

# Design and Sizing of Solar Collector for Air Conditioned Space, Using Vapor Absorption System

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## Abstract

Most advanced research has been carried out to increase the co-efficient of performance of a Li Br/H<sub>2</sub>O Vapor Absorption System (VAS). VAS uses thermal energy to generate heat which is used in a generator to evaporate refrigerant out of the solution. The thermal energy produced by burning the fossil fuel causes a lot of CO<sub>2</sub> emission to the environment. The OPEX to generate heat for VAS from fossil fuel is always high. The aim of this paper is to use the thermal energy from renewable solar energy, which has zero ozone layer depletion potential and zero global warming potential. This paper will give a fair idea about the economic constraints and the payback calculation of a solar based VAS over a conventional fossil fuel based VAS. The present research work mainly consists of three parts. In the first part, calculation of heat load is carried out for a single-storey conference hall with 100 occupants. In the second part of the study, calculation of heat required at the generator in a VAS to maintain the desired temperature in the conference hall is calculated. In the third part of the study, selection of a solar collector to generate the thermal energy required at the generator is done and the solar collector area is calculated. The results and calculation derived from the study gives a fair idea about the viability of VAS system with solar energy over conventional system.

## 1. Introduction

### 1.1 Background and Need

The need of alternative energy source over the conventional/ current source for generation of electricity through fossil fuel like coal, gas or its byproducts is essential to reduce various pollution level and global warming. We should move towards renewable source of energy like solar, tidal, biomass and hydropower. But solar energy is a consistent and steady source of energy throughout the year. Out of all renewable energy solar energy can be most widely used renewable energy because of its free availability, that's why solar energy can be used as an alternative source of energy and can be a replacement of fossil fuel.

Solar energy being a best alternate source of energy can be used to produce thermal energy because of its ultra clean and safe, zero CO<sub>2</sub> emission, zero ozone layer depletion potential. India's major electricity production facility depends on fossil fuel. India being a tropical country the demand of HVAC is very high and most of the HVAC systems either use electrical chillers or Vapor Absorption system. But the normal vapor absorption system requires high grade of heat and it is generally expensive to produce these high grades of heat through fossil fuel which prohibits using Vapor Absorption system over vapor compression system which is very popular and cost effective. However, the study on vapor

absorption system by using solar energy effectively to get the high-grade energy continuously to make the vapor absorption system viable which will also help to reduce the CO<sub>2</sub> foot print on earth.

### **Need and Importance of Solar Energy**

Solar energy being an alternate source of energy gains popularity because of its versatile advantages to people, society and environment.

*Solar energy is clean and safe:* Solar energy reduces the unclean, hazardous consequences of fossil fuel. Thermal energy or electricity production from fossil fuel causes air pollution and water pollution whereas solar energy eliminates these pollution.

*Increase of CO<sub>2</sub> level:* When forests are cut to get the raw material like fossil fuel (coal and gas) the CO<sub>2</sub> level increases. The trees generally consume CO<sub>2</sub> and reduce CO<sub>2</sub> level. Deforestation for fossil fuel results to increase of CO<sub>2</sub> foot print. Switching to solar energy reduces CO<sub>2</sub> foot print.

*Climate Change:* Increase in CO<sub>2</sub> causes global warming too. The hazardous emission from fossil fuel leads to global warming which cause sudden climate change like less rain fall, draught, melting of ice in polar region which effects the livelihood of our society. Solar energy can play a vital role to cut down the emission level.

*Socio economic benefit to society:* Except the high capital investment the input for solar energy is available free of cost to

everybody in the society. Switching to solar energy will save the utility bills.

## 1.2 Air Conditioning System

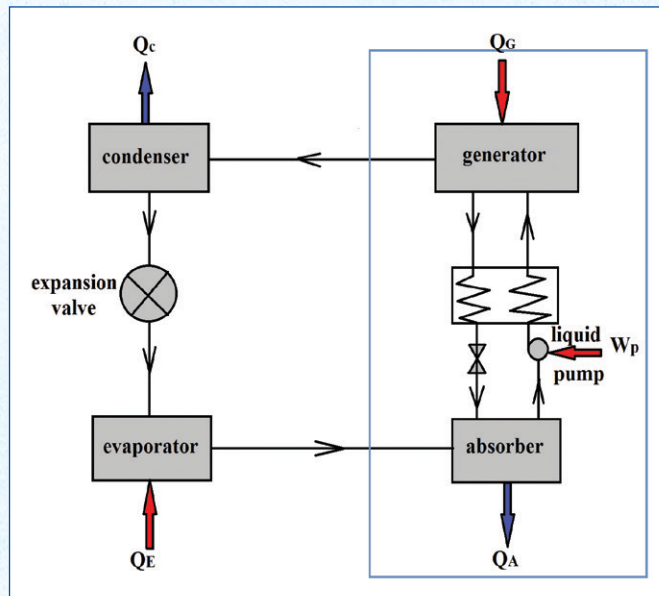
The development of air conditioning system has passed through several research over the centuries. The concept of air conditioning is used since 18<sup>th</sup> centuries when ice or snow was used for cooling by evaporative cooling method. During 18<sup>th</sup> centuries ice was harvested in winter and stored it till summer to get the comfort indoor condition. During Mid 18<sup>th</sup> centuries when ice making machine was invented the practice of harvesting of ice was replaced by machine made artificial ice. In the year 1902 Willis Carrier invented electromechanical air conditioning for printing plant for improving the manufacturing process control. Later on it became commercialized for residential and automobile application.

1.2.1. Non-Cyclic: The methods of getting cooling from melting of ice are called non-cyclic refrigeration.

1.2.2. Cyclic: Cyclic method can be classified as Vapor Cycle and Gas Cycle. Vapor cycle can further have classified as Vapor Compression System (VCS) and Vapor Absorption System (VAS). VAS was invented in early 20<sup>th</sup> century but could not become popular over VCS because of its low COP. VAS system requires lot of thermal energy. Getting heat energy from fossil fuel was always expensive over electric operated compressor in VCS.

## 1.3 Vapor Absorption System

Vapor absorption system is quite similar to vapor compression system. In VAS system, Compressor of a VCS system is replaced by absorber, a pump and a generator. VAS is always viable where ever thermal energy is readily available. VAS is very compact and less bulky than VCS for large capacities. In VAS system, there is a precise control on capacity based on the load condition by controlling the generator designated temp whereas in VCS the capacity of evaporator decreases by reduced evaporator pressure. The performance of VAS is very good in part load condition



over VCS. As there is no compressor involved in VAS system the chances of leakage of refrigerant is very less and due to absence of compressor VAS system is having less maintenance comparatively. There is no moving part except the pump which is quite small compared to the compressor of VCS, it quite in operation and very little wear and tear.

