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Determination of Cooling Load for a Solar Cooled Building in Khartoum Using a Simulation Method

Kamal Nasreldin Abdalla[1] Elamin Ahmed Osman[2]

[1]*Department of Mechanical Engineering, Faculty of Engineering, University of Khartoum Khartoum, Sudan (Email:* kamalnasr@yahoo.com>*)*
[2]*Geraif Technical College, Khartoum, Sudan (E-mail:elamin1993@gmail.com)*

Abstract: Cooling load calculations are normally carried out assuming steady state conditions. This is a simple but conservative approach that leads to overestimation of the cooling capacity. For more accurate estimation of cooling loads, one has to take into account the thermal capacity of the walls and internal heat sources, which makes the problem more complicated. In this paper, the unsteady state heat transfer formulation has been used to determine the air conditioning cooling load for a building in Khartoum for a hot summer day. An explicit method was used to calculate the nodal temperatures at each time step of the thermal network. A simulation program was developed to calculate the cooling load at each time step. The building is cooled by absorption cooling system comprised mainly of an evacuated tube solar collector, a lithium bromide –water absorption air conditioner, a storage tank fitted with an auxiliary heater. A mathematical model is built to simulate the absorption cooling system for the solar cooled building and the results showed that the absorption system driven by solar energy from the evacuated tubes solar collector is enough to cover the cooling load demand by the building more than ten hours.

Keywords: *Transient; Numerical; Explicit; Cooling Load; Simulation; Solar Cooling*.

1. INTRODUCTION

The usual steady state type of calculation in determining the air conditioning cooling and heating loads is quick but give approximate value. This approximation is mainly due to the fact that the transient thermal response of the building to solar radiation intensities and the variable outdoor temperature has not been taken into account. Transient thermal response of buildings to the external climatic variables produces the thermal storage effect of the structure. So treating the steady state heat gain as cooling load and ignoring this storage effect could lead to an appreciable error in energy estimates even for small commercial buildings. To evaluate the cooling load accurately, the transient heat transfer to which the building is exposed must be solved. Generally, this could be solved by means of numerical or analytical techniques. Numerical techniques, however, are flexible but have inherent errors; these errors could be minimized with careful programming using high storage computer facilities. There are different types of numerical techniques but in this study, the explicit finite difference method has been chosen since it is easy to formulate and the solution for the unknown quantity is given directly [1-3].

Conventional air-conditioning systems require high quality energy, electricity generated from primary energy resources. Therefore, an assessment from the ecological point of view needs to be implemented as the greenhouse gases effect remains a threat to the environment. In fact, most of the buildings, cooling demands in summer are associated with high solar energy availability, which offers an opportunity to further exploit solar energy for cooling. Solar cooling technology provides an important contribution to both economical and ecological energy supply

Operation of absorption air conditioners with energy supplied from the evacuated tubes solar collector and storage tank system is the most common approach to solar cooling [4]. In this paper, the main components of the solar cooling system for the building are the evacuated tubes solar collector, the storage tank fitted with an auxiliary heater, a LiBr–H_2O absorption air conditioner and the building itself. A mathematical model was built to simulate the absorption cooling system for the solar cooled building by modeling each component of the absorption cooling system and theoretical results were obtained. The results were discussed and some important conclusions were drawn.

2. BUILDING THERMAL NETWORK

The building under theoretical tests was chosen as small office with a net floor area of 8.5 m^2 and a volume of 25.5 m^3. All walls were made of common brick (22 cm thick), the roof and floor were made of concrete slab (15 cm each). The (1mx1 m) window was made of a single glass**.**

The conduction path of all structures of the building was approximated by a lumped parameter net work [2,4] which gives a finite difference approximation to the one dimensional transient partial differential equations of conduction heat transfer. This finite difference approximation could be obtained by using either mathematical or physical (heat balance) approach [1-3]. The latter approach was chosen since it is flexible and can easily accommodate any heat transfer complications such as variable heat transfer properties, internal heat generation, difference node sizes and shapes plus heat transfer by convection and radiation.

