CHAPTER 4

EXPERIMENTAL INVESTIGATIONS

4.1 INTRODUCTION

The thermodynamic analysis and exergy analysis on vapour absorption refrigeration system with ammonia – water as working fluids have been carried out and results are presented in chapter 3. Based on the results obtained from thermodynamic analysis, the design and fabrication of vapour absorption refrigeration system have been carried out. In this chapter, the details of design and fabrication of major components of the vapour absorption refrigeration system, the procedure for charging the working fluids, measurement of parameters, experimentation plan and procedure, and the results and discussions of experimentation are presented.

4.2 DESIGN OF THE EXPERIMENTAL SYSTEM

A biomass assisted vapour absorption refrigeration system, having 3 TR cooling capacity is designed and fabricated, based on the optimum results obtained from the thermodynamic process analysis. The heat transfer duty of the vapour absorption system components is given in Table 4.1. The relevant properties of ammonia-water are obtained from Ziegler and Trepp 1984, Patek and Klomfar 1995. The design and fabrication details of the vapour absorption system components are presented in the following sections.
Table 4.1  Heat transfer duty of absorption system components

<table>
<thead>
<tr>
<th>Component</th>
<th>Type</th>
<th>Heat transfer duty (kW)</th>
<th>Heat transfer area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condenser</td>
<td>Shell and tube</td>
<td>11.0</td>
<td>0.79</td>
</tr>
<tr>
<td>Reflux condenser</td>
<td>Shell and tube</td>
<td>3</td>
<td>0.14</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Shell and coil</td>
<td>10.64</td>
<td>0.51</td>
</tr>
<tr>
<td>Absorber</td>
<td>Vertical falling film</td>
<td>19</td>
<td>0.84</td>
</tr>
<tr>
<td>Generator</td>
<td>Shell and tube</td>
<td>23</td>
<td>4.43</td>
</tr>
<tr>
<td>Solution heat exchanger</td>
<td>Tube in tube</td>
<td>8.6</td>
<td>0.55</td>
</tr>
</tbody>
</table>

**Condenser**

The condenser is a horizontal shell and finned tube type. The cooling water is passed through the tube side and the refrigerant vapour passes through the shell side. The refrigerant enters the shell side as saturated vapour from the reflux condenser and leaves the shell as saturated liquid after condensation.

The dimensions and details of the condenser are given below.

- **Material**
  - Shell—Mild steel
  - Tube—Mild steel

- **Dimensions**
  - Shell diameter—0.22 m
  - Shell length—1.15 m
  - Tube diameter—0.0254 m
  - Tube length—1 m
  - Number of tubes—10
Figure 4.1 Condenser

A1 – Vapour NH$_3$ inlet  
B1 – Liquid NH$_3$ outlet

A2 – Cooling water inlet  
B2 – Cooling water outlet

All dimensions are in mm
Absorber

The absorber is of shell and tube type. The hot refrigerant vapour along with the hot strong solution from the solution heat exchanger enters shell side. After thorough mixing induced by the presence of baffles, the weak solution mixture passes out from the shell side. The cooling water passes through the tubes and cools the weak solution by extracting the heat of mixing.

Material : Shell—Mild steel
       Tube—Mild steel

Dimensions : Shell diameter —0.22 m
          Shell height—0.9 m
          Tube diameter—0.0254 m
          Tube length—0.75 m
          Number of tubes—14

Evaporator

The evaporator is of shell and coil type. The glycol passes through the shell and the refrigerant passes through the coil. In evaporator, the refrigerant is vapourized by absorbing heat from the glycol.

Material : Shell—Mild steel
       Coil—Mild steel

Dimensions : Shell diameter —0.24 m
          Shell height—0.33 m
          Tube diameter—0.0127 m
          Tube length—13 m
Figure 4.2 Absorber

Plan view

Sectional elevation

A1 - Strong solution inlet
A2 - Refrigerant inlet
B1 - Weak solution outlet
A3 - Cooling water inlet
B2 - Cooling water outlet

All Dimensions are in mm
A1 - Liquid NH₃ inlet       B2 - Glycol inlet
B1 - Vapour NH₃ outlet      A2 - Chilled glycol outlet

Figure 4.3 Evaporator
**Solution heat exchanger**

The solution heat exchanger is of tube in tube type. It is located in between the absorber and generator. It transfers the thermal energy from strong solution to weak solution. The weak solution passes through the inner tube and the strong solution passes through the annulus of the tube in tube heat exchanger.

Material : Both tubes— Mild steel

Dimensions :
- Inner tube diameter —0.0125 m
- Inner tube length—14 m
- Outer tube diameter—0.0254 m
- Outer tube length—14 m

**Reflux condenser**

The reflux condenser is used to remove reflux water from the vapour refrigerant stream, which leaves from the generator. The reflux condenser is a horizontal shell and tube type. The cooling water passes through the tube side and the refrigerant vapour passes through the shell side. The refrigerant absorbent vapour enters the shell side as saturated vapour from the generator and leaves the reflux condenser as saturated refrigerant vapour after rectification. The dimensions and details of the reflux condenser are given below.

Material :
- Shell-- Mild steel
- Tube-- Mild steel

Dimensions :
- Shell diameter—0.22 m
- Shell length—0.6 m
- Tube diameter—0.0127 m
- Tube length —0.51m
- Number of tubes—7
Figure 4.4 Solution heat exchanger
Figure 4.5  Reflux condenser

Elevation

A1 – Vapour NH$_3$ –H$_2$O inlet
B1 – Vapour NH$_3$ outlet
B2 – Cooling water outlet
A2 – Reflux outlet
A3 – Cooling water Inlet

All dimensions are in mm
Generator

In the generator, the refrigerant vapour is boiled off from the weak aqua ammonia solution. By burning, the producer gas, the required heat input to generator is supplied. The hot gases, which pass through, the tubes of generator heat up the weak solution and bring it to the boiling condition. The spent flue gases are then vented off through the chimney. The dimensions and details of the generator are given below.

- **Type**: Shell and tube type
- **Fluids**: Shell side—weak solution
  Tube side—hot gases
- **Material**: Shell—Mild steel
  Tube—Mild steel
- **Dimensions**: Shell diameter—0.312 m
  Shell height—1.592 m
  Tube diameter—0.0254 m
  Tube length—1.51 m
  Number of tubes—37

Solution Pump

The positive displacement diaphragm pump is used as solution pump, whose specifications are given in Table 4.4. The solution pump draws the weak solution from the solution reservoir and supplies it to generator through the solution heat exchanger. The available head at the inlet of the solution pump should be equal to the sum of the net positive suction head and
Figure 4.6 Generator

A1 - NH3-H2O vapour outlet
A2 - Reflux water inlet
A3 - Weak solution inlet
B1 - Strong solution outlet
C1 - Flue gas outlet
LG - Level gauge

All dimensions are in mm
the additional head to pump the solution through the pipelines. The pump is fixed at a suitable place, so that, it could get the required head for smooth functioning. The weak solution flow rate could be varied from 0 to 150 lit/hr, by varying the stroke length of the solution pump, with respect to the operating conditions.

4.3 BIOMASS DOWNDRAFT GASIFIER

A biomass downdraft gasifier is coupled with vapour absorption refrigeration system. The gasifier generates producer gas by gasification process. The producer gas is burnt to generate hot gas. Generated hot gas is circulated through the tubes of the generator to supply heat to the weak solution to generate refrigerant vapour.

The details of the downdraft gasifier are given below

<table>
<thead>
<tr>
<th>Mode</th>
<th>Thermal application</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Down draft type with throat</td>
</tr>
<tr>
<td>Fuel</td>
<td>Wood chips</td>
</tr>
<tr>
<td>Size of wood chips</td>
<td>20-50 mm</td>
</tr>
<tr>
<td>Rating</td>
<td>20 kW_e</td>
</tr>
<tr>
<td>Fuel loading</td>
<td>Manual</td>
</tr>
<tr>
<td>Period of operation</td>
<td>10-20 hours/day</td>
</tr>
<tr>
<td>Auxiliary power requirement</td>
<td>&lt; 1 HP</td>
</tr>
</tbody>
</table>

**Working of downdraft gasifier**

The downdraft gasifier consists of the following components as shown in Figure 4.7.
Installation of downdraft gasifier

The gasifier assembly (1) is kept inside the water tank (10) on a level surface after locating the drain plug of the water tank to a side where drain water can be let out without interference. The gas pipe (18) is connected with the gas outlet (16) of the gas generator (5) through the flange (17). The other water tank (26) is inserted under the dust remover (19). The mixing tube (22) is connected to the dust collector through gas pipe (21) and flange (20). The burner is then screwed to the elbow (28) fixed to the mixing tube (22).

