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# MODELING OF A SOLAR ASSISTED WATER-LIBR ABSORPTION REFRIGERATION SYSTEM FOR SUMMER AIRCONDITIONING OF AN OFFICE ROOM

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## ABSTRACT

Exploitation of solar energy for achieving low temperature is an active area of research at present owing to the concern over suitable alternatives to synthetic refrigerants and large electricity consumption by conventional vapour compression systems. Present work theoretically design and analyze the performance of a solar flat-plate collector-driven water-Lithium bromide absorption refrigeration system for comfort conditioning of an office room during working hours of summer months. Mathematical models of absorption chiller, flat-plate collector, water storage tank with auxiliary heating unit and conditioned-space psychrometry have been coupled with real weather data of Kolkata, India. Ten collectors of 1.5 m x 1 m each, alongside a storage tank of 15 $m^3$ capacity have been found to be adequate for the selected space. Absorption system exhibits an optimum generator temperature corresponding to maximum COP for any condenser-evaporator temperature combination and corresponding contour maps have been presented. Integrated system COP has also been found to attain maxima for a particular solar insolation level. It is possible to maintain collector efficiency greater than 40% almost throughout the day. Supply tank water temperature has been found to be slightly higher at the end of the day compared to the early morning for summer months, whereas it remains slightly lower during post-monsoon months. Use of auxiliary heaters has been suggested rather than augmenting the number of collectors.

**Keywords:** Absorption refrigeration, Solar energy, Flat plate collector, Psychrometry, Airconditioning.

## 1. INTRODUCTION

Serious environmental effects of synthetic refrigerants have forced scientists to plan for their gradual phase-out and look for suitable replacements. However, the immediately-available alternates of CFCs, in the form of HCFCs and HFCs, may have very high global warming potential and also take some part in ozone layer depletion [1-3]. Balghouthi *et al.* [4] demonstrated environmental pollution, global warming and stratospheric ozone depletion caused by extensive use of conventional vapour compression air-conditioning systems during the summer. Knowledge base about natural refrigerants like $H_2O$, $NH_3$ or $CO_2$ is yet to reach the level of practical applicability. Also the huge electricity requirements in developing countries, particularly in sub-tropical cities like Kolkata, owing to the uncontrolled use of airconditioners in peak summer is a major concern. It indirectly contributes to the environmental pollution and global warming. Hence the employment of solar-driven absorption refrigeration system with $H_2O$-LiBr as the working medium has been sough as a potent alternative.

The most attractive feature of such application is the demand is in phase with the availability of solar radiation and hence may result in large savings in primary energy production, while providing a much greener operation from environmental impact point of view.

The first law of thermodynamic analysis, concerned only with the conservation of energy, is still the most commonly used method for analysis of any thermal system. Asdrubali and Grignaffini [5] have done the experimental evaluation of the performances of a $H_2O$–LiBr absorption refrigerator under different service conditions. Banasiak and Kozio [6] have done a mathematical modeling of a LiBr–$H_2O$ absorption chiller. Accuracy of theoretical analysis of such system depends on successful evaluation of different property of the refrigeration system. Patek and Klomfar [7] presented a computationally efficient formulation of thermodynamic properties of LiBr–$H_2O$ solutions at vapor–liquid equilibrium as explicit functions of temperature and mixture composition.

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From the previous research works it is apparent that hardly any systematic theoretical effort exists to integrate solar flat plate collector with VARS and employ that for small-scale airconditioning applications. Most of the works till date generally concentrate upon large scale industrial applications and it is extremely important in current energy perspective to explore the capability of the system in small scale applications, particularly for the domestic comfort air conditioning purpose. Present study focuses on the development of a realistic model of a solar-assisted water-LiBr vapour absorption refrigeration system for comfort air conditioning of an office room during summer months.

## 2. THEORETICAL MODEL DEVELOPMENT

### 2.1 Modelling of VARS System

Figure 1 shows the schematic diagram of the complete solar-assisted LiBr- $H_2O$ absorption cooling system where 10 solar flat plate collector have been connected in parallel to power the absorption system. Load to the VARS has been provided in the form of cooling load of the office room.

