Design And Development of Small Capacity Vapor Absorption Refrigeration System

By

Abhijit. S. Musale.

11MMET08

DEPARTMENT OF MECHANICAL ENGINEERING

AHMEDABAD-382481

May 2013

Design And Development of Small Capacity Vapor Absorption Refrigeration System

Major Project Report

Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering (Thermal Engineering)

By

Abhijit. S. Musale.

(11MMET08)

Guided By

Prof. N. K. Shah (IT, NU)

Dr. V. Shrinet (ERDA)

DEPARTMENT OF MECHANICAL ENGINEERING

AHMEDABAD-382481

May 2013

Declaration

This is to certify that

1. The thesis comprises my original work towards the degree of Master of Technology in Thermal Engineering at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgment has been made in the text to all other material used.

Abhijit.S.Musale

11MMET08

Undertaking for Originality of the Work

I, **Musale Abhijit Shirish**, Roll No. **11MMET08**. give undertaking that the Major Project entitled **"Design and Development of Small Capacity Vapor Absorption Refrigeration System."** Submitted by me, towards the partial fulfillment of the requirements for the degree of **Master of Technology in Mechanical Engineering (Thermal Engineering)** of Nirma University, Ahmadabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

____
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Date : ____________________
Place : NU, Ahmedabad.

____
Endorsed by:

Institute - Guide
Prof. N.K.Shah
Assistant Professor and Guide,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmadabad

Industry - Guide
Dr. V. Shrinet
Co- Guide, Dy. Director & Head (R&D)
Electrical Research and Development
Association (ERDA),
Makarpura GIDC,
Vadodara.

CERTIFICATE

This is to certify that the Major Project Report entitled **"Design and Development of Small Capacity Vapor Absorption Refrigeration System"** submitted by **Mr. Abhijit S. Musale (11MMET08)**, Towards the partial fulfillment of the requirements for the award of Degree of Master of Technology in Mechanical Engineering (Thermal Engineering) of Institute of Technology, Nirma University, Ahmadabad is the record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination. The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institution for award of any degree.

Prof. N.K.Shah
Assistant Professor and Guide,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmadabad

Dr. V. Shrinet
Co- Guide, Dy. Director & Head (R&D)
Electrical Research and Development
Association (ERDA),
Makarpura GIDC,
Vadodara.

Dr. R. N. Patel
HOD,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmedabad.

Dr K Kotecha
Director,
Institute of Technology,
Nirma University,
Ahmedabad.

Acknowledgments

With immense pleasure, I would like to present this report on the dissertation work related to **"Design and Development of Small Capacity Vapor Absorption Refrigeration System."**. I am very thankful to all those who helped me for the successful completion of the dissertation and for providing valuable guidance throughout the project work.

I offer my sincere and heartfelt gratitude to **Dr. V.Shrinet**, Dy. Director and Head (R&D), Electrical Research and Development Association (ERDA), Vadodara, for his valuable guidance, motivation and encouragement. I would also like to express my deepest gratitude to **Shri U. N. Puntambeker** and **Shri K. D. Chudasama** for their active and continuous help to provide required materials and facilities for the project work.

I grab the opportunity to express my sense of gratitude and indebtness to **Shri T. P. Govindan**, Director, Electrical Research & Development Association (ERDA), Vadodara, for giving me the opportunity to carry out my dissertation work at ERDA and for allowing me to be a part of that reputed research center.

I truly believe that my work would not have been easier without the helps and co-operations rendered by **Shri Pravinbhai Rathod, Shri S.K.Sing, Shri Vijaybhai Baria, Shri Amit Chauhan** and others.

I would like to give my special thanks to my Faculty Guide, **Prof N.K. Shah**, Assistant Professor, Department of Mechanical Engineering, Institute of Technology, Nirma University, Ahmadabad for his valuable guidance and continual encouragement throughout the Major project.

My sincere thanks and gratitude to **Dr. V. J. Lakhera**, PG co-ordinator of thermal group and professor, Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad and **Dr. R. N. Patel**, Head of Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad for their continuous encouragement and motivation throughout the dissertation work. I also Thank the faculty members of Mechanical Engineering Department for their support during this work.

I would also like to thank **Dr. K. Kotecha**, Director, Institute of Technology, Nirma University for allowing me to carry out my project work in industry. I am thankful to Nirma University for Providing all kind of required resources.

I would like to express my gratitude to my parents, my family members and my friends for their support and encouragement. At the end I would like to express my deepest gratitude to almighty God for making this dissertation work one of the most important and valuable experience of my life.

Abhijit S. Musale

11MMET08

Abbreviation

WSST	=	Weak Solution Storage Tank
SSST	=	Strong Solution Storage Tank
RST	=	Refrigerant Storage Tank.
VAR	=	Vapor Absorption Refrigeration
VCR	=	Vapor Compression Refrigeration
COP	=	Co-efficient Of Performance
24H-SVAR	=	24 Hour Solar Vapor Absorption Refrigeration.
TR	=	Tonn of Refrigeration
AST	=	Apparant Solar Time
LST	=	Local Standard Time

Nomenclature

Q	=	Heat Transfer (kW)
T	=	Temperature (^oC)
P	=	Pressure (bar)
H	=	Enthalpy (kJ/kg), Hour Angle
M	=	Mass Flow Rate (kg/s)
M_{AW}	=	Atomic weight (gm/mol)
S	=	Concentration of LiBr in Solution (%)
f	=	Friction Factor , Mass of solution per kg of refrigerant vapor
$\triangle P$	=	Pressure Difference (N/m^2)
ρ	=	Density (kg/m^3)
g	=	Gravitational acceleration ($9.81\,m/s^2$)
P_{ow}	=	Pumping Power (kW)
σ_t	=	Tensile Stress (psi)
σ_y	=	Yeild Stress (psi)
σ	=	Surface Tension (N/m)
k	=	Thermal conductivity W/mK
do	=	Outer Diameter (mm)
di	=	Inner Diameter (mm)
D_{hyd}	=	Hydraulic Diameter (mm)
μ	=	Dynamic Viscosity ($Pa.s$)
v	=	Kinematic Viscosity (m^2/s)
Cp	=	Specific Heat (kJ/kg)
α	=	Thermal Diffusivity (m^2/s)
$A_{c/s}$	=	Cross Sectional Area (m^2)
A_{HT}	=	Heat transfer Area (m^2)
V	=	Velocity (m/s)
Re	=	Renold Number
Pr	=	Prandlt Number
Nu	=	Nusselt Number
Gr	=	Grashoff Number
Ra	=	Rayleigh Number
hi	=	Heat Transfer co-efficeient inside pipe (W/m^2K)
ho	=	Heat Transfer co-efficeient outside pipe (W/m^2K)
h	=	Water Column Head (m)
Le	=	Effective length (m)
L_{pipe}	=	Calculated Length of Pipe
U	=	Overall Heat transfer co-efficient.(W/m^2K)
$LMTD$	=	Logarithmic Mean Temperature Difference
I_{DN}	=	Direct Norma Radiation (W/m^2)
$I_{d\theta}$	=	Amount of Diffuse Radiation (W/m^2)
$I_{t\theta}$	=	Total Solar Radiation (W/m^2)
q_u	=	Usable Solar Energy obtained from collactors

Abstract

Vapor absorption refrigeration methods are usually used to take care of bigger cooling loads ranging from 10TR to 3500TR according to which their size and construction changes. Another convention about such systems is that they use waste heat or in some cases directly fuel to satisfy the requirement of heat supply in generator. In present work Lithium Bromide and water based vapor absorption refrigeration system has been designed which can use solar energy as an heat input and can work 24 hours even in absence of Sun hours and can produce 1TR refrigeration. Various ways of energy storage for the system to work in absence of Sun have been studied in literature review and out of them refrigerant storage method have been chosen. For solar operated 24hrs VAR system the excess amount of refrigerant that is required for after sun hours operation of system, is being produced by condenser and generator which is stored in refrigerant storage tank. Strong and weak solutions of LiBr-water required after sun hours for absorber use, are stored in two different tanks. Thus system is designed in such a way that after sun hours sufficient amount of refrigerant is available to produce cooling effect.

Three types of system have been designed such as, system which is completely based on solar energy, hybrid system which is working on solar energy in day time and on fuel gas in night time, and system which is completely based on gas fuel. Based on design in present work the theoretical COP for solar based VAR system is 0.67 and for hybrid system and fuel based system it is 0.87. The purpose behind design for this various options is to observe the impact on cost and size of the system. Simple cost analysis carried out for all three types of VAR systems with compare to VCR system working 24 hours and 365 days. It was found that hybrid VAR system and fuel based VAR system are not economic compared to VCR system. Solar based VAR system is economically more viable and has simple payback period of 2.5 years.

Prototype of LPG fuel based VAR system has been fabricated to check technical feasibility of development of small capacity VAR system. Best efforts have been made to make the design based on right data and comply with standards and codes such as ASHRAE and ASME.

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Chapter 1

Introduction

Vapor absorption cooling has some very attractive features such as it can work with the help of heat instead of electricity, has no moving part except one pump, easy to maintain, noise free operation, environment friendly, sturdy construction and durability and most attractive of all in which we are interested is it can use solar energy directly to operate.

Vapor absorption technology is nearly more than a century old. Absorption cooling is said to have been invented by french scientist Ferdinand Carre in 1858. Since then many modification of the systems have been developed, many patents have been filed for this system, inverters have used various arrangements and chemical pairs for this system, and still it is area of interest for engineers even today. Mostly because of the solar energy use in the Absorption Refrigeration promises some good outcomes. Such as solar based absorption refrigeration can provide refrigeration in the ruler areas of the country where the electricity grid is not available or not reachable. Even in cities it could be helpful to reduce the cooling load on electricity. According to an estimation by The International Institute of Refrigeration (IIR), 15% of total electricity produced is used for the refrigeration or air conditioning purpose.[21]

1.1 Solar Energy

Solar energy which is a free energy, abundantly available and easily accessible can be most useful to run this type of system. Solar Energy is already being successfully used in residential and industrial settings for cooking, heating, cooling, lighting, space technology, and for communications among other uses. In fact, fossil fuels are also one form of solar energy stored in organic form. With fossil fuels making major impact on the environment and raising issues of pollution and global warming, solar energy has increased in its importance to industries and homes. While the reserves of fossil fuels are restricted, there is no limitation to the availability of solar energy.With improvement in solar energy technology and the increase in prices of fossil fuel, solar energy is gradually becoming more and more affordable. In addition, there is additional cost in the form of importation and transportation, required for oil, coal and gas.

On a surface on the Earth's orbit, normal to the sun, solar radiation hits at the rate of 1,366 Watt Per Meter Square. This is known as solar constant. While 19% of this energy gets absorbed in the atmosphere, 35% gets reflected by clouds. So, the solar energy that reaches sea level is much reduced.[21]

1.1.1 Advantages of Solar Energy

Solar Energy has more advantages than the disadvantages. Once the initial cost of installation is met, the electricity generated by solar panels is free of cost. In a stand-alone solar power system you don't have to pay any utility bills. Another positive in installing solar power systems is that government offers lots of rebates and incentives to cover the initial cost. You can also sell the additional electricity generated by your system. Sometimes the utility company will give you credits for selling the excess amount of electricity.

Another good thing about solar power is that the cost of the technology is decreasing almost every few months and the efficiency is improving significantly. Today, you can find different types of solar solutions that are more convenient to install. Solar Energy is a renewable and clean source of energy and you don't have to pay any transmission cost. This is because the energy would be produced and consumed at the same place.

Depending upon your budget, you can get all or a part of your electricity requirements fulfilled by solar energy. When batteries are used in the system to store electricity you can become entirely independent of the grid. This also means that you don't have to get bothered by power failures in the grid, as you would be able to enjoy seamless supply of electricity.

1.1.2 Disadvantages of Solar Energy

Even though Solar Energy has several disadvantages, they are not major in degree and the benefits of this renewable source of energy far outweigh them.

The initial cost of the installation and equipment is high. When you compare fossil fuel technology with solar power technology, the former will seem to be far more affordable. Even though Solar Energy is being used at an increasing rate, the initial costs don't encourage the maximum users to switch to this renewable source of energy.

If there is shortage of space for installing solar panels, you would not be able to generate sufficient amount of Solar Energy required to meet your electricity requirements.

Considering all these factors, it doesn't require much judgment to see that solar energy is the mainstay of energy requirement in both households and industries, both in the future and in the next few years. It is only a matter of time before Solar Energy technology becomes easily affordable to make this untapped and unending source of energy a common phenomenon in every household.

1.2 Vapor Sorption Technology For Cooling

Even though vapor absorption technologies are much more older than the vapor compression refrigeration technologies their development is not much because of the fast developments of vapor compression technology. Vapor compression technology has some good advantages such as its size is much smaller for the same cooling load that is being handled by absorption machine. And it is much more economical, having much less initial cost and easy installation and maintenance. But days are coming nearer when the world is going to face energy crises and in those days conventional fuels are going to be much more expensive than today. Even the electricity will also become expensive since it is much depending on fossile fuels such as coal and petroleum. And vapor compression system needs electricity to run the compressor which is provided by running an electric motor.

1.2.1 Comparison Between Vapor Absorption and Vapor Compression Technology.

(1) Vapor absorption refrigeration (VAR) system can use low grade energy directly such as wast heat, exhaust of IC engine etc. where as VCR uses high grade energy such as electricity.

(2) VAR system has only one moving part pump, which has very less electrical power consumption because it handles liquid, and so operation is smooth and noise free. Where as VCR system works with help of compressor, which requires more electricity beacuse it handles refrigerent vapor, and so operation is very noisy.

(3) VCR system has less components then the VAR system. VCR system has only four components namely evaporator, compressor, condenser and expansion device. where as in VAR system compressor is replaced by four other components and they are absorber, pump, generator and pressure reducing device.

(4) VCR system uses just one working fluid refrigerant, whereas VAR system has to use minimum two working fluids namely absorbent and refrigerant. However in case of adsorption there is just refrigerant as a working fluid and the adsorbing media could be any porous materials like silica gel and zeolites.

(5) VCR systems are small, economic and can give higher COP where as VAR systems are bulky, expensive and gives less COP with compare to VCR systems. But still in some cases VAR systems are used over VCR systems because of ability of VAR systems to use low grade energy. In some industries many times exhaust flue gas of boiler or furnace have completed its potency and thoroughly used, then such hot gases can be used to produce cooling effect before just throwing them out in the atmosphere.

(6) Even though VCR technology is more affordable, in some ruler areas where the electricity grid is not available VAR technology can work with some consumption of coal, wood or even biomass. In such remote places where cooling is required in some essential applications such as storage of medical drugs, milk dairy products, cold storage to keep the vegetable fresh.

(7) For smaller cooling loads VCR technology may be good option but for bigger cooling load such as 3500TR VAR technology is better option. because to handle more amount of

refrigerant bigger compressors are required and they are very expensive and maintenance seeking.

(8) In VCR system problem of washing of lubricant oil with refrigerant is well known. This oil mixes with refrigerant and goes in to the condenser and further piping and clots in the expansion valve. This problem have been solved with the use of the special lubricant oils which does not evaporate with refrigerant vapor. In cases of VAR system where the liquid absorbent is used some times in generator absorbent also evaporates with the refrigerant and condenses in the condenser. Ammonia and water absorption system is example of this where water being absorber evaporates with ammonia in generator and condenses in condenser and forms ice in expansion device by blocking it. To separate water from the ammonia special equipment called rectifier is used whose structure is much like distillation column of the petroleum.

1.2.2 Principle of Operation of Vapor Absorption Refrigeration

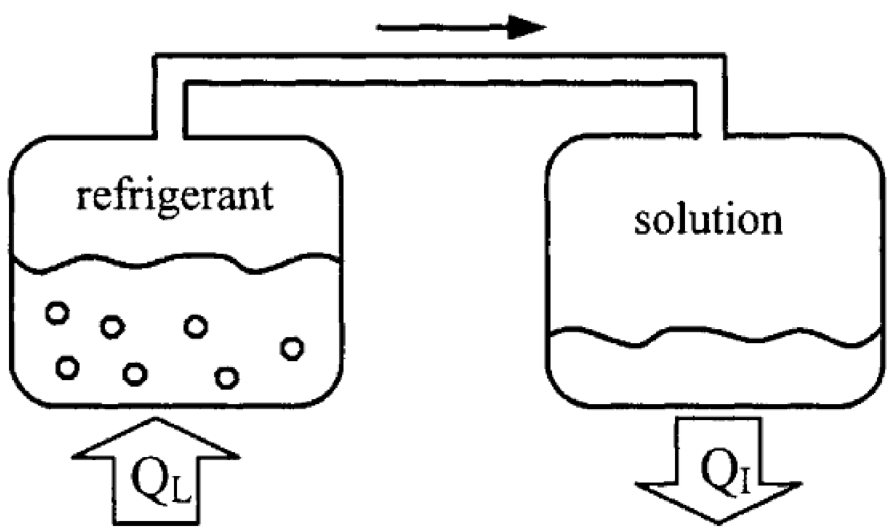

Figure 1.1: Principle of operation of Evaporator and Absorber.[22]

In Figure 1.1 two pressure sealed vessels are shown and they are lower pressure then the atmospheric pressure. In left vessel a pure liquid refrigerant is filled which is in our case we can consider water. And in right vessel a binary solution is filled of which refrigerant is one part. Now these two vessels are connected with leak proof duct. Refrigerant vapor lies in the upper empty part of both the vessels and the duct. Initially there exist equilibrium in this system. The binary solution is highly concentrated is at atmospheric temperature. So, at atmospheric temperature it has the tendency to absorb more refrigerant vapor from the empty space above and make it self dilute. Now when this vapor is absorbed by the solution pressure drops in both the vessels. In the left vessel where the refrigerant is there comes a change in the partial pressure of the vapor at that temperature so to balance that pressure

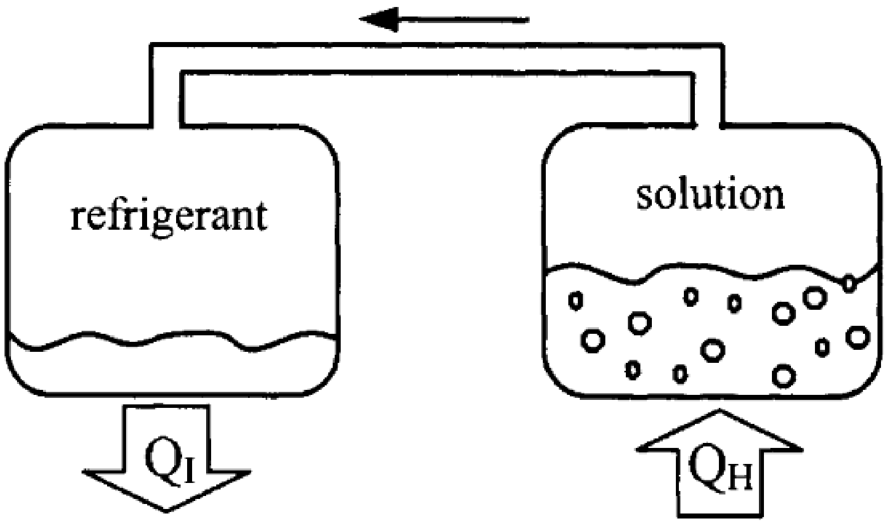

Figure 1.2: Principle of operation of Generator and Condenser.[22]

gap more evaporation of the refrigerant takes place. When this vapor evaporates to balance the pressure, it takes heat of evaporation from its surroundings and there by it creates the cooling effect. evaporated vapor is again absorbed by the solution and again more refrigerant evaporated to balance the pressure by creating cooling effect as an out come. This process goes on until the solution becomes dilute and unable to absorb any more refrigerant. To keep this process continuously working we must keep the concentration of the solution constant, which can be arranged by continuously adding the strong binary solution and removing dilute binary solution. While solution absorbs refrigerant vapor its temperature rises because of the chemical reaction this additional heat we must take away with the help of cooling water otherwise higher temperature will cause solution to stop absorbing more refrigerant vapor which will be the cause to stop refrigeration effect. The example of this two vessels explains the working principle of evaporator and absorber which are components of actual working systems.

In Figure 1.2 there are same two pressure sealed containers which are joined by duct, but now the process is opposite. This time the vessel on right side is filled with a dilute solution. Now, we are supplying heat to this solution to heat it. Since this is binary solution only refrigerant evaporates and the other component which is known as absorbent remains in the vessel and increases the concentration of the solution. Evaporated vapor goes to the vessel on the left side where with the help of cooling water heat is being rejected. The refrigerant vapor cools down and condenses. This way refrigerant is being separated from the binary solution. Pressure in both the vessels depends on the temperature of cooling water. This example of two vessels explains the working principle of Generator and Condenser.

1.2.3 Types of Vapor Absorption Systems

Some chemicals such as water has strong affinity for certain chemical such as ammonia. Water can absorb ammonia in good quantity, similarly some salts like LiBr can absorb water. Some organic chemicals such as methanol can be adsorbed by some porous material such as activated carbon or silica gel. This characteristic phenomena is base for the development of the sorption technologies. Mainly there are three kind of sorption processes.

(1) Absorption of liquid refrigerant in to liquid absorbent. Ammonia and water pair is a example of such cases.

(2) Absorption of liquid refrigerants in solid salts. Lithium Bromide and water pair is example of this type.

(3) Adsorption of refrigerant vapor in to some porous media such as silica gel, zeolites, activated carbons etc. Absorption and Adsorption processes together referred as Sorption processes.

Apart from these there are other types such as Single Effect Vapor absorption refrigeration system, Multi Effect Vapor absorption refrigeration systems, Half effect VAR, VAR with absorber heat recovery, Combined absorption and compression cycle, Dual cycle absorption refrigeration, Combine ejector absorption refrigeration cycle. etc. and there are various applications that are using vapor absorption technology are mentioned in chapter of literature review.

Chapter 2

Literature Review

As mentioned earlier vapor absorption technology is an old technology and much work has been done on it to serve various applications. Some of researchers have already made use of solar energy to use in absorption refrigeration systems. Paper from S. B. Riffat [17] gives information on the use of solar energy in absorption refrigeration to get the desalinated water. It is an open cycle system. The evaporator is supplied with brine or impure water which is to be pureified, at low pressure. Now because of the low pressure in evaporator this water evaporates and goes to the absorber, leaving behind concentrated waste water. From absorber to condenser the circuit is same. Water vapor is absorbed by LiBr or any other absorber that can be used in place of LiBr. Solar heat is used in generator to separate water from absorbent. When this water vapor is condensed in condenser we have pure desalinated water. Which can further be used in various applications such as working fluid in power plant, feed water in boilers, in food industries, pharmaceutical purposes etc.

Paper by Dr. R.E.Critoph [18] gives information about one of the structure of adsorption refrigeration system. Adsorption is different than that of absorption. Here ammonia is used as a refrigerant and activated carbon is used as an adsorbent. The ammonia vapor accumulates on the surface of the porous carbon and this effect is called adsorption. Ammonia does not mix with the carbon chemically like other absorption process pairs, but only gets accumulated on the surface. It is mentioned in this paper that the monolithic carbon adsorbent-aluminums composites can give 1kW cooling effect for one Kg of carbon. One UK based company named Sutcllife Speakman Carbon have achieved the technology to make the solid blocks of any shape of almost any type of carbon. The idea is to use the circular monolithic carbon plates with a hole in the center as adsorber. The paper gives the dimensions of the plates as 100mm diameter, 15mm thick and having a central hole of 25mm. These plates are staked together and to be put in one metal pipe. The holes in the center of the plates coinsides forms one passage through which the refrigerant can pass. This kind of system gives the cooling effect in the absence of the heat source, and if that heat source is sun than it will give the cooling effect in night. During the day time vapor evaporates from the porous disks and enters the passage between them and goes to condenser to get condensed and thus refrigerant accumulates in the storage tank. In the night time pressure in the system is reduced and the vapor in evaporator evaporates producing cooling effect and comes through the same passage to get absorbed in the carbon disks. This kind of system has to be provided two sets of this type of pipes in order to enable it work continuously. Heat source could be other than the solar. If hot steam or flue gases are used, than one of these pipes is to be kept in the hot stream and the other in the atmospheric temperature. So that when one is absorbing the refrigerant the other would be desorbing refrigerant, and the process can work continuously.

Paper by D. I. Tchernev [19] gives the application of zeolites in the adsorption processes. It is mentioned that the zeolites can adsorb gases or vapor of the refrigerants up to its 30% weight. Zeolites-water combination gives most efficient system and requires small quantity of zeolites for the process. Detail explanation is given in the paper about working of system. Zeolites during the day time absorb the solar heat and desorb the water vapor. Once the pressure of water vapor has reached to the condenser pressure it is cooled in the condenser and then it is stored in the storage tank. And now during night time the zeolite gets cooled and so the pressure in the vessel reduces and the liquid in the storage tank begins to evaporate and produces the cooling effect. During the whole night the zeolites adsorbs the vapor and by the morning it is ready start the new cycle.

A papers by Said, S.A.M., El-Shaarawi, M.A.I., and Siddiqui, M.U. [24] gives various methods by which solar energy can be used for 24 hour working absorption system. These authors have suggested mainly four methods to make the system work for 24 hours. According to them the methods have been classified in two different types. Continuous based operation and intermittent based operation. According to the discription given in the paper for these two systems continuous based systems are those in which vapor production in the generator and the cooling produced in the evaporator take place simultaneously and intermittent operation means vapor production and the cooling effect does not take place simultaneously. In intermittent systems vapor generation take place in the day time and the cooling take place in the night time. Which feels like absorption process and adsorptions process discussed in the earlier papers.

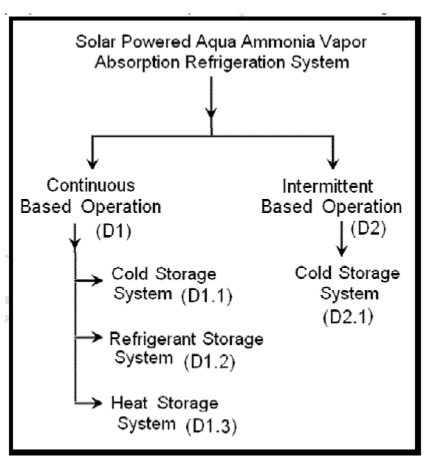

Figure 2.1: Solar energy storing methods for 24 hour working system[24]

They have given three methods for the continuous system and one for the intermittent systems. These three methods are (1) cold storage method, (2) refrigerant storage method,

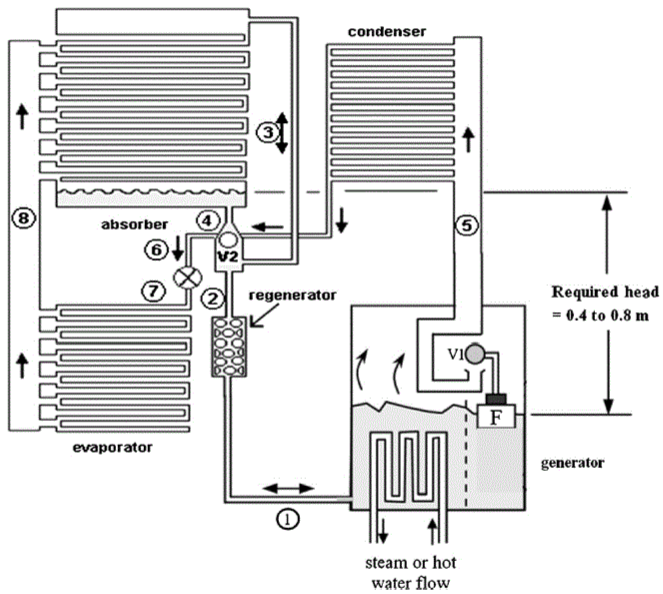

Figure 2.2: Concept of Thermo Gravity Pumping[20]

and (3) heat storage method. (1) in cold storage method the continuous cold storage system operates at the daytime and produces the required refrigeration effect for the daytime as well as store the additional cooling effect in a cold storage unit. The cold storage unit then fulfills the cooling requirement at the night time when the solar energy is not available. This absorption system remains in operation at the day time and cease to operate during the night time. (2) In refrigerant storage method mass flow in condenser is kept higher than the mass flow in the evaporator. During the day time when the cycle is operational condenser condenses additional amount of refrigerant to store in the refrigerant storage tank. This tank stores the refrigerant that is to be used for the operation of the night time. With the refrigerant tank there is also need for a tank for the storage of the absorbent and refrigerant solution because during night time the generator and condenser won't work. (3) in the heat storage system solar heat is stored in different tank with the help of heat solution. It is also mentioned about this system that this need the solar panels which can work at higher temperatures. And the storage tank has to be highly insulated. So this is rather expensive option. On the other hand for the intermittent system there is only one method of cold storage system is given.

A novel concept can be found in the paper by A. Paurine, G.G. Maidment, I.W. Eames, J.F. Missenden.[20] This paper gives information about very novel concept of thermo-gravity pumping mechanism. By applying this system in the water lithium bromide system we can eliminate the need of using the pump in the system. In this entire system no pump is used to deliver the fluid from the low pressure to the high pressure side.

How this system works and what are the different components of this system is explained in detail in the paper. In normal system the fluid flows from absorber to the generator with the help of pump. But here in this system it flows through the pipe 1,2,3,4 shown in the figure. The flow can be in both directions. It depends on the hydrostatic head that

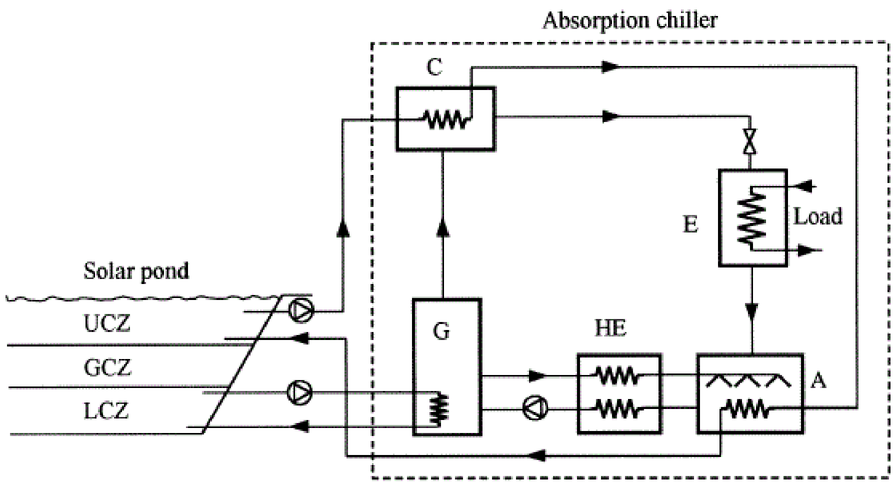

Figure 2.3: VAR system based on solar pond.[23]

is gravity and the pressure difference between the absorber and the generator. The flow from absorber to generator is achieved by the gravity. this height difference is around 0.4m and tests and experiments have shown that it is less likely to be more than 0.8m. The structure of valve V1 and V2 plays also very important role in the system. Initially when the generator has the rich solution the valve v1 remains open and allowes the vapor to pass to the condenser. After some time when considerable amount of water is evaporated the water level in generator drops causing the valve v1 to close. Because of the valve v1 is closed and heat is still being supplied more vapor is generated which increases pressure in generator. This pressure causes the week solution flow back through line 1 – 2 – 3 back to the absorber. Now the moment this pressure gets weaker than the hydrostatic head, valve V2 opens and the strong solution from the absorber flows to the generator with the help of gravity. And that's how the cycle keeps running. This process cannot be called continuous process like the other conventional systems in which pump is used. Here generator pressure keeps changing so the heat required to evaporate the water also keeps changing, also the consecration of the solution keeps changing with time. In conventional system, in steady state we can assume that the concentration of solution in generator and the absorber does not change with respect to time. Because there is continuous supply of the weak solution to the absorber and strong solution to the generator.

A Paper by Z.F. Li and K. Sumathy [23] gives detail about the use of solar pond in vapor absorption refrigeration technology. If solar pond could provide sufficient hot fluid and the cooling water then this system can work the same way as it works for the power plant. Solar ponds are also used to produce the steam and run small capacity power plant. In solar ponds there are more than one layer of water of different concentration. The bottom of the solar ponds are painted black and so by collecting solar radiation the bottom gets heated. This hot bottom floor heats the water in its immediate contact. This water in its immediate contact does not go up and mix with the colder water on the surface. Upper layer of cold water does not let the heat of hot water go in to the air and so the temperature of the brine at the bottom of the lake increases. The heat from hot brine can be used in generator to separate the refrigerant from the absorber. The cold water in the upper layer can be used as cooling water for condenser and the absorber.

Paper by Pongsid Srikhirin, et. al. [22] gives the review of various absorption refrigeration technologies available in world. It mentions total 12 different vapor absorption cycle arrangements and gives the description of their working. It also gives valuable references that can be used for the further detailed study. Now these 12 types of arrangements are as follows. (1) Single effect absorption system. (2) Multi-effect absorption refrigeration cycle. (3) Absorption refrigeration cycle with GAX (GAX stands for "Generator/Absorber Heat Exchanger"). (4) Absorption refrigeration cycle with an absorber-heat-recovery. (5) Half-effect absorption refrigeration cycle (6) Combined vapor absorption-compression cycle (7) Sorption-desorption cycle (8) Dual-cycle absorption refrigeration (9) Combined ejector-absorption refrigeration cycle (10) Osmotic-membrane absorption cycle. (11) Self-circulation absorption system using LiBr/water (12) Diffusion Absorption Refrigeration system (DAR).

Much detailed literature review has been done and list of references and the literature is so long that it would be impossible to mention all of them in detail in this thesis. Best effort has been made to cover most important and topic related literature in this thesis.

2.1 Motivation

VCR technology uses electricity and costly refrigerants. Also that these refrigerants causing problems to environmental integrity. These drawbacks can be overcome by VAR technology. Advantages of vapor absorption refrigeration system mentioned in introduction section and it is highly motivating to design a absorption refrigeration system that can use solar energy and work for 24 hours. VAR technology can use low grade energy such as solar energy directly to convert heat in to cooling effect this feature makes it independent of electricity grid. It can work in remote ruler areas where electricity is not easily available, it can serve the needs like storage for dairy products and fresh vegetables or the store for medical drugs that are needed to be kept in cool temperatures. Refrigeration and air conditioning applications consumes almost 15 to 25 % of total electricity production. If the liability of cooling is lifted from the electricity then that electricity can be used in more useful purposes.