The air inlet pipe (15) is then connected with the air manifold (12) through the valve (13). The air duct (11) is connected to the air manifold (12) and the blower outlet (3). One end of the primary air duct (23) is connected with the air manifold (12) through valve (14). The other end is connected with the mixing nozzle (24) nipple. Water is then poured into both tanks (10, 26) up to the brim. Finally the motor (27) is connected to the power line. The unit is now ready for charging. The operation of the gasifier involves various stages to be dutifully carried out so that smooth operation of the unit is maintained. The operational procedure is given below.

Fuel preparation

The gasifier fuel is a fire wood chip. Any species of wood can be used but the average size should be around 20 to 50 mm. Using the woodcutter the long pieces are cut to the required suitable and convenient dimension.
Figure 4.7 Biomass Downdraft Gasifier

Start up

The gasifier (1) is filled up to 50mm above the firing port (6) level with charcoal via the inspection door (8). The gasifier hopper (4) is then loaded with wood chips through charging door (7). Water is poured into both the water tanks (10, 26) up to the brim.
After charging completely the charging door (7) and the inspection door (8) are locked with the clamps. The blower (3) is started by switching on the motor (27). The gas control valve (13) is kept close and the pressure control valve (14) is opened fully. A lighting wick is placed near the firing port (6) and is left in place for about five minutes till the coal inside starts to burn. This is found by visual observation via the firing port (6). After three more minutes in the same manner the firing port is closed with a dummy. The gas control valve (13) is slowly opened and the gas at the burner (2) is lighted with a lit wick. The valves (13, 14) are adjusted so as to get the required burning characteristics such as flame length and temperature and the burner (2). The flame volume is positively controlled by the gas control valve (13) and the flame pressure is controlled by the pressure control valve (14). The shaking rod (9) attached to the gas generator (5) is shaken back and forth periodically to remove any clinker formed inside the reaction chamber. This is usually done when the flame length reduces.

**Shut down**

The blower motor (27) is first shut down and then both the valves (13, 14) are closed. The flame will grow small and slow down and finally extinguish itself completely.

**Recharging**

After the motor (27) has been shut down and valves (13, 14) are closed the gasifier (1) is allowed to cool it for around 15 to 30 minutes. After the gasifier is cooled sufficiently, the charging door (7) is unscrewed and removed. The wood chips are loaded into the hopper and the door is closed and clamped.
4.4 EXPERIMENTAL SETUP

All the vapour absorption refrigeration system components are fabricated as per design specifications and assembled after leak testing procedures. This vapour absorption refrigeration system is coupled with 20 kW_e downdraft gasifier and again subjected to final leak testing, before experimentation. The schematic diagram of the system is given in Figure 4.8. Figures 4.9 and 4.10 show the photographic view of the experimental setup before insulation. Figures 4.11, 4.12, 4.13 and 4.14 show the photographic view of the experimental setup after insulation. Figure 4.15 shows the photographic view of the combustion of producer gas after integration with vapour absorption refrigeration system. The gasifier is charged with wood chips completely and fired for gasification. An air blower supplies the required air for gasification. A U-tube manometer is used to measure the required air to be supplied for gasification. The producer gas, which is the end product of gasification process, is burnt in the furnace shell (F) and as a result, the hot gases are generated. The hot gases then pass through the generator (G) and transfer its heat load to the refrigerant and absorbent mixture. As the boiling point of the refrigerant is less than the absorbent, the refrigerant is vapourized from absorbent as saturated vapour and flows into the reflux condenser (RC). The reflux condenser is used to remove the reflux water from the vapour refrigerant stream, which leaves from the generator. This has been done by circulating cooling water (CT) through the reflux condenser by a circulating pump. The rectified pure vapour refrigerant then flows into the condenser (C). In condenser, by circulating cooling water (CT), the saturated vapour refrigerant is condensed into saturated liquid refrigerant. The condensate is then collected, in the refrigerant reservoir (RR).
High pressure saturated liquid refrigerant from the refrigerant reservoir passes to the evaporator (E) at low pressure through an expansion valve (ED). In evaporator the required cooling load \( Q_e \) is obtained by absorbing heat from the glycol, which is to be cooled, by continuous vapourization of refrigerant. The glycol is supplied from a glycol tank (GT), which is well insulated. Then the saturated vapour refrigerant flows to the absorber (A), where it is absorbed by a strong solution (strong in absorbent) and becomes weak solution (weak in absorbent). The driving potential for this flow is the difference in the pressure of the refrigerant in evaporator and vapour pressure of the absorbent-refrigerant solution in the absorber. This absorption process is exothermic reaction. The heat released during the absorption process, should be removed as heat sink \( Q_a \) to maintain the required low temperature, for continuous absorption. The cooling water to extract the heat from absorber is circulated by circulating pump (CT). The resulting weak solution is collected in a solution reservoir (SR), and then pumped by the solution pump (SP) to the generator at high pressure. The solution pump is of positive displacement diaphragm type. By varying the stroke length of the pump, the quantity of weak solution supplied to the generator is varied. With the flow rate of weak solution, the refrigerant mass flow rate and strong solution flow rate are calculated.

In generator, heat input \( Q_g \) is supplied by combusting the producer gas as said earlier. Strong solution left in the generator is throttled back to the absorber through a pressure-reducing valve (PRV). A solution heat exchanger (SHX), existing between the generator and absorber is enhancing the heat transfer from strong solution to weak solution. So, that the heat energy to be supplied to the generator is reduced. The flow measurements are done to measure the cooling water flow rates supplied to cool the respective components. A separate valve is provided to vary the
absorber cooling water flow rate. The needle valves are used as expansion
deVICES in refrigerant and solution flow circuits. The temperature and pressure
measuring points are represented as T, P in Figure 4.8.

All the components are fixed rigidly on a platform with suitable
supports. The provisions are made to charge the refrigerant in refrigerant
reservoir and the absorbent in the solution receiver. The relative vertical
locations are determined and fixed for the components to exploit the gravity
force for the flow of solution and refrigerant through the system.
Figure 4.8 Schematic arrangement of the experimental setup
Figure 4.9 Photographic view of the experimental setup before insulation (left)
Figure 4.10 Photographic view of the experimental setup before insulation (right)
Figure 4.11  Photographic view of the experimental setup after insulation (right)

1. Generator
2. Reflux condenser
3. Condenser
4. Absorber
5. Refrigerant reservoir
6. Solution reservoir
7. Furnace shell
8. Downdraft gasifier
Figure 4.12 Photographic view of the experimental setup after insulation (left)

1. Generator
2. Reflux condenser
3. Condenser
4. Refrigerant reservoir
5. Solution reservoir
6. Furnace shell
7. Fire hole
8. Heat exchanger
9. Absorber
10. Downdraft gasifier
Figure 4.13  Photographic view of the experimental setup after insulation (front)
Figure 4.14 Photographic view of the experimental setup after insulation (back)
Figure 4.15  Photographic view of the combustion of producer gas, after coupled with vapor absorption refrigeration system
4.5  CHARGING PROCEDURE

The system components are fabricated as per design specifications and subjected to leak testing before being assembled. First of all, all the components are subjected to hydraulic testing at 25 bar. The pressure is maintained at the same level, for 3 days to check for any leakage. Then, the leak test is carried out with nitrogen gas at a pressure of 25 bar. Again the pressure is maintained at the same level for 3 days. The system components are assembled and instrumented with calibrated pressure gauges, thermocouples and flow meters at the required points as shown in the schematic diagram of the experimental set up Figure 4.8. The system is again subjected to leak testing following the above procedure. The system is then evacuated to an extent of 30 mmHg and the vacuum is maintained for 3 days. Then, the system is charged with refrigerant vapour, to remove all non-condensable gases that may be present and again evacuated. Finally, the system is charged with the calculated quantities of refrigerant and absorbent. The calculation procedure to find out the required quantity of refrigerant and absorbent is given in Appendix 3.

4.6  MEASUREMENTS OF PARAMETERS

Details of the instrumentation are presented in this section. The various quantities to be measured are pressure and temperature. The location of the measuring parameters is shown in Figure 4.8. The Tables 4.2 and 4.3 give the description of the quantity to be measured at different locations.

Pressure Measurement

Since the system encounters only positive pressures, pressure measurement is made by bourdon gauge. These gauges are individually
calibrated using dead weight gauge. The range is from 0 to 30 bar. These gauges are fixed at 6 points in the absorption system and marked as P for pressure in Figure 4.8. The uncertainty associated with pressure measurement is ± 0.05 bar on low pressure side and ± 0.1 bar on high pressure side.

Table 4.2 Pressure measurement locations

<table>
<thead>
<tr>
<th></th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>P₁</td>
<td>Refrigerant liquid outlet pressure of the condenser</td>
</tr>
<tr>
<td>P₂</td>
<td>Refrigerant liquid outlet pressure of the refrigerant reservoir</td>
</tr>
<tr>
<td>P₃</td>
<td>Refrigerant Vapour at the inlet of the absorber</td>
</tr>
<tr>
<td>P₄</td>
<td>Weak solution outlet pressure of solution pump</td>
</tr>
<tr>
<td>P₅</td>
<td>Vapour aqua ammonia outlet pressure of generator</td>
</tr>
<tr>
<td>P₆</td>
<td>Strong solution outlet pressure of generator</td>
</tr>
</tbody>
</table>

Temperature measurement

Temperature measurements are made using 30 gauge PTPE coated copper-constantan thermocouple. These thermocouples are fixed at 26 points in the experimental setup and marked as T for temperature in Figure 4.8. The uncertainty associated with the measurements is ± 0.5°C.

