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Exergy And Energy Analysis Of A Solar Powered NH_3–H_2O Vapor-Absorption Refrigeration System Using Parabolic Collector

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Abstract

Solar powered Vapour absorption refrigeration system proves to be an environment friendly alternative when compared with traditional cooling system using CFC. They can also operate on low grade heat source. Ammonia being eco- friendly and high efficiency refrigerant is widely used various application. In this paper the exergy and energy analysis, of a 10-KW single-effect NH_3–H_2O vapor-absorption refrigeration system (VARS) powered by solar energy through a parabolic collector is presented. The system features a vertical generator (15-cm diameter, 82-cm height), water-cooled absorber, air-cooled condenser, and chilled-water evaporator operating at flow rates of 27.5–40 L/min.

A comprehensive steady-state thermodynamic model was developed in Excel to perform mass, energy, and exergy analyses of all system components. The model incorporates heat rejection in the absorber and condenser, thermal load in the generator, evaporator chilled-water dynamics, and parabolic collector performance including useful heat gain and optimal-thermal efficiency prediction. Exergy analysis reveals that collector coupled with generator and absorber account for highest amount of exergy destruction. It provides a useful insights for solar assisted vapour absorption refrigeration system.

Keywords- Solar absorption refrigeration, NH_3–H_2O working pair, Exergy analysis, Energy analysis, Parabolic solar collector, Solar cooling system, Chilled water temperature

NOMENCLATURE

Symbol	Description	Unit
A_{ap}	Aperture area of parabolic collector	m²
A_{gen}	External heat transfer area of generator	m²
C	Concentration ratio (A_{ap}/A_{gen})	–
COP	Coefficient of performance (Q_e/Q_g)	–
e	Specific exergy	$kJ \cdot kg^{-1}$
e_{ph}	Physical exergy	$kJ \cdot kg^{-1}$
e_{ch}	Chemical exergy	$kJ \cdot kg^{-1}$
e_{tot}	Total specific exergy	$kJ \cdot kg^{-1}$
$\dot{E}$	Exergy rate	kW
$\dot{E}_D$	Exergy destruction rate	kW
G	Direct Normal Irradiance (solar)	$W \cdot m^{-2}$
h	Specific enthalpy	$kJ \cdot kg^{-1}$
h_0	Specific enthalpy at dead state	$kJ \cdot kg^{-1}$
$\dot{m}$	Mass flow rate	$kg \cdot s^{-1}$
p	Pressure	bar
$\dot{Q}$	Heat transfer rate	kW
Q_e	Cooling capacity of evaporator	kW
Q_g	Heat supplied to generator	kW
R	Universal gas constant	$kJ \cdot kmol^{-1} \cdot K^{-1}$
s	Specific entropy	$kJ \cdot kg^{-1} \cdot K^{-1}$
s_0	Specific entropy at dead state	$kJ \cdot kg^{-1} \cdot K^{-1}$
T	Temperature	K or °C
T_0	Ambient (dead-state) temperature	K
T_m	Mean temperature of heat source (generator boundary)	K
$\dot{W}$	Work rate (pump)	kW
x	Ammonia mass fraction in solution	–

Greek symbols

Symbol	Description	Unit
η_{col}	Collector efficiency	–
η_{ex}	Exergy efficiency	–
ψ	Specific flow exergy	$kJ\cdot kg^{-1}$

Subscripts

0	Dead-state (ambient) condition
abs	Absorber
amb	Ambient
col	Collector
cond	Condenser
cw	Chilled water
e	Evaporator
eff	Efficiency
ex	Exergy
gen	Generator
in	Inlet
out	Outlet
ph	Physical
pump	Solution pump
sol	Solution (NH_3–H_2O)
NH3	Ammonia
shx	Solution heat exchanger

1. INTRODUCTION:

There is a sharp increase in global energy usage and is projected to increase up to 50% by 2040. [1] This increased demand aroused the need for energy substitution especially from conventional sector to sustainable, clean and efficient energy as carbon emissions increased at an average rate of 0.6% per annum over the past five years (2019-2024). [2] This energy demand from refrigeration and air conditioning has been observed exponential growth in India which has led to considerable increase in electricity consumption.[3] Traditional vapour compression refrigeration systems (VCRS) still dominate most of the refrigeration and cooling sectors use HFC and CFC's which has adverse impact on environment and are of high ODP. The surge in electricity demand during heat waves in developing economies like India contribute to grid instability. Consequently, there has been increasing research interest in solar cooling technologies that can utilize low carbon. [4]

Absorption refrigeration technologies address these energy crisis and environmental problems in a feasible manner as they can convert low grade thermal energy into refrigeration effect with least requirement of electrical input [5].Compared with many working-fluid pairs ammonia- water (NH_3–H_2O) system have high efficiency, zero ozone depletion potential, zero global warming potential (GWP easy to detect during leakage , high latent heat of ammonia, and superior performance at sub-zero refrigeration temperatures[6]