## 1.4 Solar Collector

Solar collector captures the heat from sun rays and transfers the heat to a medium (water, fluid, air). There are four type of solar collectors:

- Low temperature unglazed
- Concentrating
- Flat plate
- Evacuated type

The heart of solar collector is absorber and it is black in color as black is having high degree of absorption and is made up of several narrow metal pipes through which the working fluid is passed. To reduce the loss through emission a special surface coat is applied on absorber. This coating acts in both ways; it reduces emission and converts high amount of solar radiation into heat.

## 1.5 Need for the Present Study

Global warming is seriously affecting human, animal, plant and very serious impact on earth. Increase in sea level, drastic change in weather, draught, hurricanes, diseases, are the resultant of global warming and a threat to human as well as to another habitat. Still most of the developed countries of the world use fossil fuel to produce electricity. However fossil fuel produces several hazardous gases like carbon dioxide and sulfur dioxide to environment which causes global warming.

Thus, it is important to investigate how to reduce the greenhouse effect and solar energy is being a clean and sustainable natural source of energy which can fulfill all future need of energy on earth. Solar being a feasible and huge source of energy proved to be efficient as compared to electrical energy. Moreover, it is necessary to study the comparative performance of solar assisted vapor absorption system and conventional vapor absorption system.

## 1.6 Objectives of the Present Study

The objective of this thesis is to design an economic solar assisted environment friendly vapor absorption system.

1. To calculate the heat load of a conference hall.
2. To calculate the heat required at generator to remove the heat load from the conference hall by using vapor absorption system
3. To study the vapor absorption systems to get optimum COP.
4. To study the minimum area required by solar flat plate collector to produce the amount of heat required by the generator.
5. To make an economic analysis of solar assisted VAS vs VAS system runs on fossil fuel.
6. To make an economic analysis of solar assisted VAS vs VCS system
7. ROI calculation of Solar assisted VAS

## 2. Design of an Air Conditioning System based on Solar Powered Vapor Absorption System

### 2.1 Description of Room Set Up

A water- Lithium Bromide vapor absorption system has been used for air conditioning for 100 persons in a single story 4.5 m high building is having 35 m wide and 30 m long wall. The north and south wall have two doors on each wall respectively each of which 2 m wide and 1.5 m high. Similarly, north, south and east wall have four, three, two windows on each wall having 1.5 m wide and 2 m high. The place is Bhubaneswar (Latitude 21.12° N and longitude 85.65° E) located in south east India. The inside design conditions are 24°Cdbt and 50% RH and the outdoor design conditions are 40°C dbt, 27°C wbt, 12°C daily range and 25 km/h wind velocity. The walls are made up of 228.6mm common brick with 12.7mm plaster on both sides. The roof is made of 127mm RCC and 12.7mm plaster inside. The floor is having 150 mm concrete. The door and window is of 40 mm hard wood and 6.35mm plane glass respectively. In this system, water acts as refrigerant while a solution of lithium bromide in water is used as an absorbent. Both absorber and condenser temperature are 40°C. The generator and evaporator temperature are 80°C and 12°C respectively.

### 2.2 Load Calculation

Most of building are having non-homogeneous wall and materials are composite in nature consisting of bricks, cement plaster, RCC and both forced convection heat transfer and natural convection heat transfer exists along outer and inner wall respectively. And the heat transfer coefficient depends on surface characteristics of wall and wind velocity.

Surface heat transfer for various characteristics of wall surface

$$h = 7.9 + 0.9V \quad \text{for smooth surface [W/m}^2\text{-K]}$$

$$h = 9.1 + 1.06V \quad \text{for smooth wood and plaster [W/m}^2\text{-K]}$$

$$h = 11.4 + 1.41V \quad \text{for cast concrete and smooth brick [W/m}^2\text{-K]}$$

$$h = 11.9 + 1.76V \quad \text{for rough surface}$$

Overall heat transfer coefficient for wall, roof, floor, door, window

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{X_p}{K_p} + \frac{X_b}{K_b} + \frac{X_p}{K_p} + \frac{1}{h_i}}$$

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{X_p}{K_p} + \frac{X_b}{K_b} + \frac{1}{h_i}}$$

$$U_o = \frac{1}{\frac{X_{rcc}}{K_{rcc}} + \frac{1}{h_i}}$$

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{X_{door}}{K_{door}} + \frac{1}{h_i}}$$

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{X_{window}}{K_{window/glass}} + \frac{1}{h_i}}$$

Wall mass, roof mass

$$m = X_b(\rho_{brick}) + 2(X_p)(\rho_{plaster})$$

$$m = X_{rcc}(\rho_{rcc}) + (X_p)(\rho_{plaster})$$

Areas of windows and doors

	Window	Door
North	No of windows (wide*height)	No of doors (wide*height)
South	No of windows (wide*height)	No of doors (wide*height)
East	No of windows (wide*height)	No of doors (wide*height)
West	No of windows (wide*height)	No of doors (wide*height)

Area of walls, floors, and ceilings excluding door and windows:

North: (Length \* Height) – (area of windows on north wall) – (area of doors on north wall)

South: (Length \* Height) – (area of windows on south wall) – (area of doors on south wall)

East: (Width \* Height) – (area of windows on east wall) – (area of doors on east wall)

West: (Width \* Height) – (area of windows on west wall) – (area of doors on west wall)

$$\text{Roof} = \text{Floor} = \text{Length} * \text{Width}$$

Correction to  $\Delta t_e$

The hourly values at various solar times are given for wall mass, m, of 319 and 532 Kg/m<sup>2</sup> and roof mass, m, 212 and 318 Kg/m<sup>2</sup> are given in table.

The limitation and corrections applied while using table are given below.

a) The temp difference between indoor and outdoor ( $t_o - t_i$ ) = 8.33 has been used. To correct for the actual temp difference, add or subtract [( $t_o - t_i$ ) = 8.33] from the tabulated value. This is referred to as  $\Delta t$  correction.

b) For dark color wall absorptivity 0.9 has been used.

c) Specific heat of 0.84KJ/KgK has been used.

d) An outdoor daily range in dry bulb temperature of 11.1 °C has been used for evaluation of sol-air temperature. This may be corrected as follows

i) For each °C DR : add 0.25°C to  $\Delta t_e$  for medium construction  
Less than 11.1°C : add 0.5°C to  $\Delta t_e$  for heavy construction  
: no correction for light construction

ii) For each °C DR : subtract 0.25°C to  $\Delta t_e$  for medium construction  
more than 11.1°C : subtract 0.5°C to  $\Delta t_e$  for heavy construction  
: no correction for light construction

iii) Maximum correction : 2°C for medium and 3°C for heavy construction

Interpolation for wall and roof:

The interpolated value for m, 458.24 Kg/m<sup>2</sup> from the tabulated value of wall mass of 319 and 532 Kg/m<sup>2</sup> is

$$\Delta t_E = \Delta t_{E1} + \frac{458 - 319}{532 - 319} (\Delta t_{E2} - \Delta t_{E1})$$

The interpolated value for m, 303.022 Kg/m<sup>2</sup> from the tabulated value of wall mass of 212 and 318 Kg/m<sup>2</sup> is

$$\Delta t_E = \Delta t_{E1} + \frac{303 - 212}{318 - 212} (\Delta t_{E2} - \Delta t_{E1})$$

Sensible Heat Gain: Heat gain from walls, roof, floor, doors and windows:

For walls and roof heat gain is calculated by  $U_o A \Delta t_e$

For floor the temperature difference is assumed to be 2.5°C.

