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# THERMAL CHARACTERIZATION OF A SOLAR PARABOLIC TROUGH COLLECTOR FORADSORPTION REFRIGERATION APPLICATION SYSTEM

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**ABSTRACT**

Thermal and structural design of a solar parabolic trough collector for adsorption refrigeration application system is carried out and the thermal performance is evaluated experimentally. The results signify that the solar parabolic trough collector is a promising option for providing energy required to drive the proposed adsorption refrigerator designed for air cooling of a room. Energy balance analysis of the collector tube reveals that heat loss by radiation and convection to the surrounding is significant for the case of a bare collector tube considered in this study. Use of a collector tube covered with an evacuated tube transparent to solar radiation would aid to develop thermally efficient parabolic trough collectors for adsorption refrigeration application systems.

**Key words:** Solar energy**,** Solar parabolic trough collector, adsorption refrigeration, thermal design.

**NOMENCLATURE**

$A_a$ Aperture area
C Heat capacity of adsorbent bed J/K
CR Concentration ratio
$C_p$ Specific heat capacity, J/kg K
m Mass, kg
*m* Mass flow rate, kg/s MF Mass fraction
Q Thermal load, W
$I_b$ Incident solar beam radiation, $W/m^2$
$U_o$ Overall convective heat transfer coefficient,$W/m^2$ K
T Temperature, K

t Time, s
x local coordinate in the steam-wise direction, m

**Subscripts**
in Inlet
out Outlet
w Wall
ad Adsorbant ref Refrigerant

**Geek symbols**
α Absorptivity
θ Dimensionless temperature, $\frac{T_w}{T_{out} - T_{in}}$
γ Thermal efficiency
ε Emissivity
ρ density, $kg/m^3$
σ Stefan-Boltzmann constant, $5.67\times10^{-8}$ $W/m^2 K^4$
τ Transmissivity

## 1.INTRODUCTION

The consequences resulting from depletion of fossil fuels and emancipation of pollutants caused by burning of fossil fuels into the environment have becomes a major concern of scientists, engineers and environmentalists. This paves the way to explore the potential of alternative and eco–friendly energy resources. Solar energy, due to its abundance and more even distribution in nature than other types of renewable energy sources, has been identified as a potential alternative and eco-friendly energy source capable to meet future demand and secure long term sustainable energy supplies. With ever increasing global warming, demand for maintenance of human comfort environment shoot-up markedly. The conventional vapor compression refrigeration systems, employing halogenated refrigerants and run on conventional energy, used for producing cooling effect have witnessed detrimental impact on environment. As an alternative to this, recently, adsorption refrigeration has been reported in literature.

Headley et al. [1] used a compound parabolic concentrating solar collector of concentration ratio 3.9 and aperture area $2m^2$ to power an intermittent solid adsorption refrigerator and ice maker using activated charcoal (carbon) as the solid adsorption refrigerator and ice maker using activated charcoal (carbon) as the adsorbing medium and methanol as the working fluid. Up to 1 kg of ice at an evaporator temperature of -6ºC was produced, with the net solar coefficient of performance being of the order 0.02. Maximum receiver/adsorbent temperature recorder was 154ºC on a day when the insolation was 26.8 $MJ/m^2$ . The temperatures in excess of 150ºC are undesirable since they favour the conversion of methanol to dimethyl ether, a non-condensable gas inhibits both condensation and adsorption. The major advantage of this system is its ability to produce ice even on overcast days (insolation – $10MJ/m^2$).There was excessive heating capacity in the system, and only 2% of the incident solar radiation was converted to the refrigeration effect. The system as it stands is therefore not economically viable. However, the cost of heating using CPC is about half as expensive as the cost of heating using electrical power. The CPC is therefore a natural candidate for industrial process heat generation in temperature region 80ºC to 200ºC.

Wang [2] experimentally investigated a double stage, four bed, non-regenerative adsorption chiller powered by solar/waste heat sources between 50ºC and 70ºC.The prototype studied produced cold water at 10ºC and have cooling power of 3.2kW with a COP of 0.36, when the heating source and sink had a temperature of 55ºC and 30ºC,respectively. A theoretical study of an adsorption system, meant for providing necessary cooling effect in the room, is then carried out to estimate the heat input required to operate the adsorption system. An adsorption system contains mainly condenser, evaporator, expansion valve, adsorbent bed, some adsorbent and some adsorbate. Adsorbents are used usually in the form of spherical pellets, rods, moldings, or monoliths with hydrodynamic diameters between 0.5 and 10 mm. There are several combination of adsorbent and adsorbate pair. In this analysis, activated charcoal and methanol pair is considered.

