Theoretical design of solar-powered vapor absorption refrigeration system coupled with latent heat energy storage

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Abstract. In this work, the vapor absorption refrigeration system (VARS) with a cooling capacity of 1 kW is designed. VARS is designed to be driven by hot water available from the solar thermal collector during the day. In the Sun's absence, the system is designed to run by a coupled latent heat energy storage system. The unit has three major components: evacuated tube collector, latent heat energy storage, and VARS. The system is designed for a cooling capacity of 1 kW. The size of other components has been calculated by adopting the heat exchanger design approach and the relevant equations presented in the published works. The corresponding size of absorber, generator & condenser comes out to be 1.412 kW, 1.46 kW & 0.96 kW, respectively for the cooling capacity of 1 kW. The evacuated tube collector of 100 LPD capacity is sufficient to provide the required heat to run the VARS and to charge the thermal energy storage. Latent heat energy storage of 2.25 kW charging and discharging time of 3 hours is coupled with the system. The designed system gives the COP of 0.875.

Keywords: Solar vapor absorption refrigeration system, solar thermal collector, latent heat energy storage, evacuated tube collector.

1. Introduction

Refrigeration has become an essential part of human life in one or another way. The process which cools down or maintains the temperature of the substance, lower than surrounding is called refrigeration. The most conventional system to achieve the refrigeration is vapor absorption refrigeration system (VARS). It is a low-grade heat-driven system. Conventionally, electric energy is the prime source to operate it, but limited electricity availability provokes to think of an alternate source of energy for the energy-consuming system. In this work, the author presented a theoretical design of water LiBr VARS, operated on solar energy during sunny days, and latent heat energy storage system during non-sunny hours.

Solar energy emerges as a great alternative to conventional energy. It is a clean and eco-friendly source of energy. It is abundant and reported that if only 5% of incident solar radiations are captured and utilized efficiently, the energy demand will be fulfilled worldwide. Though solar is clean and eco-friendly, it is non-continuous during the night and sun-off days[1]. The energy storage could have answered this issue of discontinuity and will make the system run without hindrance. The current research shows that many researchers did a numerical and experimental study on either solar
refrigeration or latent heat energy storage\cite{2,6}. Lie et al.\cite{2} presented phase change material (PCM) TRNSYS model portable refrigeration. An off-vehicle refrigeration unit charged the system, and cooling is provided when PCM discharges. In such a system, PCM is directly used for cooling. Other solar-assisted refrigeration works are reported in references \cite{5,7}.

The theoretical work on solar-driven water- LiBr VARS unit with latent heat energy storage is not available. This urges the authors to present a theoretical design of a 1kW (0.25 TR) VARS system driven by hot water supplied by the solar collector during daytime and by PCM-based thermal energy storage system Sun-off days or night time. The layout of the solar-powered vapor absorption refrigeration system coupled with latent heat energy storage is shown in Figure 1.

2. Vapour Absorption Refrigeration System

The water- lithium bromide refrigeration system has become popular in the last decades for space cooling application because of low-grade energy operation. In this system, water is used as a refrigerant and a homogenous solution of LiBr as absorbent. When the water-LiBr solution is heated in a generator, LiBr does not evaporate, and water gets evaporated thus, both can be separated without the use of any additional unit of rectification. The single effect VARS considered for calculation is presented in Figure 2.