2.2 Problem Description

The aim of this project work is to design and build solar based vapor absorption refrigeration system which can work 24 hours by using solar energy. LPG fuel based prototype to be designed and built to check technical feasibility and then it is to be develope further to enable it use solar energy.

2.3 Objectives of Project

Objective above can be subdivided in following activities.

(1) Detailed literature review.

(2) Thermal and Physical Design calculations for system.

(3) Preparation of 3D Models and detailed drawings on CAD software.

(4) Preparing of bill of material and Market survey for cost data.

(5) Costing and selection of certain model out of feasible models.

(6) Procurement and purchase of items of bill of materials.

(7) Construction and assembly of system.

(8) Experimentation and analysis of result.

Chapter 3

Thermal Design.

Three design of vapour absoption refrigeration system is presented here. Three systems are classified based in heat supply technology used. In system one solar enery is used for suppling heat wherein system two solar and fuel heat is used and system three only fuel is used for heating.

3.1 Design Methodology for Solar Operated VAR System (Model 1)

This section is about design of the Solar Based Vapor Absorption Refrigeration System. While reviewing this design calculations, for subscripts used in nomenclature, the schematic diagram no.1 is to be kept in view for quick reference which is attached in Appendix A.

3.1.1 Thermodynamic Modeling and Heat Balance.

3.1.1.1 Evaporator Heat and Mass Balance

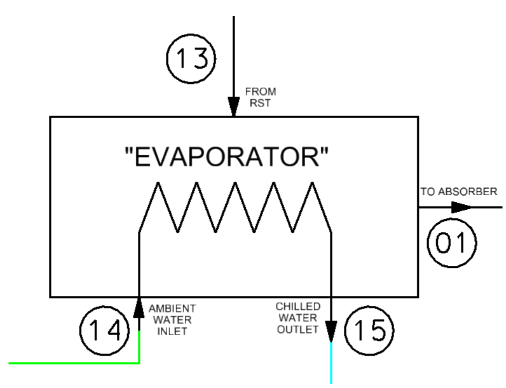

Figure 3.1: Evaporator Schematic Diagram

This is a Schematic diagram of evaporator. Such diagrams have been provided for every component in this section to be able to co relate nomenclature. As shown point 13 brings the liquid refrigerant from the refrigerant storage tank via expansion valve. This is a flooded type of evaporator so mass flow of 13 will be only as much to maintain the level of refrigerant in the evaporator. point 1 takes the saturated refrigerant vapor from evaporator to the absorber. The coil shown here is actually a copper pipe in which water at ambient temperature flows, which cools down to give us chilled water. This chilled water then further can be used to cool the air in various equipments like air washer or air conditioners.

The properties of water used as refrigerant given bellow in the table.

Table 3.1: Properties of Refrigerant in Evaporator.

Property	Nomenclature	Value	Unit	Reference
Desired Cooling Load	Q_E	1	TR	Input
Evaporator Temperature	$T_{13,1}$	5	^oC	Input
Evaporator Pressure	P_E	0.00872575	bar	[1]
Inlet Enthalpy of Refrigerant	H_1	2510.06	kJ/kg	[1]
Outlet Enthalpy of Refrigerant	H_{13}	21.02	kJ/kg	[1]
Le tent Heat of Evaporation	H_{fg}	2489.04	kJ/kg	[1]

Table 3.2: Properties of Water at Ambient Temperature.

Property	Nomenclature	Value	Unit	Reference
Water Ambient Temperature	T_{14}	35	^oC	Input
Chilled Water Outlet Temperature	T_{15}	10	^oC	Input
Enthalpy of water at 35 ^oC	H_{14}	146.634	kJ/kg	[1]
Enthalpy of water at 10 ^oC	H_{15}	42.0213	kJ/kg	[1]

Mass Flow Rate of Refrigerant

$$M_{13,1} = \frac{Q_E}{H_{fg}} = \frac{3.5}{2489.04} = 0.00140616\frac{kg}{s} = 5.062\frac{kg}{hr} \tag{3.1}$$

Mass Flow Rate of Hot Water

$$M_{14,15} = \frac{Q_E}{(H_{14} - H_{15})} = \frac{3.5}{(146.634 - 42.0213)} = 0.03345\frac{kg}{s} \tag{3.2}$$

Heat Balance of Evaporator

$$M_1H_1 = M_{13}H_{13} + Heat\ Absorbed \tag{3.3}$$

$$Heat\ Absorbed = M_{1,13}(H_1 - H_{13}) = (0.00140616)(2510.06 - 21.02) = 3.5kJ/s \tag{3.4}$$

3.1.1.2 Absorber And Generator Heat and Mass Balance

Now, Since we are designing a system which can work for 24 hours, we need to make sure that we have sufficient amount of Refrigerant, Strong solution and Weak Solution to use in hours of the night in which solar energy is not Available. Here we need to make assumption that solar Energy is available for 8 Hours per day. Say Morning 9am to 5pm. And for rest of the 16 hours solar energy is not available. Our requirement is to make generator and condenser produce more amount of refrigerant, then required by the evaporator, and store it in the Refrigerant Storage Tank (RST). So, mass flow rate of refrigerant in generator and condenser shall be more than evaporator and absorber, which has been calculated as bellow.

Mass Flow rate of refrigerant in Evaporator is Give by eq. 3.1. So, mass of refrigerant required per Day will be...

$$M_{Ref/day} = M_{13,1}(3600)(24) = 121.49kg \tag{3.5}$$

Above is the mass of refrigerant Generator and the condenser have to handle during 8 hours of solar availability. so, the mass flow rate of refrigerant vapor in generator and condenser will be.

$$M_{10,11} = \frac{121.49}{(8)(3600)} = 0.00421840\frac{Kg}{s} = 15.18\frac{kg}{hr} \tag{3.6}$$

Figure 3.2 is the Equilibrium diagram of the $LiBr.H_2O$. This Diagram gives us Concentration level of the solution in absorber and generator at Various temperature and pressures.

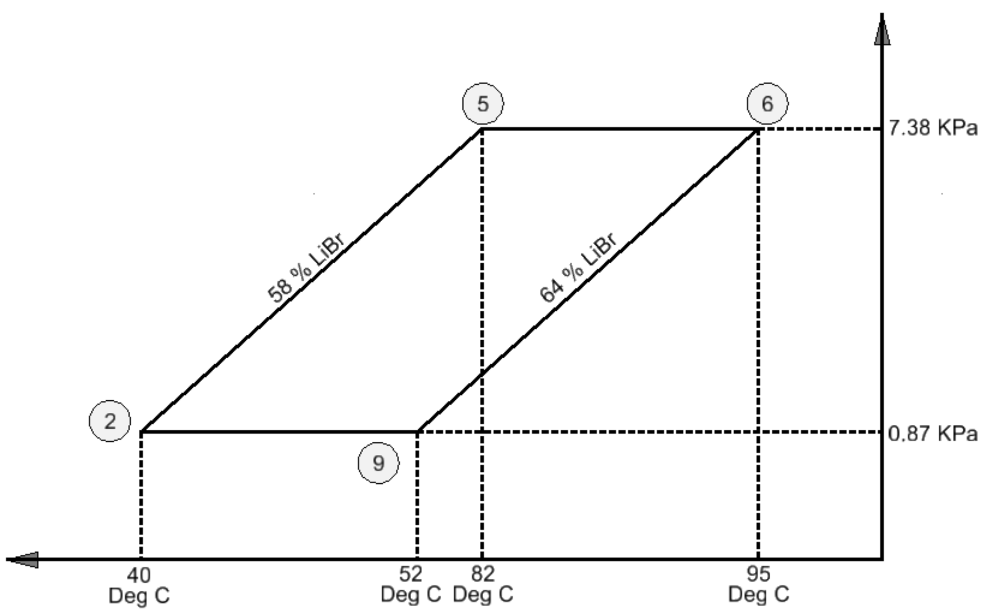

Figure 3.2: $LiBr.H_2O$ Equilibrium Diagram.

Here point 2 is exit of Absorber and point 6 is the exit of Generator. Point 5 may occur some where in heat exchanger (if used) or in generator it self. Similarly point 9 may occur in exchanger or in absorber it self depending up on which fluid has more heat capacity.

Absorber Temperature is to be maintained at 40^oC and Generator temperature is to be maintained at 95^oC. The generator temperature can easily be achieved if we are using Fuel such as Natural Gas or hydrogen as a heat source. where as solar heat collectors are comparatively expensive option.

Table 3.3: $LiBr.H_2O$ Solution Property Table for Generator and Absorber.

Property	Nomenclature	Value	Unit	Reference
Absorber Temperature	T_{Abs}	40	^oC	Input
Generator Temperature	T_{Gen}	95	^oC	Input
Temperature of Solution at the Exit of Absorber	T_2	40	^oC	Input
Pressure of Absorber	P_A	0.00872575	bar	[1]
Concentration of solution in Absorber.	S_{Abs}	58	%	[2]
Enthalpy of Solution at exit of absorber	H_2	110	kJ/Kg	[2]
Temperature of solution at the exit of Generator	$T_{6,7}$	95	^oC	Input
Pressure of generator	P_{Gen}	0.0738494	bar	[1]
Concentration of solution in Generator	S_{Gen}	64	%	[2]
Enthalpy of Solution at exit of Generator	$H_{6,7}$	230	kJ/Kg	[2]
Temperature at Point 5	T_5	82	^oC	[2]
Enthalpy at Point 5	H_5	180	kJ/Kg	[2]
Temperature At Point 9	T_9	52	^oC	[2]
Enthalpy at Point 9	H_9	150	kJ/Kg	[2]
Enthalpy of Vapor Leaving Generator	H_{10}	2667.61	kJ/Kg	[1]

Mass flow rate of the solution between absorber and generator is calculated for per kg of refrigerant vapor is being handled by the solution. concentration of refrigerant in the solution that is going from absorber to generator will be 42% and same way refrigerant content in the solution coming from the generator will be 36%.

So, mass flow rate of solution from absorber to generator is given by...

$$f = \frac{1-(0.36)}{(0.42)-(0.36)} = 10.66\frac{kg}{kg\,of\,Vapor} \tag{3.7}$$

$$M_2 = (M_{13,1})(f) = (0.00140616)(10.66) = 0.0149896\frac{kg}{s} = 53.96\frac{kg}{hr} \tag{3.8}$$

$M_2$is flow rate from the Absorber to Weak solution Storage Tank (WST). From WST to generator mass flow will be...

$$M_{3,4,5} = (M_{10,11})(f) = (0.00421840)(10.66) = 0.0449681\frac{kg}{s} = 161.88\frac{kg}{hr} \tag{3.9}$$

Similarly mass flow rate from generator to Strong Solution Storage Tank (SST) is given by...

$$f - 1 = 10.66 - 1 = 9.66 \frac{kg}{kg\, of\, Vapor} \tag{3.10}$$

$$M_6 = (M_{10,11})(f - 1) = (0.00421840)(9.66) = 0.040749 \frac{kg}{s} = 146.69 \frac{kg}{hr} \tag{3.11}$$

$M_6$is Flow rate from generator to Strong solution storage tank (SST). From SST to Absorber mass flow will be...

$$M_{7,8,9} = (M_{13,1})(f - 1) = (0.00140616)(9.66) = 0.013583 \frac{kg}{s} = 48.90 \frac{kg}{hr} \tag{3.12}$$

This way during the day time the weak solution tank will gradually become empty and strong solution tank will be filled up. At the beginning of the night we will have full strong solution storage tank, full refrigerant storage tank, and empty weak solution tank. During the night hours Evaporator will keep working as it was working in day time. and absorber will consume the strong solution from SST and Refrigerant from RST to produce weak solution, which will be filled up in the WST. The Pump, the Generator, the condenser will not work during the night.

The strong solution storage tank shall be filled up with the rate of $M_6 - M_{7,8,9} = 146.69 - 48.90 = 97.79 kg/hr$. and generator works for 8 hours so total storage capacity of Strong Solution Storage Tank shall be $(97.79)(8) = 782.32 kg$. Say 790Kg.

Similarly the weak solution storage tank shall be emptied with the rate of $M_{3,4,5} - M_2 = 161.88 - 53.96 = 107.92 kg/hr$. and for 8 hours of weak solution supply will require storage capacity of $(107.92)(8) = 863.36 kg$. Say 870kg.

Now, Heat balance of Absorber

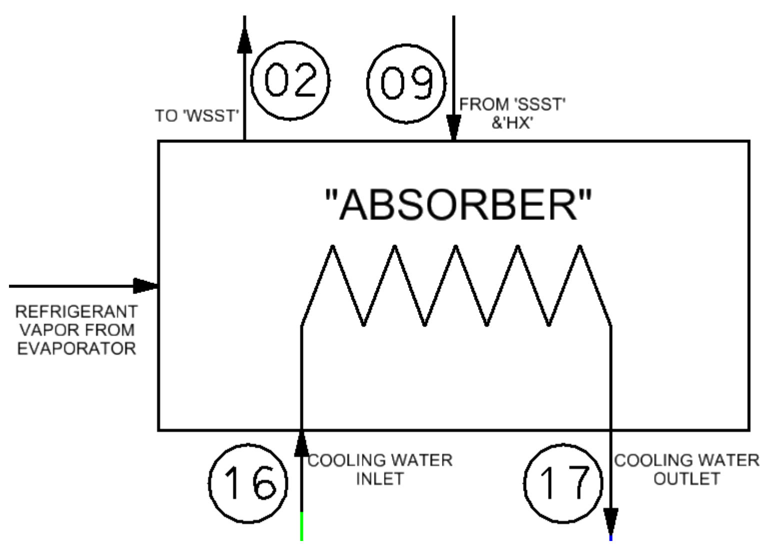

Figure 3.3: Absorber Schematic Diagram

$$M_1H_1 + M_9H_9 + Heat\,of\,Reaction = M_2H_2 + Heat\,Rejection \tag{3.13}$$

Heat of Reaction is an amount of heat that is generated during the chemical bonding of molecules of LiBr and H_2O. The Details of which can be found in the Perry's chemical hand book [3]. In there Heat of reaction is give as $5.30\,kCal/mol$ which is equivalent to $22.19\,kJ/mol$. The Atomic Weight of $LiBr.H_2O$ is $104.83gm/mol$ which is equivalent to $0.10483kg/mol$. So, the heat of reaction will be...

$$\frac{22.19}{0.10483} = 211.67\frac{kJ}{kg}$$

Now, heat is generated in absorber because of this reaction is...

$$Heat\,of\,Reaction = (M_1)(211.67) = (0.00140616)(211.67) = 0.2976\,kW \tag{3.14}$$

So, heat to be rejected from the absorber is...

$$Heat\,Rejection = M_1H_1 + M_9H_9 + Heat\,of\,Reaction - M_2H_2$$

$$Heat\,Rejection = (0.00140616)(2510.06)+(0.013583)(150)+0.2976-(0.0149896)(110) = 4.21\,kW \tag{3.15}$$

Now, Heat Balance of Generator

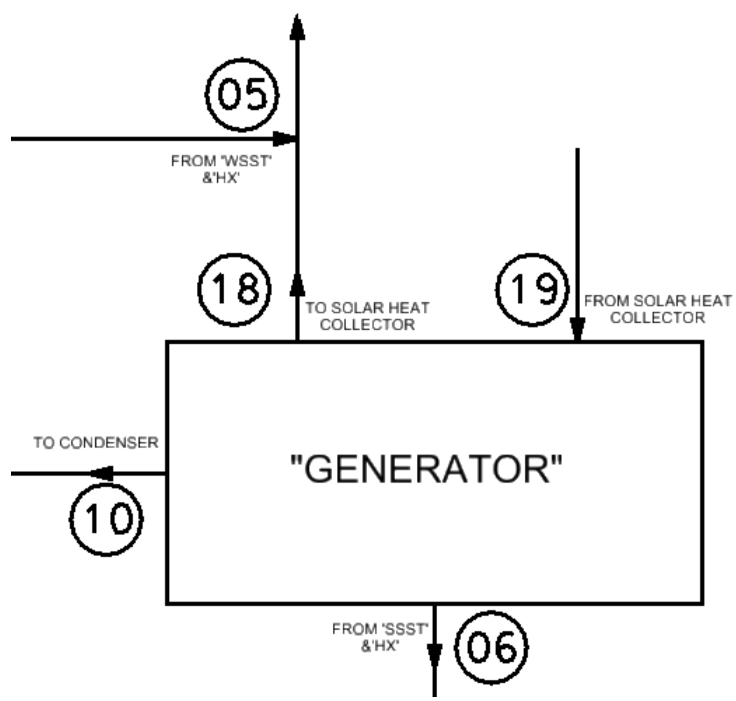

Figure 3.4: Generator Schematic Diagram.

$$M_5 H_5 + Heat\,to\,be\,Supplied = M_{10} H_{10} + M_6 H_6 \tag{3.16}$$

$$Heat\,to\,be\,Supplied = M_{10} H_{10} + M_6 H_6 - M_5 H_5$$

$$Heat\,to\,be\,Supplied = (0.00421840)(2667.61) + (0.040749)(230) - (0.0449681)(180) = 12.53\,kW \tag{3.17}$$

Additional 3.14 kW of heat to be added if Heat Exchanger is not used. If used then 2.47 kW of additional heat to be added. So, the maximum possible heat requirement of generator is 12.53 + 3.14 = 15.67 kW. To see the further details of this calculations please refer 3.1.4.

3.1.1.3 Condenser Heat and Mass Balance

The heated refrigerant vapor at 95 Deg C will enter the condenser, which is being cooled by water at 35 Deg C. Here it is perceived that heat rejection will be of two type. Latent heat rejection and sensible heat rejection. 95 deg C saturated vapor will condense and form 95 deg C hot fluid which is latent heat rejection, and 95 deg C hot fluid will further be cooled down to 40 Deg C which is sensible heat rejection. Some part of this heat rejection can take place in Refrigerant Storage Tank (RST) if it is made of steel or some conductive material instead of plastic or PVC. Same way it is recommended to make Absorber vessel out of metal than the PVC so that heat rejection could be an easier task.

Table 3.4: Condensate property table for condenser

Property	Nomenclature	Value	Unit	Reference
Temperature of Vapor entering condenser	T_{10}	95	^oC	[2]
Enthalpy of vapor entering condenser	H_{10}	2667.61	kJ/kg	[1]
Enthalpy of Saturated condensate	H_{con}	398.089	kJ/kg	[1]
Final condensate Temperature	T_{11}	40	^oC	Input
Final Enthalpy of condensate	H_{11}	167.533	kJ/kg	[1]

Now, Heat balance of condenser is...

$$M_{10} H_{10} = M_{11} H_{11} + Heat\,Rejection \tag{3.18}$$

Here, $M_{10}$and $M_{11}$is same, the heat rejection will be...

$$Heat\,Rejection = M_{10,11}(H_{10} - H_{11}) = (0.00421840)(2667.61 - 167.533) = 10.54\,kW \tag{3.19}$$

$$Ltent\,Heat\,Rejection = M_{10,11}(H_{10} - H_{con}) = (0.00421840)(2667.61 - 398.089) = 9.57\,kW \tag{3.20}$$

$$Sensible\,Heat\,Rejection = M_{10,11}(H_{con} - H_{11}) = (0.00421840)(398.089 - 167.533) = 0.97\,kW \tag{3.21}$$

3.1.1.4 Pump Work Calculations

Evaporator and Absorber Pressure is governed by Evaporator temperature which is in our case 5^oC. And Generator and Condenser pressure is governed by the final temperature of the condensate which is in our case 40^oC. So, Generator and condenser will be at high pressure side and the evaporator and absorber will be at low pressure side.

Generator pressure = $0.0738494\,bar$.

Absorber pressure = $0.00872575\,bar$.

Pressure difference = $0.06512365\,bar = 6512.365\,N/m^2$

$$Pressure\,Head = H_{head} = \frac{\triangle P}{\rho g} = \frac{6512.365}{(1670)(9.81)} = 0.3975m \tag{3.22}$$

Pumping Power required will be...

$$Pumping\,Power = Pow = \frac{(M_{3,4,5})(g)(H_{head})}{1000} = \frac{(0.0449681)(9.81)(0.3975)}{1000} = 1.7535\,X\,10^{-4}kW \tag{3.23}$$

Thus, pumping power required is very less. It may even be neglected for further calculations.

3.1.1.5 Theoretical COP of the System

The co-efficeint of performance of the system is a data that can give us the idea of how the system will be beneficial to us. It is a ratio of the cooling effect produced per unit of heat energy input. In our system cooling effect produced = 3.5 kW. and summation of energy input in Generator and pump work is = 15.67 kW. So, the COP comes 3.5/15.67 = 0.2233.

But in this system cooling effect is produced for 24 hours and heat supply of 15.67 kW is to be given for only 8 hours of the day. So, in whole day total heat absorbed by the evaporator is (3.5)(3600)(24)= 302.4 MJ. where as total Solar energy used is (15.67)(3600)(8) =451.296 MJ. Now the ratio of this two energy comes...

$$COP = \frac{Cooling\,effect\,produced\,in\,whole\,day}{Solar\,energy\,used\,in\,whole\,day} = \frac{(3.5)(3600)(24)}{(15.67)(3600)(8)} = 0.67 \tag{3.24}$$

3.1.1.6 Copper Pipe Data

Here the dimensions of the copper pipe is taken from the Standard pipe size table. It may change slightly from one manufacture to other. It would not affect the calculation much.

Table 3.5: Copper pipe data for calculation.

Property	Nomenclature	Value	Unit	Reference
Material of pipe		copper (C10200)		[4]
Tensile Strength	σ_t	30000	psi	[4]
Yield Strength	σ_y	9000	psi	[4]
Thermal conductivity of pipe	k_{pipe}	401	W/mK	
Outer diameter of the pipe	d_o	15	mm	
Inner diameter of the pipe	d_i	12.5	mm	

In our system the the difference between evaporator pressure and the atmospheric pressure is $1.013 - 0.00872575 = 1.00427bar$, which is as good as atmospheric pressure. Any selected pipe's wall thickness is sufficient to hold the atmospheric pressure because the thickness comes after calculations is very very low for 1.013 bar pressure. Calculations for which is given in 4.4.1.

3.1.2 Evaporator Design

Evaporator is a part of the system where refrigerant evaporates (boils) at low temperature because of the very low pressure is maintained in the vessel. In this system evaporater is considered as flooded type evaporator where the hot fluid at ambient pressure and temperature flows inside the copper pipe and surrounding this copper pipe is a pool of refrigerant which is at very low pressure. because of this low pressure the fluid in evaporator begins to boil or evaporate when it comes with the contact of copper tube at ambient temperature. It takes heat of evaporation from this copper tube and cause its temperature to fall. This copper tube in turn cools the fluid flowing within it and so is cooling effect is produced.

3.1.2.1 Thermophysical Properties of Refrigerant and Water in Evaporator.

As mentioned earlier that the evaporator works at 5 Deg C temperature and the refrigerant is being water its properties as follows

Table 3.6: Thermophysical properties of refrigerant in evaporator.

Property	Nomenclature	Value	Unit	Reference
Evaporator inlet temperature (sat. liquid)	T_{13}	5	^oC	Input
Evaporator outlet temperature (sat. vapor)	T_1	5	^oC	Input
Density of liquid	$\rho_{l,13}$	999.917	kg/m^3	[1]
Dynamic viscosity of liquid	$\mu_{l,13}$	0.00151829	Pa.s	[1]
Specific Heat of Liquid	$Cp_{l,13}$	4.2054	kJ/Kg	[1]
Thermal conductivity of liquid	$k_{l,13}$	0.570	W/mK	[1]
Density of vapor	$\rho_{v,1}$	0.0068022	kg/m^3	[1]
Dynamic viscosity of vapor	$\mu_{v,1}$	0.00000933	Pa.s	[1]
Thermal conductivity of vapor	$k_{v,1}$	0.01733	W/mK	[1]
Surface Tension	σ	0.0749417	N/m	[1]
Thermal Diffusivity	α_l	$1.3555\,X\,10^{-7}$	m^2/s	Calculated
Atomic weight of water	M_{AW}	18	gm/mol	

Hot water at Ambient temperature is at 35 Deg C and its properties are as follows.

Table 3.7: Thermophysical properties of water inside tube.

Property	Nomenclature	Value	Unit	Reference
Inlet temperature of hot water	T_{14}	35	^oC	input
Inlet Enthalpy of hot water	H_{14}	146.634	kJ/kg	[1]
Outlet temperature of hot water	T_{15}	10	^oC	input
Outlet Enthalpy of hot water	H_{15}	42.02	kJ/kg	[1]
The bulk mean temperature	T_b	22.5	^oC	
Density	$\rho_{14,15}$	997.614	kg/m^3	[1]
Dynamic viscosity	$\mu_{14,15}$	0.000943215	Pa.s	[1]
Specific Heat	$Cp_{14,15}$	4.1828	kJ/kg	[1]
Thermal conductivity	$k_{14,15}$	0.602	W/mK	[1]

3.1.2.2 Evaporator Inside Pipe Calculations

Heat transfer, inlet and outlet temperatures and Specific heat of hot water is known. So, the mass flow rate of hot water in evaporator is required is.

$$M_{14,15} = \frac{Q_E}{(H_{14} - H_{15})} = \frac{3.5}{(146.634 - 42.02)} = 0.033456\frac{kg}{s} = 120.44\frac{kg}{hr} \tag{3.25}$$

Cross sectional flow area of pipe is $A_{c/s} = (\pi/4)(di^2) = 0.00012271\,m^2$

Velocity of the fluid in side pipe will be

$V = M_{14,15}/[(\rho_{14,15})(A_{c/s})] = 0.033456/[(997.614)(0.00012271)] = 0.27327\,m/s.$

Inside pipe Renold number

$$Re_{14,15} = \frac{(\rho_{14,15})(V)(di)}{(\mu_{14,15})} = \frac{(997.614)(0.27327)(0.0125)}{(0.000943215)} = 3612.88 \tag{3.26}$$

Inside pipe Prandlt Number

$$Pr_{14,15} = \frac{(\mu_{14,15})(Cp_{14,15})}{(k_{14,15})} = \frac{(0.000943215)(4.1828)(1000)}{(0.602)} = 6.55 \tag{3.27}$$

The Renold number is Greater than 2300. so, to find Nusselt number (Nu) Gnielinski equation as given in ASHRAE Fundamentals [6] is used as follows.

$$Nu_{14,15} = \frac{(f/2)(Re - 1000)(Pr)}{1 + 12.7(f/2)^{0.5}(Pr^{2/3} - 1)} \tag{3.28}$$

Where,

$$f = (1.58\ lnRe - 3.28)^{-2} \tag{3.29}$$

$$f = [\{(1.58)(ln(3612.88))\} - 3.28]^{-2} = 0.01070$$

$$\frac{hi.di}{k_{14,15}} = Nu_{14,15} = \frac{(0.01070/2)(3612.88 - 1000)(6.55)}{1 + 12.7(0.01070/2)^{0.5}(6.55^{2/3} - 1)} = 27.55 \tag{3.30}$$

convective heat transfer co-efficient inside pipe is

$$hi_{14,15} = \frac{Nu_{14,15}.k_{14,15}}{di} = \frac{(27.55)(0.602)}{0.0125} = 1326.80\ \frac{W}{m^2K} \tag{3.31}$$

3.1.2.3 Evaporator Out Side Pipe Calculation

Out side of copper pipe is refrigerant pool. To find the convective heat transfer co-efficient outside pipe three type of cases have been considered. (1)Natural convection or Convective boiling (2) Nucleate Boiling. (3) Film Boiling.

(A) Natural convection or Convective Boiling

For convective boiling Nusselt Number is a function of Grashoff Number and Prandlt Number.

Grashoff Number,

$$Gr = \frac{L_e^3.g.\beta.\triangle T}{\nu^2} = \frac{(0.015^3)(9.81)(1/286.75)(17.5)}{(0.00151829/999.917)^2} = 876386.84 \tag{3.32}$$

Where, $L_e$is effective length. which in case of horizontal cylinder is the diameter of the cylinder. β is reciprocal of average of both bulk mean temperatures. $\triangle T$ is temperature difference between both bulk mean temperature. and ν is kinematic viscosity.

Prandlt Number,

$$Pr = \frac{\mu_{l,13} Cp_{l,13}}{k_{l,13}} = \frac{(0.00151829)(4.2054)(1000)}{(0.570)} = 11.20 \tag{3.33}$$

To calculate Nusselt Number from this two data chapter of heat transfer in ASHRAE fundamentals [6] gives equation for horizontal cylinders as follows.

$$Nu = \left[0.6 + \frac{0.387(Gr.Pr)^{1/6}}{[1+(0.559/Pr)^{9/16}]^{8/27}}\right]^2 \tag{3.34}$$

$$Nu = \left[0.6 + \frac{(0.387)[(876386.84)(11.20)]^{1/6}}{[1+(0.559/11.20)^{9/16}]^{8/27}}\right]^2 = 35.81 \tag{3.35}$$

Convective Heat transfer co-efficient is...

$$ho_1 = \frac{Nu.k_{l,13}}{do} = \frac{(35.81)(0.570)}{(0.015)} = 1360.70 \, \frac{W}{m^2 K} \tag{3.36}$$

(B) Nucleate Boiling

In ASHRAE Fundamentals chapter 5 of two phase flow there are total three equations for the nucleate boiling are mentioned. These equations are namely Stephan and Abdelsalam equation, Cooper equation, and Gorenflo equation [7]. Now in these equations the heat flux (Q/A) is very important parameter. But here we can not decide the heat flux since we do not have the heat transfer area. So, in following calculations value of heat flux (Q/A) is assumed and taken as $6461.14 W/m^2$. In this assumption the value of A is calculated based on ho_1.

(i) Stephen and Abdelsalam Equation.

$$\frac{ho.Dd}{k_{l,13}} = 0.0546 \left[\left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{q}{A}.\frac{Dd}{K_{l,13} T_{sat}}\right)\right]^{0.67} \left(\frac{h_{fg}.Dd^2}{\alpha_l^2}\right)^{0.248} \left(\frac{\rho_l - \rho_v}{\rho_l}\right)^{-4.33} \tag{3.37}$$

Where,

$\alpha_l = Thermal\,Diffusivity$ and $\sigma = Surface\,Tension$

$$Dd = (0.0208)(\theta) \left[\frac{\sigma}{g(\rho_l - \rho_v)}\right]^{0.5} \tag{3.38}$$

here in eq. 39 $\theta$is always taken as 35^o, which in radians comes to be 0.6108 rad.

$$Dd = (0.0208)(0.6108)\left[\frac{(0.0749417)}{(9.81)(999.917 - 0.0068022)}\right]^{0.5} = 3.5116\,X\,10^{-5} \tag{3.39}$$

$$\frac{ho.Dd}{k_{l,13}} = 0.0546\left[\left(\frac{0.0068022}{999.917}\right)^{0.5}\left(6461.14.\frac{(3.5115\,X\,10^{-5})}{(0.570)(278)}\right)\right]^{0.67}\left(\frac{(2489.04\,X\,10^{3})(3.5116\,X\,10^{-5})^{2}}{(1.355\,X\,10^{-7})^{2}}\right)^{0.248}(1)^{-4.33}$$

$$\frac{ho.Dd}{k_{l,13}} = 7.654518\,X\,10^{-3} \tag{3.40}$$

Convective heat transfer co-efficient,

$$ho2 = \frac{(7.654518\,X\,10^{-3})(0.570)}{(3.5116\,X\,10^{-5})} = 124.24\frac{W}{m^2K} \tag{3.41}$$

(ii) Cooper Co-relation

Cooper in 1984 [7] gave the correlation as follows...

$$h = 55\left(Pr^{0.12-0.4343.ln(Rp)}\right)(-0.4343.ln(Pr))^{-0.55}M^{-0.5}\left(\frac{q}{A}\right)^{0.67} \tag{3.42}$$

Here Pr does not mean prandlt number but it is a ratio of saturation pressure and the critical pressure. and Rp is a surface roughness which is to be taken as 1 if it is unknown.

$$Pr = \frac{P_{sat}}{P_{crit}} = \frac{0.008725}{220.6} = 3.95512\,X\,10^{-5} \tag{3.43}$$

Convective heat transfer co-efficent is...

$$h = 55\left((3.95512\,X\,10^{-5})^{0.12}\right)\left(-0.4343.ln\left(3.95512\,X\,10^{-5}\right)\right)^{-0.55}18^{-0.5}6461.14^{0.67} \tag{3.44}$$

$$h = 607.066\frac{W}{m^2K} \tag{3.45}$$

It is also said in reference [7] , to multiply 'h' with 1.7 in case of copper surfaces.

$$ho3 = 1032.012\frac{W}{m^2K} \tag{3.46}$$

(iii) Gorenflo Equation

Gorenflo in 1993 gave following equation.

$$h = h_o.F_{PF}\left(\frac{q/A}{(q/A)_o}\right)^{nf}\left(\frac{R_p}{R_{po}}\right)^{0.133} \tag{3.47}$$

Where, $F_{PF} = 1.2(Pr)^{0.27} + 2.5(Pr) + \frac{Pr}{1-Pr}$ and $nf = 0.9 - 0.3(Pr)^{0.15}$. Here, same as before Pr is a ratio of saturation pressure and the critical pressure and not the prandlt number. According to reference [7] value of $(q/A)_o$is to be taken as 20000 and $R_{po}$is 0.4. So, this gives us value of $F_{PF} = 0.07774$ and $nf = 0.8344$. The value of $h_o$in eq. 3.47 is taken from the reference [8].

$$ho4 = (5600)(0.07774)\left(\frac{6461.14}{20000}\right)^{0.8344}\left(\frac{1}{0.4}\right)^{0.133} = 191.55\frac{W}{m^2K} \tag{3.48}$$

So, all above three equations are used for the nucleate boiling phenomena. The more detailed description of all equations can be found in reference [8].

(C) Film Boiling.