Kalogirou et al., [3] applied for a patent of a silican gel-water adsorption chiller driven by a low temperature heat source that was used to cool a grain depot in th Jiangsu Province, China. This chiller has two identical chambers and a second stage evaporator with methanol as working fluid. Experiments performed when hot water at 85ºC was used to drive the chiller, resulted in a cooling power close to 4.96 kW, with the corresponding COP around 0.32.

Khattab[4] presented the operation of a simple structure, low cost solar-powered adsorption refrigeration module with the solid adsorption pair of local domestic type charcoal and methanol.

Kim and Seo [5] investigated numerically and experimentally, the thermal performance of a glass evacuated tube solar collector. The solar collector considered consisted of a two-layered glass tube and an absorber tube. Air is used as the working fluid. The length and diameter of this glass tube are 1200 and 37 mm, respectively. Four different shapes of absorber tubes are considered, and the performances of the solar collectors are studied to find the best shape of the absorber tube for the solar collector. Beam irradiation, diffuse irradiation, and shade due to adjacent tubes are taken into account for a collector model to obtain a realistic estimation.

A solar powered compound system used for heating and cooling was developed and successfully implemented in a golf course in Taiwan. An integrated two bed, closed type adsorption chiller working on silica gel/water system was developed for the same by Industrial Technology Research Institute in Taiwan. The solar powered system used to provide hot water at 50ºC to the dormitories and to provide chilled water to the restaurant. It had a cooling power of 9kW and a COP of 0.37.It also had a specific cooling power of about 72W/kg. However, in the field tests performed from July to October, the average cooling power was found to be 7.79kW and average COP of 0.403, this was investigated by Chang et al.[6].

The central attention of this study is to design and analyze, experimentally, the thermal performance of a solar parabolic trough collector intended to use as an energy source for driving an adsorption refrigeration system. The energy input for adsorption refrigeration depends on cooling load requirements and, therefore, the first step in the design procedure was to decide the capacity of the system. With this in objective, the cooling load requirement for a room of 200×300×480 cm size with four occupants and the accessories was calculated and energy input to drive the system was estimated. It must be noted that, this study, an adsorption system was chosen to produce the required cooling effect because adsorption system is free of moving parts. The load on an air-conditioning system can be divided into the following sections: (i) sensible transmission through glass,(ii) solar gain through glass,(iii) internal heat gains, (iv) heat gain through walls, (v) heat gain through roof and (vi) ventilation and/ or infiltration gains. Heat gains are calculated and displayed in table shown below.

Table.1 Heat gain from various elements

| Heat Gain | Watt | % |
|---|---|---|
| 1. Sensible transmission through glass | 100 | 2.5 |
| 2. Solar gain through glass | 1600 | 39.6 |
| 3. Internal | 1245 | 30.9 |
| 4. External walls | 630 | 15.6 |
| 5. Roof | 240 | 5.9 |
| 6. Ventilation | 220 | 5.5 |
| Total | 4035 | 100% |

## 2.ADSORPTION REFRIGERATION SYSTEM

In an ideal cycle depicted in Fig.1, point 1indicate that the adsorbent is at low temperature $T_d$ and at low pressure $P_d$ (evaporating pressure). Process 1- 2 represents heating of adsorbent along with adsorbate. While heating the adsorbent, the connection between the adsorbent bed and condenser is made open and the progressive heating of the adsorbent from 2 to 3 causes some adsorbate to be desorbed and its vapour to be condensed (point c). When the adsorbent reaches its maximum temperature $T_3$ desorption ceases. The liquid adsorbate is then transferred into the evaporator (point d) and the adsorbent bed is closed and cooled. The decrease in temperature from $T_3$ to $T_4$ results in pressure drop from $P_c$ to $P_d$. The adsorbent bed is then connected to the evaporator and vapour from the evaporator reaches the adsorbent bed and gets adsorbed while the adsorbent bed is being cooled from 4 to 1. During this cooling period heat is withdrawn to decrease the temperature of the adsorbent.