For door, it is calculated by  $U_o A \Delta t$

For windows the solar heat gain is calculated by multiplying SHGF to the area of the glass.

The heat gain from wall, floor, roof, door and windows can be calculated at various solar time and the maximum value can be taken for further calculation.

Infiltration: Windows:

The windows are double hung, non-weather stripped and poor fit.

Interpolating data from table for wind velocity 25Kmph

$$\text{Infiltration Rate} = 10.3 + \frac{14.3 - 10.3}{32 - 24}$$

$$\text{Total crack length} = [3 * \text{height} + 2 * \text{width}] * \text{No. of windows}$$

Infiltration through all the windows: Infiltration rate \* Total crack length

Infiltration: Doors:

The doors are non-weather stripped and poor fit.

Interpolating data from table for wind velocity 25Kmph

$$\text{Infiltration Rate} = 0.17 + \frac{0.24 - 0.17}{32 - 24}$$

$$\text{Total crack length} = [3 * \text{height} + 2 * \text{width}] * \text{No. of doors}$$

Infiltration through all the windows: Infiltration rate \* Total crack length

Considering the application as bank/office number of opening of the door will be same as no of persons and the infiltration rate for bank/ office application can be taken from table as 11 m<sup>3</sup>/h.

*Infiltration due to door opening = infiltration rate for application \* number of occupants*

*Total Infiltration = infiltration through all windows + infiltration through all the doors + infiltration due to door openings*

Cooling Load due to Infiltration:

For outdoor temperature and humidity

$$h_o(t) = 2500 + 1.88 * t$$

$$W_o = 0.622 + \frac{RH * P_{ws}}{P - RH * P_{ws}}$$

For indoor temperature and humidity

$$h_i(t) = 2500 + 1.88 * t$$

$$W_i = 0.622 + \frac{RH * P_{ws}}{P - RH * P_{ws}}$$

Sensible Heat Gain due to Infiltration:

$$Q_s = OASH = 20.4 * Q_v * \Delta t \text{ Watt}$$

Latent Heat Gain due to Infiltration:

$$Q_l = OALH = 50000 * Q_v * \Delta W \text{ Watt}$$

Occupancy Load:

Assuming the occupants are standing and walking on the floor and the sensible and latent heat can be taken from the table for the application per person. Therefore the occupancy load for total number occupants can be calculated.

*Occupancy load, Sensible load = sensible heat load per person \* number of occupants*

*Occupancy load, Latent load = latent heat load per person \* number of occupants*

Lighting Load:

The luminous energy produced by electrical lighting is dissipating heat energy. The illumination of a room depends on the area of surface in the room. The fluorescent fitting requires 20% extra power than the rated capacity.

The gain from lighting load can be calculated as

$$Q_s = W * F_u * F_s * (CFL)$$

W = Lamp rating

F<sub>u</sub> = The ratio of wattage in use to total installed wattage

F<sub>s</sub> = Special allowance factor

CFL = Cooling load factor from table

Room Sensible Heat Gain (RSH):

Room sensible heat gain is the summation of total heat gain from wall, floor, windows, doors, outside air sensible heat gain (OASH), sensible occupancy load, lighting sensible load. Safety factor of 5% is also added to the RSH. In system heat gain 6.5% of RSH value is also added to total RSH. 6.5% of RSH is the summation from 0.5% of RSH as leakage heat transfer, 1% of RSH as leakage of air supply, 5% RSH as fan motor.

Room Latent Heat Gain (RLH):

Room latent heat is the summation of outside air latent heat load and latent occupancy load. A safety factor of 5% is also considered. In system heat gain leakage in 10% of the supply air duct outside the non-conditioned space is 1% of RLH is added to total RLH.

Ventilation:

Ventilation is the fresh air requirement for persons. The minimum value from the table in Carrier Handbook is 0.33 cmm per person

*Fresh outdoor air requirement for total occupants, Q<sub>v</sub> = 0.33 \* No. of occupants*

Outside air sensible heat load (OASH)

$$OASH = 20.4 * Q_v * \Delta t \text{ Watt}$$

Outside air latent heat load (OALH)

$$OALH = 50000 * Q_v * \Delta W \text{ Watt}$$

Total Sensible Heat Load (TSH)

$$TSH = RSH + OASH$$

Total Latent Heat Load (TLH)

$$TLH = RLH + OALH$$

Grand Total Heat Load (GTH)

$$GTH = TSH + TLH$$

Supply Air Temperature:

Supply air temperature can be determined by effective sensible heat factor method (ESHF).

$$\text{Effective Room Sensible Heat (ERSH)} = RSH + X(OASH)$$

$$\text{Effective Room Latent Heat (ERLH)} = RLH + X(OALH)$$

For a typical air conditioning system By pass factor(X) is assumed as 0.15.

$$\text{Effective Room Total Heat (ERTH)} = ERSH + ERLH$$

$$ESHF = \frac{ERSH}{ERTH}$$

Apparatus dew point table for 25°C and 50% RH has taken from Carrier Handbook and interpolating in this table for ESHF we will get,

