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Investigation of Effect of Rock Storage System Parameters on Thermal Cooling Performance

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Abstract: We investigate the effects of key parameters of the rock bed system on thermal cooling performance of the system after a fixed time of operation. The method of solving the mathematical model uses a semi-discretization finite difference approximation for discretizing space in solid problem domain. A finite element approximation is used in the fluid problem domain. Graphical results on the effects of parameter variation on damping and time delay on the peaking of the outlet air temperature through the bed are presented and discussed.

Keywords: Rock Beds, Effects, Thermal Storage Systems, Parameters

1. Introduction

Passive cooling systems using building fabric storage have been used and researched over years, as means of providing building cooling using much less energy than electricity powered individual air-conditioning unit. The need to use sustainable energy technology is of great interest. Using renewable energy and energy efficiency in buildings is techno-economically feasible, and significantly cheaper than building new power stations. A long-term solution in the form of new low-cost technology that relies only on renewable sources of energy and on locally available materials is needed. Sustainable energy sources researches are continuing to address the concern over climate changes, pollution and non-renewable sources (Anderson *et al*., 2015).

The use of thermal energy storage (TES) plays an important role in building energy conservation initiatives, which are greatly enhanced by incorporation of latent heat storage in building products. Thermal energy storage deals with selecting of media/devices that absorb and store heat/cold during peak power operation and releases the same during reduced power operation. Building fabric or packed beds (rock pile, pebble bed and rock bed) materials are some of the thermal storage devices (Ravikumar and Srinivasan, 2005). However, storage and recovery of thermal energy must be done efficiently to achieve high capacity factors and low levelized cost of electricity. As described in the review of Kuravi *et al*., (2013) thermal energy technology should meet several requirements; high energy density, good heat transfer between the heat transfer fluid and solid storage media, stability of the storage medium, low thermal losses, low cost and reversibility through many charging and discharging cycles.

The thermal energy storage capacity of buildings if increased can increase human comfort by decreasing the frequency of internal air temperature swings so that indoor air temperature is closer to the desired temperature for a longer period of time (Dincer *et al*., 1997). TES systems provide the potential to attain energy savings, which in turn reduce the environmental impact related to non-renewable energy use. In fact, these systems can reduce time or rate mismatch that is often found between energy supply and energy demand (Ataer, 2006; Pasupathy and Velraj, 2006; Manohar and Adeyanju, 2009).

The thermal mass is that it can be used in conjunction with night ventilation of a building to provide passive cooling. Outside air is circulated through the building where it comes into contact with and cools the building fabric. The cooling that is stored in the building fabric is then available to offset the heat gains of the following day and keep temperatures within comfort limits (Barnard, 2006). A comparison of thermal energy storage design is given by Li *et al.* (2012). TES can be done with sensible heat storage systems or latent heat storage. The present study explores sensible thermal energy storage and the solid arrangement studied here is to store the heat/cool in a packed bed.

According to Coutier and Farber (1982) packed bed (rock bed) represents the most sensible energy storage unit for air based systems. A packed bed is a volume of porous media obtained by packing particles of selected materials into a container. A packed bed storage system consists of loosely packed solid materials through which the heat transport fluid is circulated. Heated fluid (usually air) flows from solar collectors into a bed of graded particles from top to bottom in which thermal energy is transferred during the charging phase (Balaras, C. A. 1996; Singh *et al*, 2010; Beasley and Clack, 1984).

Aly and El-Sharkawy (1990) studied the effects of storage material properties on the thermal behaviour of packed beds during charging. A transient one-dimensional two-phase mathematical model was used to describe the temperature fields in the air and solid media constituting the bed. They found out that one of the main sets of parameters affecting the design of solid packed beds is the physical properties of the solid phase used as storage material thus choice of material is particularly important.

The aim of this study is to investigate the effect of key parameters of the rock bed system on the amount of heat stored after a fixed time of operation. The rock bed acts as regenerative heat exchanger. Cool air is used to lower the temperature of the rocks at night so that rocks absorb the heat from the higher temperature outside air during the day.

