

# **Thermodynamic Analysis Of Half To Double Effect Solar Operated Absorption System For Mechanical Engineering Department Seminar Room**

In the fulfillment of the requirement of the degree of

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I hereby declare that the literature survey of the project entitled “**Thermodynamic Analysis Of Half To Double Effect Solar Operated Absorption System For Mechanical Engineering Department Seminar Room**” being submitted by me is a authentic work carried out under the supervision of Prof (Dr.) **R. S. Mishra** , Department of Mechanical Engineering, Production and Industrial engineering, Automobile Engineering , Delhi Technological University, Delhi.

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## ABSTRACT

Thermodynamic analysis of half, single and double effect vapor absorption LiBr-H<sub>2</sub>O solar powered cooling system is carried out on the basis of refrigeration load calculation of a specific building (seminar hall of mechanical department DTU). Computational analysis has done for all systems using mass flow and energy balance equations. The heat transfer and exergy equations have been developed to get exergetic efficiency. The comparison of these cycles has been done on the basis of generator temperature and refrigeration load. The COP, exergetic efficiency, heat transfer rates and exergy destruction rates have also been calculated to understand cycle performance. Cooling load calculation also provides required solar panel area. As expected, COP and exergetic efficiency increases as number of effect increases. In all cycles maximum exergy loss occurs in generator and absorber and increases with generator temperature and evaporator load. Among all configurations single effect is most commonly used because of having moderate temperature operating range and ease in operation.

## ACKNOWLEDGEMENT

Generally, individuals set aims, but more often than not, their conquest are by the efforts of not just one but many determined people. This complete project could be accomplished because of contribution of a number of people. I take it as a privilege to appreciate and acknowledge the efforts of all those who have, directly or indirectly, helped me achieving my aim.

I take great pride in expressing my unfeigned appreciation and gratitude to my supervisor **Prof.(Dr) R.S. Mishra**, Department of Mechanical Engineering, for his invaluable inspiration, guidance and continuous encouragement throughout this project work. His critics and suggestions on my experiments have always guided me towards perfection. This work is simply the reflection of their thoughts, ideas, concepts and above all his efforts. Working under his guidance has been a privilege and an excellent learning experience that I will cherish for a long time.

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# NOMENCLATURE

<b>Symbol</b>	<b>Quantity</b>	<b>Si.unit</b>
$\dot{m}$	Mass flow rate	Kg/sec
X	Solution concentration	Kg(strong sol)/kg(ref)
h	Enthalpy	kJ/kg
Q	Heat Interaction	kJ/sec
W	Work Interaction	kJ/sec
$\dot{E}_D$	Exergy Destruction Rate	kJ/sec
e	Specific Exergy	kJ/kg
T	Temperature	°C
P	Pressure	KPa
s	Specific entropy	kJ/kg
$\eta_{exe}$	Exergetic efficiency	%
COP	Coefficient of performance	
A	Area	met <sup>2</sup>

# CHAPTER 1: INTRODUCTION

## 1.1 INTRODUCTION

World wide in building areas like commercial, residential , the air conditioning emerges as a growing market and its demand also increases day by day. This increased energy demands for air-conditioners are because of so many reasons like the increased thermal loads, increased living standards of the people and occupant comfort demands in commercial fields as well as trends in building architectural, like in popular glass building's building envelope there is an increased ratio of transparent to opaque surfaces. The cooling demand is increasing because of so many factors like globalization and development of many countries in the world which leads to increased working hours and flexible working timings and comfort expectations as well. The global warming is also one of the major reasons for increased cooling demands. Summer air conditioning involves both humidity and temperature control of inside environment and as having used extensively worldwide ,is also one of the major cause of electricity consumption in many parts of the world. This increased electricity demand directly affects the energy sources who generate the electricity and due to the great use of these conventional energy sources like coal, nuclear, fossil fuels they are depleting more quickly. So it is very necessary to imply other means to generate power to drive air conditioners to save the conventional fuels. A big change is needed in the energy structure of the countries. The way to achieve this is to introduce other means of energy techniques like solar, wind, water, geothermal etc in the energy structure system of the countries. These technologies are not only environmentally friendly but also saves conventional fuels .among them solar energy is the most available type of energy on the earth surface and is also a sustainable energy resource. Using solar powered air conditioning system can reduce the fossil fuel consumption and green house gas emission to the atmosphere. This technology is in its initial stage and different type of projects are implemented in this area. In future this can become a technological alternative to conventional technologies.

Vapor absorption systems using water-lithium bromide as working fluid pair are being extensively used for cooling purpose. Its different configuration, single effect and double effect are, in practical used in air conditioning units. However, there are few works relatively available on triple effect setup of absorption cooling system. The main advantage to go for triple effect is that these systems can be able to attend even higher efficiency values than double effect system. The only condition required to run the triple effect system is to operate chiller at elevated range of temperatures. This obviously limits the choices available in materials and working pair. The difference between vapor absorption refrigeration system and vapor compression refrigeration system is that in absorption system the compressor is replaced by the combination of absorber pump and generator. Absorption system has many perks over compression system like it can make effective refrigeration using low grade energy, solar energy but compression system uses high grade mechanical energy. Also the operation is noise free and very smooth as the only moving part in the entire system is a small pump which takes a small amount electricity and requires less and cheap maintenance.

In compression system as the evaporate pressure decreases the cop also get decreased along with it but the absorption system can work on less evaporator pressure and temperature without effecting its cop.

The main aim of any higher effect working cycle is to increase performance of the system with the use of heat source at high temperature. In double effect and triple effect system configuration of cycle is in such a way that the heat that is rejected from the stage at higher temperature is further used as heat input to the lower temperature cycle stage which helps in generating additional refrigeration effect. In double-effect cooling technology, the major design choice is the choice opted for solution circuit connection. The main options available are series flow or parallel flow. However, system with parallel configuration provides higher performance than the system with series working configuration at same operating conditions. The half effect cooling system is used when available heat source temperature is relatively low to run single or multi effect system but has lowest valued cop among all configurations.

Absorption system runs on solar energy may be most commonly in use in near future. They are attractive alternative for the conventional refrigeration systems as they minimize environmental impact and fuel cost, having minimum running expense and maintenance cost with one time investment only. Nowadays solar powered absorption systems are used in air conditioning, food processing, commercial purposes. With change in the configuration of system from half to triple according to the heat source temperature, the performance of the system can be varied.

## 1.2 ABSORPTION SYSTEM

A system that uses a heat source (e.g. solar, waste, geothermal) to provide required cooling is known as absorption system. Solar vapor absorption system is the solar heat operated absorption system which is very much similar to vapor compression refrigeration system.

The basic absorption cycle works in three different phases;

- **EVAPORATION** : In this phase the liquid refrigerant evaporates in the evaporator at a low partial pressure environment by taking heat from the surrounding and turned into gaseous phase.
- **ABSORPTION** : The Gaseous refrigerant is absorbed and dissolved in to another liquid, reducing its partial pressure in evaporator and allowing more liquid to evaporate.
- **REGENERATION** : Then the refrigerant laden liquid is heated causing the refrigerant to vapor out and condense in condenser and rest of solution comes down to absorber through heat exchanger.

The absorption system commonly uses two different type of refrigerant-absorbent combination, ammonia-water and water-lithium bromide. In former the ammonia acts as a refrigerant and water as an absorbent but in later one water acts as a refrigerant and lithium bromide as an absorbent.

The refrigerant liquid in the evaporator changes its phase from liquid to gas by taking heat from the medium which needed to be cold then this gaseous

refrigerant goes to absorber where it is mixed with the absorbent which decreases solution's partial pressure and more refrigerant gets vapor off from the evaporator and comes into the absorber.

A cooling medium is also provided to release heat from the absorber in order to insure maximum mixing of refrigerant into absorber and to maintain low partial pressure at evaporator side and also to remove the heat generated by exothermic reaction as refrigerant vapor is absorbed into the absorbent. Then this liquid solution is goes to generator through a pump where it is get heated from some external heat source like solar, waste heat or geothermal heat. This causes more refrigerant vapors to boiled off. This vapor refrigerant goes to condenser where it changes it's phase from gas to liquid by exchanging heat with some external cooling arrangements, water usually and then through an expansion valve the reduced pressure and temperature liquid refrigerant again introduced into evaporator. The absorber rich solution from generator flows down to the absorber via an expansion valve and meets with the vapor coming from evaporator and thus completing the cycle.

A heat exchanger can be added between the absorber and generator to minimize the heat input to the generator. The basic principle of absorption system is shown in figure 1.1. From the figure it can also be seen that there are two distinctive circuits inside the system. First one is refrigerant circuit from generator to evaporator and other one is absorbent-refrigerant solution circuit from evaporator to generator. Strong (refrigerant rich) solution goes from absorber to generator and weak (refrigerant weak) solution comes down to absorber again through an expansion valve after sending some of the refrigerant to the condenser.

Basic two types of absorption systems are most commonly used in refrigeration and air conditioning units:

- Ammonia-water vapor absorption system.
- Lithium bromide-water vapor absorption system.

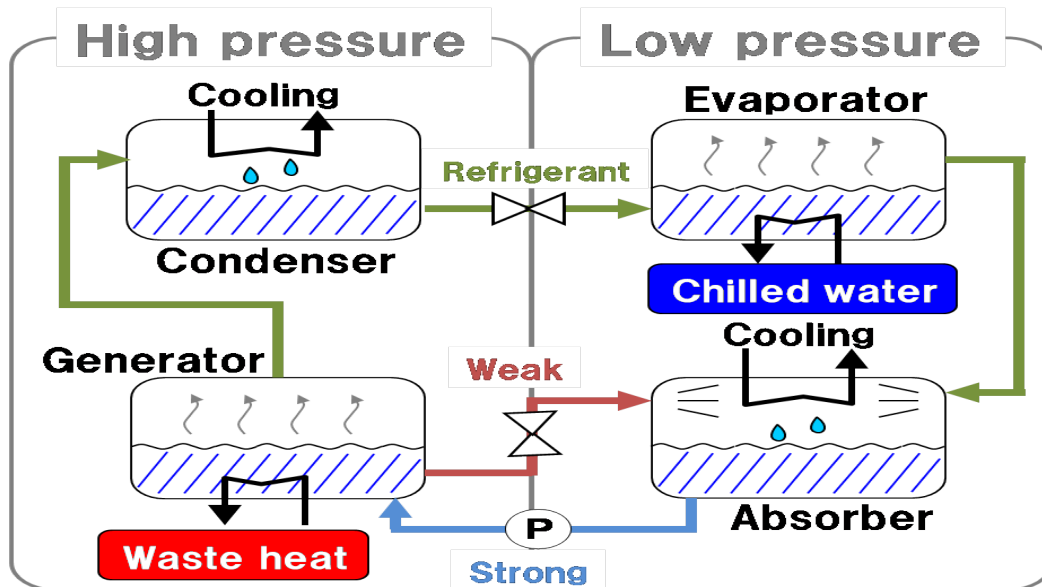


Fig.1.1: Principle of Absorption System

- a. **AMMONIA-WATER ABSORPTION SYSTEM:** It uses water-ammonia pair as the working fluid pair, water acts as an absorbent and ammonia is the refrigerant. This system very efficient as freezing point of ammonia is very low ( $-77^{\circ}\text{C}$ ), therefore, it can provide very low temperature cooling effect. The only disadvantage with this system is that the absorbent and refrigerant both are volatile in nature. The use of rectifier is must to stop the water flowing and accumulating inside evaporator which damages the system as well as reduce system performance.
- b. **LITHIUM BROMIDE-WATER ABSORPTION SYSTEM:** The system uses water as a refrigerant and lithium bromide as absorbent. The system restricted at low temperature cooling as freezing point of water is  $0^{\circ}\text{C}$ . However, it eliminates the corrosion and damage of system parts.

### 1.3 COOLING LOAD CALCULATION

Cooling load can be define as the rate of heat which must be taken out from a specific area in order to maintain the required comforting temperature and moisture inside the area. There are ample number of factors that affects the required cooling and hence



should be included while calculating the cooling load. One has to keep so many factors in mind during calculation like, the outside temperature of air, the outside humidity ratio, the occupancy and activity of people in the room, heat dissipating devices used in the room, type of material the room has made of, the orientation of room to evaluate average solar radiation falling upon it etc. as those parameters are usually confusing and hard to define precisely so they usually interrelated to each other in calculation. For an example the some factors like the outside temperature, room heat radiation rate etc vary in their magnitudes over a broad range during course of time very much like a cyclic manner if we consider them during 24h of period. As uncertainty remains with each factor as it changes over time which causes some discontinuity between these load components. The load components are not usually in the phase with each other. So in order to carry out a cooling load calculation each component must be analyzed regarding to the maximum cooling load for a specific area. The one major problem in the inside load calculation is the thermal storage capacity of the wall, floor and roof material of the building. So we have to take care of these effect of thermal accumulation into the calculating procedure. Therefore in order to calculate the cooling load with utmost precision various models and assumptions are developed and followed.

The results found by this approach are the estimated value. The analysis done is not exact but estimated analysis, as mentioned above so many load parameters are keep changing all the time. Therefore at the specific time of calculation the expected values are generally taken. These approx estimation of values leads to different outcomes for each time we calculate the cooling load as these values are taken at some particular time of the day usually the time when the heat load can be maximum. Therefore noon time is best in order to take parameters for cooling load calculations. A standard must be followed in order to avoid this uncertainty in results. Therefore cooling load calculation mostly uses the data from the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) standards. These parameters act as reference parameters for load calculation worldwide. The data ASHRAE use is based on experimental results and analysis of the collected data from all over the world. It has all the data values based on the latitude, month, countries, which help to easily locate the data. It also has some standard reference for internal cooling load calculations like

lightning, appliances, occupancy wattage values. Like the outdoor parameters, indoor parameters are also unpredictable. Estimated values can only be taken for them. therefore, due to this uncertainty, the cooling load result can not come accurate. There is always difference each time we calculate cooling load of the same area. One way to minimize these uncertainties is to provide calculation some allowance for leftover parameter's effect. This is done by adding some amount of load in the calculation as a factor of safety.

### **1.3.1 Climate/Outdoor data requirements**

In HVAC design outdoor climate has a very important role. It affects the performance of HVAC system. It is also relevant for thermal comfort of the people. The climatically designed HVAC not only improves the comfort level but also makes system work more effectively by reducing the cost of electricity. Therefore we can say HVAC design on the basis of climatic data is more economical. Some key factors of climatically designed HVAC include;

- 1) It consumes less energy, therefore reduces total energy cost of any building.
- 2) Instead of high grade energy, it uses low grade energy.
- 3) It increases comfort level of people.

### **Different climates categories**

- 1) Warm-humid climate – The area under this climate region is 15°south and 15°north from the equator.
- 2) Hot-dry desert – These deserts are spread from 15°to 30° S and N.
- 3) Hot-dry maritime desert – They are in the nearby reasons of hot dry deserts.
- 4) Tropical uplands – these are basically over tropic of Cancer and tropic of Capricorn regions but prevails at some specific height from sea level (900-1200 met)
- 5) Warm-humid Island – The equatorial zones are warm humid island. As on being equatorial sides these places are always warm. Trade wind zones are also warm humid island.
- 6) Composite Monsoon – Composite monsoon climate prevails mainly over the tropic of Cancer and tropic of Capricorn region. New delhi is an example of this composite monsoon climate condition.

## 1.3.2 CLASSIFICATION OF COOLING LOADS

### I. COOLING LOAD SOURCES

Cooling load requirements cause by different parameters. These parameters are the different sources which adds up heat inside the room. Based on type source we can divide Cooling loads in to the following areas:

- 1) Conduction heat transfer in the building from the outside environment. Transfer occurs as soon as the inside temperature drops below the outside temperature.
- 2) There is also radiation heat transfer which is also known as solar heat transfer. This occurs through any opaque surface in the building like glass or any other transparent material.
- 3) Some gaps or voids left in the building structure add up heat by passing outside infiltrate air inside the room.
- 4)The occupancy inside the building, their activities, appliances, lightning also added heat in to the building.

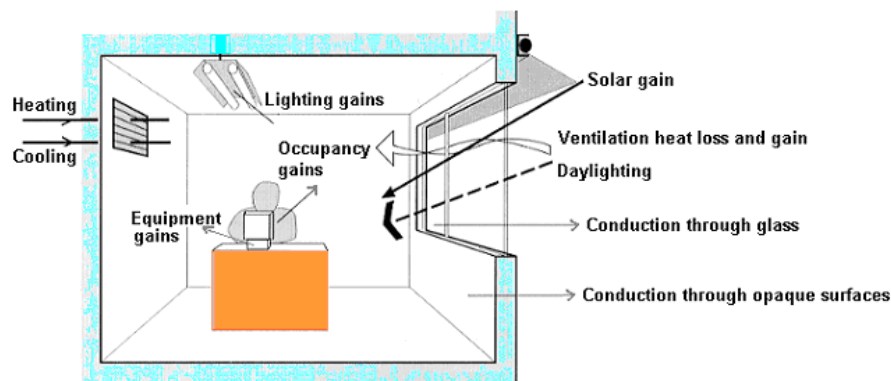


Fig.1.2: Heat load components for a room

### II. HEAT GAIN COMPONENTS

The heat can be gained in any system by two means;

- (1) the sensible heat gain
- (2) the latent or vapor heat gain.

## Sensible heat gain

Also known as sensible load, the sensible heat gain is the one when added to the space, results only in the increase of dry bulb temperature or room's normal temperature. There are many factors that affect this type of heat gain;

- 1) Building envelope gain : The solar heat can be transferred to building envelope like from outside walls of the building, from the roof, ceilings , glazed part of the building, roof, floors etc.
- 2)heat gain due to occupancy inside the space.
- 3) Heat gain due to appliances used in space.
- 4) Heat gain due to partitions in a space that separate areas of two different temperatures.
- 5) Heat gain due to infiltrated and ventilated air which sneaks through some cracks inside the space.
- 6) Heat gain due to the wattage of Lights.