Flow measurement

The mass flow rates of cooling water and glycol are calculated individually by measuring the time required for the collection of 10 litres of the respective cooling water and glycol. The uncertainty associated with cooling water and glycol flow rate measurements is of ± 3%.
### Table 4.3 Temperature measurements locations

<table>
<thead>
<tr>
<th>Sensor location</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>T₁</td>
<td>Saturated aqua ammonia Vapour at the exit of generator</td>
</tr>
<tr>
<td>T₂</td>
<td>Saturated Vapour ammonia at the exit of reflux condenser</td>
</tr>
<tr>
<td>T₃</td>
<td>Saturated liquid ammonia at the exit of the condenser</td>
</tr>
<tr>
<td>T₄</td>
<td>Saturated liquid ammonia at the exit of the refrigerant reservoir</td>
</tr>
<tr>
<td>T₅</td>
<td>Vapour-liquid ammonia at the inlet of evaporator</td>
</tr>
<tr>
<td>T₆</td>
<td>Saturated Vapour ammonia at the exit of evaporator</td>
</tr>
<tr>
<td>T₇</td>
<td>Saturated liquid weak solution at the exit of absorber</td>
</tr>
<tr>
<td>T₈</td>
<td>Sub cooled liquid weak solution at the exit of solution reservoir</td>
</tr>
<tr>
<td>T₉</td>
<td>Sub cooled liquid weak solution at the exit of solution pump</td>
</tr>
<tr>
<td>T₁₀</td>
<td>Sub cooled liquid weak solution at the outlet of heat exchanger</td>
</tr>
<tr>
<td>T₁₁</td>
<td>Sub cooled liquid weak solution at the inlet of generator</td>
</tr>
<tr>
<td>T₁₂</td>
<td>Saturated liquid strong solution at the outlet of generator</td>
</tr>
<tr>
<td>T₁₃</td>
<td>Saturated liquid strong solution at the inlet of heat exchanger</td>
</tr>
<tr>
<td>T₁₄</td>
<td>Sub cooled liquid strong solution at the outlet of heat exchanger</td>
</tr>
<tr>
<td>T₁₅</td>
<td>Vapour-liquid solution at the inlet of absorber</td>
</tr>
<tr>
<td>T₁₆</td>
<td>Saturated liquid solution at the inlet of generator</td>
</tr>
<tr>
<td>T₁₇</td>
<td>Coolant water inlet of the reflux condenser</td>
</tr>
<tr>
<td>T₁₈</td>
<td>Coolant water outlet of the reflux condenser</td>
</tr>
<tr>
<td>T₁₉</td>
<td>Coolant water inlet of the condenser</td>
</tr>
<tr>
<td>T₂₀</td>
<td>Coolant water outlet of the condenser</td>
</tr>
<tr>
<td>T₂₁</td>
<td>Glycol at the inlet of the evaporator</td>
</tr>
<tr>
<td>T₂₂</td>
<td>Glycol at the outlet of the evaporator</td>
</tr>
<tr>
<td>T₂₃</td>
<td>Coolant water inlet of the absorber</td>
</tr>
<tr>
<td>T₂₄</td>
<td>Coolant water outlet of the absorber</td>
</tr>
<tr>
<td>T₂₅</td>
<td>Ambient condition</td>
</tr>
<tr>
<td>T₂₆</td>
<td>Producer gas at the exit of the gasifier</td>
</tr>
</tbody>
</table>
The specifications of the cooling water pump, glycol pump, solution pump and air blower motor are given below.

**Table 4.4 Specification of the equipments**

<table>
<thead>
<tr>
<th>Make</th>
<th>Condenser</th>
<th>Make</th>
<th>Evaporator</th>
<th>Make</th>
<th>Absorber</th>
</tr>
</thead>
<tbody>
<tr>
<td>Make</td>
<td>Best centrifugal pump</td>
<td>Make</td>
<td>Sharp monoblock pump</td>
<td>Make</td>
<td>Ventura centrifugal pump</td>
</tr>
<tr>
<td>Volt</td>
<td>200V</td>
<td>Volt</td>
<td>230V</td>
<td>Volt</td>
<td>240V</td>
</tr>
<tr>
<td>Power</td>
<td>0.37kW</td>
<td>Power</td>
<td>0.09 kW</td>
<td>Power</td>
<td>0.37 kW</td>
</tr>
<tr>
<td>Speed</td>
<td>2760 r.p.m</td>
<td>Speed</td>
<td>2700</td>
<td>Speed</td>
<td>2800</td>
</tr>
<tr>
<td>Current</td>
<td>3.7Amps</td>
<td>Current</td>
<td>0.6 A</td>
<td>Current</td>
<td>2.8A</td>
</tr>
<tr>
<td>Head/Discharge</td>
<td>6/30M</td>
<td>Discharge</td>
<td>10000lph</td>
<td>Discharge</td>
<td></td>
</tr>
<tr>
<td>Make</td>
<td>Crystal monoblock pump</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-----------------------</td>
<td>------------------------</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Volt</td>
<td>230V</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Power</td>
<td>0.1 kW</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Speed</td>
<td>2850</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Current</td>
<td>0.7A</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Head</td>
<td>75 feet</td>
<td></td>
<td></td>
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</table>

**Solution pump**

<table>
<thead>
<tr>
<th>Make</th>
<th>Swelore, WV/3000/33</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Diaphragm pump</td>
</tr>
<tr>
<td>Discharge pressure</td>
<td>35kg/cm²</td>
</tr>
<tr>
<td>Flow rate</td>
<td>150 lit/hr</td>
</tr>
<tr>
<td>Stroke length</td>
<td>100 strokes/min (max)</td>
</tr>
<tr>
<td>Power</td>
<td>0.37 kW</td>
</tr>
</tbody>
</table>

**Blower motor**

<table>
<thead>
<tr>
<th>Make</th>
<th>New Bharat, Ahmedabad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volt</td>
<td>230V</td>
</tr>
<tr>
<td>Power</td>
<td>0.7 kW</td>
</tr>
<tr>
<td>Speed</td>
<td>280</td>
</tr>
<tr>
<td>Current</td>
<td>6.5A</td>
</tr>
<tr>
<td>Discharge</td>
<td>10000lph</td>
</tr>
</tbody>
</table>
4.7 EXPERIMENTAL PLAN

The experiments are carried out to evaluate the performance of the system by varying the biomass flow rate, weak solution flow rate, cooling water temperature and its flow rate. The cooling water flow rate for condenser and reflux condenser is kept constant at 0.212 kg/s and glycol flow rate is kept constant at 0.256 kg/s throughout the experimentation. The temperature ranges are fixed based on the practical limitations.

1. Cooling water temperatures are varied from 20$^\circ$C to 35$^\circ$C.
2. The weak solution flow rate is varied from 0.013 kg/s to 0.036 kg/s.
3. The absorber cooling water flow rate is varied from 0.4 kg/s to 0.6 kg/s.
4. Biomass flow rate is varied from 2 to 15 kg/h to demonstrate the capacity.

4.8 EXPERIMENTAL PROCEDURE

The firewood is charged in the downdraft gasifier and gasified. An air blower with a power rating of 0.7 kW is employed to supply air at the required rate for gasification process. The gasifier is started up and allowed to run for a specified time. The parameters such as temperature of output gas and primary air pressure drop are observed periodically. Once the flow rate of the gas and its temperature are stabilized, the gas is ignited in the furnace of the absorption unit. The temperature and pressure in the generator are monitored continuously. When the pressure reaches the corresponding condenser pressure, the outlet valve is opened for the flow of refrigerant vapour to the condenser via reflux condenser to improve the vapour quality of ammonia.
vapour. Solution is circulated between the generator and the absorber by the solution pump, for the continuous generation of refrigerant vapour. When the pressure of the condenser reaches the saturation pressure of the required condenser temperature and the condenser pressure reaches a value greater by 1 bar than that of the condenser, all the valves in the solution and refrigerant circuit are opened. The cooling water is circulated in the condenser and absorber to remove heat. Flow rate of water to the absorber, condenser and reflux condenser are adjusted until its exit temperature is greater than the preset temperature.

Glycol flow rate to the evaporator from the glycol tank is adjusted so that the constant evaporator capacity can be maintained. The expansion device between the refrigerant reservoir and evaporator is adjusted in such a way that a constant flow refrigerant liquid is attained. The flow rate and temperature of chilled glycol are measured continuously to maintain constant evaporator temperature, and hence constant evaporator pressure. Additional electrical load is also provided to simulate the actual cold room conditions. Readings are noted periodically. The level of solution in both the reservoirs is maintained constant for steady state operation, regulating the flow of strong solution from the generator into the absorber through needle valves. When two to three successive readings are almost the same, it can be concluded that steady state has been reached. It took 40 minutes to 1 hour to attain steady state. The temperature, pressure, and flow rates, in both the solution circuits are noted periodically.