$$ADP = ADP1 + \frac{ESHF - 0.84}{0.94 - 0.84}(ADP2 - ADP1)$$

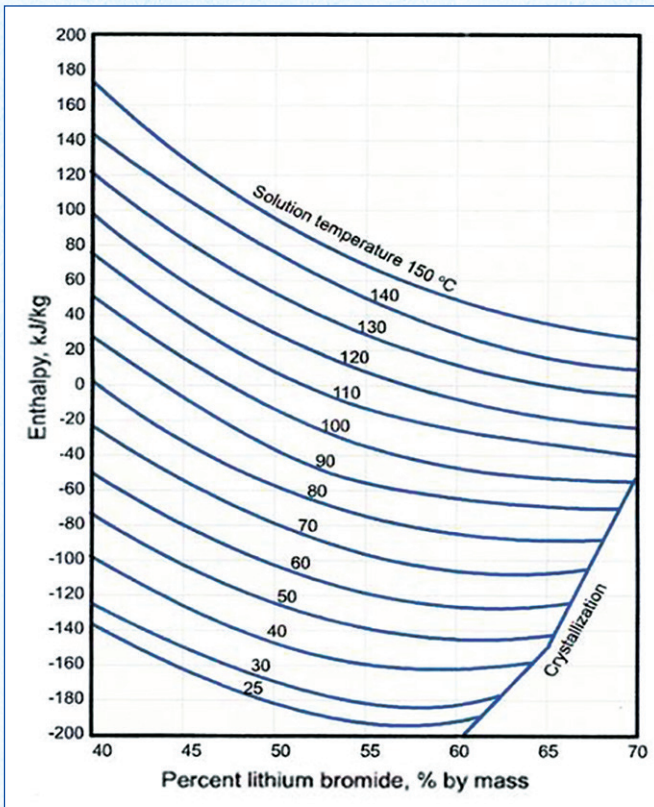
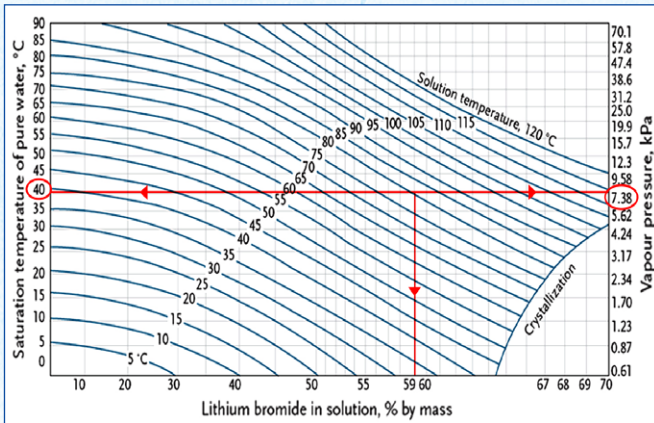
Dehumidified temperature rise under this assumption

$$\Delta t = (1-X)(t_i - t_{ADP})$$

Air entering temp to room,  $t = \text{Room Temperature} - \Delta t$

### 2.3 COP Calculation for Vapor Absorption System

If the evaporator and condenser temperature are known the concentration ( $X_1$  &  $X_2$ ) of a saturated liquid leaving the generator and saturated liquid leaving the absorber can be obtained from Duhring Plot. Once the concentration is known,



the corresponding enthalpy value can be taken from Enthalpy-Temperature-Concentration diagram for water-lithium bromide solution graph.

At generator exit, enthalpy of water vapor,  $h_3 = 2501 + 1.88(\text{generator temperature})$

At evaporator exit, enthalpy of water vapor,  $h_6 = 2501 + 1.88(\text{evaporator temperature})$

At condenser exit, enthalpy of saturated liquid,  $h_4 = 2501 + 1.88(\text{condenser temperature})$

For a specific cooling load in TR,

$$\dot{m}_6 = \frac{TR}{(h_6 - h_4)} \text{ Kg/s}$$

Overall mass conservation for the absorber,

$$\dot{m}_2 + \dot{m}_6 = \dot{m}_1 \dots \dots \dots (1)$$

At evaporator exit we have pure water vapor, hence concentration of LiBr = 0,  $X_6 = 0$

$$\dot{m}_2 * X_2 + \dot{m}_6 * (0) = \dot{m}_1 * X_1 \dots \dots \dots (2)$$

Solving equation 1 and 2 we will get  $\dot{m}_2$  and  $\dot{m}_1$

Energy balance for the absorber,

$$Q_a = \dot{m}_2 * h_2 + \dot{m}_6 * h_6 - \dot{m}_1 * h_1$$

Similarly, for generator overall mass conservation,

$$\dot{m}_2 + \dot{m}_3 = \dot{m}_1$$

$$\text{and } \dot{m}_3 = \dot{m}_6$$

$$Q_g = \dot{m}_2 * h_2 + \dot{m}_3 * h_3 - \dot{m}_1 * h_1$$

$$COP = \frac{Q_e}{Q_g}$$

### 2.4 Calculation of Area Required for Solar Panel

The total thermal heat energy ( $Q_g$ ) required by generator must be generated by the solar panel ( $Q_s$ ).

$$Q_s = (m C_p) \Delta t, \text{ and } Q_s = Q_g$$

$Q_s = \text{Total heat generated by solar panel which is required by generator}$

$Q_g = \text{Heat required by generator}$

$m = \text{Volume of water required}$

$C_p = \text{Heat capacity of water (1.16 kWh/m}^3 \text{ K)}$

$\Delta t = \text{Temperature difference of hot water and cold water.}$

Collector yield can be calculated;  $C_Y = S_R * \tilde{\eta}_K * \tilde{\eta}_{\text{System}}$

$\tilde{\eta}_K = \text{Efficiency of collector}$

$\tilde{\eta}_{\text{System}} = \text{Efficiency of System}$

Solar Radiation  $S_R$

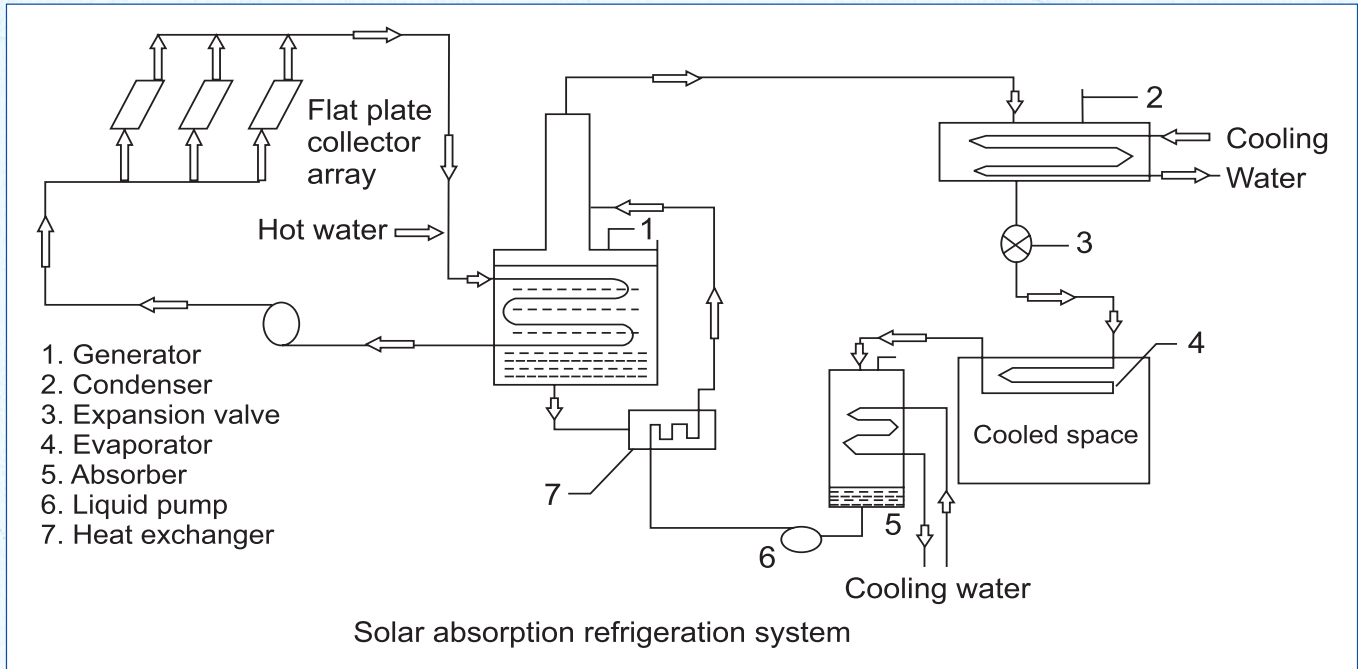
$$= \frac{\text{jan} + \text{feb} + \text{mar} + \text{apr} + \text{may} + \text{jun} + \text{jul} + \text{aug} + \text{sep} + \text{oct} + \text{nov} + \text{dec}}{12}$$