Late

## Latent heat gain

If a heat added to space cause no change of space temperature but changes the humidity inside the space, this type of heat load known as latent heat load or latent heat gain. This is the heat which the water vapor carries. It is also a very major part for heat load accumulation inside the building. It's sources are:

- 1) Ventilated and infiltrated air coming into room from cracks in wall, door or windows contains some moisture in it.
- 2) Occupancy in space, breathing causes moisture addition to the space.
- 3) Cooking appliances.
- 4) Activities like floor washing, cooking, house keeping add vapor to the space.
- 5) Any equipment or machine that uses water.

## TOTAL COOLING LOAD

In the calculation of cooling load both sensible and latent cooling loads are taken under consideration. The total load is the summation of hidden and sensible loads.

### III. COOLING LOAD FROM OUTSIDE AND INSIDE ENVIRONMENT

Any space can be divided into two regions for heat interaction, external or envelope region and internal region. The envelope region deals with the heat flows in external areas outside the space or outside environment or surroundings. The former deals with all the heat transfers inside the space itself which are in the form of sensible and latent heat gains.

#### External heat gain

External heat loads are made up of the following:

- 1) Sensible heat gain (roofs, cracks, walls, ceilings, floors)
- 2) Sensible heat gain from transparent surfaces (glasses, windows, glazed areas)
- 3) Sensible Ventilation loads and infiltration loads
- 4) Latent Ventilation loads and infiltration loads
- 5) Load due to air leakage.

There are always some differences in the calculation of external loads. Various methods are used to calculate these loads:

##### a) *Conduction via building outer envelope*

The equation is used to determine various loads from *Roofs, ceilings, External Walls, Glasses*. These are opaque surfaces.

##### b) *Solar heat gain via Glasses*

This deals with the radiation of solar heat through glasses, plastic surfaces and transparent surfaces.

##### c) *Ceilings, floors and partitions*

Equation determines heat gain through floors and ceilings.

##### d) *Ventilated & Infiltrated Air*

Ventilation air is required inside of the room to manage comfort condition along with the purity of the inside air. This is a part of the outdoor air. It also compensates the indoor air which escapes away from any opening in the room.

#### Internal heat gain

It has the following components:

- 1) Sensible and latent heat gain from occupancy.
- 2) Sensible heat gain due to lights.

- 3) Sensible heat gain from power consuming machinery like motors, fans etc as they dissipate heat inside the space.
- 4) Equipment loads.

Calculation of interior heat gain is completely depends upon one's judgment. Internal loads generally shares more than 50% of the total load. The information about it's weightage and the share in total cooling load is usually very less at the initial stage of calculation. Therefore setoff standard data are use as a reference parameters in order to continue the calculation of loads.

### **1.3.3 HEAT GAIN FOR HVAC SYSTEM**

#### **1.3.3.1 Supply and return Fan Heat Load**

Supply fans and return fans used to supply air into the room or circulate it through out the room. If these fans are located inside the system than the heat dissipated from them directly goes to the room itself. If they are placed outside than there is no heat accumulation of fan's heat inside the room. There are so many ways by which the fan's heat accumulated inside the room:

Sudden rise in the temperature of air if fan's efficiency is very poor.

Temperature rise of air when it comes to static equilibrium. At this time all the static and kinetic energies contain in the air turns into heat. Supply fan and return fan if located inside the conditioned space add heat component to the system but if located outside, they do not involve in any heat load component inside the room.

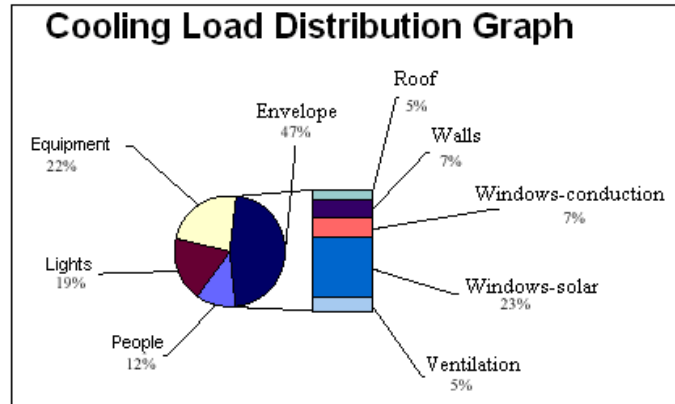


Fig.1.3 Cooling load distribution graph

### 1.3.3.2 SOLAR LOAD FROM WALLS, ROOF, PARTITIONS AND CEILINGS

#### a) Solar Gain from Walls and roofs

In order to ensure the complete effect of heat gain from in to the room some modification is done on the cooling load calculation equation. One new term is added which is known as cooling load temperature difference. CLTD is an empirical value. The use of CLTD in the equation compensates the heat storage capacity of walls and roofs and time lag in heat transfer due to the storage capacity. ASHRAE fundamental handbook has listed number of tables for CLTD values. The CLTD values vary with outside and inside temperature and it's range, the material and construction manner of roof and walls, , geographic condition of the place like latitude and direction of walls, direction of sunlit walls, time of year, month of year, tim of the day etc.

#### b)Solar Gain through Ceilings and Partitions

In cooling load calculation partition can be referred to some area beyond which some non air conditioning space is available. In the calculation any heat interaction into the room from partitions and ceilings is determined.

### 1.3.3.3 HEAT LOAD FROM GLAZED SURFACES (WINDOWS AND GLASSES)

Conduction and transmission two type of heat load occur through windows.Windows also act as a partition for the room as they separate outside environment from the inside conditioned space. But other than transferring heat conductively, they are transparent

too, hence transfer heat through radiation too. The heat transmitted through the windows is very large as compare to conduction heat transfer from them as windows are made up of glasses and glasses are very poor insulator of heat. Therefore the largest portion of heat load inside the room comes from these transmitted surfaces.

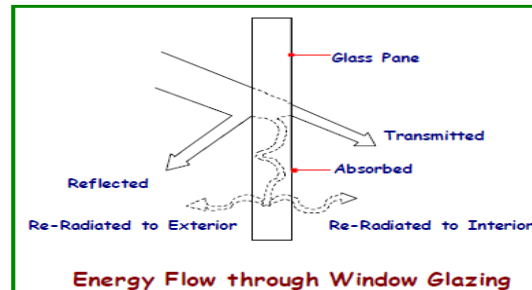


Fig. 1.4: Energy flow through glazing

## HEAT GAIN THROUGH GLAZING

The solar glazing heat gain includes radiation that has been absorbed and the radiation that has been transferred through the glass in to the inside of the room. Therefore two different heat load calculation must be done to find overall heat gain through glazing.

- 1) Conduction heat gain calculations
- 2) Radiation heat gain calculations

### Leakage of inside air

Cracks on the building envelope and windows usually transfer outside air in or insider air out. Due to this transfer heat gain or loss can occur inside the conditioned space. In order to compensate the problem of air leakage in the space ASHRAE has set some standard infiltration rates for air known as air leakage rates. It is equivalent to the one sq foot area through which one cubic feet of air passes. The value of this air leakage rate must be set to the minimum. The lower the value, less the heat load will be. Usually the value of air leakage rate is take 0.1-0.4 (cfm/sqft)



#### **1.3.3.4 HEAT LOAD FROM LIGHTS**

The lights due to their wattage capacity dissipate heat in to the room. One watt is generally taken as 3.14 Btu. Heat gain from lighting depends upon:

- 1) Fixture used.
- 2) Type of HVAC system.
- 3) Different furnishing inside the room.
- 4) System's ventilation rate.
- 5) Type of air conditioning system used.

#### **1.3.3.5- HEAT LOAD FROM APPLIANCES**

There are many different appliances and equipments are used in every area like household, industrial etc. As those power appliances heats up they dissipate heat in to the space where they are kept.

#### **1.3.3. 6- HEAT LOAD FROM PEOPLE**

Two types of heat load the space can incur, sensible and latent heat loads. ASHRAE has given list of some estimated values of sensible and latent heat gain according to the number of occupancy, activities inside the space etc. One new term cooling load factor is included in the calculation which shows the time at which the first person enters the space to the time of the staying of the last person.

#### **1.3.3.7- HEAT LOAD FROM INFILTRATION AND VENTILATION**

The requirement of ventilation air is not constant. It changes as the internal condition changes. For more occupancy, high level people's activity more fresh air will be needed so that people can feel comfort inside the room. Calculations of these loads are, therefore, based on one's judgment and understanding.

# CHAPTER 2 : LITERATURE REVIEW

## 2.1 LITERATURE REVIEW

The study covers the need and importance of solar based cooling system that can play a very prominent role in rarefy energy crisis with the help of solar energy. The study investigated and evaluated the feasibility of an absorption air conditioning unit working under solar power. This chapter presents a comprehensive study of different absorption cooling systems and their working techniques. The performance of absorption refrigeration system can be changed by changing the number of pressure levels. Therefore a proper study has done on different type of absorption systems i.e. half, single and double effect absorption systems and the way some important parameters affect the system performance is investigated. Each of the above mentioned points are extensively reviewed to conclude the optimum results for the absorption refrigeration systems.

A number of experimental and theoretical studies of solar-powered air- conditioning systems have been done in the past.

Atamush Seddiqui and Mohamad Siraj [2013] have done analysis of single and double effect, parallel flow LiBr-H<sub>2</sub>O system. They reduced generator heat input in order to improve performance of the absorption system. They have also improve the refrigerant distribution ratio of single and double effect system.<sup>[1]</sup>

Shylash Kumar and Srenesh Valiyandia [2001] have done simulation and determined the heat transfer and thermodynamic properties for all the main components. The heat loads were calculated and energy and mass balance equations had carried out for each of the component like generator, absorber, evaporator, condenser, heat exchanger and expansion valve. They have come to the result and relate evaporator temperature to the COP stated that COP of the system increases as we increase evaporator temperature. It was also seen that as the temperature of generator gets higher, the increase in COP by increasing the evaporator temperature is very slow. It happens because as we increase the generator temperature the condenser and absorber temperature also get increases along with it. They also stated that COP increases much more drastically if evaporator temperature is low. COP also increases with the Increase of generator

temperature and this happens very fast if temperatures of evaporator is high. The effect passes off as we decrease the temperature of evaporator.<sup>[2]</sup>

S. Singh and Sachin Kaushik [2014] studied absorption system. With the use of energy and exergy equations they derived the limits of performance for the system. The range of COP along with the maximum and minimum possible value was determined. They found out that this range of COP was dependent upon many variables of the system like atmospheric conditions, system condition, refrigerant conditions etc. The main purpose of the study is to design an absorption refrigeration system. The system cooling capacity was set to 5KW. All the steps involved in the analysis were shown in paper which eventually leads to the determination of system's COP.<sup>[3]</sup>

Hameed Safari and Saied Seigh [2011] analyzed both of the configuration of double effect absorption systems i.e. series and parallel. They analyzed both of the systems by energy and exergy equations. A mathematical model is developed for the stimulation of these equations. By stimulation heat load values and performance of the system were determined. These values are compared between the cycles in order to analyze the best behavior of cycle under a certain condition. The given result had shown that between parallel and series the energy performance in parallel is better than in series.<sup>[4]</sup>

Hameed Safari and Saied Seigh [2011] also done the first law analysis of single effect and half effect absorption systems. They have developed a series of codes to determine the values at each point of the system. With some constrained input values rest of the cycle was analyzed with the help of this coding. With the help of mass and energy balance the performance and heat transfer rates in all the components of the system was determined. Comparison has done between half and single effect cycles on the basis of some common parameters. They showed that as the performance is directly linked to the generator temperature and as this temperature increases system performance also increases. This effect is same in half as well as single effect vapor absorption system. They also found out that with the increase of condenser temperature and absorber temperature, the performance of the system decreases but the absorber temperature increase is much more dominant than the condenser temperature increase.

<sup>[5]</sup>

Mukesh Paidha and Lovenish Talwaar have done the energy and exergy analysis of Water- lithium bromide absorption system. the course of study is about 30 days. The heat source used to give heat to generator is steam. With the help of exergy analysis they determined the exergy destruction rate in each of the components of the absorption system and it's second law efficiency as well.<sup>[6]</sup>

N.MEHTA and A.KORIGA, analyzed single effect absorption system. They analyzed the effect of solution concentration on the performance of the single effect system i.e. how performance varies with the variation of solution concentration. For finding cop temperature of condenser varies and the other parameters (temperature of generator, temperature of absorber, capacity of evaporator, temperature of evaporator temperature of absorber) remain unchanged.<sup>[7]</sup>

Mark Merano, Brhene H. Gebrislasi and Diter Boer [2009] conducted first and second law analysis for half, single, double and triple effect Water–Lithium bromide vapor absorption systems. The systems performance along with the second law efficiencies and loss in each component for every system was determined. They analyzed the performance of the system on the basis of heat input temperature and compared this among the different number of effect cycles. They found out that the COP increases with source temperature in all of the system. they have shown that the performance of system increases as the number of effect increases and the same applies for the exergetic efficiency too.<sup>[8]</sup>

F.Sadighpour and B. Borzo studied double effect absorption system and developed a mathematical model for system's analysis. They studied effect of different variables on the water-lithium bromide absorption system and done analysis based on sensitivity of the system.<sup>[9]</sup>

S. Ouasket and R. Fati analyzed the single effect absorption system. the heat input source to the system is solar energy. They relate COP of the system to the temperature of different components of the system. They also used two heat exchangers which are connected in series. First one is located in between the condenser and generator and the second one is in between the generator and evaporator. They calculated value of COP and energy interaction in each of the components of the system and connect it to

the temperatures of the respective components. This analysis compared with an already better absorption system which also uses two heat exchangers.<sup>[10]</sup>

S. Singh, Alkaa Solanki and Yash Paal [2015] have done some case study on the water-lithium bromide cooling absorption system which is solar operated and situated in NSI(Gurgon, Haryan, India). It is a triple effect operated plant. They studied how the seasonal variations effects the performance of the system. The analysis was done during the months of april to july that is particularly the summer season. They found out that the second law efficiency of the system is 0.29 for the system, which occurs in april month but the maximum performance rating or COP of the cooling plant is 1.56 and it is in June. <sup>[11]</sup>

M. M. Abolazm, A.I.Shahanta and A. F. Elsfty [2012] have done first and second law analysis of the single effect and of the double effect systems. They have calculated the performance of the system along with all the heat transfer rates. Then they analyzed the effect of the temperature of various components on the COP and second law efficiency of the systems. Their analysis showed that maximum value of COP and second law efficiency can only be obtained at lower temperature values of absorber and generator for single effect system. This tendency is same in the double effect system too. Out of all the components the found that the absorber is the one that affects the second law efficiency and COP in a greater extent therefore this is the one that requires the maximum attention in design phase. <sup>[12]</sup>

Abhijeet Padhiary and Dillieep Kumar Mohanti [2015] have designed and carried out the analysis of absorption system. the system is single effect and works on solar power. The study of coefficient of performance is done and connection of COP with generator and absorber temperature is obtained. The solution concentration effect on the performance is also determined. For the precision and accuracy of the results they carried out an error analysis as well. <sup>[13]</sup>

V. K. Bajpai [2012] has designed and studied an absorption cooling system. The system uses R-717 refrigerant also known as NH<sub>3</sub>. This is an eco friendly type of refrigerant. The capacity of system analyzed was one ton. The source of heat was taken as hot water. this system was tested in different operating conditions. The system performance was checked and analyzed on the basis of various operating temperatures of the

components of the system such as generator temperature, heat source temperature, absorber temperature, condenser temperature and evaporator temperature. The analysis is done a solar heating panel which was installed on the hostel roofs of an institute.<sup>[14]</sup>

Poonam Rnpariya and Haardik Joshi [2015] have designed a waste heat or solar energy driven absorption refrigeration system.the system capacity was one unit. The refrigerant used to run the system was ammonia or R-717. They calculated the COP of the absorption system at various temperatures of generator and absorber. They found out that the most dominating factors on the COP are generator temperature and effectiveness of the heat exchanger. The required size of the collector was also calculated at optimum COP value. They conducted series of various iterations by changing out the generator's temperature and also the heat exchanger's effectiveness and concluded that the COP of the system increases with the increase of generator temperature. This increment in the value of COP only remains up to some certain temperature value and after that increase in generator temperature decreases COP while COP keeps on increasing with the increase in the heat exchanger effectiveness.