4.9 DATA REDUCTION

The data measured during experimentation at various operating conditions are to be reduced to get the required results such as heat loads at various components, cooling load, coefficient of performance and second law
efficiency. The model calculation with a set of measured data is given in Appendix 4. The heat quantities and the performance parameters calculated with reference to Figure 4.8 are as follows.

Thermodynamic heat quantities are calculated as given in the following equations

Heat input to evaporator

\[ Q_e = m_6 h_6 - m_5 h_5 \]  
(4.1)

Heat rejection in condenser

\[ Q_c = m_4 h_2 - m_3 h_3 \]  
(4.2)

Heat rejection in reflux condenser

\[ Q_{rec} = m_i h_i - m_2 h_2 - m_{16} h_{16} \]  
(4.3)

Heat rejection in absorber

\[ Q_a = m_6 h_6 + m_{15} h_{15} - m_7 h_7 \]  
(4.4)

Heat input in generator

\[ Q_s = m_i h_i + m_{12} h_{12} - m_{10} h_{10} - m_{16} h_{16} \]  
(4.5)

Heat transferred in heat exchanger

\[ Q_{sh} = m_{ws} (h_{10} - h_9) \]  
(4.6)
Performance Parameters are calculated as given in the following correlations.

Circulation ratio (CR)
\[
\lambda = \frac{m_{in}}{m_r} \tag{4.7}
\]

Coefficient of performance
\[
\text{COP}_{th} = \frac{Q_e}{Q_g} \tag{4.8}
\]

Carnot coefficient of performance
\[
\text{COP}_c = \left( \frac{T_g - T_a}{T_g} \times \frac{T_c}{T_c - T_e} \right) \tag{4.9}
\]

Second law efficiency
\[
\eta_{II} = \frac{\text{COP}_{th}}{\text{COP}_c} \tag{4.10}
\]

The actual heat quantities are estimated as given in the following equations.

Generator heat input,
\[
Q_{g,a} = m_{hot \text{ gas}} \cdot CV \tag{4.11}
\]
Evaporator heat input,

\[ Q_{e,a} = m_{e,g} C_p (T_{21} - T_{22}) \]  \hspace{1cm} (4.12)

Condenser heat load,

\[ Q_{c,a} = m_{c,w} C_p (T_{20} - T_{19}) \]  \hspace{1cm} (4.13)

Absorber heat load,

\[ Q_{a,a} = m_{a,w} C_p (T_{24} - T_{23}) \]  \hspace{1cm} (4.14)

Rectifier heat load,

\[ Q_{rec,a} = m_{rec,w} C_p (T_{18} - T_{17}) \]  \hspace{1cm} (4.15)

The performance parameters based on the actual heat quantities are calculated as follows:

Actual coefficient of performance

\[ COP_a = \frac{Q_e}{Q_g} \]  \hspace{1cm} (4.16)

Real coefficient of performance

\[ COP_r = \frac{Q_e}{Q_g + W} \]  \hspace{1cm} (4.17)
Exergy analysis

The exergy change in the components, the internal irreversibility, the heat transfer irreversibility and the exergetic co-efficient of performance have been estimated. The equations for the exergy analysis of the vapour absorption refrigeration system have been given with reference to Figure 4.8. (Aphornratana et al 1995; Ataer et al 1991; Park et al 1991).

Condenser

Exergy change of working fluid in the condenser is given by

\[ \Delta E_c = m_r \left( E_2 - E_3 \right) \]  
(4.18)

Internal irreversibility is difference between exergy change in working fluid and exergy of heat flux \( Q_c \).

\[ \Delta E_{c,\text{int}} = m_r \left( E_2 - E_3 \right) - Q_c \left( 1 - \frac{T_o}{T_c} \right) \]  
(4.19)

The heat transfer irreversibility is the difference between the exergy of heat flux \( Q_c \) and the exergy change in the cooling water.

\[ \Delta E_{c,\text{bt}} = Q_c \left( 1 - \frac{T_o}{T_c} \right) - m_{c,w} \left( E_{20} - E_{19} \right) \]  
(4.20)

In a similar manner, exergy change and the irreversibilities for all the remaining components of biomass assisted vapour absorption refrigeration system are given below.
Evaporator

\[ \Delta E_e = m_r (E_3 - E_6) \]  
(4.21)

\[ \Delta E_{e, \text{int}} = m_r (E_3 - E_6) - Q_e \left(1 - \frac{T_a}{T_e}\right) \]  
(4.22)

\[ \Delta E_{e, \text{ht}} = Q_e \left(1 - \frac{T_a}{T_e}\right) - m_{e, \text{gly}} (E_{21} - E_{22}) \]  
(4.23)

Absorber

\[ \Delta E_a = m_a E_6 + m_{15} E_{15} - m_7 E_7 \]  
(4.24)

\[ \Delta E_{a, \text{int}} = m_a E_6 + m_{15} E_{15} - m_7 E_7 - Q_a \left(1 - \frac{T_a}{T_a}\right) \]  
(4.25)

\[ \Delta E_{a, \text{ht}} = Q_a \left(1 - \frac{T_a}{T_a}\right) - m_{a, \text{w}} (E_{24} - E_{23}) \]  
(4.26)

Rectifier

\[ \Delta E_{\text{rec}} = m_1 E_1 + m_r E_2 - m_{16} E_{16} \]  
(4.27)

\[ \Delta E_{\text{rec, int}} = m_1 E_1 + m_r E_2 - m_{16} E_{16} - Q_{\text{rec}} \left(1 - \frac{T_a}{T_{\text{rec}}}\right) \]  
(4.28)

\[ \Delta E_{\text{rec, ht}} = Q_{\text{rec}} \left(1 - \frac{T_a}{T_{\text{rec}}}\right) - m_{\text{rec, w}} (E_{18} - E_{17}) \]  
(4.29)

Generator

\[ \Delta E_g = m_{16} E_{16} + m_{10} E_{10} - m_4 E_4 - m_{12} E_{12} \]  
(4.30)
\[
\Delta E_{\text{g, int}} = m_{16}E_{16} + m_{40}E_{10} - m_{12}E_{12} - m_tE_t - Q_g \left(1 - \frac{T_o}{T_g}\right)
\]  
(4.31)

\[
\Delta E_{\text{g, ht}} = Q_g \left(1 - \frac{T_o}{T_g}\right) - m_{\text{hot gas}} \left(\Delta E_{\text{hot gas}}\right)
\]  
(4.32)

Solution heat exchanger

\[
\Delta E_{\text{sh}} = m_3E_g + m_{13}E_{13} - m_{10}E_{10} - m_{14}E_{14}
\]  
(4.33)

Solution valve

\[
\Delta E_{SV} = \frac{m_{14} (p_{14} - p_{15})}{\rho_{14}}
\]  
(4.34)

Refrigerant valve

\[
\Delta E_{RV} = m_3 (E_3 - E_5)
\]  
(4.35)

Solution pump

\[
\Delta E_p = \frac{m_8 (p_8 - p_5)}{\rho_8}
\]  
(4.36)

Exergetic coefficient of performance

\[
ECOP = \frac{m_6 (E_6 - E_4)}{m_1E_1 + m_{12}E_{12} - m_{10}E_{10} - m_{16}E_{16}}
\]  
(4.37)

4.10 RESULTS AND DISCUSSION

The experiments have been carried out on the fabricated system as outlined in the previous sections. The operating parameters such as sink temperature, weak solution flow rate, absorber cooling water flow rate and
biomass flow rate have been varied to analyze their influence on the performance system. The effects of operating parameters on circulation ratio, COP and second law efficiency have been obtained. The actual heat quantities obtained are compared with thermodynamic heat quantities. The detailed discussions on the results obtained are presented in this section.

4.10.1 Time Temperature history of components

Figure 4.16 shows the component temperature history with respect to time at the rate of 8 kg/h of biomass consumption. It is observed that the generator temperatures increase at a faster rate initially and continue to increase until the steady state is reached. As the load on the absorber and condenser are increased, the temperatures at the outlet of these components increase before reaching a steady state. It can be observed that the system attained steady state in about an hour, and achieved an evaporator temperature of –5°C with the circulating glycol temperature around 8°C.

![Figure 4.16 Time temperature history of absorption system components](image)

Figure 4.16 Time temperature history of absorption system components
4.10.2 Circulation ratio

In this section effect of operating parameters over circulation ratio is discussed. Figure 4.17 shows the variation of the circulation ratio with the increase in weak solution flow rate, at different sink temperature and constant absorber cooling water flow rate. When the weak solution flow rate increases at constant sink temperature and at constant absorber cooling water flow rate, the circulation ratio decreases. When the sink temperature increases at constant weak solution flow rate and constant absorber cooling water flow rate, the circulation ratio increases.

