Design& Analysis Of Solar Vapour Absorption System Using Water And Lithium Bromide

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Abstract

This paper presents the design and analysis of the solar assisted vapour absorption system of 3.5 ton.The results presented in this paper are based on the analysis which is calculated by hand on the basis of different research paper and data taken by different books.

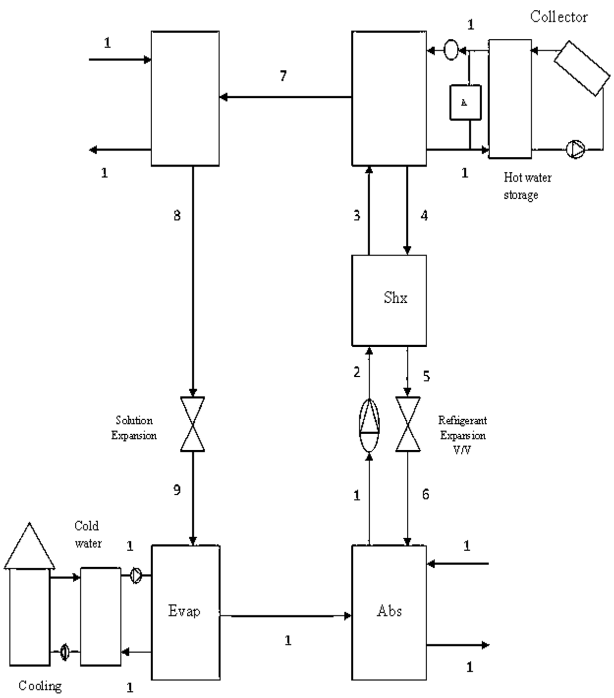
1. Introduction

The solar lithium bromide-water vapour absorption system shown in Fig. 2.1 is mainly composed of two systems. First is the solar collector-tank system composed of flat plate collector and hot water storage tank, and the second is the absorption refrigeration system composed of condenser, evaporator, absorber, solution heat exchanger and generator. Each system has its own loop and characteristics. The first lop is form the solar collectors to the solar tank and the second loop is the absorption's system loop, and it starts from the storage tank to complete the refrigeration cycle.

The solar energy is gained through the collector and is accumulated in the storage tank. Then the hot water in the storage tank is supplied to the generator to boil off water vapour from a solution of lithium bromide-water. The water vapour is cooled down in the condenser by rejecting heat to cooling water and then passed to the evaporator where it again evaporated at low pressure, thereby providing air conditioning to the required space. The strong solution leaving the generator to the absorber passes through a heat exchanger in order to preheat the weak solution entering the generator. In the absorber, the strong solution absorbs water vapour leaving evaporator. The mixing process of the absorbent and refrigerant vapour generates latent heat of condensation and this heat rejected to cooling water system. An auxiliary energy source is provided, so that the hot water is supplied to the generator, when solar energy is not sufficient to heat the water to the required temperature needed by the generator. Solar heated water can be provided at

2. Mathematical Modeling and Design of an Absorption System

The simplified schematic diagram of the system for analyzing and designing purpose is shown in Fig. 2.1. With reference to Fig. 2.1 at point (1) the solution is rich in refrigerant and pump forces the liquid through a heat exchanger to the generator (3). The temperature of the solution in the heat exchanger increases. In the generator thermal energy is added and refrigerant boil off the solutions. The refrigerant vapour (7) flows to the condenser, where heat is rejected as the refrigerant condenses. The condensed liquid (8) flows through expansion valve to the evaporator (9). In the evaporator, the heat from the load evaporates the refrigerant, which flows back to the absorber (10). At the generator exit (4), the steam consists of absorbent-refrigerant solution, which is cooled in the heat exchanger. From points (6) to (1), the solution absorbs refrigerant vapour from the evaporator and rejects heat through a heat exchanger. In order to estimate the size of various component of single-effect water-lithium bromide absorption system i.e. condenser, evaporator, absorber, solution heat exchanger, generator and finding effect of operating system following assumptions must be considered.



The assumptions are:

- Generator and condenser as well as evaporator and absorber are under same pressure.
- Refrigerant vapour leaving the evaporator is saturated pure water.
- Liquid refrigerant leaving the condenser is saturated. Refrigerant vapour leaving the generator has the equilibrium temperatures of the weak solution at generator pressure.
- Weak solution leaving the absorber is saturated.
- No liquid carryover from evaporator.
- Pump is isentropic.
- No jacket heat loss.
- The LMTD expression adequately estimate the latent changes.