<sup>[15]</sup>

S.C. Kaushik and Akhilesh Arora [2009] carried out, the energy and exergy analysis of a single effect and a series flow, double effect absorption systems. The refrigerant used in the system is water- lithium bromide. A computer based model was developed for the parametric study of the systems. The heat transfer and exergy equations are coded in the form of property equations. These equations are computationally efficient. The analysis involved the determination of how the energetic and exergetic efficiencies are affected by various system component's temperatures, for both of the systems. The effect of the pressure drop inbetween the evaporator and the absorber and heat exchangers effectiveness were investigated. Finally the COP, exergy destructions, efficiency defects in the systems and exergetic efficiencies of the systems were calculated.<sup>[16]</sup>

Riad Hakimi and Robah Gomree [2008] have carried out exergy analysis of double effect lithium bromide and water absorption system. The system has a generator which introduced as a second effect in between the condenser and the generator of a single

effect lithium bromide-water absorption system. The single effect system also includes two heat exchangers in between absorber and also has two generators. Computer stimulation of the system was done by using some set of new computationally efficient formulas. These coding were based on the thermodynamic properties of the lithium bromide-water absorption system. The analysis of exergy was also carried out for various components of the absorption system. All exergetic losses that can exist in double effect absorption system are calculated. The amount of exergy in all the components of the double effect system was also calculated along with the COP and exergetic efficiency.<sup>[17]</sup>

Muhammad Imran [2013] has done second law analysis of a double effect absorption system. the system was a direct fired system using lithium bromide-water as refrigerant solution. The arrangement of the system was parallel reverse flow. System analysis performed on the basis of four various operating conditions of the system chiller. Specific enthalpies, mass flow rate, energy balance, specific entropies and the irreversibility changes of the system were calculated. The calculation was done for all of the components of system. The results showed that the irreversibility changes for the evaporator, condenser and the heat exchangers were very less than the irreversibility changes of generator and absorber. Results also showed that system COP increases with the increase of high temperature generator's temperature. The change in irreversibility also increases with the increase in generator temperature therefore, reducing the amount of available energy. Economic optimization of lithium bromide-water absorption chiller was done with the help of these results. Thus the results play a very prominent role in improving the absorption system. It was also observed that the analysis of the absorption system by using this irreversibility method could help improve the potential performance thermodynamically, that is impossible to achieve from numeric techniques.<sup>[18]</sup>

Abbaas Alpaslan and Murat Ozturk have studied and developed a double effect water-lithium bromide absorption system and also compared effects of some operating parameters on the performance of the absorption system, in order to achieve improved potential. The schematic model of double effect system was designed to achieve this purpose. The determination of second law efficiencies and exergy loss was also done

for all the system components. This further used to analyze the system based on different configurations of the same system with different operating conditions.<sup>[19]</sup>

Kenan Saka, Omer Kaynakli and Farukh Kaynakli [2016] have performed the energy and the exergy analysis of a double effect absorption system. The system was of series flow having water-lithium bromide as working pair of fluid. This system runs on different heat sources like hot air, hot water and steam in generator operating in high pressure level (HPG). The exergy destruction rates in HPG, coefficient of performance and mass flow rates of various heat sources were calculated. A study was conducted in order to investigate the effect of heat capacity. The comparative analysis was also carried out among different sources of heat to the generator based on exergy destruction and the mass flow rates of the heat source. From the analysis, it was observed that at the higher temperature of heat sources, absorber and condenser exergy destruction in HPG increases. Same effect was observed with the lower temperature in LPG, HPG and evaporator.<sup>[20]</sup>

Recep Yamankaradeniz and Omer Kaynakli have performed the energy and exergy analysis of a single-stage operated absorption cycle having water-lithium bromide as working pair of fluid. They calculated the thermodynamic properties for each point of the cycle with the help of equations of states. The heat transfer rates of each of the components for the cycle and performance parameters of the cycle were obtained from first law analysis while with the help of second law analysis they calculated entropy generation for every component of the system. The total amount of entropy generation of the system was also determined. Performance variations and the total entropy generation were analyzed with different operating conditions. Results explained that the high performance values (COPs) can be obtained only at high generator temperature values and high evaporator temperatures values. Same results can be found if we operate the system at low condenser temperatures and low absorber temperatures. They also showed that the total entropy generation decreases with the increase of generator temperature. The maximum value of entropy generation was occurred in generator when it was operated at different operating conditions. However, the entropy generation in heat exchanger, solution pump and expansion valve was very small.<sup>[21]</sup>



A. Huicochea, G. Gutiérrez-Urueta and W. Rivera [2011] have done energy and exergy analysis of a single effect water-lithium bromide absorption system. This work was for the heating and cooling applications. Analyzed parameters were coefficient of performance (COP) and the exergy efficiency. The operating temperatures influence on these parameters was observed. Individual component analysis was also done. In the case of exergy efficiency of generator and absorber, the effect was most noticeable. That parameter increases with the increase in absorber temperature. However, opposite effect was observed as generator temperature increases. Obtained results were allowed the parameter's identification which influence the exergetic efficiency of the absorption system. Among all of the system's components, the absorber had lowest value of exergetic efficiency therefore, was the first candidate for optimization. It was seen that for absorber, the circulation ratio was emerged as new parameter also the heat exchanger was susceptible for optimization.<sup>[22]</sup>

Kamal A. Yakut, Arzu Şencan and Soteris Kalogirou [2005] have done some exergy analysis in a single effect absorption system working with water- lithium bromide pair. System was used for cooling as well as for heating applications. Exergy loss, entropy, enthalpy, temperature, heat rates and mass flow rates in each system component were evaluated. From the obtained results, it can be seen that the exergy loss and heat load in condenser and evaporator were very less than the losses in absorber and generator. This happened because of mixing heat in the solutions, which was not the case of pure fluids. A program for simulation was also written and was used for determination of coefficient of performance of the system (COP) and the exergetic efficiency of the system at various operating conditions. It was obtained that the heating and cooling COP of system increase slowly with source temperature increase. However, his increase with the exergy efficiency of system decreases and it happened for both heating and cooling applications.<sup>[23]</sup>

M. Frihat, H. Altahaineh and M. Alrashdan [2013] have carried out thermodynamic analysis of a single effect water-lithium bromide absorption system having capacity of 1 kW and 10 kW. This analysis consist of system analysis through the first law as well as second law of thermodynamics. The exergy coefficient of performance, system

coefficient of performance and exergetic losses in each system component were obtained at different cooling capacities and operating conditions. [24]

M. Ameri and T. Avanesian [2014] have done analysis and comparison of different type of water cooled absorption systems working under various operating conditions and climatic conditions. They also probed the exergy analysis of the system by considering effect of chemical exergy on the LiBr–water circuit solution. Results have shown that the EUF values increase with the generator and the evaporator temperature increase and also the values decrease with ambient if we increase the relative humidity and temperature of ambient air. The total exergetic efficiency always increases with increase of the ambient air temperature or generator temperature and it decreases with the increase of evaporator temperature and changes very slightly with the relative humidity. [25]

B. Agnew and M.M. Talibi [2000] have carried out second law analysis of a single-effect absorption cycle with lithium-bromide-water as working fluid pair. The numeric results from the cycle were collected and tabulated. Specific design procedure was applied to lithium bromide-water absorption system and an optimization procedure determining the entropy, enthalpy, mass flow rates, temperature, heat rates in each of the components with the coefficient of performance calculations was performed. [26]

M. A. Algoul, J. M. Abdulatef, A. Zaharim, K. Sopian and Ranj Sirwan have carried out the thermodynamic analysis and exergy analysis of a single effect absorption cooling system. the system was solar power driven and was operating with the pair of Lithium bromide-water fluids. MatLab based programming was developed in order to determine the thermodynamic properties of working fluid and the performance results under varying operating conditions. [27]

## **2.2 CONCLUSION FROM LITERATURE REVIEW**

Several analysis have been done on vapor absorption system and it's different effects. From all the studies it is very clear that COP increase as the number of effect increases. However this increment in COP is not directly related to the increment in the effect. As number of effect increase, the increment in COP is not as high as it is for single effect. Also higher number of effect needed more components to run on. Therefore, the

commercial use of absorption cycle is limited to double effect only though triple or quadruple effect has more COP. There analysis can only be done on paper and efforts are applying to run them practically too. It can also be seen from the studies that the performance and exergy loss rates of the system is widely influenced by generator and absorber temperature and increase as these temperature increases. Choice of the effect also depends on the type of heat source. Heat source type ensures the available temperature to run the system. Half effect works on very low heat input with very low energy performance while double effect works on high heat input and performance of double effect is also high.

### **2.3 LITERATURE GAP**

In literature reviews the studies and analysis have been done on lithium bromide- water vapor absorption system. The coefficient of performance of the system and exergy destruction rates of system components have been determined with the help of energy and exergy equations. The cooling capacity of the system is taken directly in these calculations. Whereas in other papers the cooling load is calculated separately and run only in one system to optimize it's parameters. The analysis can also be done by combining the load calculation of a specific area followed by the designing and comparison of different configuration of vapor absorption system based on the load calculated. It can help to give us a clear idea of how various systems and there parameters vary at same cooling load requirement.

### **2.4 OBJECTIVES**

The main of cooling load calculation is to calculate the cooling load required for mechanical engineering department's seminar room and use this load to design and analysis of a solar operated vapor air conditioning system based on half, single and double effect vapor absorption water-LiBr system and to carry out first and second law analysis of the systems using refrigeration capacity required for the seminar room.

To achieve this aim, the project objectives are set out below.

- Study of solar based technology advancement and it's advantage.

- Study of theoretical and experimental research work on absorption refrigeration cycles.
- Study of the effect of various parameters on the system performance.
- Study of cooling load calculation methodology.
- Done load calculations by considering seminar hall datas, outdoor, indoor, weather conditions by using ASHRAE Fundamental Handbook.
- Done the theoretical analysis for half, single, double effect Water-LiBr absorption system to obtain the optimum results.

## **CHAPTER 3: METHEDODOLOGY**

### 3.1 COOLING LOAD CALCULATIONS

In order to calculate cooling load ASHRAE has provide three distinct methods in ASHRAE Fundamentals. All methods yield almost the same results with minor differences. For a complete calculation of load zones and entire building loads any one of these three methods can be applied. The methods are following:

- I. **Transfer Function Method or TFM:** This method is considered the most complex method out of all of the methods that are proposed by ASHRAE, therefore requires some high level advanced computational programs use and advanced level of spread sheets.
- II. **Cooling Load Temperature Differential method or CLTD:** CLTD method is also called cooling load factor (CLF) and solar cooling load factor (SCL) method. This is the most used method in cooling load methodology. Derived from TFM method, this method presents the load data in much simpler form. It uses data in tabulated form in order to simplify the load calculation process. Due to it's simplicity, CLTD method can easily be transferred into fairly simple spreadsheet programs but using tabulated data sets some limitations in the calculation procedure.
- III. **Total Equivalent Temperature Differential method or TETD:** Also known as time averaging computational method (TA), this was the method used before the introduction of CLF/CLTD methods. In this method simple hand based spreadsheet calculations were used to done for cooling load calculation.

All these three methods are very well acknowledged in ASHRAE Handbook Fundamentals.

The CLTD/SCL/CLF method is known as a very reasonable and precise approximation of total heat gain from a building envelope for the sizing purpose of HVAC units. This method, therefore, can be used as the best alternative method to some difficult and cumbersome calculation methods like TFM or TETD methods also the errors in calculations by using CLTD/SCL/CLF method tend to be very less than other methods. Due to all these advantages the CLTD method is used in our calculation to determine cooling load required for mechanical engineering department seminar room (DTU).

## III.2 LITHIUM BROMIDE-WATER ABSORPTION SYSTEM

This system is a type of absorption system that uses lithium bromide-water as a working fluid pair. The water acts as a refrigerant and lithium bromide acts as an absorbent. The system configuration can be changed in many ways to attain desired performance. On major classification of the vapor absorption system is based upon the number of pressure levels or effect in the system such as single, double, triple, half etc. As this number of effect increases, system performance also increases and heat input to the system decreases. In our study the following three type of system is analyzed:

### III.2.1 SINGLE EFFECT VAPOR ABSORPTION SYSTEM

A single-effect absorption refrigeration system is the most commonly used vapor absorption system. It has a very simple design with less number of component than other configurations. It has different design configurations based on what working fluid is used. Fig. 3.1 shows a single-effect vapor absorption system.

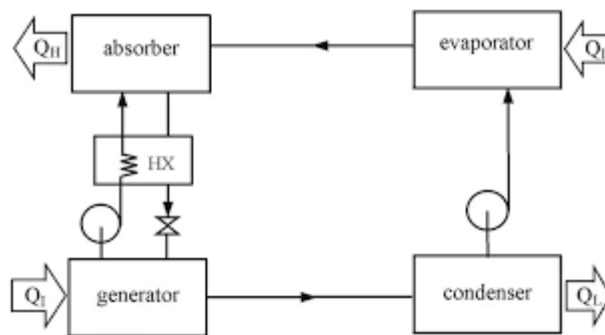


Fig.3.1: Single-effect vapor absorption system

The working of single effect vapor absorption system is same as mention above as this is a very basic system having minimal number of components. Other effects are slight variations of this single effect system. As the high temperature refrigerant goes to low temperature condenser or rejects heat at the condenser

and also rejects heat to the absorber. This causes an irreversibility in high temperature generator heat as it is wasted out at the condenser surrounding and the absorber surrounding. So a solution heat exchange is introduced in between the generator and absorber to reduce this irreversibility. The heat exchanger heats up the solution from the absorber as it is going to generator by using the heat from the hot solution coming to absorber from the generator. As heat given to generator is reduced cop increase with it. We can also reduced the size of the absorber as heat rejected from the absorber is less now. This decreases overall system size and system can be compact. Studies shows that by using solution heat exchanger cop of the single effect system can be increased upto 60%. We can secure the operation by using one more element, rectifier. Rectifier is used only when volatile absorber such as water is used in the system. It is placed right after the generator and it purifies the refrigerant entering to condenser. As of being highly volatile water vapors off and mixes with ammonia vapor. when this mixture goes to evaporator the water starts accumulating inside it and causing the performance to drop. So the rectifier condenses water vapor by introducing large surface area to the solution, water as of being much volatile condensed out and goes to the absorber along with the weak solution and the absorbent free refrigerant (ammonia) goes directly to the condenser.

### **III.2.2 DOUBLE EFFECT ABSORPTION SYSTEM**

The double effect absorption system has two generators and two condensers arranged in different combinations i.e. series, parallel etc. It can be seen as a combination of two single effect absorption system. The term double effect implies that the heat input given to the high temperature generator is used twice in the single cycle to allow more vapor to boil off from the solution. Hence the system having higher values of cop as compare to the single effect absorption system. The vapor that is boils off in the high temperature generator goes to the high temperature condenser at the same pressure. During the condensation in high temperature condenser the vapor rejects heat which is at considerably high temperature. This high temperature heat is further utilized in low temperature

generator as a heat input to it and thus using the single heat input given at high temperature generator twice.

The above mention configuration is known as series flow double effect absorption system hence the double effect absorption system is nothing but the combination of two single effect absorption systems. Fig. 3.2 here has just shown the arrangement of double effect system(series flow).

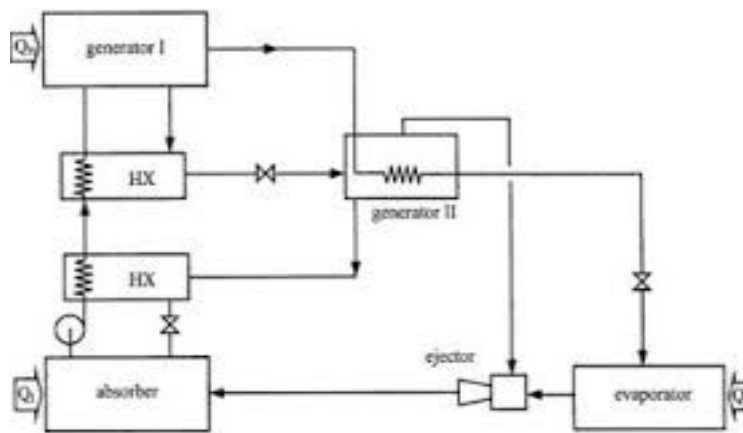


Fig.3.2: Double effect vapor absorption system

Double effect is a three pressure level machine. The absorber and evaporator work on evaporator pressure level, the low temperature generator and condenser work on condenser pressure level and the high temperature generator and the high temperature condenser work on high temperature generator level. There is an internal heat transfer in system is also occurred from high temperature condenser to low temperature generator.

The parallel arrangement of double effect system in which the system is connected differently. The solution is first go through low temperature generator to high temperature generator but in the series flow the solution is first send to high temperature generator than to low temperature generator. Also in parallel flow configuration the mass flow rate or solution mass fraction is also divided to two parallel streams low s it is flowing from absorber to low temperature generator and then from low temperature generator to high temperature



generator. However the energy and exergy efficiency find out greater in parallel configuration than in series.

### III.2.3 HALF EFFECT ABSORPTION SYSTEM

The half effect vapor absorption cycle can only be used when the temperature of the available heat source is less than the minimum necessary to drive a single-effect cycle. This cycle having the maximum number of components out of all the configurations of absorption system, is a three pressure level machine.

Like double effect, the low temperature absorber and evaporator work on evaporator pressure level, the low temperature generator and high temperature absorber work on low temperature generator pressure level and the high temperature generator and the condenser work on high temperature generator or condenser pressure level. The heat is given to the cycle at two places, at high temperature generator and at low temperate generator. There is no internal heat transfer phenomenon as in case of double effect absorption system. Hence the half effect system requires larger heat input than the rest of effects. Which result as the minimum cop of the system. The low cop in half effect absorption system is also due to the fact that it rejects much more amount of heat as compare to any other way configured absorption system. The rejected heat can be as high as 50% of the total heat supplied in the cycle. Hence system is not efficient enough to get commercial use out of it. The high and the low pressure levels of the half effect are same as of single effect. The intermediate pressure level is the new feature in half effect absorption system. At the intermediate pressure level, the refrigerant vapor is delivered from low temperature generator to the high temperature absorber. The absorption system can only be effectively operated when solution refrigerant contain in absorber is richer than it is in the generator. But as we increase the temperature or decrease the pressure of the solution there is less mixing of refrigerant into absorbent hence the solution becomes much weaker or the refrigerant contain in the solution get reduced.

Likewise if we reduce generator temperature to achieve the required cooling effect now we need more solution to run through the system which causes the

increase of solution circulation rate and decrease in cop. On the other hand the system would stop working under very low solution circulation rates. The main purpose to use the half-effect absorption system was to operate an absorption system with a relatively low-temperature values heat input. In half effect system the heat at very elevated temperature is transferred to both of the generators by means of some external heat source. The source can be of any low grade energy source like solar energy, waste or geothermal heat. The condenser and Both high pressure and low pressure absorbers reject heat out in the surroundings by using some heat dissipating medium (water). the circulation rate between low pressure generator and low pressure absorber and between high pressure generator and high pressure absorber is maintained at predetermined level as high pressure absorber and low pressure generator are at an intermediate pressure level. Despite of having a very low value of cop, the system can be operated in areas where relatively very low temperature heat sources are available that can not use to drive single effect systems. The diagram of a half effect absorption system is shown in Fig. 3.3.