Like above two cases ASHRAE Fundamentals, "Two Phase Flow" [7] Chapter gives the formula for convective heat transfer co-efficient for case of film boiling from the horizontal cylinder surface. Which is as follows...

$$h = 0.62\left[\frac{k_v^3.\rho_v.H_{fg}.g.(\rho_l - \rho_v)}{\mu_v.(T_s - T_{sat}).do}\right]^{0.25} \tag{3.49}$$

Heat transfer co-efficient...

$$ho = 0.62\left[\frac{(0.01377)^3(0.0068022)(2489.04)(1000)(9.81)(999.917 - 0.0068022)}{(0.00000933)(22.5 - 5)(0.015)}\right]^{0.25}$$

$$ho5 = 71.51\frac{W}{m^2K} \tag{3.50}$$

3.1.2.4 Evaporator Overall Heat Transfer Co-efficient.

From all above calculation we have got following heat transfer co-efficient for evaporator.

Table 3.8: Heat transfer co-efficents for evaporator design.

	Co-efficient	Value	Unit	Reference
Inside pipe	hi	1326.80	W/m^2K	eq.3.31
Outside pipe				
Convective Boiling	$ho1$	1360.70	W/m^2K	eq.3.36
Nucleate Boiling	$ho2$	124.24	W/m^2K	eq.3.41
	$ho3$	1032.012	W/m^2K	eq.3.46
	$ho4$	191.55	W/m^2K	eq.3.48
Film Boiling	$ho5$	71.51	W/m^2K	eq.3.50

Which value of 'ho' is to be taken in the calculation of overall heat transfer co-efficient depends on the fact that which kind of boiling will take place in the evaporator. For that we need to have the boiling curve of water at evaporator pressure which is in our case 0.00872575 bar. All the water boiling curves that were available, were made for atmospheric pressure. And according to them at 30^oC temperature difference heat flux is maximum. Now, logic says as the pressure decreases evaporation (boiling) takes place at lower and lower temperatures. So, to make the water boil at 5^oC and 0.00872575 bar, temperature difference needed should be even less than 30^oC. In Evaporator we are having maximum temperature difference of 30^oC at the binging of the pipe where hot water at 35 deg C enters the copper pipe and out side that 5 Deg C water is available. Here at this point there should be a phenomena of Film Boiling. And we have minimum temperature difference of 5 ^oC at the end of the tube where chilled water at 10^oC comes out and out side that 5^oC water is available. So, here most probably we will be having convective boiling. So, we can surely say that at various points of pipe the boiling pattern will be different. Thus, for calculation I have taken an average value of all the five 'ho' values in table 3.8 for the calculation of overall heat transfer, which is $556.00\,W/m^2K$.

Over all Heat Transfer Co-efficient...

$$U_{eva} = \left[\frac{1}{hi} + \frac{1}{ho} + \frac{ln(do/di)}{2\pi kl}\right]^{-1} \tag{3.51}$$

$$U_{eva} = \left[\frac{1}{1326.80} + \frac{1}{556.00} + \frac{ln(15/12.5)}{2\pi(401)(1)}\right]^{-1} = 381.00\frac{W}{m^2K}$$

3.1.2.5 Evaporator Required Heat Transfer Area

To find heat transfer area we need Logarithmic Mean Temperature Difference (LMTD). which is found as $LMTD = (\triangle T_{inlet} - \triangle T_{outlet})/ln(\triangle T_{inlet}/\triangle T_{outlet}) = 13.95$ for our case.

Required heat transfer area...

$$A_{HT} = \frac{Q}{U_{eva}(LMTD)} = \frac{3.5(1000)}{381.00(13.95)} = 0.6585\,m^2 \tag{3.52}$$

Required pipe length...

$$L_{pipe} = \frac{A}{\pi.do} = \frac{(0.6585)}{\pi(0.015)} = 13.97\,m \tag{3.53}$$

3.1.3 Absorber Design

As it has been shown in heat and mass balance of absorber and generator, absorber has to reject 4.21kW of heat. In absorber heat comes mainly from two sources. One is from the strong solution that is coming from the generator and SSST and the other is the heat of reaction which is a result of exothermic chemical reaction between LiBr and water molecules. It is required to maintain the temperature of absorber around 40^oC. For which mass flow rate of cooling water can be increased or decreased. Here, outside of pipe normal convective heat transfer will take place.

3.1.3.1 Absorber Thermophysical Properties of Cooling Water and Solution.

Few thermophysical properties of $Libr.H_2O$ solution and water at ambient conditions is given in table 3.3 and table 3.7, some of which shall be repeated in the following property tables.

Here, we don't have either mass flow rate or exit temperature of the cooling water. So, we need to make an assumption about one of them. One of the known fact about the cooling water is its temperature does not rise much during the cooling process. Its mass flow rate is kept comparatively very high. Usually its temperature rises no more than two to three degrees. So here lets make an assumption that the temperature rise of cooling water is 2^oC. So, the outlet temperature of the cooling water is assumed to be 37^oC. Which would give us the bulk mean temperature of 36^oC on which all the thermo physical properties of cooling water shall be taken.

Table 3.9: Thermophysical properties of cooling water in Absorber.

Property	Nomenclature	Value	Unit	Reference
Inlet Temperature of CW	T_{16}	35	^oC	input
Inlet Enthalpy of CW	H_{16}	146.634	KJ/Kg	[1]
Outlet Temperature of CW	T_{17}	37	^oC	input
Outlet Enthalpy of CW	H_{17}	154.993	KJ/Kg	[1]
Bulk mean temperature of CW	$T_{bulk,16,17}$	36	^oC	
Density	$\rho_{16,17}$	993.643	Kg/m^3	[1]
Dynamic Viscosity	$\mu_{16,17}$	0.00068661	Pa.s	[1]
Specific Heat	$Cp_{16,17}$	4.1794	$KJ/Kg.K$	[1]
Thermal Conductivity	$k_{16,17}$	0.6247	W/mK	[1]

Hot solution from the SSST will come to absorber at 52^oC and we are to maintain 40^oC Temperature in absorber so the properties of solution is taken at 46^oC. Properties of $LiBr.H_2O$ Solution can be found in reference [9, 10].

Table 3.10: Thermophysical properties of $LiBr.H_2O$ solution in Absorber.

Property	Nomenclature	Value	Unit	Reference
Inlet Temperature of Solution	T_9	52	^oC	[2]
Outlet Temperature of Solution	T_2	40	^oC	[2]
Bulk Mean Temperature	$T_{bulk,9,2}$	46	^oC	
Consentration of solution in abs.	S_2	58	%	[2]
Density of Solution	ρ_2	1660	Kg/m^3	[9]
Dynamic Viscosity	μ_2	0.0043	Pa.s	[9]
Specific Heat of Solution	Cp_2	1.94	$KJ/Kg.K$	[9]
Thermal Conductivity	k_2	0.430	W/mK	[10]

3.1.3.2 Absorber Inside Pipe Calculations

Pipe selected for the absorber is same as the pipe selected for evaporator, with same dimensions and material. Details of which can be found in table 3.5.

First we need to determine mass flow rate of cooling water.

$$M_{16,17} = \frac{Heat\,Rejection}{(H_{17} - H_{16})} = \frac{(4.21)}{(154.993 - 146.634)} = 0.5036Kg/s \tag{3.54}$$

Velocity of fluid in pipe is...

$$V_{16,17} = \frac{M_{16,17}}{\rho_{16,17}.A_{c/s}} = \frac{(0.5036)}{(993.643)(0.0001227)} = 4.130\,m/s \tag{3.55}$$

Renold Number,

$$Re = \frac{\rho_{16,17}.V_{16,17}.di}{\mu_{16,17}} = \frac{(993.643)(4.130)(0.0125)}{(0.00068661)} = 72741.84 \tag{3.56}$$

Prandlt Number,

$$Pr = \frac{\mu_{16,17}Cp_{16,17}}{k_{16,17}} = \frac{(0.00070519)(4.1794)(1000)}{(0.6242)} = 4.71 \tag{3.57}$$

To find Nusselt Number we can use equation 3.28 and 3.29.

$$f = (1.58\,ln(72741.84) - 3.28)^{-2} = 4.8174\,X\,10^{-3} \tag{3.58}$$

$$Nu = \frac{[(4.8174\,X\,10^{-3})/2].(72741.84 - 1000).(4.721)}{1 + (12.7).[(4.8174\,X\,10^{-3})/2]^{0.5}(4.721^{2/3} - 1)} = 386.09 \tag{3.59}$$

Convective heat transfer co-efficent inside pipe

$$hi = \frac{(386.09)(0.6247)}{(0.0125)} = 19295.45 \frac{W}{m^2 K} \tag{3.60}$$

3.1.3.3 Absorber Outside Pipe Calculation

Out side of pipe is a pool of weak solution. Here natural convection is considered to be taking place.

Grashoff Number

$$Gr = \frac{L_e^3 . g . \beta . \triangle T}{\nu^2} = \frac{(0.015)^3 (9.81)(1/314)(10)}{(0.0043/1660)^2} = 157142.04 \tag{3.61}$$

Prandlt Number,

$$Pr = \frac{\mu . Cp}{k} = \frac{(0.0043)(1.95)(1000)}{(0.430)} = 19.4 \tag{3.62}$$

Rayleigh Number, $Ra = Gr.Pr = (157142.04)(19.4) = 3048555.57$.

Nusselt Number equation for horizontal cylinder,

$$Nu = \left[0.6 + \frac{(0.387)(Ra)^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right]^2 \tag{3.63}$$

$$Nu = \left[0.6 + \frac{(0.387)(3048555.57)^{1/6}}{[1 + (0.559/19.4)^{9/16}]^{8/27}} \right]^2 = 25.88 \tag{3.64}$$

Convective heat transfer co-efficient,

$$ho = \frac{Nu . k}{do} = \frac{(25.88)(0.430)}{(0.015)} = 741.89 \, \frac{W}{m^2 K} \tag{3.65}$$

3.1.3.4 Absorber Overall Heat Transfer Co-efficient

$$U_{abs.} = \left[\frac{1}{hi} + \frac{1}{ho} + \frac{ln(do/di)}{2 \pi k l} \right]^{-1} \tag{3.66}$$

$$U_{abs} = \left[\frac{1}{19295.45} + \frac{1}{741.89} + \frac{ln(15/12.5)}{2 \pi (401)(1)} \right]^{-1} = 679.30 \frac{W}{m^2 K} \tag{3.67}$$

3.1.3.5 Absorber Required Heat transfer Area

$$LMTD = \frac{\triangle T_{inlet} - \triangle T_{outlet}}{ln(\triangle T_{inlet}/\triangle T_{outlet})} = 9.10 \tag{3.68}$$

Heat Transfer Area,

$$A_{HT} = \frac{Heat\,Rejected}{U_{abs}(LMTD)} = \frac{(4.21)(1000)}{(679.30)(9.10)} = 0.6810\,m^2 \tag{3.69}$$

Requred pipe length,

$$L_{pipe} = \frac{A_{abs}}{\pi.do} = \frac{(0.6810)}{\pi(0.015)} = 14.45\,m \tag{3.70}$$

3.1.4 Heat Exchanger Design

Heat Exchanger is a component which transfers heat from the hot solution coming from SSST to the cold solution comming from WSST. This component is not always be necessary to keep in the system. Its presence in the system depends on how much it reduces the heat requirement of generator. And if it is not much then it is avoided. It can be decided whether to keep heat exchanger in system or not based on following calculations.

Heat available with hot solution comming from SSST,

$$Q_{hot,7} = M_{7,8,9}(H_7 - H_8) = (0.013583)(230 - 150) = 1.0866KW \tag{3.71}$$

Heat required by cold fluid going from WSST to generator,

$$Q_{cool,3} = M_{3,4,5}(H_5 - H_4) = (0.0449681)(180 - 110) = 3.1477KW \tag{3.72}$$

Now, if we choose to put a heat exchanger here, then, only some amount of heat we will be able to transfer from hot to cold solution. It is not possible to transfer all the 1.0866kW of heat to cold solution. Even if we put an exchanger which would give good heat transfer we have to give 3.1477-1.0866 = 2.0611kW of extra heat from solar heat collectors, and if we don't put heat exchanger we have to give all the 3.1477 kW extra from the solar heat collector. Puting an exchanger would perhaps save one solar collector but the exchanger it self would be more expensive then the collector. But for design the calculationis done which is as follows.

Assuming that all the 1.0886 kW heat is being transferred to cold solution then what will be the temperature (T_5) and enthalpy of cold solution coming out of heat exchanger.

$$T_5 = \frac{Q}{M_{3,4}Cp_2} + T_{4,2} = \frac{(1.0866)}{(0.0449681)(1.94)} + 40 = 52.45\,^oC \tag{3.73}$$

Enthalpy of 58% rich $LiBr.H_2O$ solution at 52.45^oC according to [2] is 125 kJ/kg. Now, according to our previous calculations of heat and mass balance of generator, the solution that reaches to generator should have temperature of 82^oC and enthalpy of 180 kJ/kg. So, we need to put additional $(0.0449681)(180 - 125) = 2.47KW$ of heat through solar heat collectors.

Pipe Selection for Heat Exchanger

This heat exchanger is assumed to be double pipe heat exchanger. Where inner pipe is made of copper and outer pipe could be of steel.

Inner diameter of inner pipe $di = 12.5mm$

Outer diameter of outer pipe $do = 15mm$

Inner diameter of outer pipe $Di = 28mm$

Outer diameter of outer pipe $Do = 33.40mm$

3.1.4.1 Heat Exchanger Thermophysical Properties for Hot and Cold solutions

The cold solution is entering the exchanger at 40^oC. and leaves at 52.45^oC. Similarly hot solution is entering at 95^oC and leaving at 52^oC. The properties are taken at their bulk mean temperatures which are 46^oC and 74^oC. The properties of cold fluid shall be same as mentioned in Table3.10. The properties of hot fluid is given in table 3.11.

Table 3.11: Thermophysical Properties of $LiBr.H_2O$ solution at 74^oC.

Property	Nomenclature	Value	Unit	Reference
Inlet Temperature of Solution	T_7	95	^oC	[2]
Outlet Temperature of Solution	T_8	52	^oC	[2]
Bulk Mean Temprature	$T_{bulk,7,8}$	74	^oC	
Concentration of solution in abs.	$S_{7,8}$	64	%	[2]
Density of Solution	$\rho_{7,8}$	1720	Kg/m^3	[9]
Dynamic Viscosity	$\mu_{7,8}$	0.0035	Pa.s	[9]
Specific Heat of Solution	$Cp_{7,8}$	1.82	$KJ/Kg.K$	[9]
Thermal Conductivity	$k_{7,8}$	0.460	W/mK	[10]

3.1.4.2 Heat Exchanger Inside Pipe Calculations

Cross sectional area of inner pipe is $A_{c/s} = (\pi/4)(0.0125^2) = 0.00012271\, m^2$

Velocity of cold fluid inside the pipe $V = M_{3,4,5}/\rho_2.A_{c/s} = (0.0449681)/(1660)(0.00012271) = 0.2207m/s$

Renold Number inside pipe,

$$Re = \frac{\rho_2.V.di}{\mu_2} = \frac{(1660)(0.2207)(0.0125)}{(0.0043)} = 1065.00 \quad (3.74)$$

Prandlt Number inside pipe,

$$Pr = \frac{\mu_2 Cp_2}{k_2} = \frac{(0.0043)(1.94)(1000)}{(0.430)} = 19.4 \quad (3.75)$$

Flow will be laminar, it would have been laminar even if we had used 1/8" pipe which is smallest availabel standard size.

According to reference [6] equation for Nusselt number for laminar flow is...

$$Nu = 3.66 + \frac{0.065.(D/L).Re.Pr}{1 + 0.04[(D/L).Re.Pr]^{2/3}} \quad (3.76)$$

here, D is diameter of the pipe and the length is taken as 1.

$$Nu = 3.66 + \frac{0.065(0.0125/1)(1065.00)(19.4)}{1 + 0.04[(0.0125/1)(1065.00)(19.4)]^{2/3}} = 9.45 \quad (3.77)$$

convective heat transfer co-efficient is.

$$hi = \frac{Nu.k_2}{di} = \frac{(9.45)(0.430)}{(0.0125)} = 325.08 \frac{W}{m^2K} \quad (3.78)$$

3.1.4.3 Heat Exchanger Out Side Pipe Calculation

Cross sectional area of Annulus $A_{c/s} = (\pi/4)(Di^2 - do^2) = (\pi/4)(0.028^2 - 0.015^2) = 5.59\,X\,10^{-4} m^2$

Hydraulic Diameter of Annulus $D_{hyd} = 4.A_{c/s}/Wetted\,Perimeter = 4(5.59\,X\,10^{-4})/[\pi(0.015 + 0.028)] = 0.01655\,m$

Velocity of hot fluid in Annulus $V = M_{7,8}/\rho_{7,8}.A_{c/s} = (0.013583)/(1720)(5.59\,X\,10^{-4} m^2) = 0.01412\,m/s$

Renold Number in Annulus,

$$Re = \frac{\rho_{7,8} V.D_{hyd}}{\mu_{7,8}} = \frac{(1720)(0.01412)(0.01655)}{(0.0035)} = 114.83 \quad (3.79)$$

Prandlt Number,

$$Pr = \frac{\mu_{7,8} Cp_{7,8}}{k_{7,8}} = \frac{(0.0035)(1.82)(1000)}{(0.460)} = 13.84 \quad (3.80)$$

To find Nusselt Number we will use equation 3.76.

$$Nu = 3.66 + \frac{0.065(0.01655/1)(114.83)(13.84)}{1 + 0.04[(0.01655/1)(114.83)(13.84)]^{2/3}} = 4.92 \tag{3.81}$$

Convective heat transfer co-efficient is,

$$ho = \frac{Nu.k_{7,8}}{D_{hyd}} = \frac{(4.92)(0.460)}{(0.01655)} = 136.74 \frac{W}{m^2K} \tag{3.82}$$

3.1.4.4 Heat Exchanger Overall Heat Transfer Co-efficient

$$U_{hx.} = \left[\frac{1}{hi} + \frac{1}{ho} + \frac{ln(do/di)}{2\pi kl}\right]^{-1} \tag{3.83}$$

$$U_{hx} = \left[\frac{1}{328.08} + \frac{1}{136.74} + \frac{ln(15/12.5)}{2\pi(401)(1)}\right]^{-1} = 95.84 \frac{W}{m^2K} \tag{3.84}$$

3.1.4.5 Heat Exchanger Heat Transfer Area Required

Logarithmic Mean Temperature Difference,

$$LMTD = \frac{\triangle T_{inlet} - \triangle T_{outlet}}{ln(\triangle T_{inlet}/\triangle T_{outlet})} = 24.28 \tag{3.85}$$

Heat Transfer Area,

$$A_{HT} = \frac{Q}{U_{hx}(LMTD)} = \frac{(1.0886)(1000)}{(95.84)(24.28)} = 0.4678\, m^2 \tag{3.86}$$

Length of pipe required,

$$L_{pipe} = \frac{A}{\pi.do} = \frac{(0.4678)}{\pi(0.015)} = 9.92m \tag{3.87}$$

3.1.5 Solar Heat Collector Requirement.

According to reference [13] liquid flat plate collector can give temperature up to 95^oC. Their Advantages are simple construction, relatively low cost, ease of maintainer and repair, no moving parts and durability. The current value of solar constant, which is defined as the intensity of solar radiation to the on a surface normal to the sun, just beyond the atmosphere is $1367W/m^2$.

ASHRAE 2007 HVAC Application hand book's chapter 33,[14] gives the specific directions on how to decide total solar heat collector area. Solar heat collector area requirement calculation done based on following facts and assumptions.

3.1.5.1 Calculations For Flate Plate Solar Heat Collectors.

The Location of experiment is Vadodara city which has coordinates $22.3^o N$ and $73.19^o E$. Indian Standard Time meridian is known to be $82.5^o E$. For the sake of calculation, randomly chosen date of experiment is $1^{st} March$ which is a 60^{th} day of the year. Time of the experiment is chosen to $9:00AM$, because at this time of morning, it is assumed, that lowest possible radiation is available on which our system can work, and if area of heat collectors can fulfill the requirement at 9am then it can also fulfill the requirement for rest of the day. Collector's direction and position is imagined to be facing exactly south and tilting $30^o$from the horizontal. Atmospheric temperature is considered to be $35^o C$ as earlier cases. Average temperature of the heat collector plate is considered to be $70^o C$.

3.1.5.2 Apparant Solar Time.

From eq.4 of[14] we can find Apparant Solar Time (AST) which will be useful to calculate hour angle.

Apparant Solar Time,

$$AST = LST + Equation\, of\, Time + 4min.(LST\, meridian - Lcocal\, meridian) \tag{3.88}$$

$$AST = 9:00 + (-12.5) + 4(82.5 - 73.19) = 9:24:45 \tag{3.89}$$

So, we are $12:00:00 - 09:24:45 = 02:35:15$ away from solar noon. or say $2.58 hours$ away from solar noon.

3.1.5.3 Calculations for Various Solar Angles.

Now, eq.1 of [14] provides the solar declination which changes every day because of earth's rotation about sun and its tilted axis.

Declination angle,

$$\delta = 23.45\, Sin \left[(360)\, X\, \frac{(284) + (N)}{(365)} \right] = -8.29^o \tag{3.90}$$

Where, Year's day N = 60.

From Table 2 in [14] we can find equation of time to be -12.5 for our case.

Eq. 3 of [14] gives the hour angle H,

$$H = (Number\, of\, hours\, from\, solar\, noon)\, X\, 15^o = (2.58).15^o = 38.7^o \tag{3.91}$$

Eq. 5 of [14] gives Altitude angle β,

$$Sin\beta = Cos(LAT).Cos\delta.cosH + Sin(LAT).Sin\delta \tag{3.92}$$

$$\beta = Sin^{-1}(0.6716) = 42.19^0 \tag{3.93}$$

Where, $LAT = 22.3^oE$.

Eq.6 of [14] gives Azimuth angle ϕ,

$$Sin\phi = Cos\delta.SinH/Cos\beta \tag{3.94}$$

$$\phi = Sin^{-1}(0.8130) = 54.39^o \tag{3.95}$$

Eq.8 of [14] gives solar incident angle θ,

$$Cos\theta = Cos\beta.Cos\gamma.Sin\varepsilon + Sin\beta.Cos\varepsilon \tag{3.96}$$

$$\theta = Cos^{-1}(0.8873) = 27.46^o \tag{3.97}$$

Where, Tilt from horizontal$\varepsilon = 30^o$, and surface solar azimuth $\gamma = \phi$in our case, because our collector is exactly facing south.

3.1.5.4 Calculations for Total Solar Radiation

Now, from table 1 of [14] three constants can be found. They are Apparent solar irradiation at zero air mass $A = 1186W/m^2$ for month of march. Atmospheric extinction co-efficient $B = 0.156$ and Ratio of diffused radiation and direct radiation $C = 0.071$.

Now, Direct normal radiation I_{DN} can be calculated with the help of eq.11 of [14],

$$I_{DN} = A.e^{-B/Sin\beta} = (1186)e^{-0.156/Sin42.19} = 940.166W/m^2 \tag{3.98}$$

Amount of Diffuse Radiation $I_{d\theta}$can be calculated with the help of eq.14 of [14],

$$I_{d\theta} = C.I_{DN}.F_{ss} = (0.071)(940.166)(0.939) = 62.67W/m^2 \tag{3.99}$$

Where, $F_{ss} = (1 + Cos\varepsilon)/2 = 0.9390$

So, total solar radiation can be found with the help of eq.12 of[14],

$$I_{t\theta} = I_{DN}.Cos\theta + I_{d\theta} = (940.166)(Cos27.46) + 62.67 = 895.54W/m^2 \tag{3.100}$$

3.1.5.5 Useful Solar Radiation.

This is the value of total solar radiation that is available to the surface of the flat plate collector. Now, collector can not transfer all this energy to the fluid because it has some efficiency. Eq.24 of [14] gives usable energy that can be obtained from collectors.

$$q_u = I_{t\theta}(\alpha\tau)_\theta - U_L(T_p - T_{at}) \tag{3.101}$$

Where, $\alpha$is Absorptance and τ is Transmittance. Their values can be found from the table 4 of [14] for various incident angles. For our case their values are $\alpha = 0.95$ and $\tau = 0.87$.

$U_L$is collector lose co-efficient which accounts for the heat lose that take place in various cases like wind, snow etc. For our case its value from figure 12 of [14] comes to be $U_L = 7.6W/m^2K$.

T_p is an average plate temperature, which in our case we have taken as 70^oC, and T_{at} is atmospheric temperature, which in our case we have taken as 35^oC.

$$q_u = [(894.54)(0.95)(0.87)] - [7.6(70 - 35)] = 473.33W/m^2 \tag{3.102}$$

Our maximum possible heat supply requirement in generator is 15.64 kW as mentioned earlier. Total heat collector area required would be $15.64/0.4733 = 33.04m^2$. Now, it is to be kept in mind that the date and the time of experiment is chosen such that when solar radiation is not so strong. If instead of 9:00 am we take the time around 11:30am then with same calculation area requirement comes to be $24.07m^2$. and if the day is chosen of mid summer such as any day of months May or June when the solar radiation is at its full height the collector area will come even less. But, if we need our system to work in all seasons then $34\,or\,36m^2$of area is suggested. If Solar heat collectors of 3m x 2m area is used then five or six such collector should be sufficient. Market survey and selection or proper heat collector shall be mentioned in later chapters.

3.1.6 Generator Design

The generator and the SSST shall be physically the same component. A Tank which is containing strong solution, which can be used for 16 hours, can also be used as a generator. This tank should be at higher elevation than the solar heat collectors. A pipeline will take strong solution from tank to the solar heat collectors at lower elevations, where it will be heated and return to the SSST. This flow shall take place with the help of thermosiphan effect. Natural convection gives very low heat transfer co-efficient. so, to induce the flow we could joint the line coming from the pump directly to the pipe that is taking strong solution from generator to solar heat collectors as shown in schematic diagram 1. This way the flow in heat collectors can be induced a little bit and better heat transfer co-efficient can be achieved.

Here, the heat transfer surface is heat collectors it self. So, the generator design shall include the number of solar heat collectors required and its piping. Calculations for solar flate plate collectors area have been done in previous section and calculations for piping and arrangement of the same shall be included in chapter of physical design.

3.1.7 Condenser Design

Various aspects of condensation have been explained in various books and references [11, **?**]. For condensation in cases such as high vapor velocity vapor shear effect and vapor separation effect has to be taken in consideration. Such situation occurs where vapor is flowing with high velocities, like in condensers in power plant where steam turbine exhaust have very high velocity. If there are multiple tubes one over other then the inundation phenomena have to be taken in consideration which is well explained in [11].

But in our case refrigerant vapor mass flow rate is very low and it has insignificant velocity and we will also see at the end of the calculation that the heat transfer area requirement can be satisfied with just one pipe.

Pipe Selection for Condenser

The pipe selected for condenser is of 1" copper pipe instead of 0.5" in all other components. The reason for which is any smaller pipe than this diameter increases the cooling water velocity more than 3m/s (with 0.5" it gives about 6m/s) which is undesirable and not a standard practice. Flow velocity in pipe is usually kept near or bellow 2m/s. In selection of 1" pipe this criteria is satisfied.

Outside diameter of pipe $do = 25\,mm$

Inside diameter of pipe $di = 24\,mm$

C/S Area of pipe $A_{c/s} = (\pi/4)(0.024^2) = 0.0004523\,m^2$

3.1.7.1 Thermophysical Properties of Fluids in Condenser.

Thermophysical properties of cooling water shall be according to Table 3.9. Cooling water is same that is used for Absorber.

Thermophysical properties of the refrigerant vapor is given bellow...

Table 3.12: Thermophysical properties of refrigerant vapor in Condenser.

Property	Nomenclature	Value	Unit	Reference
Vapor inlet Temperature	T_{10}	95	^oC	[1]
Vapor inlet Enthalpy	H_{10}	2667.61	KJ/Kg	[1]
Condensate outlet Temperature	T_{11}	95	^oC	[1]
Condensate outlet Enthalpy	H_{11}	398.089	KJ/Kg	[1]
Letent heat of evaporation	H_{fg}	2269.521	KJ/Kg	[1]
Density (Liquid)	ρ_l	961.88	Kg/m^3	[1]
Density (Vapor)	ρ_v	0.5049	Kg/m^3	[1]
Dynamic Viscosity (Liquid)	μ_l	0.000297278	Pa.s	[1]
Dynamic viscosity (Vapor)	μ_v	0.0000120988	Pa.s	[1]
Specific Heat (Liquid)	Cp_l	4.2102	$KJ/Kg.K$	[1]
Specific Heat (Vapor)	Cp_v	2.0606	$KJ/Kg.K$	[1]
Thermal conductivity (Liquid)	k_l	0.6773	W/mK	[1]
Thermal conductivity (Vapor)	k_v	0.02454	W/mK	[1]

3.1.7.2 Condenser Inside Pipe Calculations

According to our previous calculations in heat and mass balance of condenser section, we need to reject 10.54kW of heat.

So, required mass flow of cooling water is,

$$M_{20,21} = \frac{Q}{Cp.\triangle T} = \frac{(10.54)}{(4.179)(2)} = 1.2610\, kg/s \tag{3.103}$$

Velocity of cooling water inside pipe is,

$$V = \frac{M_{21,22}}{\rho_{21,22}.A_{c/s}} = \frac{(1.2610)}{(993.643)(0.0004523)} = 2.80\, m/s \tag{3.104}$$

Renold Number,

$$Re = \frac{\rho.V.di}{\mu} = \frac{(993.643)(2.80)(0.025)}{(0.000705192)} = 94687.41 \tag{3.105}$$

Prandlt Number,

$$Pr = \frac{\mu.Cp}{k} = \frac{(0.000705192)(4.179)(1000)}{(0.642)} = 4.59 \tag{3.106}$$

Nusselt number calculations,

$$f = (1.58.ln(94687.41) - 3.28)^{-2} = 4.5505\, X\, 10^{-3} \tag{3.107}$$

$$f/2 = 2.2752\,X\,10^{-3} \tag{3.108}$$

$$Nu = \frac{(f/2)(Re - 1000)Pr}{1 + 12.7(f/2)^{0.5}(Pr^{3/2} - 1)}$$

$$Nu = \frac{(2.2752\,X\,10^{-3})(94687.41 - 1000)(4.59)}{1 + 12.7(2.2752\,X\,10^{-3})^{0.5}(4.59^{2/3} - 1)} = 473.26 \tag{3.109}$$

Heat Transfer Co-efficient,

$$hi = \frac{Nu.k}{di} = \frac{(473.26)(0.642)}{(0.023)} = 12659.70\,\frac{W}{m^2K} \tag{3.110}$$

3.1.7.3 Condenser Outside Pipe Calculation.

As mentioned in 3.1.1.1 heat rejection shall be of two types. sensible heat rejection and the latent heat rejection. The latent heat rejection part shall be taken care by the condenser. The sensible heat is very less which will be rejected in piping to RST or RST it self.

In our case for heat transfer co-efficient calculations two equations can be used from [11, **?**]. which are as follows...

(A) Average Heat Transfer co-efficient Equation by [11].

$$\frac{h.do}{k_l} = 0.728\left[\frac{\rho_l.(\rho_l - \rho_g).g.h_{fg}.do^3}{\mu_l.(T_{sat} - T_w).k_l}\right]^{1/4} \tag{3.111}$$

$$\frac{h.do}{k_l} = 0.728\left[\frac{(961.88)(961.88 - 0.5049)(9.81)(2269.521)(1000)(0.025)^3}{(0.000297278)(95 - 36)(0.6773)}\right]^{1/4} = 295.31$$

Heat Transfer Co-efficient,

$$ho1 = \frac{(295.31)(0.6773)}{(0.025)} = 8000.53\,\frac{W}{m^2K} \tag{3.112}$$

(B) Dhir and Lienhard (1971) equation.[?]

$$ho2 = 0.729\left[\frac{k_l^3.\rho_l.(\rho_l - \rho_v).g.h_{fg}}{\mu_l^2.(T_{sat} - T_s).do}\right]^{1/4} \tag{3.113}$$

Heat Transfer Co-efficient,

$$ho2 = 0.729\left[\frac{(0.6773^3)(961.88)(961.88 - 0.5049)(9.81)(2269.521)}{(0.000297278)^2(95 - 36)(0.025)}\right]^{1/4} = 10850.21\,\frac{W}{m^2K} \tag{3.114}$$

Average of both heat transfer co-efficient is $9425.37\,W/m^2K$. which shall be used in further calculations.

3.1.7.4 Condenser Overall Heat Transfer Co-efficient.

$$U_{con.} = \left[\frac{1}{hi} + \frac{1}{ho} + \frac{ln(do/di)}{2\pi kl}\right]^{-1} \tag{3.115}$$

$$U_{con} = \left[\frac{1}{12659.70} + \frac{1}{9425.37} + \frac{ln(25/23)}{2.\pi.(401)(1)}\right]^{-1} = 4967.96\,\frac{W}{m^2K} \tag{3.116}$$

3.1.7.5 Condenser Required Heat Transfer Area.

LMTD of the condenser shall be 59.

Required Heat Transfer area is,

$$A_{HT} = \frac{Q}{U_{con}(LMTD)} = \frac{(10.54)(1000)}{(4967.96)(59)} = 0.03595\,m^2 \tag{3.117}$$

Required length of pipe,

$$L_{pipe} = \frac{A}{\pi.do} = \frac{0.03595}{\pi(0.025)} = 0.457m \tag{3.118}$$

3.2 Design Methodology for Solar and Fuel Operated VAR System (Model 2)

Solar and gas hybrid system is a modification of the all design mentioned from section 3.1.1 to 3.1.7 Solar and Gas hybrid system is not designed to work 24 hours on solar energy. Here, in day time the system would work on solar energy but during the hours of the night it will used LPG, hydrogen or other combustible source of energy.

Because of this following modifications will take place in original Solar Operated VAR system.

(1) SSST, RST and WSST will be removed from the system because it does not required to store solar energy in the form of refrigerant or strong or weak solution. Therefore the size of the system shall reduce.

