# Design and Performance Validation of Vapour Absorption Solar Air Conditioning System

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***Abstract:*** The Solar based absorption systems are used for the reduction of power load that is caused because of the utilization of the compressors. There have been different absorption sets that are tried for space cooling applications by bringing in different solar based heat inputs. The $NH_3 – H_2O$ based absorption system need a working temperature of 120°C to 150°C, and requires concentrators with tracking and attracting higher qualities, however $NH_3 – H_2O$ vapour absorption systems can operate at lower temperatures and can utilize FPC or ETC solar water heating frameworks as generators. In the near future, this will bring in low cost and low maintenance. The single effect 1 KW, $NH_3 – H_2O$ absorption system with evacuated tube solar collector is attempted in this research work. The testing of the system and comparison of COPs with standard vapour absorption Solar Air Conditioning system are done.

***Keywords:*** *Absorption; solar energy; Ammonia - Water; $NH_3 – H_2O$; evacuated tube collector; space cooling; coefficient of performance*

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## I.INTRODUCTION

Many number of small scale office buildings today are installed with a conventional cooling technique which normally utilizes an compressor system that is electrically driven and shows several disadvantages such as higher power consumption, higher peak load electricity demands and typically it uses refrigerants that causes harmful impacts over the environment. There are various refrigerant – absorbent sets utilizes among which the most widely recognized are water-lithium bromide and ammonia water. The Single Effect Chillers have an operating condition of hot water temperature varying from 75°Cto120°C when water is pressurized. The COP of the system ranges from 0.65 to 0.75.

## II.METHODOLOGY

The simplified schematic diagram of the system for analyzing and designing purpose is shown in Fig. 1 at point 1 the solution is rich in refrigerant and pump forces the liquid through a heat exchanger to the generator . 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 [1]. 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. [2] From points 6 to 1, the solution absorbs refrigerant vapour from the evaporator and rejects heat through a heat exchanger. [3] In order to estimate the size of various component of single-effect aqua ammonia absorption system i.e. condenser, evaporator, absorber, solution heat exchanger, generator and finding effect of operating system following assumptions must be considered. [4,5]

**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:

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

$$m_7 = m_8$$

Also, $m_{15} = m_{16}$

Energy balance:

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

**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:

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

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/mK.

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

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

The equation (3,4) is applicable, if following condition fulfills:

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

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)

$$Re_{D_i} = D_i * \vartheta * \rho \quad (6)$$

Hence,

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

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

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

Here N = Number of tube in a vertical row. The physical property in equation (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:

$$Q_{evap} = m_{10} h_{10} - m_9 h_9 \quad (11)$$

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* was developed for water falling in laminar regime with no nucleation. The equation is:

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

The physical property in Eqn. (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. 13. And hence, overall heat transfer coefficient based on the outside surface of tube is determined by Eqn. (13). The LMTD for the evaporator which is used in the calculation of evaporator area can be obtained from equation below:

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

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

**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:

$$Q_{abs} = m_{10} h_{10} + m_6 h_6 - m_1 h_1 \quad (16)$$

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: $m_{17} = \frac{Q_{abs}}{C_p * (T_{14} - T_{13})}$ (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 . For this the heat transfer coefficient *ho* is needed. Wilke's correlation is used to calculate $h$ 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 . 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:

$$Re_s = \frac{4*\Gamma}{\rho} \quad (20)$$

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:

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

**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

$$m_3 x_3 = m_4 x_4 \quad (26)$$

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. 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 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 from pressure level P to P 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**[3]

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.

$$COP = \frac{Q_{evap}}{Q_{gen}} \quad (Because\ W_{pump} \ll Q_{gen}) \quad (32)$$

A 1 KW $NH_3 - H_2O$ vapour absorption system is constructed using ETC based Solar Hot Water System considering the

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solar data for ChennaiIndia with Latitude 13.0827° N , Longitude 80.2707° E for final details.

