Thermodynamic Analysis of Cogeneration Cycle by Utilizing Waste Heat of Blow Down Water

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Abstract. In the present work, the potential usage of blow down water from National Thermal Power Corporation Limited, Badarpur is investigated in the running of double effect H$_2$O-Li-Br vapor absorption refrigeration system. So mass of blow down water available is computed. The blow down water is at 342.45 °C which can be used for process heating. So that after process heating water enters high pressure generator at 155°C and transfer heat in such a way that at exit of high pressure generator water temperature is 140°C. The heat recovered is used as a source of energy in high pressure generator of vapor absorption refrigeration system. Results of parametric analysis show that coefficient of performance increases with increase in evaporator temperature and low pressure generator temperature; whereas, coefficient of performance decreases with increase in absorber temperature, condenser temperature and high pressure generator temperature. Parametric analysis also shows that low pressure generator temperature is most significant factor on which coefficient of performance depends. Exergy analysis of individual components of vapor absorption refrigeration system shows exergy destruction is maximum in absorber (26.17 kW) and least in high pressure generator (2.377 kW). The given model is validated with the research work carried out by Kaushik and Arora.

Keyword: High pressure generator, Exergy, Low pressure generator, VARS, Cogeneration

Nomenclature

| Symbol | Description                              | Latin Symbol |
|--------|------------------------------------------|--------------|
| C$_{pw}$ | Specific heat of water                    | abs          |
| C      | Specific heat of Li-Br mixture            | c            |
| COP    | Coefficient of Performance                | e            |
| h      | Specific enthalpy                         |              |
| HPG    | High pressure generator                   |              |
| I      | Irreversibility                           | $\varepsilon_1$ |
| LPG    | Low pressure generator                    | $\varepsilon_2$ |
| Li-Br  | Lithium Bromide                           | $\eta_p$     |
| m      | Mass                                     | $\rho$       |
| P      | Pressure                                  |              |
| Q      | Heat transfer                             | 1, 2, 3, State points |
| s      | Specific entropy                          |              |
| T      | temperature                               |              |
| X      | mass fraction of Li-Br                   |              |
| VARS   | Vapor absorption refrigeration system     |              |
1. Introduction

The economic growth of any country is reflected by its consumption of energy but it comes with a price that ever increasing demand of fossil fuel lead to depletion of fossil fuel reserves as well as enhances the release of harmful gases as a by-product of energy production. These gases are responsible for the increasing menace of global warming. Simultaneously the quest for human comfort led to ever increasing refrigeration load which further led to increase in power consumption as well as serious environmental issues i.e. global warming and ozone depletion. Thus, a need for a new system was felt which produces power as well as refrigerating effect simultaneously from the same primary energy source that is combined cooling, heating and power system.

Combined cooling, heating and power (CCHP) system mean simultaneously producing of power, heating as well as cooling from same primary energy source.

The cogeneration system offer many benefit with respect to conventional power plant

1. First and foremost benefit of CCHP system is less primary energy requirement to simultaneously produce power as well as Cooling. This result in less fuel requirement.
2. The second benefit of distributed CCHP systems is emission reduction .Since Cogeneration require less fuel it mean that combustion of “less fuel” results less emission of harmful gases like NOx, CO2.
3. Last, but of equal importance, CCHP systems increase the reliability of the energy supply network. CCHP system give necessary flexibility to replace large centralized conventional power system with smaller diverse power systems. It mean that power supply would remain less affected by snag in the system.

A typical CCHP system consists of five basic elements: the prime mover; electricity generator; heat recovery system; thermally activated equipment and the management and control system. According to current technologies, options in prime movers can be steam turbines, reciprocating internal combustion engines, combustion turbines, micro turbines, Stirling engines and fuel cells; the last three prime movers are relatively new technologies developed in last decade. Any of these options can be selected to meet diverse demands and limitations from site-to-site, especially local heat and electricity profiles, regional emissions and noise regulations and installation restrictions. Thermally activated equipment is another part of CCHP systems, to provide cooling or dehumidification. Commercialized thermally activated technologies include absorption chillers and desiccant dehumidifiers; moreover, novel adsorption chillers—currently almost entirely for commercial use—can be another choice for small CCHP systems.