Condenser

A liquid state of a refrigerant is must in order to run the refrigeration process. Hence, the vapour phase of a refrigerant from the generator is altered to liquid by condenser. The condensing process of a high pressure refrigerant vapour is done by rejecting the vapour' latent heat to the sink.

The rate of heat rejection is given by:

$$Q_{cond} = m_7 h_7 - m_8 h_8 \qquad (1)$$

$$m_7 = m_8$$

Also,

$$m_{15} = m_{16}$$

Energy balance:

$$m_{15} = \frac{Q_{cond}}{c_p * (T_{16} - T_{15})} \qquad (2)$$

Condenser heat exchanger design

Water cooled horizontal shell and tube type heat exchanger is considered. The overall heat transfer coefficient based on the outside surface of tube is defined as:

$$U = \frac{1}{\frac{D_o}{D_i * h_i} + \frac{D_o}{D_i * F_i} + \left(\frac{D_o}{2*k}\right) * \ln\left(\frac{D_o}{D_i}\right) + F_o + \frac{1}{h_o}} \qquad (3)$$

The value of the fouling factor (F_i, F_o) at the inside and outside of surfaces of the tube can be taken as 0.09 m^2 K/kW and *k* for copper 386 W/ m K.

The heat transfer coefficient for turbulent flow inside the tube is expressed by well-known Petukhov-Popov equation:

$$Nu_i = \frac{\left(f/8\right) * Re_{D_i} * (Pr)}{1.07 + 12.7 \left(f/8\right)^{1/2} \left(Pr^{2/3} - 1\right)} \qquad (4)$$

The equation (3.4) is applicable, if following condition fulfills:

Reynolds Number: $10^4 < Re_{D_i} < 5 * 10^6$

Prandtl Number: 0.5 <*Pr*<2000

Where f is the friction factor and for smooth tubes can be obtained from the following relation:

Here, $f = (0.790 * \ln(Re_{D_i}) - 1.64)^{-2}$ (5)

$$Re_{D_i} = \frac{D_i * v * \rho}{\mu} \qquad (6)$$

Hence,

$$h_i = \frac{Nu_i(k)}{D_i} \qquad (7)$$

Nusselt's analysis of heat transfer for condensation on the outside surface of a horizontal tube gives the average heat transfer coefficient as:

$$h_o = 0.725 * \left[\frac{g * \rho_l(\rho_l - \rho_v) h_{fg} * k_l^3}{(N * \mu_l * D_o(T_v - T_w))}\right]^{0.25} \qquad (8)$$

Here N = Number of tube in a vertical row.

The physical property in equation (2.8) should be evaluated at the mean wall surface and vapour saturation temperature.

The LMTD (log mean temperature difference) for the condenser which is used in the calculation of condenser area can be obtained from equation below:

$$LMTD_{cond} = \frac{T_{16} - T_{15}}{\ln\left[\frac{T_8 - T_{15}}{T_8 - T_{16}}\right]} \quad (9)$$

$$Q_{cond} = U * A_{cond} * LMTD_{cond} \quad (10)$$

Evaporator

The evaporator is component of the system in which heat is extracted from the air, water or any other body required to be cooled by the evaporating refrigerant. Temperature of the evaporator regulates lower pressure level of the absorption system.

The rate of heat absorption is given by:

$$Q_{evap} = m_{10}h_{10} - m_9h_9 \quad (11)$$

Mass balance is:

$$m_9 = m_{10}$$

Also, $m_{17} = m_{18}$

Energy balance:

$$m_{17} = \frac{Q_{evap}}{c_p * (T_{17} - T_{18})} \quad (12)$$

Evaporator heat exchanger design

A falling film evaporator with vertical tubes, housed in cylindrical shell is considered. The correlation employed to determine the heat transfer coefficient h_o was developed for water falling in laminar regime with no nucleation. The equation is:

$$h_o = 0.606(k_l)\left(\frac{\square_l^2}{g * \rho_l^2}\right)^{-1/3}\left(\frac{\Gamma}{\square_l}\right)^{-0.22} \quad (13)$$