(2) Refrigerant flow rate will be same in entire system. Unlike earlier design it is not required to keep different mass flow rates in high pressure side and low pressure side. This will be most helpful in reducing the size of the condenser and generator.

(3) Heat Supply requirement in generator shall be less with compare to earlier system. Earlier we were supplying 15.64kW of heat in generator because it has to evaporate more mass of refrigerant, But if mass of refrigerant vapor evaporated is less then the heat supply required will also be less.

(4) Solar heat plate collector area shall be reduced because of less heat requirement of the generator. Solar heat plate collectors and storage tanks are most expensive items in bill of material and if they can be reduced or eliminated then capital cost of system can be reduced.

The purpose behind the calculation of this arrangement is to observe the difference in economical aspect of all three arrangements.

During reviewing the calculations for model 2 the schematic diagram 2 for should be kept in sight for quick review which is attached in Appendix A.

3.2.1 Evaporator Design.

Evaporator design shall remain same as 3.1.1.1 and 3.1.2. There will be no change in any parameters of evaporator.

3.2.2 Absorber Design.

Absorber Design also shall remain same as 3.1.1.2 and 3.1.3.

3.2.3 Heat Exchanger Design.

Design of evaporator and absorber mentioned in section 3.1 does not change for model 2 and 3 because refrigerant mass flow rate does not change in model 2 and 3. However the design of heat exchanger, condenser, generator and solar heat collector area required will change for model 2 and 3. In Model 1 the heat exchanger has to deal with higher mass flow rate of the cold LiBr-water solution that is running from absorber to generator, however in model 2 this mass flow rate is much less with compared to model 1 because higher mass flow rate of refrigerant in generator and condenser is not required in this model.

3.2.3.1 Heat and Mass Balance of heat exchanger for model 2.2.

Heat available with hot solution coming from Generator,

$$Q_{hot,7} = M_{7,8,9}(H_7 - H_8) = (0.013583)(230 - 150) = 1.0866KW \tag{3.119}$$

Heat required by cold fluid going from Absorber to Generator via pump,

$$Q_{cool,3} = M_{3,4,5}(H_5 - H_4) = (0.0149889)(180 - 110) = 1.0492KW \tag{3.120}$$

Hot solution has more heat than the cold solution requirement. so, it can be assumed that this time the heat requirement of the cold fluid shall be fulfilled to reach the state mentioned in the equilibrium diagram. So, generator need not to put any extra heat input to the solution.

Pipe Selection for Heat Exchanger

In design of model 1 (Solar operated VAR) double pipe heat exchanger was considered in which there was only one internal pipe (1/2" Dia.) and one external pipe (1" Dia.). But for model 2 (Solar & Fuel operated) and model 3 (Fuel operated) different arrangement of heat exchanger has been considered. Here the heat exchanger will have seven internal copper pipes (1/2" Dia.) inside of larger steel pipe (4" Dia.). Detail arrangement drawings of which can be found in Appendix B. The intension behind this selection is to reduce the number of threaded joints. The double pipe heat exchanger has many threaded joints which increases the possibility of vacuum leakages.

Inner diameter of inner pipe $di = 12.5\,mm$

Outer diameter of inner pipe $do = 15\,mm$

Inner diameter of outer pipe $Di = 100\,mm$

Outer diameter of outer pipe $Do = 102\,mm$

3.2.3.2 Heat Exchanger Thermophysical Properties for Hot and Cold solutions

The operating temperatures of generators and the absorber will not be different from model 1. So, the thermophysical properties for hot and cold solution shall be same as 3.11 and 3.10.

3.2.3.3 Heat Exchanger Inside Pipe Calculations

Cross sectional area of inner pipe is $A_{c/s} = (\pi/4)(0.0125^2) = 0.00012271\,m^2$

Velocity of cold fluid inside the pipe $V = M_{3,4,5}/\rho_2.A_{c/s} = (0.0149889)/(1660)(0.00012271) = 0.07358\,m/s$

Renold Number inside pipe,

$$Re = \frac{\rho_2.V.di}{\mu_2} = \frac{(1660)(0.07358)(0.0125)}{(0.0043)} = 355.06 \tag{3.121}$$

Prandlt Number inside pipe,

$$Pr = \frac{\mu_2 Cp_2}{k_2} = \frac{(0.0043)(1.94)(1000)}{(0.430)} = 19.4 \tag{3.122}$$

Flow will be laminar, it would have been laminar even if we had used 1/8" pipe which is smallest available standard size.

According to ASHRAE Fundamentals chapter 4, Table 8, eq.(T8.4b) Nusselt number for fully developed laminar flow is...

$$Nu = 4.36 \tag{3.123}$$

convective heat transfer co-efficient is...

$$hi = \frac{Nu.k_2}{di} = \frac{(4.36)(0.430)}{(0.0125)} = 149.98\,\frac{W}{m^2K} \tag{3.124}$$

3.2.3.4 Heat Exchanger Out Side Pipe Calculation

Cross sectional area of annulus is $A_{c/s} = (\pi/4)(0.1^2 - 7(0.015^2)) = 0.0066169\,m^2$

Hydrolic diameter of the annulus is $D_{hyd} = 4A_{c/s}/Wetted\,Perimeter = (4\,X\,0.0066169)/\pi(0.1+(7\,X\,0.015)) = 0.04109\,m$

Velocity of hot fluid inside the annulus $V = M_{3,4,5}/\rho_2.A_{c/s} = (0.01326)/(1753.81)(0.0066169) = 0.0011426\,m/s$

Renold Number inside annulus,

$$Re = \frac{\rho_2.V.D_{hyd}}{\mu_2} = \frac{(1753.81)(0.0011426)(0.04109)}{(0.00455)} = 18.096 \tag{3.125}$$

Prandlt Number inside annulus,

$$Pr = \frac{\mu_2 Cp_2}{k_2} = \frac{(0.00455)(1.75)(1000)}{(0.424)} = 18.779 \tag{3.126}$$

According to ASHRAE Fundamentals chapter 4, Table 8, eq.(T8.4b) Nusselt number for fully developed laminar flow is...

$$Nu = 4.36 \tag{3.127}$$

convective heat transfer co-efficient is...

$$ho = \frac{Nu.k_2}{di} = \frac{(4.36)(0.424)}{(0.04109)} = 44.99\,\frac{W}{m^2K} \tag{3.128}$$

3.2.3.5 Heat Exchanger Overall Heat Transfer Co-efficient

Over all heat transfer co-efficient shall be different than model 2.1.

$$U_{hx.} = \left[\frac{1}{hi} + \frac{1}{ho} + \frac{ln(do/di)}{2\pi kl}\right]^{-1} \tag{3.129}$$

$$U_{hx} = \left[\frac{1}{149.98} + \frac{1}{44.99} + \frac{ln(15/12.5)}{2\pi(401)(1)}\right]^{-1} = 34.52\,\frac{W}{m^2K} \tag{3.130}$$

3.2.3.6 Heat Exchanger Heat Transfer Area Required

Logarithmic Mean Temperature Difference,

$$LMTD = \frac{\triangle T_{inlet} - \triangle T_{outlet}}{ln(\triangle T_{inlet}/\triangle T_{outlet})} = 24.28 \tag{3.131}$$

Heat Transfer Area,

$$A = \frac{Q}{U_{hx}(LMTD)} = \frac{(1.0492)(1000)}{(34.52)(24.28)} = 1.2518\,m^2 \tag{3.132}$$

Length of pipe required,

$$L_{pipe} = \frac{A}{\pi.do} = \frac{(1.2518)}{\pi(0.015)} = 26.56\,m \tag{3.133}$$

3.2.4 Generator Design.

Here also the generator will be connected with solar heat collectors directly from higher elevations so that heating can take place with the help of thermosiphan effect. But since the mass flow rate of the refrigerant vapor generator has to produce has changed in model 2, its heat supply requirements shall also be different.

$$M_5H_5 + Heat\,to\,be\,Supplied = M_{14}H_{14} + M_7H_7 \tag{3.134}$$

$$Heat\,to\,be\,Supplied = M_{14}H_{14} + M_7H_7 - M_5H_5$$

$$Heat\,to\,be\,Supplied = (0.00140616)(2667.61){+}(0.013583)(230){-}(0.0149889)(180) = 4.17KW \tag{3.135}$$

3.2.5 Solar Heat Collector Requirement

Here solar heat collector calculations shall remain same as mentioned in section 3.1.5. Only difference is heat input requirement in generator is much less with compare to earlier model. In model 1 our maximum possible requirements was 15.64kW where as hear it is 4.17kW. which is nearly one forth of earlier requirement.

Solar heat collectors can give $473.33W/m^2$ according to calculations we had in 3.1.5. So, required solar heat collector area in this case shall be $4.17/0.47333 = 8.80m^2$, or Say $10m^2$.

The economic implications of this fact is this model will be much more cheaper than the earlier model. In earlier model our requirement was 33 to 35 sq. meter of area where as in this case it is only 10 sq. meter. which is substantial saving in solar heat collectors expenses.

3.2.6 Condenser Design.

Here also the method for condenser design shall remain same as mentioned in 3.1.7 and 3.1.1.3. only difference being is the mass flow of refrigerant that condenser has to handle is less with compare to model 1 because of which many parameters of the condenser design shall change. which are as follows.

3.2.6.1 Condenser Heat and Mass Balance

Now, Heat balance of condenser is...

$$M_{15}H_{15} = M_{16}H_{16} + Heat\,Rejection \tag{3.136}$$

Here, $M_{15}$and $M_{16}$is same, the heat rejection will be...

$$Heat\,Rejection = M_{15,16}(H_{15}{-}H_{16}) = (0.00140616)(2667.61{-}167.533) = 3.51KW \tag{3.137}$$

$$Ltent\,Heat\,Rejection = M_{10,11}(H_{10} - H_{con}) = (0.00140616)(2667.61 - 398.089) = 3.19KW \tag{3.138}$$

$$Sensible\,Heat\,Rejection = M_{10,11}(H_{con}{-}H_{11}) = (0.00140616)(398.089{-}167.533) = 0.32KW \tag{3.139}$$

Pipe Selection for Condenser

Pipe selection for the condenser is same as model 1.

3.2.6.2 Thermophysical Properties of Fluids in Condenser.

Thermophysical properties of cooling water shall be according to Table 3.9. And thermophysical properties of the refrigerant vapor shall be same as mentioned in 3.12.

3.2.6.3 Condenser Inside Pipe Calculations

According to our previous calculations in heat and mass balance of condenser section, we need to reject 10.54kW of heat.

So, required mass flow of cooling water is,

$$M_{18,19} = \frac{Q}{Cp.\triangle T} = \frac{(3.51)}{(4.179)(2)} = 0.4199Kg/s \tag{3.140}$$

Velocity of cooling water inside pipe is,

$$V = \frac{M_{18,19}}{\rho_{18,19}.A_{c/s}} = \frac{(0.4199)}{(993.643)(0.0004523)} = 0.9343\, m/s \tag{3.141}$$

Renold Number,

$$Re = \frac{\rho\, V\, di}{\mu} = \frac{(993.643)(0.9343)(0.023)}{(0.0007051)} = 30282.64 \tag{3.142}$$

Prandlt number shall remain same as equation 3.106. which is 4.59,

Nusselt Number calculations,

$$f = ((1.58\,(ln(30282.64))) - 3.28)^{-2} = 5.8963\, X\, 10^{-3} \tag{3.143}$$

$$f/2 = 2.9481\, X\, 10^{-3}$$

$$Nu = \frac{(2.9481\, X\, 10^{-3})(30282.64 - 1000)(4.59)}{1 + 12.7(2.9481\, X\, 10^{-3})^{0.5}(4.59^{2/3} - 1)} = 178.89 \tag{3.144}$$

$$hi = \frac{Nu\, k}{di} = \frac{(178.89)(0.642)}{(0.023)} = 4993.36\, \frac{W}{m^2 K} \tag{3.145}$$

It can be seen from equation 3.111 and 3.113 that the convective heat transfer co-efficient for out side pipe (ho) does not depend on the mass flow rate of refrigerant. convective heat transfer co-efficient shall remain same as model 1 for outside pipe calculation which is $ho = 9425.37\, W/m^2K$. Value of LMTD shall also remain same which is 59.

$$U = \left[\frac{1}{4993.36} + \frac{1}{9425.36} + \frac{ln(25/23)}{2\,\pi\,(401)} \right]^{-1} = 2945.88\,\frac{W}{m^2K} \tag{3.146}$$

Required heat transfer area,

$$A = \frac{Q}{U(LMTD)} = \frac{(3.51)(1000)}{(2945.88)(59)} = 0.02019\,m^2 \tag{3.147}$$

Required length of pipe,

$$L_{pipe} = \frac{A}{\pi\,do} = \frac{(0.02019)}{\pi\,(0.025)} = 0.2570\,m$$

3.2.7 Requirement of Fuel.

As we have seen in generator heat balance calculation, we need to supply 4.17kW of heat in generator. According to our assumption only 8 hours of day solar energy is available and remaining 16 hours we need to use fuel. So, total heat supply per day to be given is $4.17\,X\,3600\,X\,16 = 240.192 MJ/day$. Table 3.13 gives Lower calorific values for various fuels and their required quantity for operation for model 1 and 3.

3.2.8 Theoretical COP of Model 2

In model 1 there was much bigger heat supply with compare to this model because of its refrigerant storage feature. But, in this model the heat supply required in generator is 4.21kW only. So, COP of this system will be much higher than the model 1.

COP of the system is, $3.5KW/4.21KW = 0.831$. in which pump work is neglected since its being much smaller. Model 3 Shall also have the same COP.

3.3 Design Methodology for Fuel Operated VAR System (Model 3)

Fuel based system is also 24 hour working except it does not use solar energy and depends only on fuel gas. This system has smaller size with compare to earlier two systems. Solar heat collector will completely be removed since they are of no use any more and they were the most expensive part of the system too. The fuel based system will be the cheapest among all the systems and also the smallest but its operating cost will be much higher compared to other two because of fuel cost.

It is to be noted that there will be no difference in design calculations done in model 2 and and model 3 except the mass flow rate of the fuel required, because in this case system will work 24hours on fuel.

3.3.1 Requirement of Fuel.

In This case system will use fuel for 24 hours. so, total heat supply requirements per day will be $4.17\,X\,3600\,X\,24 = 360.288MJ/day$.

Table 3.13: Fuel Requirement For system 2 and system 3

Fuel	LHV	Approx. price	Model 2.2		Model 2.3	
	MJ/kg	Rs/kg	kg/Day	Rs/day	kg/Day	Rs/day
Hydrogen*	119.96	200	2.00	400.00	3.00	600.00
LPG*	45.75	56	5.24	293.44	7.87	440.72
Kerosene*	43	30	5.58	167.40	8.37	251.10
Petrol*	44	75	5.45	408.75	8.18	613.50
Diesel*	43	53	5.58	295.74	8.30	439.90

*Note: Prices of all fuels are approximate. Values of density and LHV are refered from wikipedia.com and engineeringtoolbox.com.

Any of this fuel can be used to supply the heat by one gas heater with temperature control device which is placed in the path of weak solution. Hydrogen here may seem to be most expensive option but among these five fuels hydrogen is only one which is reproducible, clean fuel, environment friendly and having much higher calorific value with compare to others.

Chapter 4

Physical Design

The physical design mentioned in this chapter is for the model 3 which is completly fuel based. It was decided to build model 3 this year and check its technical feasibility. Solar panels shall be added in future to convert it in to model 2.

4.1 Evaporator and Absorber Unit

4.1.1 Schematic Arrangement of Evaporator and Absorber Unit.

Evaporator and Absorber use to have same pressure and they are most commonly in same vessel. Here in our model too we have kept evaporator and absorber in same vessel as shown in the picture.

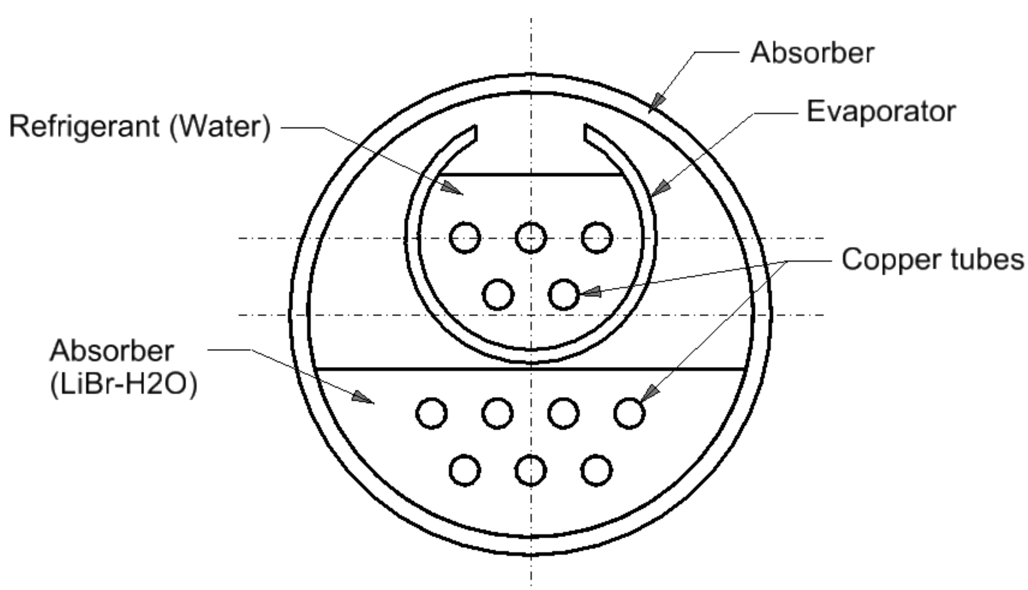

Figure 4.1: Schematic arrangement of Evaporator and Absorber

Figure 4.1 is a cross section of the Evaporator and absorber unit. Here evaporator is made of smaller pipe which has been kept open from the top through out its length and absorber is the larger outer pipe. Water at atmospheric temperature and pressure will flow from the five copper tubes that has been kept in evaporator. These five copper tubes will provide required heat transfer area for the evaporator. Once the required pressure (Vacuum) has been created in the empty space in bigger pipe, the water in evaporator will evaporate with the help of the heat received from the five copper tubes. This heat is taken from the water that is flowing inside the five copper tubes and hence that water will be cooled, and this way the cooling effect will be achieved. The evaporated water vapor will enter in the empty space of absorber from the open slot at the top of evaporator pipe. Larger pipe's lower half is filled with Absorbent (LiBr-water solution). This evaporated water vapor will get absorbed in the Absorbent and decrease the concentration of the absorbent. During this absorption process the temperature of the absorbent rises because of the two reasons. One reason is the chemical reaction between LiBr and water. The other reason is the returning quantity of hot solution form the generator. To maintain the absorbent temperature there are seven copper pipes have been passed within the absorbent solution from which the cooling water will flow. The number of copper tubes have been decided based on the calculations for heat transfer area requirement in previous chapter.

4.1.2 Dimensional Calculations

(1) Copper tubes as heat transfer surface.

Thermal conductivity of the copper is around $401\, W/mK$ where as thermal conductivity of aluminum is around $205\, W/mK$. so the selection of copper material has been done as heat transferring material. Dimension and the class of the copper tubes have been taken from "The Copper Tube Hand Book" [15] in which the standard sizes of the copper tubes have been mentioned in the various tables.These dimensions of the copper tubes are in accordance with the ASTM B88 standard. To ensure the minimum size of the whole equipment we have chosen the smallest pipe of 1/4" mentioned in table 2a in [15]. According to Subsection 3.1.2.5 Required heat transfer area for evaporator is$0.6585\, m^2$. Copper tubes has Inner Diameter $ID = 12.5\, mm$ and Outer Diameter $OD = 15.00\, mm$. So, our required length of the tube for evaporator comes $13.97\, m$. Similarly required heat transfer area for absorber is $0.6810\, m^2$according to subsection 3.1.3.5. Copper tube of same dimension have been used in absorber. So, our required length of the absorber comes $14.45\, m$.

(2) Absorber Vessel

Figure above shows the 3d CAD model of the absorber internal arrangement. For the sake of good visibility the transparency of pipe and the blind flange have been increased.

There were two options to fabricate the evaporator and absorber vessels. one is in which we have to chose the steel sheet of required thickness and role it in our required diameter and make an cylindrical vessel out of it and the other is the purchase the ready made pipe

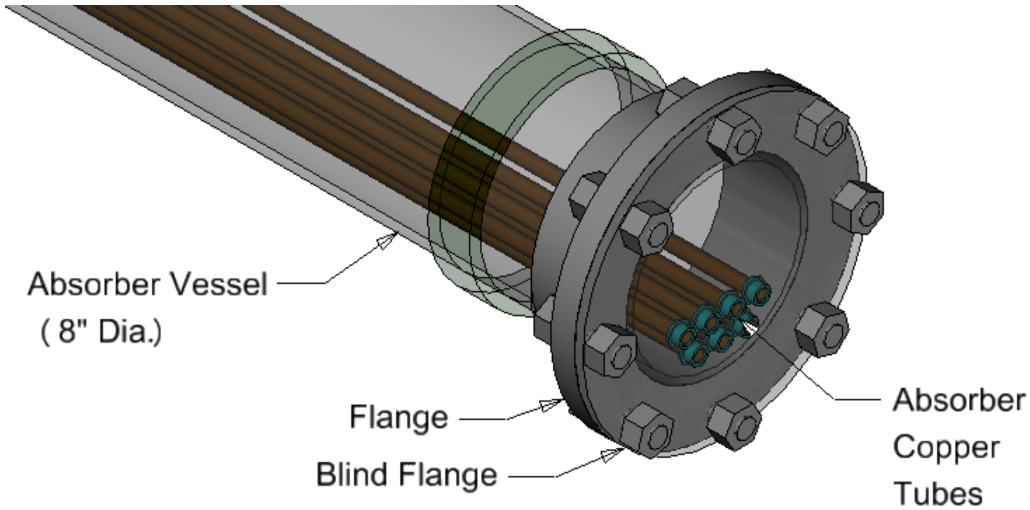

Figure 4.2: 3D Model of Absorber internal arrangement.

of that diameter from the market. Vessel thickness calculations which shall be covered in coming sections have reviled that the vacuum does not require the the thickness of the vessel more than one millimeter. So we could use the thin non standard pipe which can have the thickness of 2mm. If we have chosen to make the pipe of desired diameter out of steel sheet it would have required the equipments and precise skill of forming and welding the two mm thick sheet. So the choice of ready made pipe was cheaper and less time consuming. Pipes are available in market in standard size and length. The largest easily available size was 8" dia. If the sizes are larger than this than they will have to be manufactured and purchased by special order and it has to be in large quantity. Whereas our requirement was hardly 3 to 4 meters. Therefore it was decided to use 8" diameter pipe for absorber vessel. and to close that pipe from both the ends it was decided to use flange and blind flange. That blind flange will have total seven holes to pass the absorber copper pipes and one larger pipe for the evaporator. These copper pipes are fixed with the blind flange with the help of threaded joint. And the bushings of 1/4" X 1/2" will be used for that. The length of the absorber vessel was decided based on the the length of the copper tube required in absorber to maintain its temperature at absorber temperature which is 3.5 m.

(3) Evaporator Vessel

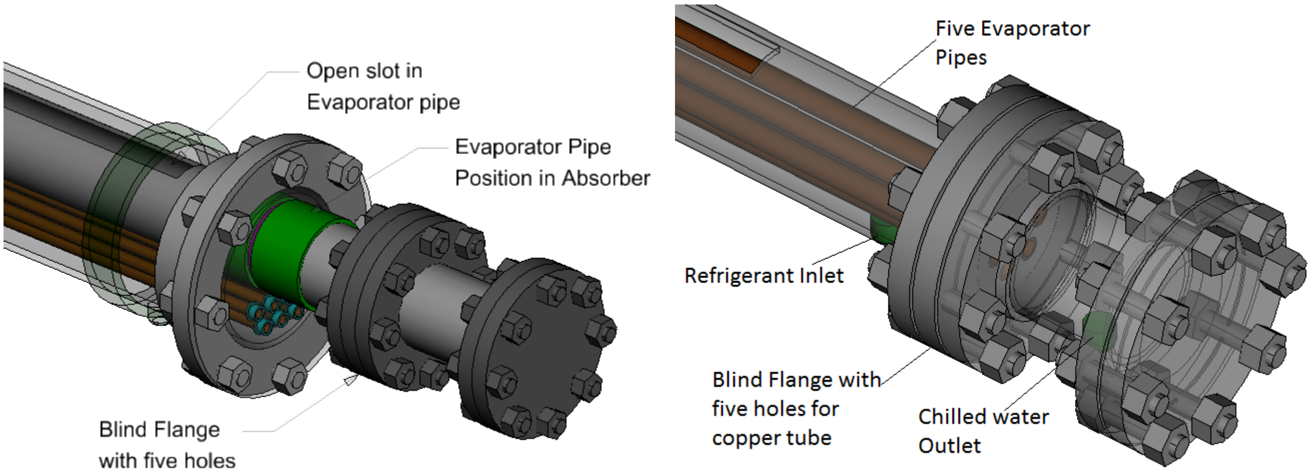

Figure 4.3: 3D Model of Evaporator internal arrangement.

Above images shows the position of the evaporator vessel in the Absorber vessel. As it can be seen in first image that the evaporator runs through the absorber and the slotted portion of evaporator remains in side the absorber vessel. The working significance have been explained at the start of this section. Required length of copper tubes for evaporator comes to be 13.97 m. As it can be seen from above images that the pipe larger than 4" would not leave much space in the absorber vessel. So the evaporator vessel will have to be made of 4" diameter pipe, and as it can be seen that the 4" diameter pipe will be able to accommodate 5 pipes in its lower half, which can remain submerged in the refrigerant (water). Therefore the required length of evaporator comes to be $13.97/5 = 2.794\,m$. But the flange of the evaporator has to be at certain minimum distance from the absorber's blind flange otherwise it would create a blockage for those seven pipes of the absorber. Therefore the length of the evaporator was kept 4m.

4.1.3 General Arrangement and Detailed Drawings.

Here is a drawing of the absorber and Evaporator with their dimensions. The details of their joints are shown in detail sketches. For more detail drawings please refer appendix.

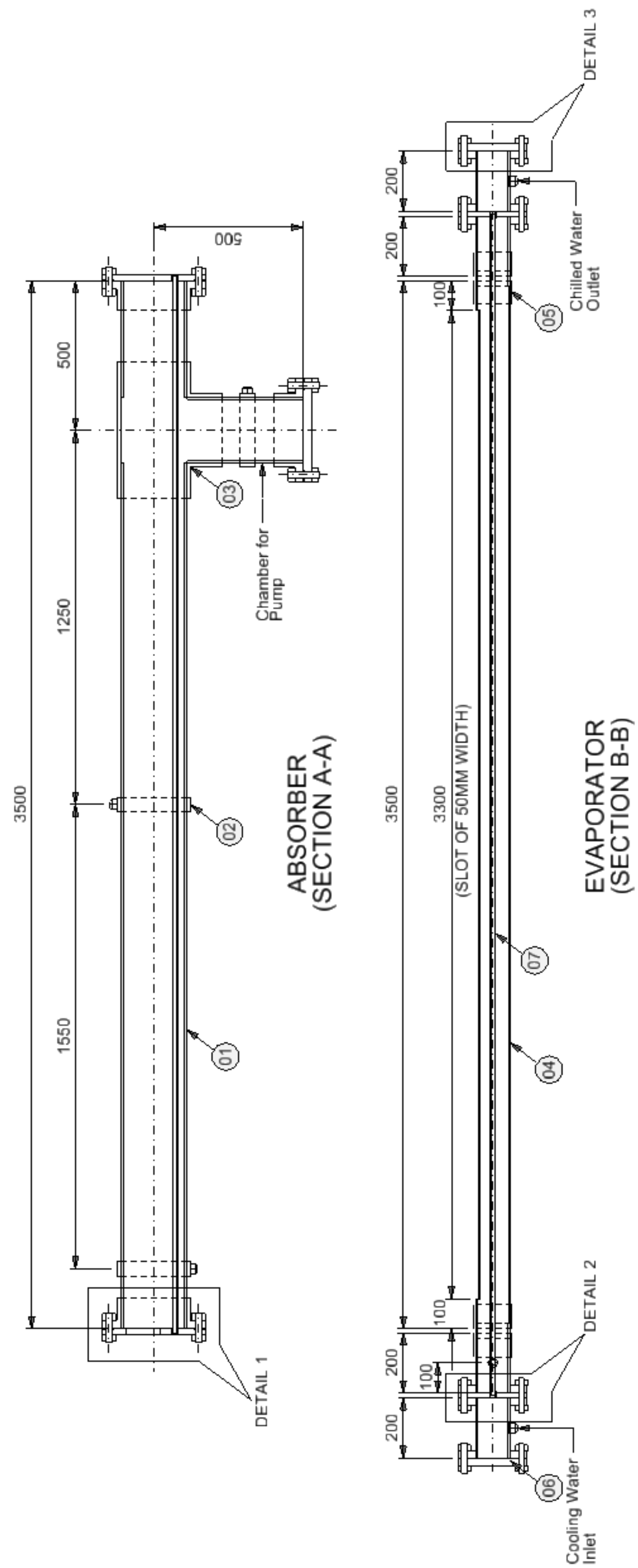

Figure 4.4: Dimensional Drawing of Absorber and Evaporator

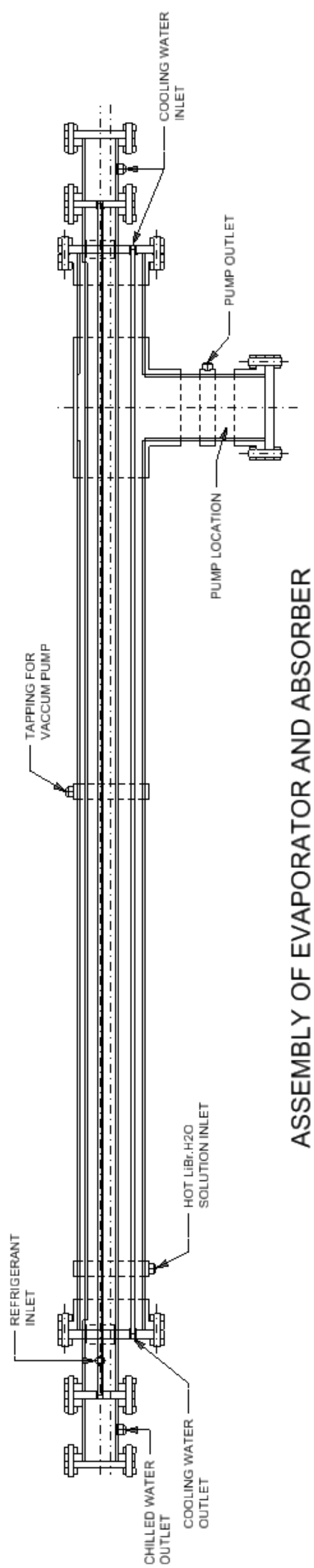

Figure 4.5: Mid section of Absorber Evaporator Assembly

4.2 Heat Exchanger Unit

The Heat Exchanger used here is simple pipe in pipe single pass heat exchanger. Heat exchanger thermal calculations are done in previous chapter. The pipe which can be used as a heat exchanger shell is 4” diameter pipe. In this diameter of pipe seven copper pipes can be accommodated. According to previous chapter calculations the required heat transfer area for the heat exchanger is $1.2518\,m^2$and the required length of copper pipe is $26.56\,m$. With this information the required length of heat exchanger comes to be $26.56/7 = 3.794\,m$.

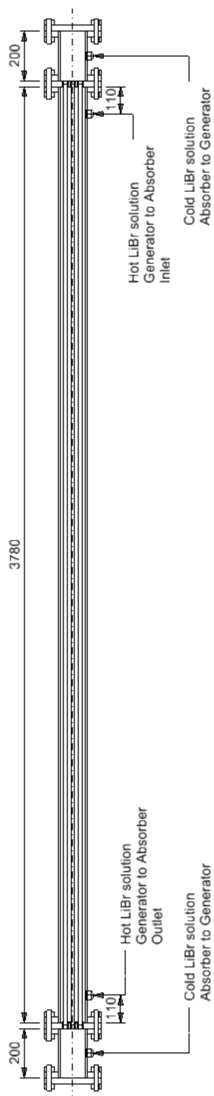

Figure 4.6: Mid Section of Heat Exchanger

4.3 Condenser and Generator Unit

4.3.1 Schematic Arrangement of Condenser and Generator Unit

Condenser and generator has same pressure so they have been kept in the same vessel just like absorber and evaporator, but the pressure of generator and condenser will be higher than the evaporator pressure. The schematic arrangement of the condenser generator unit is given in the figure bellow.

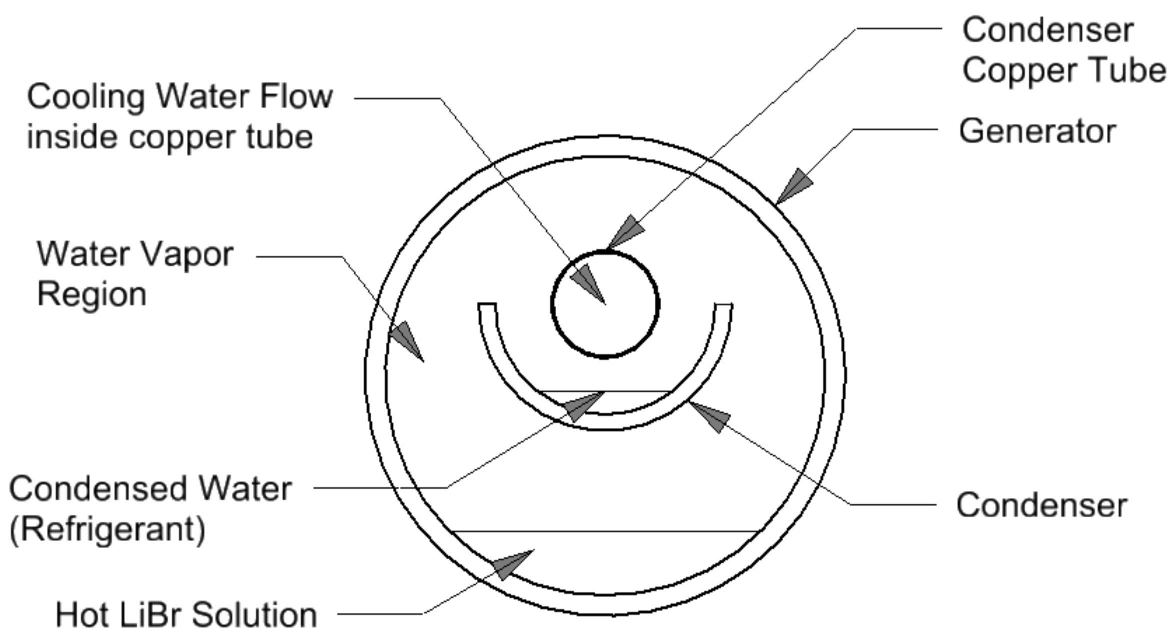

Figure 4.7: Schematic arrangement of Condenser Generator Unit.