2. The Mathematical Model

The problem considered is a mathematical representation of heat transfer in the packed bed unit. It predicts the response of the unit to the time dependent fluid inlet temperature. The rock bed thermal storage unit under investigation is shown diagrammatically in Figure 1. The rocks are contained in a rectangular shaped wire cage set against the sides of a brick underground. The bed is used as a thermal storage to regulate temperature in the room space environment for a typical building. The energy transporting fluid is air, and energy transporting materials are rocks.

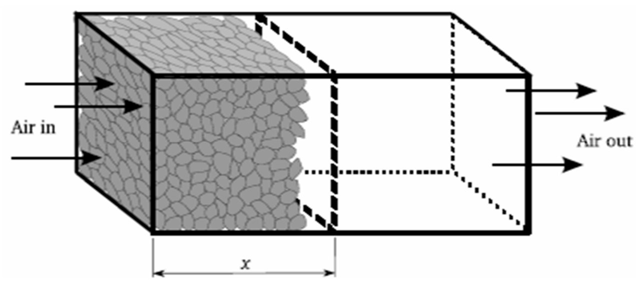

Figure 1. *The model of a packed bed.*

In the model, the air flow is assumed to be evenly spread in cross-section and rocks at a cross-section are assumed to have the same temperature distribution. For the sake of ease of analysing the temperature distribution within the rock, the shape is assumed to be spherical. Thus, for the mathematical model the bed is assumed to be evenly packed with spherical rocks of equal radius (that is, we assume the bed contains the same mass and size of rock with uniform cross-section area), with a fluid (air) spread uniformly in cross-section, flowing through the bed starting from the initial time $(t=0)$.

The temperature of the rocks satisfies the heat diffusion equation. The energy equation for the fluid contains diffusion and heat transport terms. Coupling of the solid (rock) and fluid phases is through heat convection between the rock surface and the fluid, which is represented by a term in the equation of each phase. The boundary conditions for the fluid are that the initial temperature is known, and at the outlet end the fluid is assumed not to lose heat significantly to the duct. An additional term in the fluid equation allows for heat loss to the environment. This takes the form of convection to the surrounding earth or bricks which are at a known constant temperature. In the development of the model, the following assumptions are taken into consideration. The bed material temperature $T_m(r,x,t)$, satisfies the one-dimensional parabolic equation (1) for $0 \le x \le L$.

$$\rho_m c_m \frac{\partial T_m}{\partial t} = k_m \frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2 \frac{\partial T_m}{\partial r}\right), \quad 0 < r < R, \quad 0 \le x \le L, \quad t > 0, \tag{1}$$

with boundary conditions:

$$-k_m \frac{\partial T_m}{\partial r}(r=R,x,t) = h_c\left(T_{ms}(x,t) - T_f(x,t)\right), \tag{2}$$

$$\frac{\partial T_m}{\partial r}(r=0,x,t) = 0, \qquad t > 0, \tag{3}$$

and initial condition:

$$T_m(r,x,t=0) = T_0, \qquad 0 \le r \le R, \quad 0 \le x \le L. \tag{4}$$

also the fluid temperature, $T_f(x,t)$ satisfies,

$$\dot{m}_a c_f L \frac{\partial T_f}{\partial x} = k_f S_{fr} \varepsilon L \frac{\partial^2 T_f}{\partial x^2} + h_c A\left(T_{ms} - T_f\right) + UPL\left(T_{env} - T_f\right),\ t > 0, \tag{5}$$

subject to the boundary conditions:

$$T_f\left(x = 0,\ t\right) = T_{fi}\left(t\right), \quad t \ge 0, \tag{6}$$

$$\frac{\partial T_f}{\partial x}\left(x = L, t\right) = 0, \quad t > 0, \tag{7}$$

and initial condition:

$$T_f\left(x, t = 0\right) = T_0, \qquad 0 \le x \le L, \tag{8}$$

where, A is the area of convective heat transfer, c is the specific heat capacity, ε is the bed void fraction (which is defined as the volume of gas per unit volume of rock-bed), h_c is the convective heat transfer coefficient, k is the thermal conductivity, L is the bed length, P is the bed perimeter, r is the radial distance, ρ is the density, R is the radius of the average solid particles (rocks), S_{fr} is the surface frontal area, T is the temperature, t is time, U is the heat loss coefficient from bed to surrounding, x is the distance in the direction of fluid flow, and $\dot{m}_a$ is the mass flow rate of heat transfer fluid. For subscripts: f stands for fluid, fi is fluid inlet, m is bed material and s stands for surface of a rock.