Vapour absorption systems with single-effect NH_3–H_2O usually operate with coefficient of performance in the range of 0.45–0.65 when supplied with heat at temperatures between 80 and 150 °C, making them well suited for coupling with solar thermal collectors, industrial waste heat, and combustion-based heat sources [7]

The diverse cooling requirements have been addressed by several absorption configurations such as cascade VARS, hybrid absorption–compression systems, solar-driven absorption systems, and externally fired absorption units. Hybrid refrigeration system is energy efficient and cost effective but require additional electrical input.[8]. Fully powered biomass has less capital cost as compared to solar biomass powered systems but are not carbon neutral.[9] Cascade absorption systems can achieve ultra-low evaporator temperatures but suffer from high system complexity [10]. In contrast, solar absorption system a sustainable cooling is achieved with higher sunshine, particularly in regions with high direct normal irradiance (DNI) [11][12]

The systems combined with parabolic collectors as compared to other solar collectors has although high initial cost but has high economic and thermodynamic efficiency in operating conditions [13] These studies also indicate that system performance is highly sensitive to operating parameters such as evaporator temperature, condenser pressure, absorber temperature and are focused on energy flow and exergy destruction analysis.[14]

Energy analysis alone is insufficient for identifying the fundamental thermodynamic limitations of absorption systems, as it does not account for irreversibility within individual components. Exergy analysis determines the irreversibilities of the absorption system identifying major sources of entropy generation within the system [15]. Previous studies applying exergy analysis to NH3 -H2O absorption cycles have shown that the generator and absorber contribute the largest share of total exergy destruction, systems operating under high ambient temperatures.[16]

Despite significant progress, several research gaps remain. Many existing investigations are limited to nominal operating conditions and do not systematically evaluate the combined effects of chilled-water temperature variation, generator heat input fluctuation, and evaporator solution level under realistic climatic conditions. Furthermore, limited studies integrate a detailed solar collector performance model with absorption system analysis for Indian subtropical climates, where high ambient temperatures and seasonal DNI variability strongly influence absorber performance and overall exergy efficiency. Comparative assessments of solar-driven VARS against cascade, hybrid, and externally fired absorption systems under identical cooling capacity constraints are also scarce.

In this context, the present study develops a comprehensive thermodynamic and exergy model of a 10 KW solar-assisted NH_3–H_2O vapor absorption refrigeration system driven by a parabolic trough collector under climatic conditions representative of Durg, Chhattisgarh, India. The system comprises an air-cooled absorber and a water-cooled condenser, reflecting practical constraints for decentralized cooling applications. A detailed parametric investigation is conducted by varying chilled-water inlet temperature, generator heat input, and evaporator solution level. Energy and exergy analyses are employed to evaluate system performance, component-wise irreversibility, and solar collector–generator matching. The results provide design and operational insights and enable a comparative evaluation of solar-based absorption refrigeration relative to cascade, hybrid, and externally fired VARS configurations for sustainable cooling deployment.

1.1Comparison of VCRS and VARS

Typically Vapor absorption refrigeration systems use absorber generator pair instead of compressor as used in conventional vapour compression refrigeration systems. Although VCRS can be setup with low initial cost but have high complexity and vibration due to mechanical parts. Absorption systems can operate at reduced evaporator pressure and temperature by increasing steam pressure to the generator with very little decrease in capacity whereas capacity of compression systems rapidly drops at lowered evaporator pressure [17] VCRS is energy efficient and has higher COP but are exergy inefficient due to large amount amount of exergy destruction [18]. Detailed 1st- and 2nd-law analysis of VCRS; shows good energy (COP) performance, but significant exergy destruction in compressor, condenser, expansion valve, etc., highlighting limited exergy efficiency when using high-grade electrical work[19] .Review paper that explicitly states VARS is well suited to waste heat, solar, geothermal, biomass, and highlights that exergy analysis is the best tool to evaluate efficient use of low-grade heat; also emphasises environmental advantages (natural working pairs, low GWP/ODP) over many VCR systems.[20]. Higher electric rates and ability of the absorption system to use low grade energy makes absorption system most sustainable option in recent scenario.