To apply an explicit in practice, using heat balance approach, consider the nodal sub volume of **Fig. 1**. This element can be considered as an arbitrary sub volume of an outer part of a building wall. Assume one dimensional heat transfer flow and node i to represent the sub volume. Node i has a thermal capacity C_i and connected by resistances R_{ij1} and R_{ij2} which stand for convective and conductive resistances respectively. If node i is exposed to a solar heat input , q_i, and has a temperature T_i^t at time t, then for a time interval Δt , the temperature of node i (T_i^{t+1}) could be shown as follows[4]:

$$T_i^{t+1} = \frac{\Delta t}{c_i}\left[q_i + \frac{T_{j1}^t}{R_{ij1}} + \frac{T_{j2}^t}{R_{ij2}} - T_i^t\left(\frac{1}{R_{ij1}} + \frac{1}{R_{ij2}}\right)\right] + T_i^t \qquad (1)$$

If node i is connected to n number of nodes, then equation (1) becomes:

$$T_i^{t+1} = \frac{\Delta t}{c_i}\left[q_i + \sum_{j=j1}^{jn} \frac{T_j^t}{R_{ij}} - T_i^t\left(\sum_{j=j1}^{jn} \frac{1}{R_{ij}}\right)\right] + T_i^t \qquad (2)$$

The conduction path of the four walls, roof, and floor were approximated by a lumped parameter network which gives a

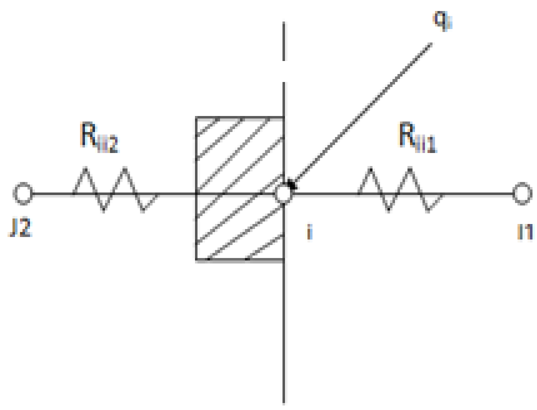

Fig. 1. Illustration of heat balance approach

finite difference approximation to the partial differential equation concerned. A complete view of the thermal network of the whole brick building is shown in Fig. 2 which consists of eight parallel circuits, all of which terminate at node (36) which represents the inside room air node, for more details of the thermal network of the building , the reader is referred to [1, 4].

To evaluate the transient heat response of the office thermal network which is explained in **Fig. 2**, the following assumptions are considered [4, 5]:

- One dimensional conduction heat transfer through the office structure.
- Explicit method is used to solve transient heat conduction.
- Thermal properties of all materials are constant. All inside surfaces radiate and reflect thermal radiation.
- Inside temperature of the space is remained constant