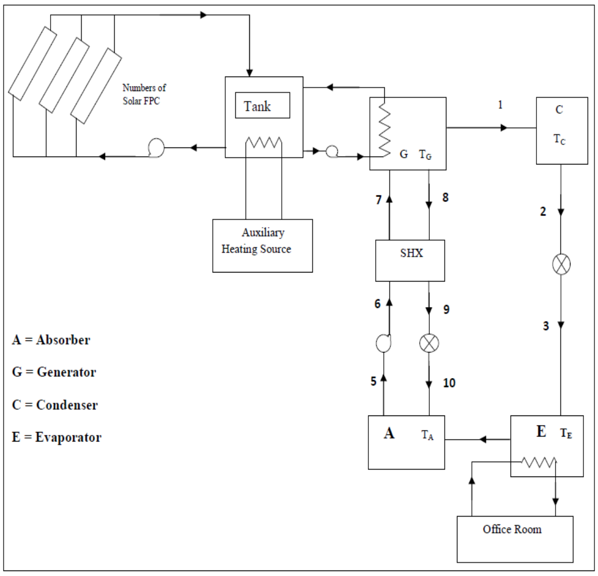


Fig.1. Schematic diagram of complete solar-assisted airconditioning system

A thermodynamic analysis of the system has been done by separately performing mass and energy balance across different components and considering the overall conservation as a whole with the help of following equations:

$$Q_C + m_R h_2 = m_R h_1 \quad (1)$$
$$Q_E + m_R h_3 = m_R h_4 \quad (2)$$
$$Q_G + m_{SS} h_7 = m_{WS} h_8 + m_R h_1 \quad (3)$$
$$Q_A + m_{SS} h_5 = m_{WS} h_{10} + m_R h_4 \quad (4)$$
$$m_{SS}(h_7 - h_6) = m_{WS}(h_8 - h_9) \quad (5)$$
$$W_P + m_R h_5 = m_R h_6 \quad (6)$$
$$W_P = (0.001) m_{SS}(P_C - P_E) \quad (7)$$
$$m_R + m_{WS} = m_{SS} \quad (8)$$
$$COP = \frac{Q_E}{Q_G + W_P} \quad (9)$$

### 2.2 Modelling of Solar Flat Plate Collector

Solar flat plate collectors made of GI absorber plate and with two glass covers of size 1.5 m × 1 m has been considered. Total incident solar radiation and total flux by solar FPC is estimated as a function of surface inclination and location [8, 9] by the following equations:

$$I_T = I_b r_b + I_d r_d + (I_b + I_d) r_r \quad (10)$$
$$S = I_b r_b (\tau\alpha)_b + \{I_d r_d + (I_b + I_d) r_r\}(\tau\alpha)_d \quad (11)$$

The amount of useful absorbed energy transferred to the storage fluid after occurrence of various modes of heat losses to the environment, overall heat transfer coefficient, collector efficiency and the outlet temperature of fluid flowing through the flow channels are calculated by different equations given below. To do the analysis following assumptions have been made.

a) Solar radiation on the collector is uniform and steady
b) Collector performs in steady – state condition
c) Absorber plate and the transparent covers are at uniform temperature
d) Energy loss through the transparent cover and bottom insulation and side insulation is one dimensional
e) Same environment temperature prevails all around the collector.

Accordingly the top loss coefficient based on collector gross area is given by [10]:

$$U_t = \left[ \frac{M}{\frac{c}{T_p}\left(\frac{T_p - T_a}{M + f}\right)^{0.252}} + \frac{1}{h_a} \right]^{-1} + \frac{\sigma(T_p + T_a)(T_p^2 + T_a^2)}{[\varepsilon_p + 0.0425(1-\varepsilon_p)]^{-1} + \frac{2M + f - 1}{\varepsilon_c} - M} \quad (12)$$