Some existing systems also apply electric chillers, or engine-driven chillers integrated with prime movers, to fulfil cooling demands, which, combined with thermally activated technologies, are the cooling or dehumidification options of CCHP systems in some of the literature. Different prime movers, connecting with different cooling or dehumidification options, can result in various kinds of CCHP systems in theory, but only several modes of combination are widely adopted in commercial markets; other promising possibilities are being investigated to overcome technological or economic problems.

Yiping dai.et.al [1] worked on Rankine cycle with ejector refrigeration cycle and they included exergy analysis to determine exergy loss in various component of the system and conducted parametric study to optimize the system. A.Khaliq.et.al [2] investigated exhaust from industry driven Rankine cycle as well as single effect vapour absorption cycle. A thermodynamic analysis through energy and exergy was employed, and a comprehensive parametric study was performed to investigate the effects of exhaust gas inlet temperature, pinch-point, and gas composition on energy efficiency, power-to-cold ratio, and exergy efficiency of the cogeneration cycle and exergy destruction in each component. Ricardo Vasquez Padilla.et.al [3] considered Rankine cycle with vapour absorption cycle for their work and presented a thermodynamic study of power and cooling cogeneration. The performance of the cycle for a range of boiler pressures, ammonia concentrations and isentropic turbine efficiencies were studied to find out the sensitivities of net-work, amount of cooling and effective efficiencies. The roles of rectifier and superheater on the cycle performance were also investigated. Andre Alexio Manzela.et.al [4] presented an experimental study of an ammonia–water absorption refrigeration system using the exhaust of an internal Combustion engine as energy source. The engine was tested for 25%, 50%, 75% and wide-open throttle valve. The refrigerator reached a steady state temperature between 4 and 13°C about 3 h after system start up, depending on engine throttle valve opening. T. Guo et.al [5] worked on geothermal
source with vapor absorption cycle. The main objective of the paper was to find out suitable working fluid and optimized cycle parameters (power output per unit mass flow rate of heat source, \( \Delta W_{\text{net}} \)). H. Abed et al. [6] developed a model to perform a parametric analysis to evaluate the effects of important parameters on the performance of the cycle, which was a combination of Rankine and absorption refrigeration cycle. Propane-decane was used as an organic dual working fluid. Armando Fontalvo et al. [7] investigated the effect of pressure ratio, ammonia mass fraction at the absorber and turbine efficiency on the total exergy destruction. The effect of internal as well as external cooling for rectification was also investigated. A.T. Régo et al. [8] presented the experimental analysis of an absorption refrigeration system. The exhaust system of an automotive internal combustion engine was connected to the generator element of an absorption. Mohdparvez and Abdul Khaliq [9] researched on a biomass fuelled cogeneration cycle for the combined production of power and refrigeration. The system was an integration of combined gas-steam cycle and absorption refrigeration cycle. Xingyang Yang et al. [10] conducted a theoretical study on a combined power and ejector refrigeration cycle using zeotropic mixture iso-butane/pentane. The performances of different mixture compositions were compared. An exergy analysis was conducted for the cycle. Rahul Goyal et al. [11] discussed the performance and emission characteristics of a micro cogeneration system based on a single cylinder diesel engine. In this cogeneration system, in addition to the electricity generated from the gen-set, waste heat from hot exhaust gas of diesel engine was used to drive a combination of four units of Electrolux vapor absorption (VA) system for space cooling. K. Talukdar and T.K Gogoi [12] conducted thermodynamic analysis with different value HPG temperature. Single effect was also compared with double effect. They worked on Boiler flue gas driven double effect vapour absorption cycle. Pouya Ifaei et al. [13] worked on bleed steam driven vapor absorption cycle. Steam power plant-Natural draft wet cooling tower (SPP-NDWCT) was compared with Vapor compression refrigeration (VCR)-SPP-NDWCT and Absorption heat pump (ABHP)-SPP-NDWCT with respect to energy analysis as well as water consumption. Omendra Kumar Singh [14] worked on combined cycle with ammonia-water absorption cycle. In this work, energy as well as exergy analysis of the system and effect of climatic condition was also considered. L. Kairouani and E. Nehdi [15] calculated COP for different pure refrigerant while working with geothermal with vapor absorption system. Jouan Rashidi et al. [16] worked on Kalina cycle with single effect vapour absorption cycle. In this investigation, Kalina cycle was taken as base system and it was compared with Kalina Power Cooling Cycle (KPCC) and Kalina Absorption Cooling Cycle (KLACC) with respect to thermodynamic and economic aspect as well as parametric study was also performed.

