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Theoretical Study of Heat Transfer through a Sun Space Filled with a Porous Medium

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Abstract

A theoretical study had been conducted to detect the effect of using a porous medium in sunspace to reduce heating load and overcoming coldness of winter in the cold regions. In this work, the heat transferred and stored in the storage wall was investigated. The mathematical model was unsteady, heat conduction equation with nonlinear boundary conditions was solved by using finite difference method and the solution technique of heat conduction had based on the Crank Nicholson method. The results had adopted on the aspect ratio (H/L=30), Darcy number ($Da=10^{-3}$), porosity ($\varphi=0.35$) and particle to fluid thermal conductivity ratio ($k_p/k_f=38.5$). The results showed that using the porous medium had enhanced the heat transferred and stored in the storage wall. For the outside storage wall temperature, an increase of 19.7% was achieved by using the porous medium instead of the air, while it was 20.3% for the inside storage wall temperature.

Keywords: Sun Space, Rectangular Enclosure, Porous Medium, Free Convection.

دراسة نظرية لانتقال الحرارة خلال حيز شمسي مملوء بوسط مسامي

الخلاصة

أُجريت دراسة نظرية لتحري تأثير إستخدام وسط مسامي في حيز شمسي لتقليل حمل التدفئة والتغلب على برودة الشتاء في المناطق الباردة. في هذا البحث، تم إستقصاء الحرارة المنتقلة والمخزونة في جدار الخزن. النموذج الرياضي كان غير مستقر، معادلة التوصيل الحراري ذات الشروط الحدية الغير خطية حُلت باستخدام طريقة الفروقات المحددة وإستندت تقنية الحل للتوصيل الحراري على طريقة (Crank Nicholson). إعتمدت النتائج على النسبة الباعية (H/L=30)، عدد دارسي ($Da=10^{-3}$)، المسامية ($\varphi=0.35$) ونسبة الموصلية الحرارية للجزيئات إلى المائع ($k_p/k_f=38.5$). أظهرت النتائج أن إستخدام الوسط المسامي حسن من الحرارة المنتقلة والمخزونة في جدار الخزن. بالنسبة لدرجة حرارة سطح جدار الخزن الخارجي تم الحصول على زيادة مقدارها (19.7%) عند استخدام الوسط المسامي بدلاً من الهواء، بينما كانت (20.3%) لدرجة حرارة سطح جدار الخزن الداخلي.

الكلمات الدالة: حيز شمسي، حيز مستطيل، وسط مسامي، حمل حر.

Nomenclature

C_P Specific heat at constant press. (J/kg K)
Da Darcy number (-)
d_p Diameter of porous particles (m)
g Acceleration of gravity (m/s^2)
H Height of the wall (m)
h Heat transfer coefficient (W/m^2 K)
hs Heat storage rate per unit area (W/m^2)
I Intensity of solar radiation per unit area (W/m^2)
K Permeability of the porous medium (m^2)
k Thermal conductivity (W/m K)
L Width of the sunspace (m)
Nu Nusselt number (-)
n Time step (s)
q Rate of heat per unit area (W/m^2)
Ra Rayleigh number (-)
T Temperature (K)
t Time(s)
t_g Glass thickness (m)
t_w Wall thickness (m)
V_w Wind speed (m/sec)
x x-Space coordinate in Cartesian system (m)

Greek Symbols

α Absorptivity (-), or Thermal Diffusivity (m^2/s)
β Volume coefficient of expansion (K^{-1})
ε Emissivity(-)
λ Thermal conductivity ratio (-)
ν Kinematic viscosity (m^2/s)
ρ Reflectivity (-), or Density (kg/m^3)
σ Stefan-Boltzmann constant (W/m^2.K^4)
τ Transmissivity(-)
φ Porosity (-)

Subscripts

a Ambient
e Effective
f Air at film temperature
g Glass
i Inside the room
m Medium
N Last node
o Outside the room
p Porous particles
r Room
w Wall

Introduction

The waning of the fossil fuel resources besides increasing the oil prices has led to looking for other resources of energy. The solar energy, i.e., the renewable energy, represented a suitable resource since it is sustainable, free, clean energy and available everywhere in the earth.

Utilizing passive solar energy in heating had attracted a lot of attention because it provides auxiliary cheap clean energy. The sunspace is one effective way used to heat the rooms and reduction the large oscillations in the room air temperature [1].

Sitar and Krajnc [2], had suggested two scenarios of technological renewal. The first one was a classical one using additional thermal insulation of the building envelope and fitting of new structural elements such as windows, doors, balconies and windbreaks. The second scenario, however, included the sunspace construction used as a new passive solar structural element, modifying the envelope. The energy efficiency of the suggested scenarios were calculated according to the procedures, given in the EN832 standard, considering the attached sunspace as an integral part of the building in the first case and a passive solar object adjacent to the thermal envelope of the building in the second case. The results showed that the last case had yielded the most energy efficient renewal of the existing residential building. Oliveti,et al.[3], had proposed a simplified model for the evaluation of the solar energy absorbed by a sunspace using an effective absorption coefficient defined as the ratio of the solar energy absorbed by the internal surfaces of the room and the solar energy entering through the glazed surfaces. Reference was made to simple geometries constituting volumes with glass surfaces of various sizes, number and orientation. In total, ten different geometrical configurations were considered. An extensive parametric analysis was carried out by means of a simulation code with the aim of highlighting the range of variability of the absorption coefficient and the most significant variables on which it depends: the latitude, geometrical characteristics of the volume, type of glass, exposure and the optical properties of the opaque surfaces. The results obtained showed that the absorption phenomenon can be extremely adequately parameterized using the simplified models that were proposed.