The parts such as generator, condenser, evaporator, absorber and hear exchanger are designed, fabricated and assembled for test experiment as shown in Fig. 2 [6]

## III.TEST SETUP

The generator and the condenser are oriented horizontally and the evaporator and the absorber are oriented vertically in the process diagram. The high temperature water will be circulated in to the storage tank, and is sent to the collector after successful utilization for the purpose of heating. The temperature of the water in the storage tank is increased and the water is circulated into the generator using a pump [7].

For the purpose of compensation of hear or temperature in the storage tank, an auxiliary heater is used inside the tank. This is needed in the system for the vaporization in the generator. The absorber sends the mixture into the generator. The vapour refrigerant will get converted into cold liquid over there and is sent to the evaporator though capillary tubes. This low temperature liquid in the evaporator will absorb the temperature of the cold water and itself will get included into the absorber. Then, this refrigerant will be converted into high temperature refrigerant and is sent into the generator for the purpose of recycling.

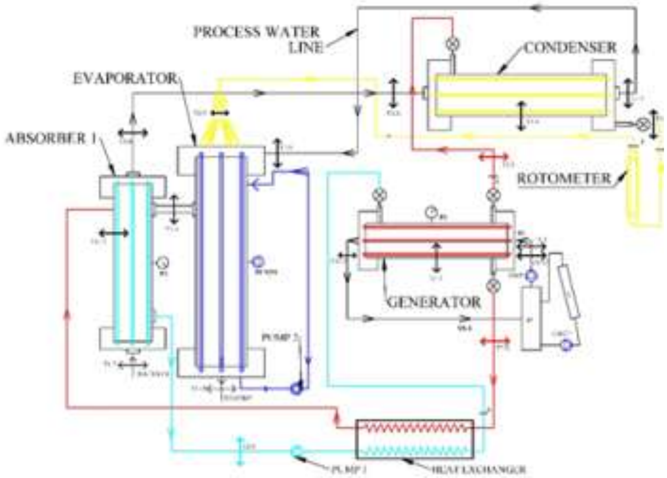


**Fig-1**: Process instrumentation and test setup

All the parts of the system in which the generator and the condenser are horizontally placed and the absorber and the evaporator are placed vertically[8] are shown in Fig-2.

**Fig-2**: Assembly of Experimental setup

**Testing**

The experiment is carried out under controlled condition of various generator temperatures such as 55, 60, 65, 70, 75, 80, 85, 90 and 95°C. The high temperature water from the ETC Hot Solar Water System is collected and coursed into the system from the generator [9]. The performance is assessed against the input conditions such as evaporator, generator and temperature of the inlet. The high temperature water is gathered from the evacuated tube collector into the generator. The testing is carried out for Generator Temperature Vs COP of the system.

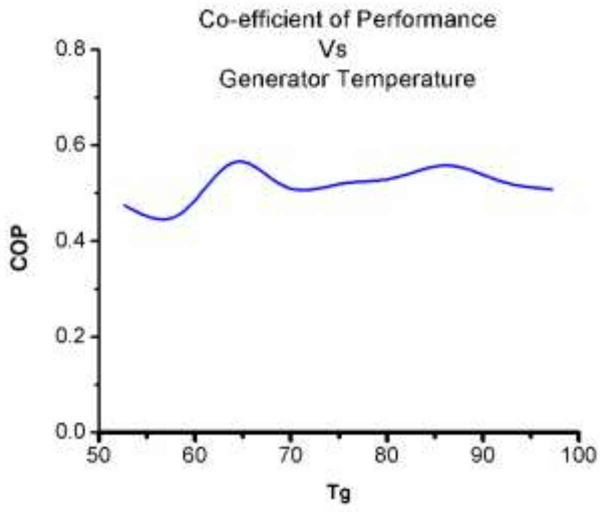


**Fig-3**:Coefficient of Performance Vs Generator temperature

## IV. ANALYSIS OF THE SYSTEM

To study the influence of generator temperature on efficiency of the system, the performance analysis is done which helps in deducing the design criteria for $NH_3 – H_2O$ vapour absorption system for solar Air Conditioning. Here the inlet temperature of the generator changes depending up on the available solar radiation intensity at the different locations [10]. The temperature of the generator is controlled to obtain constant and uniform system performance and to accomplish power and energy conservation.