The physical property in Eqn. (2.13) should be evaluated at the mean wall surface and water saturation temperature. The heat transfer coefficient for turbulent flow inside the tube is determined by Eqn. (2.7). And hence, overall heat transfer coefficient based on the outside surface of tube is determined by Eqn. (2.3). The LMTD for the evaporator which is used in the calculation of evaporator area can be obtained from equation below:

$$LMTD_{evap} = \frac{T_{17} - T_{18}}{ln\left[\frac{T_{17} - T_{10}}{T_{18} - T_9}\right]} \quad (14)$$

$$Q_{evap} = U * A_{evap} * LMTD_{evap} \quad (15)$$

Absorber

Absorber is a chamber where the absorbent and the refrigerant vapour are mixed together. It is equipped with a heat rejection system and operates under a low pressure level which corresponds to the evaporator temperature. The absorption process can only occur if the absorber is at sensible low temperature level, hence the heat rejection system needs to be attached. The mixing process of the absorbent and the refrigerant vapour generate latent heat of condensation and raise the solution temperature. Simultaneously with the developmental processing of latent heat, heat transfer with cooling water will then lower the absorber temperature and solution temperature, creates a well-blended solution that will be ready for the next cycle. A lower absorber temperature means more refrigerating capacity due to a higher refrigerant's flow rate from the evaporator.

The rate of heat rejection is given by:

$$Q_{abs} = m_{10}h_{10} + m_6h_6 - m_1h_1 \quad (16)$$

At steady state, the net mass flow into each component must be zero. Furthermore, since it is assumed that no chemical reaction occurs between the water and lithium bromide, the net mass flow of these species into any component must also be zero. Since there are two species (water and lithium bromide), there are only two independent mass balances.

Mass balance is:

$$m_1 = m_6 + m_{10}$$

Mass flow equilibrium between the refrigerant and the absorbent that flows in and out of the absorber is a function of the concentration of lithium bromide.

$$m_1 * x_1 = m_6 * x_6$$

Also, $m_{13} = m_{14}$

Energy balance:

$$m_{17} = \frac{Q_{abs}}{c_p * (T_{14} - T_{13})} \quad (17)$$

Absorber heat exchanger design

Water cooled shell and vertical tube type heat exchanger has been used. To design an absorber the total length of a tube bundle need to be calculated for obtaining a certain outlet concentration for a given inlet concentration and solution mass flow rate. This length is then considered together with the tube diameter to see if the exchanger area is sufficient to release the heat of absorption Q_{abs}. For this the heat transfer coefficient *ho* is needed. Wilke's correlation is used to calculate h_o and valid for constant heat flux wall with progressively decreasing difference from isothermal wall outside the entrance region, can be used for the falling film. It is assumed that the flow is fully developed in a wavy, laminar regime and that the bulk solution temperature profile is linear with respect to the transverse coordinate [6]. The Wilke's correlation is:

$$h_o = \frac{k_s}{\delta}[0.029(Re_s)^{0.53}(Pr_s)^{0.344}] \quad (18)$$

Film thickness is given by:

$$\delta = \left(\frac{3 ▯ \Gamma}{\rho^2 g}\right)^{1/3} \quad (19)$$

And the solution Reynolds number:

$$Re_s = \frac{4 * \Gamma}{▯} (20)$$

The properties of solution should be evaluated at temperature of a weak solution and average concentration of LiBr. The heat transfer coefficient for turbulent flow inside the tube is determined by And hence, overall heat transfer coefficient based on the outside surface of tube is determined by equation. The LMTD for the absorber which is used in the calculation of absorber area can be obtained from equation

$$Q_{abs} = U * A_{abs} * LMTD_{abs} \quad (21)$$

Solution Heat Exchanger

A solution heat exchanger is a heat exchange unit with the purpose of pre-heating the solution before it enters the generator and removing unwanted heat from the absorbent. The heat exchange process within the solution heat exchanger reduces the amount of heat required from the heat source in the generator and also reduces the quantity of heat to be rejected by the heat sink (cooling water) in the absorber as well. The heat exchange process occurs between the low temperature of the working fluid and the high temperature of the absorbent which will benefit both.

$$T_5 = T_2(\varepsilon) + (1-\varepsilon)T_4 \quad (22)$$

The overall energy balance on the solution heat exchanger is satisfied if:

$$m_3(h_3 - h_2) = m_4(h_4 - h_5) \ (23)$$

Generator

The desorption process generates vapour and extracts the refrigerant from the working fluid by the addition of the external heat from the heat source; i.e. desorption of water out of a lithium bromide-water solution. The refrigerant vapour travels to the condenser while the liquid absorbent is gravitationally settled at the bottom of the generator; the pressure difference between the generator and the absorber then causes it to flow out to the absorber through an expansion valve. The rate of heat addition in the generator is given by the following equation:

$$Q_{gen} = m_7 h_7 + m_4 h_4 - m_3 h_3 \quad (24)$$

Mass balance is

$$m_3 = m_4 + m_7 \quad (25)$$

And $m_3 * x_3 = m_4 * x_4$

Also, $m_{11} = m_{12}$

Energy balance:

$$m_{11} = \frac{Q_{gen}}{c_p * (T_{11} - T_{12})} \quad (27)$$

Generator heat exchanger Design

The falling film type shell and tube heat exchanger is used as generator. A survey on the literature has not produced useful correlations for calculating the exact heat transfer coefficient h_o. Nevertheless the mass transfer characteristics of this kind of exchanger depend upon a wide range of parameters and in this regard it is not possible to make any prediction for a novel design generator. However, for preliminary design calculation, the value of overall heat transfer coefficient will be used as 850 W/m^{2o}C.