1.2 Comparison of vars system based of heat input to the generator

System	Heat Source	COP Range	Exergy Efficiency	Complexity	Best Suited For
Cascade VARS	Waste heat, high-temp heat	0.2–0.35	Low	Very High	Ultra-low temperatures (−40°C and below)
Hybrid VARS	Heat + electricity	0.8–1.4	Moderate	High	High-performance HVAC, industrial cooling
Solar VARS	Solar thermal	0.45–0.65	18–35%	Moderate	Off-grid cooling, agricultural cold chain
Externally Fired VARS	Gas/biomass/diesel combustion	0.45–0.55	15–25%	Low	Remote industrial cooling, reliable 24/7 operation

1.3 Scope of research and formulation of research statement –

1. **To develop a validated thermodynamic model** of a 10 KW single-effect NH_3–H_2O vapour absorption refrigeration system driven by solar thermal energy, using experimentally established mass, energy, and exergy balance equations.
2. **To integrate a parabolic trough solar collector model** with the absorption system, enabling realistic estimation of generator heat input under representative DNI and ambient temperature conditions.
3. **To investigate the influence of chilled-water inlet temperature** on system energy and exergy performance, over a wide range relevant to comfort cooling and low-temperature refrigeration applications.
4. **To analyse the effect of generator heat input variation** on cooling capacity, coefficient of performance (COP), and exergetic efficiency, identifying optimal operating ranges.
5. **To quantify component-wise exergy destruction** in the generator, absorber, condenser, evaporator, and auxiliary components, and to identify the dominant sources of thermodynamic irreversibility.

2.1 System details and overview – The system investigates a Solar driven vapor absorption refrigeration system using NH3-H2O as working pair designed for 10KW evaporator load. The thermodynamic behaviour of experimental VARS system which is externally fired has been replaced here by solar driven model and is validated through simulation framework in excel (visual basics)

Experimental investigation of a vapour absorption refrigeration system [21] approaches the gas fired VARS system uses ammonia as refrigerant designed for cooling capacity of 10KW. It uses a vertical generator of closed steel cylinder 15cm in diameter and 82 cm in height. Ethylene glycol is added to work at low temperature ranges. It uses an air-cooled absorber and a water-cooled condenser. This study provides COP ranges and generator evaporator interactions with that of heat input for comparing with the present mathematical model.

The experimental setup of external fired heat input is replaced here by solar heat input through parabolic trough collector

This system uses:

- **Parabolic trough collector** delivering 10–25 kW heat input
- **Vertical cylindrical generator** (15 cm Ø × 82 cm height)
- **Water-cooled condenser**, **air-cooled absorber**
- **Chilled water inlet temperature** varied from –2 to 22 °C
- **Maximum Chilled water level in Evaporator drum – 29cm**
- **Ambient conditions** typical of Durg, Chhattisgarh (In winter for an average of 6.00am to 6.00pm) (33–38°C, 45–60% RH)

2.2 Thermodynamic model Formulation

2.2.1 Assumptions

1. The system operates under steady state conditions.

2. Ammonia leaving the generator and entering the condenser is assumed to be pure refrigerant, ensured by an ideal rectification process.

3. Changes in kinetic and potential energy of the working fluids are assumed negligible compared to changes in thermal and flow energies.

4. The work input to the solution pump in the absorption system is assumed negligible compared to the compressor power and generator heat input.

5. The fluctuations due solar irradiance is neglected. No thermal storage tank has been taken into account as solar input is treated as constant heat source from design point. The system is assumed to be in steady state condition.

6. Ambient temperature and pressure conditions are considered to be constant and all variations are neglected.

7. All components except the evaporator, condenser, absorber, generator, and solar collector are assumed to be perfectly insulated, with no heat loss to the surroundings.

8. Thermal storage effects due to metal components and piping are ignored as metal piping are used in experimental conditions.

2.2.2 Thermodynamic Modelling

The thermodynamic analysis of a vapour absorption system is based on the following three equations:

(i) Mass balance

$$\sum \dot{m}_{\text{in}} x_{\text{in}} = \sum \dot{m}_{\text{out}} x_{\text{out}}$$

(ii) Material balance (or partial mass balance)

$$\dot{m}_s x_s = \dot{m}_w x_w + \dot{m}_r$$

(iii) Energy balance: General steady-state exergy balance for a control volume:

$$\sum \dot{E}_{Q,j} + \sum \dot{m}_{\text{in}} \psi_{\text{in}} + \dot{E}_W = \sum \dot{m}_{\text{out}} \psi_{\text{out}} + \dot{E}_D$$

Solution mass balances (generator & absorber)

At the absorber (strong solution leaving, weak solution entering):

- Mass balance:

$$\dot{m}_s = \dot{m}_w + \dot{m}_r$$

- NH_3 mass balance:

$$\dot{m}_s x_s = \dot{m}_w x_w + \dot{m}_r$$

At the generator (weak solution leaving, strong solution entering):

Component energy and exergy balances

Using the equations from

Evaporator (EVA)

Energy balance (rate form)

- On refrigerant side:

$$\dot{Q}_{\text{evap}} = \dot{m}_r (h_1 - h_4)$$

- On chilled-water side:

$$\dot{Q}_{\text{evap}} = \dot{m}_{cw} c_{p,cw} (T_{cw,in} - T_{cw,out})$$

Exergy balance

Heat transfer occurs at an approximate mean evaporator temperature T_{evap}:

- Exergy input with heat:

$$\dot{E}_{Q,\text{evap}} = \dot{Q}_{\text{evap}} (1 - \frac{T_0}{T_{\text{evap}}})$$

- Stream exergy balance:

$$\dot{E}_{Q,\text{evap}} + \dot{m}_r \psi_4 = \dot{m}_r \psi_1 + \dot{m}_{cw} \psi_{cw,out} - \dot{m}_{cw} \psi_{cw,in} + \dot{E}_{D,\text{evap}}$$

Often for **system-level COP** $_{\text{ex}}$, we treat the **useful exergy output** as:

$$\dot{E}_{\text{useful,evap}} \approx \dot{m}_r (\psi_1 - \psi_4)$$

or equivalently the exergy change of chilled water.

Condenser (CON)

Energy balance

$$\dot{Q}_{\text{cond}} = \dot{m}_r(h_3 - h_4)$$
$$\dot{Q}_{\text{cond}} = \dot{m}_c c_{p,c}(T_{c,out} - T_{c,in})$$

Exergy balance

At mean condenser boundary temperature T_{cond}:

$$\dot{E}_{Q,\text{cond}} = \dot{Q}_{\text{cond}}(1 - \frac{T_0}{T_{\text{cond}}})$$
$$\dot{m}_r\psi_3 + \dot{m}_c\psi_{c,in} + \dot{E}_{Q,\text{cond}} = \dot{m}_r\psi_4 + \dot{m}_c\psi_{c,out} + \dot{E}_{D,\text{cond}}$$

Because condensation is rejection to environment, $\dot{E}_{Q,\text{cond}}$is usually **negative** relative to the system (exergy out).

Absorber (ABS)

Energy balance

$$\dot{m}_r h_1 + \dot{m}_w h_{w,in} = \dot{m}_s h_{s,out} + \dot{Q}_{\text{abs}}$$

with mass and NH_3 balances (given above) also enforced.

Exergy balance

Heat rejection exergy term:

$$\dot{E}_{Q,\text{abs}} = \dot{Q}_{\text{abs}}(1 - \frac{T_0}{T_{\text{abs}}})$$

Exergy balance:

$$\dot{m}_r\psi_1 + \dot{m}_w\psi_{w,in} = \dot{m}_s\psi_{s,out} + \dot{E}_{Q,\text{abs}} + \dot{E}_{D,\text{abs}}$$

Because $\dot{Q}_{\text{abs}}$is usually close to ambient temperature, $\dot{E}_{Q,\text{abs}}$is small but $\dot{E}_{D,\text{abs}}$tends to be large (big irreversibility).

Generator (GEN)

Energy balance

$$\dot{Q}_{\text{gen}} + \dot{m}_s h_{s,in} = \dot{m}_w h_{w,out} + \dot{m}_r h_3$$

Exergy balance

$$\dot{E}_{Q,\text{gen}} = \dot{Q}_{\text{gen}}(1 - \frac{T_0}{T_{\text{gen}}})$$
$$\dot{E}_{Q,\text{gen}} + \dot{m}_s\psi_{s,in} = \dot{m}_w\psi_{w,out} + \dot{m}_r\psi_3 + \dot{E}_{D,\text{gen}}$$

This $\dot{E}_{D,\text{gen}}$ is usually one of the largest terms in the whole system.

Solution Heat Exchanger (SHX)

Energy balances

Neglecting heat losses to surroundings:

$$\dot{m}_w(h_{w,h,in} - h_{w,h,out}) = \dot{m}_s(h_{s,c,out} - h_{s,c,in})$$

Exergy balance

No external heat/work (ideal insulated SHX), so:

$$\dot{m}_w\psi_{w,h,in} + \dot{m}_s\psi_{s,c,in} = \dot{m}_w\psi_{w,h,out} + \dot{m}_s\psi_{s,c,out} + \dot{E}_{D,\text{SHX}}$$

Expansion Valves (Throttling devices)

- Solution expansion (if present): from generator pressure to absorber pressure.

Assumed **isenthalpic** and adiabatic.

Energy relation

$$h_{\text{out}} = h_{\text{in}}$$

Exergy balance (per valve)

No heat, no work:

$$\dot{m}\psi_{\text{in}} = \dot{m}\psi_{\text{out}} + \dot{E}_{D,\text{valve}}$$

So:

$$\dot{E}_{D,\text{valve}} = \dot{m}(\psi_{\text{in}} - \psi_{\text{out}}) \geq 0$$

Solution Pump

The pump raises solution pressure from absorber to generator pressure.