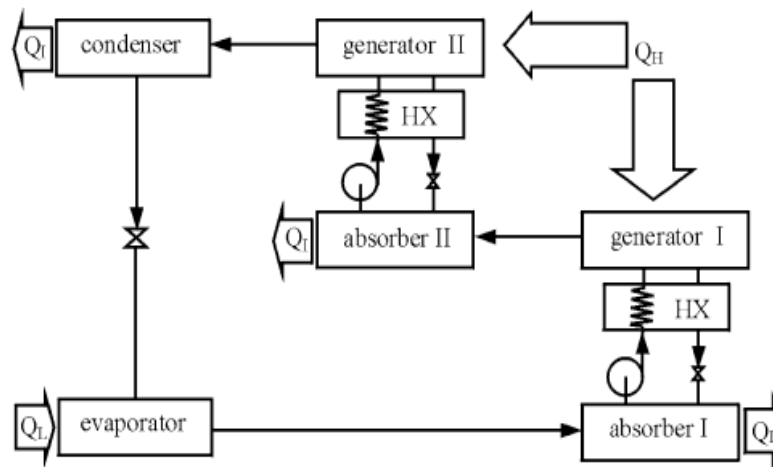


Fig.3.3: Half effect vapor absorption system

## CHAPTER 4 : COOLING LOAD CALCULATION

### 4.1 COOLING LOAD CALCULATION

**External loads:**

**a) Heat load from opaque surfaces:** The heat load from various opaque surfaces like walls, doors, roof etc.

$$Q_{\text{opaque}} = U.A.CLTD$$

U = overall heat transfer coefficient

A = area for heat transfer of the surface

CLTD = cooling load temperature difference.

**For sunlit surface:** CLTD is obtained from the CLTD table. Adjustment needed in the values which are obtained by ASHRAE Handbook if the real conditions are different from the ones according to which CLTD tables are prepared.

**For non sunlit surface:** the CLTD values are directly equal to the temperature difference across the desired areas.

$$CLTD = T_{\text{out}} - T_{\text{in}}$$

$T_{\text{out}}$  = outdoor dry bulb temperature

$T_{\text{in}}$  = indoor dry bulb temperature

**FOR FLOOR:** The CLTD values depend on the location of the space and also given based on different times of the day as these values vary with time.

**b) Heat load from fenestration:** it deals with the heat transfer rates from transparent surfaces like glasses, windows etc. It includes:

1. Conduction heat transfer occurs due to difference of temperature across surfaces:

$$Q_{\text{cond,fs}} = U.A.CLTD$$

2. heat load due to radiation through the windows:

$$Q_{\text{rad,fs}} = A.SC.SCL$$

SC = Shading Coefficient, from ASHRAE tables depend upon the orientation, direction of the windows, month, location of the building also on the glass type and internally applied shading devices.

SCL = solar cooling load factor also known as cooling load factor (CLF), as all radiation energy enter in to the space at a particular duration of time do not become the part of the heat load instantly.

Cooling load factor values for different surfaces can be seen as a function of solar time and orientation of the surface. The values are available in the table form and can be obtained from ASHRAE Handbooks.

**c) Heat load through infiltration:** Infiltration heat transfer has both the sensible as well as the latent heat components.

**Sensible load:**

$$Q_{s,inf} = 1.10. \text{ cfm}_{inf} \cdot (T_{out} - T_{in})$$

**Latent load:**

$$Q_{l,inf} = 4840. \text{ cfm}_{inf} \cdot (w_{out} - w_{in})$$

**Total load:**

$$Q_{tot,inf} = 4.5. \text{ cfm}_{inf} \cdot (h_{out} - h_{in})$$

$\text{cfm}_{inf}$  = infiltration cfm from ASHRAE standard

**d) Heat load due to ventilation:** Like infiltration, ventilation heat transfer also has both the sensible and the latent heat components.

**Sensible load:**

$$Q_{s,ven} = 1.10. \text{ cfm}_{ven} \cdot (T_{out} - T_{in})$$

**Latent load:**

$$Q_{l,ven} = 4840. \text{ Cfm}_{ven} \cdot (w_{out} - w_{in})$$

**Total load:**

$$Q_{tot,ven} = 4.5. \text{ cfm}_{ven} \cdot (h_{out} - h_{in})$$

$\text{Cfm}_{ven}$  = ventilation cfm from ASHRAE standard 62

**Estimation of internal loads:**

**a) Load due to occupants:** The internal cooling load caused by the occupants has two components of loads, sensible cooling load and latent cooling load. Amount of load greatly depends upon the activities people are doing inside the space and the number of occupants inside the space. There is one more component in this process, the radiant heat load as every body at any temperature (other than 0K) radiates heat in a certain amount. Therefore, a Cooling Load Factor (CLF) needed to be used in the load calculation similar to that one used in the case of fenestration heat transfer.

**Sensible load:**

$$Q_{s,p} = N.(SHG).CLF_s$$

**Latent load:**

$$Q_{l,p} = N.(LHG).CLF_l$$

N = number of people

SHG = sensible heat gain /person

LHG = latent heat gain /person

CLF<sub>s</sub>= sensible cooling load factor, based on the number of hours from the entry of the first person into the space to the last person leaving the space or the total hours spent by the occupants inside the space and type, material of the building. CLF values are obtained for different type of buildings and are tabulated in ASHRAE fundamental handbook.

CLF<sub>l</sub>= latent cooling load factor, for instantaneous heat gain CLF value is taken as 1.

**b) Lighting Loads:** Lightings add only sensible heat load to the space.

$$Q_{s,light} = 3.41.W.F_{uf}.F_{sa}.. CLF$$

W = wattage installed

F<sub>uf</sub> = use factor, account for any of the lamps that are installed but not switched on at the time when the load calculations are performed.

= 1.00 (fluorescent lights)

F<sub>sa</sub> = special allowance factor, takes into account of the load imposed by ballasts

= 1.25 (fluorescent lights)

CLF= cooling load factor for lightning, it depends on the number of hours after lights are turned on, type of fixtures used for lightning and the hours of operation the lights are used. These values are available in the form of tables in ASHRAE handbooks.

**c) Internal loads due to appliances and equipment:** The equipment and appliances used in the space may add both the sensible as well as the latent heat loads to the space. Like in the case of fenestration heat transfer, the sensible load can be in the form of only convection or of both, radiation and convection.

**Sensible load:**

$$Q_{s,app} = W \cdot F_U \cdot CLF$$

W = wattage installed

$F_{Uf}$  = use factor, based on the type of appliance or equipment.

CLF= cooling load factor for appliances, which are available in the form of tables in . ASHARE handbooks

**Latent load:**

$$Q_{l,app} = W \cdot (\text{latent heat fraction})$$

## 4.2 SYSTEM DISCRIPTION

Specification of cooling space is the most important part of cooling load calculations. The table 4.1 shows dimension and directions of all the component of the seminar room. The room structure is further illustrated by making room diagrams from different angles in diagrams.

The outside of the room from different directions is shown in figures below (Fig. 4.1-4.4). It shows the orientation, shape and size of the windows and doors of the room. Figures contain different angles from top view and front view of the room.

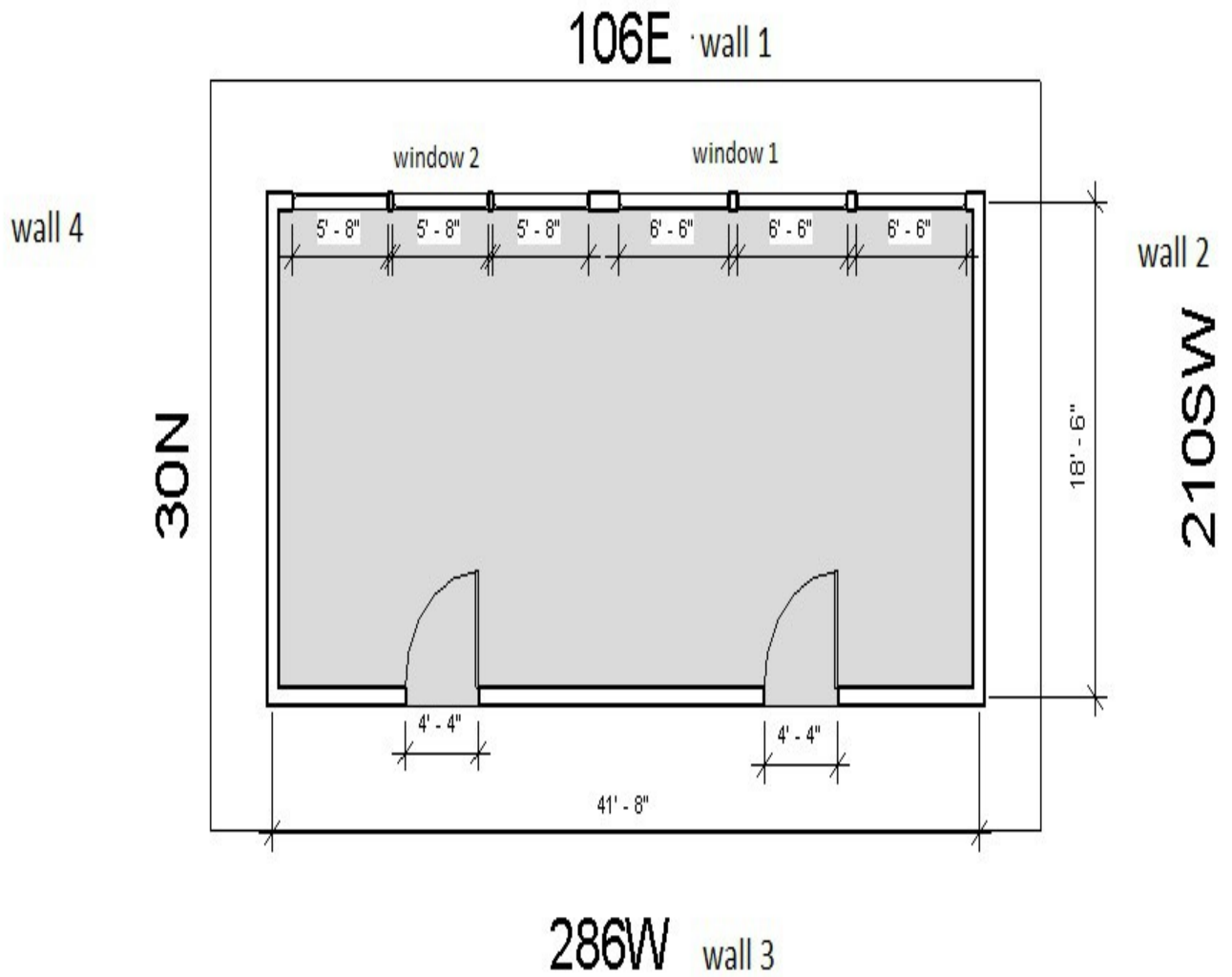


Figure 4.1: Schematic diagram of mechanical engineering department seminar room (DTU)

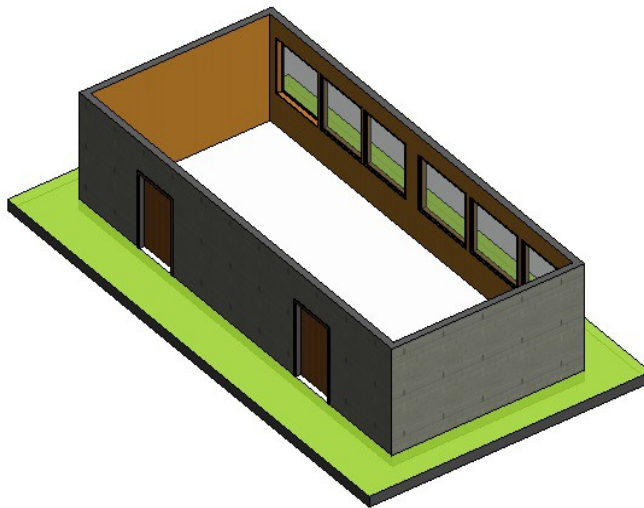
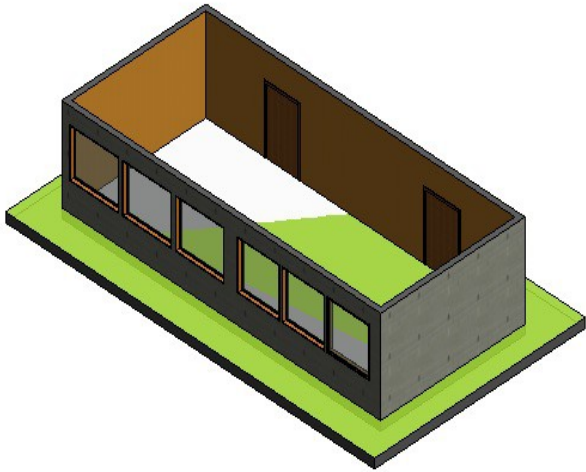
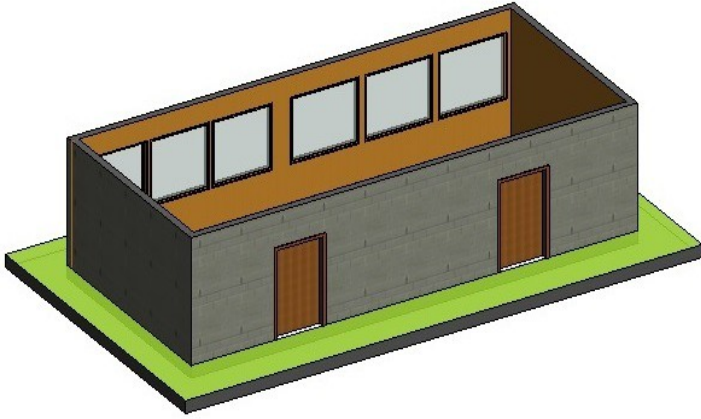


Fig 4.2: Top view of seminar room (DTU)

Fig 4.3: Top view of seminar room (DTU)





seminar room

Fig 4.4: Top view of seminar room (DTU)



Fig 4.5:Side view of  
(DTU)

The location, dimension and orientation of different walls , windows etc are shown below in the table 4.1.

TABLE 4.1: Dimensions of seminar room, Mechanical Engineering Department (DTU)

COMPONENT		LENGTH(met)	WIDTH(met)	HEIGHT(met)	DIRECTIO N
ROOM		12.7	5.64	2.64	—
WALL 1		12.7	—	2.64	106° E
WALL 2		—	5.64	2.64	210° SW
WALL 3		12.7	—	2.64	286° W
WALL 4		—	5.64	2.64	30° N
DOOR 1		1.32	—	2.13	286° W
DOOR 2		1.32	—	2.13	286° W
WINDOW (×3)	1	1.98	—	1.6	106° E
WINDOW (×3)	2	1.73	—	1.6	106° E

The seminar room has one sunlit wall (wall 1) and three non sunlit walls and roof. Out of which one is in direct contact of outside ambient and other two are in contact with the other rooms. It has six windows on the sunlit wall side and two doors at the wall number 3.

In order to calculate heat transfer through walls we need the values of overall heat transfer coefficient of all the walls, roof and floor, which, in turn depends upon the material construction of the walls, room and floor. The table 4.2 below shows the component materials.

TABLE 4.2: Component materials of seminar room, Mechanical Engineering Department (DTU)

COMPONENTS	ROOM SIDE	MIDDLE LAYER	OTHERSIDE
FOUR WALLS	1in plywood	4in face brick	$\frac{3}{4}$ in plaster
FLOOR	Clay tiles	—	12in heavy weight concrete block
ROOF	Clay tiles	—	8in light weight concrete
GLASS	$\frac{1}{2}$ in single plate (heat absorbing)		

The external loads depend upon the structure materials, windows or fenestration areas , infiltrated and ventilated air quantity and location of the cooling space. But in turn to

calculate the internal load one should take care of all the possible internal heat sources i.e appliances, equipments, people occupy the space, lightning loads etc. The load can be of any form, sensible heat load or latent heat load or both.

The internal load members are shown in table 4.3.

ITEMS	NUMBER
LAPTOPS	2
FANS	8
LIGHTS	18
PEOPLE	63

TABLE 4.3: Internal heat load members in seminar room

**Desired condition to be maintained at cooling space:**

25.5 °C with 50% RH and  $w = 0.0102075$  kg vapor/kg dry air

**Outdoor weather condition in delhi on june 2015 :**

45 °C with 20% RH, daily range of 14.5 C

$w = 0.010982$  kg vapor/kg dry air

Outside heat transfer coefficient = 20 W/m<sup>2</sup>K

**4.3 EXTERNAL COOLING LOAD CALCULATION:**

**1. HEAT TRANSFER THROUGH OPAQUE SURFACES:**

**WALLS:**

**WALL 1:** There is only one sunlit wall, wall no. 1 in the entire room. The R- value is chosen by taking wall’s material under consideration from ASHRAE fundamental handbook. The appropriate value range is,

R-Value Range = 2.5 – 3.0

( table 15,ch 26)The corresponding wall type in accordance with this R range and the wall material is Wall no. 5.

( table 33B,ch 26)

Now this wall number is used to calculate CLTD for the wall heat transfer calculation. By inspection the cooling load from the sunlit wall can be expected to be the variable making the greatest contribution to the overall cooling load for the building. Therefore the time of maximum cooling load occurrence will probably be close to the time of max CLTD for the sunlit wall.

From ASHRAE Table no. 32 the CLTD for wall no. 5 at 40° N latitude, 78 F indoor , 95 F max outdoor, 85 F mean and 21 F daily range temperature has a max tabulated value of 40 F at hrs 1300 and 1400.