The LMTD for the generator which is used in the calculation of generator area can be obtained from equation below:

$$LMTD_{gen} = \frac{(T_{11} - T_4) - (T_{12} - T_7)}{ln\left[\frac{T_{11} - T_4}{T_{12} - T_7}\right]} \quad (28)$$

$$Q_{gen} = U * A_{gen} * LMTD_{gen} \quad (29)$$

Expansion valve

An expansion valve is a component that reduces the pressure and splits the two different pressure levels. In a simple model of a single effect absorption refrigeration system, the pressure change is assumed only to occur at the expansion valve and the solution pump. There is no heat added or removed from the working fluid at the expansion valve. The enthalpy of the working fluid remains the same on both sides. The pressure change process between the two end points of the expansion valve, while there is no mass flow change and the process is assumed as an adiabatic process.

Solution Pump

Although the main distinction between compression and absorption refrigeration is the replacement of the mechanically driven system by a heat driven system, the presence of a mechanically driven component is still needed in an absorption system. A solution pump will mainly circulate and lift the solution from the lower pressure level side to the higher pressure level side of the system. To maintain this pressure difference, a centrifugal type pump is preferable. Assuming the solution is an uncompressible liquid, in other words the specific volume of the liquid (v) will not change during the pumping process, and the power requirement to lift the solution with mass flow m_1 from pressure level P_1 to P_2 and certain pump efficiency is calculated by following equation:

$$W_{pump} = \frac{m_1 * v_1(P_2 - P_1)}{\eta_{pump}} \quad (30)$$

COEFFICIENT OF PERFORMANCE

Efficiency of an absorption refrigeration system can be easily expressed by a Coefficient of Performance (COP) which is defined as the ratio between the amount of heat absorbed from the environment by the evaporator and the heat supplied to the generator to operate the cycle and pump work.

$$COP = \frac{Q_{evap}}{Q_{gen} + W_{pump}} \quad (31)$$

As the work supplied to the absorption system is very small compared to the amount of heat supplied to the generator, generally the amount of work is often excluded from the calculation.

$$COP = \frac{Q_{evap}}{Q_{gen}} \qquad (\text{Because} W_{pump} \ll Q_{gen})$$

THE ENTHALPY AND CONCENTRATION

The accurate measurement of enthalpies and concentration at various state points (from 1 to 10 as shown in Fig. 2.1) is very important for simulation of vapour absorption system.

The Enthalpy

The enthalpies of the solution of the refrigerant and absorbent at points 1 to 6 is a function of the solution temperature and concentration as given in ASHRAE chart (Appendix A). For computer simulation enthalpies at various points can be expressed in multi variable polynomials in terms of the solution temperature (T °C) and concentration (X % LiBr).

$$h = \sum_0^4 A_n X^n + T \sum_0^4 B_n X^n + T^2 \sum_0^4 C_n X^n \quad (2.33)$$

A_n, B_n and C_n are constant coefficient. In extended form the above equations can be written as:

$$h = A_0 + A_1X^1 + A_2X^2 + A_3X^3 + A_4X^4 + T(B_0 + B_1X^1 + B_2X^2 + B_3X^3 + B_4X^4) + T^2(C_0 + C_1X^1 + C_2X^2 + C_3X^3 + C_4X^4) \quad (2.34)$$

The value of constants is given in Table 2.1. The Eqn. (2.44) is valid /applicable if:

Concentration range: 40 %< X<70 % LiBr.

Temperature range: 15 °C< T < 165 °C.