Batainеha and Fayez [4], had investigated the thermal performance of a sunspace attached to a living room located in Amman–Jordan. DEROB-LTH was used to estimate the thermal performance in terms of cooling and heating loads required for the indoor climate. The annual heating and cooling loads were obtained under climatological prevalent conditions. The main contribution of this passive solar design was to reduce heating loads in winter and minimize overheating during the summer period. Six configurations that differ by the ratio of glazed surface area to opaque surfaces area were studied. The effect of orientation of sunspace, opaque wall and floor absorptivity coefficients, and number of glass layers on the thermal performance was evaluated. The results showed that the sunspace reduced the heating load during the winter while it created a serious overheating problem during the summer. The contribution of reducing heating requirement increased with increase of the ratio of glazed surface area to opaque surface area. Also, the optimal contribution obtained when sunspace oriented to the south. Two passive cooling techniques were proposed and evaluated to overcome the summer overheating problem, as well as a passive heating technique was proposed to minimize the thermal losses during winter nighttime. Internal shading and night ventilation successfully minimized the overheating problem. The results showed that when single clear glass sunspace oriented to the south as high as 42% reductions in annual heating and cooling load can be achieved.

Guanghua et al. [5], had analyzed the effectiveness of warming in a classroom with attached sunspace based on solar radiation received at unit area. It was obvious that the attached sunspace has a possibility to receive the effective solar radiation of average 41.6MJ/m^2.day and achieved indoor warming of daily average 13 ^{o}C in heating period.

Bataineha and Fayez [6], had predicted through simulations that the sunspace could be significant contribute to the reduction of heating load during the winter while creating a serious overheating problem during the summer. It was found that the amount of reduction of the heating load increased with increase of the ratio of the glazed surface area to opaque surface area. The maximum reduction of heating load obtained when the sunspace was oriented to the south. Besides, increasing the absorption coefficient had increased the reduction of the heating load. Also, using a double glazed window instead of a single glass improved the thermal behavior of the sunspace.

Oliveti,et al. [7], had evaluated the solar heat gains obtainable from attached sunspaces to air-conditioned rooms by means of the solution to the optical problem of incident solar radiation absorption through the windows and of the temperature field in the shell separating the sunspace from outdoors and adjacent spaces. The effective absorption coefficient of the sunspace was used for these evaluations, as well as the ratio of the absorbed energy of the internal surfaces to the solar energy entering, and utilization the factor of solar contributions that represent the fraction of the absorbed energy supplied to the indoor air. With reference to a pre-established geometry and a system of windows made up of clear double-glazing, the solar gains of the sunspace and the adjacent spaces were calculated for some Italian localities at variation of exposure, optical properties and thermal capacity of the opaque surfaces, the amount of ventilation and shading of the device. Finally, they determined the operative temperature to estimate the comfort acceptability conditions in the sunspace.

In this paper, a theoretical study was done to investigate the influence of filling a sunspace with a transparent porous medium (glass balls) instead of air on the storage wall temperature and the heat stored in it and transferred through it.

Mathematical Model

Energy Balances

In this work, two cases were dealt with a gap fill with a porous medium (glass balls), and an air gap. This problem is a one-dimensional heat flow and homogenous layers with assuming constant properties to formulate the mathematical equations.

The heat transfer through the glazing was considered at unsteady condition, all the surfaces are "gray bodies" with diffuse reflection and emission, considering the air as a non-participating medium in radiation heat exchange, the dwelling air temperature was thermostatically controlled and constant, and the sunspace floor and ceiling were well insulated, so there were no lateral losses.

The incident radiation on the glass (I) was assumed to be isotropic. The amount of solar energy absorbed by the glass layer was taken as ($\alpha_g I$), the reflected energy as

($\rho_g I$), and transmitted energy as ($\tau_g I$). Hence the energy balances for the glass layer shown in Figure (1), can be analyzed as:

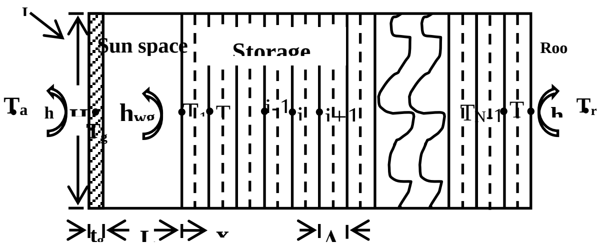

Fig. 1. The energy balance for the sun space

$$\alpha_g I + h_{wg}\left[T_1^n - T_g^n\right] + \alpha_g.(\rho_w + \rho_p).\left[\tau_g I\right] = h_o\left[T_g^n - T_a^n\right] + \sigma\varepsilon_g\left[T_g^{n^4} - T_a^{n^4}\right] + \rho_g C_{Pg} t_g \left[\frac{T_g^{n+1}-T_g^n}{\Delta t}\right] \quad \text{.......(1)}$$