2. Description of the system

Figure 1 show schematic diagram of the proposed Cogeneration System. In this weak Li-Br solution coming out of absorber state 4 is pressurised upto HPG pressure by the pump. This weak solution at state 5 then passes through solution heat exchanger 1. Here it recovers heat from strong Li-Br and heated upto T6. This mixture then passes through solution heat exchanger 2 where temperature is further raised upto 7. Then it passes through HPG where it is externally heated from blow down water of steam drum. This generate refrigerant (water vapor) \( m_{11} \) as well as strong Li-Br mixture. The mass \( m_{11} \) at state 11 then passes through LPG. LPG act like an internal heat exchanger and recover heat from mass \( m_{11} \) to generate refrigerant \( m_{14} \) as well as strong Li-Br mixture. The strong Li-Br from HPG i.e. \( m_8 \) passes solution heat exchanger 2 for heat exchange. The strong Li-Br from LPG i.e. \( m_{15} \) passes solution heat exchanger 1 for heat exchange. The refrigerant \( m_{14} \) and \( m_{11} \) then enter condenser where it is externally cooled by water. This condenses refrigerant \( (m_{14}+m_{11}) \) and this combined refrigerant of mass \( m_1 \) is throttled upto \( P_{\text{evap}} \). The refrigerant then enter evaporator and cool external water from temperature \( T_{22} \) to \( T_{23} \). The refrigerant recover this heat and get converted into saturated water vapour. This saturated water passes through absorber where it mixes with strong Li-Br mixture returning from solution heat exchanger 1 to from weak mixture \( m_4 \). This cycle then repeats.
Figure 1. The schematic illustration of double effect absorption refrigeration system [17]

2.1. Procedure for modelling a double effect refrigeration system
1. Give the input value of cooling load, condenser temperature, evaporator temperature, LPG temperature, HPG temperature, blow down water inlet temperature, blow down water outlet temperature, effectiveness of solution heat Exchanger 1, and effectiveness of solution heat Exchanger 2.
2. Give input value of inlet and exit temperature of external fluid.
3. Set absorber temperature equal to evaporator temperature.
4. Calculate saturation pressure corresponding to evaporator temperature, condenser temperature and absorber temperature.
5. Set LPG pressure equal to condenser pressure.
6. Determine mass fraction of Li-Br in the mixture of Li-Br and H2O at LPG generator, absorber.
7. Calculate specific enthalpy and specific entropy at relevant points.
8. Assume mass fraction of Li-Br at exit of HPG generator.
9. Write down conservation of mass equation, conservation of concentration equation and energy balance equation to calculate mass flow rate and heat transfer for condenser, absorber, HPG and LPG.
10. Reassume concentration of HPG generator so that net heat transfer of the LPG is within 10^-2 kW.
11. Calculate COP of the system and irreversibility of each component of the system.
12. Calculate mass of blow down water available of NTPC, Badarpur.
13. Calculate mass of blow down water required by using energy balance of HPG generator.
14. If mass of blow down water required is more than mass of blow down water available, decrease the cooling load to lower value so that mass of blow water required is less than or equal to mass of blow down water available.

Table 1. Fixed data used in simulation
| Sr. No. | Quantity | Description                        | Value(s)   |
|--------|----------|------------------------------------|------------|
| 1      | T<sub>1</sub> | Condenser temperature              | 40°C       |
| 2      | T<sub>3</sub> | Evaporator temperature             | 10°C       |
| 3      | T<sub>4</sub> | Absorber temperature               | 40°C       |
| 4      | T<sub>5</sub> | High pressure generator temperature | 148.9°C    |
| 5      | T<sub>6</sub> | Low pressure generator temperature  | 95°C       |
| 6      | T<sub>7</sub> | Blow down inlet water temperature  | 155°C      |
| 7      | T<sub>8</sub> | Blow down outlet water temperature | 140°C      |
| 8      | T<sub>9</sub> | Cooling water inlet temperature to condenser | 25°C |
| 9      | T<sub>10</sub> | Cooling water outlet temperature from condenser | 30°C |
| 10     | T<sub>11</sub> | Water inlet temperature to evaporator | 15°C |
| 11     | T<sub>12</sub> | Water outlet temperature from evaporator | 20°C |
| 12     | T<sub>13</sub> | Cooling water inlet temperature to absorber | 25°C |
| 13     | T<sub>14</sub> | Cooling water outlet temperature from absorber | 30°C |
| 14     | Q<sub>e</sub> | Cooling load                        | 300 KW     |
| 15     | T<sub>15</sub> | Effectiveness of solution heat exchanger 1 | 0.7 |
| 16     | T<sub>16</sub> | Effectiveness of solution heat exchanger 2 | 0.7 |
| 17     | T<sub>17</sub> | Isentropic efficiency               | 0.95       |