The problem variable $T_m\left(r, x, t\right)$ is defined for any x in the range $0 \le x \le L$, although the spheres occur in random discrete positions in the bed. Thus T_m represents the temperature in an average sphere at position x as it varies with time. Similarly $T_f\left(x, t\right)$ is the fluid temperature at position x varying with time t. The term on the left hand side of equation (5) represents an energy rate associated with the motion of the fluid through the bed. On the right hand side, the first term represents fluid heat dispersion and, the second and third terms represent convective heat transfer on the fluid-solid interface and, thermal interaction with the environment, respectively.

3. Numerical Procedures

The mathematical model in equations (1)-(4) is solved numerically using semi-discretization finite difference approximation. For equations (5)-(8) finite element approximation is used. In order to develop a numerical method which uses either finite differences or finite elements, nodes are introduced for the space dimension along the bed and along the radius of the spheres. We define the following discretization in space for fixed time t, $r = i\Delta r, \quad i = 0, 1, 2, \ldots, M - 1$, $x = j\Delta x, \quad j = 0, 1, 2, \ldots, N - 1,$ where $\Delta r = R / \left(M - 1\right)$ and $\Delta x = L / \left(N - 1\right)$ are node spacing and M, N are the number of node points used in discretization of the sphere and bed respectively. Consider approximating the space derivatives with finite differences. Equation (1) is equivalent to

$$\frac{\partial T_m}{\partial t} = \alpha\left(\frac{\partial^2 T_m}{\partial r^2} + \frac{2}{r}\frac{\partial T_m}{\partial r}\right), \quad 0 < r < R, \quad t > 0, \tag{9}$$

where $\alpha = \frac{k_m}{\rho_m c_m}$ and satisfies the boundary condition $\frac{\partial T_m}{\partial r}\left(0, x, t\right) = 0$. A consideration of the symmetry of the solution shows that $\frac{\partial T_m}{\partial r} = 0$ at the origin. Keeping t constant, we treat T_m as a function of r only and expand the equation on the right hand side of equation (1) using a Taylor series around $r = 0$ to obtain

$$\frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2 \frac{\partial T_m}{\partial r}\right) \approx 3\frac{\partial^2 T_m}{\partial r^2}\left(0\right) \tag{10}$$

which transforms equation (1) to

$$\frac{\partial T_m}{\partial t} = 3\alpha \frac{\partial^2 T_m}{\partial r^2} \tag{11}$$

Using central difference quotient, at the first node, equation (11) becomes

$$\frac{dT_m^{(1,j)}}{dt} = 3\alpha\left(\frac{-2T_m^{(1,j)} + T_m^{(2,j)}}{\left(\Delta r\right)^2}\right) \tag{12}$$

since $T_m^{(0,j)} = T_m^{(2,j)}$ which follows from boundary condition (6). For the middle nodes the discrete form of equation (1) is

$$\frac{dT_m^{(i,j)}}{\partial t} = \alpha\left(\frac{T_m^{(i-1,j)} - 2T_m^{(i,j)} + T_m^{(i+1,j)}}{\left(\Delta r\right)^2} + \left(\frac{2}{r_i}\right)\frac{T_m^{(i+1,j)} - T_m^{(i-1,j)}}{2\Delta r}\right) \tag{13}$$

and at node M is

$$\frac{dT_m^{(M,j)}}{dt} = \alpha\left(\frac{T_m^{(M-1,j)} - 2T_m^{(M,j)} + T_m^{(M+1,j)}}{(\Delta r)^2} + \left(\frac{2}{R}\right)\frac{T_m^{(M+1,j)} - T_m^{(M-1,j)}}{2\Delta r}\right) \tag{14}$$