From Fig 3.1: $h_1 = h_2$ and $h_5 = h_6$

Table 2.1: The value of constants

$A_0 = -2024.33$	$B_0 = 18.2829$	$C_0 = -3.7E-2$
$A_1 = 163.309$	$B_1 = -1.16$	$C_1 = 2.88E-3$
$A_2 = -4.88161$	$B_2 = 3.24E-2$	$C_2 = -8.15E-5$
$A_3 = 6.308E-2$	$B_3 = -4.034E-4$	$C_3 = 9.9E-7$
$A_4 = -2.91E-4$	$B_4 = 1.85E-6$	$C_4 = -4.4E-2$

The enthalpies of refrigerant (at state points 7 to 10 as shown in Fig 2.1) are calculated from steam table. However, for computer simulation enthalpies at various points can be expressed as [2]:

The enthalpy at exit of generator (state 7) is given as:

$$h_7 = A + B * T_{gen} + C * T_{gen}^2 \quad (2.35)$$

The value of constant is given as:

A = 2533.996; B = 0.97901595 and C = 0.00493525

The enthalpy at exit of condenser (state 8) is given as:

$$h_8 = A + B * T_{cond} + C * T_{cond}^2 \quad (2.36)$$

The value of constant is given as:

A = 0.1534285; B = 4.1956286 and C = 0.000257

$h_9 = h_8$

The enthalpy at exit of evaporator (state 10) is given as:

$$h_{10} = A + B * T_{evap} + C * T_{evap}^2 \quad (2.37)$$

The value of constant is given as:
A = 2510.2114; B = 1.8614286and C = 0.00071429

The concentration

The solution concentration, "X" is defined as the ratio of the mass of LiBr to the mass of solution of water and LiBr. The computation of the concentration depends upon the temperature. The concentration is usually determined by ASHRAE charts, allocating the other two properties of the state point.

MODELLING OF FLAT PLATE SOLAR COLLECTOR

In this section a mathematical model is developed to simulate the flat-plate collector. Various collector areas are considered and the effect of collector areas is evaluated against the auxiliary heat required.

In this system inlet hot water temperature is 78 ℃ which is being fed to generator supplied by solar collector .this temperature is used to heat the weak solution in generator .FPC (flat plate collector) works between 30℃ to 90 ℃ temperature due to simple design of it is recommend for this system

General energy equation for flat plate collector is given by

$Q = I \times A_c$
(2.38)

$Q =$ Amount of solar radiation received by collector (in W)

$I =$ Intensity of solar radiation (W/m^2)

A_c =Collector surface area (m^2)

$Q = \dot{m} \times C_p \times (T_p - T_a)$
(2.39)

From equation (1) and (2)

$\dot{m} \times C_p \times (T_p - T_a) = I \times A_c$

Where m = mass flow rate of water flowing in the tube of collector

Useful heat gain is expressed as

$$Q_u = F_r[I \times \tau_\alpha \times A_c - U_l \times A_c(T_i - T_\alpha)]$$

$$\dot{m} \times C_p \times (T_{11} - T_{12}) = F_r[I \times \tau_\alpha \times A_c - U_l \times A_c(T_i - T_\alpha)]$$

$F_r =$ Heat removal factor

$$F_r = \frac{\dot{m} \times C_p \times (T_{11} - T_{12})}{[I \times \tau_\alpha \times A_c - U_l \times A_c(T_i - T_\alpha)]}$$

F_r From duffci and Beckman

$$F_r = m \times C_p \times (1 - \exp(\frac{-A_u \times U_l}{m \times C_p})$$

Collector efficiency -

$$(\eta) = \frac{Q_u}{IA_c}$$

Results

Table 1: Thermodynamic properties of state points corresponding to input data

state	h(kj/kg)	m (kg/s)	p (kpa)	T°C	%LiBr (byweight)
1	58.76	0.0218	1.22	30	49
2	58.76	0.0218	1.22	30	49
3	76.96	0.0218	1.22	38.4	49
4	173.58	0.0178	4.30	70	60
5	150.42	0.0178	4.30	58	60
6	150.43	0.0218	1.22	58	60
7	2626.71	0.004	4.30	70	0
8	126.04	0.004	4.30	30	0
9	126.04	0.004	1.22	30	0
10	2528.89	0.004	1.22	10	0

Table 1: Component Capacities

Evaporator capacity	10.5KW
absorber heat rejected	11.51KW
heat input to generator	11.5KW
condenser heat rejected	10.0KW
COP	0.9130

RESULTS AND DISCUSSION

A mathematical model has been developed and parametric study of lithium bromide-water vapour absorption air conditioning system is carried-out. The effect of generator temperature on ratio of the rate of solution circulation and on the coefficient of performance for different condenser temperature, evaporator temperature, absorber temperature has been evaluated and discussed. After analysis, the design of different component of 10.50 kW, vapour absorption system i.e. the condenser, evaporator, absorber, solution heat exchanger and generator has been presented. Finally, the availability of solar energy, useful energy and auxiliary heat requirement for running the absorption air-conditioning system from 09:00 hrs to 18:00 hrs (9 hours/day) has been calculated. The above time frame has been selected judiciously as the energy requirement for cooling purposes attains its maximum value for months April to June (Summer Season).