2.2. Assumptions
1. The whole system operate under steady state condition.
2. The composition of Li-Br does not change with time.
3. There is no pressure loss in heat exchangers and connecting piping.
4. The specific heat of water is assumed to be constant.
5. There is no change of concentration of Li-Br mixture while transferring heat in solution heat exchanger.
6. The refrigeration load is constant.
7. Only physical exergy loss is considered.
8. There is no heat transfer between connecting pipes and atmosphere.
9. The water vapour is saturated at exit of evaporator.
10. The water is saturated liquid at exit of Condenser.
11. The ambient temperature is 298K (T<sub>a</sub>) for exergy analysis.

3. Thermodynamic Analysis
In the thermodynamic analysis energy and exergy analysis of the system is done. For the energy analysis, cop of double effect Li-Br VARS is computed at base conditions. For the parametric analysis effect of generator temperature (LPG and HPG), condenser temperature, absorber temperature and evaporator temperature on cop is computed. In this analysis effect of 1°C change in generator temperature (LPG and HPG), condenser temperature, absorber temperature and evaporator temperature on COP is determined.

\[ \text{COP} = \frac{\dot{Q}_e}{\dot{Q}_{HPG}} \]  

For the exergy analysis exergy destruction of each component of the system is computed as well as total exergy destruction of whole system is computed at base conditions. Percentage contribution of individual component in exergy destruction is computed.

\[ I = T_0 \times S_{gen} \]
Here, $T_o$ is ambient temperature

$$S_{gen} = S_{out} - S_{in}$$  \tag{3}$$

$S_{in}$ = Entropy at inlet to the component.
$S_{out}$ = Entropy at outlet from the component.

### 3.1. Calculation of mass of blow down water available

- Inner diameter of steam drum = 1676 mm
- Length of steam drum = 15700 mm
- Power generated = 210 MW
- Pressure in steam drum = 150 bar
- Level of saturated liquid water in steam drum = 584 mm
- Level of saturated water vapor in steam drum = 1092 mm
- Specific volume of saturated liquid water in steam drum = 0.0016579 m$^3$/kg
- Specific volume of saturated water vapor in steam drum = 0.010340 m$^3$/kg
- Mass of saturated liquid water in steam drum = 2536.62 kg
- Mass of saturated water vapor in steam drum = 1422.04 kg
- Total mass of wet steam = 3958.66 kg
- Specific work output = 1430.98 kJ/kg
- Mass of steam required to generate 210 MW = 210000/1430.98 = 146.75 kg/sec
- Blow down water flow rate (3% of 146.75 kg/sec) = 4.4034 kg/sec

### 3.2. Governing Equations

#### Absorber

Conservation of mass equation gives

$$m_1 + m_{15} = m_4$$  \tag{4}$$

Conservation of Li-Br concentration equation gives

$$m_4 \times X_4 = m_{15} \times X_{15}$$  \tag{5}$$

Conservation of Energy equation gives

$$Q_{abs} = m_1 \times h_3 + m_{15} \times h_{17} - m_4 \times h_4$$  \tag{6}$$

#### Pump

$$\delta P = P_{hp} - P_{abs}$$  \tag{7}$$

$$W_p = \frac{m_4 \times \delta P}{\rho_4 \times \eta_p}$$  \tag{8}$$

$$h_5 = h_4 + W_p$$  \tag{9}$$

#### Solution heat exchanger 1

$$\epsilon_1 = \frac{T_{15} - T_{16}}{T_{15} - T_5}$$  \tag{10}$$

$$m_4 \times C_5 \times (T_6 - T_5) = m_{15} \times C_{15} \times (T_{15} - T_{16})$$  \tag{11}$$