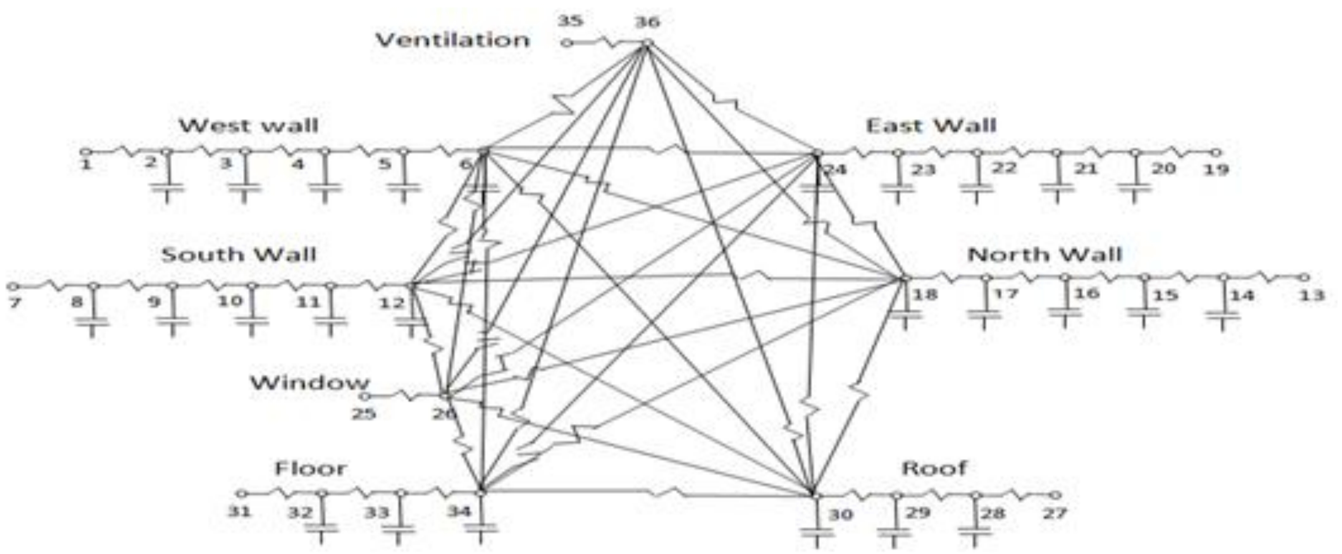

Fig. 2. Building thermal network

The following variables are used to solve the thermal network: Ambient air temperatures, thermal resistances, and thermal capacitances and sol-air temperatures.

3. SIMULATION OF COOLING LOAD

The air conditioning cooling load is basically composed of the following items: direct solar radiation through window, heat transmission through structure, internal heat gains, infiltration and ventilation.

The previous loads can be computed if the basic information about minimum time step and nodal temperature at each time step are known. The minimum time step can be calculated from the following equation to ensure stability and convergence of all calculations:

$$\Delta t_i < \frac{C_i}{\sum_{j=1}^{n} C_{ij}} \tag{3}$$

The new temperature at each succeeding time step of any node in contact with the structure can be obtained from Equation (2) provided that q_i is set to zero since sol-air temperature takes into account all radiation heat exchange. For nodes not into contact with the building, the new temperature will be read directly from the input data.

The total cooling load was obtained by summing the previous four loads at each time step. A simulation program was developed to calculate the cooling loads.

4. SIMULATION OF SOLAR COOLED BUILDING

Fig. 3 shows a schematic diagram of the solar cooling system [6,7] and the different components of this system was modeled as follows:

The solar collector was modeled in a manner in which the useful heat gained by the collector, Qu, is given by[8]:

$$Q_u = F_R A_c \left(H - U_L (T_i - T_a)\right) \tag{4}$$

The tilt angle is set to be equal to the latitude and taken as 15^0 N, which is the latitude of Khartoum, when the collector of the fluid outlet temperature ,T_o , is less than the storage tank temperature ,T_s, there will be no flow of fluid through the collector and as a control function F is set equal to zero. Then, the temperature of the fluid in the solar collector is given by the solution of the following energy equation [4, 8]:

$$m_f C_{pf} \frac{dT_o(t)}{dt} = A_c H - A_c U_L (T_o - T_a) \tag{5}$$

It is assumed that when T_0 is greater than T_s by a prescribed amount ,ε , then the collector pump was set into action as a control function F=1 and the amount of useful energy delivered to the storage tank , Q_u, is given by Equation (4).
When the pump is in work, the temperature of the fluid entering the collector is assumed equal to the temperature of the storage tank and the out let temperature of the collector , T_o, is :

$$T_o(t) = T_i + \frac{Q_u(t)}{\dot{m}_f C_{pf}} \tag{6}$$

Smith and Gupta [9] gave the following energy equation for the storage tank:

$$m_s C_{ps} \frac{dT_s(t)}{dt} = F Q_u(t) - F_1 \dot{m}_s C_{ps} \Delta T_{HE} - A_s U_s (T_s(t) - T_a) \tag{7}$$