Suppose an artificial node $M+1$ is introduced, from the surface boundary condition equation (2) we use central difference quotient for the first derivative for the surface boundary condition equation (5) to obtain,

$$T_m^{(M+1,j)} = 2\sigma\Delta r\left(T_{ms}^{j} - T_f\right) + T_m^{(M-1,j)}, \quad \sigma = -\frac{h_c}{k_m} \quad . \tag{15}$$

By substituting this result into equation (14) we get

$$\frac{dT_m^{(M,j)}}{dt} = \alpha\left(\frac{2}{(\Delta r)^2}T_m^{(M-1,j)} + \left(\frac{2\sigma\Delta r - 2}{(\Delta r)^2} + \frac{2\sigma}{R}\right)T_m^{(M,j)} - 2\sigma\left(\frac{1}{\Delta r} + \frac{1}{R}\right)T_f\right) \tag{16}$$

where $T_{ms}^{j} = T_m^{(M,j)}$ is the interface rock temperature.

Equations (12) through (16) form a system of equations for the thermal energy material (solid) and may be written in matrix form as

$$\dot{\mathrm{T}}_m(t) = B\,\mathrm{T}_m(t) - T_f(t)\,\mathrm{b} \tag{17}$$

where,

$$\dot{\mathrm{T}}_m(t) = \left(\dot{T}_m^{(1,1)}, \dot{T}_m^{(2,1)}, \ldots, \dot{T}_m^{(M,1)}, \dot{T}_m^{(1,2)}, \dot{T}_m^{(2,2)}, \ldots, \dot{T}_m^{(M,2)}, \dot{T}_m^{(1,N)}, \dot{T}_m^{(2,N)}, \ldots, \dot{T}_m^{(M,N)}\right)^T,$$

$$\mathrm{T}_m(t) = \left(T_m^{(1,1)}, T_m^{(2,1)}, \ldots, T_m^{(M,1)}, T_m^{(1,2)}, T_m^{(2,2)}, \ldots, T_m^{(M,2)}, T_m^{(1,N)}, T_m^{(2,N)}, \ldots, T_m^{(M,N)}\right)^T,$$

$\mathrm{b} = \left(0,0,0,\ldots,-2\alpha\sigma\left(\frac{1}{\Delta r} + \frac{1}{R}\right)\right)^T$ (The constant vector from the convection boundary condition) and B is an $MN \times MN$ matrix shown below:

$$\mathbf{B} = \alpha\begin{pmatrix} -2\delta & 2\delta & & & & & \\ a_{t2} & b_t & c_{t2} & & & & \\ & a_{t3} & b_t & c_{t3} \cdots & & & \\ & \vdots & \ddots & & & & \\ & & a_{ti} & b_t & c_{ti} \cdots & & \\ & & \vdots & \ddots & & & \\ & & & a_{t(M-1)} & b_t & c_{t(M-1)} & \\ & & & & -b_t & \varphi & \end{pmatrix}, \tag{18}$$

where, $\delta = \frac{3}{(\Delta r)^2}$, $b = -\frac{2}{(\Delta r)^2}$,

$$a_{ti} = \left(\frac{1}{(\Delta r)^2} - \frac{1}{(\Delta r)r_i}\right),$$

$$c_{ti} = \left(\frac{1}{(\Delta r)^2} + \frac{1}{(\Delta r)r_i}\right), \quad i = 2, 3, \ldots, M-1,$$

$$\phi = \left(\frac{2\sigma\Delta r - 2}{(\Delta r)^2} + \frac{2\sigma}{R}\right).$$

The system of equations (17) is consistent and consists of MN equations in MN unknowns for the solid.