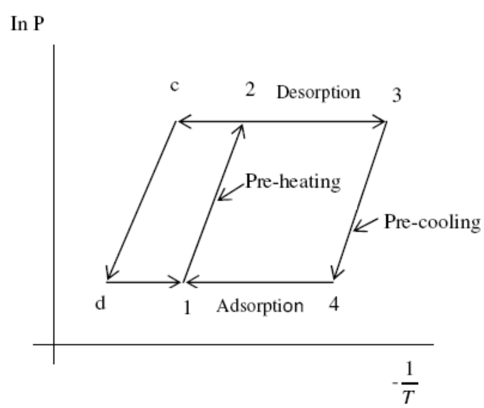


FIG.1 THE IDEAL ADSORPTION COOLING CYCLE

## 3.SOLAR COLLECTORS

Two main categories of solar collectors are solar collectors without concentrators and solar collectors with concentrators. Flat plate collector is an example of solar collector without concentrator. Parabolic dish and parabolic trough collectors are the examples of solar collectors with concentrators. Parabolic trough power plants use parabolic trough collectors to concentrate the direct solar radiation onto a tubular receiver. Parabolic troughs can be used to provide process heat for industrial processes at lowest cost compared to any other solar collectors. However, they are less efficient when sky is cloudy or in the absent direct solar radiation. Since air cooling of buildings becomes necessitate under hot climate, under such climatic conditions parabolic trough collector would be a viable option. This logical reasoning leads to the selection parabolic trough collector for generate requited heat input.

**Thermal design**

The primary function of Parabolic Trough Collector (PTC) is to absorb and transfer the concentrated energy to the fluid flowing through the receiver tube of the trough collector. The heat gained by the fluid is utilized for heating the adsorbent bed during the heating phase of the bed. The amount of heat required to heat the adsorbent bed depends on the heat capacity of the bed and temperature rise above the ambient. The adsorbent bed is assumed to be made of thin walled stainless steel tube of diameter 3 inch, containing copper tubes of ½ inches diameter. Thus, the total sensible thermal capacity of an adsorber is the sum of the thermal capacity of metals (steel and copper), adsorbent, and average amount of liquefied refrigerant.
the adsorbent bed and gets adsorbed while the adsorbent bed is being cooled from 4 to 1. During this cooling period heat is withdrawn to decrease the temperature of

$$C=(m\times c_p)_{steel}+(m\times c_p)_{Cu}+(m\times c_p)_{ad}+ 0.5(MF_{max}+MF_{min})\times(m\times c_p)_{ref} \quad (1)$$

The sensible heating rate for the adsorber during the heating phase is calculated as:

$$Q_{sh}=\frac{C\times T}{t} \quad (2)$$

Where, T is the required temperature rise and‘t’ is the time for heating the adsorber.

The latent heat rate during the heating phase is given by:

$$Q_l=\frac{m_r\times(MF_{max}-MF_{min})\times h_{fg}}{t} \quad (3)$$

The total heat input to the adsorber during the heating phase is:

$$Q=Q_{sh}+Q_l \quad (4)$$

The total heat input to the adsorber is calculated as 5462.35W.

**Selection of PTC dimension:** The PTC dimension is chosen in accordance with the availability of open space on the roof floor of Mechanical engineering building, College of Engineering, Trivandrum. The description of the PTC is given in Table 2. The thermal performance the PTC is determined in terms its optical efficiency, instantaneous efficiency and the system efficiency. The Optical efficiency, $\eta_o$, is the ratio of energy absorbed by the absorber to the energy incident on the concentrator aperture as given by:

$$\eta_o=\frac{Q_a}{I_b\times A_a} \quad (5)$$

Table 2 Specifications of the PTC

| Sl. No. | Item | Value |
|---|---|---|
| 1 | Geographical location | 8° 30' 24" N / 76° 57' 24" E |
| 2 | Collector aperture | 1.2 m |
| 3 | Collector length | 3 m |
| 4 | Collector aperture area | 3.6 $m^2$ |
| 5 | Receiver area | 0.142 $m^2$ |
| 6 | Concentration ratio | 25.48 |
| 8 | Rim angle | $90^o$ |
| 8 | Focal distance | 0.3 m |