Energy

For the solution stream:

$$\dot{W}_{\text{pump}} = \dot{m}_s(h_{\text{out}} - h_{\text{in}})$$

Exergy

$$\dot{E}_{W,\text{pump}} = \dot{W}_{\text{pump}}$$
$$\dot{m}_s\psi_{\text{in}} + \dot{E}_{W,\text{pump}} = \dot{m}_s\psi_{\text{out}} + \dot{E}_{D,\text{pump}}$$

Often approximated by:

$$\dot{W}_{\text{pump}} \approx \frac{\dot{m}_s(p_{\text{gen}} - p_{\text{abs}})}{\rho_s\,\eta_{\text{pump}}}$$

Solar Collector (Parabolic Trough) – Thermal Side

The collector provides heat $\dot{Q}_{\mathrm{col}}$to the generator through a heat-transfer fluid loop.

Using Hotter- Whiller model

Collector energy

$$\dot{Q}_{\mathrm{col}} = \eta A_{\mathrm{ap}} G$$

with

$$\eta = \eta_0 - a_1 \frac{T_m - T_a}{G}$$

Collector–generator coupling

You enforce:

$$\dot{Q}_{\mathrm{gen}} \leq \dot{Q}_{\mathrm{col}}$$

and usually for “maximum load”:

$$\dot{Q}_{\mathrm{gen}} \approx \dot{Q}_{\mathrm{col}}$$

Exergy of collector heat

At mean collector fluid temperature $T_m \approx T_{\mathrm{gen}}$:

$$\dot{E}_{Q,\mathrm{col}} = \dot{Q}_{\mathrm{col}}(1 - \frac{T_0}{T_m})$$

If you treat the collector + generator as a combined control volume, the **external exergy input** is $\dot{E}_{Q,\mathrm{col}}$.

Circuit-level balances

Overall energy COP

For a cooling capacity fixed at $\dot{Q}_{\mathrm{evap}}$:

$$COP_{\mathrm{en}} = \frac{\dot{Q}_{\mathrm{evap}}}{\dot{Q}_{\mathrm{gen}} + \dot{W}_{\mathrm{pump}}}$$

In a purely thermal VARS, $\dot{W}_{\mathrm{pump}}$is small, often neglected:

$$COP_{\mathrm{en}} \approx \frac{\dot{Q}_{\mathrm{evap}}}{\dot{Q}_{\mathrm{gen}}}$$

Overall exergy COP

Define useful exergy output as exergy associated with evaporator effect (or chilled-water cooling):

$$\dot{E}_{\mathrm{cool}} \approx \dot{m}_r(\psi_1 - \psi_4)$$

Exergy input from heat (dominant):

$$\dot{E}_{\text{in}} \approx \dot{E}_{Q,\text{gen}} + \dot{E}_{W,\text{pump}}$$

Then:

$$COP_{\text{ex}} = \frac{\dot{E}_{\text{cool}}}{\dot{E}_{\text{in}}}$$

You can also define a **solar exergy efficiency** for the collector–generator–evaporator chain:

$$\eta_{\text{ex,solar}} = \frac{\dot{E}_{\text{cool}}}{\dot{E}_{Q,\text{col}}}$$

Total exergy destruction

Sum over all components:

$$\dot{E}_{D,\text{total}} = \dot{E}_{D,\text{evap}} + \dot{E}_{D,\text{cond}} + \dot{E}_{D,\text{abs}} + \dot{E}_{D,\text{gen}} + \dot{E}_{D,\text{SHX}} + \dot{E}_{D,\text{valves}} + \dot{E}_{D,\text{pump}} + \dot{E}_{D,\text{collector (thermal)}}$$

You can report **fractional contribution** as:

$$\Phi_i = \frac{\dot{E}_{D,i}}{\dot{E}_{D,\text{total}}}$$

for each component $i =$ gen,abs,cond, ...

Table 1: Equations of all components with energy and exergy balance equations

Component	Energy Balance	Exergy Balance	Key exergy Destruction source
Evaporator (EVA)	$\dot{Q}_{evap} = \dot{m}_r(h_1 - h_4)$	$\dot{E}_{Q,evap} + \dot{m}_r\psi_4 = \dot{m}_r\psi_1 + \dot{E}_{D,evap}$	Finite ΔT between chilled water and refrigerant
Condenser (CON)	$\dot{Q}_{cond} = \dot{m}_r(h_3 - h_4)$	$\dot{m}_r\psi_3 + \dot{E}_{Q,cond} = \dot{m}_r\psi_4 + \dot{E}_{D,cond}$	Heat rejection at elevated temperature
Absorber (ABS)	$\dot{m}_r h_1 + \dot{m}_w h_{w,in} = \dot{m}_s h_{s,out} + \dot{Q}_{abs}$	$\dot{m}_r\psi_1 + \dot{m}_w\psi_{w,in} = \dot{m}_s\psi_{s,out} + \dot{E}_{Q,abs} + \dot{E}_{D,abs}$	Mixing + heat rejection near ambient
Generator (GEN)	$\dot{Q}_{gen} + \dot{m}_s h_{s,in} = \dot{m}_w h_{w,out} + \dot{m}_r h_3$	$\dot{E}_{Q,gen} + \dot{m}_s\psi_{s,in} = \dot{m}_w\psi_{w,out} + \dot{m}_r\psi_3 + \dot{E}_{D,gen}$	Large ΔT between solar source and solution
Refrigerant Expansion Valve	$h_{in} = h_{out}$	$\dot{m}_r\psi_{in} = \dot{m}_r\psi_{out} + \dot{E}_{D,valve}$	Throttling (pressure drop)
Solution Expansion Valve	$h_{in} = h_{out}$	$\dot{m}_s\psi_{in} = \dot{m}_s\psi_{out} + \dot{E}_{D,valve}$	Throttling losses
Solution Pump	$\dot{W}_{pump} = \dot{m}_s(h_{out} - h_{in})$	$\dot{m}_s\psi_{in} + \dot{W}_{pump} = \dot{m}_s\psi_{out} + \dot{E}_{D,pump}$	Mechanical & hydraulic losses