#### Solution heat exchanger 2

$$\epsilon_2 = \frac{T_8 - T_9}{T_8 - T_6}$$  \tag{12}$$

$$m_4 \times C_6 \times (T_7 - T_6) = m_8 \times C_8 \times (T_8 - T_9)$$  \tag{13}$$

#### HPG

$$m_8 + m_{11} = m_4$$  \tag{14}$$

$$Q_{hp} = m_{11} \times h_{11} + m_8 \times h_8 - m_4 \times h_7$$  \tag{15}$$

#### LPG
\[ m_8 = m_{14} + m_{15} \]  
\[ m_8 \times X_8 = m_{15} \times X_{15} \]  
\[ Q_{lpg} = m_{11} \times (h_{11} - h_{12}) + m_8 \times h_{10} - m_{15} \times h_{15} - m_{14} \times h_{14} \]

Condenser

\[ m_4 = m_{11} + m_{14} \]  
\[ Q_c = m_{14} \times h_{14} + m_{11} \times h_{13} - m_1 \times h_1 \]

Evaporator

\[ m_1 = \frac{Q_e}{(h_3 - h_2)} \]  
\[ COP = \frac{Q_e}{Q_{hp}} \]

Second law analysis

Absorber

\[ I_{abs} = T_o \times ((m_4 \times S_4 - m_{15} \times S_{17} - m_3 \times S_3) + m_{24} \times C_{pw} \times \ln\left(\frac{T_{25}}{T_{24}}\right)) \]  
\[ C_{pw} = 4.187 \]  
\[ Q_{abs} = m_{24} \times C_{pw} \times (T_{25} - T_{24}) \]

HPG

\[ I_{hp} = T_o \times ((-m_4 \times S_7 + m_8 \times S_{8} + m_{11} \times S_{11}) + m_{18} \times C_{pw1} \times \ln\left(\frac{T_{19}}{T_{18}}\right)) \]  
\[ Q_{hp} = m_{18} \times C_{pw} \times (T_{18} - T_{19}) \]

Condenser

\[ Q_c = m_{20} \times C_{pw} \times (T_{21} - T_{20}) \]  
\[ I_{cond} = T_o \times ((-m_4 \times S_{14} - m_{11} \times S_{13} + m_1 \times S_1) + m_{20} \times C_{pw} \times \ln\left(\frac{T_{21}}{T_{20}}\right)) \]

Evaporator

\[ Q_e = m_{22} \times C_{pw} \times (T_{22} - T_{23}) \]  
\[ I_{evap} = T_o \times (m_1 \times (S_3 - S_2) + m_{22} \times C_{pw} \times \ln\left(\frac{T_{23}}{T_{22}}\right)) \]

Expansion Device

\[ I_{evap} = T_o \times (m_1 \times (S_2 - S_1)) \]

LPG

\[ I_{lpg} = T_o \times (m_{35} \times S_{15} + m_{11} \times S_{12} - m_{11} \times S_{11} - m_8 \times S_{10} + m_{14} \times S_{14}) \]

Solution heat exchanger 1

\[ I_1 = T_o \times (m_{15} \times C_{pw} \times \ln\left(\frac{T_{16}}{T_{15}}\right) + m_4 \times C_{pw} \times \ln\left(\frac{T_6}{T_5}\right)) \]

Solution heat exchanger 2

\[ I_2 = T_o \times (m_8 \times C_{pw} \times \ln\left(\frac{T_6}{T_5}\right) + m_4 \times C_{pw} \times \ln\left(\frac{T_6}{T_6}\right)) \]