The energy equation for the auxiliary tank is given by:

$$m_A C_{pA} \frac{dT_A(t)}{dt} = F_2 (Q_A - \dot{m}_A C_{pA} \Delta T) \tag{8}$$

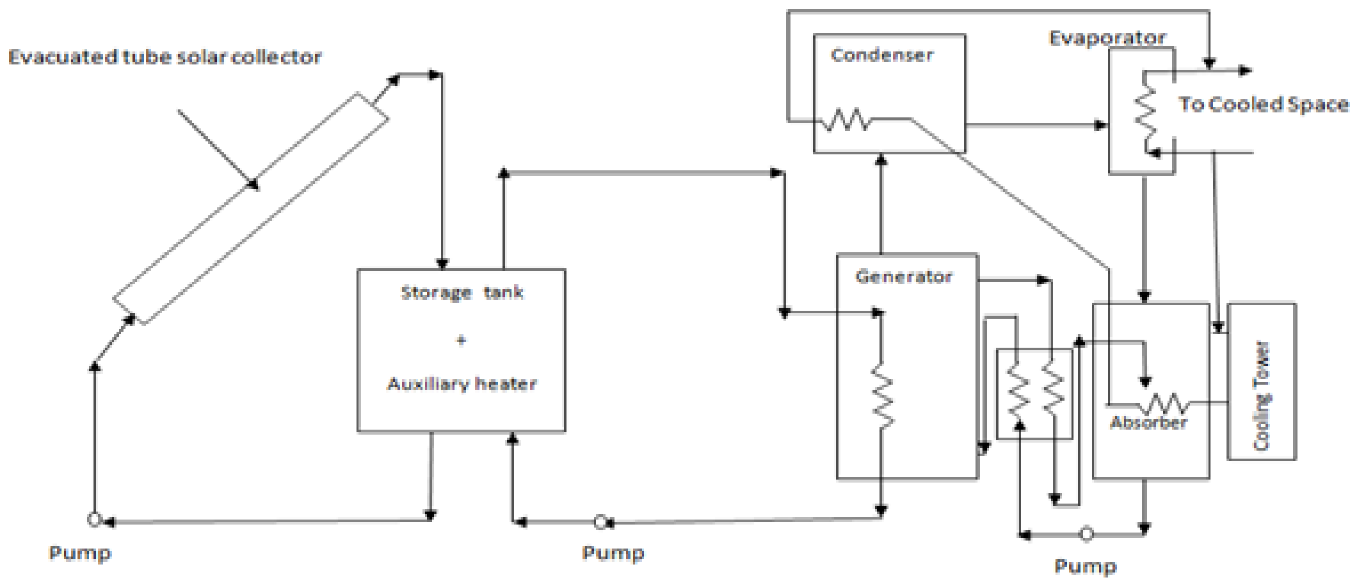

Fig. 4. Schematic of a solar operated absorption of air conditioner

A lithium bromide – water absorption air conditioner with a coefficient of performance of 0.65 was selected. The total load on the cooling coil was modeled by simply multiplying the energy transferred from either the storage or the auxiliary tank to the absorption machine by 0.65 so,

$$\text{The cooling coil load} = 0.65(F1+F2).\dot{m}_s.C_{ps}.\Delta THE \qquad (9)$$

By neglecting ducting and fan power gains, the building heat gain was obtained by subtracting the fresh air load , $m_a\Delta h$, from the total cooling coil load so,

$$Q_, = 0.65(F1+F2).m_s.\ C_{ps}.\Delta THE- m_a.\ \Delta h \qquad (10)$$

$$(\ \Delta h=h_o-h_r\ ) \qquad (11)$$

The energy balance of the building accounts for the heat supplied by either storage or auxiliary tank, the instantaneous heat gain or loss from the building and the cooling load supplied by the absorption air conditioner, therefore the energy equation for the building takes the form:

$$m_b C_{ps} \frac{dT_b(t)}{dt} = (F1+F2)\dot{m}_s C_{ps} \Delta T_{HE} + Q_{total} - F3(0.65(F4F5)\dot{m}_s C_{ps} \Delta T_{HE} - \dot{m}_a \Delta h \qquad (12)$$