4. Finite Element Solution of the Fluid Equation

Finite element method is used with the fluid temperature approximated by linear variation on each element. This has been chosen as the temperature along the bed does not vary sharply. Using iso-parametric formulation of the element stiffness matrix and Garlekin's approach (Logan, 2007) we obtain the following matrix form.

$$\left[\frac{\gamma}{L}\begin{pmatrix} 1 & -1 \\ -1 & 1 \end{pmatrix} + \frac{b}{2}\begin{pmatrix} -1 & 1 \\ -1 & 1 \end{pmatrix} + \frac{\mu L}{6}\begin{pmatrix} 2 & 1 \\ 1 & 2 \end{pmatrix}\right]\begin{bmatrix} T_{f,1} \\ T_{f,2} \end{bmatrix} = \frac{\beta T_{env} L}{2}\begin{pmatrix} 1 \\ 1 \end{pmatrix} + \frac{\delta T_{ms} L}{2}\begin{pmatrix} 1 \\ 1 \end{pmatrix} \tag{19}$$

where $\gamma = k_f S_{f r} \varepsilon L$, $b = \dot{m}_f c_f L,$, $\mu = h_c A + UPL$ while $T_{f,1}$ and $T_{f,2}$ are the nodal temperatures to be determined. Applying the element equation $\{f\} = [K]\{\hat{T}\}$ to equation (19), we see that the element stiffness matrix is now composed of three parts:

$$[K] = [K_c] + [K_h] + [K_m] \tag{20}$$

where,

$$[K_c] = \frac{\gamma}{L}\begin{pmatrix} 1 & -1 \\ -1 & 1 \end{pmatrix}, \quad [K_h] = \frac{\mu L}{6}\begin{pmatrix} 2 & 1 \\ 1 & 2 \end{pmatrix},$$
$$[K_m] = \frac{b}{2}\begin{pmatrix} -1 & 1 \\ -1 & 1 \end{pmatrix} \tag{21}$$

and the element force matrix and unknown temperature matrix are $\{f\} = \frac{\beta T_{env} L}{2}\begin{pmatrix} 1 \\ 1 \end{pmatrix} + \frac{\delta T_{ms}}{2}\begin{pmatrix} 1 \\ 1 \end{pmatrix}$ and $\{\hat{T}\} = \begin{Bmatrix} T_{f,1} \\ T_{f,2} \end{Bmatrix}$ respectively. From equation (21) we observe that the mass transport stiffness matrix $[K_m]$ is asymmetric and, hence $[K]$ is asymmetric. Thus, the equations for fluid without boundary conditions becomes,

$$K\mathrm{T}_f = \mathrm{f} \tag{22}$$

The stiffness matrix K comes from the terms involving the fluid temperatures, and the element nodal force matrix $\mathbf{f}$ from the convection and environmental terms. The condition that $T_{f1} = T_{fi}$ has to be imposed, that is, we set $K(1,1) = 1$ and zero for the rest of the rows and $\mathbf{f}(1) = T_{fi}$. If equation (22) is multiplied by K^{-1} and the derivative boundary condition is incorporated the resulting equation has the form:

$$\mathbf{T}_f = K^{-1}\mathbf{f} + T_{fi}\mathbf{g} \tag{23}$$

where $\mathbf{g} = [1 \ 0 \ 0 \ \ldots \ 0]^T$.

5. Semi-analytic Method for Solution in Time

The rock and fluid temperatures are assumed to be known at a particular time t_1 , and we needed to find the rock temperatures at $t_2 = t_1 + \Delta t$. The fluid temperatures are known at t_2 from the given inlet temperature. We assume that the fluid temperature T_f varies linearly between times t_1 and t_2 , that is, the fluid temperature responds very quickly to a change in input temperature because of the flow speed. This contrasts with the change of temperature in the rocks. That is,

$$T_f(t) = T_{f1} + \frac{t - t_1}{\Delta t}\left(T_{f2} - T_{f1}\right) \tag{24}$$