Theoretical calculation of efficiency of a flat plate collector:

$$\tilde{\eta} = 0.848 - 3.80 * \frac{(T_m - T_a)}{G} - 0.0012 \frac{(T_m - T_a)^2}{G}$$

Collector area required:

$$C_A = \frac{Q}{C_Y}$$



**2.5 Results and Discussion**

Convective heat transfer for cast concrete and smooth brick

$$h = 11.4 + 1.41V$$

$$h_o = 11.4 + 1.41 * 25 \text{ (Km/h)} = 46.65 \text{ W/m}^2 \text{ }^\circ\text{K}$$

Convective heat transfer for cast concrete and smooth brick

$$h = 7.9 + 0.9V$$

$$h_i = 7.9 + 0.9 * 25 \text{ (Km/h)} = 8.8 \text{ W/m}^2 \text{ }^\circ\text{K}$$

Overall heat transfer coefficient for wall, roof, floor, door,

window

$$U_o = \frac{1}{\frac{1}{8.8} + \frac{0.0127}{1.15} + \frac{0.2286}{0.77} + \frac{0.0127}{1.15} + \frac{1}{46.65}}$$

$$U_o = 2.20244 \text{ W/m}^2 \text{ }^\circ\text{K}$$

$$U_o = \frac{1}{\frac{1}{8.8} + \frac{0.0127}{1.15} + \frac{0.127}{1.73} + \frac{1}{46.65}}$$

$$U_o = 4.55526 \text{ W/m}^2 \text{ }^\circ\text{K}$$

$$U_o = \frac{1}{\frac{1}{8.8} + \frac{0.15}{1.73}}$$

$$U_o = 4.99148 \text{ W/m}^2 \text{ }^\circ\text{K}$$

$$U_o = \frac{1}{\frac{1}{8.8} + \frac{0.04}{0.158} + \frac{1}{46.65}}$$

$$U_o = 2.57575 \text{ W/m}^2 \text{ }^\circ\text{K}$$

$$U_o = \frac{1}{\frac{1}{8.8} + \frac{0.00625}{0.78} + \frac{1}{46.65}}$$

$$U_o = 6.98258 \text{ W/m}^2 \text{ }^\circ\text{K}$$

Wall mass, roof mass

$$m = (0.2286)(1800) + 2(0.0127)(1860)$$

$$m = 458.724 \text{ kg/m}^2$$

$$m = (0.127)(2200) + (0.0127)(1860)$$

$$m = 303.022 \text{ kg/m}^2$$

Areas of windows and doors

	Window	Door
North	4 (2 * 1.5) = 12 m <sup>2</sup>	2 (1.5 * 2) = 6 m <sup>2</sup>
South	3 (2 * 1.5) = 9 m <sup>2</sup>	2 (1.5 * 2) = 6 m <sup>2</sup>
East	2 (2 * 1.5) = 6 m <sup>2</sup>	0 (1.5 * 2) = 0 m <sup>2</sup>
West	0 (2 * 1.5) = 0 m <sup>2</sup>	0 (1.5 * 2) = 0 m <sup>2</sup>

Area of walls, floors, and ceilings excluding door and windows:

$$\text{North: } (35 * 4.5) - (12) - (9) = 139.5 \text{ m}^2$$

$$\text{South: } (35 * 4.5) - (9) - (6) = 142.5 \text{ m}^2$$

$$\text{East: } (30 * 4.5) - (6) - (0) = 129 \text{ m}^2$$

$$\text{West: } (30 * 4.5) - (0) - (0) = 135 \text{ m}^2$$

$$\text{Roof = Floor = } 30 * 35 = 1050 \text{ m}^2$$

Correction to  $\Delta t_E$

$$t_o - t_i = 40 - 24 = 16^\circ\text{C}$$

$$\text{Correction for } (t_o - t_i) = 16 - 8.33 = 7.7^\circ\text{C}$$

$$\text{Correction for daily range, DR} = 0.25 (12 - 11.1) = 0.225$$

$$\text{Total correction to } \Delta t_E = 7.7 - 0.225 = 7.475^\circ\text{C}$$

Interpolation for wall mass:

$$\Delta t_E = \Delta t_{E1} + \frac{458 - 319}{532 - 319} (\Delta t_{E2} - \Delta t_{E1})$$

Solar time	North Wall	$\Delta t_E$
2.00 pm	$\Delta t_{E1}$	3.3
	$\Delta t_{E2}$	1.1
3.00 pm	$\Delta t_{E1}$	4.4
	$\Delta t_{E2}$	1.7
4.00 pm	$\Delta t_{E1}$	5.5
	$\Delta t_{E2}$	2.2
5.00 pm	$\Delta t_{E1}$	6.1
	$\Delta t_{E2}$	2.8
6.00 pm	$\Delta t_{E1}$	6.7
	$\Delta t_{E2}$	2.8
7.00 pm	$\Delta t_{E1}$	6.7
	$\Delta t_{E2}$	2.8

Solar time	South Wall		$\Delta t_E$
2.00 pm	$\Delta t_{E1}$	13.3	8.970524
	$\Delta t_{E2}$	6.7	
3.00 pm	$\Delta t_{E1}$	13.9	10.22651
	$\Delta t_{E2}$	8.3	
4.00 pm	$\Delta t_{E1}$	14.4	10.7921
	$\Delta t_{E2}$	8.9	
5.00 pm	$\Delta t_{E1}$	12.3	10.79124
	$\Delta t_{E2}$	10	
6.00 pm	$\Delta t_{E1}$	11.1	10.37842
	$\Delta t_{E2}$	10	
7.00 pm	$\Delta t_{E1}$	8.3	8.3
	$\Delta t_{E2}$	8.3	

Solar time	East Wall		$\Delta t_E$
2pm	$\Delta t_{E1}$	7.8	11.40789671
	$\Delta t_{E2}$	13.3	
3pm	$\Delta t_{E1}$	7.2	9.758326761
	$\Delta t_{E2}$	11.1	
4pm	$\Delta t_{E1}$	6.7	8.864738028
	$\Delta t_{E2}$	10	
5pm	$\Delta t_{E1}$	7.2	8.315168075
	$\Delta t_{E2}$	8.9	
6pm	$\Delta t_{E1}$	7.8	7.8
	$\Delta t_{E2}$	7.8	
7pm	$\Delta t_{E1}$	7.2	7.593588732
	$\Delta t_{E2}$	7.8	