TABLE 4.4: CLTD values for sunlit wall

TIME	1200	1300	1400	1500
CLTD	39	40	40	38
$t_r$	78	78	78	78
$t_m$	100	100	100	100
Corrected CLTD	54	55	55	53

$t_r$  = inside temperature

$t_m$  = mean outdoor temperature = max outdoor temperature – (daily range)/2

Corrected  $CLTD_{max} = CLTD_{max} + (78 - t_r) + (t_m - 85)$

The value of overall heat transfer coefficient can be calculated from the wall material description which comes out to be 2.34 W/m<sup>2</sup>K (Table 4, ch22)

$Q_{wall\ 1} = 243.09\ W$

**WALL 2:** It is the non sunlit wall hence CLTD equals to the temperature difference across the wall. The other side of wall can be expected to be at the ambient temperature.

CLTD = 35

$Q_{wall}$

$2 = 688.36\ W$

**WALL 3, WALL 4:** As the wall is attached to another room which are also air conditioned, the CLTD can be directly estimate zero in this case.

$$CLTD = 0$$

$$Q_{\text{wall } 3} = Q_{\text{wall } 4} = 0 \text{ KW}$$

**ROOF:** As the space is on ground floor in a three story building having the other room above it and the temperature at the other side of roof is same as that of the conditioned space as that room is also air conditioned, the CLTD in this can assume to be zero.

$$CLTD = 0$$

$$Q_{\text{roof}} = 323.39 \text{ W}$$

$$\text{FLOOR: } U = 0.314 \text{ W/m}^2\text{K}$$

$$CLTD = 26.1$$

(ASHRAE Handbook ch 26 )

$$Q_{\text{roof}} = 327.08 \text{ W}$$

## 2. HEAT TRANSFER THROUGH FENESTRATION:

### Radiation heat transfer:

The load component from solar heat gain is calculated by determining values of SC and SCL. A shading coefficient of 0.87 is used for heat absorbing ½ in glass.

(Table 11, ch 27)

SCL values are taken after identifying the appropriate zone type for solar load.

For bottom floor, two sides exposed, uncarpeted floor with masonry partition and partially inside shaded window, the zone type selected for various loads are shown in table 4.5.

TABLE 4.5: Zone types for use with SCL and CLF tables

ZONE PARAMETERS				ZONE TYPE		
No Walls	Floor Covering	Partition Type	Inside Shade	Glass Solar	People and Equipments	Lights
1 or 2	Uncarpeted Vinyl	Concrete block	Half or None	D	D	D

Cross checking the variation of SCL for glass solar Zone type D, glass facing east, the maximum cooling load from these windows would be slightly more at early hours of the day than at noon or 1300 for 40° N latitude.

ZONE TYPE D: For zone D, SCL values are tabulated in east direction at different solar time hours in table 4.6.

TABLE 4.6: SCL values for ZONE TYPE D

Glass Face	Solar Time (Hour)					
	9	10	11	12	13	14
Direction East	124	110	85	65	60	57

By taking SCL value 60 the radiation cooling load from fenestration areas is shown in table 4.7.

TABLE 4.7: Radiation cooling load from fenestration areas

Section	Number	Area (m <sup>2</sup> )	SC	SCL	Cooling Load (W)
Window 1	3	1.98 × 1.6	0.87	60	1605.4
Window 2	3	1.73 × 1.6	0.87	60	1410.8

**Conduction heat transfer:**

For single plate glass, U = 4.857 W/m<sup>2</sup>K

CLTD value should be taken at the maximum cooling load hours i.e 1300 and 1400.

TABLE 4.8: CLTD values for conduction heat transfer

TIME	1100	1200	1300	1400
CLTD	7	9	12	13
t <sub>r</sub>	78	78	78	78
t <sub>m</sub>	100	100	100	100
Corrected CLTD	22	24	27	28

t<sub>r</sub> = inside temperature t<sub>m</sub>

= mean outdoor temperature = max outdoor temperature – (daily range)/2

Corrected CLTD<sub>max</sub> = CLTD<sub>max</sub> + (78 – t<sub>r</sub>) + (t<sub>m</sub> – 85)

Cooling load values due to conduction fenestration from the windows are tabulated in table 4.9.

TABLE 4.9: Cooling load through conduction fenestration

Section	Number	Area (m <sup>2</sup> )	CLTD	CLTD <sub>corr</sub>	Cooling Load (W)
Window 1	3	1.98 × 1.6	12	27	705.82
Window 2	3	1.73 × 1.6	12	27	588.18

**3. Heat transfer due to infiltration:**

Window infiltration is considered zero since the windows are sealed. Infiltration through wall surfaces is also neglected as insignificant, particularly plastered outer and wooden interior surfaces. Calculation of door infiltration however, should take under consideration. In this case the door infiltration is included as a part of cooling load, estimating 80 ft<sup>3</sup> / person /door passage. Further estimating outside door use at 10 person/h and insider door use at 30 person/h, generates the following infiltration rate:

$$cfm_{inf} = 40 \times 80/60 = 53.33 \text{ cfm}$$

**Sensible load:**

$$Q_{s,inf} = 601.73 \text{ W}$$

**Latent load:**

$$Q_{l,inf} = 58.58 \text{ W}$$

**4. Heat transfer due to ventilation:**

From ASHRAE standard 62, a ventilation load of 15 cfm/person is selected as representative of a general office building. With 63 people, the total ventilation air quantity is:

$$Cfm_{ven} = 63 \times 15 = 945 \text{ cfm}$$

**Sensible load:**

$$Q_{s,ven} = 10662.65 \text{ W}$$

**Latent load:**

$$Q_{l,ven} = 1038.17 \text{ W}$$

**3.4 INTERNAL COOLING LOAD CALCULATION:**

**a) Load due to occupants:**

The heat gain rates for seated and standing occupants doing light office work is shown in table 4.10.

TABLE 4.10: Heat gain rates for occupants

No	Degree of Activity	Total Heat (W)	Sensible Heat (W)	Latent Heat (W)	% SH that is Radiant	
					Low V	High V
60	Seated, very light work	132	72	30	18	8
3	Standing, light	160	74	30	17	11



	work, walking					
--	---------------	--	--	--	--	--

The CLF for the sensible component is taken from ASHRAE Handbook, table 37 ch 26 as 0.87, for a condition of 10 total hours in a zone type D and a load calculation taken 8 hours after entry. Cooling load from people is thus estimated at:

$$Q_{s,p} = 3968.18 \text{ W}$$

$$Q_{l,p} = 1938.66 \text{ W}$$

**b) Load due to lighting:**

Assuming that use factor is 1.0 and the special allowance factor is 1.25 for 20 W fluorescent bulbs. The CLF for the lightning is taken from ASHRAE Handbook, table 38 ch 26 as 0.90, for a condition of 10 total hours in a zone type D and a load calculation taken 10 hours after entry. Cooling load from people is thus estimated at:

$$Q_{s,light} = 404.74 \text{ W}$$

**c) Internal loads due to equipment and appliances:**

As there is no radiation from smaller objects like laptop and fans therefore the CLF values for these objects can be taken as 1.0. The latent heat load due to appliances is zero as there is no latent heat producing appliances in the space, thus the sensible heat load only is calculated here and tabulated in table 4.11.

TABLE 4.11: Internal loads due to equipment and appliances

Appliance	Number	Installed Wattage	CLF	Cooling Load (W)
Laptop	2	60	1	1200
Fan	4	28	1	112

# **CHAPTER 5 : THERMODYNAMIC ANALYSIS**

## **5.1 ASSUMPTIONS**

For developing the model for analyzing the absorption system some input data are taken under consideration and assumed that cycle works under ideal conditions. The mathematical model is analyzed by using mass balance and energy transfer equations between the internal and external streams for each component. Several assumptions and hypothesis are considered for the systems

- The pressure drop in the pipes and vessels is negligible.
- The heat losses from the generator to the surroundings and the heat gains to the evaporator from the surroundings are negligible.
- The expansion process of the expansion device is at constant enthalpy.
- The water at the outlet of evaporator is saturated vapor.
- The water at the outlet of condenser is saturated liquid.
- The water vapor at the outlet of generator is superheated.
- The mass accumulation in the generator, absorber, condenser, evaporator and the solution heat exchanger is neglected.

The outlet temperatures from the absorber and from generators correspond to equilibrium conditions of the mixing and separation, respectively.

## **5.2 SINGLE EFFECT LITHIUM BROMIDE WATER VAPOR ABSORPTION SYSTEM**

The basis of thermodynamics is stated in the first and second laws. The first law describes the conservation of energy, while the second law is used to describe the quality of energy and material. For the thermodynamic analysis of the absorption system the principles of mass conservation, first and second laws of thermodynamics are applied to each component of the system. This energy and exergy equations are applied to every system component to obtain heat rates in various components and their exergy destruction rates. Concentration balance and mass balance equations are

used to determine thermodynamic properties at various state points of the system cycle. With the help of charts the properties are determined which are further used to calculate heat transfer rates by using steady flow energy equation (SFEE).

The performance coefficient and exergy efficiency can also be determined from the result data.

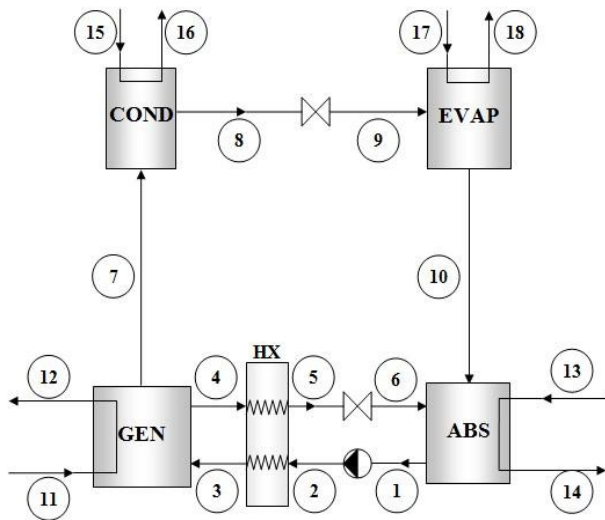


Fig. 5.1: Single effect LiBr-water circuit

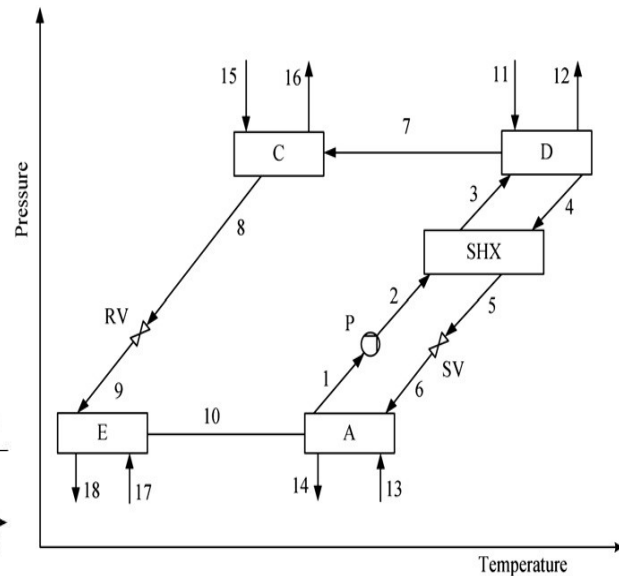


Fig.5.2: P-T diagram of Single effect circuit

### Mass conservation

Mass conservation law for each component is written as:

$$\sum \dot{m}_i = \sum \dot{m}_o$$

This law for each component of the cycle is written as:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_3$$

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_6$$

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_9 = \dot{m}_{10}$$

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_7$$

$$\dot{m}_1 = \dot{m}_6 + \dot{m}_{10}$$

### Conservation of concentration

$$\sum \dot{m}_i X_i = \sum \dot{m}_o X_o$$

$$\dot{m}_3 X_3 = \dot{m}_4 X_4$$

$$X_1 = X_2 = X_3$$

$$X_4 = X_5 = X_6$$

$$X_7 = X_8 = X_9 = X_{10} = 0$$

### The first law of thermodynamics/ Energy analysis

The first law of thermodynamics yields the energy balance of each component (each component can be treated as a control volume with inlet and outlet streams, heat transfer and work interactions) of the absorption system as follows:

$$(\sum \dot{m}_i h_i - \sum \dot{m}_o h_o) + (\sum Q_i - \sum Q_o) - W_{out} = 0$$

$$\text{Generator: } \dot{m}_3 h_3 - \dot{m}_4 h_4 - \dot{m}_7 h_7 + Q$$

$$W_{gen} = 0$$

$$\text{Condenser: } Q$$

$$Q_{cond} = \dot{m}_7 (h_7 - h_8)$$

$$\text{Evaporator: } Q$$

$$Q_{evp} = \dot{m}_9 (h_{10} - h_9)$$

$$\text{Absorber: } \dot{m}_{10} h_{10} + \dot{m}_6 h_6 - \dot{m}_1 h_1 - Q$$

$$W_{abs} = 0$$

Heat

$$\text{Exchanger : } Q$$

$$Q_{hch} = \dot{m}_2 (h_3 - h_2)$$

$$\text{COP} = Q$$

$$Q_{evp} / Q$$

$$W_{gen}$$

### Exergy analysis

Exergy analysis is the combination of the first and the second laws of thermodynamics. The total actual exergy change that occurs between the inlets and outlets of the control volume, namely the sum of the reversible and the irreversible contributions to the total actual exergy change, can be calculated by:

$$\dot{E}_D = \sum (1 - T_o/T_i) \dot{Q} + (\sum \dot{m} e)_{in} - (\sum \dot{m} e)_{out} + \dot{W}$$

By neglecting changes in kinetic energy and potential energy and due to the fact that there is no release of chemical substances from the cycle to the environment, the chemical exergy is zero and the specific exergy is calculated by:

$$e = (h-h_o) - T_o(s-s_o)$$

Therefore exergy rate at any in or out point can be written as:

$$\dot{E} = \dot{m} ((h-h_o) - T_o(s-s_o))$$

$$\text{Generator : } \Delta\dot{E}_{\text{gen}} = \dot{E}_3 + \dot{E}_{11} - (\dot{E}_7 + \dot{E}_4 + \dot{E}_{12})$$

$$\text{Condenser : } \Delta\dot{E}_{\text{cond}} = \dot{E}_{15} + \dot{E}_7 - (\dot{E}_8 + \dot{E}_{16})$$

$$\text{Evaporator : } \Delta\dot{E}_{\text{evp}} = \dot{E}_9 + \dot{E}_{17} - (\dot{E}_{18} + \dot{E}_{10})$$

$$\text{Absorber : } \Delta\dot{E}_{\text{abs}} = \dot{E}_{10} + \dot{E}_{13} + \dot{E}_6 - (\dot{E}_{14} + \dot{E}_1)$$

$$\text{Heat Exchanger : } \Delta\dot{E}_{\text{hxch}} = \dot{E}_2 + \dot{E}_4 - (\dot{E}_3 + \dot{E}_5)$$

$$\text{Expansion valve : } \Delta\dot{E}_{\text{exp}} = \dot{E}_8 - \dot{E}_9$$

### Total Exergy Loss

The total exergy loss ( $\Delta ET$ ) of the absorption chiller cycle is the sum of the exergy losses in each component of the cycle:

$$\Delta\dot{E}_T = \Delta\dot{E}_{\text{gen}} + \Delta\dot{E}_{\text{cond}} + \Delta\dot{E}_{\text{evp}} + \Delta\dot{E}_{\text{abs}} + \Delta\dot{E}_{\text{hxch}} + \Delta\dot{E}_{\text{exp}}$$

**Exergatic efficiency** can be calculated by:

$$\eta_{\text{exe}} = (\dot{Q}_{\text{evp}} | 1 - T_o/T_{\text{evp}} |) / (\dot{Q}_{\text{gen}} (1 - T_o/T_{\text{gen}})) = \text{Exergy input to the system} / \text{Exergy output from the system}$$

## 5.3 DOUBLE EFFECT LITHIUM BROMIDE WATER VAPOR ABSORPTION SYSTEM

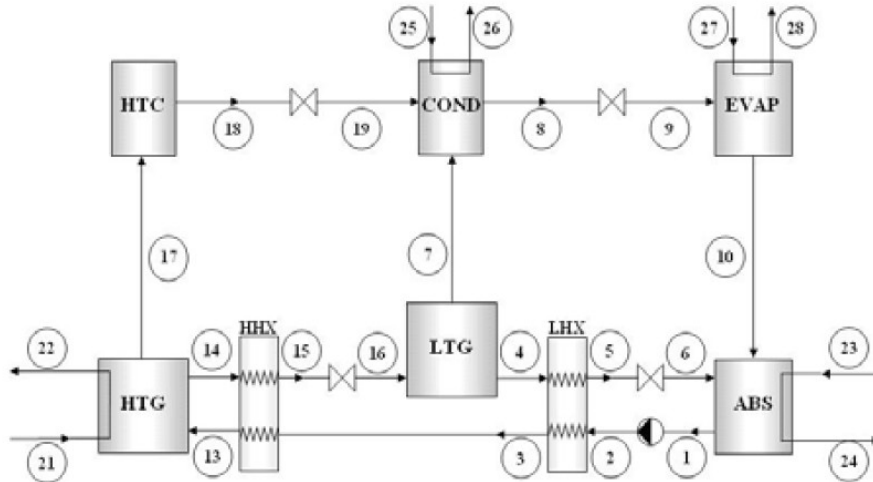


Fig. 5.3: Double effect LiBr-water system circuit

### Mass conservation :

$$\dot{m}_{12} = \dot{m}_{13} = \dot{m}_1 = \dot{m}_2 = \dot{m}_3$$

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_6 = \dot{m}_7$$

$$\dot{m}_8 = \dot{m}_9 = \dot{m}_{10}$$

$$\dot{m}_{15} = \dot{m}_{16} = \dot{m}_{17}$$

$$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_{13}$$

### Conservation of concentration

$$\dot{m}_{10} X_{10} = \dot{m}_{15} X_{15}$$

$$\dot{m}_7 X_7 = \dot{m}_8 X_8$$

$$X_4 = X_5 = X_6 = X_7$$

$$X_8 = X_9 = X_{10}$$

$$X_{15} = X_{16} = X_{17}$$

$$X_1 = X_2 = X_3 = X_{14} = X_{12} = X_{13} = X_{11} = 0$$

### Energy analysis

$$\text{HPGenerator} : \dot{m}_7 h_7 - \dot{m}_{11} h_{11} - \dot{m}_8 h_8 + \dot{Q}_{\text{HPgen}} = 0$$

$$\text{LPGenerator} : \dot{m}_{11} h_{11} - \dot{m}_{12} h_{12} + \dot{m}_{10} h_{10} - \dot{m}_{14} h_{14} - \dot{m}_{15} h_{15} = 0$$

$$\text{Condenser} : \dot{m}_{13} h_{13} + \dot{m}_{14} h_{14} - \dot{m}_1 h_1 - \dot{Q}_{\text{cond}} = 0$$

$$\text{Evaporator} : \dot{Q}_{\text{evp}} = \dot{m}_2 (h_3 - h_2)$$

$$\text{Absorber: } \dot{m}_{17} h_{17} + \dot{m}_3 h_3 - \dot{m}_4 h_4 - \dot{Q}_{\text{abs}} = 0$$

$$\text{LPHeat Exchanger I: } \dot{Q}_{\text{LPhxch}} = \dot{m}_5 (h_6 - h_5)$$

$$\text{HPHeat Exchanger II: } \dot{Q}_{\text{HPPhxch}} = \dot{m}_6 (h_7 - h_6)$$