This energy balance was for the porous medium case, and for the air case, the energy balance can be analyzed as:

$$\alpha_g I + h_{wg}\left[T_1^n - T_g^n\right] + \frac{\sigma}{\frac{1}{\epsilon_g}+\frac{1}{\epsilon_{wo}}-1}\left[T_1^{n^4} - T_g^{n^4}\right] + \alpha_g.\rho_w.\left[\tau_g I\right] = h_o\left[T_g^n - T_a^n\right] + \sigma\varepsilon_g\left[T_g^{n^4} - T_a^{n^4}\right] + \rho_g C_{Pg} t_g \left[\frac{T_g^{n+1}-T_g^n}{\Delta t}\right] \quad \text{.......(2)}$$

The energy equation, which is in one dimension through the wall layer, shown in Figure (1) can be written as:

$$\frac{\partial T}{\partial t} = \alpha_w \frac{\partial^2 T}{\partial x^2} \quad \text{.......(3)}$$

For the outside wall surface, the amount of solar energy absorbed was taken as α_w ($\tau_g I$) and the reflected energy as ρ_w ($\tau_g I$) which is absorbed again by the glass layer. The energy balance for this layer of the wall in the porous medium case was given by:

$$\alpha_w.\left[\tau_g I\right] = h_{wg}\left[T_1^n - T_g^n\right] + \frac{k_w}{\Delta x}\left[T_1^n - T_2^n\right] + \rho_w C_{Pw}\frac{\Delta x}{2}\left[\frac{T_1^{(n+1)}-T_1^n}{\Delta t}\right] \quad \text{.......(4)}$$

and for the air case:

$$\alpha_w.\left[\tau_g I\right] = h_{wg}\left[T_1^n - T_g^n\right] + \frac{\sigma}{\frac{1}{\epsilon_g}+\frac{1}{\epsilon_{wo}}-1}\left[T_1^{n^4} - T_g^{n^4}\right] + \frac{k_w}{\Delta x}\left[T_1^n - T_2^n\right] + \rho_w C_{Pw}\frac{\Delta x}{2}\left[\frac{T_1^{(n+1)}-T_1^n}{\Delta t}\right] \quad \text{...(5)}$$

For the inside wall surface which is illustrated in Figure (1), the energy balance for both cases is:

$$k_w\left[\frac{T_{N-1}^n - T_N^n}{\Delta x}\right] = h_i(T_N^n - T_r^n) + \sigma\varepsilon_{wi}(T_N^{n^4} - T_r^{n^4}) + \rho_w C_{Pw}\frac{\Delta x}{2}\left[\frac{T_N^{(n+1)}-T_N^n}{\Delta t}\right] \quad \text{.......(6)}$$

where T_{N-1} and T_N are the temperatures of the pre and the last node.

The solution procedure is achieved by solving Equations (1) and (2) for (T_g^{n+1}), Equations (4) and (5) for (T_1^{n+1}), and Equation (6) for (T_N^{n+1}).

The heat transferred to the storage wall per unit area (q_w) for the porous case is:

$$q_w = \alpha_w\left[\tau_g I\right] - h_{wg}\left[T_1^{(n+1)} - T_g^{(n+1)}\right] \quad \text{.......(7)}$$

and for the air case (q_w) is:

$$q_w = \alpha_w\left[\tau_g I\right] - h_{wg}\left[T_1^{(n+1)} - T_g^{(n+1)}\right] - \frac{\sigma}{\frac{1}{\epsilon_g}+\frac{1}{\epsilon_{wo}}-1}\left[T_1^{(n+1)^4} - T_g^{(n+1)^4}\right] \quad \text{......(8)}$$

The heat transfer to the room per unit area (q_r) is calculated from:

$$q_r = h_i\left[T_N^{(n+1)} - T_r^{(n+1)}\right] + \sigma\varepsilon_{wi}\left[T_N^{(n+1)^4} - T_r^{(n+1)^4}\right] \quad \text{.......(9)}$$

The rate of heat storage per unit area (hs) inside the wall is computed from the difference between the incident energy on the outside wall surface (q_w) and the heat transfer (useful energy) to the room (q_r), as follows:

$$hs = q_w - q_r \quad \text{.......(10)}$$

Convection Heat Transfer Coefficients

For the heat loss calculations from the glass exposed to wind, the heat transfer coefficient (h_o) in (W/m^2.K) is found by Frank and Kreider [8]:

$$h_o = 5.7 + 3.8\,V_w \quad \text{.......(11)}$$

For free convection heat transfer through the porous enclosure in the gap between the wall and glass, the recommended relation for heat transfer coefficient (h_{wg}) is given by Ali, *et al*. [9]:

$$Nu = \frac{Ra_m^{\frac{0.43078}{0.15773\left(\frac{H}{L}\right)}} \varphi^{0.1887\left(\frac{H}{L}\right)^{0.28584}}}{\left(\frac{k_p}{k_f}\right)^{0.02684\left(\frac{H}{L}\right)^{0.21648}} Da^{0.0572\left(\frac{H}{L}\right)^{0.17416}}} \qquad (12)$$

