SOLAR HEATING AND RADIATIVE COOLING USING UNCOVERED FLAT-PLATE COLLECTORS UNDER SYRIAN CLIMATIC CONDITION

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Abstract

There are several methods to heat or to reduce overheating of inner spaces in buildings, one of the promising techniques is the use of simple uncovered roof collector as nocturnal radiators in areas where clear sky condition are frequent. Simple uncovered roof collectors acting also as night-sky radiators and heat exchangers with the ambient air offer a simple and cheap solution to maintain human comfort in buildings among Syrian climatic conditions.

A computerized design method has been worked out to predict the heating and cooling performance of roof collectors. The method is based on mass flow network model, the simultaneous model is built up from thermal and hydrodynamical part models connected by dual-function elements.

The solar system consists of two parts: the hydrodynamical part model is a mass flow network model, modelling also the executional functions of the control system and serving to determine the branch mass flows.

The temperature variations of the flowing medium are determined by means of the thermal part model built up from the heat flow network model of the storage, heat exchanger and consumption.

I. Part model of the collector

The construction of the uncovered flat-plate collector is shown in Fig. 1.

Along the flow direction the collector is lumped into discrete parts (Fig. 1) and for each the heat transport in the direction perpendicular to the base plane is described by a nodal type heat balance equation as follows:

$$I_o \cdot \Delta x_j(2h + D) = h_{in}D_{in} \cdot \pi \cdot \Delta x_j \cdot (T_{wj} - T_{avj}) + 2h \cdot k_1 \cdot \Delta x_j \cdot (T_{avj} - T_0) + \\ + D \cdot \pi/2 \cdot k_1 \cdot \Delta x_j \cdot (T_{wj} - T_0) + h_{1j} \cdot 2h \cdot \Delta x_j(T_{av} - T_j) + \\ + h_{sj} \cdot = \Delta x_j \cdot D \cdot \pi/2 \cdot (T_{wj} - T_s) \qquad (1)$$

Where: $k_1 = h_0 + k$

$$h_{1j} = \varepsilon \cdot \sigma_0 \cdot (T^4_{avj} - T^4_s)/(T_{avj} - T_0)$$

$$h_{sj} = \varepsilon \cdot \sigma_0 \cdot (T_{wj} + T_s) \cdot (T^2_{wj} + T^2_s) .$$

In the j-th discrete part of ΔA_j surface the heat-carrying medium is warmed up by $\Delta T_{l_j} = (\dot{Q}_j/\dot{m} \cdot c_p)$ (2), where $\dot{Q}_j$ can be calculated using the resultant tube wall temperature from (1) and either a predicted average fluid temperature in the j-th section or the fluid outlet temperature from the $(j - 1)$-th part.

The algorithm is advancing along flow direction step-by-step solving the nonlinear heat balance Equation (1).

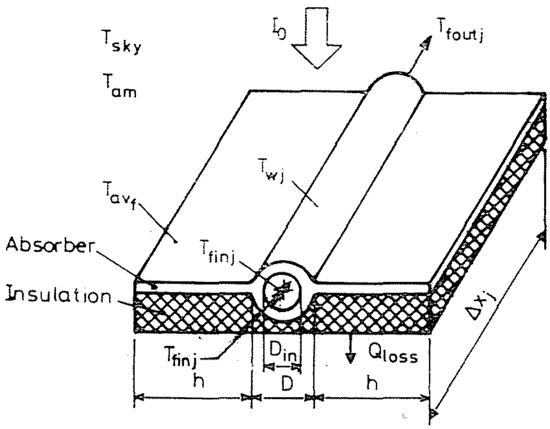

Fig. 1. Element of uncovered collector

II. Thermal part model of the storage

The storage unit may be a fluid tank, a solid body (wall or rock-bed) or a phase change material. In fluid storage units the fluid usually gets mixed if the heat is introduced from below.