### Exergy analysis

$$\text{HPGenerator: } \Delta \dot{E}_{\text{HPgen}} = \dot{E}_7 + \dot{E}_{18} - (\dot{E}_{11} + \dot{E}_8 + \dot{E}_{19})$$

$$\text{LPGenerator: } \Delta \dot{E}_{\text{LPgen}} = \dot{E}_{10} + \dot{E}_{11} - (\dot{E}_{12} + \dot{E}_{14} + \dot{E}_{15})$$

$$\text{Condenser: } \Delta \dot{E}_{\text{cond}} = \dot{E}_{13} + \dot{E}_{14} + \dot{E}_{20} - (\dot{E}_{21} + \dot{E}_1)$$

$$\text{Evaporator: } \Delta \dot{E}_{\text{evp}} = \dot{E}_{22} + \dot{E}_2 - (\dot{E}_{23} + \dot{E}_3)$$

$$\text{Absorber: } \Delta \dot{E}_{\text{abs}} = \dot{E}_3 + \dot{E}_{17} + \dot{E}_{24} - (\dot{E}_4 + \dot{E}_{25})$$

$$\text{Heat Exchanger I: } \Delta \dot{E}_{\text{LPhxch}} = \dot{E}_5 + \dot{E}_{15} - (\dot{E}_{16} + \dot{E}_6)$$

$$\text{Heat Exchanger II: } \Delta \dot{E}_{\text{HPPhxch}} = \dot{E}_6 + \dot{E}_8 - (\dot{E}_9 + \dot{E}_7)$$

$$\text{Expansion valve: } \Delta \dot{E}_{\text{exp}} = \dot{E}_1 - \dot{E}_2$$

### Total Exergy Loss

$$\Delta \dot{E}_T = \Delta \dot{E}_{\text{HPgen}} + \Delta \dot{E}_{\text{LPgen}} + \Delta \dot{E}_{\text{cond}} + \Delta \dot{E}_{\text{evp}} + \Delta \dot{E}_{\text{abs}} + \Delta \dot{E}_{\text{LPhxch}} + \Delta \dot{E}_{\text{HPPhxch}} + \Delta \dot{E}_{\text{exp}}$$

### Exergatic efficiency

$$\eta_{\text{exe}} = (\dot{Q}_{\text{evp}} | 1 - T_o/T_{\text{evp}} |) / (\dot{Q}_{\text{HPgen}} (1 - T_o/T_{\text{HPgen}}))$$

## 5.4 HALF EFFECT LITHIUM BROMIDE WATER VAPOR ABSORPTION SYSTEM

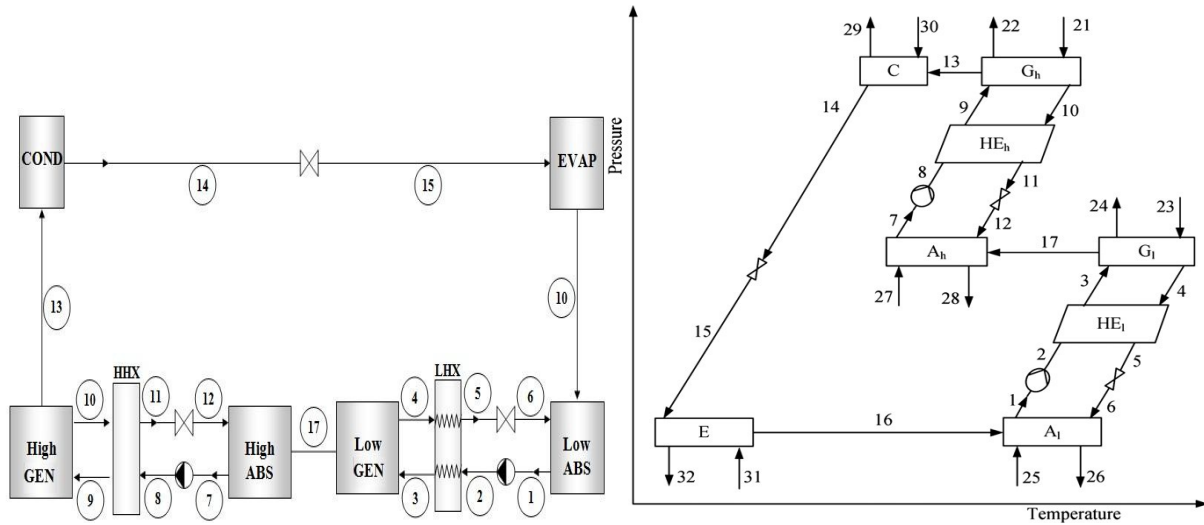


Fig. 5.4 : Half effect LiBr-water system circuit

Fig.5.5: P-T diagram of Half effect

### Mass conservation :

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_3$$

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_6$$

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_9$$

$$\dot{m}_{10} = \dot{m}_{11} = \dot{m}_{12}$$

$$\dot{m}_{13} = \dot{m}_{14} = \dot{m}_{15} = \dot{m}_{16}$$

### Conservation of concentration

$$\dot{m}_1 X_1 = \dot{m}_6 X_6$$

$$\dot{m}_9 X_9 = \dot{m}_{10} X_{10}$$

$$X_1 = X_2 = X_3$$

$$X_4 = X_5 = X_6$$

$$X_7 = X_8 = X_9$$

$$X_{10} = X_{11} = X_{12}$$

$$X_{13} = X_{14} = X_{15} = X_{16} = X_{17} = 0$$

### Energy analysis

$$\text{HPGenerator} : \dot{m}_9 h_9 - \dot{m}_{10} h_{10} - \dot{m}_{13} h_{13} + \dot{Q}_{\text{HPgen}} = 0$$

$$\text{LPGenerator} : \dot{m}_3 h_3 - \dot{m}_4 h_4 - \dot{m}_{17} h_{17} + \dot{Q}_{\text{LPgen}} = 0$$

$$\text{Condenser} : \dot{m}_{13} h_{13} - \dot{m}_{14} h_{14} - \dot{Q}_{\text{cond}} = 0$$



$$\text{Evaporator: } \dot{Q}_{\text{evp}} = \dot{m}_{15} (h_{16} - h_{15})$$

$$\text{LP Absorber: } \dot{m}_1 h_1 - \dot{m}_6 h_6 - \dot{Q}_{\text{LPabs}} = 0$$

$$\text{HP Absorber: } \dot{m}_{17} h_{17} + \dot{m}_{12} h_{12} - \dot{m}_7 h_7 - \dot{Q}_{\text{HPabs}} = 0$$

$$\text{LP Heat Exchanger I: } \dot{Q}_{\text{LPPhxch}} = \dot{m}_2 (h_3 - h_2)$$

$$\text{HP Heat Exchanger II: } \dot{Q}_{\text{HPPhxch}} = \dot{m}_8 (h_9 - h_8)$$

### Exergy analysis

$$\text{HP Generator: } \Delta \dot{E}_{\text{HPgen}} = \dot{E}_9 + \dot{E}_{21} - (\dot{E}_{22} + \dot{E}_{10} + \dot{E}_{13})$$

$$\text{LP Generator: } \Delta \dot{E}_{\text{LPgen}} = \dot{E}_3 + \dot{E}_{23} - (\dot{E}_{17} + \dot{E}_{24} + \dot{E}_4)$$

$$\text{Condenser: } \Delta \dot{E}_{\text{cond}} = \dot{E}_{13} + \dot{E}_{30} - \dot{E}_{29}$$

$$\text{Evaporator: } \Delta \dot{E}_{\text{evp}} = \dot{E}_{15} + \dot{E}_{31} - (\dot{E}_{16} + \dot{E}_{32})$$

$$\text{LP Absorber: } \Delta \dot{E}_{\text{LPabs}} = \dot{E}_6 + \dot{E}_{25} + \dot{E}_{10} - (\dot{E}_1 + \dot{E}_{26})$$

$$\text{HP Absorber: } \Delta \dot{E}_{\text{HPabs}} = \dot{E}_{12} + \dot{E}_{27} + \dot{E}_{17} - (\dot{E}_7 + \dot{E}_{28})$$

$$\text{LP Heat Exchanger: } \Delta \dot{E}_{\text{LPPhxch}} = \dot{E}_2 + \dot{E}_4 - (\dot{E}_3 + \dot{E}_5)$$

$$\text{HP Heat Exchanger: } \Delta \dot{E}_{\text{HPPhxch}} = \dot{E}_8 + \dot{E}_{10} - (\dot{E}_{11} + \dot{E}_9)$$

$$\text{Expansion xvalve: } \Delta \dot{E}_{\text{exp}} = \dot{E}_{14} - \dot{E}_{15}$$

### Total Exergy Loss

$$\Delta \dot{E}_T = \Delta \dot{E}_{\text{HPgen}} + \Delta \dot{E}_{\text{LPgen}} + \Delta \dot{E}_{\text{cond}} + \Delta \dot{E}_{\text{evp}} + \Delta \dot{E}_{\text{LPabs}} + \Delta \dot{E}_{\text{HPabs}} + \Delta \dot{E}_{\text{LPPhxch}} + \Delta \dot{E}_{\text{HPPhxch}} + \Delta \dot{E}_{\text{exp}}$$

### Exergatic efficiency

$$\eta_{\text{exe}} = (\dot{Q}_{\text{evp}} \left| 1 - T_o/T_{\text{evp}} \right|) / (\dot{Q}_{\text{HPgen}} (1 - T_o/T_{\text{HPgen}}) + \dot{Q}_{\text{LPgen}} (1 - T_o/T_{\text{HPgen}}))$$

**Area of solar collector:** Sun light falls on solar collector which is being used to heat the water. Let's consider heat absorbing capacity of insulating material as 100%. This heated water is stored in hot water tank.

Its efficiency mainly depends on insulating material and its thickness. It has efficiency of 85%.

Energy absorbed by the collector plate equals to the energy removed from the generator, and is given by

$$Q_g = K \times M \times A$$

K=efficiency of collector plate (assume k=0.85)

M= maximum solar heat falling on earth's surface=7.2 KWH/m<sup>2</sup>/day= 900 W/m<sup>2</sup> ( 8 hrs)

A=Area of collector plates

Qg =Heat req. in the generator

## CHAPTER 6 : RESULT AND DISCUSSION

### 6.1 COOLING LOAD

The maximum cooling loads, sensible heat ratio of the sample space calculated using cooling load temperature difference (CLTD) method. All the values can be shown as a tabulated form.

#### COOLING LOAD CALCULATION FOR MECHANICAL ENGINEERING DEPARTMENT SEMINAR ROOM

Table 6.1 shows the outdoor and indoor desired condition of the space.

TABLE 6.1: outdoor and indoor conditions

	Dry Bulb °C	RH %	Specific Humidity kg vap/kg dry air
Outdoor Conditions	45	20	0.010982
Indoor Conditions	25	50	0.010207
Difference	20	30	0.000775

Table 6.2 shows the sensible heat load calculation of the room from different heat parts of the room. The final result shows that the total cooling load from all of the sensible heat load parts, inside the building comes out to be 20.35 KW. This table does not show the heat load from latent heat sources inside the building. A separate table is made to show various latent heat load components in the building and their values.

TABLE 6.2: Sensible load calculation of mechanical engineering department seminar room

<b>SENSIBLE COOLING LOAD AT 1300 h</b>	<b>Watt</b>
Roof and exposed walls	
Roof	
East (sunlit wall)	2430.96
SW wall	688.36
West wall	_____
North wall	_____
Fenestration Areas	
East window 1	
705.82	
East window 2	
588.18	
Internal Sources	
People	
3968.18	Fluorescent lights
1200.00	Fan
112.00	Laptop
120.00	
Outside Air	
Infiltration	
675.00	Ventilation
10662.65	
<b>TOTAL</b>	

Table 6.3 shows the sensible heat load calculation of the room from different heat parts of the room.

TABLE 6.3: Sensible load calculation of mechanical engineering department seminar room

<b>LATENT COOLING LOAD AT 1300 h</b>	
<b>Watt</b>	
People	1938.66
Infiltration	58.58
Ventilation	1038.17
<b>TOTAL</b>	<b>3035.42</b>

Total sensible and latent loads are obtained by summing-up all the sensible and latent load components (both external as well as internal) as:

$$Q_{S, total} = 20356.03 \text{ W}$$

$$Q_{L, total} = 3035.42 \text{ W}$$

$$Q_{total} = 23391.46 \text{ W}$$

Room Sensible Heat Factor (RSHF) is given by:

$$RSHF = Q_{s, total} / Q_{total} = 0.8702$$

To calculate the required cooling capacity, one has to know the losses in return air ducts, supply ducts and fan heat gain. Hence using a safety factor of 1.25, the required cooling capacity is:

$$\text{Required cooling capacity} = 23391.46 \times 1.25 = 29.09 \text{ KW} = 8.31 \text{ TR}$$

The required cooling load for the seminar building is 8.31TR or 29.09 KW. This refrigeration load further be used in the analysis of solar assisted vapor absorption system.

## **5.2 LITHIUM BROMIDE-WATER ABSORPTION SYSTEM**

By analyzing the thermodynamic model, the performance of each component of the lithium bromide absorption cycle has been determined. Based on different working temperatures (generator temperature) and fixed values for condenser absorber and evaporator temperatures the coefficient of performance (COP) and exergetic efficiency of each system were determined.

The fix data used for the calculation are mentioned in tables 6.4.

The state points and values of different properties like temperature, pressure, solution concentration, enthalpy and entropy are calculate and tabulated. Which further will be

used for first and second law analysis. The fixed values taken for calculation have shown in table 6.4:

TABLE 6.4: Fixed conditions used for the analysis of LiBr-water absorption system

Cooling load	29.09 KW
$T_{evp}$	8 ( $^{\circ}$ C)
$T_{cond}$	35 ( $^{\circ}$ C)
$T_{abs}$	35 ( $^{\circ}$ C)
$T_{gen}$ (half effect)	80 ( $^{\circ}$ C)
$T_{gen}$ (single effect)	110 ( $^{\circ}$ C)
$T_{gen}$ (double effect)	130 ( $^{\circ}$ C)
$T_{amb}$	25 ( $^{\circ}$ C)
$P_{amb}$	101(Kpa)

## 6.2.1 THERMODYNAMIC PROPERTIES AT ALL STATE POINTS OF LITHIUM BROMIDE-WATER ABSORPTION SYSTEM:

### 6.2.1.1 Single effect LiBr-water absorption system

The values of all the thermo dynamical properties like temperature, pressure, specific enthalpy, specific entropy and mass flow rates for the single effect absorption system are calculated and shown in table 6.5.

TABLE 6.5: Thermodynamic properties of single effect absorption system

State point	T( $^{\circ}$ C)	P(Kpa)	X	m(kg/sec)	h(kj/kg)	s(kj/kg-K)
1	35	1.073	0.54	0.046	79.31	0.222
2	35	1.073	0.54	0.046	79.31	0.222
3	55	5.627	0.54	0.046	121	0.354
4	110	5.627	0.73	0.033	332.6	0.508
5	75.21	5.627	0.73	0.033	275.6	0.349
6	75.21	1.073	0.53	0.033	275.6	0.349
7	110	5.627	—	0.012	2706	8.764
8	35	5.627	—	0.012	146.6	0.505
9	8	1.073	—	0.012	146.6	0.523
10	8	1.073	—	0.012	2515	8.984
11	130	270	—	0.616	546.4	1.635
12	115	270	—	0.616	483.4	1.475
13	27	101	—	1.446	113.1	0.395
14	33	101	—	1.446	138.2	0.477
15	27	101	—	1.247	113.1	0.395
16	33	101	—	1.247	138.2	0.477
17	16	101	—	1.154	67.1	0.239
18	10	101	—	1.154	42	0.151

### 6.2.1.2 Double effect LiBr-water absorption system

The values of different properties of double effect series flow vapor absorption system at each state point are mentioned in table 6.6.