Now the bottom, side and overall loss coefficient, total heat lost from the collector, heat removal factor, and useful heat gain by the fluid flowing medium, collector efficiency are estimated by following equations:

$$U_B = \frac{K_i}{\delta_B} \quad (13)$$
$$U_s = \frac{(l + w) H_{bb} K_i}{l w \delta_s} \quad (14)$$
$$U_l = U_t + U_B + U_s \quad (15)$$
$$Q_l = U_l A_p (T_{pm} - T_a) \quad (16)$$

$$F_R = \frac{m_f c_{pf}}{U_l A_p}\left[1-\exp\left\{-\frac{F' U_l A_p}{m_f c_{pf}}\right\}\right] \quad (17)$$

$$Q_u = F_R A_c\left[S - U_l\left(T_{fi} - T_a\right)\right] \quad (18)$$

$$T_{pm} = T_{fi} + \left[\frac{Q_u}{F_R A_c U_l}\right]\left(1 - F_R\right) \quad (19)$$

$$T_{fo} = T_{fi} + \frac{Q_u}{m_f \times c_{pf}} \quad (20)$$

$$\eta_I = \frac{Q_u}{A_p \times I_T} \quad (21)$$

### 2.3 Generator Heat Input Assembly

An ideally-insulated hot water storage tank of 15 $m^3$ capacity is designed from where water is supplied through a piping system with negligible piping loss to a few numbers of solar flat plate collectors, connected in refrigeration system to give the desired heat input. Following transient approach, the temperature variation of the tank is calculated throughout the day time by using the following equation:

$$T'_{\text{tank}} = T_{\text{tank}} + \frac{\Delta t}{m_t c_{pt}}\left(Q_u - Q_G\right) \quad (22)$$

### 2.4 Psychrometric Calculations for the Room

A schematic representation of the conditioned office space has been shown in Fig. 2 and detailed specifications used for the calculations have been shown in appendix. Some assumptions are made during the design of the room and its load calculation which are given below.

- Only 10% of the supply duct is outside the conditioned space.
- No return duct outside the conditioned space.
- A total load due to fan hp, leakage, safety factor etc. is taken as 5% of the room sensible heat.
- Only the west wall is exposed to direct sun light.
- Rooms below and above the conditioned space and on the outer side of the partition wall and also the corridor on the outer side of the west wall are not air conditioned. All these spaces are assumed to be at a temperature of 3 °C less than the ambient

The actual load on the evaporator, the apparatus dew point (effectively required evaporator temperature) and the supply condition of the air entering office room at design condition is calculated through psychrometric calculation [11,12] by using the following equations:

$$RSH = Q_{gr} + Q_{ga} + Q_{gc} + Q_w + Q_{floor} + Q_{pw} + Q_s + Q_{light} + Q_{fan} + Q_{appl} + Q_{\text{inf}} \quad (23)$$

$$Q_L = \frac{LHL}{Person} \times TotalOccupants \quad (24)$$

$$Q_{L\text{inf}} = INF_{Total}\left(\omega_{or} - \omega_{ir}\right)\frac{\rho_a \times c_{pa}}{60} h_{fg} \quad (25)$$

$$RLH = Q_L + Q_{L\text{inf}} \quad (26)$$

## 3. COMPUTATIONAL PROCEDURE

A computer simulation program has been developed along with the supporting indigenous $H_2O$-LiBr mixture property code with the help of the correlations proposed by Patek and Klomfar [7]. The solution heat exchanger effectiveness (ε) is assumed to be 0.8. Computer code is also developed to calculate the properties of water following IAPWS-IF97 [13]. The properties of air have been calculated with the help of some curve-fitted relations [14], which compare well with standard resources [8,9]. An iterative method [8] is followed to calculate the outlet temperature of the fluid stream under given conditions of solar radiation and information about date and time. Psychometric analysis has been performed to calculate the generator heat requirement depending upon the weather condition and comfort requirement of the targeted space over the entire working period of 8 AM – 17 PM. The whole air-conditioning system is then analyzed during the months of March to October with real weather data of Kolkata.