![Graph showing the effect of weak solution flow rate over circulation ratio.](image)

**Figure 4.17 Effect of weak solution flow rate over circulation ratio**

At constant sink temperature and constant absorber cooling water flow rate, when weak solution flow rate increases, the concentration of weak solution increases. Therefore the mass flow rate of the refrigerant also increases. Since the absorber cooling water flow rate and sink temperature are constant, the rate of increase in the refrigerant flow rate is more than the rate
of increase in the weak solution flow rate. Therefore as the weak solution flow rate increases circulation ratio decreases.

At constant weak solution flow rate and constant absorber cooling water flow rate, when the sink temperature increases, the absorber temperature also increases. So the concentration of weak solution, which leaves from the absorber decreases causing the concentration spread to decrease. Therefore the circulation ratio increases.

**Figure 4.18**  Effect of absorber cooling water flow rate over circulation ratio

Figure 4.18 shows the variation of the circulation ratio with increase in the absorber cooling water flow rate, for various sink temperature and constant weak solution flow rate. When the absorber cooling water flow rate increases at constant sink temperature and constant weak solution flow rate, the circulation ratio decreases. At constant absorber cooling water flow
rate and constant weak solution flow rate, as the sink temperature increases the circulation ratio increases.

At constant sink temperature and constant weak solution flow rate, when the absorber cooling water flow rate increases, the cooling water absorbs more heat energy. So the weak solution concentration increases. The mass flow rate of refrigerant also increases. Since the weak solution flow rate is constant, as mass flow rate of refrigerant increases, the circulation ratio decreases.

When the sink temperature increases, the weak solution concentration decreases, which in turn decreases the refrigerant flow rate. Since the flow rate of weak solution is constant, when refrigerant flow rate decreases, the circulation ratio increases.

Figure 4.19 shows the effect of sink temperature over circulation ratio at various weak solution flow rates and at constant absorber cooling water flow rate. At constant sink temperature and constant absorber cooling water flow rate as the weak solution flow rates increases the circulation ratio decreases. At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases the circulation ratio increases.

At constant absorber cooling water flow rate and constant weak solution flow rate, when the sink temperature increases the weak solution concentration decreases. Therefore the mass flow rate of refrigerant will decrease. So the circulation ratio increases. At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases the refrigerant flow rate increases. Therefore the circulation ratio decreases.
4.10.3 Coefficient of performance

The influence of operating parameters over coefficient of performance of the vapour absorption refrigeration system is analyzed.

Figure 4.20 shows the variation of the COP with increase in the weak solution flow rate, for various sink temperature and constant absorber cooling water flow rate. When the weak solution flow rate increases at constant sink temperature and constant absorber cooling water flow rate the COP increases. When the sink temperature increases, at constant weak solution flow rate and constant absorber cooling water flow rate the COP decreases.
Figure 4.20 Effect of weak solution flow rate over COP

At constant weak solution flow rate and constant absorber cooling water flow rate, when the sink temperature increases, due to decrease in the concentration of weak solution, the mass flow rate of refrigerant decreases. Therefore the cooling load decreases. The required heat load to be supplied in generator, to desorb the refrigerant also decreases correspondingly. Therefore the COP decreases with increase in the sink temperature.

At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the refrigerant flow rate also increases. Therefore the cooling load increases. The generator load also increases correspondingly. Although there is an increase in the generator load, as the cooling load increases, the COP increases. Since there is heat loss in the system during experimentation, the actual COP is always less than the thermodynamic COP.
Figure 4.21 shows the variation of the COP with increase in the absorber cooling water flow rate, when the sink temperature is varied and the weak solution flow rate is kept constant. As expected, the COP remains constant with the increase in the absorber cooling water flow rate, for a particular weak solution flow rate and sink temperature. As the sink temperature decreases, the coefficient of performance of the system improves due to the higher generation of refrigerant vapour from the generator for the given heat input.

Figure 4.22 shows the effect of sink temperature over COP at various weak solution flow rates and constant absorber cooling water flow rate. At constant sink temperature and constant absorber cooling water flow rate as the weak solution flow rates increases COP increases. At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases COP decreases.
At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases the cooling load also increases. Although the generator load increases correspondingly, as the cooling load is more, the COP is increased.

When the sink temperature increases, the refrigerant mass flow rate is reduced. Therefore the cooling load is decreased. Since there is a corresponding decrease in generator load, when the sink temperature increases COP decreases. The actual COP is less than thermodynamic COP as expected

**4.10.4 Second law efficiency**

The impact of operating parameters on second law efficiency of the vapour absorption system is discussed. Figure 4.23 shows the variation of the
second law efficiency with increase in the weak solution flow rate, when the
sink temperature is varied from 26°C to 33 °C and the absorber cooling flow
rate is constant. When the weak solution flow rate increases at constant sink
temperature and constant absorber cooling water flow rate the second law
efficiency decreases. When the sink temperature increases, at constant weak
solution flow rate and constant absorber cooling water flow rate the second
law efficiency increases.

At constant absorber cooling water flow rate and constant sink
temperature, when the weak solution flow rate increases the COP_{carnot}
increases. Although the COP is increasing, the rate of increase in COP_{carnot} is
more than the COP Therefore second law efficiency decreases as weak
solution flow rate increases.

![Figure 4.23 Effect of weak solution flow rate over second law efficiency](image)

Figure 4.23 Effect of weak solution flow rate over second law efficiency
When the sink temperature increases while other parameters are constant, the second law efficiency increases. Although both COP and \( \text{COP}_{\text{carnot}} \) decrease, the rate of decrease in \( \text{COP}_{\text{carnot}} \) is more than the COP. Therefore when sink temperature increases second law efficiency increases. Since the actual COP is less than thermodynamic COP the thermodynamic second law efficiency is more than the actual second law efficiency.

Figure 4.24 shows the variation of the second law efficiency with increase in the absorber cooling water flow rate, when the sink temperature varies from 26°C to 33°C and the weak solution flow rate is 0.036 kg/s. From Figure it is seen that at constant absorber cooling water flow rate and constant weak solution flow rate, as sink temperature increases, the second law efficiency increases. And at constant weak solution flow rate and constant sink temperature, as the absorber cooling water flow rate increases, the second law efficiency decreases.

Figure 4.24 Effect of absorber cooling water flow rate over second law efficiency
At constant absorber cooling water flow rate and constant weak solution flow rate, as sink temperature increases, the COP\textsubscript{Carnot} decreases. Although, COP decreases with increase in sink temperature, the rate of decrease of COP\textsubscript{Carnot} is more than decrease of COP. Therefore, the second law efficiency increases.

At constant weak solution flow rate and constant sink temperature, as the absorber cooling water flow rate increases, the COP\textsubscript{Carnot} increases. Since absorber cooling water flow rate seems to be no effect on COP, the second law efficiency decreases with increase in absorber cooling water flow rate.

Figure 4.25 shows the effect of sink temperature over second law efficiency at various weak solution flow rates and constant absorber cooling water flow rate. At constant sink temperature and constant absorber cooling water flow rate as the weak solution flow rate increases the second law efficiency decreases. At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases second law efficiency increases.

At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases the, COP\textsubscript{Carnot} increases more than the COP. Therefore second law efficiency decreases. At constant weak solution flow rate and absorber cooling water flow rate as sink temperature increases, the COP\textsubscript{Carnot} decreases more than the COP. Therefore second law efficiency increases.
4.10.5 Producer gas flow rate

The influence of operating parameters over producer gas flow rate is discussed in this section.

Figure 4.26 shows the variation of the producer gas flow rate with increase in the weak solution flow rate, when the sink temperature is varied and the absorber cooling flow rate is 0.588 kg/s. When the weak solution flow rate increases at constant sink temperature and constant absorber cooling water flow rate the producer gas flow rate increases. When the sink temperature increases, at constant weak solution flow rate and constant absorber cooling water flow rate the producer gas flow rate decreases.
At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the weak solution concentration also increases. Therefore the heat input to be given in the generator to desorb the refrigerant is increased. Therefore the required producer gas flow rate is also increased.

At constant weak solution flow rate and constant absorber cooling water flow rate, when the sink temperature increases, the weak solution concentration from the absorber decreases. Therefore, the required generator load decreases. So that the producer gas flow rate also decreases.

Figure 4.27 shows the variation of the producer gas flow rate with increase in the absorber cooling water flow rate, when the sink temperature is varied and the weak solution flow rate is kept constant. At constant absorber
cooling water flow rate and constant weak solution flow rate, as sink temperature increases, the producer gas flow rate decreases. At constant weak solution flow rate and constant sink temperature, as the absorber cooling water flow rate increases, the producer gas flow rate increases.