Table 2. Result of properties at state points
| State Points | Temperature °K | Pressure KPa | Mass fraction Li-Br X (Kg) | Specific Enthalpy h (KJ/Kg K) | Density ρ (Kg/m³) | Specific Entropy s (KJ/Kg K) | Mass flow rate m |
|--------------|----------------|--------------|-----------------------------|-----------------------------|------------------|-----------------------------|----------------|
| 1            | 313            | 7.323        | 166.9                       | 0.1487                      | 0.1276           |                             |                |
| 2            | 313            | 43.3         | 166.9                       | 0.5923                      | 0.1276           |                             |                |
| 3            | 283            | 1.216        | 2519                        | 8.902                       | 0.1276           |                             |                |
| 4            | 313            | 1.216        | 0.5492                      | 93.79                       | 0.1276           |                             |                |
| 5            | 313            | 43.3         | 0.5492                      | 93.81                       | 0.1276           |                             |                |
| 6            | 342.7          | 43.3         | 0.5492                      | 155                         | 0.1276           |                             |                |
| 7            | 379.4          | 43.3         | 0.5492                      | 232.8                       | 0.1276           |                             |                |
| 8            | 401.8          | 43.3         | 0.5975                      | 290.1                       | 0.1276           |                             |                |
| 9            | 360.4          | 43.3         | 0.5975                      | 207.6                       | 0.1276           |                             |                |
| 10           | 360.4          | 7.323        | 0.5975                      | 207.6                       | 0.1276           |                             |                |
| 11           | 401.8          | 43.3         | 2739                        | 7.906                       | 0.1276           |                             |                |
| 12           | 350.9          | 43.3         | 325.7                       | 1.049                       | 0.1276           |                             |                |
| 13           | 350.9          | 7.323        | 325.7                       | 1.049                       | 0.1276           |                             |                |
| 14           | 368            | 7.323        | 2677                        | 8.565                       | 0.1276           |                             |                |
| 15           | 368            | 7.323        | 0.6436                      | 245.9                       | 0.1276           |                             |                |
| 16           | 329.5          | 7.323        | 0.6436                      | 176.7                       | 0.1276           |                             |                |
| 17           | 329.5          | 1.216        | 0.6436                      | 176.7                       | 0.1276           |                             |                |
| 18           | 428            | 101.325      | 653.3                       | 1.891                       | 3.438            |                             |                |
| 19           | 413            | 101.325      | 588.6                       | 1.738                       | 3.438            |                             |                |
| 20           | 298            | 101.325      | 104.7                       | 0.3648                      | 7.395            |                             |                |
| 21           | 298            | 101.325      | 104.7                       | 0.3648                      | 7.395            |                             |                |
| 22           | 293            | 101.325      | 83.74                       | 0.2941                      | 14.33            |                             |                |
| 23           | 288            | 101.325      | 62.81                       | 0.2221                      | 14.33            |                             |                |
| 24           | 298            | 101.325      | 104.7                       | 0.3648                      | 17.71            |                             |                |
| 25           | 303            | 101.325      | 125.6                       | 0.4345                      | 17.71            |                             |                |

4. Result and Discussion
The COP of double effect VARS depend upon evaporator temperature, absorber temperature, LPG temperature, HPG temperature and condenser temperature. For the parametric analysis, relevant temperature are varied and its effect is seen on COP.
The Figure 2 shows effect of change of evaporator temperature on COP. In this it can be clearly seen that COP increase with increase in evaporator temperature. The COP is 1.238 at 5°C and 1.356 at 15°C. The main reason for this increase in COP is that refrigerating effect increase with increase in evaporator temperature and cooling load (Q_c) is constant it implies that mass of refrigerant required would be less. So in present analysis evaporator pressure is considered equal to absorber pressure. Now as evaporator temperature increase evaporator pressure would also increase which implies higher value of absorber pressure for same absorber temperature. This results in higher value of mass coming out of absorber which reduces heat load of HPG due to which COP Increase.

The Figure 3 shows variation of LPG temperature on COP. In this it can be seen that COP increases with increase in LPG temperature. The COP is 1.126 at 80°C and 1.351 at 95°C. The governing reason is that mass (m_{11}) and enthalpy of steam vapor (h_{11}) from HPG increase at higher LPG temperature. It is also observed that mass and enthalpy of Li-Br mixture from HPG increase at higher LPG temperature. This effect is counter balanced by increase of mass flow rate of Li-Br mixture from absorber (m_4). The m_4 is 2.709 at 80°C and 0.8691 at 95°C. The heat transfer in HPG at 95°C is 222.1 whereas its value at 80°C is 266.4. So for given cooling load if heat transfer in HPG decrease COP will increase.
Figure 4. COP vs Absorber Temperature

The Figure 4 shows effect of absorber temperature on COP. It can be seen that as the absorber temperature increase COP decrease. The COP is 1.351 at 30°C and 1.311 at 40°C. The principal reason behind this is as temperature of absorber increase keeping effectiveness of heat exchangers and HPG temperature same, the enthalpy of weak Li-Br mixture (h7) entering the HPG would increase which in turn increase heat load in HPG. Thus, cooling load remain constant whereas thermal load in HPG increase. So COP decrease with increase of absorber temperature.