2.1. Thermal Analysis

Each component's thermal analysis evaluates the operating parameters like temperature, pressure, solution strength, mass flow rates, and heat exchanging capacity. The design calculations are based on the following requirements:

- Required Refrigeration Capacity: 1kW
- Desired Evaporator Temperature: 5°C
- Condenser Temperature: 45°C
- LiBr Strength $X_{LiBr}$: 0.55

At first, the saturation pressure and temperature are estimated at each component from the Pressure-Temperature-Concentration characteristics of Water- LiBr solution.
The following values are obtained corresponding to H$_2$O (refrigerant) at 5˚C:
Saturation Pressure = 8.7 mbar = 870 Pa and solution strength is 60%.
Thus, Evaporator Pressure ($P_E$) = Absorber Pressure ($P_C$) = 870 Pa.
Since condenser temperature is 45 ˚C, the saturation temperature of water = 9.58 kPa
Thus, vapor generator pressure $P_G$ = 9.58 kPa
For $P_G$ = 9.58 kPa and $T_G$ = 80 ˚C.
Required H$_2$O-LiBr strength $X_{LiBr}$ = 55%
Since the hot water temperature is limited to 85 ˚C, the generator's solution temperature is restricted to approx. 80 ˚C only. The next step is calculating the heat potential of working fluids in enthalpy at the different locations within the unit.

2.1.1. Enthalpy Calculation. The enthalpy of the water and water vapour can be estimated using the general expression as follow:
For water, $h = 4.1867t$ kJ/kg $\quad t$ in ˚C. (1)
For water vapour, $h = 2501 +1.88t$ kJ/kg, $t$ in ˚C. (2)
At generator exit, phase of fluid is vapour so, enthalpy of water vapour ($h_3$) at 80 ˚C is given by Eqn. (1)
At Absorber, $T_{abs} = 45$ ˚C & Pressure = 8.52 kPa at solution concentration $X_1 = 0.58$.
enthalpy $h_1$ corresponding to inlet to generator = -145 kJ/kg
While the enthalpy $h_2$ corresponding to solution at $X_2 = 0.55$ $T_G = 80$ ˚C is -70 kJ/kg.
At the evaporator exit, enthalpy of water vapour leaving the evaporator at 5 ˚C is given by Eqn. (2)
At condenser exit enthalpy of water $h_4$ = 146.5 kJ/kg
In enthalpy calculation, $h_5$ is not considered as it not the integral part of the calculation. The next step is to calculate the required mass flow rate of working fluids.

2.1.2. Mass Flow Rate Calculation. The following steps are adopted to calculate the mass flow rate at various points of the refrigeration system.
At Evaporator; $m_6 = 1/(h_6-h_7)$ = 0.00043 kg/s (3)
At Absorber; $m_2 x_2 + m_6 x_6 = m_1 x_1$ & $m_2 + m_6 = m_1$ gives $m_1 = 0.44 \times 10^{-2}$ kg/s (4)
At Generator; $x_1 m_1 = x_2 m_2 + x_3 m_3$ and $m_1 = m_2 + m_3$ gives $m_3 = 0.004$ kg/s. (5)

2.1.3. Heat Exchange Capacity
Absorber Capacity: \( Q_a = m_1 h_2 + m_2 h_3 - m_1 h_1 \) gives \( Q_a = 1.416 \text{ kW} \) 

(6)

Generator Capacity \( Q_d = m_2 h_2 + m_3 h_3 - m_1 h_1 \) gives \( Q_d = 1.46 \text{ kW} \) 

(7)

Condenser Capacity \( Q_c = m_3 (h_3 - h_4) \) gives \( Q_c = 0.965 \text{ kW} \) 

(8)

In this section capacity estimation of each component is done and the summarized capacity is presented in Table 1 as below:

| Component     | Capacity[kW] |
|---------------|--------------|
| Generator     | 1.462        |
| Condenser     | 0.965        |
| Absorber      | 1.412        |
| Evaporator    | 1            |

And, theoretical COP comes out to be 0.875 with a circulation factor of 10.232.

The next step in the calculation is calculating the dimension of the component and heat transfer arrangement.