At constant absorber cooling water flow rate and constant weak solution flow rate, as sink temperature increases, the refrigerant flow rate decreases. The required generator load to desorb the refrigerant from weak solution in generator also decreases. Therefore, the producer gas flow rate decreases. At constant weak solution flow rate and constant sink temperature, as the absorber cooling water flow rate increases, the weak solution concentration increases. As a result, the refrigerant flow rate also increases. Therefore, the required producer gas flow rate also increases.

Figure 4.27 Effect of absorber cooling water flow rate over gas flow rate
Figure 4.28 shows the effect of sink temperature over producer gas flow rate at constant absorber cooling water flow rate and at various weak solution flow rates. At constant sink temperature and constant absorber cooling water flow rate as the weak solution flow rate increases the producer gas flow rate increases. At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases producer gas flow rate decreases.

**Figure 4.28 Effect of sink temperature over gas flow rate**

At constant sink temperature and constant absorber cooling water flow rate when the weak solution flow rate increases, the mass flow rate of refrigerant to be desorbed from the generator is more. Therefore, the required producer gas flow rate is also more. When the sink temperature increases the refrigerant mass flow rate decreases. Therefore required producer gas flow rate is reduced.
4.10.6 Heat loads

The influence of operating parameters such as weak solution flow rate, absorber cooling water flow rate and sink temperature over heat loads of components are discussed in this section.

4.10.6.1 Effect of weak solution flow rate

In this section the effect of weak solution flow rate over heat loads of vapour absorption refrigeration system, when the sink temperature is varied from 26°C to 33°C and absorber cooling water flow rate is at 0.588 kg/s is discussed.

Generator load

Figure 4.29 shows the variation of the generator load with increase in the weak solution flow rate, at various sink temperature, and at constant absorber cooling water flow rate. At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the generator load also increases. At constant weak solution flow rate and constant absorber cooling water flow rate, when the sink temperature increases the generator load decreases.

At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the concentration of weak solution, which leaves the absorber and enters into the generator increases. The heat input required to desorb the refrigerant vapour from the weak solution in generator also increases. Therefore, the generator load increases with increase in weak solution flow rate. When the sink temperature increases, while the weak solution flow rate and absorber cooling
water flow rate are kept constant, the concentration of weak solution decreases. The required heat input to be supplied to generator decreases. Therefore, the generator load decreases with increase in sink temperature. The thermodynamic generator load is less than actual generator load, due to the heat loss incurred during the experimentation.

![Figure 4.29 Effect of weak solution flow rate over generator load](image)

**Figure 4.29 Effect of weak solution flow rate over generator load**

**Condenser load**

Figure 4.30 shows the variation of condenser load with increase in weak solution flow rate, for different sink temperatures and constant absorber cooling water flow rate. At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the condenser load increases. At constant weak solution flow rate and constant absorber cooling water flow rate, when the sink temperature increases the condenser load decreases.
At constant sink temperature and constant absorber cooling water flow rate, as the weak solution flow rate increases, the refrigerant flow rate also increases. Therefore the condenser load increases. At constant weak solution flow rate and absorber cooling water flow rate, when the sink temperature increases, the weak solution concentration decreases. So the generator pressure increases, causing condenser temperature to increase. Therefore, the refrigerant flow rate is decreased, which in turn decreases the condenser load. The thermodynamic condenser load is more than the actual condenser load as expected.

**Figure 4.30 Effect of weak solution flow rate over condenser load**
Evaporator load

Figure 4.31 shows the variation of the evaporator load with increase in the weak solution flow rate, when the sink temperature is varied and the absorber cooling water flow rate is kept constant. When the weak solution flow rate increases at constant sink temperature and constant absorber cooling water flow rate the evaporator load increases. When the sink temperature increases, at constant weak solution flow rate and constant absorber cooling water flow rate the evaporator load decreases.

![Figure 4.31 Effect of weak solution flow rate over evaporator load](image)

At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the refrigerant flow rate increases. Therefore the evaporator load also increases. When the sink temperature increases, at constant weak solution flow rate and at constant absorber cooling water flow rate, the concentration of weak solution
decreases, causing the refrigerant mass flow rate to decrease. Therefore the evaporator load decreases. The thermodynamic evaporator load is more than the actual evaporator load as expected.

**Absorber load**

Figure 4.32 shows the variation of absorber load with increase in weak solution flow rate, for various sink temperature and constant absorber cooling water flow rate. At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the absorber load also increases. At constant weak solution flow rate and constant absorber cooling water flow rate, when the sink temperature increases the absorber load decreases.

![Figure 4.32 Effect of weak solution flow rate over absorber load](image)

**Figure 4.32 Effect of weak solution flow rate over absorber load**
At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the absorber temperature decreases. The concentration of weak solution increases, causing the absorber load to increase. At constant weak solution flow rate and constant absorber cooling water flow rate, when sink temperature increases, the absorber temperature also increases, causing the weak solution concentration to decrease. Therefore, the absorber load decreases. The actual absorber load is less than thermodynamic absorber load due to heat loss as expected.

**Rectifier load**

Figure 4.33 shows the variation in rectifier load with increase in weak solution flow rate, for various sink temperature and constant absorber cooling water flow rate. When the weak solution flow rate increases at constant sink temperature and constant absorber cooling water flow rate the rectifier load increases. When the sink temperature increases, at constant weak solution flow rate and constant absorber cooling water flow rate the rectifier load decreases.

At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the refrigerant flow rate increases. Therefore the rectifier load also increases. When the sink temperature increases, at constant weak solution flow rate and at constant absorber cooling water flow rate, the refrigerant mass flow rate decreases. Therefore the rectifier load decreases. The thermodynamic heat load is more than the actual heat load as expected.
4.10.6.2 Effect of absorber cooling water flow rate

In this section, the influence of absorber cooling water flow rate over heat loads of vapour absorption system, when the sink temperature is varied from 26°C to 33°C, and the weak solution flow rate is at 0.036kg/s is discussed.

Generator load

Figure 4.34 shows the variation of the generator load with increase in the absorber cooling water flow rate, for various sink temperature and constant weak solution flow rate. When the absorber cooling water flow rate increases at constant sink temperature and constant weak solution flow rate, the generator load increases. And at constant absorber cooling water flow rate and constant weak solution flow rate, as the sink temperature increases the generator load decreases.
At a constant weak solution flow rate and sink temperature, when the absorber cooling water flow rate increases, the weak solution concentration also increases. Therefore the heat input to desorb the refrigerant from the solution mixture in generator is increasing.

![Figure 4.34 Effect of absorber cooling water flow rate over generator load](image)

**Figure 4.34** Effect of absorber cooling water flow rate over generator load

At constant weak solution flow rate and constant absorber cooling water flow rate, when the sink temperature increases, the weak solution concentration decreases. The refrigerant mass flow rate also decreases. Therefore the required heat input decreases. Due to heat loss the thermodynamic generator load is less than the actual generator load as expected.
Condenser load

Figure 4.35 shows the variation of the condenser load with increase in the absorber cooling water flow rate, at various sink temperature and constant weak solution flow rate. When the absorber cooling water flow rate increases at constant sink temperature and constant weak solution flow rate, the condenser load increases. At constant absorber cooling water flow rate and constant weak solution flow rate, as the sink temperature increases the condenser load decreases.

![Diagram showing condenser load variation](image)

**Figure 4.35  Effect of absorber cooling water flow rate over condenser load**

At constant sink temperature and weak solution flow rate, when the absorber cooling water flow rate increases, the weak solution concentration increases. So the refrigerant mass flow rate increases. Since the mass of the refrigerant to be condensed is more the condenser load is increased.
When the sink temperature increases weak solution concentration decreases, which in turn decreases the refrigerant mass flow rate. Therefore the condenser load decreases. Due to heat loss the actual condenser load is less than the thermodynamic condenser load as expected.

**Evaporator load**

Figure 4.36 shows the variation of the evaporator load with increase in the absorber cooling water flow rate, when the sink temperature is varied and the weak solution flow rate is constant. When the absorber cooling water flow rate increases at constant sink temperature and constant weak solution flow rate, the evaporator load increases. At constant absorber cooling water flow rate and constant weak solution flow rate, as the sink temperature increases the evaporator load decreases.

![Figure 4.36 Effect of absorber cooling water flow rate over evaporator load](image)

**Figure 4.36 Effect of absorber cooling water flow rate over evaporator load**
At constant sink temperature and weak solution flow rate, when the absorber cooling water flow rate increases, the weak solution concentration increases. So the refrigerant mass flow rate increases. Since the mass flow rate of the refrigerant is more the evaporator load is increased. When the sink temperature increases, weak solution concentration decreases which in turn decreases the refrigerant mass flow rate. Therefore the evaporator load decreases.

**Absorber load**

Figure 4.37 shows the variation of the absorber load with increase in the absorber cooling water flow rate, at constant weak solution flow rate and various sink temperature. When the absorber cooling water flow rate increases at constant sink temperature and constant weak solution flow rate, the absorber load increases. At constant absorber cooling water flow rate and constant weak solution flow rate, as the sink temperature increases the absorber load decreases.