The above equation valid for medium Rayleigh number ($Ra_m = \frac{g\beta_f(T_1-T_g)KL}{\alpha_e \nu_f}$) at a range of ($5 \le Ra_m \le 2000$), porosity ($\varphi$=0.35, 0.45, 0.55), particle to air-which its thermal conductivity is 0.026W/mK thermal conductivity ratio (k_p/k_f=5.77, 38.5, 1385.5), Darcy number (Da=10^{-3}, 10^{-4}, 10^{-5}), and aspect ratio-the ratio between the height of the wall (H) and the width of the sunspace (L)- of a range ($5 \le H/L \le 30$).

For a packed-sphere bed, the permeability (K) is:

$$K = \frac{d_p^2 \varphi^3}{150\,(1-\varphi)^2} \qquad (13)$$

The effective thermal diffusivity of the saturated porous medium (α_e) and the Darcy number (Da) are defined as:

$$\alpha_e = \frac{k_e}{(\rho C_P)_f} \qquad (14)$$

$$Da = \frac{K}{L^2} \qquad (15)$$

Where the effective thermal conductivity of the saturated porous medium (k_e) is obtained by Kaviany [10]:

$$\frac{k_e}{k_f} = 1 - \sqrt{(1-\varphi)} + \frac{2\sqrt{(1-\varphi)}}{1-\lambda B}\left[\frac{(1-\lambda)B}{(1-\lambda B)^2}\ln\left(\frac{1}{\lambda B}\right) - \frac{B+1}{2} - \frac{B-1}{1-\lambda B}\right] \qquad (16)$$

$B=1.25[(1-\varphi)/\varphi]^{10/9}$

$\lambda = k_f/k$

$$Nu = \frac{h_{wg} L}{k_e}$$

While for the air gap case (Nu) can be found by Zrikem and Bilgen [11]:

$$Nu = \max\left\{\begin{matrix} 1 \\ 0.288\left(\frac{Ra}{H/L}\right)^{0.25} \\ 0.039 Ra^{0.333} \end{matrix}\right\} \qquad (17)$$

where

$$Nu = \frac{h_{wg} L}{k_f},$$

$$Ra = \frac{g\beta_f (T_1 - T_g) L^3}{\nu_f \alpha_f}$$

Also, k_f and α_f are the thermal conductivity, the effective thermal diffusivity of air at film temperature, respectively.

For free convection between the inside storage wall surface and the room, the value of the average heat transfer coefficient (h_i) is found by Holman [12]:

$$Nu = \left\{\begin{matrix} 0.59\, Ra_H^{0.25} & 10^4 < Ra_H < 10^9 \\ 0.10\, Ra_H^{0.33} & 10^9 < Ra_H < 10^{12} \end{matrix}\right\} \qquad (18)$$

where Nusselt number (Nu) is given by (h_i.H/k_f), and Rayleigh number (Ra_H) is given by $\frac{g\beta_f(T_N - T_r)H^3}{\nu_f \alpha_f}$.

Solution Methods

Taking the passive solar heating from the view of the instantaneous change in energy incidence, the unsteady heat conduction equation with nonlinear boundary conditions was solved using finite difference method.

Representing the second order derivative in Equation (3), in the following form:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{2}\left[\frac{\partial^2 T}{\partial x^2}\right]_{n+1} + \frac{1}{2}\left[\frac{\partial^2 T}{\partial x^2}\right]_n \qquad (19)$$

From Equations (3) and (19), it can be get:

$$\frac{\partial T}{\partial t} = \frac{1}{2}\alpha_w\left[\left(\frac{\partial^2 T}{\partial x^2}\right)_{n+1} + \left(\frac{\partial^2 T}{\partial x^2}\right)_n\right] \qquad (20)$$

Expressing the first order time derivative of equation (20) in forward difference scheme gives:

$$\frac{\partial T}{\partial t} = \frac{\left[T_i^{(n+1)} - T_i^n\right]}{\Delta t} \qquad (21)$$

The second order derivatives in the central difference scheme are:

$$\left(\frac{\partial^2 T}{\partial x^2}\right)_n = \frac{[T_{i-1}^n - 2T_i^n + T_{i+1}^n]}{(\Delta x)^2} \qquad (22)$$

and:

$$\left(\frac{\partial^2 T}{\partial x^2}\right)_{n+1} = \frac{\left[T_{i-1}^{(n+1)} - 2T_i^{(n+1)} + T_{i+1}^{(n+1)}\right]}{\Delta x^2} \qquad (23)$$