Solar Collector	$\dot{Q}_{col} = \eta AG$	$\dot{E}_{Q,col} = \dot{Q}_{col}(1 - \frac{T_0}{T_m})$	Optical + thermal losses
Whole System	$COP_{en} = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}}$	$COP_{ex} = \frac{\dot{E}_{cool}}{\dot{E}_{Q,gen}}$	Sum of all component destructions

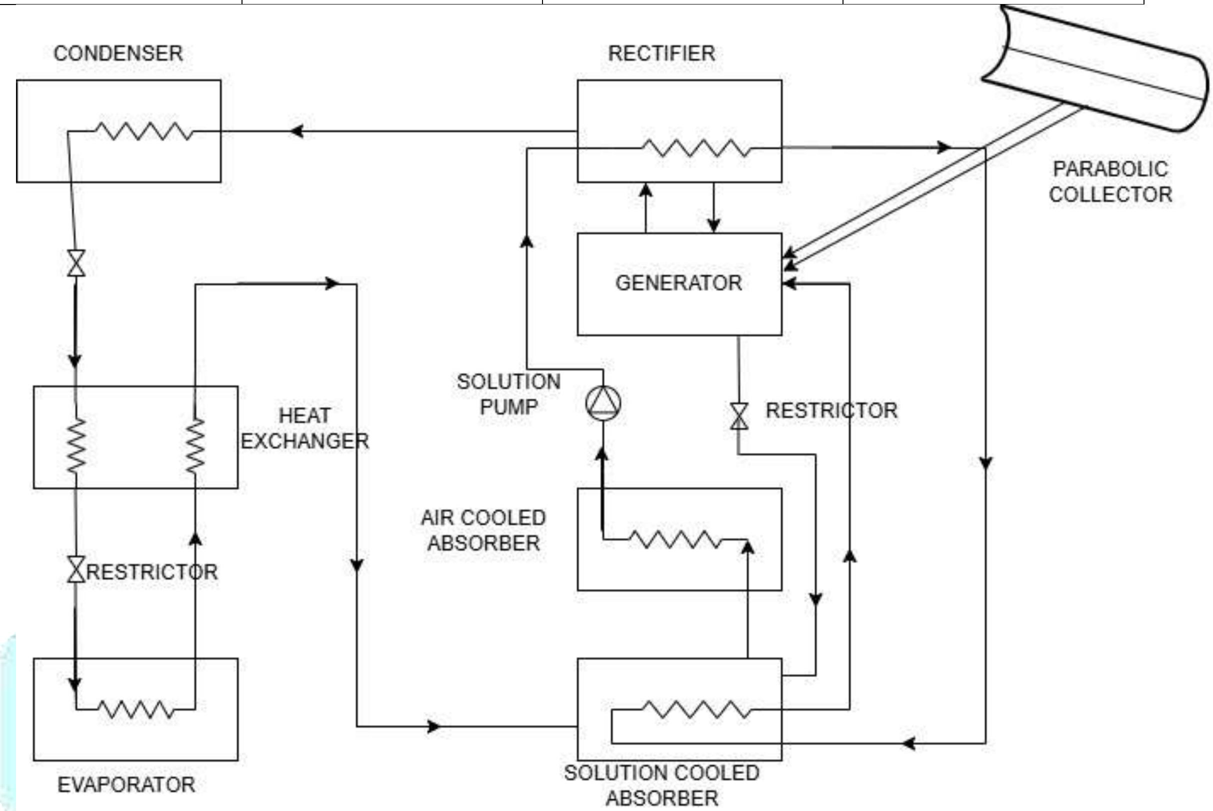

Fig.1 Schematic diagram of analyzed vapour absorption refrigeration system

3.Results and Discussion

3.1 Variation of COP for heating with chilled water Temperature

Effect of Chilled Water Temperature

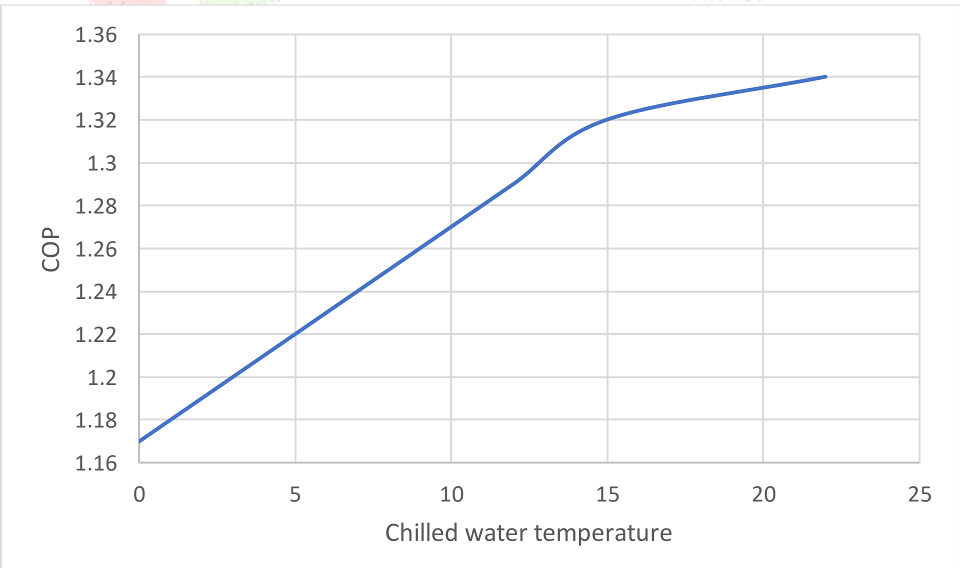

Fig.1 Coefficient of performance against Chilled Water inlet temperature

The coefficient of performance of the ammonia–water vapour absorption refrigeration system increases with an increase in chilled water inlet temperature. A higher chilled water temperature results in an increased evaporator saturation temperature and pressure, which reduces