Then equation (17) becomes,

$$\frac{d\,\mathbf{T}_m(t)}{dt} = B\,\mathbf{T}_m(t) - \mathbf{b}\left[T_{f1} + \left(T_{f2} - T_{f1}\right)\left(\frac{t - t_1}{\Delta t}\right)\right] \tag{25}$$

for $t_1 \le t \le t_2$ with $\mathbf{T}_m(t_1) = \mathbf{T}_{m1}$ known and $\Delta t = t_2 - t_1$. Setting $P = -\mathbf{b} T_{f1}$ and, $Q = -\frac{\mathbf{b}\left(T_{f2} - T_{f1}\right)}{\Delta t}$ equation (25) becomes

$$\dot{\mathbf{T}}_m(t) = B\mathbf{T}_m(t) + P + Q(t - t_1) \qquad t_1 \le t \le t_2 \tag{26}$$

Let $\mathbf{T}_a(\tau) = \mathbf{T}_m(t) - \mathbf{T}_{m1}$, where the subscript a stands for "additional" and $\tau = t - t_1$. Then, equation (26) changes to

$$\dot{\mathbf{T}}_a(t) = B\mathbf{T}_a(\tau) + B\mathbf{T}_{m1} + P + Q\tau \ , \quad \mathbf{T}_a(0) = 0 \tag{27}$$

Suppose X is a matrix that diagonalizes B , then $X^{-1}BX = D$ (the diagonal matrix) and since $BX = XD$ then equation (27) becomes

$$\dot{\mathbf{z}}(\tau) = D\mathbf{z}(\tau) + X^{-1}\left(B\mathbf{T}_{m1} + P\right) + X^{-1}Q\tau \tag{28}$$

Letting $H = X^{-1}\left(B\mathbf{T}_{m1} + P\right)$ and $S = X^{-1}Q$ equation (28) can be written as

$$\dot{\mathbf{z}}(\tau) = D\mathbf{z}(\tau) + H + S\tau, \qquad \mathbf{z}(0) = \mathbf{0}, \tag{29}$$

This is a set of uncoupled equations for the temperature at each x along the rock cage. The general equation of uncoupled system is

$$\dot{z}_j = \lambda_j z_j + H + S\tau \tag{30}$$

Equation (30) is a first order differential equation and can be solved using the usual methods. For each row of the system (30) the solution is, at $\tau = \Delta t$, we have

$$z_j = \alpha_j\left(1 - \exp(\lambda_j \Delta t)\right) + \beta_j \Delta t, \quad i = 1, 2, \ldots, M$$

In general, we have

$$\mathbf{z}(\Delta t) = \alpha\left(1 - \exp\left(\Delta t D\right)\right) + \beta \Delta t, \tag{31}$$

and $\mathbf{T}_a(\Delta t) = X\mathbf{z}$, $\mathbf{T}_{m2} = \mathbf{T}_{m1} + \mathbf{T}_a$

6. Results and Discussions

The results of mathematical simulation of the system have been discussed in the system. The effect of the system parameters (convective heat transfer coefficient h_c, mass flow rate $\dot{m}_a$, the length of the bed L, and rock sizes (stones dimension)) on the thermal performance of the system has been investigated. It is thought that the temperature distribution in the bed is the most aspect, it is important to discuss the effect of system and operating variable on temperature distribution. Mathematical simulation has been used to yield the temperature of element of storage system during charging at a given instant of time.

Table 1. *Fixed Parameters of bed.*

Rock	Air	Other parameters
$c_m = 800$ J/kg °C	$c_f = 1007$ J/kg °C	$h_c = 6$ W/m^2 °C
$\rho_m = 2700$ kg/m^3	$\rho_f = 1.106$ kg/m^3	$\varepsilon = 0.5$
$k_m = 2.1$ W/m °C	$k_f = 0.25$ W/m °C	$L = 2$ m
$R = 0.1$ m		$S_{fr} = 5.4\,\text{m}^2$
		$A = 3\dfrac{(1-\varepsilon)}{R} S_{fr} L$ m^2

Table 2. *Range of system parameters.*

Parameter	Range
Coefficient of heat transfer coefficient, (h_c)	6 to 12
Length of rock bed (m)	2 to 3
Air flow rate ($\dot{m}_a$)	0.4 to 0.8
Radius (m)	0.05 to 0.2

6.1. Effects of Convective Heat Transfer Coefficients

Figure 2 depicts the transient response of the packed unit for different peak values of the convective heat transfer coefficient. Calculation of the fluid temperature leaving the rock bed for various values of convective heat transfer coefficient (h_c) shows that this parameter has significant influence on the thermal performance of the system. A period of 18 h is chosen to allow closer look at the behaviour rather than show the whole fortnight. This covers two periods of fan operation from 12.00 noon to 15: 30 hrs, and 22.00 hrs to 6.00 hrs.