2.2. Sizing of VARS Component

The sizing of each component is based on the heat exchanger design approach. The standard assumptions and approximation are made as per requirement. The standard equations used for each component are:

Heat exchange capacity,
\[ Q = UA \Delta T_{lmtd} \] 

(9)

The overall heat transfer coefficient,
\[ \frac{1}{U} = \frac{1}{h} + \frac{r_i \ln \left( \frac{r_o}{r_i} \right)}{k_{sol}} + \frac{r_o}{r_o h_{sol}} \] 

(10)

Reynolds Number is given by \( \text{Re} = \frac{4m}{\pi D \mu} \) 

(11)

Logarithmic Temperature Difference is \( \text{LMTD} = \frac{T_h - T_{h_{\text{avg}}}}{\ln \left( \frac{T_h - T_{h_{\text{avg}}}}{T_{c_{\text{avg}}} - T_{c_{\text{avg}}}} \right)} \) 

(12)

2.2.1. Generator. Since the generator's theoretical heat exchange capacity is 1.5 kW, all the calculations are done with respect to that capacity.

The calculation for heat transfer coefficient of single-phase flow of water inside the tube \( h_i \):

The heat transfer coefficient of water inside the tube \( h_i \) is given by Gnielinski [8]:
\[ h_i = \frac{f}{2} \left( \frac{\text{Re} - 1000}{ \text{Pr} } \right), \quad \frac{k}{D} \]

(13)

**The calculation for heat transfer coefficient of H\textsubscript{2}O-LiBr solution outside the tube \( h_{sol} \)**

The following equation calculates the heat transfer coefficient of H\textsubscript{2}O-LiBr solution outside the tube:

\[ h_{sol} = 9.7 \text{Pr}^{0.5} (1.8 \text{Pr}^{0.17} \frac{4}{10} \text{Pr}^{1.2} 10\text{Pr}^{10}) \left( \frac{qD}{\mu_j h_f} \right)^{0.67} \frac{C_f f}{k_f} \frac{k_f}{D} \]

(14)

Substitute the value of \( h_i \) and \( h_{sol} \) in the Eqn. 10 of U and solve for area \( A \) in Eqn. 9.

\[ A = \pi d l \]

where \( l = \) length of the tube

On solving the above Eqn 9-14, we get the characteristics listed in Table 2.

**Table 2. Characteristics of Vapor Generator**

| Parameter | Value | Unit |
|-----------|-------|------|
| \( h_i \) | 6756.6 | W/m\(^2\)K |
| \( h_{sol} \) | 2102.40 | W/m\(^2\)K |
| \( U \) | 1535.70 | W/m\(^2\)K |
| LMTD | 14.427 | K |
| Length of Tube(\( l \)) for a tube diameter of 12.5 mm | 1.72 | m |

2.2.2. Condenser

To estimate the overall heat transfer coefficient \( U \) as per Eqn 10, we need to solve for heat transfer coefficient inside tube( \( h_i \)) and outside the tube (\( h_{sol} \)). If Reynolds number (estimated from Eqn. 11) lies in \( 10^4 < \text{Re} < 5 \times 10^6 \) and \( \text{Pr} = 0.5 < \text{Pr} < 2000 \). Then the Nusselt number equation given by Petukhov-Popov Equation is used [9]:

\[ Nu = \frac{f}{8} \frac{\text{Re} \cdot \text{Pr}}{K_1 + K_2 \left( \frac{f}{8} \right)^{0.8} (\text{Pr}^3 - 1)} \]

(15)

\[ f = \left[ 1.82 \log(\text{Re}) - 1.64d \right]^2 \]

\[ K_1 = 1.04 \text{ and } K_2 = 11.7 + \frac{1.8}{\text{pr}^3} \]

(16)

Eqn. 16 will give \( h_i \).

**The calculation for heat transfer coefficient of water outside the tube \( h_o \)**

The heat transfer coefficient of water outside the tube \( h_o (h_{sol}) \) is estimated by Nusselt Theory for average heat transfer coefficient [9] and equation is given as

\[ \frac{h_o D}{k_f} = 0.728 \left[ \frac{\rho_i (\rho_i - \rho_s) g h_s d_0^3}{\mu_i (T_{w} - T_s) k_f} \right]^{1/4} \]

(17)

\( D \) and \( d_0 \) are shell diameter and outer diameter of tube respectively.