Substituting Equations (21), (22) and (23) into Equation (20) and simplifying to get:

$$-\Lambda T_{i-1,n+1} + 2(1+\Lambda)T_{i,n+1} - \Lambda T_{i+1,n+1} = \Lambda T_{i-1,n} + 2(1-\Lambda)T_{i,n} + \Lambda T_{i+1,n} \qquad (24)$$

where:

$$\Lambda = \frac{\alpha_w . \Delta t}{\Delta x^2} \quad \text{……………………………..(25)}$$

Equation (25) was applied to the internal nodes (from 2 to N-1) to give a system of simultaneous equations for T_2^{n+1}, T_3^{n+1}….. T_{N-1}^{n+1}, that were solved by subroutine of tridiagonal system of equations by gauss elimination technique. Equation (24) was solved by the Crank Nicholson method. The above solution has the powerful property of being that the solution is always stable for any value of (Λ).

Results and Discussions

The using of a porous medium instead of air in a sunspace was investigated theoretically. The glass balls were used as a porous medium, and the storage wall was made of concrete. The recorded data of solar radiation, ambient temperature and wind speed by the Rain Wise Portable Weather Logger (Portlog) in Tikrit-Iraq (43º 38̀ E and 34º 39̀ N) were used from (February 8, 2012) to (February 21, 2012).

Figure (2) shows the temperature of the outside storage wall (sunspace side). As general behavior, the temperature was low before sunrise then increases during the shining hours and after sunset the temperature decreased again. This behavior is expected since the sunspace work depends on solar radiation.

When filling the sunspace gap with a porous medium, amount of the heat transferred to the wall increased because with presence the porous medium the conductance was achieved which increased the heat transfer through the gap and consequently increased the temperature of the outside storage wall comparing with the air case. The maximum temperature difference between the porous medium case and the air case was noticed in (February 10, 2012) at (16:11) where the difference was 9.4ºC while the minimum difference was 5.5ºC in (February 16, 2012) at (9:56).

Figure (3) shows the inside storage wall temperature (room side). The temperature was low, before and about, two hours after sun rise. This happened because the heat needs some time to conduct through the wall. Also, it was observed that the decreasing in the inside storage wall temperature has not immediately occurred after sunset, but it has delayed about two hours after sunset which decreases the need for auxiliary heating during nights.

Furthermore, in all days, the inside storage wall temperature did not fall below 20ºC for the porous medium case and in most days for the air case.

As well, using the porous medium had the advantage with a maximum difference 4.7ºC in (February 12, 2012) at (8:58) and minimum difference 3.1ºC in (February 21, 2012) at (13:57).

Figure (4) shows the heat transferred through the sunspace to the storage wall. In the air case, the radiation reflected by the outside storage wall went back to the glass and lost to the ambient while, in the porous medium case, it was absorbed by the porous medium. This means that more heat was kept inside the sunspace then this heat transferred again to the storage wall, leading to increase the heat transferred to the storage wall which decreases the power needed to heat the room.

For all days, except (9 and 19), significant fluctuations were observed during the day because of the clouds that covered the sky which affected on the solar radiation.

Figure (5) shows the heat stored by the storage wall, since the ambient temperature during the night was low compared with it in the daytime, the heat losses from the storage wall through the gap during the night was greater and consequently the heat stored in the storage wall was low, but in the day with the sun shining those losses became lower and the heat stored in the storage wall become higher, with presence of the porous medium more heat was stored comparing with the air case. The stored heat can be increased by covering the glass during the night and the cloudy days.

Figure (5) shows the heat stored by the storage wall, since the ambient temperature during the night was low compared with it in the daytime, the heat losses from the storage wall through the gap during the night was greater and consequently the heat stored in the storage wall was low, but in the day with the sun shining those losses became lower and the heat stored in the storage wall become higher, with presence of the porous medium more heat was stored comparing with the air case. The stored heat can be increased by covering the glass during the night and the cloudy days.