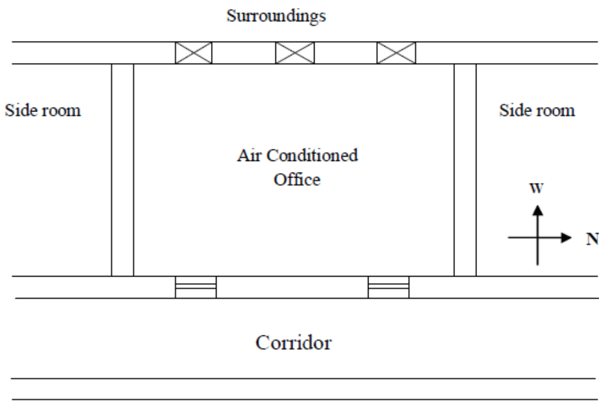


Fig.2. Schematic representation of the location of the conditioned space

## 4. RESULTS AND DISCUSSION

Figure 3 shows that for a fixed evaporator and condenser temperature (same as absorber temperature), as generator temperature increases the COP increases initially up to a maximum value and then decreases gradually. The generator temperature corresponding to this maximum COP indicates the optimum generator temperature and corresponding heat input into generator is lowest. Increase in condenser temperature forces an increase in the optimum generator temperature and also lowers the maximum system COP. Higher condenser temperature enhances the temperature range of the system. As the absorber temperature is increasing with condenser, mixture concentration in the solution circuit is also affected and hence the COP level comes down quite appreciably.

The reverse can be observed with increase in evaporator temperature. Higher evaporator temperature allows the system to operate over a lower temperature range and hence increases the maximum COP. Corresponding optimum generator temperature also decreases significantly. Hence the optimum generator temperature and concerned maximum COP is dependent on condenser-evaporator temperature combination.

A contour plot for selection of optimum generator temperature has been shown in Fig. 4 for any given condition of condenser and evaporator temperatures. Only a few selected contour lines have been shown for clarity. The desirable value of optimum generator temperature can be identified from the contours, which can ensure maximum value of COP corresponding to the design conditions. Corresponding contours of maximum possible COP is shown in Fig. 5 which gives an estimate about the theoretical limit of COP that can be achieved for any given combination of condenser and evaporator temperatures.

