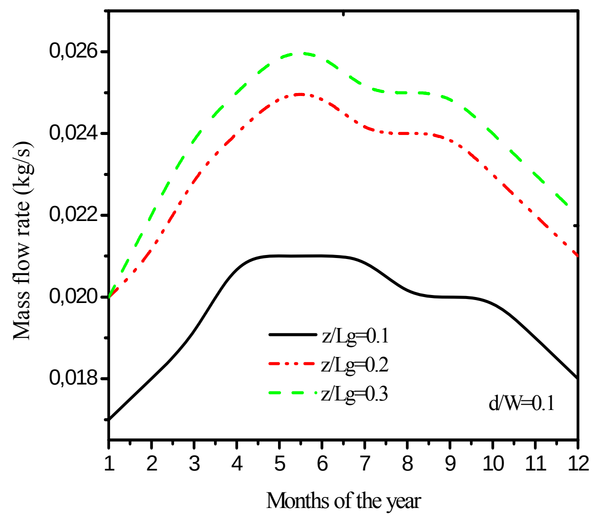

Fig. 8 Monthly evolution of mass flow rate for different values of z/Lg and for : d/W= 0.1

IV. Conclusion

The work presented in this study is related to an energy system analysis based on passive cooling system for dwellings. It consists to solar chimney energy performances determination versus geometrical and environmental considerations. Adrar site located in the southern region of Algeria is chosen for this study according to ambient temperature and solar irradiance technical data availability.

The glazing temperature distributions, the chimney mass flow rate, the internal wall temperatures and the air room change per hour (ACH) parameter are presented and discussed.

It s shown that the influence of the incident solar radiation is the important parameter on energy performances analysis of the chimney and an optimum design between the width of the chimney and the aperture of the absorber wall may exist for increasing ACH parameter.

A good agreement is observed between the results obtained in this study and those of the literature [5, 8].

APPENDIX

U	Overall heat transfer coefficient, W. $m^{-2}.K^{-1}$
H	Incident solar radiation, W. m^{-2}
A	Area, m^2
h	Heat transfer coefficient, W.$m^{-2}.K^{-1}$
S	Solar radiation heat flux, W. m^{-2}
V	Velocity, m.s^{-1}
T	Temperature, °C
C_p	Specific heat, J.$kg^{-1}.K^{-1}$
C_d	Discharge coefficient = 0.57
$\dot{m}$	Mass flow rate, kg.s^{-1}
K	Thermal conductivity, W.$m^{-1}.K^{-1}$
d	Chimney width, m
L	Length, m
ACH	Air change per hour, h^{-1}
Nu	Nusselt number
Pr	Prandtl number
Gr	Grashof number

Greek symbols

ρ	Density, kg.m^{-3}
μ	Dynamic viscosity, kg.$m^{-1}.s^{-1}$
η	Instantaneous efficiency of solar chimney, %
ε	Emissivity
α	Absorptivity
τ	Transmitivity

Subscripts

o	Outlet to the air channel
i	Inlet to the air channel
r	Room
a	Ambient air
g	Glass
f	Fluid
s	Sky
w	Wall
wind	Wind
ins	Insulation
c	Convection
rs	Radiative between glass and sky
rwg	Radiative between wall and glass

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