The inner node temperature of the building was assumed to be kept within a comfortable range 22^0C to 24^0C. The energy equation for each component

5. RESULTS AND DISCUSSION

The cooling profiles for the different walls, roof, window, ventilation and the total cooling load for the building are shown in **Figs 4** and **5.** Since the initial temperatures of the building structures nodes of the thermal network were assumed based on steady state heat transfer, it was expected that some errors would arise from this assumption. To make these errors quite negligible, the programs calculating the cooling load were run for a period of three days using the same data and it is clear that from **Figs 4** and **5** that the results of the second and third day are similar i.e. the errors in these figures damp out in the second day .Accordingly all discussion will be based on the results of the third day. **Fig. 4** shows the cooling load for the west, south, east, north walls and roof for the building and it is clear that the roof gave the maximum one which is approximately 0.52 kW at 65 hr or 66 hr. This is mainly because the roof was exposed to higher sol-air temperature during the day and secondly due to the earlier assumption that the roof is only a surface which loses heat to the sky by long wave radiation. The curves for east and north walls are slightly similar which gave the maximum cooling load at 67 hr or 68 hr but the cooling load for the north wall is greater than that of east wall since the area subject to the sun radiation of the north wall is greater than the east wall. The cooling load for the south wall is smaller than that of the north wall since the average sol-air temperature is slightly lower (located in the shadow) compared to the sol-air temperature of the north wall. The cooling load for the west wall gave the minimum one which is approximately 0.19 kW at 68 hr or 69 hr. This is mainly because the west wall was exposed to lower sol-air temperature during the day. **Fig. 5** shows the cooling load for ventilation, the window and the total cooling load, the area under the curve for the total cooling load of the second and third day are approximately equal this means that the same total amount of energy (Approximately 3.8 kW) must be removed from the building during the day, however, a large amount of energy is removed during the noon hours. The peak cooling load occurs approximately at the same hour of the window peak load but the peak cooling loads of other contributors occur at different hours, this mainly because the window has negligible storage effect and therefore has zero time lag.

The variation of inside air temperature with time is shown in Figure 6. It can be seen that sharp and abrupt fluctuations occur at certain time steps. The main reasons for these fluctuations could be discussed as follows:

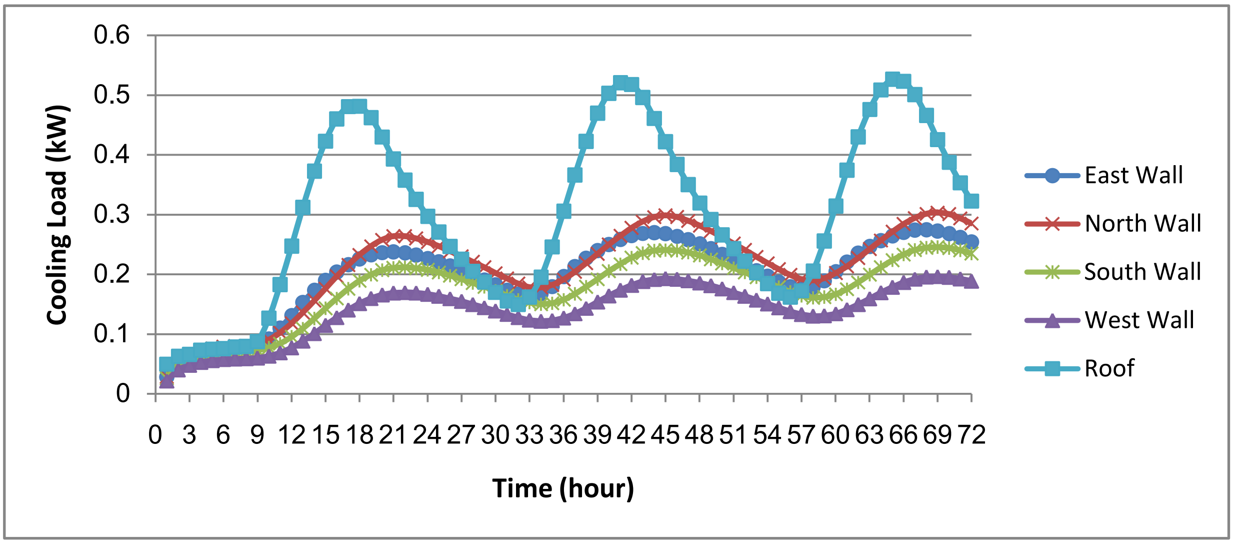

Fig. 4. Cooling load profiles for walls and roof

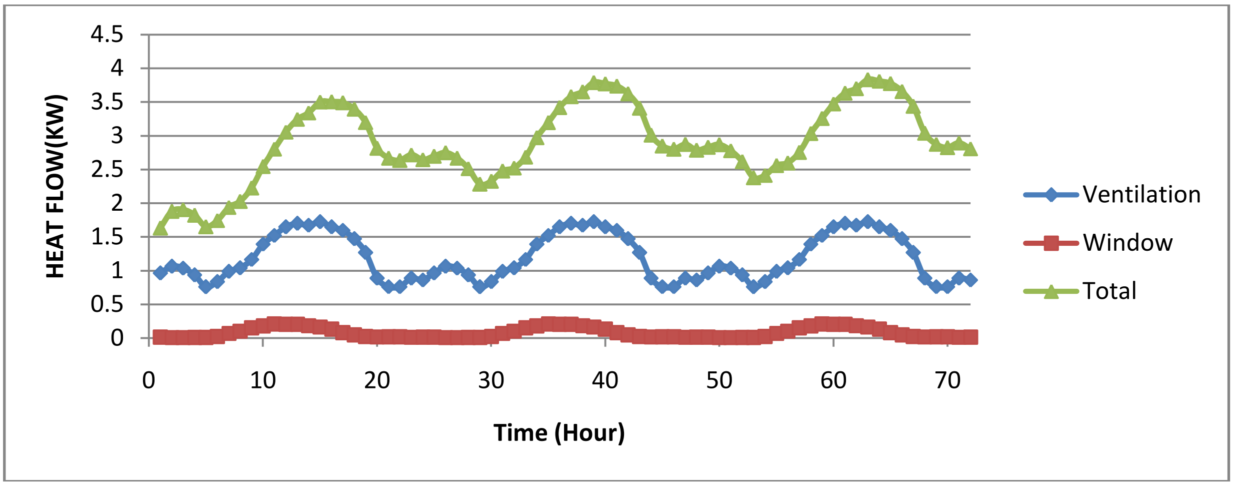

Fig. 5. Cooling load profiles for window, ventilation and total cooling load

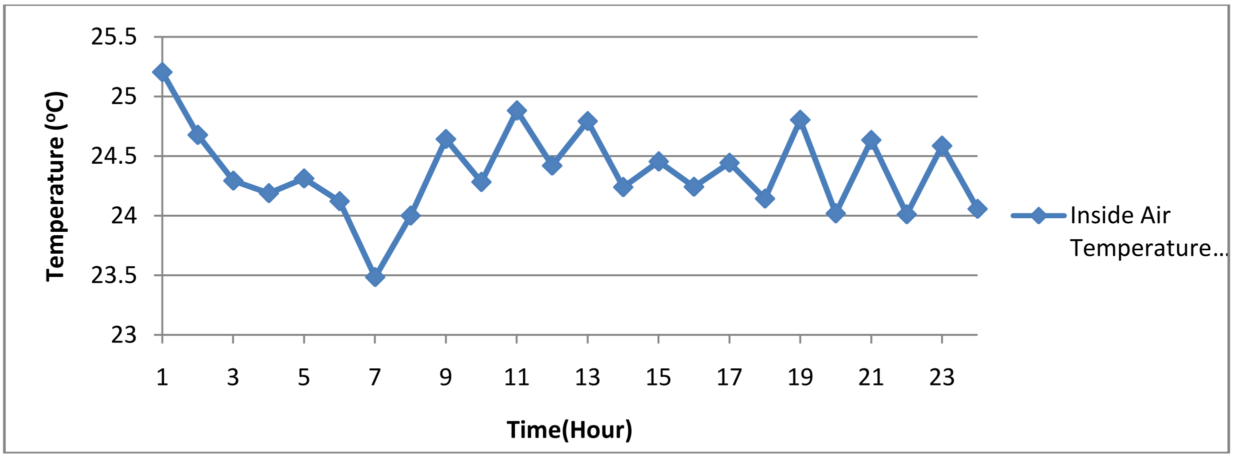

Fig. 6. Variation of inside air temperature with time

The inside air building temperature, TBO, is described by Equation (11) and it is a function of three distinct terms, the first term gives the amount of heat supplied by either storage or auxiliary tank ,the second term is the instantaneous heat gain or loss from the building while the third term gives the amount of cooling introduced to the building from the absorption air conditioner utilizing either the heat coming from the storage or auxiliary tank .if TBO is above 22 ^{o}C and below 24 ^{o}C ,the control function governing the cooling term would be set to zero, so the building temperature is only function of total cooling load or instantaneous heat gain .If the cooling load is increased the inside temperature of the building would increase and vice versa .therefore a sharp and un predictable fluctuations in TBO could happen. If TBO is above 24 ^{o}C the cooling term would be activated. The maximum fluctuation for the inside air temperature of the building is approximately 1.5 ^{o}C and the fluctuations during any single time step is less than 0.8 ^{o}C which is quite reasonable during an hour.

6. CONCLUSIONS

- An explicit numerical technique utilizing thermal network of electrical analogy was used to calculate transient cooling load for a building in Khartoum area .The maximum cooling load for the selected building was approximately found to be equal to 3.8 kW.
- A general simulation program for a solar cooled building has been developed and can be used for any combination of solar collector, building type, storage system, auxiliary heating or cooling system.