The optical efficiency range of most solar concentrators is 0.6 - 0.7[7]. The optical efficiency depends on several factors and its dependence on those factors is expressed as:

$$\eta_o = K(\phi)\_\rho(\tau\alpha)_n\_\gamma \qquad (6)$$

This definition of the optical efficiency allows a clear distinction of the factors contributing to it. The first bracketed term accounts for all the incidence angle related effects. The second bracketed term represents the optical effects of the physical properties of the materials used in receiver and reflector construction. The last term, the intercept factor, $\gamma$, contains the effects of various manufacturing, operation and materials imperfections/errors. $\phi$ is the angle of incidence of the sun's rays on the collector aperture measured from the normal to trough aperture.

The useful thermal energy, $Q_u$, delivered by the concentrator is obtained from the energy balance analysis.

$$Q_u = \eta_o I_b A_a - U_o A_{ra} (T_{rs} - T_\infty) - \sigma\varepsilon A_{ra} (T_{rs}^4 - T_\infty^4) \qquad (7)$$

The instantaneous thermal efficiency, $\eta$, of the PTC is expressed as

$$\eta = \frac{Q_u}{I_a A_a} = \eta_o - \frac{U_o (T_{rs} - T_\infty)}{I_a CR} - \frac{\sigma\varepsilon (T_{rs}^4 - T_\infty^4)}{I_a CR} \qquad (8)$$

The system thermal efficiency, $\eta_s$, is determined using the expression;

$$\eta_s = \frac{\dot{m} c_p (T_{out} - T_{in})}{I_b A_a} \qquad (9)$$

## Structural design

Structural design mainly involves design of structure for supporting the reflector and the receiver. Oxy-acetylene flame hardened mild steel angle of 1 inch size is used for the construction of structure. The mild steel angle was bent into the desired parabolic shape as described in table 2. It is flame hardened to prevent the structural deformation. The structural design is carried out by taking into consideration of various loads such as wind load, weight of receiver together with the weight of water, weight of receiver support and weight of mirror material. Finally, the static load on the bearing is calculated.

## 4.RESULTS AND DISCUSSION

Experiments are conducted to estimate the performance of the PTC. Figure 2shows the photo graph of the experimental setup. The receiver tube is made of copper and Acrylic sheet mirror is used as the reflecting surface. The wall temperature of the receiver tube at four different locations is measured by T-type thermocouples fixed into the blind hole made on the tube. A high conducting thermal bond is used to fix the thermocouples into the holes. The inlet and outlet temperature of the water flowing through the receiver tube is recorded using thermocouples placed at the inlet and outlet sections of the tube. All thermocouples are connected to a data acquisition system. The experiments are conducted during the second week of March 2013, from 11 to 14 hr, when intensity of solar radiation does not expect to vary appreciably. As per data from the Meteorological Department, (weather Station: Thiruvananthapuram (latitude 8-28N, longitude 76-57E), the average intensity of direct solar radiation is 865W/$m^2$.

FIG. 2 PHOTOGRAPH OF THE PARABOLIC COLLECTOR

Figure 3 depicts the variation of dimensionless temperature as a function of dimensionless distance of the receiver tube for three Reynolds numbers. It is observed that tube wall temperature increases along the length up to half-length of the tube, beyond that the temperature decreases. For constant heat flux wall boundary condition, the fluid temperature is expected to increase along the flow direction. Clearly, the wall temperature must increase along the length of the tube However, a small drop in temperature at outmost section due to end losses is appreciable. It is, therefore, inferred that the decrease in wall temperature beyond half-length of the tube is due to misalignment of the receiver tube with the focus of the parabolic concentrator. As far as a parabolic trough collector is concerned a small deviation of receiver tube from the focus will result in significant loss in energy absorbed by the receiver tube. In view of this, the experimental result implies that an adjusting mechanism needs to be incorporated with the existing setup to align the receiver tube exactly with the focus of the parabolic collector.