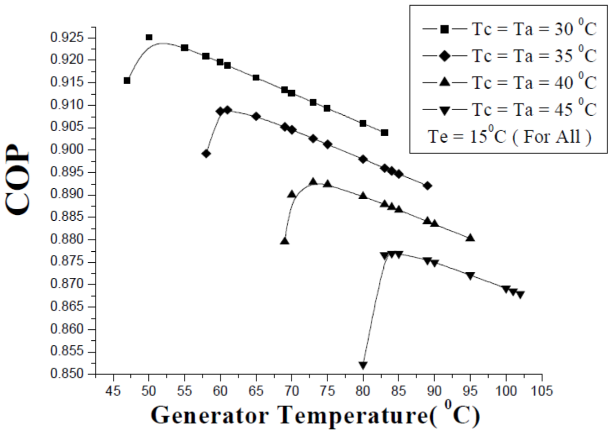


Fig.3. Effect of generator temperature on COP

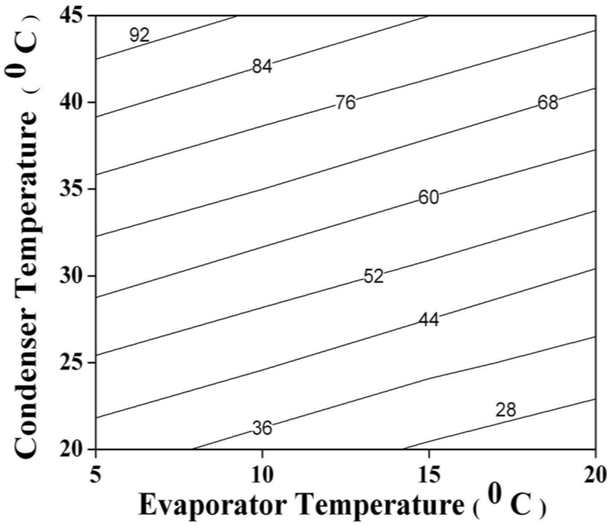


Fig.4. Contour of optimum generator temperature

Figure 6 shows the variation of COP of the solar assisted cooling system with the variation of solar radiation. It shows that with the increase of solar radiation the COP initially increases up to a maximum value and then decreases for a particular design condition (fixed cooling load and condenser, evaporator, absorber temperature). Increase in insolation continually increases the supply water temperature, which effectively increases the generator temperature of VARS part. As system COP achieves maximum for a particular generator temperature, COP variation with solar radiation also exhibits a peak. Efficiency of solar flat plate collector is another important factor to contribute on the performance of integrated system. Collector efficiency increases with solar radiation intensity, as is shown in Fig. 7. However the rate of increase of efficiency gradually comes down with insolation level. Higher radiation causes an increase in the plate mean temperature, which reduces the heat loss coefficients and hence fraction of energy lost from the collector decreases gradually, causing the increase in the collector efficiency. In fact, for very low insolation values, the hypothetical situation of negative efficiency may be calculated, which physically indicates that amount of energy extracted by collector fluid from sunrays is lesser than that lost to much cooler ambient. Such situation may appear during early morning and late afternoon. However, present model is concerned with office hour operation (8 AM – 5 PM) and hence such periods are not of consideration.

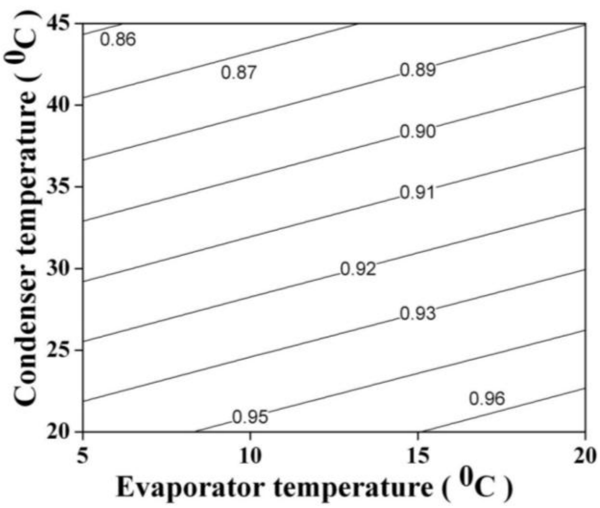


Fig.5. Contour of maximum COP at optimum generator temperature

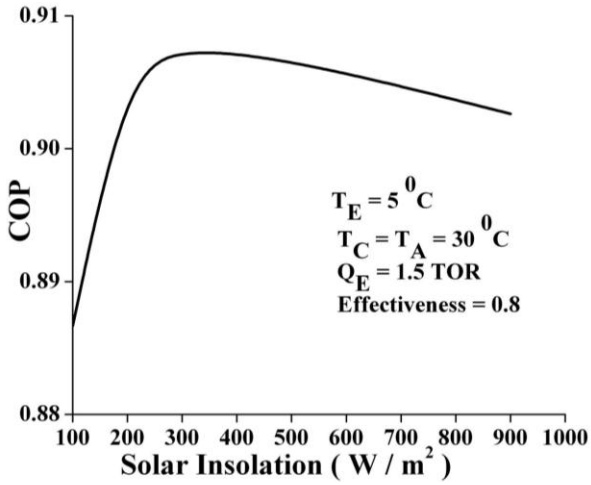


Fig.6. Variation of COP with solar radiation on fixed date and time

As the solar radiation level varies over a typical day, due to the change in collector efficiency and associated change in generator temperature, system performance differs widely. In order to evaluate the real-life feasibility of the proposed system, the whole system is simulated using the weather data of Kolkata (82º 22' E & 22º 34' N) during the summer months of March to October. It is assumed that water flow through the solar collector and storage tank assembly would be activated or deactivated depending on the availability of sunlight by photosensitive sensor and valve assembly. The variations in collector efficiency for the summer months (March to June) and monsoon and autumn months (July to October) are presented in Fig. 8 and 9 respectively. They exhibit identical trends, with October showing highest efficiency during forenoon, owing to lower ambient temperature. Efficiency curves are very flat for the months of May to July due to consistently high temperature. It is very encouraging to observe that more than 40% efficiency can be achieved for majority of the working hours for almost all the months.