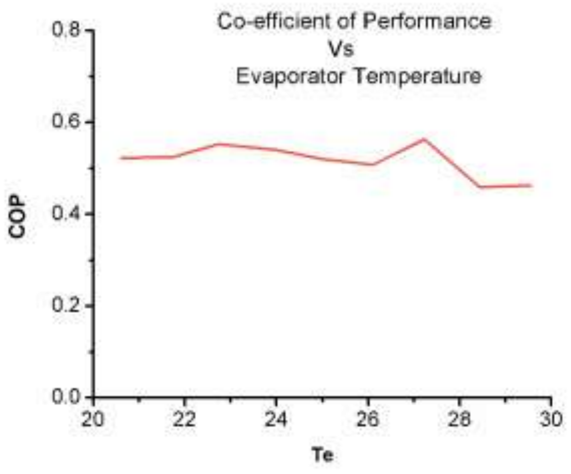


**Fig-4**: Coefficient ofPerformanceVs. Evaporator Temperature

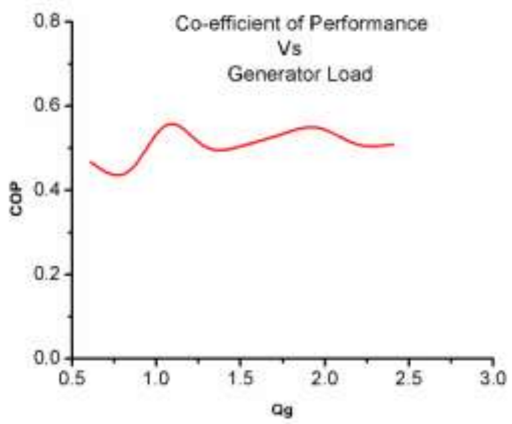


**Fig-5**:CoefficientofPerformanceVsGeneratorLoad(Qg)

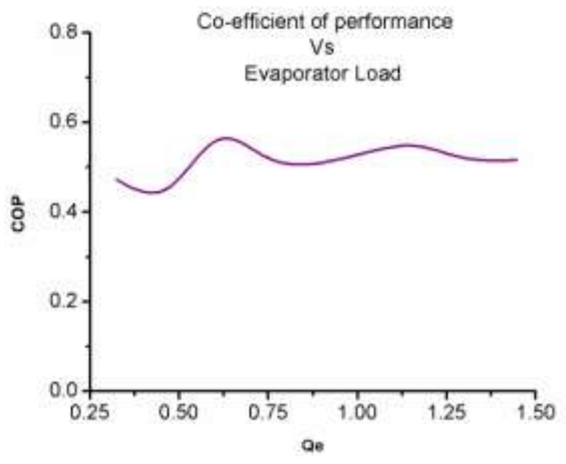


**Fig-6**:Coefficientof PerformanceVs Evaporatorload (Qe)

## V. CONCLUSION

The $NH_3$ – $H_2O$ vapour absorption system is designed and evaluated for the required cooling load with various components such as generator, evaporator, condenser and heat exchanger with appropriate vacuum and water circulation framework.

The increase in Coefficient of Performance with increase in Generator Temperature is seen, however the Coefficient of Performance settles and becomes stable at higher Generator Temperature. The optimum Coefficient of Performance is accomplished at Generator Temperature of 83.2°C and Evaporator Temperature of 23.59°C in the current case.

The cooling load correspondingly varies between 0.95 to 1.58 kW. The Coefficient of Performance of the system is comparable with the current $NH_3$ based absorption systems.

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