![Figure 4.37 Effect of absorber cooling water flow rate over absorber load](image)
At constant sink temperature and constant weak solution flow rate, as the absorber cooling water flow rate increases the heat of absorption by the cooling water from the absorber increases. The refrigerant mass flow rate is increased. Therefore the absorber load increases. When the sink temperature increases, the absorber temperature also increases. The weak solution concentration decreases, causing the refrigerant flow rate to decrease. Therefore the absorber load decreases. Due to heat loss the actual absorber load is less than thermodynamic absorber load.

**Rectifier load**

Figure 4.38 shows the variation of the rectifier load with increase in the absorber cooling water flow rate, for various sink temperature and constant weak solution flow rate.

![Figure 4.38](image)

**Figure 4.38** Effect of absorber cooling water flow rate over rectifier load
When the absorber cooling water flow rate increases at constant sink temperature and constant weak solution flow rate, the rectifier load increases. At constant absorber cooling water flow rate and constant weak solution flow rate, as the sink temperature increases the rectifier load decreases.

At constant sink temperature and constant weak solution flow rate, when the absorber cooling water flow rate increases, the saturated vapour aqua ammonia flow rate increases. Therefore the rectifier load increases. As the sink temperature increases, while other parameters are constant, aqua ammonia flow rate decreases. Therefore the rectifier load decreases. Due to heat transfer losses actual rectifier load is less than the thermodynamic rectifier load.

4.10.6.3 Effect of sink temperature

In this section, the effect of sink temperature over the heat loads of absorption system, when weak solution flow rate is varied from 0.013 kg/s to 0.036 kg/s and absorber cooling water flow rate is at 0.588 kg/s is discussed.

Generator load

Figure 4.39 shows the effect of sink temperature over generator load at various weak solution flow rates and constant absorber cooling water flow rate. From Figure it is seen at constant sink temperature and constant absorber cooling water flow rate as the weak solution flow rate increases the generator load increases. At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases the generator load decreases.
At constant sink temperature and constant absorber cooling water flow rate as weak solution flow rate increases, the concentration of weak solution increases. Therefore the heat input to be given in generator to desorb the refrigerant is more. When the sink temperature increases, the weak solution concentration decreases, correspondingly the required generator load also decreases. Since the heat loss is incurred during experimentation, the thermodynamic generator load is less than the actual generator load.

Figure 4.40 shows the effect of sink temperature over condenser load when weak solution flow rate is varied and absorber cooling water flow rate is kept constant. At constant sink temperature and constant absorber cooling water flow rate as the weak solution flow rates increases the condenser load increases. At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases the condenser load decreases.
Figure 4.40 Effect of sink temperature over condenser load

**Condenser load**

At constant sink temperature and absorber cooling water flow rate when the weak solution flow rate increases the weak solution concentration increases. The refrigerant flow rate is also increased. Therefore the condenser load is increased. When the sink temperature increases, due to decrease in the weak solution concentration the condenser temperature increases which in turn decreases the condenser load.

**Evaporator Load**

Figure 4.41 shows the effect of sink temperature over evaporator load at constant absorber cooling water flow rate and at various weak solution flow rates. At constant sink temperature and constant absorber cooling water
flow rate as the weak solution flow rate increases the evaporator load increases.

![Figure 4.41 Effect of sink temperature over evaporator load](image)

At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases the evaporator load decreases. At constant sink temperature and absorber cooling water flow rate when the weak solution flow rate increases the weak solution concentration increases. The refrigerant flow rate is also increased. Therefore the evaporator load is increased. When the sink temperature increases, at constant weak solution flow rate and constant absorber cooling water flow rate due to decrease in the weak solution concentration, the refrigerant flow rate decreases. Therefore, the evaporator load decreases.
**Absorber load**

Figure 4.42 shows the effect of sink temperature over absorber load at various weak solution flow rates and constant absorber cooling water flow rate. At constant sink temperature and constant absorber cooling water flow rate as the weak solution flow rates increases the absorber load increases. At constant weak solution flow rate and constant absorber cooling water flow rate as sink temperature increases the absorber load decreases.

At constant sink temperature and constant absorber cooling water flow rate when the weak solution flow rate increases, the mass flow rate of the refrigerant is more. Therefore more absorber load is enhanced. When the sink temperature increases, the absorption of refrigerant in absorber is decreased. Therefore the absorber load decreases.
Figure 4.43 shows the effect of sink temperature over rectifier load by varying the weak solution flow rate and keeping the absorber cooling water flow rate as constant. At constant sink temperature and constant absorber cooling water flow rate, as the weak solution flow rate increases, the rectifier load increases. At constant weak solution flow rate and constant absorber cooling water flow rate, as sink temperature increases, the rectifier load decreases.

At constant sink temperature and constant absorber cooling water flow rate, when the weak solution flow rate increases, the mass flow rate of the saturated vapour aqua ammonia, which enters the reflux condenser for rectification is more. Therefore the rectifier load increases. When the sink temperature increases, the mass flow rate of saturated vapour aqua ammonia decreases, which in turn decreases the rectifier load.

![Figure 4.43 Effect of sink temperature over rectifier load](image)

**Figure 4.43 Effect of sink temperature over rectifier load**
4.10.6.4 Effect of biomass consumption over cooling load and COP

Figure 4.44 depicts the effect of the biomass consumption in kg/h on the cooling capacity and COP. It is observed that the cooling capacity and COP increases with increase in biomass consumption rate. The increase in biomass consumption rate increases the heat input to the generator, it leads to more generation of refrigerant vapour, for a constant solution flow rate. At the increased refrigerant flow rate, it has the capacity of absorbing more heat. Hence, the cooling capacity increases at higher biomass consumption. The COP is also increasing due to the same fact as discussed above. It is found that the experimental COP of around 0.4 is attained under the conditions analyzed for the fabricated system.
4.10.7 Exergy analysis

The exergy analysis is carried out with the experimental observations, made as per the experimental plan given in section 4.7. The results of the experimental exergy analysis of biomass assisted single stage vapour absorption refrigeration system are discussed here.

Internal irreversibility

In this section influence of operating parameters over internal irreversibility of the system is discussed. Figure 4.45 shows the effect of sink temperature over internal irreversibility of the components, while other parameters are constant. The internal irreversibility increases with increase in sink temperature. When the sink temperature increases, the absorber temperature increases. Therefore the weak solution concentration decreases causing the circulation ratio to increase. The increase in circulation ratio causes increase in strong solution flow rate. The decrease in heat loads and increase in strong solution flow rate cause increase in the internal irreversibility.

Figure 4.46 shows the effect of absorber cooling water flow rate over internal irreversibility, while other parameters are kept constant. From the Figure, it is understood that the internal irreversibility decreases with increase in absorber cooling water flow rate. When absorber cooling water flow rate increases, the cooling water absorbs more heat energy from the absorber. Therefore weak solution concentration increases which in turn decreases circulation ratio. Moreover, the heat loads in the components increase with absorber cooling water flow rate. Therefore, the internal irreversibility decreases in the system components.
Figure 4.45 Effect of sink temperature over internal irreversibility

Figure 4.46 Effect of absorber cooling water flow rate over internal irreversibility
Figure 4.47 shows the effect of weak solution flow rate over internal irreversibility of system components, keeping other parameters as constant. The increase in weak solution flow rate decreases the internal irreversibilities in the components of the system. When the weak solution flow rate increases the heat loads in the components increase. Therefore the internal irreversibility of the system components decrease.

Figure 4.47 Effect of weak solution flow rate over internal irreversibility

Heat transfer irreversibility

In this section the influence of operating parameters over heat transfer irreversibility is discussed. Figure 4.48 shows the effect of sink temperature over heat transfer irreversibility in the absorption system components. From the figure, it is seen that the heat transfer irreversibility decreases with increase in sink temperature in all the components. When the sink temperature increases, the weak solution concentration which leaves
from absorber decreases causing the refrigerant flow rate to decrease. Therefore, the heat load to be given in generator is reduced. As result, the heat loads in other components also decrease causing heat transfer irreversibility to decrease.

**Figure 4.48** Effect of sink temperature over heat transfer irreversibility

Figure 4.49 shows the effect of absorber cooling water flow rate over heat transfer irreversibility in the components of the absorption system. From the figure, it is seen that the heat transfer irreversibility increases with increase in absorber cooling water flow rate except evaporator. When the absorber cooling water flow rate increases, the concentration of weak solution increases. Therefore, the heat load to be given in generator for the desorption of refrigerant increases. The heat loads in other components also increase. Therefore, the heat transfer irreversibility increases with absorber cooling water flow rate.
Figure 4.49  Effect of absorber cooling water flow rate over heat transfer irreversibility

Figure 4.50  Effect of weak solution flow rate over heat transfer irreversibility
Figure 4.50 shows the effect of weak solution flow rate over heat transfer irreversibility in the absorption system components. From the figure, it is depicted when the weak solution flow rate increases, the heat transfer irreversibility increases. When the weak solution flow rate increases, the heat loads increase in all the components as expected. Therefore, the heat transfer irreversibility from the components increase.