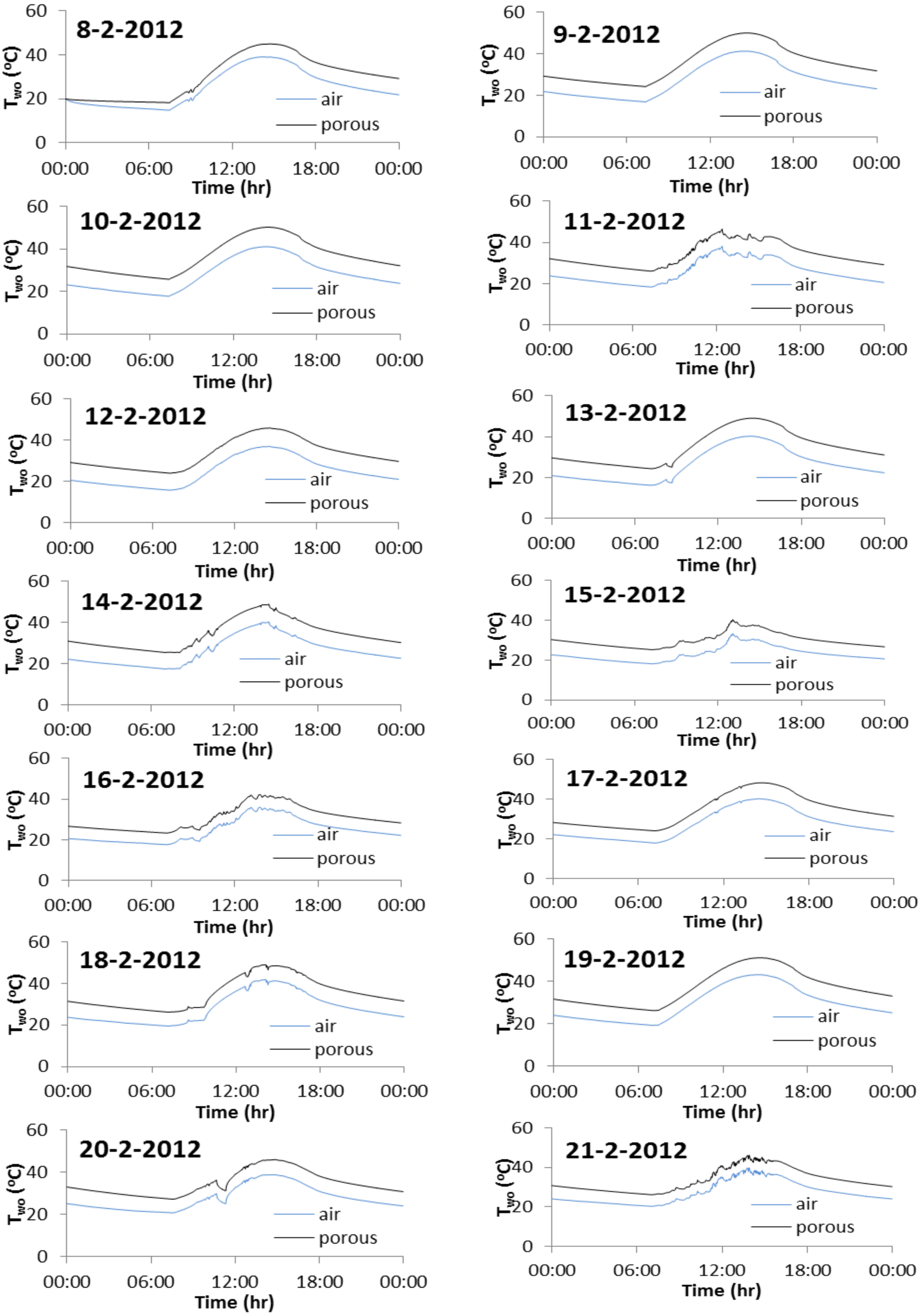

Fig. 2. The outside storage wall temperature along the day

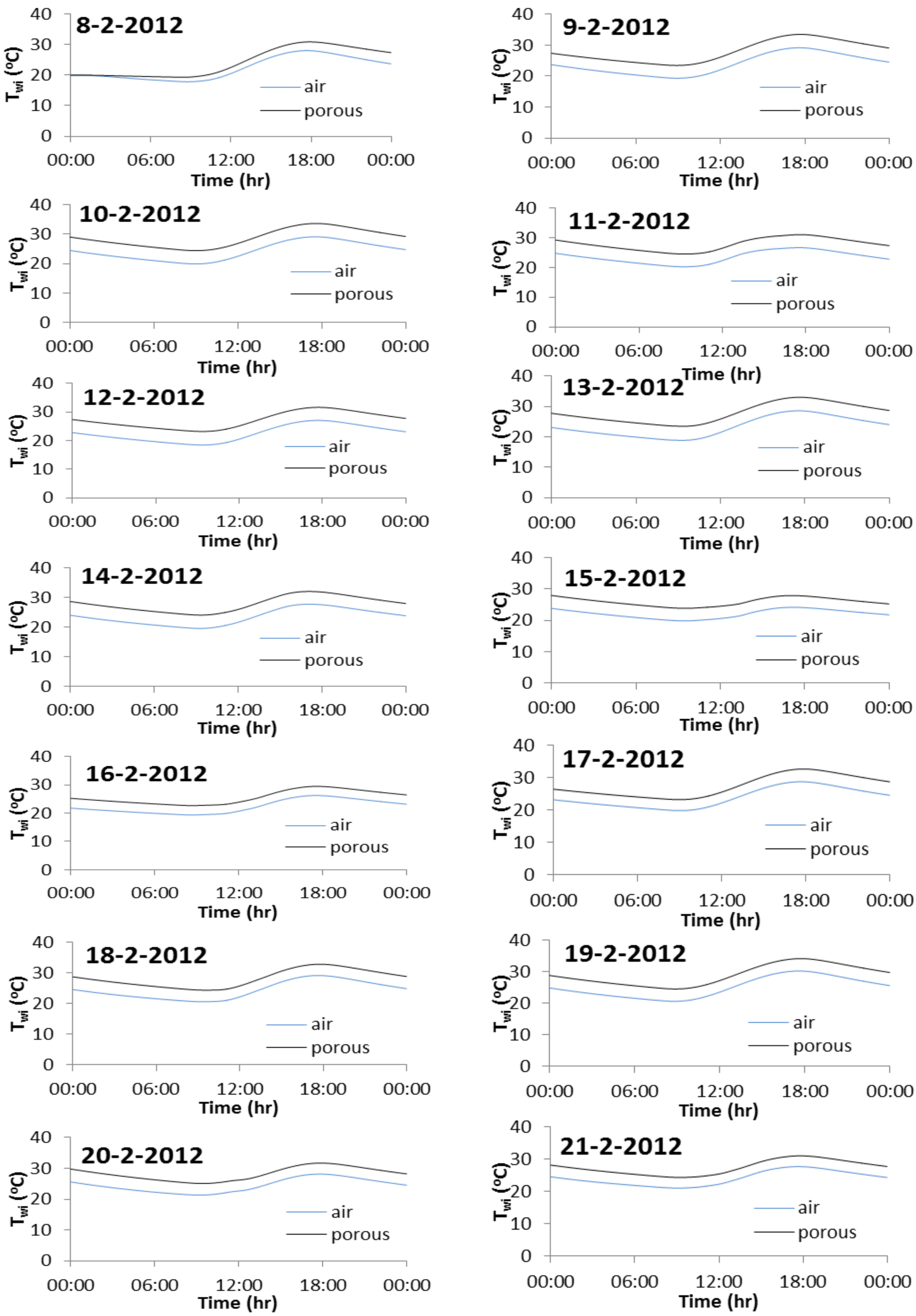