3.1 PARAMETRIC STUDY OF ABSORPTION SYSTEM

3.1.1 Effect of generator temperature on COP for different temperature of condenser

Figure 3.1 shows the variation of the coefficient of performance with generator temperature (T_{gen}) for different condenser temperature (T_{cond}) with absorber temperature as (T_{abs} = 30°C) and evaporator temperature as (T_{evap} = 10°C). The graph plotted has been keeping T_{cond} as 25 ^{o}C, 30^{o}C and 35^{o}C.From figure it is evident that the COP initially exhibits an appreciable increase with the rise in the generator temperature which results in increase in COP due to the fact that when generator temperature increases the mass flow of the refrigerant also increases which results in increase in cooling capacity and hence COP of the system increases. However, for T_{gen}< 65 ^{o}C curve corresponding to condenser temperature (T_{cond} = 35 ^{o}C) COP is least due to the fact that as T_{cond} increases, the condensing temperature increases and hence causes less heat transfer in the condenser. This results in an increase in the temperature and in enthalpy of the refrigerant at the condenser outlet. Hence, the cooling capacity decreases as does the COP.

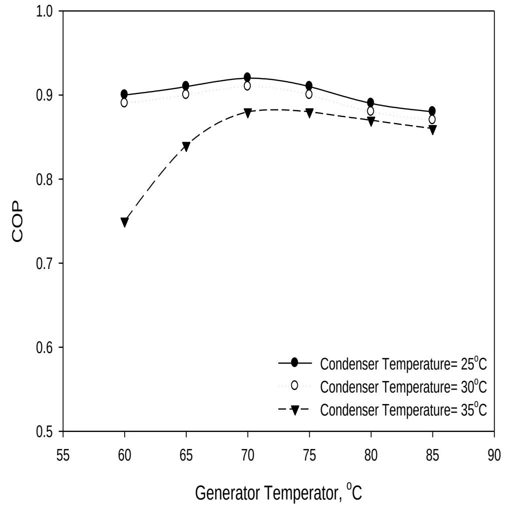


Fig. 3.1 Variation of COP with generator temperature for different condenser temperature at T_{abs} = 30°C and T_{evap} = 10°C.

3.1.2 Effect of generator temperature on COPfor different temperature of evaporator

Figure 3.2 shows the variation of generator temperature (T_{gen}) with COP respectively for various evaporator temperature (T_{evap}) at keeping absorber temperature as (T_{abs}=30°C) and condenser temperature as (T_{cond}=30°C). The different evaporator temperatures taken are 8°C, 10°C and 12°C. From the figure 3.2 it is clear that with increase in T_{evap} leads to significant increase in COP.

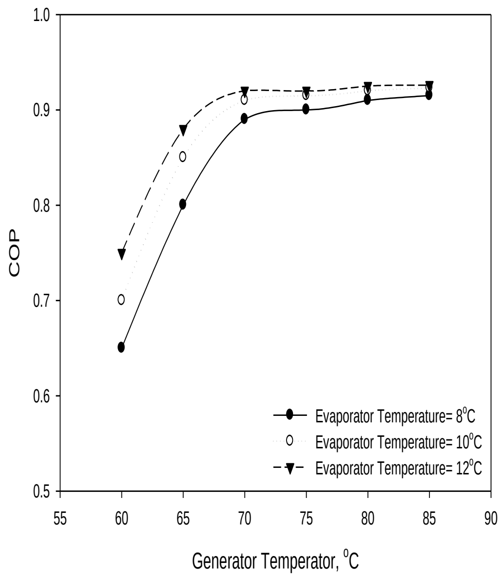


Fig. 3.2: Variation of COP with generator temperature for different evaporator temperature at T_{abs}=30°C and T_{cond}=30°C.

3.1.3 Effect of generator temperature on COP and heat input for different absorber temperature

Figure 3.4 plots graph between COP and generator temperature (T_{gen}). Figure 3.5 graphically shows the variation between Q_{gen} and generator temperature (T_{gen}). Both the graphs have been plotted for different absorber temperature (T_{abs}) at keeping condenser temperature as (T_{cond}=30°C) and evaporator temperature as (T_{evap}=10°C). The different absorber temperatures taken are 25°C, 30°Cand 35°C.