Solar time	West Wall		$\Delta t_E$
2pm	$\Delta t_{E1}$	5.4	4.744019
	$\Delta t_{E2}$	4.4	
3pm	$\Delta t_{E1}$	10.6	7.254496
	$\Delta t_{E2}$	5.5	
4pm	$\Delta t_{E1}$	14.4	9.348945
	$\Delta t_{E2}$	6.7	
5pm	$\Delta t_{E1}$	18.9	12.66818
	$\Delta t_{E2}$	9.4	
6pm	$\Delta t_{E1}$	22.2	14.91861
	$\Delta t_{E2}$	11.1	
7pm	$\Delta t_{E1}$	22.8	16.96177
	$\Delta t_{E2}$	13.9	

Interpolation for roof mass:

$$\Delta t_E = \Delta t_{E1} + \frac{303 - 212}{318 - 212} (\Delta t_{E2} - \Delta t_{E1})$$

Solar time	Roof		$\Delta t_E$
2pm	$\Delta t_{E1}$	15	14.48478
	$\Delta t_{E2}$	14.4	
3pm	$\Delta t_{E1}$	17.2	16.68478
	$\Delta t_{E2}$	16.6	
4pm	$\Delta t_{E1}$	19.4	18.02608
	$\Delta t_{E2}$	17.8	
5pm	$\Delta t_{E1}$	21.1	19.64021
	$\Delta t_{E2}$	19.4	
6pm	$\Delta t_{E1}$	21.7	20.75543
	$\Delta t_{E2}$	20.6	
7pm	$\Delta t_{E1}$	21.7	20.75543
	$\Delta t_{E2}$	20.6	

Applying correction to equivalent temperature by adding 7.4°C we will get  $\Delta t_E$  corrected.

Solar time	North Wall	South Wall	East Wall	West Wall	Roof	Floor Assumed
2pm	9.26	16.3	18.7	12.1	21.8	2.5
3pm	10.03	17.6	17.1	14.6	24	2.5
4pm	10.7	18.2	16.2	16.7	25.4	2.5
5pm	11.3	18.2	15.7	20.1	27	2.5
6pm	11.5	17.7	15.2	22.3	28.15	2.5
7pm	11.5	15.7	14.9	14.3	28.15	2.5

Solar time	2	3	4	5	6	7
% DR	3%	0%	3%	10%	21%	34%
%DR =	0.36	0	0.36	1.2	2.52	4.08
to °C	39.64	40	39.64	38.8	37.48	35.92
$\Delta t = (t_o - t_i)$	15.64	16	15.64	14.8	13.48	11.92
SHGFNorth	60	79	104	129	88	
SHGFSouth	44	44	38	28	9	
SHGF East	44	44	38	28	9	
SHGF west	303	451	505	467	256	

Sensible Heat Gain from Wall, Roof and Windows						
Solar time	2	3	4	5	6	7
Wall	Q = UoA $\Delta t_e$					
North	2396.945	2595.881	2777.912	2932.523	2985.712	2985.712
South	4338.497	4670.444	4819.927	4819.699	4710.594	4161.283
East	6290.629	5740.218	5442.055	5258.681	5086.785	5017.912
West	4228.172	5099.016	5825.546	6976.934	7757.571	8466.31
Roof	104890.6	115413.3	121828.8	129549.2	134883.3	134883.3
Floor	13102.62	13102.62	13102.62	13102.62	13102.62	13102.62
Window						
North	1310.49	1340.655	1310.49	1240.106	1129.502	998.7877
South	982.8674	1005.491	982.8674	930.0792	847.1262	749.0908
East	655.245	670.3273	655.245	620.0528	564.7508	499.3939
West	0	0	0	0	0	0
Door	483.4159	494.5431	483.4159	457.4524	416.6526	368.4346
Glass						
North	720	948	1248	1548	1056	
South	396	396	342	252	81	
East	264	264	228	168	54	
West	0	0	0	0	0	
<b>Total</b>	<b>140059.5</b>	<b>151740.5</b>	<b>159046.8</b>	<b>167855.3</b>	<b>172675.6</b>	<b>171232.8</b>

The maximum heat gain from walls, roof, floor, window, glasses at 6.00 pm = 172675.6 watt

Infiltration: Windows:

$$\text{Infiltration Rate} = 10.3 + \frac{14.3 - 10.3}{32 - 24}$$

$$= 10.8 \text{ m}^3/\text{hr per crack length}$$

## Design and Sizing of Solar Collector for Air Conditioned Space, Using Vapor Absorption System

Total crack length for 9 windows =  $(3 * 1.5 + 2 * 2) * 9 = 76.5$  m  
 Infiltration through all the window =  $10.8 * 76.5 = 826.2$  m<sup>3</sup>/hr  
 Infiltration: Doors:

$$\text{Infiltration Rate} = 0.17 + \frac{0.24 - 0.17}{32 - 24}$$

$$= 0.17875 \text{ cmm per min} = 10.725 \text{ m}^3/\text{hr}$$

Total crack length for all the doors =  $(3 * 2 + 2 * 1.5) * 4 = 36$  m

Infiltration through all the window =  $10.725 * 36 = 386.1$  m<sup>3</sup>/hr

Infiltration due to door openings due to 100 occupants =  $100 * 11 = 1100$  m<sup>3</sup>/hr

Total Infiltration,  $Q_v = 826.2 + 386.1 + 1100 = 2312.3$  m<sup>3</sup>/hr = 38.5383 cmm

Cooling load due to infiltration

Outdoor temp = 40°C, RH = 50%

At 40°C,  $P_{ws} = 0.07375$

$$h_g(40) = 2500 + 1.88 * 40 = 2575.2$$

$$W_o = 0.622 + \frac{RH * P_{ws}}{P - RH * P_{ws}}$$

$$W_o = 0.02349 \text{ KgW/Kga}$$

Indoor temp = 24°C, RH = 50%

At 24°C,  $P_{ws} = 0.02982$

$$W_i = 0.622 + \frac{RH * P_{ws}}{P - RH * P_{ws}}$$

$$W_i = 0.00929 \text{ KgW/Kga}$$

Sensible heat gain due to infiltration

$$Q_s = \text{OASH} = 20.4 * 38.5 * (40 - 24) = 12578.9 \text{ watt}$$

Latent heat gain due to infiltration

$$Q_L = \text{OALH} = 50000 * 38.5 * (0.02349 - 0.00929) = 27365.1 \text{ Watt}$$