Generator is made of 4" MS steel pipe inside which 3/4" copper pipe has been passed which shall work as a condensing surface. Another half cut 2" pipe has been passe through the generator vessel bellow the copper tube to accumulate the condensate dripping from the copper tube surface. This condensate (Refrigerant / Water) will than be taken out side the vessel and sent to the evaporator through the capillary tube which will work as a expansion device. LiBr solution will be heated in the gas heater and then immediately enter the generator vessel where the refrigerant water will vaporize and the concentration of the solution will increase. This water vapor will spread in the whole empty volume of the generator and then will come in contact with the outer surface of copper tube. Cooling water flows continuously inside this copper tube which keeps the copper surface cold. The water vapor will condense on the surface and get accumulated in the half cut MS pipe.

4.3.2 Dimensional Calculations

(1) Copper Tube Selection.

The sizing of the generator and absorber vessel depends upon the length of the copper tube length which will act as a condenser. In condenser required mass flow of the cooling water is 605.703 kg/hr. This much mass flow rate can not flow through the copper pipe of 15mm OD and 12.5mm ID without the increasing the velocity more than 2 m/s. Therefore the larger diameter of the copper tube has to be selected for the condenser. Therefore the copper tube selected for the condenser has the OD of 25mm and ID of 23mm.

(2) Generator and Condenser Length

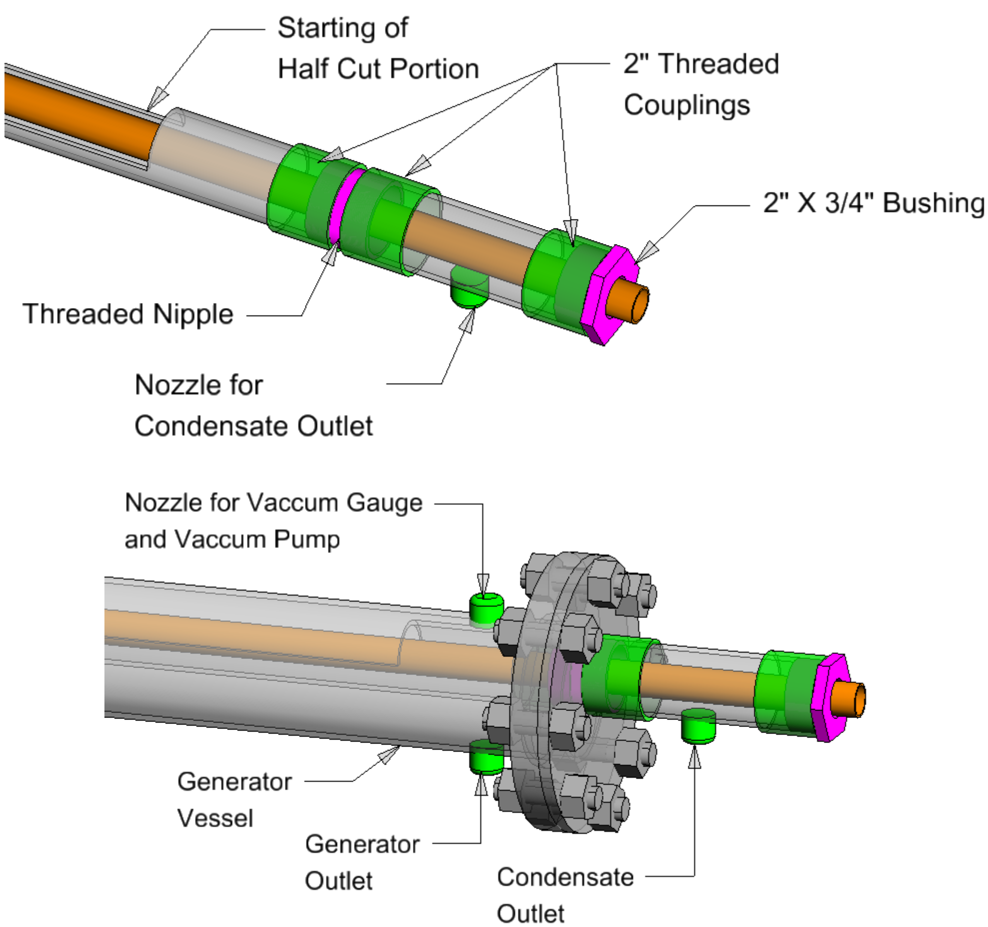

Figure 4.8: 3D Model of Condenser and Generator

According to the calculation done in the previous chapter the heat transfer area required for the condenser is $0.02019\,m^2$and the required length of the pipe is $0.257\,m$. Say 0.4m. In this system the LiBr solution is getting heated directly by the gas heater, but if the solution was to be heated indirectly like with hot water flowing inside tube ans the solution is outside the tube then the required length of the generator vessel comes around 1.5m. the calculations of which have not been shown in this report since we are not using this kind of generator vessel. In future it is intended to change the heat source from fuel to solar energy, in which case the bigger generator vessel will be required. This is why the length of the generator vessel has been kept 2.65m.

4.3.3 General Arrangement, Detailed Drawings.

Following are the general arrangement and assembly drawings of Condenser and Generator Unit. For detailed drawings please refer the drawings in Appendix.

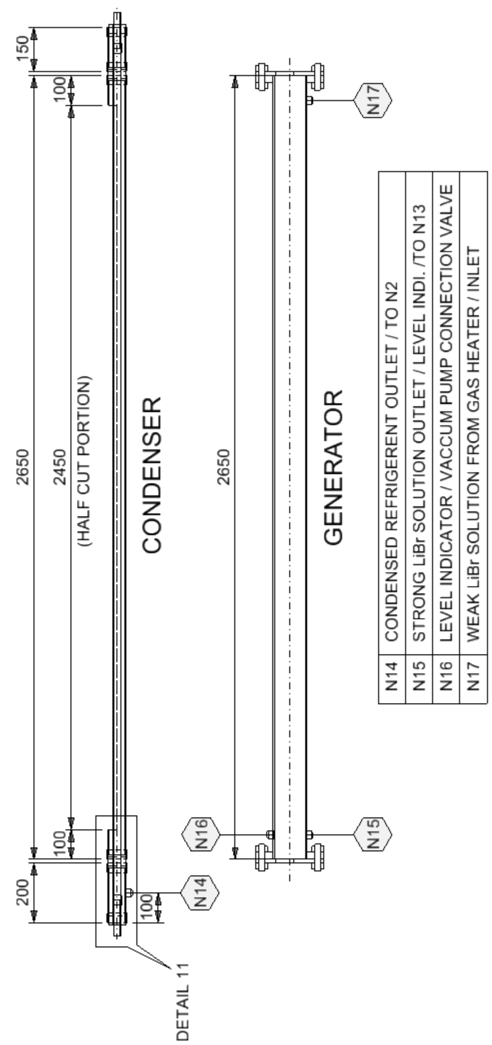

Figure 4.9: Dimensional Drawing of Condenser and Generator

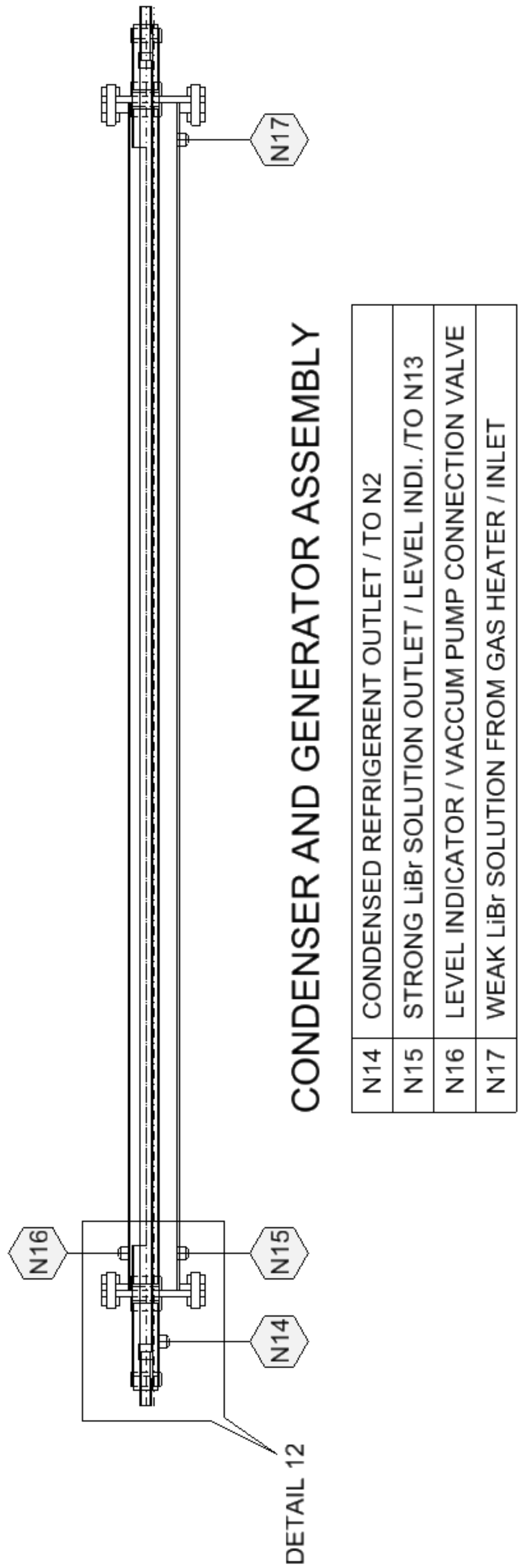

Figure 4.10: Mid Section of Condenser Generator Assembly

4.4 Wall Thickness Calculation

Wall thickness calculations for cylindrical vessels subjected to internal and external pressure can be found in ASME section 8 Div.1, UG-27,28. In our system there are total two cylindrical components which are going to experience different pressures on their either sides. (1) Copper tubes in evaporator and absorber vessel will have atmospheric pressure inside them and 0.008725 bar (absolute) outside. Similarly the copper tube of condenser will experience the atmospheric pressure inside and 0.0732 bar (absolute) outside. These two copper tubes shall be considered as a cylindrical vessels subjected to the internal pressure. (2) The absorber vessel will have 0.008725 bar absolute pressure in side and the atmospheric pressure outside. Similarly the heat exchanger and the generator vessel will have 0.0732 bar absolute pressure inside and the atmospheric pressure outside. These three vessels shall be considered as cylindrical vessels subjected to external pressure.

4.4.1 For Vessels Subjected to Internal Pressure.

Here there are two dimensions of copper tube is used. Coper tubes in Evaporator, Absorber and Heat Exchanger has the Outer Diameter OD = 15 mm and the Inner Diameter ID = 12.5 mm, Where as copper tube in Condenser has the Outer Diameter OD = 25 mm and Inner Diameter ID = 23 mm. Both copper tubes are having same internal pressure but different external pressure. Properties of copper are shown in the table bellow.

Property	Nomenclature	Value	Unit	Reference
Material of pipe		copper (C10200)		[4]
Tensile Strength	$\sigma_{tensile}$	30000	psi	[4]
Yield Strength	σ_{yeild}	9000	psi	[4]
Thermal conductivity of pipe	k_{pipe}	401	W/mK	

According to ASME Section 8, Div 1 [5] following are the formulas to find out wall thickness for circumferential stress and longitudinal stress.

Wall thickness for circumferential stress...

$$t_c = \frac{PR}{SE - 0.6P} = \frac{(101300)(0.00625)}{[(6.2052\,X\,10^7)(0.7)] - [(0.6)(101300)]} = 0.00145mm \tag{4.1}$$

Here, P is atmospheric pressure in Pa. R is a radius of pipe in m. S is yield stress in Pa. E is joint efficiency which is taken as 70%. same goes for next equation.

Wall thickness for longitudinal stress...

$$t_l = \frac{PR}{2SE + 0.4P} = \frac{(101300)(0.00625)}{[2(6.2052\,X\,10^7)(0.7)] + [(0.4)(101300)]} = 0.00728mm \tag{4.2}$$

Our pipe thickness is 1.25mm. so design is safe. For the copper pipe of the condenser if we replace the value of R in above equation with the 0.0125 m we will get the thickness for circumferential stress is 0.002919 mm and the thickness for longitudinal stress is 0.01456 mm.

4.4.2 For Vessels Subjected to External Pressure.

The calculation procedure for the shell thickness has been mentioned in UG-28 of ASME section 8 div.1. The applicable formula for our case is...

$$P_a = \frac{2AE}{3(D_o/t)} \tag{4.3}$$

Where,

P_a= Calculated value of maximum allowable external working pressure for the assumed value of t.

A = A factor determined from Fig. G in Subpart 3 of ASME Section II, Part D.

E = Young's Modulus of material

D_o= External Diameter of the vessel

t = Shell thickness

(1) Generator and Heat Exchanger Thickness Calculation

According to the procedure mentioned in standard the thickness first has to be guessed. Here we guess the thickness of the generator vessel to be 1mm. The generator vessel is made of 4" diameter pipe so its external diameter will be 100mm. This gives us the $L/D_o = 2.6m/0.1m = 26$ and $D_o/t = 0.1m/0.001m = 100$. Based on this data we can find the factor A from the fig. G in Subpart 3 of ASME Section II, Part D. The value of factor A is found to be 0.0001. The young's modulus of mild steel is known be 210 GPa. Putting these values in equation above...

$$P_a = \frac{2\,(0.0001)(210\,X\,10^9)}{3(0.1/0.001)} = 140000\,N/m^2 = 1.4\,bar$$

Which means the 1mm thickness is of generator vessel is sufficient to take the external pressure of 1.4 bar. Where as the atmospheric pressure is not more than 1.013 bar. We may add another one mm thickness for the corrosion allowence. The required thickness of the generator and heat exchanger shall be 2mm.

(2) Absorber Vessel Thickness Calculation

We have chosen 8" (200 NB) diameter pipe to be our absorber vessel. Here if the guessed value of the thickness is taken 1mm and repeated the calculation above, it can be found that the maximum allowable external pressure comes to be 0.7 bar which is less than the atmospheric pressure of 1.013 bar. So, the thickness of 1mm is not acceptable for the Absorber vessel. Now, if we take the guessed value of thickness to be 2mm and the outer diameter of the pipe to be 200mm then we can have $L/D_o = 3.5m/0.2m = 17.5$ and $D_o/t = 0.2/0.002 = 100$. The value of factor A will not differ much from previous value and will be taken as 0.0001. Young's modulus is taken as 210 GPa. Putting all these values in equation above the maximum allowable external pressure comes to be 1.4 bar. So, the thickness of 2mm is acceptable. We may add another 1mm thickness for the corrosion allowance. The required thickness of Absorber vessel will be 3mm.

Another alternative material for the absorber vessel was considered to be HDPE. The pipe of 8" diameter made of HDPE was considered to be viable option because of its cheapness. The young's modulus of the HDPE material is 0.8 GPa. If we guess the thickness of the HDPE pipe to be $t = 12\,mm$, and take $D_o = 200\,mm$ and the length of the pipe $L = 3.5m$, then the $L/D_o = 17.5$ and $D_o/t = 16.66$. Based on this data the factor A comes to be 0.005. Putting all these values in equation above the maximum allowable external pressure comes to be 1.6 bar. So, the thickness of 12mm is acceptable. Standard for HDPE pipes is IS 4984. According to this standard we need pipe of PE 80 material grade and PN 6 pressure rating, which has minimum wall thickness 11.4mm and maximum wall thickness 12.8mm.

4.5 Pumping Power and Head Lose Calculation

Location of pump, as mentioned before is in the chamber made in absorber vessel. This vertical chamber made in absorber vessel is 500mm long from center line so that the pump of 300mm length and 170mm width can be accommodated in it. The reference for the pumping power calculation is taken from [12].

The LiBr-water solution in the absorber is to be pumped from Absorber to Generator via Heat Exchanger and Heater. This piping rout can be divided in to Four parts. (1) From Absorber to Heat Exchanger entrance. The pump being located in side the Absorber it self it has no suction piping. The pump is of submersible kind and it has been chosen so to avoid the leakage through pump casing and pump-moter connection in case of non submersible pump. Now, from Absorber to Heat Exchanger the LiBr solution will have 40^0C and accordingly other thermal properties. This temperature will change as it flows further. (2) The Heat Exchanger, in which the cold LiBr solution shall flow in the tube side. There will be 4m long seven copper tubes through which flow has to pass parallaly during which its temperature should rise up to 76^0C. (3) From Heat Exchanger to Gas Heater. During this path the temperature will remain 76^0C and its thermal properties accordingly. (4) From Gas Heater to Generator. In this path the Libr solution will have reached up to $90 - 95^0C$ and also will have started to form refrigerant vapors.

(1) Pressure lose from Absorber to Heat Exchanger

The input data for calculations is shown in table bellow.

Table 4.1: Input data for pressure drop calculation from Absorber to Heat Exchanger

Description	Nomenclature	Value	Unit
Absorber Pressure	P_{abs}	0.00863	bar
Generator Pressure	P_{gen}	0.07326	bar
Temperature of $LiBr - H_2O$	T	40	^0C
Density of $LiBr - H_2O$	ρ_2	1659.53	kg/m^3
Dynamic Viscosity of $LiBr - H_2O$	μ_2	0.004366	Pa.s
Mass Flow Rate of $LiBr - H_2O$	M	0.014669	kg/s
Number of 90^0Elbows		4	Nos.
Hinge Type Check Valve		1	Nos.
Diameter of GI pipe	D	0.015	m
Diameter of copper tubes in H.E.	D_{copper}	0.00925	m
Length of Straight pipe	L_{pipe}	1.10	m

c/s Area of pipe

$A = \left(\frac{\pi}{4}\right) D^2 = 1.7671\, X\, 10^{-4}$

Volume flow rate

$Q = M/\rho_2 = 0.014669/1659.53 = 8.83924x10^{-6} m^3/s$

Velocity of fluid in side pipe

$V = Q/A = 8.83924\, X\, 10^{-6}/1.7671\, X\, 10^{-4} = 0.05002 m/s$

Renold Number

$Re = \rho_2 V.D/\mu_2 = (1659.53)(0.05002)(0.015)/0.004366 = 285.19$

For Laminar flow the Friction factor is

$f = 16/Re = 16/285.19 = 0.05610$

L/D ratio for elbows and check valve is 60 and 110 according to [12].

Equation for pressure drop according to [12] is,

$$\triangle P_1 = 4.f.\rho_2 \frac{V^2}{2} . \sum \left[\frac{L}{D}\right] \tag{4.4}$$

$$\triangle P_1 = 4(0.05610)(1659.53) \frac{(0.05002^2)}{2} \sum \left[4(60) + 1(110) + \frac{1.10}{0.015}\right] = 197.21 Pa$$

Head lose due to this piping,

$$h_1 = \frac{\triangle P_1}{\rho_2 g} = \frac{(197.21)}{(1659.53)(9.81)} = 0.01211 m$$

(2) Pressure lose in Heat Exchanger

The heat exchanger consists of 4" dia outer pipe and seven 1/4" dia inner pipes. The inner pipes are of copper. Pressure drop shall be calculated for one copper pipe and will be multiplied by seven.

The mass flow rate in single copper pipe is

$M = 0.014669/7 = 2.09563\,X\,10^{-3}\,kg/s$

Diameter of copper pipe is 9.25mm so, c/s area of the copper pipe is

$A_{c/s} = 6.7200\,X\,10^{-5}\,m^2$.

Length of the copper tube is 4m

Inlet temperature of the heat exchanger is 40^0C and outlet temperature of the heat exchanger is 76.19^0C so the thermal properties like density and viscosity are taken at the average temperature of 58.09^0C.

Density of the solution is $\rho_{he} = 1648.49\,kg/m^3$And viscosity of solution is $\mu_{he} = 0.003403\,Pa.s$.

Volume flow rate of solution in one tube

$Q = M/\rho_{he} = 2.09563\,X\,10^{-3}/1648.49 = 1.271242\,X\,10^{-6}\,m^3/s$

Velocity of solution in tube

$V = Q/A_{c/s} = 1.271242\,X\,10^{-6}/6.7200\,X\,10^{-5} = 0.01891\,m/s$

Renold number

$Re_{he} = \rho_{he}V.D/\mu = (1648.49)(0.01891)(0.00925)/(0.003403) = 84.73$

For Laminar flow the the friction factor is

$f = 16/Re_{he} = 16/84.73 = 0.188835$

Pressure drop can be found by useing the 4.4.

$$\triangle P_2 = 4(0.188835)(1648.49)\frac{(0.01891^2)}{2}\left[\frac{4}{0.00925}\right] = 96.27\,Pa$$

This is a pressure drop for one copper pipe, for seven copper pipes the pressure drop will be

$\triangle P_2 = 96.27\,X\,7 = 673.90\,Pa$

So, The total head lose in heat exchanger is

$h_2 = \frac{\triangle P_2}{\rho_{he}g} = \frac{(673.90)}{(1648.49)(9.81)} = 0.01013\,m$

(3) Pressure lose from Heat Exchanger to Heater

There are total two elbows and approximately 2.5m length of straight half inch pipe from heat exchanger to gas heater. Calculations for this pipe rout is done separately because the temperature of the fluid will now be 76.19^0C and accordingly the thermal properties will be different from the other pipe routs. The diameter and mass flow rate of the pipe shall remain same as the case (1).

Density of the solution will be $\rho_3 = 1637.46\, kg/m^3$and Dynamic viscosity will be $\mu_3 = 0.002440\, Pa.s$

Volume flow rate

$Q = M/\rho_3 = 0.014669/1637.46 = 8.958386\, X\, 10^{-6}\, m^3/s$

Velocity of the solution is

$V = Q/A_{c/s} = 8.958386\, X\, 10^{-6}/1.7671\, X\, 10^{-4} = 0.05069\, m/s$

Renold Number

$Re_3 = (1637.46)(0.05069)(0.015)/(0.002440) = 510.263$

Friction factor

$f = 16/Re_3 = 16/510.263 = 0.031356$

For pressure drop using equation 4.4

$$\triangle P_3 = 4(0.031356)(1637.46)\frac{(0.05069^2)}{2}\left[2(60) + \frac{2.5}{0.015}\right] = 75.63\, Pa$$

Head lose

$h_3 = \triangle P/\rho.g = (75.63)/(1637.46)(9.81) = 0.00470m$

(4) Pressure lose from Heater to Generator

There is one gate valve, three elbows and approximately 2m long pipes make the pipe routing from heater to generator. Here the temperature of the fluid will be 95^0C.

Density of the solution will be $\rho_4 = 1627.503\, kg/m^3$and Dynamic viscosity will be $\mu_4 = 0.00191371\, Pa.s$

Volume flow rate

$Q = M/\rho_4 = (0.014669)/(1627.503) = 9.01319\, X\, 10^{-6}\, m^3/s$

Velocity of the solution

$V = Q/A_{c/s} = (9.01319\, X\, 10^{-6})/(1.7671\, X\, 10^{-4}) = 0.051005$

Renold Number

$Re_4 = (1627.503)(0.051005)(0.015)/(0.00191371) = 650.65$

Friction factor

$f = 16/Re = 16/650.65 = 0.02459$

L/D ratio for fully open gate valve is 7 according to [12].

For pressure drop using equation 4.4

$$\triangle P_4 = 4(0.02459)(1627.503)\frac{(0.051005^2)}{2}\left[3(60) + 1(7) + \frac{2}{0.015}\right] = 66.70\, Pa$$

Head lose

$h_4 = \triangle P/\rho.g = (66.70)/(1627.503)(9.81) = 0.0041776\,m$

So, Total Pressure and Head Lose due to friction is

$\triangle P_f = \triangle P_1 + \triangle P_2 + \triangle P_3 + \triangle P_4 = 1013.44\,Pa$

$h_f = h_1 + h_2 + h_3 + h_4 = 0.0311\,m$

Head lose due to Pressure difference according to topic 3.1.1.4

$h_p = 0.3975\,m$

The height difference between pump discharge and generator level is $h_l = 0.8m$

So, Total head lose is

$h_{Total} = h_f + h_p + h_l = 1.228\,m$

The head lose occurring in heater is unknown but it can not be more than one meter. So the maximum possible head lose in this pipe rout is 2.228 m. In this case we can safely select a pump which can give us head of 3m. and mass flow rate of 52 kg/hr. Such small capacity pumps are available in market at cost of 600/- to 800/- Rs. and have been used in this experimental set up.

4.6 Piping Design

For piping and instrumentation information please refer the schematic diagram for System 3 in the Appendix A. Three different kind of piping is required in this model. (1) LiBr-water solution piping. (2) Refrigerant piping. (3) Cooling Water piping. Figure bellow indicates the basic process flow diagram in which every location is indicated with the property of the fluid at that location.

Figure 4.12 is the Property table. Numbers indicated in red in figure 4.11 refers to the Numbers given in the first column of the property table.

4.6.1 LiBr-Water solution piping

Piping in which Lithium Bromide water solution flows shall be under vacuum (0.008725 bar) and high temperatures such as 95^oC. The maximum mass flow rate of this piping will be in the pipe running from pump to the generator which is around 52.81 kg/hr = 0.01466 kg/s. The density of the solution in this pipe shall be around $1659\,kg/m^3$. The volume flow rate will be $8.8366\,X\,10^{-6}\,m^3/s$, which is 0.0088366 liter/s. Now if we assume the velocity of the fluid flowing in the pipe to be 2 m/s then the cross section area of the pipe shall be $4.4183\,X\,10^{-6}\,m^2$, which gives us the required minimum diameter of the pipe to be 2.371mm. Now the smallest available pipe diameter is 1/8" which has the nominal bore of 6mm, which could have been possible to use for our application, but we have used the 1/2" pipe which has the diameter of 15mm. The reason behind choosing the larger pipe for the application

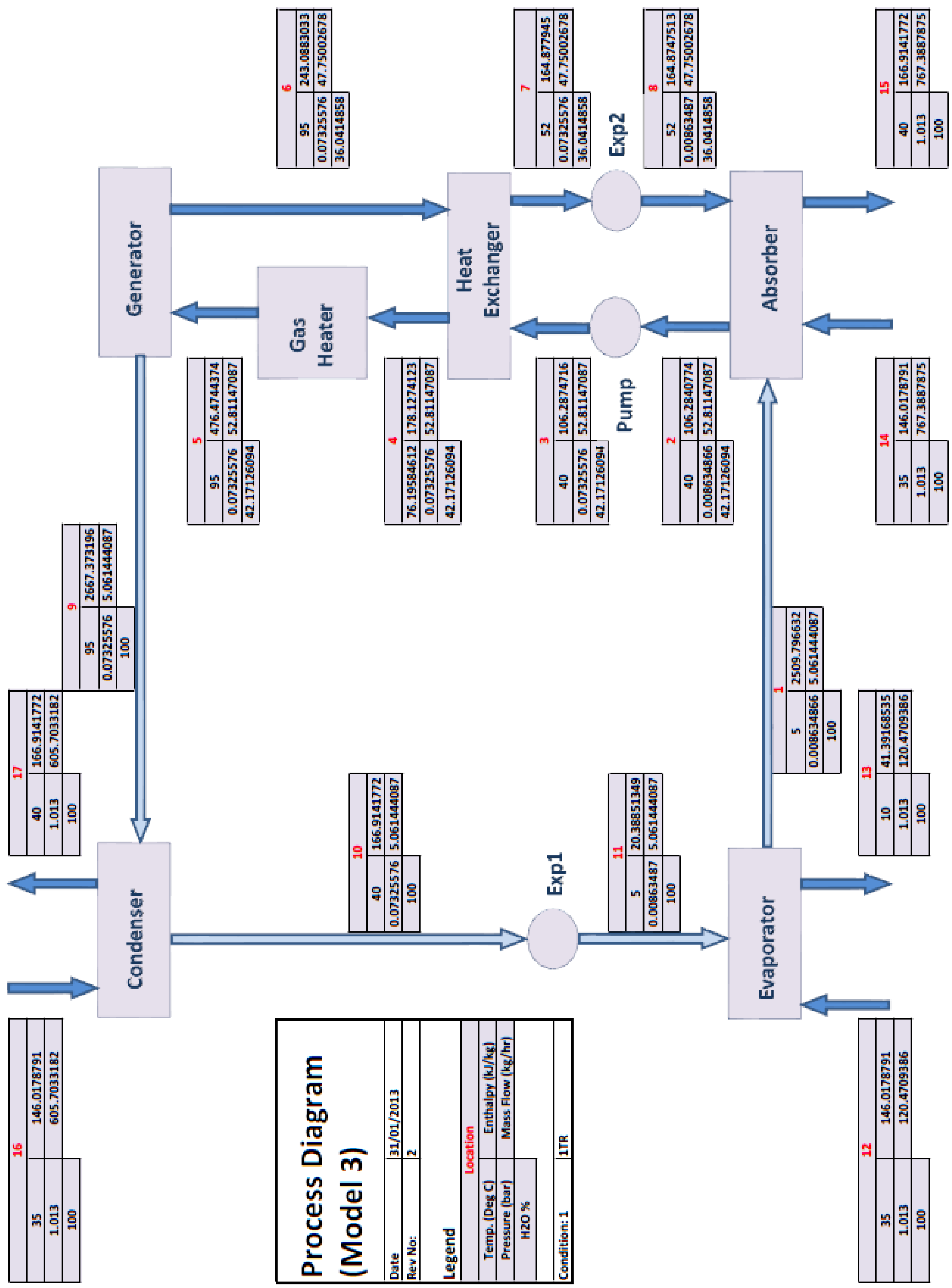

Figure 4.11: Process Diagram For Model 3

Property Table (Model 3)

	State of fluid	Mass fraction of water	Temperature	Pressure	Density	Dynamic Viscosity	Specific heat	Thermal conductivity	Enthalpy	Mass flow rate	
		%	Deg C	bar	Kg/m3	Pa.s	J/Kg K	W/m K	KJ/Kg	Kg/s	Kg/hr
1	Water Vapor	100	5.0000	0.00863487	0.006735	0.0000090858	1.8916	0.0168	2509.7966	0.0014059567	5.0614
2	Liq. Solution	42.17126094	40.0000	0.00863487	1659.531870	0.0043666597	1.9537	0.4307	106.2841	0.0146698530	52.8115
3	Liq. Solution	42.17126094	40.0000	0.07325576	1659.531870	0.0043666597	1.9538	0.4307	106.2875	0.0146698530	52.8115
4	Liq. Solution	42.17126094	76.1958	0.07325576	1637.469219	0.0024406506	2.0097	0.4558	178.1274	0.0146698530	52.8115
5	Liq. Solution	42.17126094	95.0000	0.07325576	1627.503012	0.0019137164	2.0282	0.4664	476.4744	0.0146698530	52.8115
6	Liq. Solution	36.04148576	95.0000	0.07325576	1753.818878	0.0030193108	1.8370	0.4366	243.0883	0.0132638963	47.7500
7	Liq. Solution	36.04148576	52.0000	0.07325576	1779.834501	0.0061193392	1.7944	0.4126	164.8779	0.0132638963	47.7500
8	Liq. Solution	36.04148576	52.0000	0.00863487	1779.834501	0.0061193392	1.7944	0.4126	164.8748	0.0132638963	47.7500
9	Water Vapor	100	95.0000	0.07325576	0.502283	0.0000120533	2.0581	0.0243	2667.3732	0.0014059567	5.0614
10	Water Liquid	100	40.0000	0.07325576	992.240326	0.0006545641	4.1788	0.6284	166.9142	0.0014059567	5.0614
11	Water Liquid	100	5.0000	0.00863487	999.919749	0.0015255272	4.2057	0.5720	20.3885	0.0014059567	5.0614
12	Cooling water	100	35.0000	0.05582093	994.047746	0.0007212801	4.1792	0.6218	146.0179	0.0334641496	120.4709
13	Cooling water	100	10.0000	0.01215894	999.666735	0.0013116524	4.1961	0.5816	41.3917	0.0334641496	120.4709
14	Cooling water	100	35.0000	0.05582093	994.047746	0.0007212801	4.1792	0.6218	146.0179	0.2131635521	767.3888
15	Cooling water	100	40.0000	0.07325576	992.240326	0.0006545641	4.1788	0.6284	166.9142	0.2131635521	767.3888
16	Cooling water	100	35.0000	0.05582093	994.047746	0.0007212801	4.1792	0.6218	146.0179	0.1682509217	605.7033
17	Cooling water	100	40.0000	0.07325576	992.240326	0.0006545641	4.1788	0.6284	166.9142	0.1682509217	605.7033

Figure 4.12: Property Table For Model 3

is that we already had total 12m long GI pipe available as scrape so it was not required to be purchased. Only the fittings, valves and few pipe nipples have to be purchased. Flexible tubes and hoses are not suitable for the use because high temperature and under the vacuum they will be crushed because of external atmospheric pressure.

4.6.1.1 Capillary tube calculation for LiBr Solution.

LiBr solution coming from the generator passes through the pressure reducing device in which its temperature and the enthalpy remains same but only the pressure drops. In this case like the refrigerant the phase change does not take place, so the following formula can be used to calculate the required length and the diameter of the capillary tube.

$$\triangle P = \frac{4\,f\,L\,V^2}{2\,g\,D} \tag{4.5}$$

Where $\triangle P$ is pressure difference which is known to be$0.06463\,bar = 6463\,N/m^2$. and f is a friction factor which is assumed to be 0.08. L is a length of capillary tube. V is velocity of the fluid flowing inside. g is gravitational acceleration and D is the diameter of the capillary.