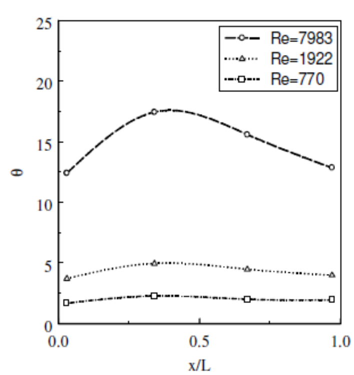


FIG.3 VARIATION OF DIMENSIONLESS TEMPERATURE AT DIMENSIONLESS DISTANCE FOR DIFFERENT REYNOLDS NUMBER

Also, an important conclusion from the fig.3 is that decrease in wall temperature is the largest for highest Reynolds number or mass flow rate considered in this study. The reason for this observed trend is attributed to the fact that cooling rate will be larger at higher flow velocity.

The instantaneous thermal efficiency obtained at different mass flow rate is represented diagrammatically in fig.4. The instantaneous efficiency increases marginally with mass flow rate as seen in fig. 4. An efficiency of 65% obtained in the present study is close to the value reported in literature [8].

The increase in instantaneous thermal efficiency with increasing mass flow rate is predictable because higher mass flow rate results in lower wall temperature, consequently convection and radiation losses to the cold ambient would expected to decrease with increase in mass flow rate. The reduction in heat losses to the surrounding causes increase in instantaneous efficiency, as described by eq. (8). It is worth to mention at this point that the estimated values of instantaneous efficiency is obtained for the case without a glass envelop surrounding the receiver tube.

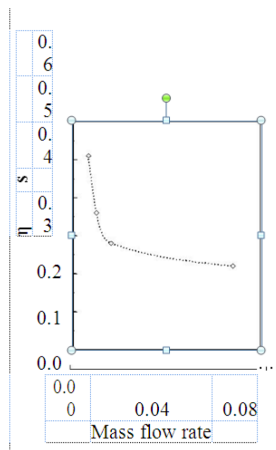


FIG.4 VARIATION OF INSTANTANEOUS THERMAL EFFICIENCY WITH MASS FLOW RATE

Improvement in instantaneous efficiency is achievable with the use of an evacuated glass envelop, the purpose of which is to minimize the energy losses to the cold ambient.
The system thermal efficiency which is defined s the ratio of energy gained by the receiver tube fluid to the energy incident on the concentrator aperture is estimated for different mass flow rate and the dependence of system thermal efficiency on the mass flow rate is expressed in fig. 5.

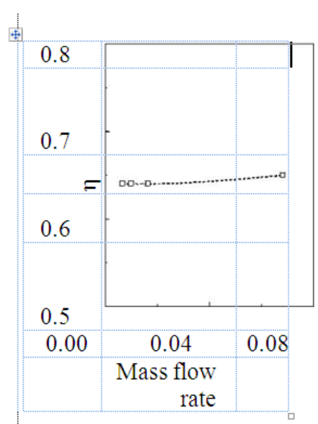


FIG.5 SYSTEM THERMAL EFFICIENCY VS MASS FLOW RATE

The result reveals that system efficiency is the maximum at minimum mass flow rate. It must be noted that lower mass flow leads to higher tube wall temperature. The higher wall temperature and more residence time available for the fluid at lower mass flow rate results in larger energy transfer rate. The overall system thermal efficiency around 54% at mass flow rate of 0.007 kg/s is an indicative of efficient thermal design. As mentioned earlier, improvement in system efficiency would be possible if the absorber tube is covered with an evacuated glass envelop.

## 5. CONCLUSIONS

Thermal and structural design of a solar parabolic trough collector for adsorption refrigeration application system is carried out and the thermal performance is evaluated experimentally. The important conclusions of the study are:

1. The instantaneous thermal efficiency increases marginally with mass flow rate. The maximum value of instantaneous efficiency obtained is close to the value reported in literature.

2. The overall system thermal efficiency is the maximum at minimum mass flow rate. The overall system efficiency around 54% at mass flow rate of 0.007 kg/s, even in the absence of evacuated glass envelop, is an indicative of efficient thermal design

3. Improvement in system thermal efficiency would be possible if the absorber tube is covered with an evacuated glass envelop.

4. The present study reveals that parabolic trough collector can provide adequate thermal energy to drive adsorption cooling system for air cooling application of a room during day time when intensity of solar radiation is at its peak. A hybrid design would be a feasible option for driving the air cooling system even when solar flux is weak or absent.

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