the solution circulation ratio and the required generator heat input for a fixed cooling capacity. Conversely, at lower chilled water temperatures, the evaporator operates at a lower pressure, increasing the circulation ratio and generator heat demand, thereby reducing the COP↓

3.2 Variation of COP with Generator heat input

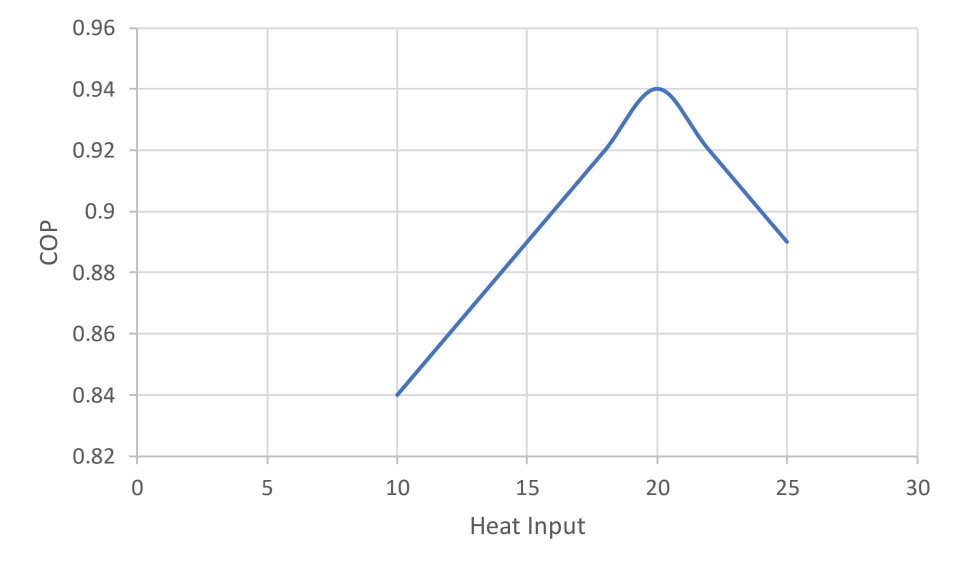

Fig. 2 Coefficient of performance against Heat Input

As heat input to the generator increases COP of the system increases but beyond an optimal range is reached the COP starts decreasing due to thermodynamic irreversibility. This is due to the increased circulation ratio decreasing the overall COP of the system. This aligns with typical NH_3–H_2O generator characteristics.

3.3 Solar Collector Performance and System Coupling

The parabolic trough collector provided **thermal input between 10 kW and 25 kW**, depending on DNI and inlet fluid temperature.