Table 4.2: Capillary Tube Sizing for LiBr Solution

Pressure Difference		Length	Diameter	C/S Area	Velocity
N/m^2		m	m	$X\,10^{-7}\,m^2$	m/s^2
6463.089	6446.40	0.137	0.0005	1.962	37.98
	6429.36	0.34	0.0006	2.826	26.37
	6461.88	1.44	0.0008	5.024	14.83
	6455.22	4.39	0.001	7.850	9.49
	6448.14	33.3	0.0015	17.663	4.22

Table above gives the various lengths for the various diameter of capillary tubes. The first column which covers all five rows, with heading pressure difference indicates the known value of pressure difference which is $6463\,N/m^2$. The length and diameter are kept as a variables and all others parameters are fixed. By changing the length and diameter at random the value of pressure difference is achieved with trial and error method with use of MS-excel sheet. The values of pressure difference so achieved are given in the second column of heading pressure difference. It can be seen that the required length of the capillary tube increases beyond 33.3m as we choose the diameter bigger than the 1.5mm. Even 1mm diameter would require 4.39 m long capillary. This happens because as we increase the diameter of the capillary tube the velocity of the flow decreases substantially. The mass flow rate is $47.75\,kg/s = 0.01326\,kg/s$, which is very low. The density of the solution at the capillary tube is $1779\,kg/m^3$.

4.6.2 Refrigerant Piping

Refrigerant piping is the shortest piping of all the piping in this system. This piping connects nozzle number N14 and N2 according to the schematic diagram given in the Appendix. This piping does not contain anything except capillary tube. According to physical arrangement and the 3D model the distance between nozzle N14 and N2 is sufficiently less such that we may connect them directly to one another with capillary tube.

4.6.2.1 Capillary tube calculation for refrigerant (water)

As refrigerant comes from condenser to evaporator it passes through an expansion device. This expansion device could be a valve if the mass flow rate is considerably high or it could be a capillary tube in case of low mass flow rates. Pressure, Temperature and Enthalpy decreases as the refrigerant comes from high pressure side to low pressure side. Theoretically the enthalpy does not decrease and remains constant but practically Enthalpy also decreases. During this process some amount of refrigerant changes the phase and vaporizes. The method to determine required length and diameter of capillary has been mentioned in reference[16] and the same has been followed here. The Calculation has been performed in Excel sheet since it requires to do several iterations. Based on these design calculations capillary of 6mm diameter and of 2m length required to be used in present work. Another option for flooded type evaporator is float valve, however literature recommends its application with high capacity refrigeration. As capillary tube size in present work is quite long and large in diameter it was decided to use float valve for this application.

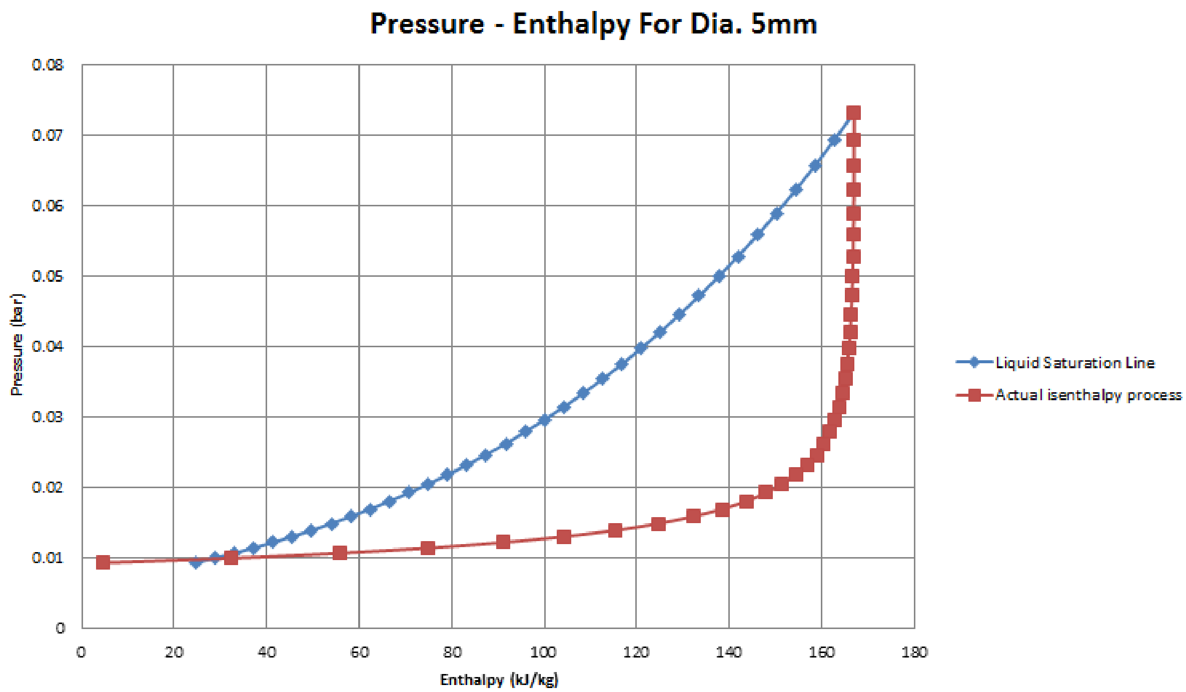

Figure 4.13: Actual Isenthalpic Process for Various Diameters of Capillary Tube For 5mm Diameter

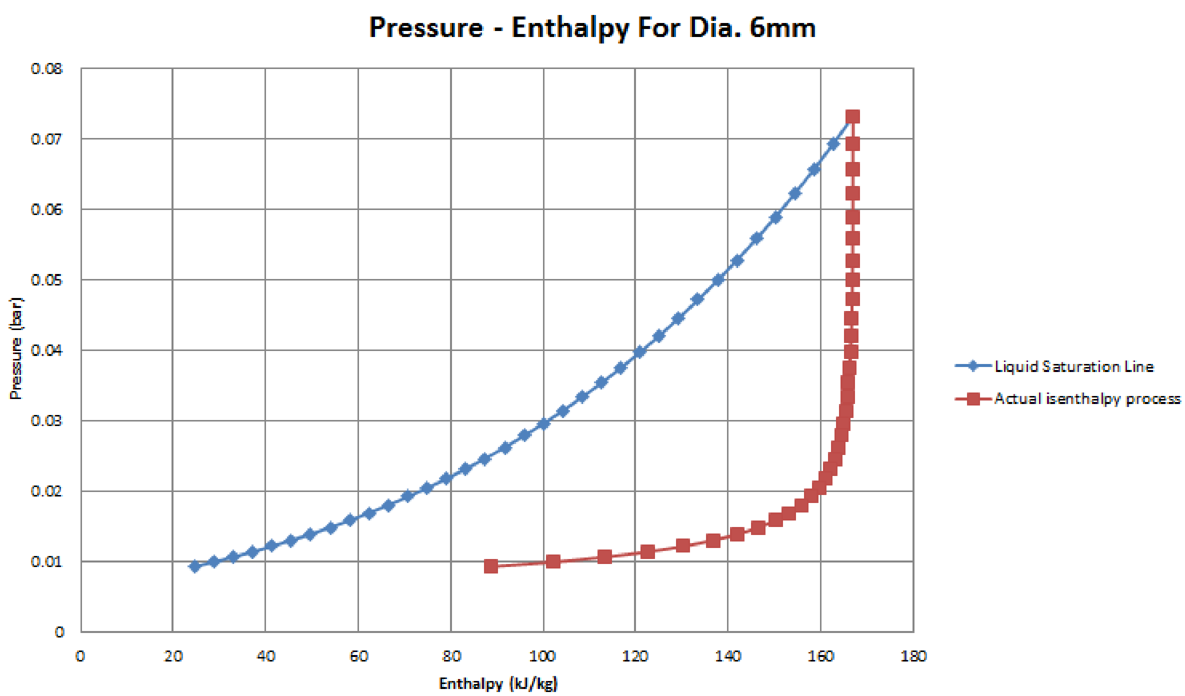

Figure 4.14: Actual Isenthalpic Process for Various Diameters of Capillary Tube For 6mm Diameter

Figure 4.13 and 4.14shows how the enthalpy changes in actual expansion process in capillary tube of various diameters. according to calculation length required for 5mm diameter capillary tube comes around 0.45m and for 6mm diameter capillary tube it comes around 1.85m. For methodology and detail calculations please refer [16] and Excel sheet in CD.

4.6.3 Cooling Water Piping

Cooling water flows at atmospheric pressure and temperature so this piping can be done with the help of PVC pipes and flexible plastic tubes. Nearly five meter long 1" pipe was found in the scrap which was used for this piping. We purchased some fittings and valves required to complete the piping. Cooling water is required at three places in system. (1) Absorber : To maintain the absorber temperature up to 40^oC. The required mass flow rate of cooling water, in absorber, according to calculation is $767.389\,kg/hr$ (2) Condenser : To take out the latent heat from the refrigerant vapor and condense it. The required mass flow rate of cooling water in condenser according to calculation is $605.70\,kg/hr$. (3) Evaporator : This cooling water will further be cooled and will be turned in to chilled water. The required mass flow rate of cooling water in evaporator, according to calculation is $120.47\,kg/hr$. This way the total mass flow rate of the cooling water supply is $1493\,kg/hr = 0.4149\,kg/s$. If we consider the density of cooling water at atmospheric pressure and temperature to be $1000\,kg/m^3$and the maximum velocity of the water in side pipe to be $2\,m/s$, then the required cross section area of the pipe is $0.0002\,m^2$and the diameter of the pipe is 16.25mm. The 1/2" pipe has the diameter of 15mm. so, the 1" diameter pipe will be most suitable for cooling water main pipe. Further detail arrangement can be found in the CAD-3D model of the system and schematic diagram in the Appendix.

4.7 Bill of Material.

Following table 4.3are the consolidated bill of material of the whole project. Apart from these materials there are some machining and fabrication processes for which we had to hire the services from out side the industry. This includes the machining of eccentric holes in steel blind flanges and its threading, Welding and brayzing, and some consumables like Teflon tape, Anabond Gasket Maker solution, m-seal thread sealent, etc.

Table 4.3: Consolidated Bill of Material

Sr.No.	Item	Qty.	Unit
1	1” Dia. PVC Coupling	3	Nos.
2	1” Dia. PVC Elbow	10	Nos.
3	1” Dia. PVC Pipe	6	m
4	1” Dia PVC Tee	5	Nos.
5	1” X 1/2” PVC Tee	4	Nos.
6	1 X 1/2” FTA	4	Nos.
7	1/2” Ball Valve	5	Nos.
8	1/2” Cap	10	Nos.
9	1/2” Gate Valve	2	Nos.
10	1/2” GI Coupling	8	Nos.
11	1/2” GI Elbow	20	Nos.
12	1/2” GI Pipe	12.5	m
13	1/2” GI Tee	26	Nos.
14	1/2” NRV	1	Nos.
15	1/2” Rubber Tube Connectors	20	Nos.
16	1/2” Rubber Tube Fasteners	34	Nos.
17	1/2” Threaded Couplings	13	Nos.
18	1/2” Transperant Rubber Tube	9.5	m
19	1/2” X 1/4” Rubber Tube Connectors	14	Nos.
20	1/2” X 1/4” Threaded Bushing	38	Nos.
21	1/4” Copper Coupling	57	Nos.
22	1/4” Copper Tubes (16 Swg.)	1.52	m
23	1/4” Copper Tubes (20 Swg.)	70	m
24	2” Dia. Steel Pipe	3	m
25	2” Dia. Threaded Coupling	6	Nos.
26	2” Dia. X 20mm Long Threaded Nipple	2	Nos.
27	2” X 3/4” Threaded Bushing	2	Nos.
28	3/4” Copper Coupling	2	Nos.
29	3/4” Copper Tube (16 Swg.)	0.240	m
30	3/4” Copper Tube (20 Swg.)	2.88	m
31	35X35X5mm Channel	9	m
32	4” Dia. Steel Pipe	12	m
33	4” Dia X 80mm Long Threaded Nipple	2	Nos.
34	4” Rubber Gasket	14	Nos.
35	4” Steel Blind Flange	10	Nos.
36	4” Steel Flange	14	Nos.

Sr.No.	Item	Qty.	Unit
37	4” Steel Threaded Couplings	4	Nos.
38	5/8” Nut	104	Nos.
39	5/8” X 4” Long Nut	104	Nos.
40	75X50mm Channel	12	m
41	8” Dia. HDPE Flange	3	Nos.
42	8” Dia. HDPE Pipe	4.1	m
43	8” Dia. HDPE Stub End	3	Nos.
44	8” Dia. HDPE Tee	1	Nos.
45	8” Dia. Rubber Gasket	3	Nos.
46	8” Dia. Steel Flange	2	Nos.
47	8” HDPE Blind Flange	1	Nos.
48	8” X 1/2” Tapping Saddle	3	Nos.
49	Capillary Tubes	2	Nos.
50	Pump	1	Nos.
51	Temperature Gauge	6	Nos.
52	Vacuum Gauge	2	Nos.
53	Gas Heater	1	Nos.

Chapter 5

Experimentation And Results

This chapter includes the operating procedure, experimentation methodology, expected results and actual results. Present status of the project is that we have been successful in achieving the designed vacuum in the system. But the vacuum in the system is difficult to maintain. Once we achieve the designed vacuum and close the valve that is connected to the vacuum pump in about an hour the vacuum is lost by 100mm of Hg. It is absolutely essential that this vacuum remains constant otherwise the system will not be able to perform. The reason for lose of vacuum is very minor leakages occurring in the threaded joints. All the welded joints and flange gasket joints are confirmed to be leak proof with the help of soap water and compressed air leakage test. The threaded joints of GI piping have been made with application of sufficient amount of Teflon and sealant, yet they are vulnerable to minor leakages.

5.1 Methodology of Experimentation.

Please refer the schematic diagram for system 3 given in the appendix A while reading the experimentation methodology. Once it is ensured that the system is achieving and maintaining the required vacuum, the system has to run with the water before we fill in the actual LiBr Solution.

5.1.1 Preparation of The System.

(1) The system shall be brought to the atmospheric pressure by keeping valve BV-1 or BV-3 open.

(2) The cap of absorber fill up point to be opened and water to be added in the absorber vessel. While water is being added it is essential that valve BV-2 and BV-4 are kept in open condition so that the water level can be seen in the level indicator LI-1. The valve GV-2 to be kept in close condition. The water will first fill up the pump chamber and absorber. Water level in indicator LI-1 should be allowed to rise so that all the seven copper pipes of

absorber are submerged by 5mm in it. But in no conditions the water level should rise far above these seven copper tubes and touch the evaporator pipe. It must be taken care of that the absorber vessel must not be filled completely, if this happens then the water will enter in to the evaporator through the 50 wide slot. which is not desirable because during actual fill up procedure in place of plain water, LiBr-water solution will be filled and it must not enter in evaporator.

(3) Once the mentioned level in absorber is achieved the pump is to be started. The pump will take the water from the pump chamber and absorber vessel and send it to the generator via heat exchanger and gas heater. The tube side of heat exchanger will be filled first and then the fluid will go to gas heater, and then it will enter the generator. The path of the water will pass through the nozzle N9-N10-N13-N17 in sequence.

(4) Once water has reached generator, it will flow down through the nozzles N15-N12-N11 with the help of gravity, and water level in level indicator LI-3 will not increase. Thus the water will reach up to the valve GV-2. Since valve GV-2 is in close condition so it will not allow water to pass. Thus first the pipe connecting the GV-2 and N11 will be filled and then the shell side of the heat exchanger shall be filled, then the pipe connecting N12 and N15 will be filled, and then the water will begin to fill the generator. Now water level can be seen in level indicator LI-3 increaseing. Water level in generator should not be more than 20mm from the bottom of the generator. Once this level is achieved the pump must be turned off. The NRV valve after the pump will not let water come back to the absorber vessel. The cap of absorber fill up point to be closed tightly with Teflon and sealent.

(5) Evaporator to be filled with water. The cap of evaporator fill up point to be opened and water to be added in the evaporator. The level of water in evaporator can be seen in the level indicator LI-2. Water level in the evaporator is to be kept only as much so that five copper pipes are completely submerged in water by 5 to 10mm. Evaporator water must not over flow and come out of 50mm wide slot to get in to absorber. Once the Evaporator is filled up the cap of evaporator fill up point to be closed tightly with Teflon and sealent.

(6) Once the system is filled with water and both the caps are closed with Teflon and sealant, we can create vacuum in the system. Now, the vacuum pump connection to be done with the ball valve BV-3. Before starting vacuum pump it must be ensure that the valves BV-2 and BV-4 are closed. If these valves are not closed and the vacuum pump is started, then the liquid from level indicator will get sucked in to the vacuum pump, which can damage the vacuum pump.

(7) Vacuum pump to be started and valve BV-3 to be kept in open condition. The pressure in generator vessel will begin to drop. Vacuum pump to be stoped when the vacuum gauge VG-2 indicates -705 mm of Hg pressure, which is equivalent to our generator pressure 0.7326 bar (54.94 mm of Hg absolute).

(8) Now the vacuum pump to be connected to the valve BV-1, which will be in open condition. The valve BV-2 will be in closed condition. The vacuum pump will be started and the vacuum gauge VG-1 will indicate the pressure inside the absorber vessel. Vacuum pump will be stoped when the vacuum gauge VG-1 indicates -754 mm of Hg pressure inside the absorber vessel, which is equivalent of 0.008725 bar (6mm of Hg absolute).

(9) Now the pump in pump chamber of absorber to be started and water to be allowed to flow in LiBr solution piping. The water may not be able to flow through the capillary tube CP-1 because it has been designed for the LiBr solution. If this happens then the valve GV-2 is to be opened partially so that some water flow can take place. This valve will remain closed in all other conditions while capillary tube is working. The mass flow rate of the fluid in this piping can be controlled with the help of valve GV-1.

(10) Once the continuous flow is established in the system we can light up the gas heater. The difference in reading of temperature gauge TG-2 and TG-6 will indicate the amount of heat that is being added to the fluid flowing inside the pipe, having known the density and specific heat of water at that pressure. Temperature gauge TG-1 will indicate the temperature of the fluid in absorber vessel. The temperature difference indicated by temperature gauge TG-1 and TG-2 can tell us how much amount of heat the fluid has received from the hot fluid that is coming from the generator. The temperature difference between TG-6 and TG-3 can tell us how much amount of heat has been lost in the generator. The temperature difference between TG-3 and TG-4 can tell us how much amount of heat the hot fluid has given to the cold fluid and how much it has lost to atmosphere through the steel pipe wall. TG-4 gives us the temperature at which the fluid is entering in the absorber. The temperature difference between TG-4 and TG-1 gives us amount of heat that has been lost to the cooling water flowing in side the absorber through seven copper pipes. TG-5 indicates the temperature of the chilled water that is coming out of the evaporator. The atmospheric temperature of the cooling water inlet will be known to us. The difference between the temperature of TG-5 and cooling water inlet temperature will tell us how much cooling effect has been achieved, assuming the mass flow rate of chilled water is measured and known. (Of-course, While we are useing water in place of LiBr solution the system will not produce cooling effect.)

Now, if the system is working properly with the water then the water can be drained and replaced by LiBr Solution and above steps to be repeated. Only in evaporator the water will be filled, in all other parts the LiBr-Solution.

5.1.2 Expected and Achieved Results.

According to the calculations done previously we have all the temperatures for various points. Once the system achieves the steady state all the temperature indicators and vacuum gauges should achieve the following readings, shown in table bellow.

Table 5.1: Expected Vs. Achieved Results

Sr.No.	Instrument	Expected Reading	Achieved Reading
1	Vacuum Gauge VG-1	-754 mm of Hg	-754 mm of Hg
2	Vacuum Gauge VG-2	-705 mm of Hg	-705 mm of Hg
3	Temperature Gauge TG-1	$40^{o}C$	Awaited
4	Temperature Gauge TG-2	$76.19^{o}C$	Awaited
5	Temperature Gauge TG-3	$95^{o}C$	Awaited
6	Temperature Gauge TG-4	$52^{o}C$	Awaited
7	Temperature Gauge TG-5	$10^{o}C$	Awaited
8	Temperature Gauge TG-6	$95^{0}C$	Awaited

Expected temperatures are shown in the table above are also shown in the figure of process diagram and the schematic diagram in the appendix. Apart from these temperatures the consumption of fuel and the mass flow rate of the chilled water is also essential to determine the performance of the system. Once these readings are achieved then the other parameters can be determined from calculations in previous chapters.

5.2 Comparison of Solar Based VAR with VCR.

In gulf countries and desert areas where temperature is quite high throughout the year, air conditioning system is required to be operated for 24 hours and 365 days. Keeping this application in mind a solar based VAR system and VCR system of 1TR are compared in this section.

Assuming VCR system's COP is 3 and refrigeration capacity is 3.5 kW, 1.166 kW power is consumed. If this system is used for 24 hours and 365 days, considering 6 Rs/unit as rate of electricity consumption, operating cost per year is approximately Rs. 61,320/-. for two years the operating cost will be Rs. 1,22,640/- and initial cost of 1TR VCR system is approximately Rs. 25,000/- therefore cost for 2 years is 1,47,640/-.

Initial cost for solar operated VAR system prototype is 1.5 to 1.75 lakh. If the system is optimized and developed further then cost definitely will reduce. Operating cost of solar based vapor absorption system is nearly zero as solar energy is freely available. Thus we can say that in 2 to 2.5 years time total cost of solar operated VAR system is same as VCR system. After 2.5 years time solar based VAR system becomes economical and more than that it conserves valuable high grade energy.

Chapter 6

Conclusion and Scope For Future Work

6.1 Conclusion & Current Status of Project Work

Design of LiBr-water vapor absorption refrigeration system has been carried out for three systems (1) Fully solar operated system. (2) Solar and fuel operated system. (3) Fully fuel operated system. Summary of the results obtained at the end of design is given in table 6.1.

Table 6.1: Summary of models

Feature	Model 1	Model 2	Model 3
Energy Source	Fully solar	Solar & Fuel	Fuel
Heat Input	15.67 kW	4.21 kW	4.21 kW
Theoretical COP	0.67	0.83	0.83
Fuel Consumption	0.000	2 kg/day H_2	3 kg/day H_2
Approx Fuel cost	0.00 Rs/day	400 Rs/day	600 Rs/day
Solar Heat collector area	35 sq.m.	10 sq.m.	0.00 sq.m.
Approx. cost	1.5 to 2 Lakh	1 to 1.25 Lakh	0.75 to 1 Lakh

It can be concluded from the table 6.1 that COP of hybrid system and fuel based system is 23% higher than solar based system. Also initial cost of solar energy operated system is quite huge compared to hybrid system and gas based system. However running cost of the solar energy based system is very low, while running cost of hybrid and fuel based system is very high with compare to VCR system. All these three systems has much higher capital cost with compare to VCR system. Comparing VCR system and solar based VAR system for 24 hour and 365 days operation based on simple payback period method, it can be concluded that by 2.5 years of time total cost of VCR system and solar operated VAR system is almost same. Thus the simple payback period of solar operated VAR system is 2.5 years and by using this system it is possible to conserve lot of conventional energy sources.

Assembly of fuel based vapor absorption system was done and required vacuum was generated in the system. However, system can not sustain the vacuum for more than 5 to 6 hours because of very minor leakages which requires special attention and remedial actions. Once this problem is rectified, the main objective of the project work to develop prototype of 1TR vapor absorption refrigeration system can be fulfilled.

6.2 Scope for Future Work

Once the system is ensured to sustain the vacuum the experimentation can be carried out. Achieved readings are to be compared with the calculated ones and determined the system performance. Once fuel based VAR system is developed and successfully and tested for its functionality, solar operated system can be developed.

Optimization of size and the performance of the system shall be the scope of future work.

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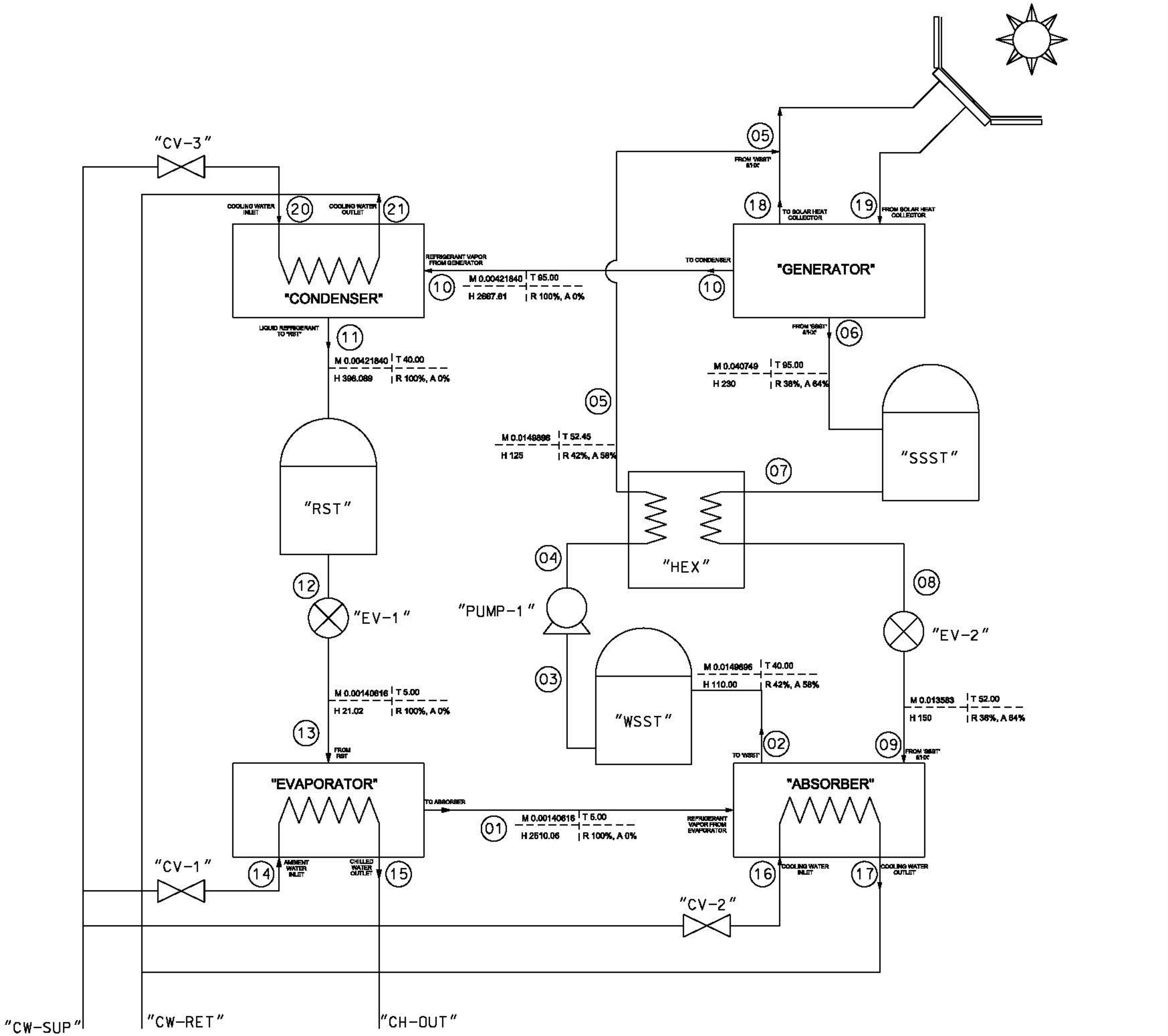
[23] Z.F. Li, K. Sumathy, “Technology Development In The Solar Absorption Air-conditioning Systems”, Renewable and Sustainable Energy Reviews 4 (2000) 267-293

[24] Said, S.A.M., El-Shaarawi, M.A.I., Siddiqui, M.U., “Alternate designs for a 24- hour operating solar powered absorption refrigeration technology”, International Journal of Refrigeration (2012).

Appendix A

Schematic Diagrams

Schematic Diagram 1 :	SVAR-M1-PI1	Schematic Diagram for Solar operated VAR System
Schematic Diagram 2 :	SVAR-M2-PI2	Schematic Diagram for Solar and Fuel operated VAR System
Schematic Diagram 3 :	SVAR-M3-PI3	Schematic Diagram for Fuel operated VAR System



01	"EVA"	EVAPORATOR
02	"ABS"	ABSORBER
03	"GEN"	GENERATOR
04	"CON"	CONDENCER
05	"HEX"	HEAT EXCHANGEER
06	"SSST"	STRONG SOLUTION STORAGE TANK
07	"WSST"	WEAK SOLUTION STORAGE TANK
08	"RST"	REFRIGERATION STORAGE TANK
09	"PUMP-1"	PUMP
10	"EV-1"	EXPANSION VALVE 1
11	"EV-2"	EXPANSION VALVE 2
12	"CV-1"	CONTROL VALVE 1
13	"CV-2"	CONTROL VALVE 2
14	"CV-3"	CONTROL VALVE 3

LEGEND:

MASS FLOW RATE M (Kg/s)	TEMPERATURE T (Deg C)
ENTHALPY H (KJ/Kg)	CONCENTRATION REFRIGERANT R (%) ABSORBER A (%)

PROJECT :-
SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

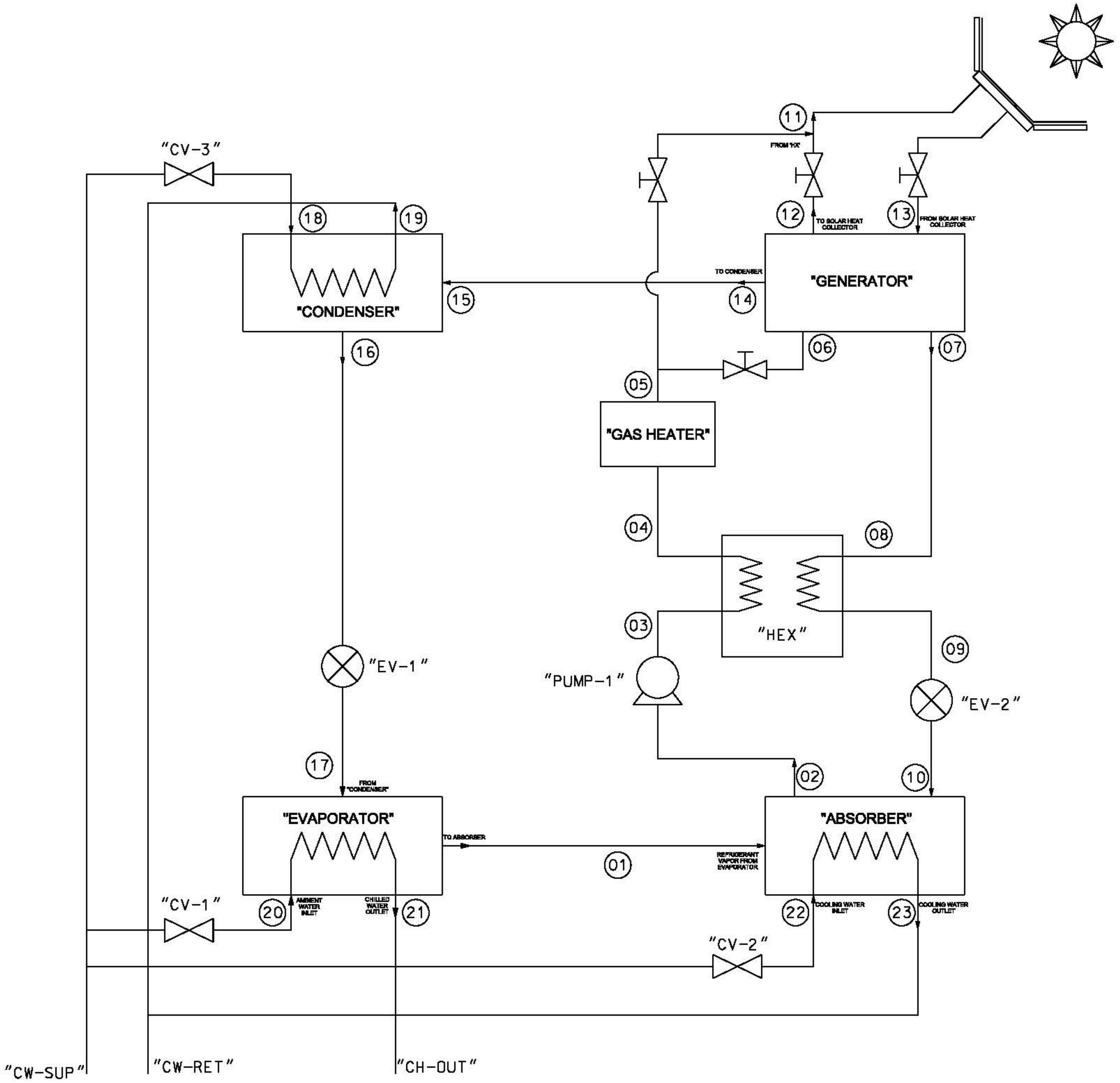
STUDENT :-
ABHIJIT. S. MUSALE.
(Roll No : 11MMET08)

COLLAGE :-
INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY
(Guide : Prof. N.K.SHAH)