Occupancy Load

Occupancy Load, Sensible load = 7300

Occupancy Load, Latent load = 7300

Lighting Load

$$12000 \text{ w Fluorescent Lamp} = 1.2 * 12000 = 14,400 \text{ watt}$$

$$5000 \text{ w Tungsten Lamp} = 5000 \text{ watt}$$

$$\text{Lighting Sensible load} = 19400 \text{ watt}$$

Room Sensible Heat Gain (RSH)		
A.	Sensible Heat gain from Walls, Roof, Floor and Window	172675.6 Watt
B.	Infiltration	12578.91 Watt
C.	Occupancy	7300 Watt
D.	Lighting	19400 Watt
Total=		211954.5 Watt
Safety Factor 5%=		10597.73 Watt
Total After SF=		222552.2 Watt
System Heat Gains		
a.	Leakage Heat Transfer	0.50% 1112.761 Watt
b.	Leakage of Supply Air	1% 2225.522 Watt
c.	Fan Motor	5% 11127.61 Watt
Room Sensible Heat Gain (RSH)		237018.1 Watt

Room Latent Heat Gain (RLH)			
A.	Infiltration	27365.06 Watt	
B.	Occupancy	7300 Watt	
Total=		34665.06 Watt	
Safety Factor 5%=		1733.253 Watt	
Total After SF=		36398.31 Watt	
System Heat gains			
a.	Leakage in 10% of the supply air duct outside the non-conditioned space = 1% of RLH	1%	363.9831 Watt
Room Latent Heat Gain (RLH)		36762.29 Watt	

Ventilation Load:

For 100 persons,  $Q_v = 0.33 * 100 = 33$  cmm

Outdoor air sensible heat load (OASH) =  $20.4 * 33 * (40 - 24) = 10771.2$  watt

Outside air latent heat load (OALH) =  $50000 * 33 * (0.02349 - 0.00929) = 23432$  watt

Total Sensible heat load =  $237018.1 + 10771.2 = 247789.3$  watt

Total Latent heat load =  $36762.29 + 23432 = 60194.72$  watt

Grand Total Heat Load (GTH) =  $307984.06$  watt = 307.984 KW = 87.595 TR

Supply Air Temperature:

$$\text{ERSH} = 237018.1 + (0.15) * 10771.2 = 238634 \text{ watt}$$

$$\text{ERLH} = 36762.29 + (0.15) * 23432 = 40277.2 \text{ watt}$$

$$\text{ERTH} = 238634 + 40277.2 = 278911 \text{ watt}$$

$$\text{ESHF} = 0.85559$$

Interpolating data for ESHF = 0.85559 in ADP table we will get

$$\text{ADP} = 12.393^\circ\text{C}$$

Dehumidified temperature rise under this assumption

$$\Delta t = (1 - 0.15) * (24 - 12.393) = 9.8659^\circ\text{C}$$

$$\text{Air enters the room} = 24 - 9.8659 = 14.1^\circ\text{C}$$

COP analysis of Vapor Absorption System:

Evaporator Temperature: 12°C

Condenser Temperature: 40°C

Absorber Temperature: 40°C

Generator Temperature: 80°C

Refrigeration Load: 307.984 KW

For evaporator temperature 12°C and absorber temperature 45°C and generator temperature 80°C and condenser temp 45°C, from P-T-X diagram we can get

$$X_1 = 0.55 \quad h_1 = -145 \text{ KJ/Kg}$$

$$X_2 = 0.59 \quad h_2 = -84 \text{ KJ/Kg}$$

At generator exit enthalpy of water at 80°C,  $h_3 = 2500 + 1.88 * (80) = 2650.4$  KJ/Kg

At evaporator exit enthalpy of water vapor at 12°C,  $h_6 = 2500 + 1.88 * (12) = 2522.56$  KJ/Kg

At condenser exit enthalpy of saturated liquid at 40°C,  $h_4 = 4.1867 * 40 = 167.468$  KJ/Kg

For 307.984 KW cooling capacity,

$$m_6 = \frac{307.984}{2522.56 - 167.468} = 130.774 \times 10^{-3} \text{ Kg/s}$$

Overall mass conservation for the absorber and at evaporator exit,  $X_6=0$ ,

$$\dot{m}_2 = \frac{m_6 \cdot X_1}{X_2 - X_1} = 1.798138 \text{ Kg/s}$$

$$\dot{m}_1 = \frac{m_6 \cdot X_2}{X_2 - X_1} = 1.928911 \text{ Kg/s}$$

Energy balance for generator

$$Q_g = 1.798138 \cdot (-84) + 0.130774 \cdot 2650.4 - 1.928911 \cdot (-145) = 475.2519 \text{ KW}$$

$$COP = \frac{307.984}{475.251} = 0.648$$

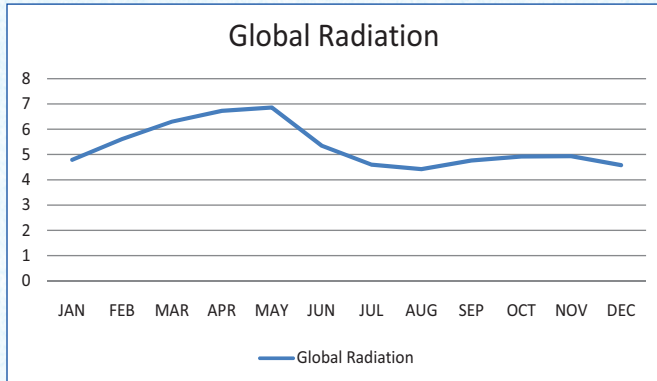
Calculation of Area required for solar panel:

$$\text{Energy demand } Q_s = Q_g = 475.2519 = m \cdot 1.16 \cdot 50$$

$$m = 8.19 \text{ m}^3 = 8190 \text{ liter}$$

Average monthly and yearly values of global solar radiation on a horizontal surface in kWh/m<sup>2</sup>.

Bhubaneswar- Global Radiation in kWh/m <sup>2</sup>											
JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
4.8	5.6	6.3	6.7	6.9	5.4	5	4.4	4.8	4.9	4.9	4.6