Fig. 3. The inside storage wall temperature along the day

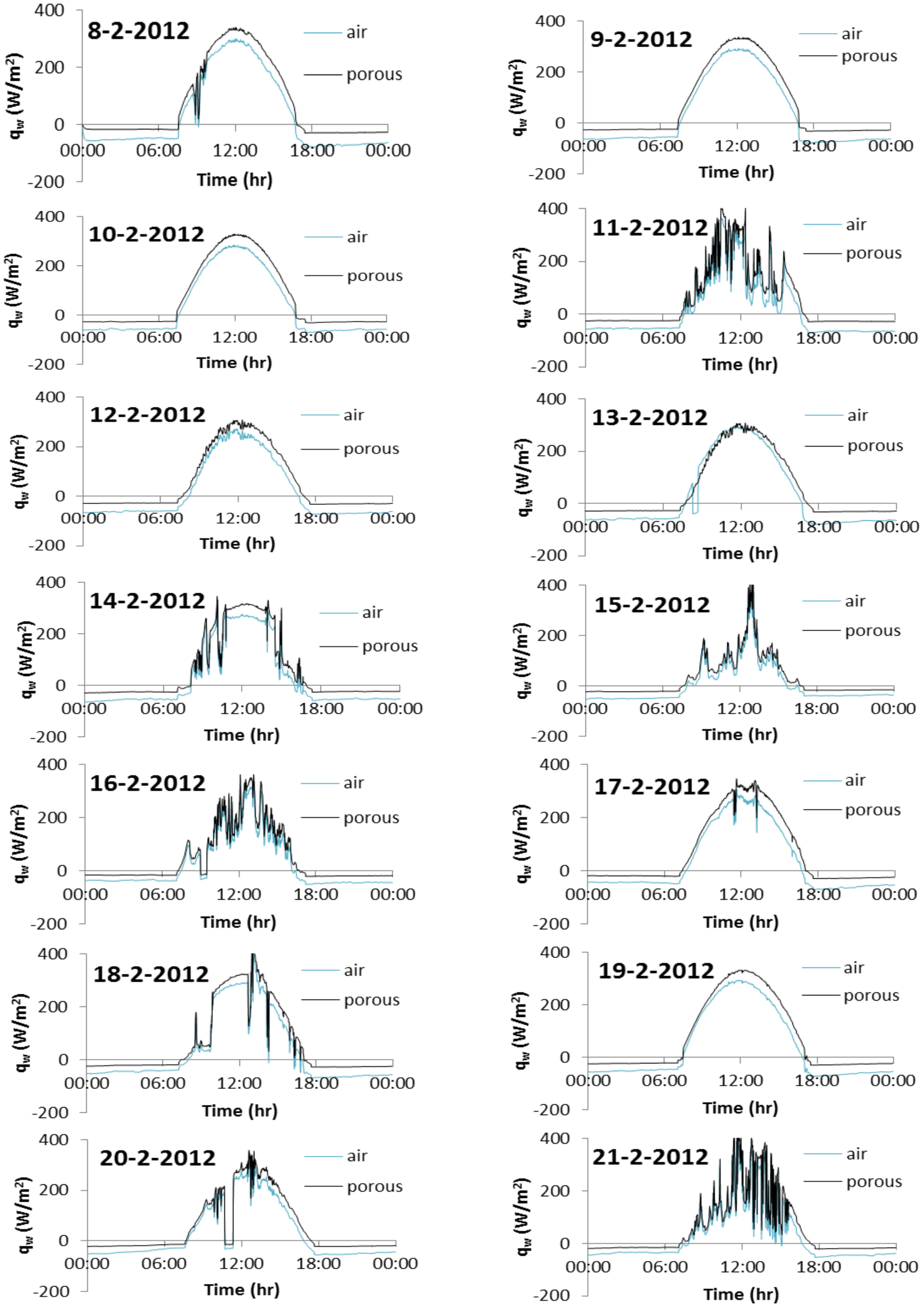

Fig. 4. The rate of heat transfer through the sun space to the storage wall along the day