3.3.1 Collector efficiency

As solar irradiance on the collector increases the efficiency of the parabolic collector increases as useful heat gain to the collector is increased. This is in line with higher DNI areas getting higher output which makes it highly useful for hot climatic conditions**.**

As the mean boundary temperature increases the efficiency of collector decreases as it accounts for higher heat loss to the environment

This matches trends reported in Indian solar cooling experimental studies where collector performance degrades sharply with temperature rise due to convective and radiative losses. Simulation results indicate this trend:

Table 2 :Mean fluid temperature to the collector and efficiency

T_m in °C	G in W/m^2	Eta Model
60	400	0.5012
60	600	0.5742
60	800	0.6106
80	400	0.3262
80	600	0.4575
80	800	0.5231

- The system produces its maximum COP when collector output maintains generator temperature at **110–120 °C**.
- Below ~100 °C generator temperature, insufficient ammonia vaporization limits cooling capacity.
- Above ~130 °C, solar losses increase more than the useful generator heat.

Therefore, **collector–generator matching is essential**, which agrees with experimental testbeds.

3.3.2 Exergy Destruction Distribution

Exergy Perspective

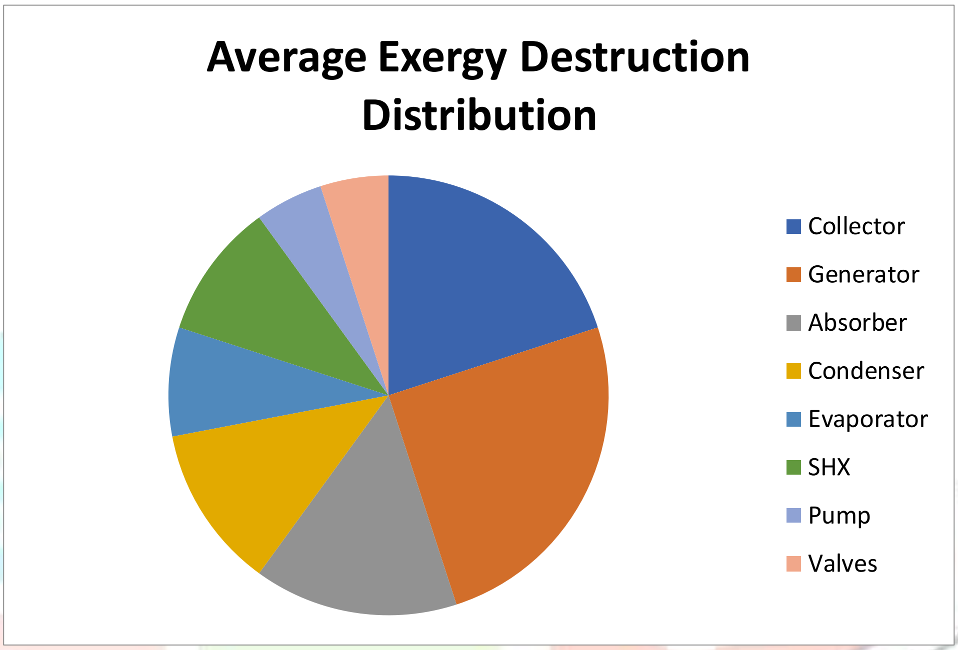

Table3: Exergy destruction component wise

Component	Average Exergy Destruction (KW)
Collector	0.709048672
Generator	0.88631084
Absorber	0.531786504
Condenser	0.425429203
Evaporator	0.283619469
SHX	0.354524336
Pump	0.177262168
Valves	0.177262168

Highest exergy destruction account for generator coupled with collector system and second largest occurs in absorber than condenser. This is due to high irreversibility associated with solar radiation absorption low temperature heat transfer and solution separation in the generator. Also the absorber experiences large entropy generation due to air cooling under high ambient temperatures typical of Chhattisgarh. It is seen that the variations in environmental temperature increases the irreversibility. The Condenser irreversibility is moderate since water cooling is more effective. The least energy destruction is observed at lowest generation temperature. The total exergy destruction (in KW) increased from 1.65 to 3.63 when heat input to the generator increased although energy COP also increased significantly. The exergetic coefficient of performance of an ammonia–water vapour absorption refrigeration system increases with increase in chilled water inlet temperature due to enhancement in evaporator exergy gain and reduction in system irreversibilities, while the generator exergy input remains nearly unchanged.

4. Recommendations and future perspective

1. Transient conditions and cloud cover variations can be included in this study for more accurate results.
2. Optical simulations, collector tilt for specific latitudes and real time solar irradiance tracking integrated with automatization can be done.
3. Development can be made on more zero ODP absorbent refrigerant pairs like nano fluids and absorbent refrigerant pairs combined with catalyst that produce enhanced COP
4. Thermo-economic analysis for payback period can be assessed.
5. Optimazation algorithms can be developed for increasing the COP.

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