It is clear from Figure 3.4 and 3.5 that for T_{gen}<65°C, COP shares inverse relation with T_{abs}. This is primarily due to the fact that an appreciable increase in heat input (Q_{gen}) is observed.

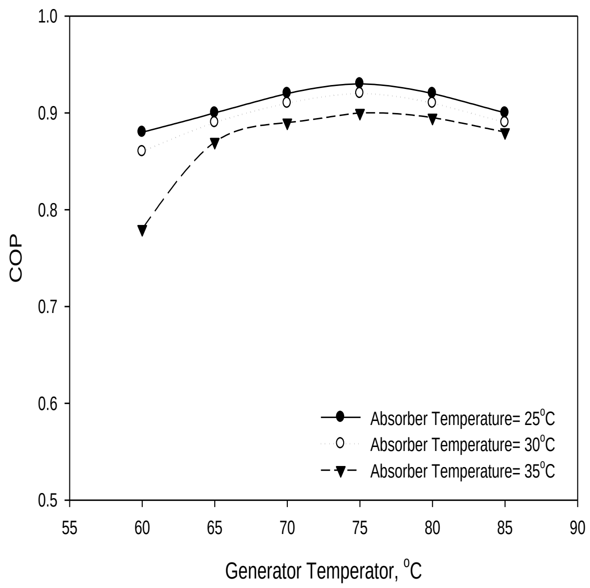


Fig. 3.4: Variation of COP with generator temperature for different absorber temperature at a particular value of T_{cond} = 30°C and T_{evap} = 10°C.

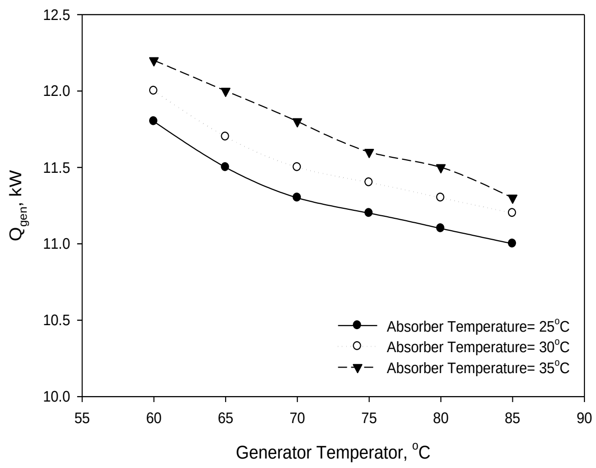


Fig. 3.5: Variation of load on generator side for different Absorber temperature at particular value of T_{evap} = 10°C and T_{cond} = 30°C

3.2 HEAT EXCHANGER DESIGN

The type of exchangers employed in each unit and their heat transfer characteristics are presented in this section. This is a critical step of planning a machine and can be rather complex. The prediction of the overall heat transfer coefficient U is affected by many uncertainties. A great number of parameters have effect on the final performance of the exchangers and unexpected results often arise in practice. All the components employ shell and tubes heat exchangers except the solution heat exchanger which is plate and frame type. Copper used for the construction of shell and tube heat exchanger and stainless steel used for plate and frame type heat exchanger. All the streams coming from the external circuits flow inside the tubes The thermodynamic property/condition from state point 1 to 18 (as shown in Fig. 2.1) required for heat exchanger design is attached as Annexure – 1.

Table 3.1: Condenser heat exchanger characteristics

Heat Exchanger Type and Specifications	Shell and Tube (horizontal tubes and two tube passes) Outside diameter (D_o)= 0.01905m Inside diameter (D_i)= 0.013m
sMass flow rate of cooling water ($\dot{m}_{15}$)	1.0 kg/sec
Condenser load (Q_{cond})	10 kW
LMTD	1.3 ℃
Overall heat transfer coefficient (U)	2.05 kW/m^2°C
Area	3.531 m^2
Length of tube	1.843 m
No. of tubes used	16

Table 3.2: Evaporator heat exchanger characteristics

Mass flow rate of water ($\dot{m}_{17}$)	0.084 kg/sec
Evaporator capacity (Q_{evap})	10.50 kW
Area	3.733m^2
Length and number of tubes	2 m, 16

Table 3.3: Absorber heat exchanger characteristics

Heat Exchanger Type and Specifications	Shell and Tube (Vertical tubes and four tube passes) Outside diameter (D_o)= 0.01905m Inside diameter (D_i)= 0.013m
Mass flow rate of cooling water ($\dot{m}_{13}$)	1kg/sec
Absorber heat rejection (Q_{abs})	11.51 kW
LMTD	3.3
Overall heat transfer coefficient (U)	1820.2 W/m^2°C
Area	1.91 m^2
Length and number of tubes	1m, 16 m