In the case of forced convection, the most conservative relation

\[ \frac{\rho_o d_0^{1/2}}{k_f} = 0.59 \text{Re} \]

(18)

3. When both velocity and gravity is considered
\[
\frac{N_u}{R_e^{1/2}} = 0.64(1 + (1 + 1.69F)^{1/2})^{1/2}
\]

(19)

\[
F = \frac{g d_o \mu_i h_{fg}}{u^* k_i \Delta T}
\]

(20)

Substitute the value of \(h_i\) and \(h_m = h_o\) in the Eqn.10 of \(U\) and solve for area \(A\).

\[A = \pi d_i l, \ l = \text{length of the tube}\]

(21)

| Parameter          | Value | Unit |
|--------------------|-------|------|
| \(h_i\)           | 2432  | W/m²K |
| \(h_o\)           | 905   | W/m²K |
| \(U\)             | 531.5 | W/m²K |
| LMTD              | 13.8  | K    |
| Length of Tube(L) for tube diameter of 12.5 mm | 3.4 | m |

Table 3. Characteristics of Condenser

2.2.3. Absorber. The absorption process of an aqueous lithium bromide refrigeration system usually occurs in the vertical tube as shown in fig. 3. The control variables of such absorption are pressure, temperature, the mass flow rate of solution, and the solution's concentration. The absorber's performance assessment parameters are mass absorption flux, the cooling of the outlet solution, and the falling film heat transfer coefficient.

![Figure 3. Schematic of Absorber][10]

The variables that affect absorber performance are pressure, solution inlet concentration, solution mass flow rate, and absorber temperature. The correlation for mass flow rate and length of the absorber is:

\[
L = a m b
\]

(22)

Where \(a = -132(\ln(100 - A_p) / 86.0), b = 1.33\) & \(m = \text{mass flow rate per width of plate (kg/m s)}\);

\[
A_p = \text{“absorption percentage”}
\]

\[
A_p = \frac{C_{in} - C_{out}}{C_{in} - C_{eq}} \times 100
\]

(23)

Where \(C_{eq} = \frac{X_{eq}}{100}\)

\(X_{eq}\) is estimated from the iterative method presented in ref [10]. \(L\)'s assumed value in Eqn. (24) calculates the mass flow rate since the external area of tube per unit length is known, so the required tube number can be estimated.

The next step to ensure whether the calculated surface area is enough for absorption or not. Eqn. 9 and 10 in coordination with the following relationships, are used to calculate the heat exchange area: The heat transfer coefficient \(h_i\) can be evaluated using Wilke’s relation.
\[ h_s = \frac{k}{\delta} (0.029(Re_s)^{0.53} Pr_s^{0.344}) \] where,
\[ \delta = \frac{(3\mu\Gamma)}{\rho^3 g} (m) \]

and the Reynolds number \((Re_s)\) of solution flow in the vertical tube is
\[ Re_s = \frac{4\Gamma}{\mu} \]

The heat transfer coefficient \(h_s\) for cooling water can be estimated from \(Nu = \frac{hD}{k}\).

The values of \(h_s\) and \(h_i\) will be a substitute in Eqn. 10 and \(U\) overall heat transfer will be available. And, the length per tube would be obtained. This length should be equal to the initial length assumed. The above calculation steps may be repeated until the length of the pipe matches.

2.2.4. Evaporator. The evaporator can be designed similarly to that of the absorber. The falling film of water around the evaporator tubes (through which water flows) extract heat to vaporize. The heat transfer coefficient is continuously increasing with distance. It is difficult to calculate all of the vaporisation characteristics theoretically because of the involvement of a large number of variables and the complexity of the multiphase flow patterns that occur as the quality of the vapour-liquid mixture increases during vaporization [10]. So, the mean heat transfer coefficient could be to be determined by experimental investigation.