TABLE 6.6: Thermodynamic properties of double effect absorption system

State point	T(°C)	P(Kpa)	X	m(kg/sec)	h(kj/kg)	s(kj/kg-K)
1	35	1.073	0.54	0.019	79.31	0.222
2	35	5.627	0.54	0.019	79.31	0.222
3	65.72	5.627	0.54	0.019	143.7	0.421
4	80	5.627	0.61	0.096	195.8	0.451
5	48.54	5.627	0.61	0.096	135.7	0.272
6	48.54	1.073	0.61	0.096	135.7	0.272
7	80	5.627	—	0.008	2643	7.611
8	35	5.627	—	0.012	146.6	0.505
9	8	1.073	—	0.012	146.6	0.523
10	8	1.073	—	0.012	2515	8.948
13	44.24	20	0.54	0.019	205	0.594
14	130	20	0.69	0.014	335.2	0.629
15	99.17	20	0.69	0.014	282.7	0.494
16	99.17	5.627	0.69	0.014	282.7	0.494
17	94.24	20	—	0.004	2675	8.095
18	94.24	20	—	0.004	251.5	0.832
19	35	5.627	—	0.004	251.5	0.845
21	145	415	—	2.422	610.7	1.791
22	140	415	—	2.422	589.7	1.748
23	27	101	—	1.402	113.1	0.395
24	33	101	—	1.402	138.2	0.477
25	27	101	—	0.807	113.1	0.395
26	33	101	—	0.807	138.2	0.477
27	16	101	—	0.866	67.1	0.239
28	10	101	—	0.866	42	0.151

### 6.1.2.3 Half effect LiBr-water absorption system

Half effect absorption system values at each state point is tabulated in table 6.7.

TABLE 6.7: Thermodynamic properties of double effect absorption system

State point	T(°C)	P(Kpa)	X	m(kg/sec)	h(kj/kg)	s(kj/kg-K)
1	35	1.073	0.54	0.071	79.31	0.222
2	35	3.5	0.54	0.071	79.31	0.222
3	56.25	3.5	0.54	0.071	123.6	1.075
4	80	3.5	0.65	0.582	221.4	0.425
5	57.75	3.5	0.65	0.582	181.8	0.309

6	57.75	1.073	0.65	0.582	181.8	0.309
7	35	3.5	0.36	0.031	77.94	0.369
8	35	5.627	0.36	0.031	77.94	0.369
9	43.82	5.627	0.36	0.031	101	0.443
10	80	5.627	0.60	0.018	195.8	0.451
11	70.3	5.627	0.60	0.018	177.1	0.397
12	70.3	3.5	0.60	0.018	177.1	0.397
13	43.82	5.627	—	0.012	2581	8.405
14	35	5.627	—	0.012	146.6	0.505
15	8	1.073	—	0.012	146.6	0.523
16	8	1.073	—	0.012	2515	8.948
17	56.25	3.5	—	0.012	2605	8.948
21	95	101	—	0.766	398	1.25
22	85	101	—	0.766	355.9	1.134
23	90	101	—	0.861	376.9	1.193
24	80	101	—	0.861	334.9	1.075
25	27	101	—	1.402	113.1	0.395
26	33	101	—	1.402	138.2	0.477
27	27	101	—	0.807	113.1	0.395
28	33	101	—	0.807	138.2	0.477
29	33	101	—	0.807	138.2	0.477
30	27	101	—	0.807	113.1	0.395
31	16	101	—	0.866	67.1	0.239
32	10	101	—	0.866	42	0.151

### 6.2.2 FIRST LAW ANALYSIS OF LITHIUM BROMIDE-WATER VAPOR ABSORPTION SYSTEM

The values of heat interactions in all the system components have tabulated in table 6.8. Direct comparison can be made by seeing the values of COP, exergetic efficiencies, heat interaction among the systems. Half effect has the minimum value of COP among all effects. This is due to the fact that the generator heat input is large in it than in any other system. Half effect uses two generators at different temperature and pressure. The cumulative heat input to the generator is grater than the single or double effect. The COP of half effect is in between the range of 0.4-0.5.

Double effect has minimum heat input given to the generator hence yield maximum COP among all the systems. The highest value of COP in double effect system can be reached theoretically upto 1.5 but for this the required generator temperature should be minimum 150 (°C) or greater than that. This is quite inconvenient as it requires expensive technologies and complex arrangements in the cycle. On the other hand in



order to achieve highest COP (0.5) in half effect system the required temperature is in between the range of 60-80 (°C). This temperature can easily obtain with less expensive solar technologies. It should be noted that the intermediate range of COP can be achieved in single effect system. The range of COP for single effect system is 0.7 -0.9. the temperature required to achieve this COP is greater than half effect but lower than the double effect. Therefore by using single effect we can get moderate value of COP with moderate generator temperature range.

Fig 6.1 shows the variation of the COP with generator temperatures for various systems. It can be seen that the COP increases from half to double effect system. As the generator temperature increase more further, for any system, COP decreases. This is because at the fix evaporator load the increased generator temperature results in increased generator heat supply. Which directly causes COP to drop. The maximum value of generator temperature is also restrained by crystallization temperature of generator. As the generator temperature and pressure increase in each half to single to double effect, COP also increases along with it. Though the increase of the COP is very small from system to system. If the evaporator load is not fixed there is a tendency in COP to increase as the generator temperature or chiller water inlet temperature increase. This is because the more heat boils off more vapor from the solution. This increases mass flow rate of refrigerant going to evaporator and more refrigerant results in more refrigeration effect.

Table 6.8 compares various parameters like heat rates, exergy destruction rates, COP and exergetic efficiency of half, single and double effect system. It can be easily seen that the value of COP and exergetic efficiency increase from half to double effect system and heat input to the generator decreases from half to double effect system.

TABLE 6.8: Thermodynamic parameters of half to double effect LiBr-water absorption system

Parameter	Half effect	Single effect	Double effect	Unit
COP	0.45	0.74	1.20	
$Q_{EVP}$	29.09	29.09	29.09	KW
$Q_{COND}$	29.9	31.43	20.35	KW

$Q_{GEN}$	—	38.8	24.22	KW
$Q_{HTG}$	32.2	—	—	
$Q_{LTG}$	36.17	—	—	
$Q_{ABS}$	—	36.45	35.34	KW
$Q_{HTABS}$	32.86	—	—	
$Q_{LTABS}$	35.89	—	—	
$Q_{HX}$	—	1.9	—	KW
$Q_{HTHX}$	0.343	—	6.662	
$Q_{LTHX}$	2.305	—	6.487	
$\Delta \dot{E}_{EVP}$	0.542	0.542	0.542	KW
$\Delta \dot{E}_{COND}$	0.481	0.679	2.689	KW
$\Delta \dot{E}_{EXPVLV}$	0.664	0.664	0.664	KW
$\Delta \dot{E}_{GEN}$	—	3.101	—	KW
$\Delta \dot{E}_{HTG}$	2.783	—	6.154	
$\Delta \dot{E}_{LTG}$	1.968	—	6.336	
$\Delta \dot{E}_{ABS}$	—	2.65	1.376	KW
$\Delta \dot{E}_{HTABS}$	2.783	—	—	
$\Delta \dot{E}_{LTABS}$	3.057	—	—	
$\Delta \dot{E}_{HX}$	—	0.195	—	KW
$\Delta \dot{E}_{HTHX}$	0.015	—	0.191	
$\Delta \dot{E}_{LTHX}$	0.091	—	0.123	
$\eta_{exe}$	15.58	19.27	26.3	%
A	89.37	50.72	31.66	m <sup>2</sup>

Fig. 6.1: Variation of COP with generator temperature

TABLE 6.9: Variation of COP with generator temperature

COP VS $T_{gen}$					
double effect		half effect		single effect	
$T_{gen}$ [°C]	COP [KW]	$T_{gen}$ [°C]	COP [KW]	$T_{gen}$ [°C]	COP [KW]
110	1.433	40	0.4604	70	0.8204
114.4	1.378	45.56	0.4538	75.56	0.804
118.9	1.325	51.11	0.448	81.11	0.7893
123.3	1.273	56.67	0.4427	86.67	0.7763

127.8	1.224	62.22	0.4379	92.22	0.7648
132.2	1.179	67.78	0.4335	97.78	0.7547
136.7	1.135	73.33	0.4296	103.3	0.746
141.1	1.095	78.89	0.4261	108.9	0.7385
145.6	1.058	84.44	0.4231	114.4	0.732
150	1.024	90	0.4204	120	0.7266

### 6.2.3 SECOND LAW ANALYSIS OF LITHIUM BROMIDE-WATER VAPOR ABSORPTION SYSTEM (Exergetic efficiency of half to double effect LiBr-water absorption system)

Fig 6.2 shows the variation of exergetic efficiency with the generator temperature for different systems. Tendencies of exergetic efficiency in all the system is same that of the COP of the systems. The minimum value of exergetic efficiency is in half effect system followed by single and the maximum value is for double effect system. For half effect system it's value is 15.58% while for single effect it is 19.27% and for double effect it is 26.3% (TABLE 6.8).

Like the COPs the difference between the exergetic efficiency is also small from system to system. The decreasing pattern for exergetic efficiency is more steep than that of COP for the same system. It can also be seen in Fig 18. This is because of the comparatively constant denominator in COP calculation which is generator heat  $Q_{GEN}$ , while the denominator value in the exergetic efficiency formula is  $\dot{Q}_{gen}(1 - T_o/T_{gen})$ .

As the nominator for both of the formulas remains constant entirely, the change in denominator of exergetic efficiency is more than in COP due to the extra  $T_{gen}$  factor in the formula. Therefore exergetic efficiency is more sensitive per unit of temperature change than the COP. It shows more steepness in the graph of exergetic efficiency as the generator temperature increases.

The extra term in exergetic efficiency calculation  $(1 - T_o/T_{gen})$  is also known as dimensionless exergetic temperature. This term increases as the temperature of the generator increase and attends a very high value at higher generator temperatures. The comparison of COP an exergetic efficiency ( $\eta_{exe}$ ) is shown in TABLE 6.8.

Fig. 6.2: Variation of Exergetic efficiency with generator temperature

TABLE 6.10: Variation of Exergetic efficiency with generator temperature

Exergetic efficiency vs Tgen					
Half Effect		Single Effect		Double Effect	
T <sub>gen</sub> [°C]	η <sub>exe</sub>	T <sub>gen</sub> [°C]	η <sub>exe</sub>	T <sub>gen</sub> [°C]	η <sub>exe</sub>
40	0.548	60	0.4096	80	0.6404
45.56	0.4012	66.67	0.3411	88.89	0.5428
51.11	0.3172	73.33	0.2923	97.78	0.4601
56.67	0.2629	80	0.2559	106.7	0.3911
62.22	0.225	86.67	0.2279	115.6	0.3341
67.78	0.197	93.33	0.2058	124.4	0.2875
73.33	0.1756	100	0.188	133.3	0.2498
78.89	0.1587	106.7	0.1735	142.2	0.2194
84.44	0.1451	113.3	0.1614	151.1	0.1949
90	0.1339	120	0.1513	160	0.2265

#### 6.2.4 EXERGY DESTRUCTION RATES

There is another method to understand the difference between cycles operating at different working conditions. Exergy destruction rate shows the rate at which exergy of any system or component is lost because of irreversibility present in the system or component. The irreversibility rate is directly equals to the difference of exergy input and exergy output rate. At ideal condition when irreversibility rate is zero, the exergy in and exergy out are equal. Therefore the exergy distruction rate can be at minimum zero ( in ideal conditions) or positive ( in real conditions) but it can never be negative.

The exergy destruction rate of all the main components of different cycles is tabulated in TABLE 6.8. The direct comparison among the cycles can be made from this. This comparison can be used to understand the working of different cycles under different operating condition and it's effect on cycle performance.

From TABLE 6.8 it can be seen that in all the cycles the exergy destruction rate of expansion valve and of evaporator remain constant. This is due to the fact that the inlet and outlet conditions of both of the components are being fixed. In other components the exergy destruction rate varies as their inlet and outlet conditions also vary.

To understand the effect of exergy destruction rate on cycles more profoundly, the each cycle is individually taken and analysis on it is done. The parameters which are taken for the analysis of exergy destruction rates in each cycle are generator temperature and evaporator load.

#### **6.2.4.1 SINGLE EFFECT LITHIUM BROMIDE-WATER VAPOR ABSORPTION SYSTEM**

The analysis of single effect system is done by considering the exergy destruction rates in different components and how it affects the system performance as some of the cycle parameters change.

In our basic calculation the cooling load remain fixed (29.09 KW). The cooling load in the system could change within some expected range. As the cooling load largely affect the entire system performance it is important to analyze system performance in the terms of changing cooling load. Fig 6.3 shows the variations in exergy destruction rate or exergy loss in all the components of the single effect absorption system with evaporator load.

From Fig 6.3, it can be seen that for all the components, the exergy loss increases with the increase of evaporator load. As it is expected, the exergy loss is highest in generator followed by absorber. The generator in the system, is dealing with the highest temperature difference heat transfer. As the cooling load increases in order to maintain the same performance the generator heat input must be increase which results in increase in irreversibility. This increases exergy loss of the system.

Absorber exergy loss is less than that of the generator. It also increases with the increase of cooling load. The increase in absorber exergy loss is as fast as the generator exergy loss. It can be seen in the Fig 19 that the steepness of absorber and generator exergy loss line is more than rest of the component's exergy loss line. This is because of high temperature difference involved in absorber for heat transfer.

Exergy loss in condenser is much lower than exergy loss in absorber. Also the increase in the exergy loss in condenser per unit increase in the cooling load is much less than absorber exergy load increase. This can be seen from the gradual increasing graph of condenser exergy loss in Fig 6.3.

Evaporator follows the condenser exergy loss. The value of heat exchanger exergy loss also increases with the increase of cooling load. This is because of the increase load also increase the temperature difference between the cooling load area and evaporator which results in greater exergy losses.

The exergy loss in heat exchanger is also increases with the evaporator load. It has a moderate value as very high temperature differences are not involved in it's working.

The expansion valve has the least value of exergy destruction among all of the system's components. This exergy loss also increase as cooling load increases.

Fig 6.3: Variation of exergy loss with evaporator load

TABLE 6.11: Variation of exergy loss with evaporator load

SINGLE EFFECT EXERGY LOSS VS RE						
RE [KW]	GEN [KW]	ABS [KW]	COND [KW]	HX [KW]	EXPVLV[KW]	EVP [KW]
20	2.132	1.822	0.4675	0.1343	0.04568	0.373
25	2.665	2.278	0.5844	0.1678	0.0571	0.4662
30	3.198	2.733	0.7012	0.2014	0.06852	0.5594
35	3.731	3.189	0.8181	0.2349	0.07994	0.6527
40	4.264	3.644	0.935	0.2685	0.09136	0.7459
45	4.797	4.1	1.052	0.3021	0.1028	0.8391
50	5.33	4.555	1.169	0.3356	0.1142	0.9324

The exergy loss in single effect is plotted against generator temperature for different components in Fig 6.4 and Fig.6.5. The exergy loss in absorber, evaporator and heat exchanger is shown in figure 6.4 and the exergy loss in generator, expansion valve and condenser is shown in figure 6.4. It can be seen from the TABLE 6.8 that the maximum exergy loss is in generator followed by absorber. This is due to the fact that the generator and absorber are the two components of the system that deals across high temperature difference input and output conditions. As high temperature generator on one side interacts with low temperature absorber and on the other side with low temperature condenser. The condenser temperature is generally fixed at atmospheric temperature +(10 to 20) and absorber is also nearby at condenser temperature. This causes great irreversibility in generator operation and results in higher exergy loss. This irreversibility also increases as generator temperature increases as the concentration difference between strong and weak solution increases.

The absorber also interacts with low temperature evaporator and high temperature generator/heat exchanger. As the evaporator is set to very low temperature range, the exergy loss in the absorber is high. The exergy loss in the absorber increases with the increase of generator temperature as more temperature difference is created for heat to transfer. In the absorber the exergy loss is also depends on solution circulation ratio. As the circulation ratio decreases losses also decreases and if it increases, losses too increases. The only way to reduce irreversibility is to reduce the circulation ratio. This can be done by increasing the concentration of the solution in the generator. In order to achieve high concentrated solution, the temperature and pressure of the generator must decrease. With the decrease of circulation ratio, the heat input to the generator and heat output from the absorber is also increased and irreversibility can minimized.

It can also be seen from the Fig 6.4 and Fig. 6.5 that the exergy loss is nearly constant or increasing very gradually in heat exchanger and condenser. We can say that for condenser and heat exchanger the exergy loss is almost independent of the generator temperature.

On the other hand the exergy losses in evaporator and expansion valve is fairly remain constant with the increase of generator temperature. It shows that the evaporator and expansion valve work independently from the generator temperature as their input and output conditions remain unchanged with any change in generator input.

Fig. 6.4: Variation of exergy loss with generator temperature

TABLE 6.12: Variation of exergy loss with generator temperature

SINGLE EFFECT EXERGY LOSS			
VS $T_{gen}$			
$T_{gen}$ [°C]	ABS [KW]	EVP [KW]	HX [KW]
70	0.864	0.5424	0.09
75.56	1.154	0.5424	0.1
81.11	1.429	0.5424	0.11
86.67	1.692	0.5424	0.12

92.22	1.944	0.5424	0.13
97.78	2.185	0.5424	0.14
103.3	2.409	0.5424	0.15
108.9	2.61	0.5424	0.1644
114.4	2.819	0.5424	0.1993
120	3.06	0.5424	0.2133

Fig. 6.5: Variation of exergy loss with generator temperature

TABLE 6.13: Variation of exergy loss with generator temperature

SINGLE EFFECT EXERGY LOSS VS $T_{gen}$			
$T_{gen}[^{\circ}C]$	GEN [KW]	EXPVLV [KW]	COND [KW]
70	1.404	0.06643	0.56
75.56	1.604	0.06643	0.59
81.11	1.804	0.06643	0.61
86.67	2.004	0.06643	0.63
92.22	2.204	0.06643	0.65
97.78	2.404	0.06643	0.66
103.3	2.604	0.06643	0.6778
108.9	2.804	0.06643	0.6788
114.4	3.104	0.06643	0.6808
120	3.304	0.06643	0.6818

#### 6.2.4.2 DOUBLE EFFECT LITHIUM BROMIDE-WATER VAPOR ABSORPTION SYSTEM

The double effect exergy loss in different components with cooling load is plotted in Fig. 6.6 and Fig 6.7. Fig 6.6 shows the exergy loss of for the double effect cycles in both high and low pressure heat exchangers, expansion valve and evaporator. It is clear from the Fig 6.6 as well as from TABLE 6.8 that among them the maximum value of exergy loss is in evaporator. which followed by high pressure heat exchanger then low



pressure heat exchanger and the minimum value of exergy loss occurs in expansion valve. Which is, in fact the smallest exergy loss value in complete cycle.