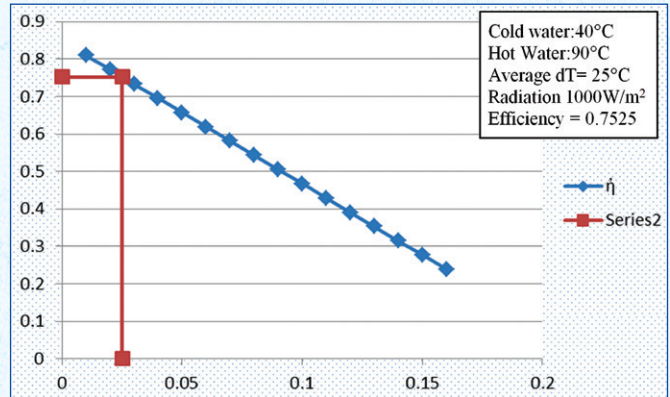
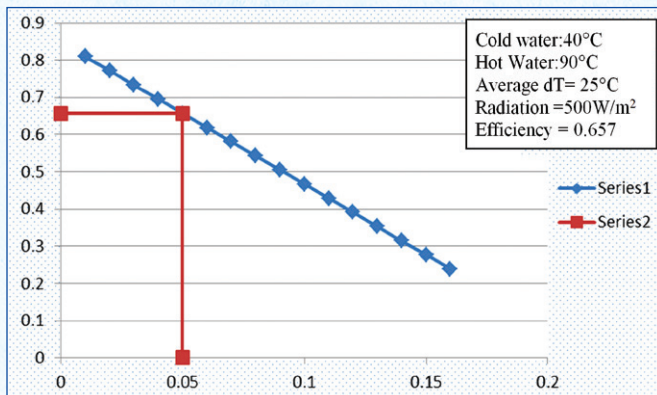
$$S_R = \frac{4.79 + 5.61 + 6.73 + 6.86 + 5.35 + 4.6 + 4.42 + 4.76 + 4.92 + 4.93 + 4.58}{12}$$

$$S_R = 5.32 \frac{\text{kWh}}{\text{m}^2 \cdot \text{day}}$$

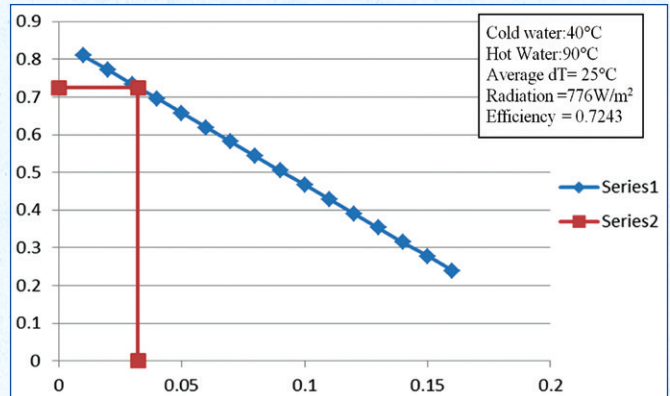
Theoretical calculation of efficiency of a flat plate collector

$$\eta_f = 0.848 - 3.80 \cdot \frac{(T_m - T_a)}{G} - 0.0012 \cdot \frac{(T_m - T_a)^2}{G}$$

Efficiency of a flat plate collector can be observed at various global radiation level



Global radiation of Bhubaneswar city has been taken from International journal of Agricultural Engineering, Vol2, No 2(Oct 2009 to Mar 2010) : 176-181 as 776 W/m<sup>2</sup> and efficiency has been calculated as follows.



By keeping system efficiency 70%, Collector yield at global radiation 776 W/m<sup>2</sup> can be calculated as follows

$$C_Y = 5.32 \cdot 0.7243 \cdot 0.7 = 2.69 \text{ kWh/m}^2$$

Collector area:

$$C_A = \frac{475.2519}{2.69} = 176.67 \text{ m}^2$$

Therefore we can use 89 collector plates having dimension of 1 X 2 sq m

### 3. Economics of Solar Collector for the Present Problem

#### 3.1 Collector Economics

With the increasing cost of electricity with each passing day people are switching towards renewable solar energy from the conventional method of getting electricity from fossil fuel. In India, interrupted power supply and dependency on diesel generated sets for backup power supply force people to think about the alternate uninterrupted source of energy called solar energy. There are several Government incentives and subsidies are provided to people to buy solar PV to promote solar PV system under the Jawaharalal Nehru National Solar Mission Scheme in India by Ministry of New and Renewable Energy (MNRE). As per

research (on e-commerce sites selling solar panel) cost of solar PV lies between Rs 10000 to 15000 per square meter.

There are two types of Solar PV Cells: Monocrystalline and Polycrystalline. The difference between the two is that Monocrystalline is made of single silicon crystal whereas Multi-crystalline PV is made up of multiple crystals. A monocrystalline is more efficient in converting solar energy into electricity per sq meter area than a multi-crystalline PV. Thus, the space required for the same amount of wattage is less in monocrystalline PV panel. Thus, it is costlier than a Multi-crystalline PV. The modules available in India are mostly "Polycrystalline". Typical efficiency of panels available in India is between 14-17%. But modules of smaller sizes have efficiencies as low as 6%. Please note that lower the efficiency, more panels you would need to get the same amount of electricity and it will occupy more rooftop area.

### 3.2 Comparison of Solar Powered VAS with Existing Approach (VCS with Use of Electricity)

In general, a hall having 90 TR air conditioning heat load requires a 132 KW electric motor connected to air cooled chiller to maintain 24°C. We can make a comparative study of operating cost of an air cooled electric chiller with capital investment made to install a solar powered vapor absorption system for a 90 TR air-conditioned space considering 8 operating hours in a day 25 days in a month.

Total operating hours in a year =  $8 * 25 * 12 = 2400$  hrs

Total electricity consumption by the electric chiller in a year =  $132 * 2400 = 316800$  kWh

Assuming Rs. 5 as unit cost of electricity,

The total operating cost of electric chiller in a year =  $316800 * 5 = \text{Rs}15.84$  Lacs/year

The approximate cost of a vapor absorption machine varies from Rs25000 to Rs 35000 per TR including installation and commissioning.

Approximate cost of vapor absorption machine =  $90 * 35000 = \text{Rs} 3150000$

Approximate cost of collector plates =  $176.67 * 15000 = \text{Rs}2650050$

Total cost of solar power vapor absorption system = Rs 5800050

Return on investment (ROI) =  $\text{Rs}5800050 / \text{Rs} 1584000 = 3.66$  year

## 4. Summary and Conclusions

### 4.1 Summary

In the above calculation, the heat load of a 30 X 35 X 4.5 m room is about 307.984 KW and to run this 307.984 kw LiBr/H<sub>2</sub>O vapor absorption system we require 475.2519 KW thermal energy at generator. And this 475.2519 KW heat is generated from solar panel by heating the water and supplied to generator. The efficiency of vapor absorption system has been calculated as 64.8%. The total area of solar PV panel calculated to generate 475.2519 KW thermal energy is 176.67 m<sup>2</sup>. By keeping the heat losses while flowing through the pipe the output temperature of hot water from solar panel is 90°C and hot water supplied to

generator is 80°C. An economic study of a conventional electrical chiller with solar powered vapor absorption system has been done.

### 4.2 Major Conclusion

In the above study, the feasibility of a solar powered LiBr/H<sub>2</sub>O vapor absorption system has been proved over electric chiller operated vapor compression system. Based on results obtained from the above calculation, the return on investment or the pay back of a solar powered vapor absorption system has been calculated as less than four years. As discussed earlier, solar energy can be a best alternate for commercial air conditioning purpose.

### 4.3 Scope of Future Work

More research and investigation need to be carried out to increase the efficiency of a solar powered LiBr/H<sub>2</sub>O vapor absorption system and specifically on the study of solar collector to make the system more cost optimized.

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