3. Latent Heat Energy Storage

The other component of the proposed system is latent heat energy storage, in which, heat is stored during the daytime and could be used during the night or cloudy days. In this section, specifications of LHES are estimated, which would be sufficient to run the VARS in the absence of solar thermal energy. The phase change material is used to store heat in latent form, this section presents the initial guess of desired properties to match the heat requirement for VARS. The present calculation is done with known properties of paraffin wax with a melting temperature of 56°C-58°C and exercised for other PCMs also. The properties of Paraffin wax are listed in Table 4.

The design of the unit is the same as that of the standard heat exchanger. The following set of equations are used to calculate the mass of the PCM, size of the container, and heat transfer area. As the generator’s capacity is 1.5 kW, considering a design factor of 1.5, the design capacity is 2.25 kW. The energy storage should be able to run the refrigeration unit for 3 hours. Thus, the amount of energy to be store in PCM will be 24300 kJ.

The amount of PCM required to store \(Q_{stored}\) will be estimated by equation [11]:
\[ Q_s = m[C_p(t_m - t_f) + fVq] + C_p(t_f - t_m)] \]

\(Q_s\) = Storage capacity in kJ, \(m\)=Mass of the storing medium in kg, \(t_m\) = The melting point of the storing medium in °C, \(t_f\) = The initial temperature of the storing medium in °C, \(C_p\) = Latent specific heat capacity, \(C_s\) = Sensible specific heat capacity.

\(f\) = Melt fraction; \(Vq\) =Latent heat of fusion, in kJ/kg.

Thus, the required mass of the PCM is approx. 95 kg.

Volume of the PCM Container = (mass/density) = 0.120 m³.

Liquid density is considered to account for maximum volume.

For, cylindrical container with height = 1000 mm.
The inner diameter of the container = 400 mm

The effectiveness of an energy storage tank can be evaluated the same as that of the heat exchanger. For this case, shell and tube arrangement is considered and the LMTD approach is adopted.
Table 4 Properties of the PCM

| Stored Heat $Q_{stored}$ | 24300 kJ |
|--------------------------|----------|
| Melting Temp. $t_m$      | 56 °C    |
| Initial Temp., $t_i$    | 30 °C    |
| Final Temp. $t_f$       | 80 °C    |
| $C_{pl}$                | 2.88 kJ/Kg K |
| $C_{ps}$                | 2.93 kJ/Kg K |
| $f$                      | 1        |
| $\dot{V}q$              | 115 kJ/kg |
| Density liquid ($\rho$) | 780 kg/m$^3$ |

For inlet and outlet condition: $T_{hi} = 85$ °C, $T_{ho} = 70$ °C, $T_{ri} = 30$ °C, $T_{ro} = 80$ °C;
From Eqn. 12, $\Delta T_{m} = 56$ °C
The inner and outer heat transfer coefficient is calculated and substitute in Eqn. 10 to get $U$.

$$h_{water} = \frac{\frac{f}{2} \left( Re^{-1000} Pr \right)}{1 + 2.7 \left( \frac{f}{2} \right)^{\frac{1}{3}} (Pr^{\frac{1}{3}} - 1)} \frac{k}{D}$$  \hspace{1cm} (27)

$D$ is the outer diameter of the water tube = 12.5 mm
Other parameters in the equation can be estimated the same as in Eqn 16 and 17.
Thus, $h_{water} = 9052$ W/m$^2$K
And $h_{pcm}$ can be calculated by using the Nusselt equation in terms of Rayleigh number as given below[12]

$$Nu = 0.133 Ra^{0.326} \left( \frac{l}{\Delta r_m} \right)^{-0.0686}$$  \hspace{1cm} (28)

$\Delta r_m$ is the thickness of heat storage material $\frac{d_o - d_i}{2}$; $l$ is the heat exchanger length.