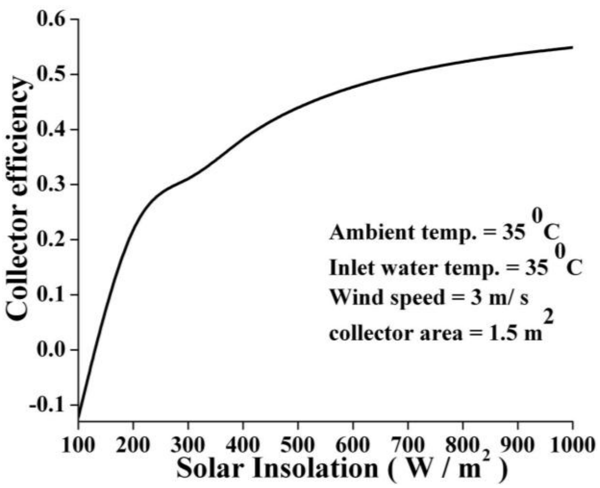


Fig.7. Variation of collector efficiency with solar radiation on fixed date and time

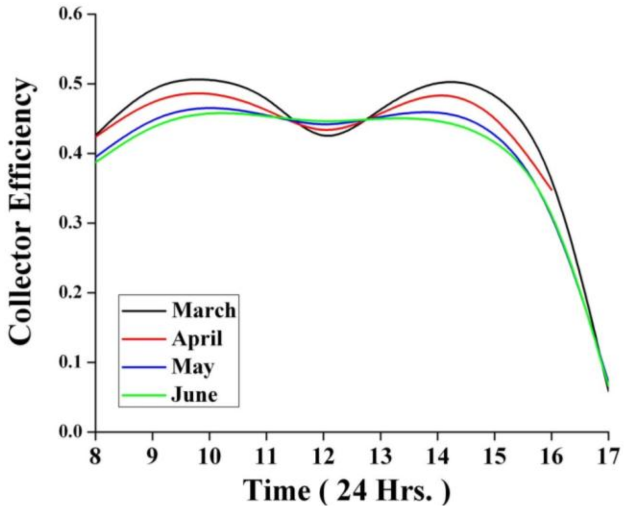


Fig.8. Variation of collector efficiency with respect to time for March to June

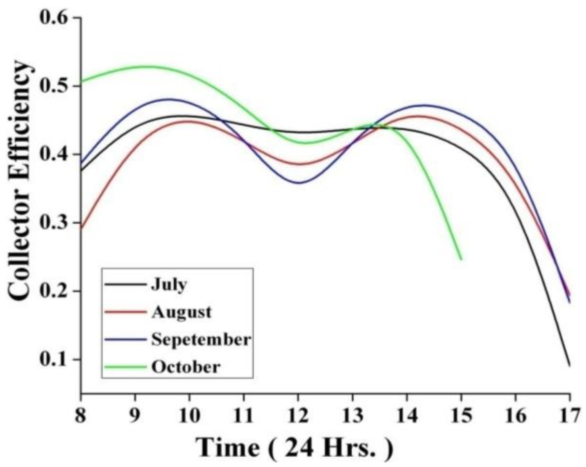


Fig.9. Variation of collector efficiency with respect to time for July to October

The variation of system COP, required evaporator and tank water temperature and collector efficiency throughout the day with prevailing real weather condition is shown for the month of March and September in table 1 and table 2 respectively. The trend of variation of different system properties is quite similar for both representative months. It has been observed that daily total solar radiation is slightly more than overall demand in March, resulting in small increase in tank water temperature at the end of the day compared to the early morning. On the other hand, solar radiation is slightly less than demand in September, causing small decrease in tank water temperature at the evening. So an auxiliary heating source is applied into the tank to increase tank water temperature to its initial value, if required, and it would also help during the cloudy or low solar radiation condition during the working hours. Increasing the number of collectors can result in sufficient energy collection during every month. However, that may unnecessary increase the water temperature during majority of summer months.