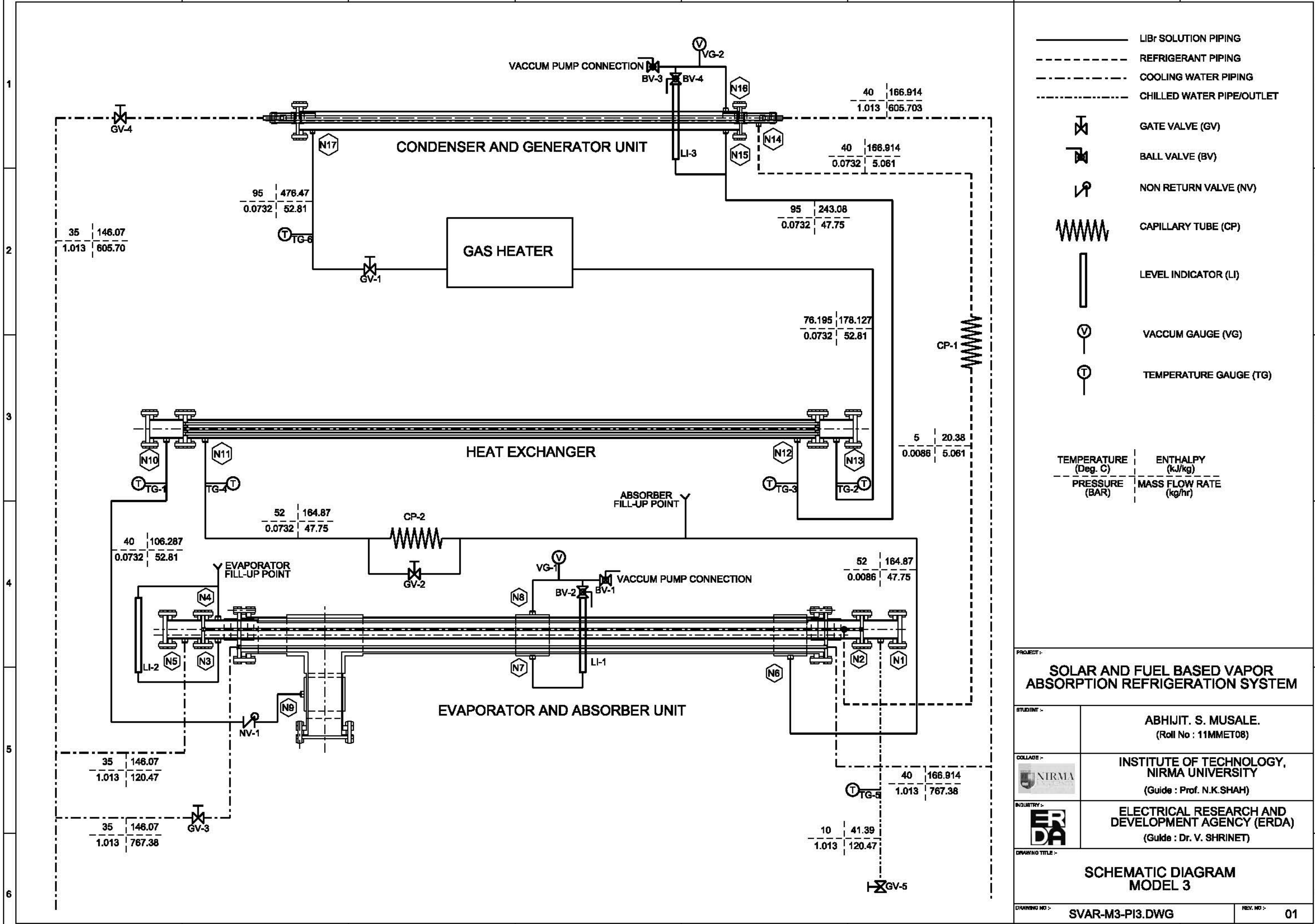
INDUSTRY :-
ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)
(Guide : Dr. V. SHRINET)

DRAWING TITLE :-
SCHEMATIC DIAGRAM
MODEL 1

DRAWING NO :- SVAR-M1-PI1.DWG
REV. NO :- 1



PROJECT :-	SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :- NIRMA UNIVERSITY	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :- ERDA	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :-	SCHEMATIC DIAGRAM MODEL 2
DRAWING NO :- SVAR-M2-PI2.DWG	REV. NO :- 1



PROJECT :-

SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-

ABHIJIT. S. MUSALE.
(Roll No : 11MMET08)

COLLAGE :-

INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY
(Guide : Prof. N.K.SHAH)

INDUSTRY :-

ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)
(Guide : Dr. V. SHRINET)

DRAWING TITLE :-

SCHEMATIC DIAGRAM MODEL 3

DRAWING NO :- SVAR-M3-PI3.DWG

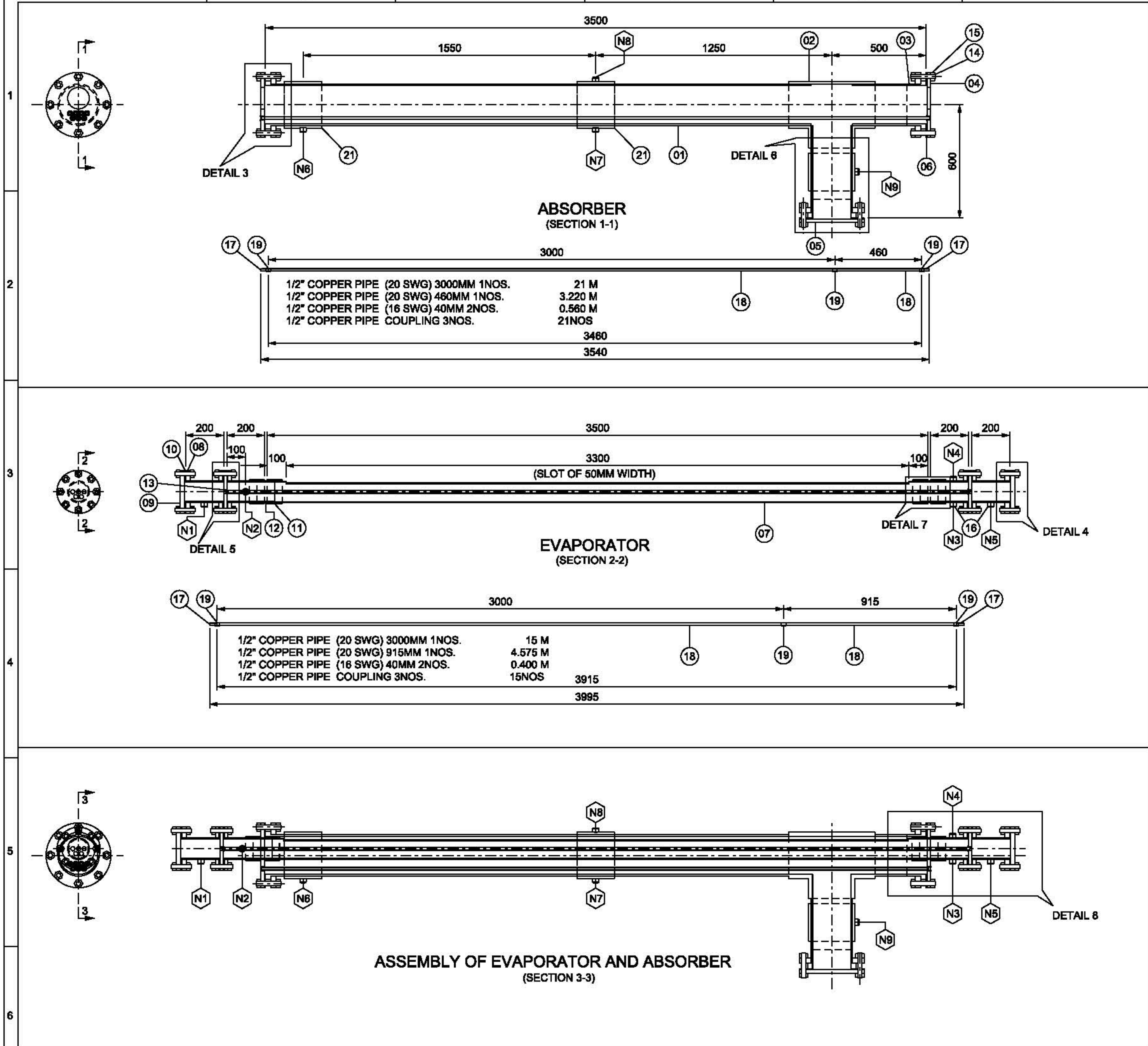
REV. NO :- 01

Appendix B

Detail Drawings

Assembly Drawing 1	Assembly Drawing for Absorber and Evaporator
Assembly Drawing 2	Assembly Drawing for Generator and Condenser

Along with these two Assembly drawings 12 separate detail drawings have been attached for reference.



LIST OF MATERAIL

NO.	ITEM	QTY.	UNIT
01	8" HDPE PIPE	4	M
02	8" HDPE TEE	1	Nos.
03	8" HDPE FLANGE	3	Nos.
04	8" MS BLIND FLANGE	2	Nos.
05	8" HDP BLIND FLANGE	1	Nos.
06	8" RUBBER GASKET	3	Nos.
07	4" MS PIPE	12	M
08	4" MS FLANGE	14	Nos.
09	4" MS BLIND FLANGE	10	Nos.
10	4" RUBBER GASKET	14	Nos.
11	4" MS COUPLING	4	Nos.
12	4" DIA. X 80MM LONG THREADED NIPPLE	2	Nos.
13	1/2" X 1/4" THREADED BUSHING	38	Nos.
14	5/8" DIA X 4" LONG BOLT	86	Nos.
15	5/8" DIA NUTS	86	Nos.
16	1/2" THREADED COUPLINGS	17	Nos.
17	1/4" COPPER TUBE (16 SWG)	3	M
18	1/4" COPPER TUBE (20 SWG)	67	M
19	1/4" COPPER COUPLING	57	Nos.
20	8" HDPE STUB END	3	Nos.
21	8" X 1/2" TAPPING SADDLE	3	Nos.
22	SUBMERSIBLE PUMP	1	Nos.
23	1/2" FLAXIBLE TUBE FOR PUMP OUTLET	1	Nos.
24	3/4" COPPER TUBE (16 SWG)	0.250	M
25	3/4" COPPER TUBE (20 SWG)	3	M
26	3/4" COPPER COUPLINGS	2	Nos.
27	2" MS PIPE	4	M
28	2" MS COUPLING	6	Nos.
29	2" X 3/4" THREADED BUSHING	2	Nos.
30	2" DIA. X 60MM LONG THREADED NIPPLE	2	Nos.

PROJECT :-

SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-

ABHIJIT. S. MUSALE.
(Roll No : 11MMET08)

COLLAGE :-

NIRMA

INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY
(Guide : Prof. N.K.SHAH)

INDUSTRY :-

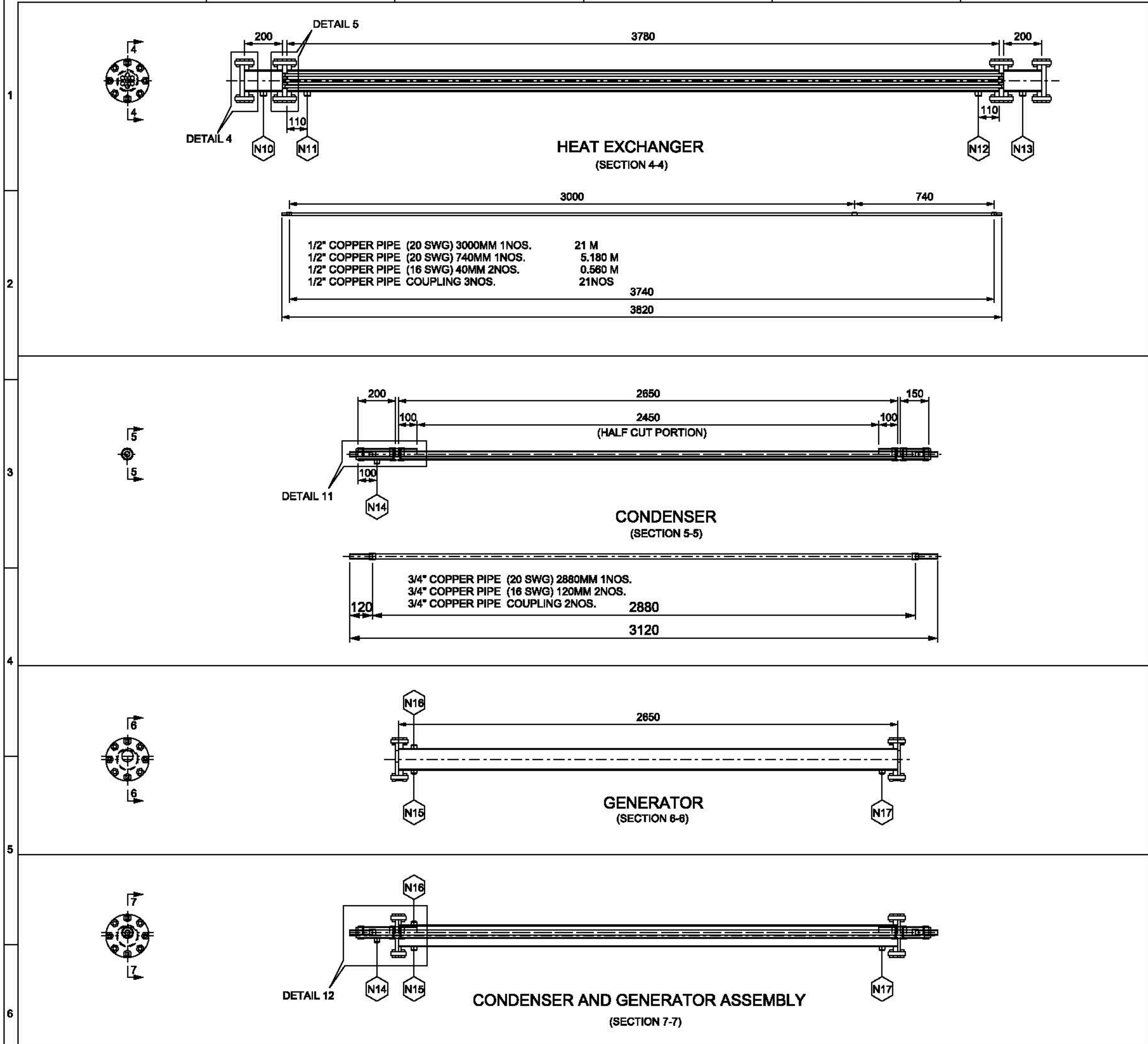
ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)
(Guide : Dr. V. SHRINET)

DRAWING TITLE :-

EVAPORATOR & ABSORBER ARRANGEMENT

DRAWING NO :- SVAR-M3-01.DWG

REV. NO :- 01



LIST OF NOZZLES

NO.	DISCRIPTION
N1	CHILLED WATER OUTLET
N2	REFRIGERENT INLET TO EVAPORATOR
N3	LEVEL INDICATOR FOR EVAPORATOR
N4	LEVEL INDICATOR & EVAPORATOR FILL UP
N5	COOLING WATER INLET
N6	HOT STRONG SOLUTION INLET
N7	LEVEL INDICATOR FOR ABSORBER
N8	LEVEL INDICATOR & VACCUM PUMP CONNECTION
N9	PUMP OUTLET
N10	WEAK LiBr SOLUTION FROM N9 / COLD SIDE INLET
N11	STRONG LiBr SOLUTION TO N6 / HOT SIDE OUTLET
N12	WEAK LiBr SOLUTION TO N17 / COLD SIDE OUTLET
N13	STRONG LiBr SOLUTION FROM N15 / HOT SIDE INLET
N14	CONDENSED REFRIGERENT OUTLET / TO N2
N15	STRONG LiBr SOLUTION OUTLET / LEVEL INDI. /TO N13
N16	LEVEL INDICATOR / VACCUM PUMP CONNECTION VALVE
N17	WEAK LiBr SOLUTION FROM GAS HEATER / INLET

PROJECT :-
SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-
ABHIJIT. S. MUSALE.
(Roll No : 11MMET08)

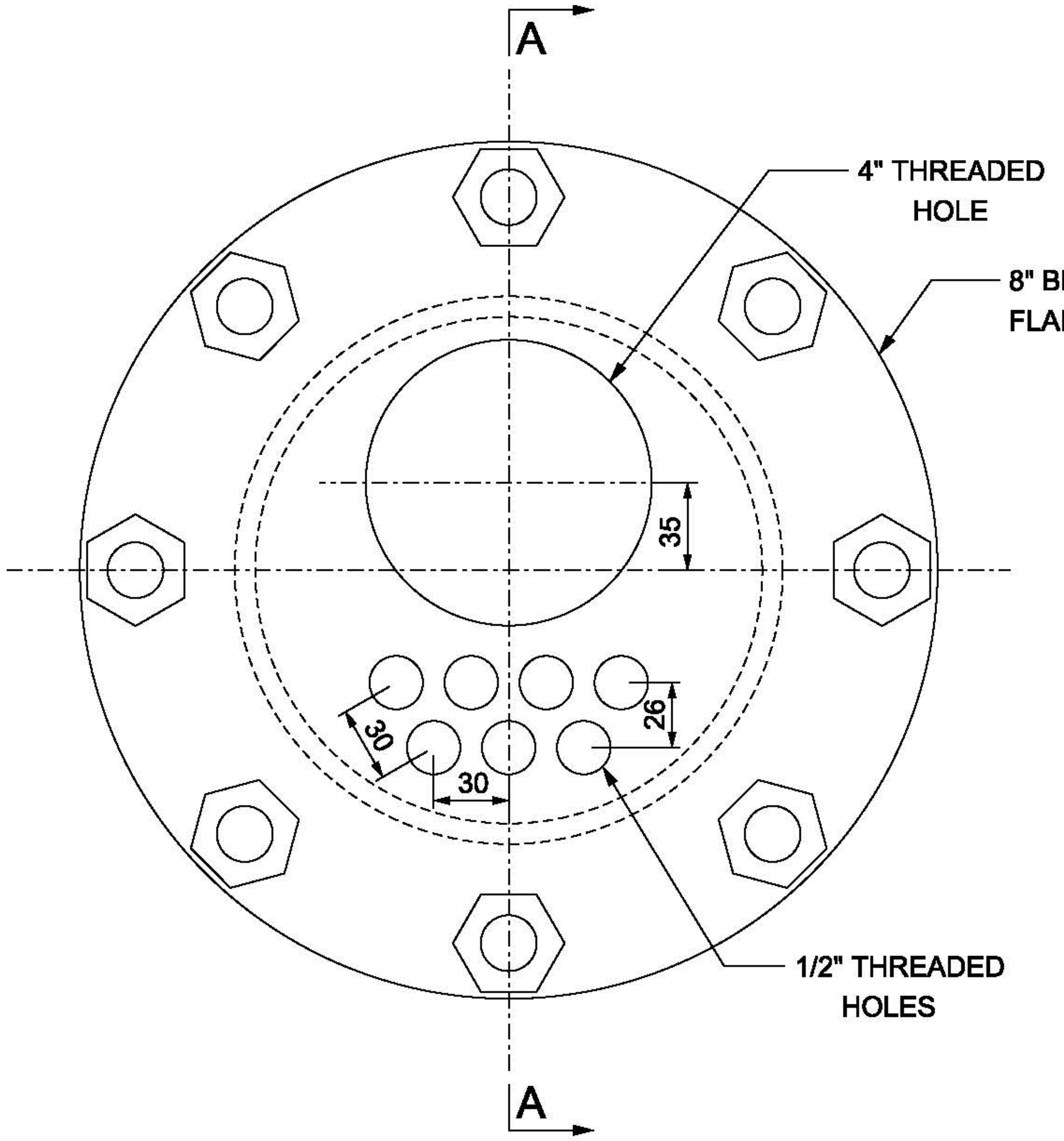
COLLAGE :-
INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY
(Guide : Prof. N.K.SHAH)

INDUSTRY :-
ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)
(Guide : Dr. V. SHRINET)

DRAWING TITLE :-
HEAT EXCHANGER, CONDENSER AND GENERATOR ARRANGEMENT

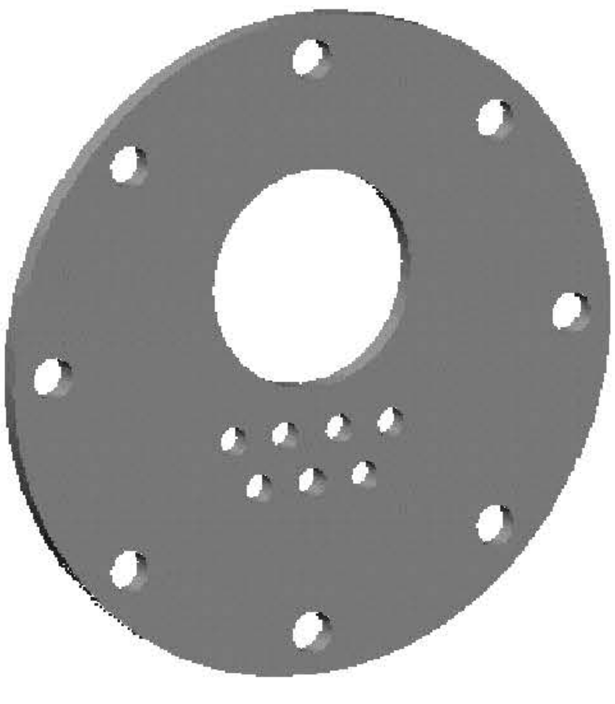
DRAWING NO :- SVAR-M3-02.DWG

REV. NO :- 01

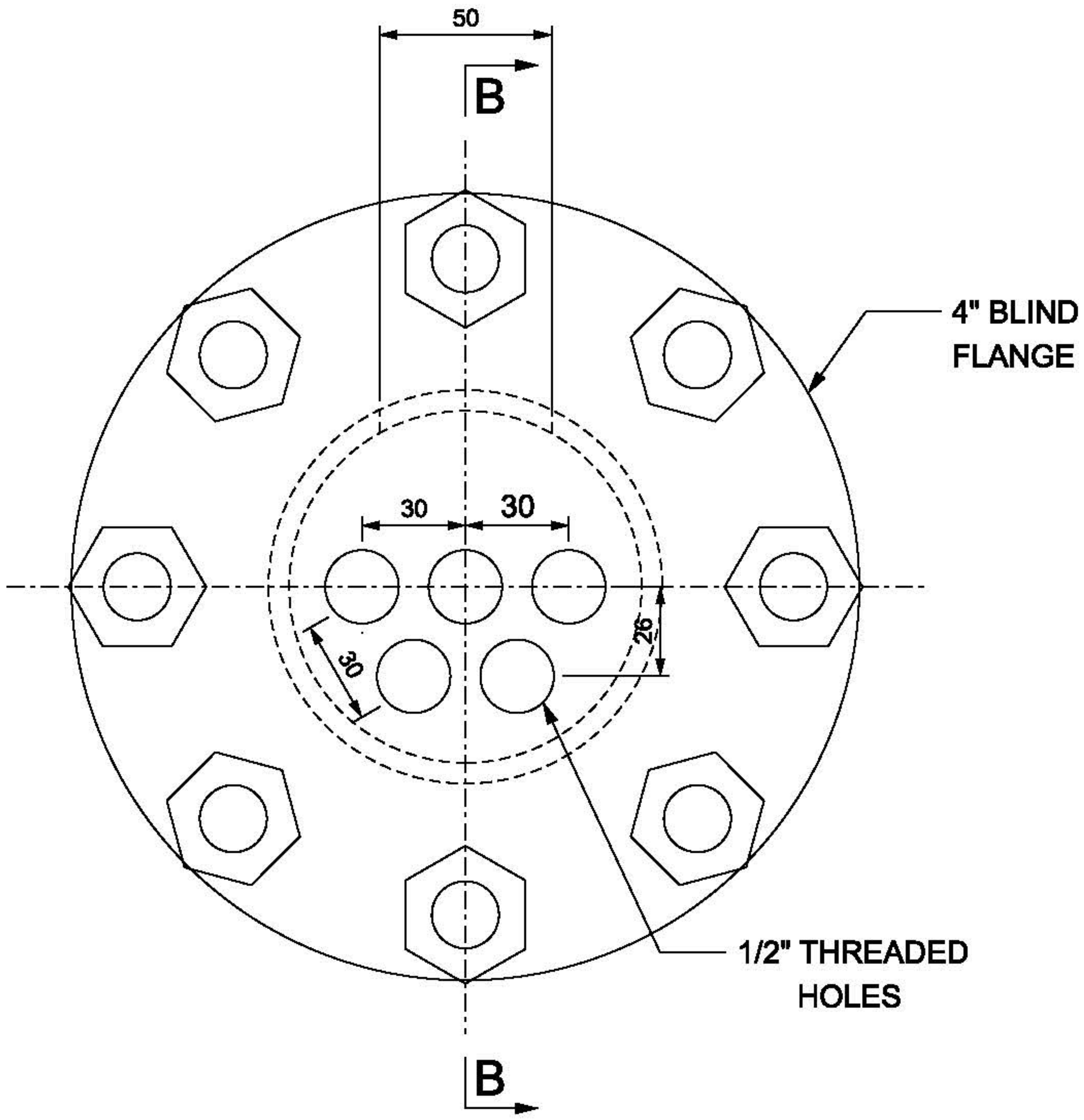


DETAIL 1

(8" BLIND FLANGE AS TUBE SHEET DETAILS)

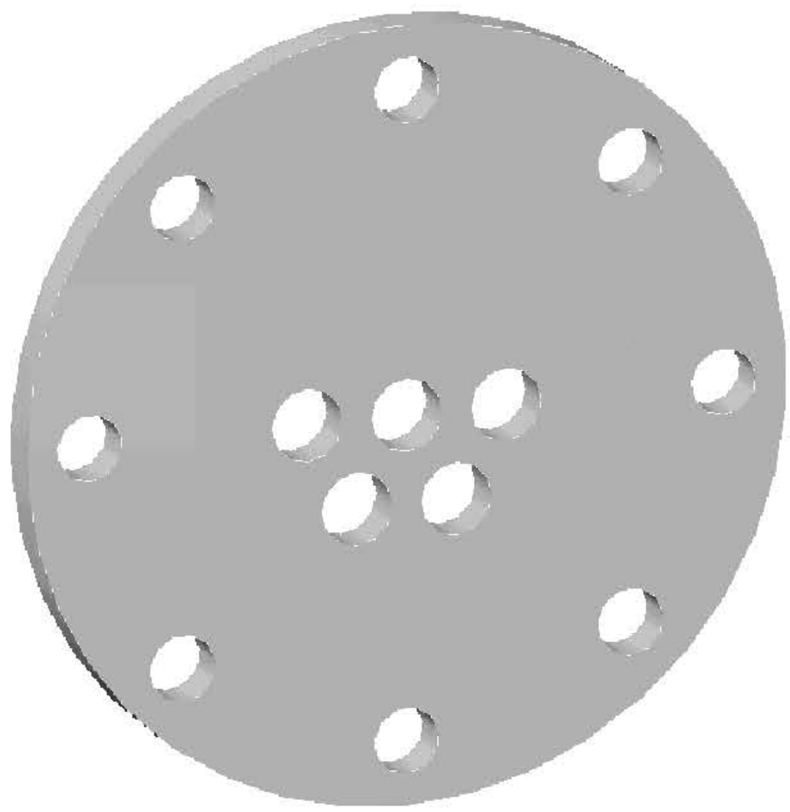


PROJECT :-	SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :-	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :-	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :-	EVAPORATOR & ABSORBER ARRANGEMENT
DRAWING NO :-	SVAR-M3-DT01.DWG
REV. NO :-	01

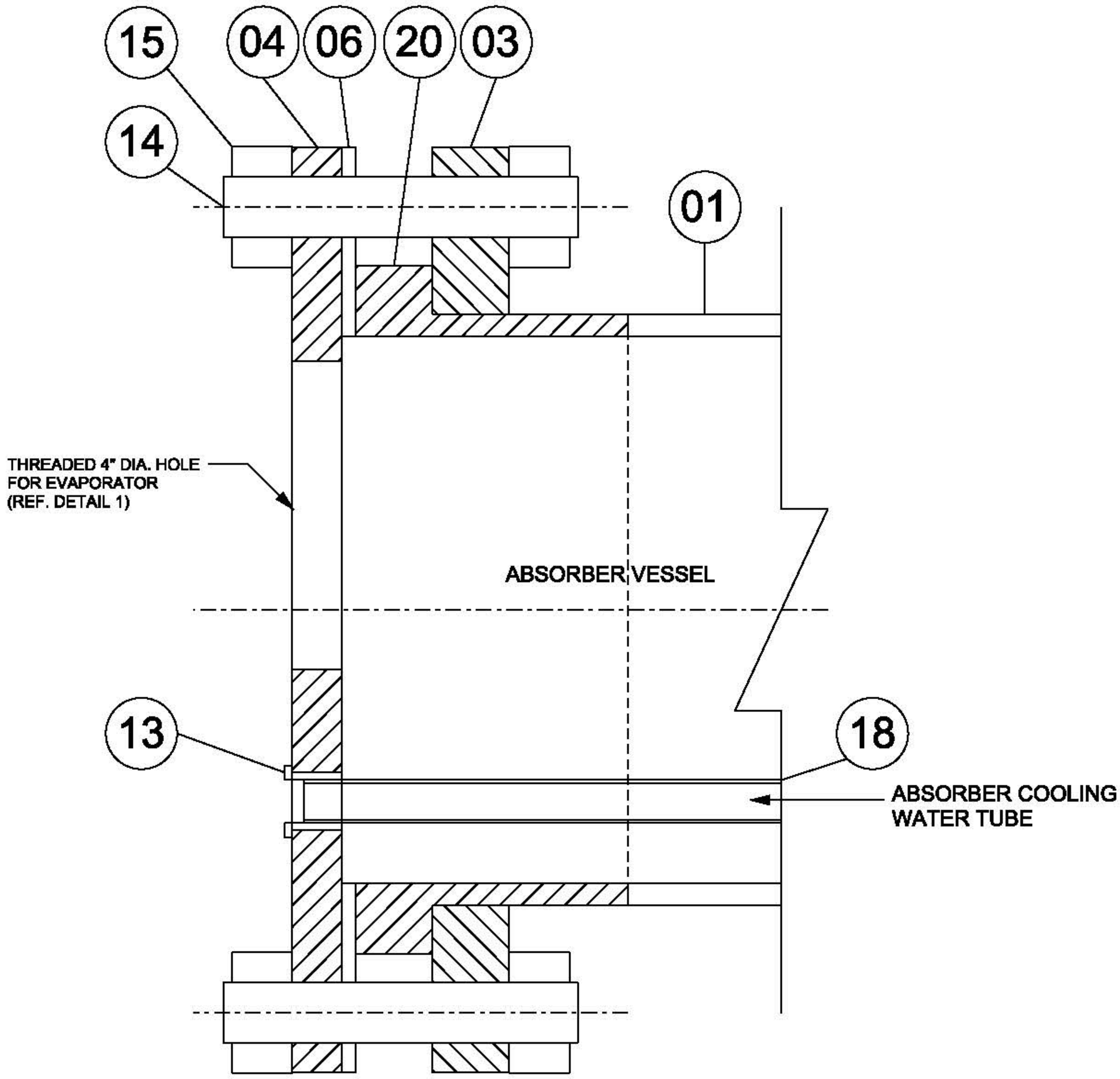


DETAIL 2

(4" BLIND FLANGE AS TUBE SHEET DETAILS)



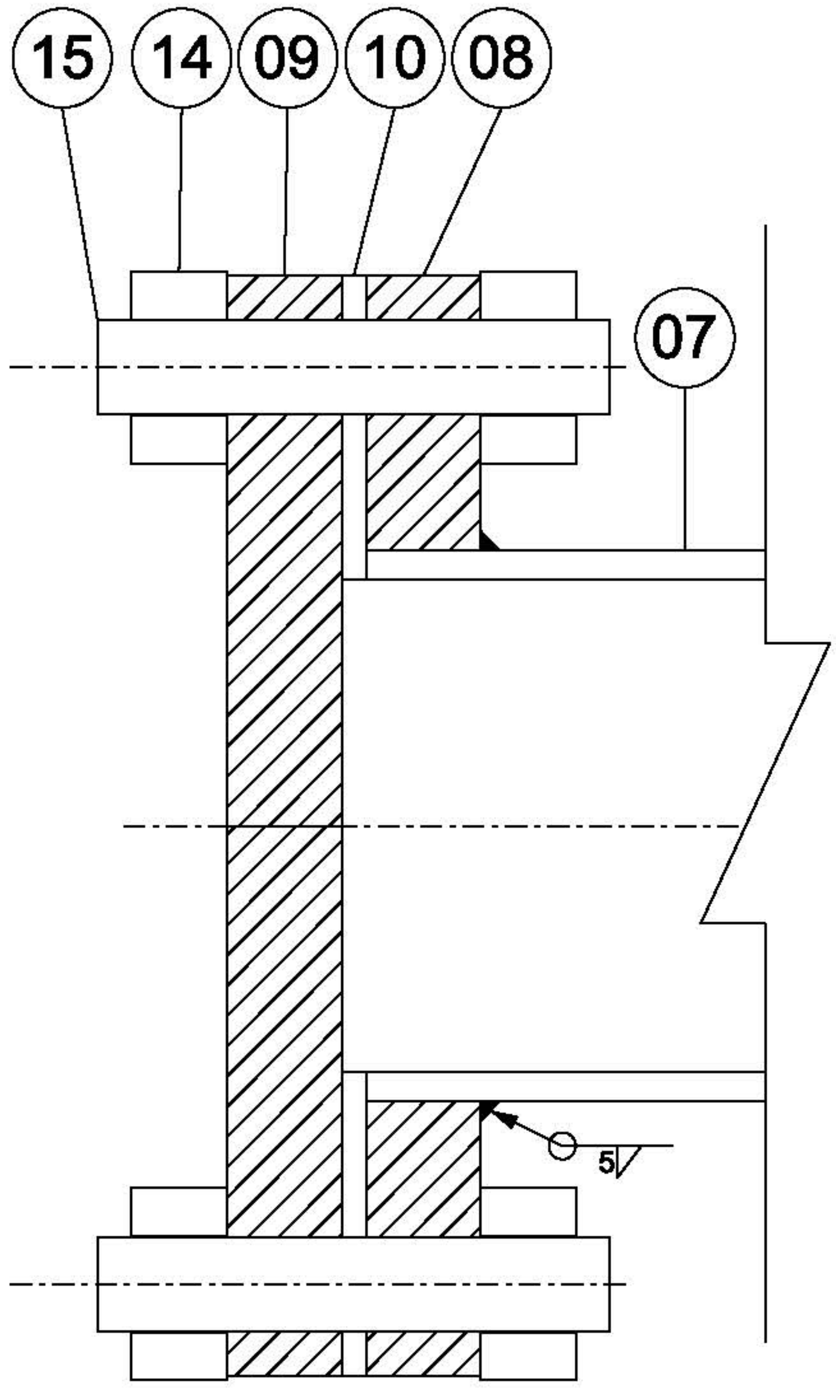
PROJECT :-	SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :- NIRMA UNIVERSITY	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :- ERDA	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :-	EVAPORATOR & ABSORBER ARRANGEMENT
DRAWING NO :- SVAR-M3-DT02.DWG	REV. NO :- 01



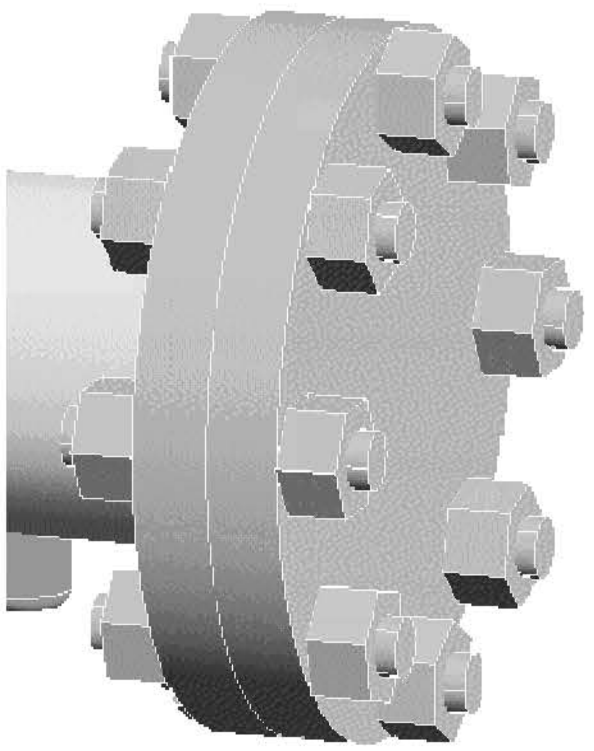
DETAIL 3

(SECTION A-A)

PROJECT :- SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM	
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :- NIRMA UNIVERSITY	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :- ERDA	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :- EVAPORATOR & ABSORBER ARRANGEMENT	
DRAWING NO :- SVAR-M3-DT03.DWG	REV. NO :- 01



DETAIL 4



PROJECT :-

SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-

ABHIJIT. S. MUSALE.