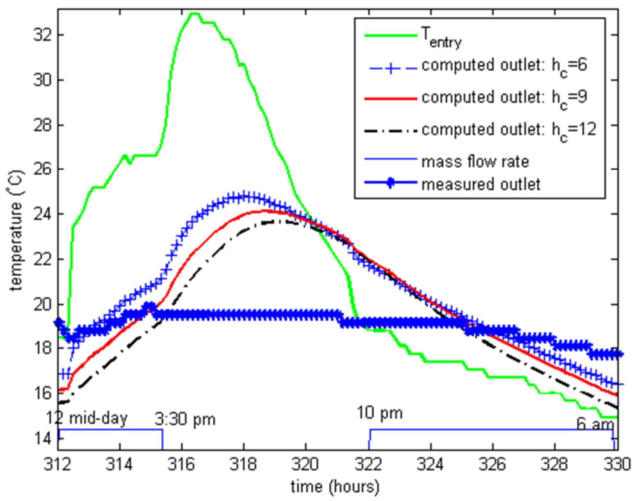

Figure 2. *Effect of varying convective heat transfer coefficient* h_c.

From the figure it can be seen that, the effect of increasing the convective heat transfer coefficient h_c is to dampen the response of the fluid outlet temperature. This is because the interaction between the rocks and the air is enhanced. The bigger the convective heat transfer coefficient the greater the gain in the fluid temperature and this brings the predicted temperature closer to the measured value. In practice, mechanical ventilation is required to achieve reasonable values of convective heat transfer coefficient. However the actual values of convective heat transfer coefficient depend not only on the air flow rate but also on the level of turbulence created in the air flow in the channels, which is a function of surface roughness and obstructions in the air flow path (Isanska-Cwiek, 2005). For each of the three convective heat transfer coefficients, the flow rate of $\dot{m}_a = 0.6\,\text{m}^3/\text{s}$ is used for forced convection and the other parameters are as given in Table 1.

6.2. Effects of the Air Flow Rates

This section investigates the effect of the air flow rate $\dot{m}_a$ by simulating low rates of $0.4\,\text{m}^3/\text{s}$, $0.6\,\text{m}^3/\text{s}$ and $0.8\,\text{m}^3/\text{s}$ to observe the agreement between predicted and measured fluid outlet temperature. Figure 3 shows the transient response of the morning unit for different peak values of $\dot{m}_a$ denoted by m_a in the figure. For each of the three air flow rates, the convective heat transfer coefficient h_c is assumed to be constant at 6 W/m^2 °C and the other parameters are as given in Table 1. Since air flow rate has significant impact on the value of the heat transfer coefficient value, the results are comparable to results given in section 6.1. This shows that the air flow rate parameter has a strong dependence on the thermal performance of the system.

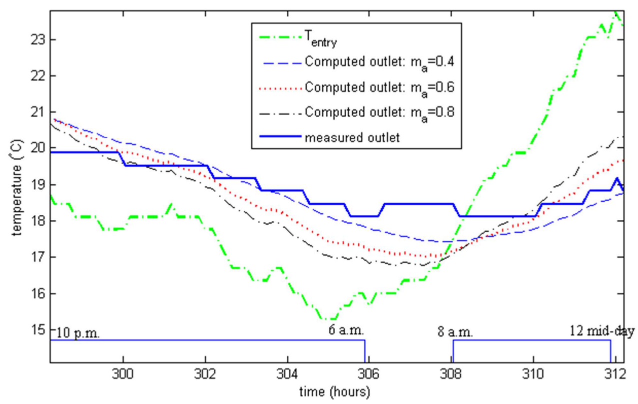

Figure 3. *Effect of varying air flow rate* $\dot{m}_a$.