Here the tank is maintained 6 to 7 °C higher than generator temperature, in order to achieve the highest possible COP of the system. The result is very evident in as the COP values remain almost unaltered throughout the day and over two different seasons.

## 5. CONCLUSIONS

From the above analysis the following conclusions can be drawn:

i) It is possible to identify an optimum generator temperature for each combination of condenser and evaporator temperatures, for which the COP of the systems is maximum.

ii) Solar FPC-assisted absorption air conditioning systems is suitable for comfort conditioning of an office room for densely-populated sub-tropical cities such as Kolkata (82º 22' E & 22º 34' N).

iii) The system shows the highest values of COP at temperatures around 75 °C which makes it possible to work this system with solar flat plate collectors.

Table 1: Variation of different property of the system for the month of March

| Time (24hrs) | COP | $T_E$ (°C) | $T_{tank}$ (°C) | Collector efficiency |
|---|---|---|---|---|
| 7 | – | – | 80.14 | 0.12514 |
| 8 | 0.894 | 15.83 | 80.16 | 0.42628 |
| 9 | 0.895 | 16.58 | 80.73 | 0.50328 |
| 10 | 0.896 | 16.85 | 81.63 | 0.51006 |
| 11 | 0.896 | 17.05 | 82.99 | 0.48973 |
| 12 | 0.896 | 17.03 | 83.43 | 0.39968 |
| 13 | 0.896 | 17.04 | 84.13 | 0.46673 |
| 14 | 0.896 | 17.01 | 84.87 | 0.51073 |
| 15 | 0.896 | 16.93 | 85.09 | 0.49540 |
| 16 | 0.895 | 16.75 | 84.63 | 0.40407 |
| 17 | 0.895 | 16.43 | 83.66 | 0.05898 |

iv) The variation in collector efficiency of the solar flat plate collector with the day time for different months is quite similar for all the months and it is very encouraging to note that more than 40% efficiency can be achieved almost throughout the day, apart from late afternoon which justifies the potential of such a system in places like Kolkata.

v) In some months slight decrease in initial tank water temperature can be overcome by using some more flat plate collector. But that will be redundant during the months of March and April.

On the other hand when the generator temperature will increase beyond the maximum COP point there is an adverse effect of generator temperature on the COP. So it is recommended to go for auxiliary heating when it is needed.

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Table 2: Variation of different property of the system for the month of September

| Time (24hrs) | COP | $T_E$ (°C) | $T_{tank}$ (°C) | Collector efficiency |
|---|---|---|---|---|
| 7 | – | – | 79.94 | -0.09662 |
| 8 | 0.889 | 14.27 | 79.04 | 0.330633 |
| 9 | 0.894 | 15.61 | 78.62 | 0.45622 |
| 10 | 0.895 | 16.17 | 78.67 | 0.476021 |
| 11 | 0.895 | 16.45 | 78.64 | 0.398606 |
| 12 | 0.896 | 16.5 | 77.93 | 0.270978 |
| 13 | 0.895 | 16.42 | 77.77 | 0.389785 |
| 14 | 0.895 | 16.29 | 77.93 | 0.484999 |
| 15 | 0.894 | 15.87 | 77.61 | 0.473634 |
| 16 | 0.893 | 15.44 | 76.92 | 0.41317 |
| 17 | 0.891 | 14.65 | 75.8 | 0.232949 |
| 18 | – | – | 75.65 | -0.25663 |