$$Ra = \frac{g \beta (T_{wall} - T_{pcm}) D_i^3}{\nu^2} \cdot Pr$$  \hspace{1cm} (29)

All the thermophysical properties are at the mean temperature.
$h_{pcm}$ can be calculated from the following equation:

$$Nu = \frac{h_{pcm} D_{eff}}{k_{pcm}}, \text{ where } D_{eff} \text{ is the effective diameter of the PCM bed and the } h_{pcm} \text{ to be 121 W/m}^2\text{K.}$$

Substituting the values of $h_{pcm}$ and $h_{water}$ in the equation xxiv to get overall heat transfer $U$
Thus, $U = 167.812$ W/m$^2$K
From Equation, the required area of heat transfer 0.8 m$^2$.
The design of latent heat energy storage is done based on PCM paraffin Wax. The same equations can be used for other suitable PCM.

4. Theoretical Result Validation
The results obtained from a theoretical calculation using the set of equations presented above needs validation. To validate the calculated results, we have compared the calculation with the values presented by Florides et al.[10] and presented in
Table 5. The result is varied less than 10% as shown in Table 5 and hence within the acceptable range.

| Component   | Florides et al. [10] | Presented Work | Difference (%) |
|-------------|----------------------|----------------|----------------|
| Evaporator [kW] | 1                   | 1              | -              |
| Absorber [kW]   | 1.28                | 1.412          | 10.3           |
| Generator [kW]  | 1.36                | 1.460          | 7.3            |
| Condenser [kW]  | -                   | 0.965          |                |

5. Result and Discussion

After the theoretical equations, assumptions and procedure used for calculation in the paper are validated. In this section, the effect of cooling capacity on the size of generator, absorber, and condenser is presented. The variation in the quantity of PCM required for the different cooling capacity has also been presented and discussed.

5.1 Effect of Generator Temperature on COP

The effect of generator temperature on COP is estimated with the following condition and assumptions and plotted in figure 4:
1. Condenser Temperature = 45°C
2. Evaporator Capacity = 1 kW at 5°C.
3. Solution Strength = 60%
4. Pressure in Evaporator and Absorber = 8.52 kPa

5.2 Effect of Absorber Temperature on COP

The effect of absorber temperature on COP is estimated with the following condition and assumptions:
1. Condenser Temperature = 45°C.
2. Evaporator Capacity = 1 kW at 5°C.
3. Solution Strength = 60%
4. Pressure in Evaporator and Absorber = 8.52 kPa

The variation of COP with absorber temperature is plotted in Figure 5.

5.3 Effect of Evaporator Temperature on COP

The effect of Evaporator temperature on COP is estimated for evaporator capacity of 1 kW, and other parameters are not changed.

| S.No. | Evaporator Temperature | COP   |
|-------|------------------------|-------|
| 1     | 5°C                    | 0.875 |
| 2     | 10°C                   | 0.890 |
| 3     | 15°C                   | 0.91  |
5.4 Effect of Component Capacity on COP

Figure 6 shows the generator, absorber, and condenser size for change in the machine's cooling capacity. The graph shows the different slope for generator, absorber and condenser for fixed change in cooling capacity. The actual sizes are presented in Figure.

5.5 Effect of Cooling Capacity on amount of PCM

Figure 7 shows the amount of required PCM for change in the cooling capacity of the VARS machine. The graphs show a linear variation in PCM quantity to cooling capacity. The higher the latent heat capacity of the PCM lower will be the mass required. The presented results are for the latent heat of fusion of 115 kJ/kg.

6. Conclusion

The presented theoretical design of solar-powered VARS coupled with PCM based latent heat energy storage is helpful to do initial sizing with some realistic assumptions and approximation. The obtained values can be used for the detailed design and sizing of the components.

The analytical design of each component is important and useful for detailed design. Component size varies linearly with cooling capacity and the amount of PCM required varies linearly with cooling capacity with different slopes.

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