Table 3.4: Generator heat exchanger characteristics

Heat Exchanger Type and Specifications	Shell and Tube vertical tube type Outside diameter (D_o)= 0.01905m Inside diameter (D_i)= 0.013m
Mass flow rate of hot water ($\dot{m}_{11}$)	0.457 kg/sec
Heat input in generator (Q_{gen})	11.5 kW
LMTD	12.6
Overall heat transfer coefficient (U)	1074.81 W/m^{2o}C
Length and number of tubes	7.089 , 16 m

Table 3.5: Solution heat exchanger characteristics

Heat Exchanger Type and Specifications	Shell and Tube vertical tube type Outside diameter (D_o)= 0.01905m Inside diameter (D_i)= 0.013m
LMTD	29.76
Overall heat transfer coefficient (U)	68.04W/m^{2o}C
Area	0.2031 m^2
Length tube per pass and no of tube	11cm 16

3.3 SIMULATION RESULTS FOR FLAT PLATE COLLECTOR

In this section a flat plate collector has been considered. The results elaborated below are obtained by considering the vapour absorption air-conditioning system working for 9 hours i.e. 9:00 am to 6:00 pm (office hours) for month April, May and June in Dehradun, Uttarakhand. The operation conditions of the absorption system is T_{gen} = 80°C, T_{evap} = 10°C, T_{cond} = T_{abs} = 30°C and Q_{gen} = 11.51 kW

.3.3.1 Effect of collector area on the collector useful heat gain

The influence of collector area on the heat gain is shown in Fig. 3.6. It is clear from the figure, an increase in the collector area results in an increased collector useful heat (Q_{useful}) gain.

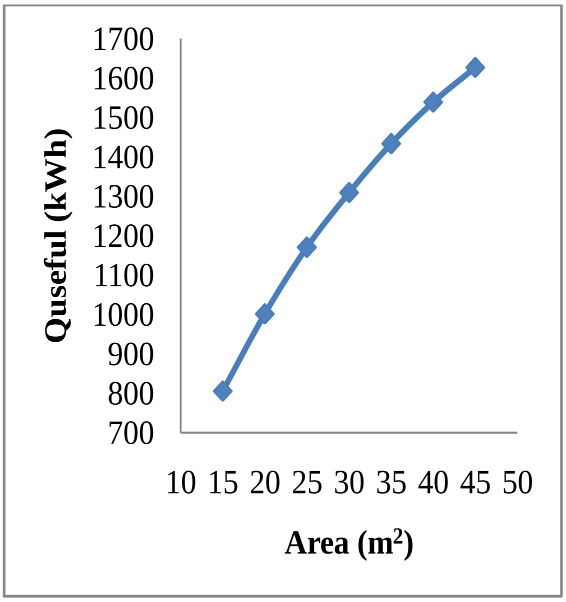


Fig. 3.6: Variation of Q_{useful} with Collector area

CONCLUSIONS

In this study modeling and design has been carried out for 10.5 kW of solar assisted vapour absorption air conditioning system using Lithium Bromide-

Water as refrigerant. Also parametric study has been carried out to study the effect of condenser temperature, generator temperature, evaporator temperature, absorber temperature on co-efficient of performance (COP) and effect of generator temperature on specific solution circulation rates (f). On the basis of the parametric study the different components of vapour absorption refrigeration system i.e. condenser, evaporator, absorber and solution heat exchanger has been designed. Also the availability of solar energy for running the air-conditioning system for 09 hours i.e. 09:00 hrs to 18:00 hrs (office hours) for April, May and June in Dehradun, Uttarakhand has been studied. The above time frame was selected judiciously as the energy requirement for cooling purposes attains its maximum value for months April to June (Summer Season).

Based upon the study following conclusions have been drawn:

1. A model has been developed which can predict the COP of the system. Each component of the absorption system i.e. generator, condenser, evaporator and absorber has been designed.
2. Higher evaporator and generator temperature, results in higher coefficient of performance (COP) of the system due to the fact that as generator temperature increases, the heat transfer to the solution in the generator increases, result in the increase of mass flow rate and so does the COP.
3. Low condensing temperature results in higher COP due to the fact that as T_{cond} increases, the condensing temperature increases and hence causes less heat transfer in the condenser, which results in an increase in temperature and enthalpy of the refrigerant at the condenser outlet. Hence, the cooling capacity decreases as does the COP.
4. For generator temperature from 65°C to 80°C, the absorption system work efficiently.

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