From Figure 3 it can be seen that using the lower flow rate of $0.4\,\text{m}^3/\text{s}$ is to dampen the fluid outlet temperature. The greater temperature gain achieved with a lower flow rate may be due to the air being given more time to be in contact with the rocks. The situation is complicated if it is assumed that convection is linked to air flow rate; then lowering the air flow rate in itself increases the peak damping, but the related result of decreasing the convective heat transfer coefficient reverses the effect. Also, the measured average value of $0.6\,m^3/s$ recommended for use in the modelling is not a correct value. The fluid must be well mixed before taking a measurement otherwise the reading is not a true representative of the fluid flow rate.

6.3. Effects of Length of the Packed Bed

Figure 4 compares the outlet air temperature from rock beds of the different lengths. The figure indicates that the effect of increasing the bed length L is to increase the peak damping of the fluid outlet temperature. That is, the temperature difference is greater as the bed length increases.

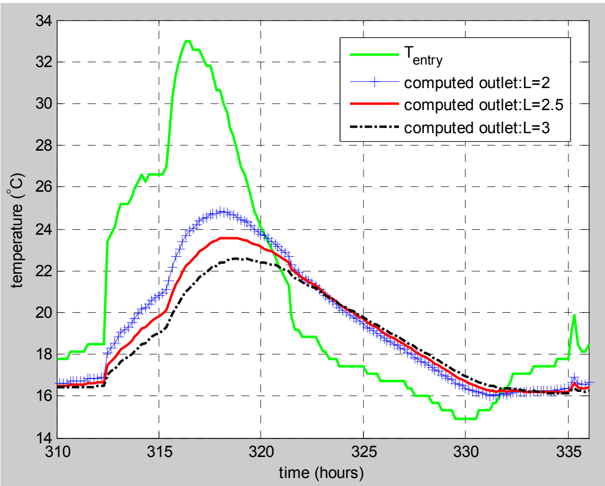

Figure 4. *Effect of varying length of the packed bed.*

6.4. Effects of Rock Size

In Figure 5 the effects of radius of rocks (rock sizes) on the outlet air temperature from packed bed is displayed. It is found from the figure that, lowering the radius of the rocks increases the peak damping of the fluid outlet temperature. Hence, for efficient operation, smaller rocks allow for more efficient storage of energy than large rocks.

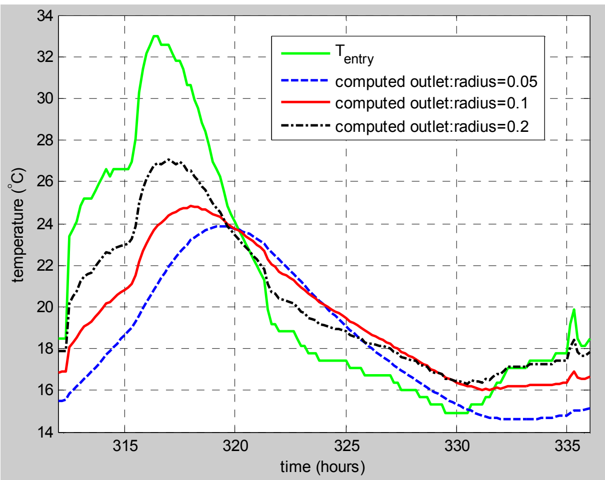

Figure 5. *Effect of varying rock sizes.*

7. Conclusions

The combined effect of convective heat transfer coefficient, air flow rates, length of the packed bed and the rock size on the transient responses of the packed bed unit to the time dependent fluid inlet temperature was investigated. The partial differential equations are solved numerically using a semi-discretization finite difference method together with finite element method. The results are summarized as follows:

- The greater convective heat transfer coefficients tend to produces the greater gain in the outlet fluid temperature leaving the rock bed.
- The greater temperature gain can be achieved with lower flow rate and this is due to the air being given more time to be in contact with the rocks.
- For the efficient operation it is found also that, long bed length and smaller rocks allow more efficient storage of energy than small bed length and large rocks respectively.

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