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## NOMENCLATURE

| | | |
|---|---|---|
| $m$ | Mass flow rate of water | (kg/s) |
| $P$ | Evaporator pressure | (Pa) |
| $Q$ | Rate of heat flow | (W) |
| $W$ | Work done | (kW) |
| $h$ | Specific enthalpy | (kJ/kg) |
| $I$ | Radiation intensity | (W/m$^2$) |
| $r$ | Tilt factor | |
| $S$ | Absorbed solar radiation | W |
| $\tau$ | Transmissivity | |
| $\alpha$ | Absorptivity | |
| $U$ | Heat loss coefficient | (W/m$^2$ K) |
| $M$ | Number of glass plate | |
| $T$ | Temperature | (°C, K) |
| $\sigma$ | Stephan Boltzmann constant | (W/m$^2$K$^4$) |
| $\varepsilon$ | Emissivity | |
| $\delta$ | Thickness of insulation | (m) |
| $K$ | Thermal conductivity | (W/m K) |
| $l$ | Length of collector box | (m) |
| $w$ | Width of collector box | (m) |
| $H$ | Height of collector box | (m) |
| $A$ | Area of collector plate | (m$^2$) |
| $c$ | Specific heat | (J/kg K) |
| $F$ | Heat removal factor | |
| $RSH$ | Room sensible heat | (W/person) |
| $LHL$ | Latent heat load | (W/person) |
| $INF$ | Amount of infiltration | (m$^3$) |
| $\rho$ | density | (kg/m$^3$) |
| $\omega$ | Relative humidity | (%) |

**Subscripts**

| | |
|---|---|
| $f$ | Flowing fluid(water) |
| $R$ | Refrigerant |
| $WS$ | Weak solution |
| $SS$ | Strong solution |
| $C$ | Condenser |
| $E$ | Evaporator |
| $A$ | Absorber |
| $G$ | Generator |
| $P$ | Pump |
| $T$ | Total radiation |
| $b$ | Beam radiation |
| $d$ | Diffuse radiation |
| $r$ | Reflected radiation |
| $p$ | Plate |
| $a$ | Ambient(air) |
| $i$ | Insulation |
| $l$ | Overall |
| $pm$ | Plate mean |
| $pf$ | Fluid at constant pressure |
| $fo$ | Fluid at inlet of flowing channel |
| $fi$ | Fluid at outlet of flowing channel |
| $u$ | Useful |
| $c$ | Collector |
| $t$ | Tank |
| $pt$ | Tank water at constant pressure |
| $ga$ | Absorption by glass |
| $gc$ | Conduction by glass |
| $w$ | wall |
| $inf$ | infiltration |
| $s$ | sensible |
| $fg$ | Difference between gaseous and liquid state |
| $L$ | latent |

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## APPENDIX

All relevant details about the construction and location of the room are given below.

- Size of the room: 8 m × 8 m × 5 m
- Outside wall : 20 cm brick veneer
- Partition wall : 10 cm brick
- Floor : 20 cm RCC slab
- Floor : 20 cm concrete
- Plaster on the all walls, roof and floors : 1.25 cm
- 3 windows of 2 m × 1 m area in the west wall
- 2 swinging revolving doors made of wood panel of 1 m × 3 m area in the east wall
- 4 mm normal glass used in the window
- $U_{glass}$ = 5.9 W/ m$^2$ K
- $U_{wood}$ = 0.63 W/ m$^2$ K
- Fresh air requirement per person for normal office work = 0.28 cmm / person
- $K_{glass}$ = 0.78 W/ m K
- $K_{concrete}$ = 9.0 W/ m K
- $K_{brick}$ = 1.32 W/ m K
- $K_{plaster}$ = 8.65 W/ m K
- Outside wall film coefficient ($fw_o$) =23 W/ m$^2$ K
- Inside wall film coefficient ($fw_i$) = 7 W/ m$^2$ K
- Outside glass film coefficient ($fg_o$) = 17.5 W/ m$^2$ K
- Inside glass film coefficient ($fg_i$) = 11.5 W/ m$^2$ K
- Infiltration for door opening = 1.9813 cmm / opening

- Infiltration through the crack = 2.5 $m^3$ / hour / m of crack
- Bypass factor = 0.15
- Occupancy load for normal office work:
  SHL = 75 W/ person and LHL = 55 W/ person

## AUTHOR BIOGRAPHY

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