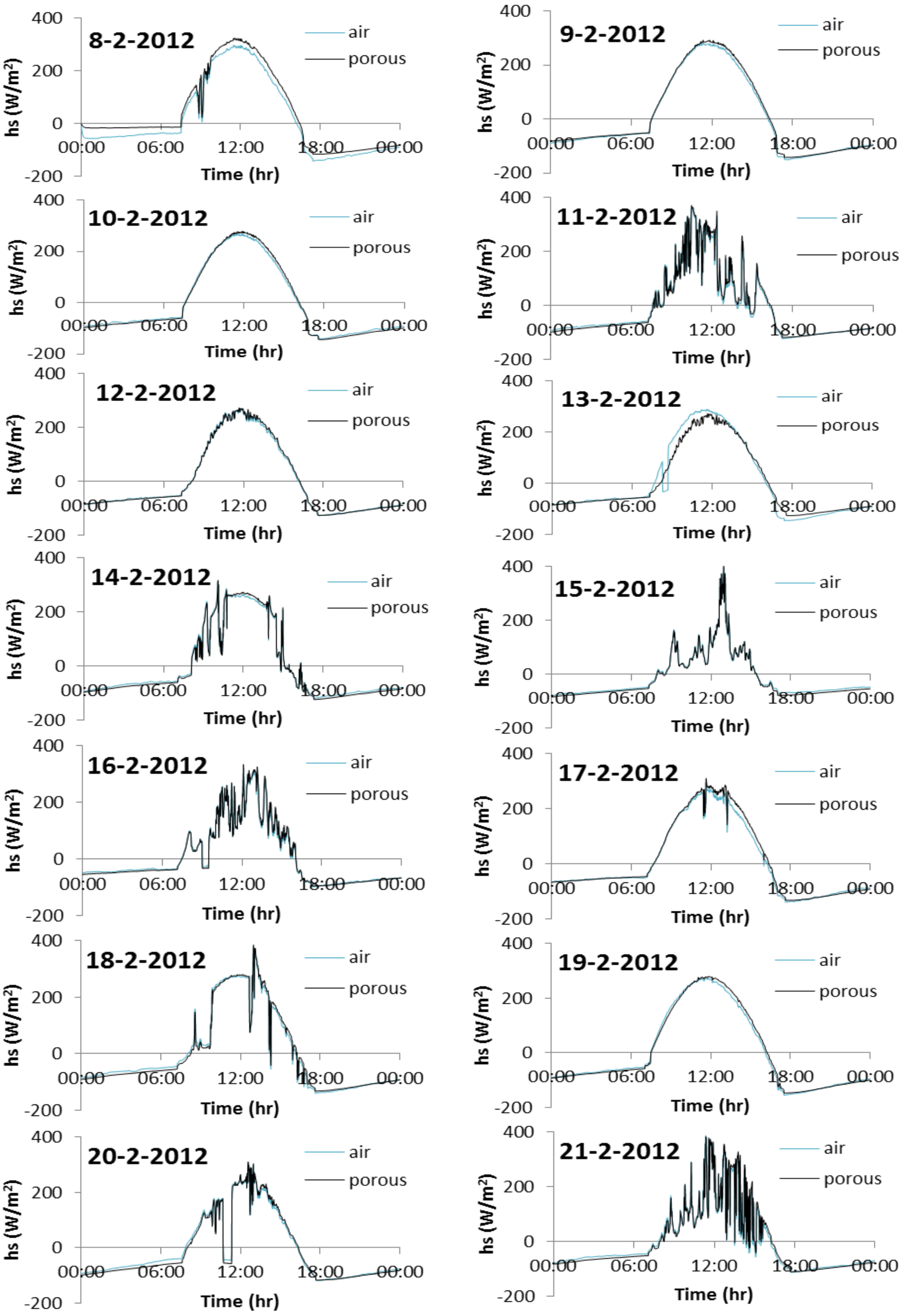

Fig. 5. The heat stored in the storage wall along the day

Conclusions

The effect of using a porous medium in a sunspace was theoretically investigated comparison with using air.

The outside and the inside storage wall temperatures increased when the porous medium was used. The maximum increasing for the outside storage wall temperature, between the porous medium case and the air case, was 19.7%, and for the inside storage wall temperature was 20.3%. The minimum increasing was 20% for the outside storage wall temperature and 11% for the inside storage wall temperature. The rate of heat transferred through the storage wall was larger, when using the porous medium than when using air, and the heat storage in the storage wall was greater. Increasing the heat storage will help in reducing the auxiliary heating during the night and in shining days, where the heating is depending on passive solar heating.

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