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Optimum selection of solar collectors for a solar-driven ejector air conditioning system by experimental and simulation study

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ABSTRACT

In this paper, three different solar collectors are selected to drive the solar ejector air conditioning system for Mediterranean climate. The performance of the three selected solar collector are evaluated by computer simulation and lab test. Computer model is incorporated with a set of heat balance equations being able to analyze heat transfer process occurring in separate regions of the collector. It is found simulation and test has a good agreement. By the analysis of the computer simulation and test result, the solar ejector cooling system using the evacuated tube collector with selective surface and high performance heat pipe can be most economical when operated at the optimum generating temperature of the ejector cooling machine.

1. Introduction

The principal operational characteristic of solar powered ejector cooling system is the use of low temperature thermal energy obtained as free solar energy. The heat from solar collector system is used to produce vapor to drive a ejector cooling system. In ejector refrigeration systems, when the high-pressure vapor passes through the primary nozzle, it expands to a supersonic flow, providing a low pressure region inside the evaporator compartment. The low-pressure secondary refrigerant from the evaporator is thus entrained and a refrigeration effect produced [1].

A number of studies have been made to investigate the solar powered ejector cooling system. Pridasawas and Lundqvist [2] compared the performance of different type solar collectors for ejector system with iso-butane as a refrigerant by year-round dynamic simulation. It is concluded that vacuum tube collector is more economically competitive than flat-type collectors due to the lower amount of auxiliary heat required. In many developed steam ejector systems, the generator temperature is designed to within 120–140 °C [3,4]. These systems would not be efficient when used with solar collectors for vapor generation because the efficiency of the solar collectors reduces with the increase of its temperature. Therefore, systems using solar energy require relatively low generator temperatures.

The objective of the present study is to select suitable solar collector to drive the ejector cooling system for Mediterranean climate. The 5 kW steam ejector using water as refrigerant was designed and manufactured for the operating conditions in a range that would be suitable for air-conditioning application using solar collectors. The outlet temperature of solar collector system operated at 100 °C to drive the ejector cooling system [5]. In the present study, a number of popular solar collector products on the market have been reviewed. Three different types of evacuated tubes with high efficiency have been selected as potential solar devices to drive the solar ejector air conditioning system for Mediterranean climate. Solar collectors used for solar cooling should meet the following criteria: (1) high efficiency, (2) long durability and (3) competitive price.

At the design condition of a collector outlet temperature of 100 °C, flat plate collectors cannot operate at high efficiency. Solar tracking concentrating solar collectors achieve high performance, but are expensive and require maintenance of the moving parts. Three different evacuated tube solar collectors with high efficiency were chosen for assessment. These are the TMA600 heat pipe solar collector with single borosilicate glass cover [6], the TZ58-1800 heat pipe collector with double-glass borosilicate glass cover [7] and the Cortec2 direct flow solar collector with single borosilicate glass cover [8]. These are selected on the basis of their suitability for operating on a Mediterranean climate.

2. Development of a mathematical model of evacuated tube solar collector

A simulation program was constructed based on the thermal performance analysis of the solar evacuated tube collector system in order to study the effect of modifying the operational and design parameters of the system. A theoretical model incorporating a set

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Nomenclature

A	absorber area of collector (m^2)
C	specific heat capacity (J/kg °C)
c	heat loss coefficient (W/m^2 °C)
D	mass flow rate (kg/s)
G	global solar irradiance (W/m^2)
Q	heat Energy (W)
T	temperature (°C)
Δt	temperature difference between fluid outlet and inlet (°C)
η	efficiency
Subscripts	
a	ambient air
ab	absorber
abs	absorption
de	demand solar energy heat for cooling system
g	gained heat of cooling fluid
in	inlet of cooling fluid
m	mean temperature of heat transfer fluid
o	optical efficiency of collector
out	outlet of cooling fluid
sc	incident heat for solar radiation
ref	reflection
t	top cover
tra	transmission

of heat balance equations was developed and used in the simulation model to analyze the heat transfer processes occurring in separate regions of the collector, i.e., the top cover, absorber and condenser/manifold areas.The heat process must achieve heat balance in order to work steadily. General assumptions in mathematical model are: ① steady state condition; ② negligible heat loss; ③ uniform temperature of the absorber; ④ heat balance of the each components. The heat process equations of a solar collector can be determined from related literature [9–11], and described briefly as the follows:

The heat processes in the top cover (Q_t) include three parts which are absorption (Q_{abs}), transmitting (Q_{tra}) and reflection (Q_{ref}) of solar energy, and given as:

$$Q_t = Q_{abs} + Q_{ref} + Q_{tra} \tag{1}$$

The obtain heat of absorber (Q_{ab}) is given by:

$$Q_{ab} = Q_{tra} - Q_{ab-t} \tag{2}$$

where Q_{ab-t} is the heat of the absorber dispersed through top cover (W).

The cooling fluid in the manifold obtains heat from the absorber according to the evaporation and condensation of working fluid in the heat pipe. The overall heat transferred from solar collector to cooling fluid is given as

$$Q_g = \sum_{i=1}^{n} Q_{g,i} = D \cdot C \cdot (T_{out} - T_{in}) \tag{3}$$

where $Q_{g,i}$ is heat transferred between a single solar collector and cooling liquid (W); C is specific heat of cooling fluid (J/kg °C), D is mass flow rate of cooling fluid (kg/s), and t_{out} and t_{in} is inlet and outlet temperature of the cooling fluid (°C). The outlet temperature of cooling water of solar collector system can be calculated by the above simulation steps. The system efficiency and heat transport energy were also be determined, and used to design the solar collector system.

The results obtained by running the computer program were used to evaluate the thermal performance of the solar collectors. For a normal solar collector, performance is usually evaluated using efficiency η, which is defined as the ratio of useful heat gained by the cooling fluid to the incident solar radiation over the same period. Eqs. (4) and (5) use these empirical values to determine the efficiency of the collector system. The efficiency of the collectors can be calculated using the following thermal performance equations:

$$\eta = \eta_0 - c_1 \cdot \frac{T_m - T_a}{G} - c_2 \cdot \frac{(T_m - T_a)^2}{G} \tag{4}$$

with

$$\eta = \frac{D \cdot C \cdot \Delta t}{A \cdot G} \tag{5}$$

where η_o is optical efficiency; T_a is ambient air temperature (°C); T_m is the average temperature of inlet and outlet temperature for the solar collector (°C); and G is global solar irradiance (W/m^2); C_1 and C_2 are the heat loss coefficient (W/m^2 °C).

The $\eta - (t_m - t_a)/G$ relations for different top cover conditions were investigated using the computer model, and the results are shown in Fig. 1. From Fig. 1, simulation results have the same trend: efficiency decreased as temperature difference increased. The TMA600 was found to be more stable than the other two collectors. At the low temperature, TZ58-1800 works better. At high temperature, TMA600 works better.

In the system design and Tunisia climate condition, the collector out temperature is 100 °C, average ambient temperature is 28 °C, and average solar radiation intensity is 700 W/m^2 at an angle of 20° facing south during summer [12]. As shown in Fig. 1, below around 60 °C, the thermal efficiency of the TZ58-1800 is the highest. Above 60 °C, the situation starts changes. At the design condition, the collector efficiencies of TMA600, Cortec2 and TZ58-1800 are 70%, 59% and 53% respectively.

Because of the 5 kW cooling load design and system design, the heat input required by the cooling system is 16.7 kW. It is approved that boiler heat input can be satisfied during cooling system laboratory test. Thus the required solar collector area is

$$\text{Area} = \frac{Q_{de}}{\eta \times Q_{sc}} \tag{6}$$

where Q_{de} is demand solar energy heat for cooling system (W); Q_{sc} is incident heat for solar radiation (W).

Calculated by the simulation program, the required area and price of TMA600 (with $F_{sol} = 1$), Cortec2 and TZ58-1800 at the design condition are shown in Table 1.

From the above result, it can be seen that the TZ58-1800's price is less than the half price of the other two collectors. For the collector maintenance, TMA600, Cortec2 and TZ58-1800 can work over 15 yr. Compared with direct flow collector, Cortec2, the heat pipe collector TMA600 and TZ58-1800 have added advantages including:

(1) Ease of evacuated tube replacement without the need for system drain-down should tube breakage occur.
(2) Lower water side pressure drop.