The Figure 4.51 shows the effect of sink temperature over exergy change in heat exchanger, solution throttling valve, refrigerant throttling valve and solution pump. When the sink temperature increases, the strong solution flow rate increases, while the weak solution flow rate is kept constant. Therefore the exergy change in heat exchanger, solution valve and solution pump increases. In refrigerant valve, the exergy change decreases, due to low refrigerant flow rate.

Figure 4.51 Effect of sink temperature over exergy change of heat exchanger, solution throttling valve, refrigerant throttling valve and solution pump
The Figure 4.52 shows the effect of absorber cooling water flow rate over exergy change in heat exchanger, solution throttling valve, refrigerant throttling valve and solution pump. From the figure it is known that the exergy change increases with increase in absorber cooling water flow rate in heat exchanger and refrigerant valve. Whereas the exergy change decreases in solution valve and solution pump. When the absorber cooling water flow rate increases, due to high generator heat load, the temperature of the strong solution increases. Therefore, the exergy change in heat exchanger increases. The increase in mass flow rate of refrigerant increases the exergy change in refrigerant valve. The decrease in the mass flow rate of strong solution causes decrease in the exergy change in solution valve. The increase in density of solution at the pump decreases the exergy change in pump.

The Figure 4.53 shows the effect of weak solution flow rate over exergy change in heat exchanger, expansion valves and solution pump. When the weak solution flow rate increases, both the strong solution flow rate and the refrigerant flow rate increase. Therefore the exergy change increases in the above components

**Exergetic coefficient of performance**

Figure 4.54 shows the variation in ECOP with sink temperature at various weak solution flow rates. From Figure, it is explained that ECOP increases with increase in weak solution flow rate. The ECOP decreases with increase in sink temperature. When the weak solution flow rate increases, circulation ratio is decreased. Therefore, ECOP increases. When the sink temperature increases, the absorber temperature increases. As a result, the circulation ratio increases. Therefore, ECOP decreases, with increase in sink temperature.
Figure 4.52  Effect of absorber cooling water flow rate over exergy change of heat exchanger, solution throttling valve, refrigerant throttling valve and solution pump

Figure 4.53  Effect of weak solution flow rate over exergy change of heat exchanger, solution throttling valve, refrigerant throttling valve and solution pump
Figure 4.54 Effect of sink temperature over ECOP

Figure 4.55 shows the variation in ECOP with absorber cooling water flow rate at various weak solution flow rates and constant sink temperature. The ECOP increases with increase in absorber cooling water flow rate and weak solution flow rate. When the absorber cooling water flow rate and weak solution flow rate increases, the weak solution concentration increases. As a result, the circulation ratio decreases. Therefore, ECOP increases, with increase in absorber cooling water flow rate and weak solution flow rate.

Figure 4.56 shows the effect of weak solution flow rates over ECOP at various absorber flow rates. From the figure it is seen that the ECOP increases with increase in weak solution flow rate and absorber cooling flow rate. When the weak solution flow rate increases, more cooling load is obtained in evaporator. Therefore, ECOP also increases. When the absorber
Figure 4.55 Effect of absorber cooling water flow rate over ECOP

Figure 4.56 Effect of weak solution flow rate over ECOP
cooling water flow rate increases, the absorber temperature decreases. The concentration of weak solution increases. Therefore, the circulation ratio decreases. As the circulation ratio decreases, the cooling load is increased. Therefore, when the absorber cooling water flow rate increases, the ECOP increases.

Exergy of producer gas

The exergy of producer gas is calculated using Gibbs theorem of gas mixtures. The sum of latent and sensible energy of producer gas at the given operating condition is the exergy of the producer gas. Figure 4.57 shows the variation in exergy in producer gas with sink temperature at various weak solution flow rates. The exergy in producer gas decreases with increase in sink temperature. The exergy in producer gas increases with increase in weak solution flow rate. When the sink temperature increases, the circulation ratio increases. Therefore, the required heat input to desorb vapour ammonia from the weak solution in generator decreases. Whereas, when the weak solution flow rate increases, the heat input required to separate the vapour ammonia in generator increases. Therefore, the exergy of producer gas increases.

Figure 4.58 shows the variation in exergy in producer gas with absorber cooling water flow rate at various weak solution flow rates. The exergy in producer gas increases with increase in absorber cooling water flow rate and weak solution flow rate. When the absorber cooling water flow rate and weak solution flow rate increase, the circulation ratio decreases. Therefore, the required heat input to desorb vapour ammonia from the weak solution in generator increases. Therefore, the exergy of producer gas increases.
Figure 4.57 Effect of sink temperature over exergy of producer gas

Figure 4.58 Effect of absorber cooling water flow rate over exergy of producer gas
Figure 4.59 shows the effect of weak solution flow rate over exergy in producer gas at various sink temperature. The exergy of producer gas increases with increase in the weak solution flow rate. The exergy of producer gas increases with decrease in sink temperature. When the weak solution flow rate increases the heat input at generator to separate refrigerant vapour is more. Therefore the required exergy of producer gas is also increasing. When the sink temperature decreases the weak solution concentration increases. Therefore the circulation ratio decreases causing exergy of producer gas to increase.

Exergy flow analysis

The Figure 4.60 shows the schematic results of exergy flow analysis, when the operating parameters are such as absorber cooling water flow rate is 0.588 kg/s, sink temperature is 26°C and weak solution flow rate is 0.036 kg/s. In this Figure, the total heat input to the system is given as exergy inflow in percentage. In addition, the exergy inflow is destructed into exergy outflow at various components is also given in percentage. The exergy outflow at
<table>
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<tr>
<th>Component Name</th>
<th>Exergy change</th>
<th>Exergy inflow (%)</th>
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<tr>
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<td>Evaporator</td>
<td>Chilled glycol exergy change</td>
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<td>Solution pump</td>
<td>Pump work</td>
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<td>Total</td>
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<table>
<thead>
<tr>
<th>Component Name</th>
<th>Exergy change</th>
<th>Exergy outflow (%)</th>
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</thead>
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<td>Generator</td>
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<tr>
<td></td>
<td>Internal irreversibility</td>
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<td></td>
<td>Internal irreversibility</td>
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<td></td>
<td>Internal irreversibility</td>
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<td></td>
<td>Cooling water exergy change</td>
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<tr>
<td>Condenser</td>
<td>Heat transfer irreversibility</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Internal irreversibility</td>
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<tr>
<td></td>
<td>Cooling water exergy change</td>
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<tr>
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<td></td>
<td>Cooling water exergy change</td>
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<td>Refrigerant Valve</td>
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<td>Solution Valve</td>
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<tr>
<td>Heat exchanger</td>
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</tr>
<tr>
<td>Total</td>
<td></td>
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</table>

Figure 4.60  Exergy flow diagram
various components has been given in terms of internal irreversibilities, heat transfer irreversibilities and exergy change. The hot gas exergy change, which takes place at generator, is the major proportion of the total exergy inflow to the system. Moreover, the exergy change in glycol and the exergy change at solution pump contribute the total exergy inflow of the system. From the exergy outflow, it is seen that the heat transfer irreversibility is more at generator and evaporator, because the heat supply takes place at these components. If the heat supply at generator is less, then the heat transfer irreversibility would also be less. It is possible, when the system is operated at low circulation ratio. Moreover, the high temperature of the weak solution that enters into generator could also reduce the heat supply at the generator. The heating of solution at generator and the absorption of high temperature solution at absorber are the reasons for high internal irreversibilities of the respective components. By reducing temperature of strong solution that enters into absorber the internal irreversibility could be reduced.

4.11 ECONOMIC ANALYSIS

Figure 4.61 shows the economic analysis of biomass assisted vapour absorption systems in comparison with vapour compression systems. It is seen that a biomass assisted vapour absorption system is more cost effective than a vapour compression system.

The calculations are based on the cost of biomass at Rs 0.75/kg and electricity cost at Rs. 3.5/kWh. The real COP and resource COP are taken into account while estimating the operating cost/h for both the biomass based system and conventional compression based cold storage unit. It can be clearly seen from the analysis that the operating cost for the developed system is around 25-30% lower than the operating cost of conventional plant for the same cooling capacity.
Figure 4.61 The economic analysis between vapour compression system and biomass assisted vapour absorption system

4.12 CONCLUSIONS

In this chapter, design of vapour absorption refrigeration system components, fabrication details, working fluids charging procedure, measurements of parameters, experimental plan and procedure have been presented. The results of the experiments carried out on biomass assisted vapour absorption refrigeration system are discussed. The influence of various parameters such as sink temperature, weak solution flow rate, absorber cooling water flow rate, and producer gas flow rate on performance of the system and economic analysis are discussed in the above sections. The results and discussions of the exergy analysis conducted with experimental system are presented in this chapter. Conclusions drawn from the theoretical and experimental analysis are listed in chapter 5.