The evaporator exergy loss and the expansion valve exergy loss remain unchanged from single to double absorption system (TABLE 6.8) as their operating condition remains unchanged. While in comparing we can see that the combined heat exchanger loss in double effect absorption system is more than that of in single effect absorption system as two heat exchangers in double effect work across wide temperature differences. Like single effect cycle, these exergy losses of heat exchangers also increase as the cooling load increases.

The variation of high pressure generator, low pressure generator, condenser and absorber in terms of cooling load is shown in Fig.6.7. the low pressure generator system is internally integrated with high pressure condenser. The cumulative exergy loss is calculated for both of them as they acts as a single unit. By taking low pressure generator and high pressure condenser cumulative exergy loss, the graph has plotted. It can be seen from the graph that the exergy loss increases as refrigerator load increases. The maximum value of exergy loss is also in the combined unit of low pressure generator and high pressure condenser. This value is followed by high pressure generator (Fig. 6.7). The pattern of exergy loss with cooling load in high pressure generator is same as that of in low pressure generator unit i.e. it also increases with the cooling load. The condenser exergy loss is greater in the double effect system than in single effect system (TABLE 6.8). The generator used in double effect system has high value of temperature than the generator used in single effect system. It increases the properties value at the inlate of condenser. As the condenser interacts with the same outside environment to dispose it's heat off, as a result exergy loss increases in condenser. And this keeps on increasing with the cooling load.

Absorber exergy load also increases with the cooling load but it is very less as compare to the single effect system.

Fig 6.6: Variation of exergy loss with evaporator load

TABLE 6.14: Variation of exergy loss with evaporator load

DOUBLE EFFECT EXERGY LOSS VS RE				
$T_{htg}[^{\circ}C]$	HPHX [KW]	LPHX [KW]	EXPVLV [KW]	EVP [KW]
20	0.1316	0.08431	0.04568	0.373
23.33	0.1535	0.09836	0.0533	0.4351
26.67	0.1754	0.1124	0.06091	0.4973
30	0.1973	0.1265	0.06852	0.5594
33.33	0.2193	0.1405	0.07614	0.6216
36.67	0.2412	0.1546	0.08375	0.6837
40	0.2631	0.1686	0.09136	0.7459
43.33	0.285	0.1827	0.09898	0.8081
46.67	0.307	0.1967	0.1066	0.8702
50	0.3289	0.2108	0.1142	0.9324

Fig 6.7: Variation of exergy loss with evaporator load

TABLE 6.15: Variation of exergy loss with evaporator load

DOUBLE EFFECT EXERGY LOSS VS RE				
$T_{htg}[^{\circ}C]$	LPG [KW]	HPG [KW]	COND [KW]	ABS [KW]
20	4.38	4.273	1.863	0.9458
23.33	5.11	4.985	2.173	1.103
26.67	5.84	5.698	2.484	1.261
30	6.57	6.41	2.794	1.419
33.33	7.3	7.122	3.105	1.576
36.67	8.03	7.834	3.415	1.734
40	8.76	8.546	3.726	1.892
43.33	9.491	9.259	4.036	2.049
46.67	10.22	9.971	4.347	2.207
50	10.95	10.68	4.657	2.365

The exergy loss of system in the terms of generator temperature is shown in Fig. 6.8, Fig. 6.9 and Fig. 6.10. Fig. 6.8 shows the exergy of high pressure heat exchanger and evaporator and the way it varies with the generator temperature. As expected the evaporator values varies in the same manner as in single effect system. There is no difference in it's variation with generator temperature. It remains constant with generator temperature.

Exergy destruction in high pressure heat exchanger is due to the larger temperature heat transfer. More the heat capacity rate mismatch, greater the temperature difference

across the heat exchanger, more the exergy loss will be. Increased generator temperature increases heat capacity difference between incoming and outgoing solutions. Therefore heat exchanger exergy loss increases with the generator temperature.

Like the evaporator, the exergy loss in expansion valve is constant through out the range of generator temperature. The fixed inlet and outlet values has fixed the exergy loss in the expansion valve.

Fig.6.9 shows the exergy of low pressure generator, high pressure generator and condenser. They all have same increasing tendency with the generator temperature. The low pressure generator comprising high temperature condenser in it, has the maximum value of exergy loss in the entire system. The main generator of double effect absorption system runs at a higher temperature (130-150 (°C)) than single effect absorption system (100-110(°C)). This higher temperature of generator increases solution concentration difference. Therefore in double effect system the loss is higher than the loss in single effect system.

Condenser in double effect system has more exergy loss than in single effect system and this loss also increases as generator temperature increases.

Absorber and low pressure heat exchanger are directly affected by low temperature generator. The variation of the change of high temperature generator is very negligible in them. Therefore the pattern of exergy loss must be find for absorber and low pressure heat exchanger by analyzing it at different low temperature generator input. Fig. 6.10 shows the exergy loss variation with low temperature generator's temperature. The absorber exergy loss increases with the generator temperature as it causes incrimination in the solution concentration difference values. More the difference between the solution concentration values, more the irreversibility will be and exergy losses will be greater in that case.

Variation in low pressure heat exchanger with the generator temperature is less steep as in the case of absorber. The less value of exergy loss in low pressure heat exchanger also shows less heat capacity rate mismatch than in high pressure heat exchanger.

Fig. 6.8: Variation of exergy loss with high temperature generator

TABLE 6.16: Variation of exergy loss with high temperature generator

DOUBLE EFFECT EXERGY LOSS VS $T_{htg}$			
$T_{htg}$ [°C]	HPHX [KW]	EXPVLV [KW]	EVP [KW]
120	0.1241	0.06644	0.5425
126.7	0.1678	0.06644	0.5425
133.3	0.2176	0.06644	0.5425
140	0.2807	0.06644	0.5425
146.7	0.3395	0.06644	0.5425
153.3	0.35	0.06644	0.5425
160	0.382	0.06644	0.5425
166.7	0.4	0.06644	0.5425
173.3	0.421	0.06644	0.5425
180	0.476	0.06644	0.5425

Fig 6.9: Variation of exergy loss with high temperature generator

TABLE 6.17: Variation of exergy loss with high temperature generator

DOUBLE EFFECT EXERGY LOSS VS $T_{htg}$			
$T_{htg}$ [°C]	LPG [KW]	HPG [KW]	COND [KW]
120	2.967	2.75	2.381
126.7	5.187	3.81	2.586
133.3	7.498	4.73	2.792
140	9.81	5.62	2.991
146.7	12.02	6.85	3.179
153.3	14.11	7.65	3.356
160	16.28	8.74	3.557
166.7	17.15	9.89	3.788
173.3	19.5	10.75	3.9
180	20.8	12.93	4.2

Fig 6.10: Variation of exergy loss with low temperature generator

TABLE 6.18: Variation of exergy loss with high temperature generator

DOUBLE EFFECT EXERGY		
LOSS VS $T_{htg}$		
	LPHX	ABS
$T_{htg} [^{\circ}C]$	[KW]	[KW]
60	0.08	0.2995
64.44	0.097	0.5577
68.89	0.1063	0.8042
73.33	0.1141	1.04
77.78	0.1203	1.266
82.22	0.1245	1.483
86.67	0.1269	1.692
91.11	0.1273	1.894
95.56	0.128	2.09
100	0.129	2.278

#### 6.2.4.3 HALF EFFECT LITHIUM BROMIDE-WATER VAPOR ABSORPTION SYSTEM SYSTEM

The graphs for half effect has plotted in Fig. 6.11-6.15. Exergy loss with respect to evaporator for all the main components is shown in Fig.6.11 and Fig. 6.12. Fig. 6.11 shows the exergy loss in low pressure heat exchanger, high pressure heat exchanger and expansion valve. It can be seen that the value of exergy loss in low pressure heat exchanger is more than the value of exergy loss in high pressure heat exchanger. The main reason for exergy loss in heat exchanger is the difference between solution capacity rate between incoming and outgoing solution circuit. This exergy loss increases as the cooling load increases. The value of exergy loss in heat exchanger is fairly low than any other system (TABLE 6.8). This is because the working temperature associate with half cycle is the lowest among all the other system. As the cooling load increases, heat exchanger exergy loss also increases.

As expected there is no change in the exergy loss of expansion valve. It remains constant for all the values of cooling load. The loss in expansion valve, like any other cycle, is the minimum among all the components. Exergy loss of low pressure

generator, high pressure generator, low pressure absorber, high pressure absorber, condenser and evaporator with variable cooling load values is plotted in Fig 6.12. It can be seen from the figure that the exergy loss in low pressure absorber is greater than the loss in high pressure absorber. Both of the values increases with the increase of evaporator load.

Half effect system uses two absorber and two generator. Exergy loss in low pressure absorber is more followed by high pressure absorber. The exergy loss in high pressure generator is equal to the exergy loss in high pressure absorber and it varies in the same manner as high pressure absorber's exergy loss varies with the evaporator load. This value of exergy loss is followed by the loss in low pressure generator. Which also increases as the cooling load increases. Exergy loss values in evaporator and condenser are very close. Both vary gradually with the evaporator load as compare to others.

Fig 6.11: Variation of exergy loss with evaporator load

TABLE 6.19: Variation of exergy loss with evaporator load

HALF EFFECT EXERGY LOSS VS RE			
RE [KW]	LPHX [KW]	HPHX [KW]	EXPVLV [KW]
20	0.0629	0.01027	0.04568
23.33	0.07338	0.01198	0.0533
26.67	0.08386	0.01369	0.06091
30	0.09435	0.0154	0.06852
33.33	0.1048	0.01711	0.07614
36.67	0.1153	0.01882	0.08375
40	0.1258	0.02053	0.09136
43.33	0.1363	0.02224	0.09898
46.67	0.1468	0.02395	0.1066
50	0.1572	0.02566	0.1142

Fig 6.12: Variation of exergy loss with evaporator load

TABLE 6.20: Variation of exergy loss with evaporator load

HALF EFFECT EXERGY LOSS VS RE						
RE	LPA	HPG	COND	LPG	EVP	HPA

[KW]	[KW]	[KW]	[KW]	[KW]	[KW]	[KW]
20	2.102	1.913	0.3306	1.353	0.373	1.913
23.33	2.452	2.232	0.3857	1.579	0.4351	2.232
26.67	2.802	2.551	0.4409	1.804	0.4973	2.551
30	3.152	2.87	0.496	2.03	0.5594	2.87
33.33	3.503	3.189	0.5511	2.255	0.6216	3.189
36.67	3.853	3.508	0.6062	2.481	0.6837	3.508
40	4.203	3.827	0.6613	2.706	0.7459	3.826
43.33	4.554	4.145	0.7164	2.932	0.8081	4.145
46.67	4.904	4.464	0.7715	3.157	0.8702	4.464
50	5.254	4.783	0.8266	3.383	0.9324	4.783

The variation of exergy loss of half effect cycle with respect to generator temperature has shown in Fig. 6.13- 6.15. Low pressure heat exchanger, high pressure heat exchanger and expansion valve exergy loss can be seen in Fig. 6.13. As expected the expansion valve exergy loss is of constant value and does not change with generator temperature until unless we change the condition of it's inlet and outlet, which is fixed in our calculation. Exergy loss in low pressure heat exchanger is greater rthan the exergy loss in high pressure heat exchanger. The major reason for this loss is the difference between heat carrying capacity of solutions going through heat exchangers.

The loss of exergy in the generators can be seen in Fig. 6.14. Both, high and low pressure generators exergy loss increases with the increase of generator temperature. The increased temperature of generator increases solution concentration difference. The increased generator also increase temperature difference for heat transfer between generator and condenser and generator and absorber. These all reasons eventually increase the irreversibility.

The low pressure absorber having the highest value of exergy loss in the cycle, shows step increase with the generator temperature. Same trend can be seen for high pressure absorber. The loss of exergy in the absorber is due to the large difference between solution concentration which increases as the temperature of the absorber increases.(Fig6.15)

Fig 6.13: Variation of exergy loss with generator temperature

Fig 6.21: Variation of exergy loss with generator temperature

HALF EFFECT EXERGY LOSS VS $T_{gen}$			
$T_{gen}$ [°C]	LPHX [KW]	HPHX [KW]	EXPVLV [KW]
40	0.036	0.004341	0.06644
45.56	0.046	0.00547	0.06644
51.11	0.056	0.008241	0.06644
56.67	0.066	0.01085	0.06644
62.22	0.067	0.0128	0.06644
67.78	0.0752	0.01404	0.06644
73.33	0.0878	0.0147	0.06644
78.89	0.09316	0.01493	0.06644
84.44	0.096	0.015	0.06644
90	0.1036	0.01621	0.06644

Fig 6.14: Variation of exergy loss with generator temperature

TABLE 6.22: Variation of exergy loss with generator temperature

HALF EFFECT EXERGY LOSS VS $T_{gen}$		
$T_{gen}$ [°C]	LPG [KW]	HPG [KW]
40	1.28	1.32
45.56	1.36	1.56
51.11	1.45	1.73
56.67	1.57	1.95
62.22	1.65	2.13
67.78	1.76	2.32
73.33	1.87	2.5
78.89	1.98	2.65
84.44	2.16	2.85
90	2.25	3.05

Fig 6.15: Variation of exergy loss with generator temperature



TABLE 6.23: Variation of exergy loss with generator temperature

HALF EFFECT EXERGY LOSS VS $T_{gen}$				
$T_{gen}$ [°C]	LPA [KW]	HPA [KW]	EVP [KW]	COND [KW]
40	0.6906	1.321	0.5425	0.409
45.56	1.104	1.639	0.5425	0.4109
51.11	1.473	1.852	0.5425	0.4209
56.67	1.816	2.039	0.5425	0.4309
62.22	2.137	2.219	0.5425	0.4409
67.78	2.441	2.396	0.5425	0.4509
73.33	2.729	2.573	0.5425	0.4609
78.89	3.003	2.748	0.5425	0.4709
84.44	3.268	2.921	0.5425	0.4909
90	3.519	3.09	0.5425	0.509

In the calculation the refrigeration load is fixed. In order to operate a solar vapor absorption system it is necessary to relate the heat required to run the system to the required cooling demand. Heat required to the system, in solar powered system is from the solar collectors. Therefore required cooling load must be analyzed with solar panel area. Fig. 6.16 shows the variation in solar panel areas with the cooling effect required. As half effect requires the maximum energy input among all the cycles (TABLE 6.8), the required solar panel area for half effect is also maximum. Double effect utilizes minimum heat among all to run the cycle therefore area required to operate double effect system is also minimum among all. Single effect system has moderate value of heat input, therefore it settles down for moderate solar panel area as well. It can directly be seen from the figure, as the refrigeration load increases, the required area of solar panel also increases. This is because to ensure the same performance generator heat input must be increased as evaporator load increases.

Fig. 6.16: Variation of solar panel area with evaporator load

TABLE 6.24: Variation of solar panel area with evaporator load

AREA VS RE			
RE	Half Effect	Single	Double

		Effect	Effect
20	61.45	34.88	21.77
25	71.69	43.6	25.39
30	81.93	52.31	29.02
35	92.17	61.03	32.65
40	102.4	69.75	36.28
45	112.7	78.47	39.9
50	122.9	87.19	43.53

## CHAPTER 7: CONCLUSION

In this thesis absorption refrigeration systems of half effect, single effect and double effect have been analyzed and cooling load is calculated for mechanical engineering department seminar room. The COP, exergetic efficiency, heat transfer rates and exergy destruction rates are also computed, compared and their variation with generator temperature and refrigeration load is plotted to understand cycle performance and their functioning. Following results is concluded from the thesis:

- I. It is observed that COP of cycle increases from half to double effect as heat required to the generator decreases. The value of COP for half effect, single effect and double effect vapor absorption refrigeration system is 0.45, 0.74 and 1.2 respectively.
- II. For variable evaporator load, it has been observed that if generator temperature increases, the COP of vapor absorption refrigeration system increases as increasing generator temperature increases mass flow rate of refrigerant in the evaporator.

- III. The second law efficiency (i.e exergetic efficiency) also increases in the same manner as the COP. The value of second law efficiency for half effect, single effect and double effect vapor absorption system is 15.58%, 19.27% and 26.3% respectively.
- IV. Maximum exergy loss is found in generator and absorber and it increases with generator temperature and evaporator load. In the evaporator and expansion valve exergy loss remains constant for all vapor absorption refrigeration system.
- V. The solar collector panel area required increases as cooling load increases. Required solar collector panel area for half effect, single effect and double effect vapor absorption refrigeration system is 89.37 m<sup>2</sup>, 50.72 m<sup>2</sup> and 31.66 m<sup>2</sup> respectively.

## **SCOPE FOR FUTURE WORK**

- Analysis of the system can be done by using various low grade heat sources like geothermal, industrial waste energy sources.
- Thermo economic analysis can be done on the absorption system to obtain optimum performance with best economy.
- COP values can be increase by using more thermal resistant material in generator construction.
- New solar technologies can be provided to store more thermal energy inside the solar collector at minimum expense.

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