- About 60 % of the total cooling demand for the building was supplied by energy from the solar collector and the other 40% portion has to be met by the auxiliary heater.
- The maximum fluctuation for the inside air temperature of the building was found to be around 1.5^{o}C and the fluctuation during any single time step was found to be less than 0.8^{o}C.

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NOMENCLATURE

Ac surface area of evacuated solar collector, m^2
As surface area of storage tank, m^2
C_{pA} specific heat of fluid in auxiliary tank, kJ/kg K
C_{pf} specific heat of fluid in storage tank, kJ/kg K
F, F1, F2, F3, F4, F5 control functions
FR evacuated solar collector efficiency, %
Qu useful energy from Solar collector, kW
H solar radiation intensities incident on the plane of evacuated solar collector, W
UL heat loss coefficient of solar collector, W/m^2
Ti temperature of fluid entering solar collector, ^oC
Ta ambient air temperature, ^{o} C
m_f mass of fluid within the collector, kg
H solar radiation intensities incident on the plane of solar collector, W
ho outside air enthalpy, kJ/kg
hr room air enthalpy, kJ/kg
Δh enthalpy drop of fresh air introduced to the building, kJ/kg
$\dot{m}_A$ mass flow rate of fluid in auxiliary tank, kg/s
m_b mass of air within the building, kg
$\dot{m}_f$ mass flow rate of fluid within the collector, kg
m_s mass of fluid within in the storage tank, kg
$\dot{m}_s$ mass flow rate of fluid within in the storage tank, kg/s
QA heat supplied by the auxiliary tank, kW
Q_{total} total instantaneous heat gain by building, kW
Qu useful energy from collector, kW
UL heat loss coefficient of solar collector, W/m^2
Us heat loss coefficient of storage tank, W/m^2
Ta ambient air temperature, ^oC
Tb inside air temperature of the building, ^oC
ΔTHE design temperature drop through air conditioner, ^oC
To outlet temperature of fluid from collector, ^oC
Ts storage tank temperature, ^oC