Fluid tanks may also operate with significant degrees of stratification, that is, with fluid of not uniform temperature along the vertical dimension of the tank.

Partially stratified water storage tank with two sections is shown in Fig. 2.

If we analyze n-section water storage tank, each section considered to be at uniform temperature $T_{s,1}$, $T_{s,2} \ldots T_{s,n}$, and we define two control functions, one for the collector side F_c and the other for the load side F_L.

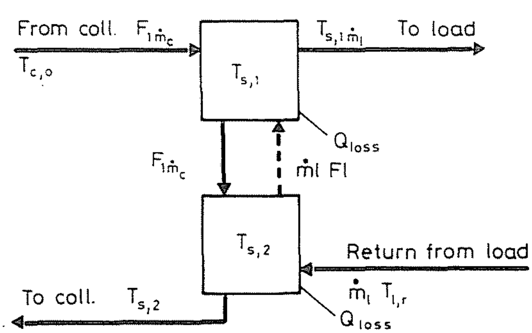

Fig. 2. Partially stratified water storage tank

For the collector, in the section i of the tank, we can define $F_c(i)$ such that:

$$F_c(i) = \begin{cases} 1 & \text{if} \quad T_{i-1} > T_{c,0} > T_i \\ \emptyset & \quad \text{otherwise} \end{cases}$$

and for the load, we can define $F_L(i)$ such that:

$$F_L(i) = \begin{cases} 1 & \text{if} \quad T_i > T_{L,R} > T_{i+1} \\ \emptyset & \quad \text{otherwise.} \end{cases}$$

With these definitions of $F_c(i)$ and $F_L(i)$ the energy balance for section i of a n-section tank is:

$$\dot{m} \cdot c_p \cdot dT(i)/dt = (\dot{m} \cdot c_p)_c \left\{ F_c(i) \cdot (T_{c,0} - T_i) + (T_{i-1} - T_i) \sum_{j=1}^{i-1} F_{cj} \right\} +$$

$$+ (\dot{m} \cdot c_p)_L \left\{ F_L(i) \cdot (T_{L,R} - T_i) + (T_{i+1} - T_i) \sum_{j=(n-i+1)}^{n} F_{Lj} \right\} +$$

$$+ U_i \cdot A_i \cdot (T_a - T_i) \, .$$

Applying this equation, for finite increments of time, we have $\Delta T_s(i)$, (the variation of temperature in the section i of the tank).

Nomenclature

a:	absorptance of absorber	
C_p:	specific heat of fluid	(kj/kg · k)
D:	tube diameter	(m)
D_{in}:	tube inner diameter	(m)
h_0:	heat transfer coefficient between air and absorber	(w/m^2 k)
h:	distance between each 2nd tubes	(m)
h_1:	heat transfer coefficient at $x\_$ div. along the plate	(w/m^2 k)
h_3:	heat transfer coefficient along the tube wall	(w/m^2 k)
h_{in}:	heat transfer coefficient between fluid and tube wall	(w/m^2 k)
I_0:	solar radiation intensity	(w/m^2)
K:	conductive heat transfer	(w/m^2 k)
$\dot{m}$:	fluid mass flow rate	(kg/s)
$\dot{m}_c$:	the collector flow rate	(kg/s)
$\dot{m}l$:	the load flow rate	(kg/s)
$\dot{Q}$:	heat transported by fluid	(w)
T_{am}:	ambient temperature	(k)
T_0:	ambient temperature	(k)
T_{av}:	plate average temperature	(k)

T_{fav}:	fluid average temperature	(k)
T_{fin}:	fluid inlet temperature	(k)
T_{fout}:	fluid outlet temperature	(k)
T_s:	sky temperature	(k)
T_w:	tube wall temperature	(k)
$T_{c,0}$:	the outlet fluid temperature from collector	(k)
$T_{1,r}$:	the return fluid temperature from load	(k)
U:	overall heat transfer coefficient	
ε:	emittance	
σ:	Stefan Boltzmann constant.	

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