Figure 5. COP vs Condenser Temperature

The Figure 5 shows effect of condenser temperature on COP. It can be seen that as the condenser temperature increase COP decrease. The COP is 1.34 at 30°C and 1.312 at 40°C. The factor responsible for this is condenser pressure increase enthalpy at exit of condenser increase whereas enthalpy at exit of evaporator is same which reduces refrigerating effect thus mass of refrigerant (m1) required for given cooling load increase thus m4 mass also increase which result in increase in pump work. Now as m4 mass increase heat load on HPG increase thus lowering COP.
Figure 6 shows increase in COP with increase in variation of HPG temperature. COP is 1.351 at 128.8°C and 1.285 at 150°C. The following factors are responsible for the same.

1. As HPG temperature increases, HPG pressure increases to keep mass fraction of Li-Br coming out of HPG constant (recorded as 43.3 kPa at 128.8°C and 87.7 kPa at 150°C). It results in increase of pump work.

2. As HPG temperature increase enthalpy of water vapour \( (h_{11}) \) and of Li-Br mixture \( (h_8) \) increase which increases the heat load in HPG, thus reducing COP.

It is worth mentioning the LPG temperature is the most significant factor which affects COP of double effect VARS. COP value is 1.271 at 85°C and 1.351 at 95°C showing a 5.92% increase in COP.

**Table 3.** Irreversibility values of various components

| Sr. No | Name of the component       | Irreversibility (KW) | Irreversibility of each component as a percentage of total irreversibility (%) |
|--------|-----------------------------|----------------------|-------------------------------------------------------------------------------|
| 1      | Absorber                    | 26.17                | 40.02                                                                         |
| 2      | Solution heat exchanger 1   | 2.8669               | 4.3845                                                                        |
| 3      | Solution heat exchanger 2   | 4.954                | 7.576                                                                         |
| 4      | High pressure generator     | 2.377                | 3.6352                                                                        |
| 5      | Low pressure generator      | 3.063                | 4.6844                                                                        |
| 6      | Condenser                   | 6.4589               | 9.877                                                                         |
| 7      | Expansion device            | 16.85                | 25.76                                                                         |
| 8      | Evaporator                  | 2.647                | 4.048                                                                         |
|        | Total Irreversibility (KW)  | 65.3868              |                                                                               |

**Figure 6.** COP vs HPG Temperature
The above pie chart shows that Irreversibility of individual component as a percentage of total irreversibility of system. It shows that irreversibility in absorber is maximum 26.17 kW (40.02%) whereas irreversibility in HPG is minimum 2.377 kW (3.6353%).

5. Model Validation
The input values are set according to paper title “Energy and exergy analysis of single effect and series flow double effect water–lithium bromide absorption refrigeration systems” by Kaushik and Arora [18]. Parameters: T8 =140.6°C, T3= 7.2°C, T1=37.8°C, T4 = 37.8°C, effectiveness of solution heat exchangers 1 and 2 is 0.7, mass flow rate of refrigerant kg/s.

Table 4. Validation of Model with Previous Research

| Sr. No | Component | Quantity | Present Work | Kaushik and Arora | Percentage Error (%) |
|--------|-----------|----------|--------------|-------------------|----------------------|
| 1      | Absorber  | Q_{abs}  | 2954 kJ      | 2942.175 kJ       | 0.40                 |
| 2      | Condenser | Q_{cond} | 1351 kJ      | 1282.052 kJ       | 5.37                 |
| 3      | HPG       | Q_{hpg}  | 1897 kJ      | 1868.1 kJ         | 1.54                 |
| 4      | COP       |          | 1.242        | 1.26              | 1.42                 |

6. Conclusions
1. The vapour absorption cycle is possible with the heat recovered from blow down water.
2. COP of the VARS increase with increase of low pressure generator temperature.
3. COP of the VARS increase with increase of evaporator temperature.
4. COP of the VARS decrease with increase of high pressure generator temperature.
5. COP of the VARS decrease with increase of condenser temperature.
6. COP of the VARS decrease with increase of absorber temperature.
7. Absorber has maximum irreversibility of all components of VARS system.
8. High pressure generator has least irreversibility of all components of VARS system.
9. The COP of the VARS varies significantly with low pressure generator temperature.

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