(Roll No : 11MMET08)

COLLAGE :-

NIRMA UNIVERSITY

INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY

(Guide : Prof. N.K.SHAH)

INDUSTRY :-

ERDA

ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)

(Guide : Dr. V. SHRINET)

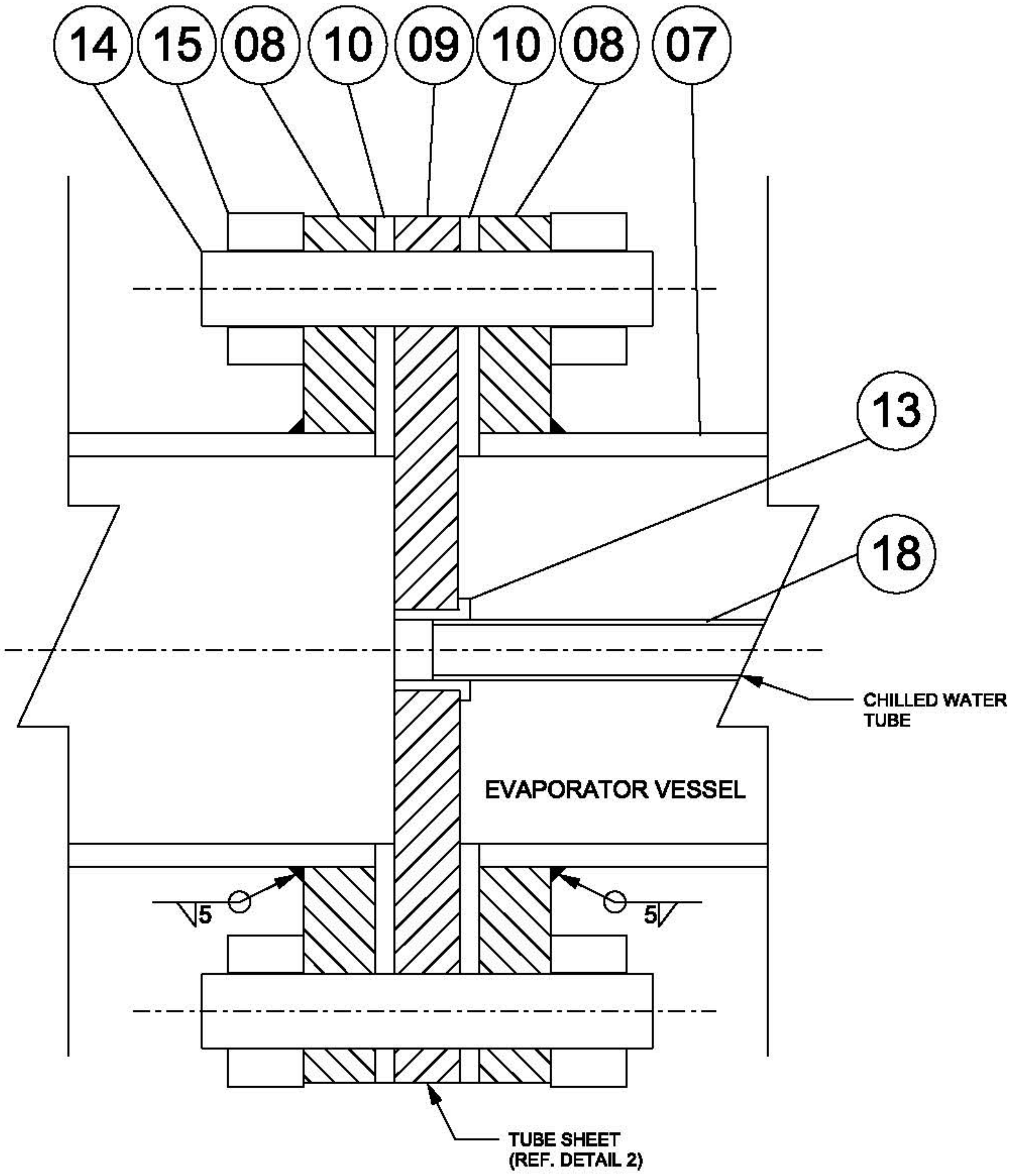
DRAWING TITLE :-

EVAPORATOR & ABSORBER ARRANGEMENT

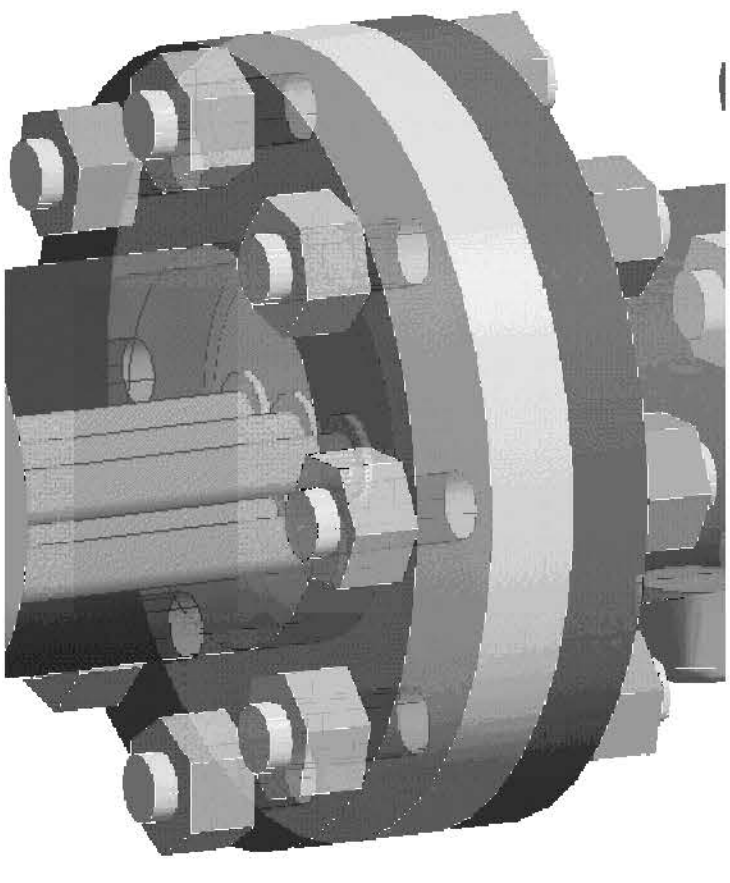
DRAWING NO :- SVAR-M3-DT04.DWG

REV. NO :- 01

E F G H

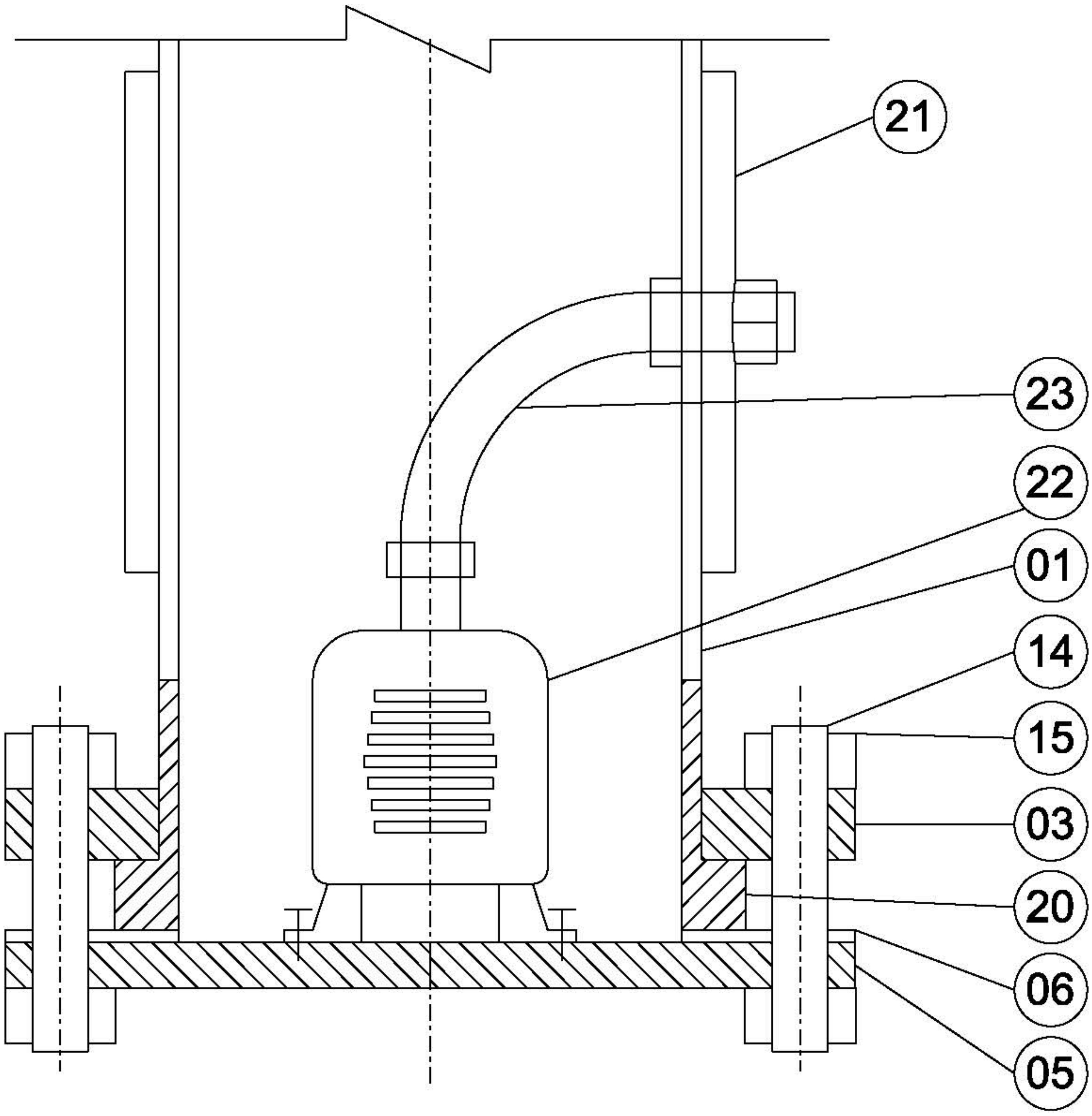


DETAIL 5
(SECTION B-B)



PROJECT :-	SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :- NIRMA UNIVERSITY	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :- ERDA	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :-	EVAPORATOR & ABSORBER ARRANGEMENT
DRAWING NO :- SVAR-M3-DT05.DWG	REV. NO :- 01

E F G H



DETAIL 6

PROJECT :-

SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-

ABHIJIT. S. MUSALE.
(Roll No : 11MMET08)

COLLAGE :-

NIRMA UNIVERSITY

INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY
(Guide : Prof. N.K.SHAH)

INDUSTRY :-

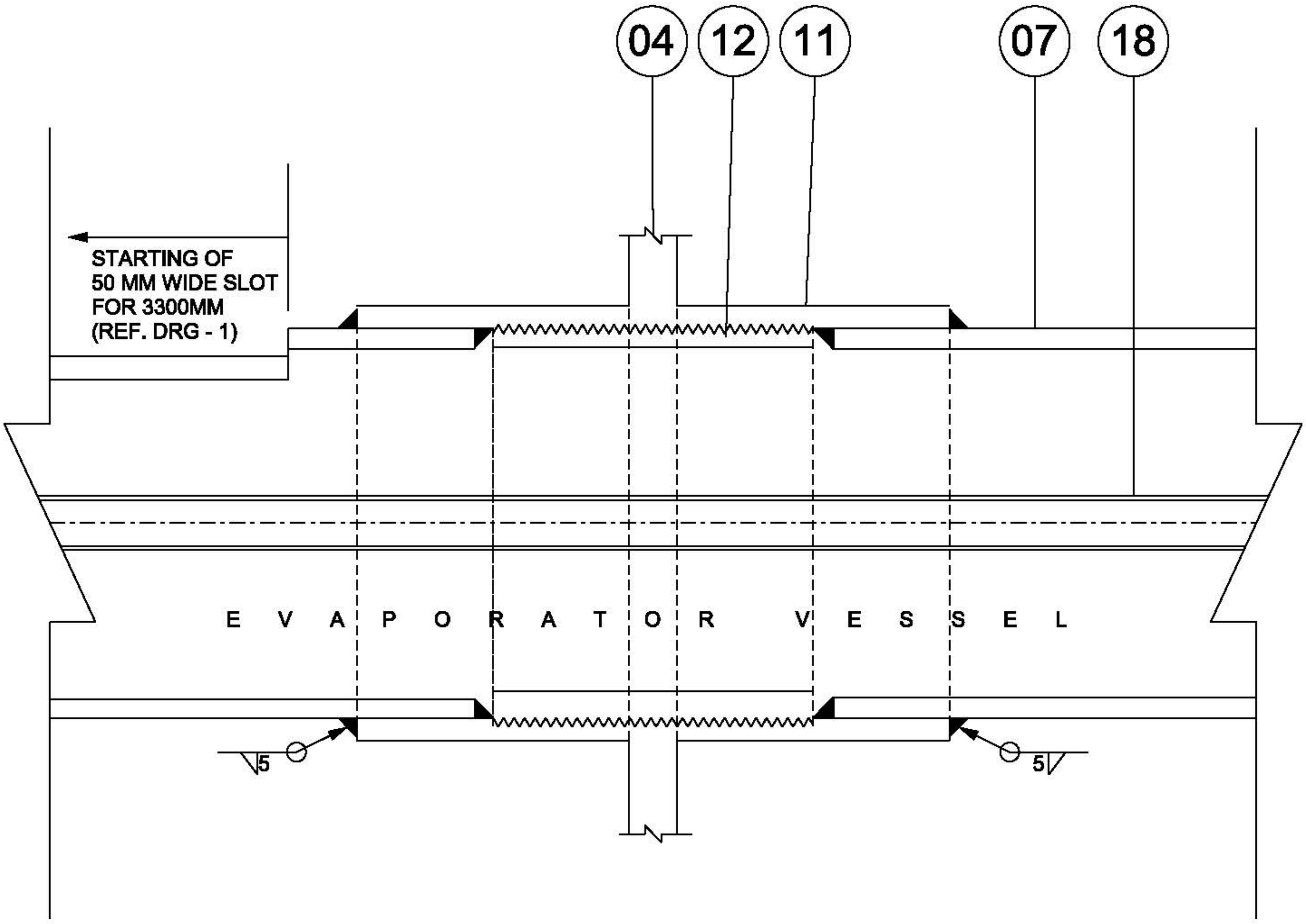
ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)
(Guide : Dr. V. SHRINET)

DRAWING TITLE :-

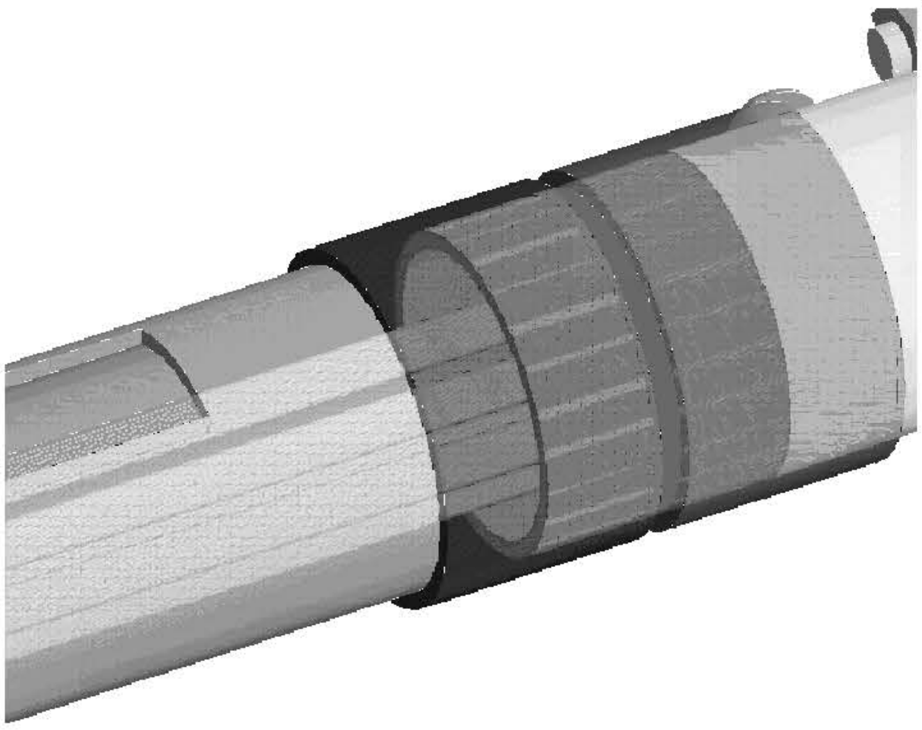
EVAPORATOR & ABSORBER ARRANGEMENT

DRAWING NO :- SVAR-M3-DT06.DWG

REV. NO :- 01



DETAIL 7



PROJECT :-

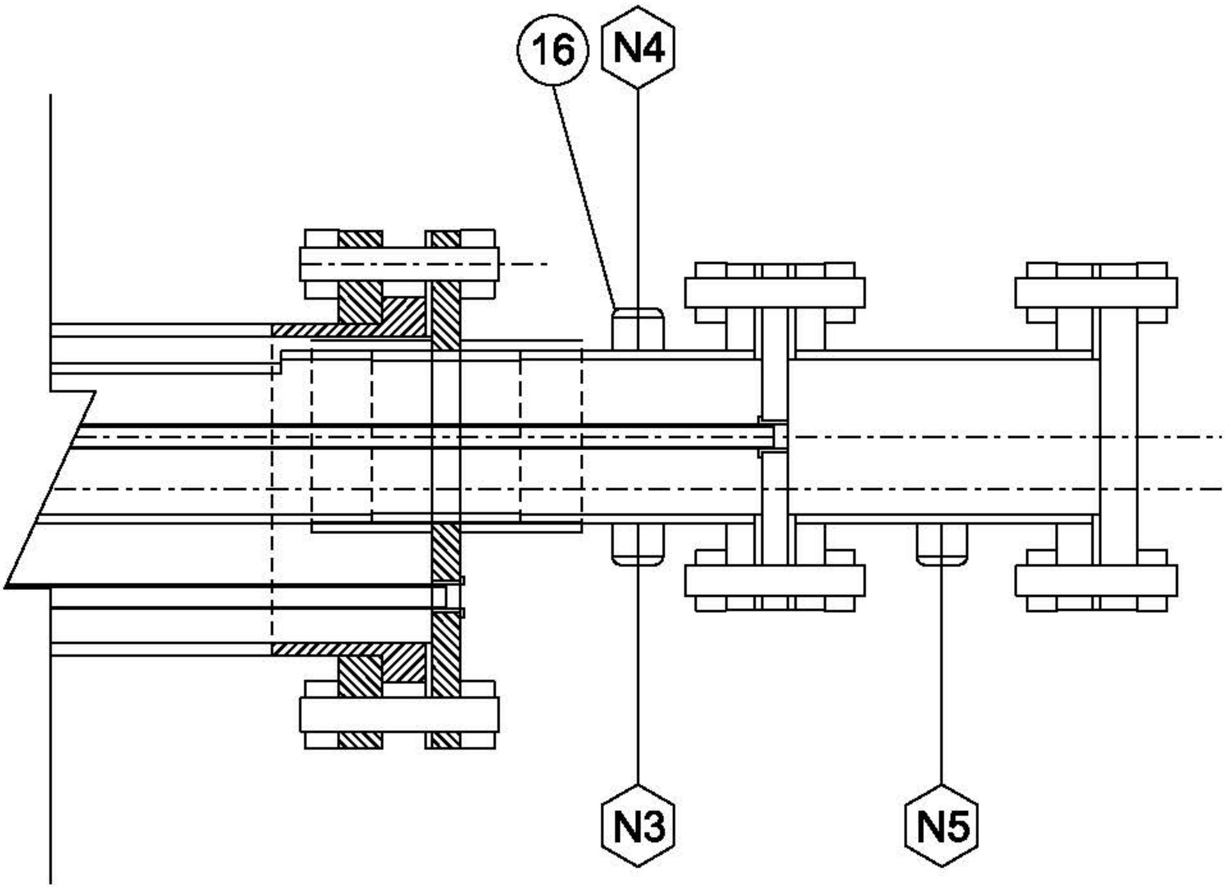
SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :- NIRMA UNIVERSITY	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :- ERDA	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)

DRAWING TITLE :-

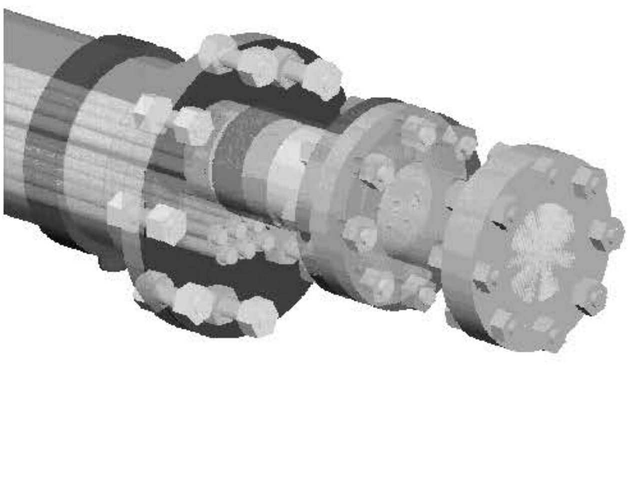
EVAPORATOR & ABSORBER ARRANGEMENT

DRAWING NO :-	SVAR-M3-DT07.DWG	REV. NO :-	01

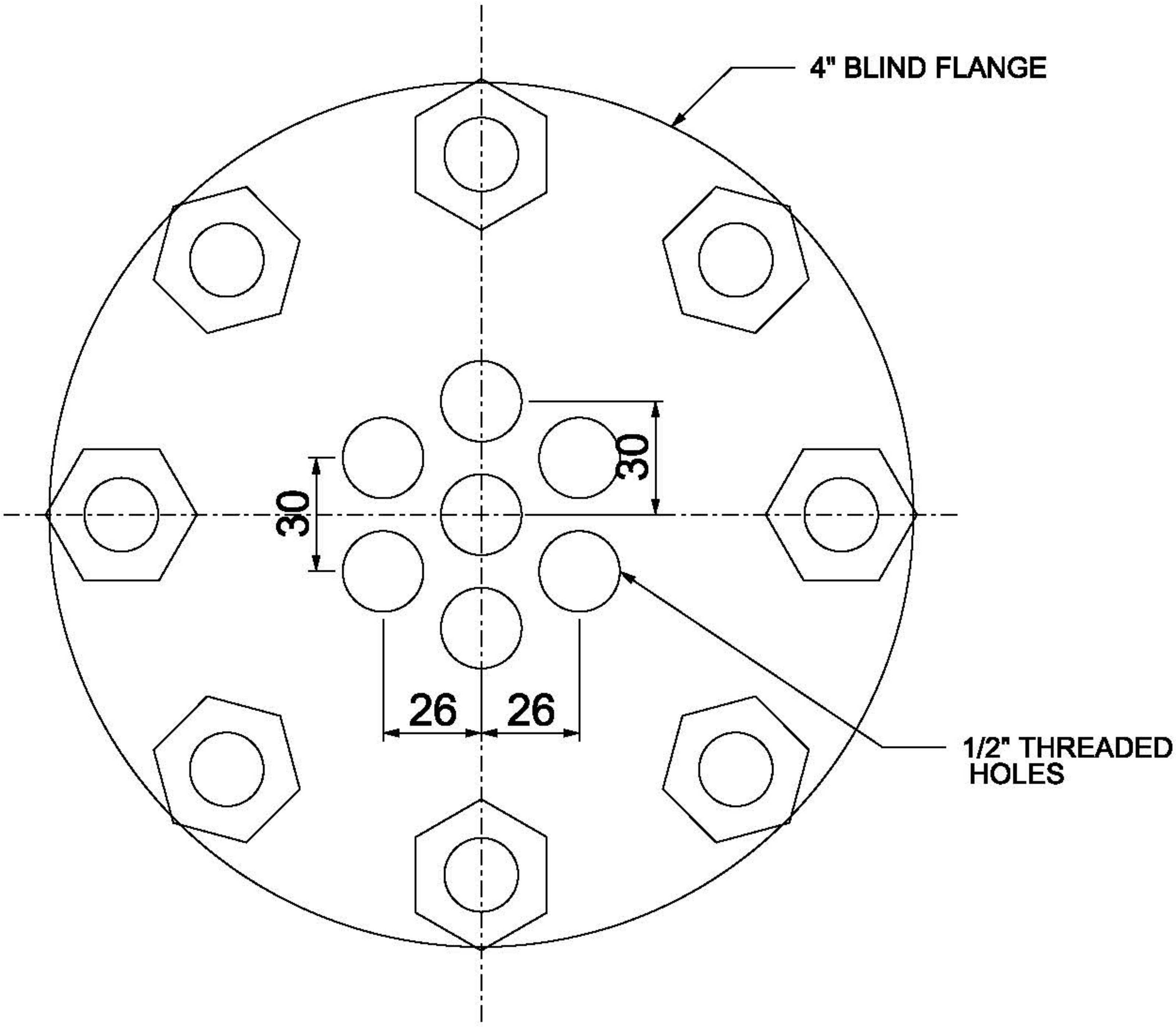


DETAIL 8

(ASSEMBLED VIEW OF DETAIL- 3,4,5,7)

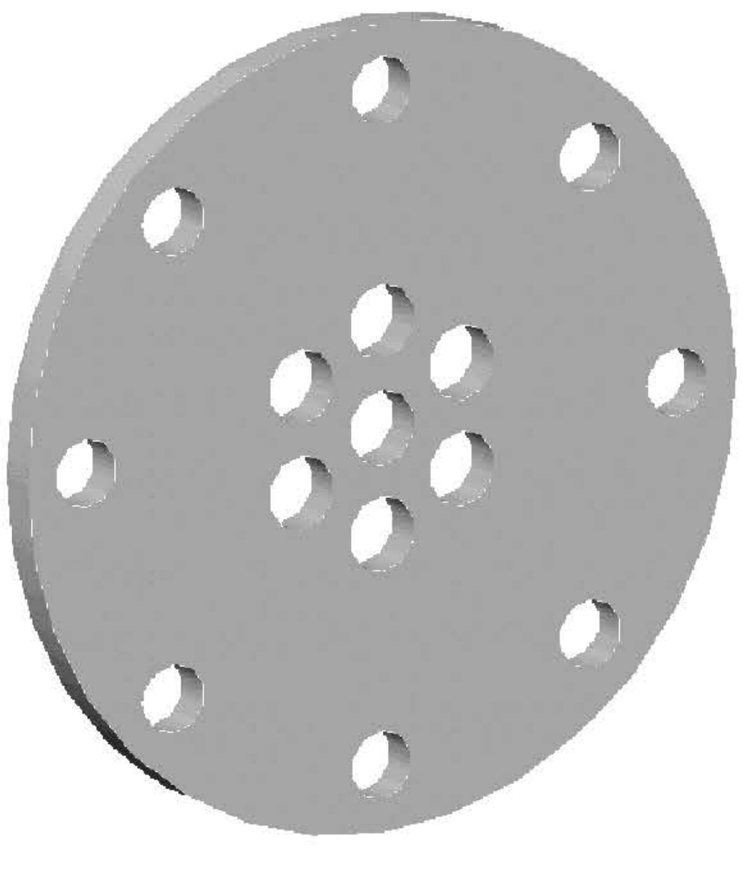


PROJECT :-	SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :- NIRMA UNIVERSITY	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :- ERDA	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :-	EVAPORATOR & ABSORBER ARRANGEMENT
DRAWING NO :- SVAR-M3-DT08.DWG	REV. NO :- 01



DETAIL 9

(4" BLIND FLANGE AS TUBE SHEET)
(FOR HEAT EXCHANGER)



PROJECT :-

SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-

ABHIJIT. S. MUSALE.
(Roll No : 11MMET08)

COLLAGE :-

INSTITUTE OF TECHNOLOGY,
NIRMA UNIVERSITY
(Guide : Prof. N.K.SHAH)

INDUSTRY :-

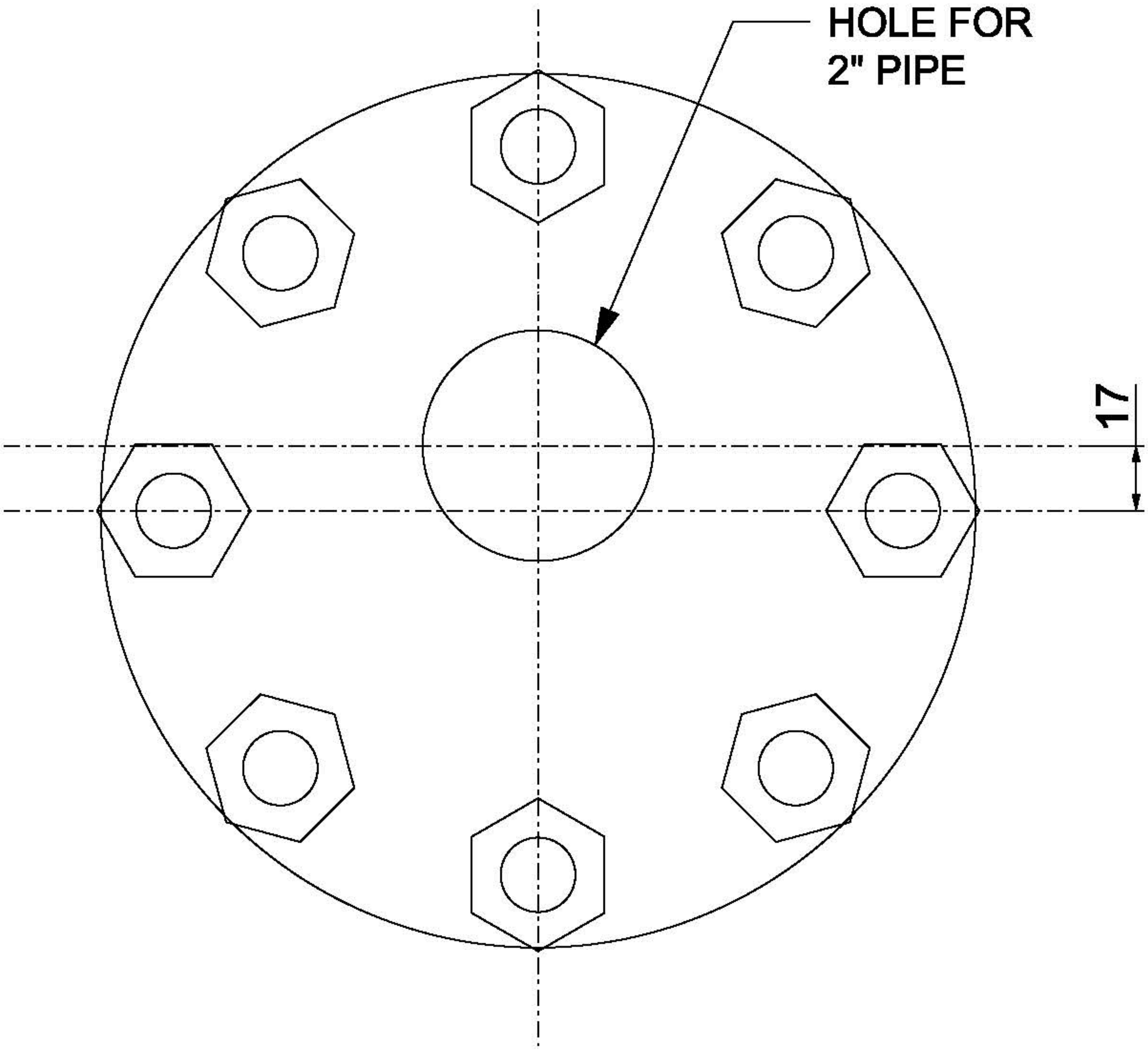
ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)
(Guide : Dr. V. SHRINET)

DRAWING TITLE :-

HEAT EXCHANGER, CONDENSER AND GENERATOR ARRANGEMENT

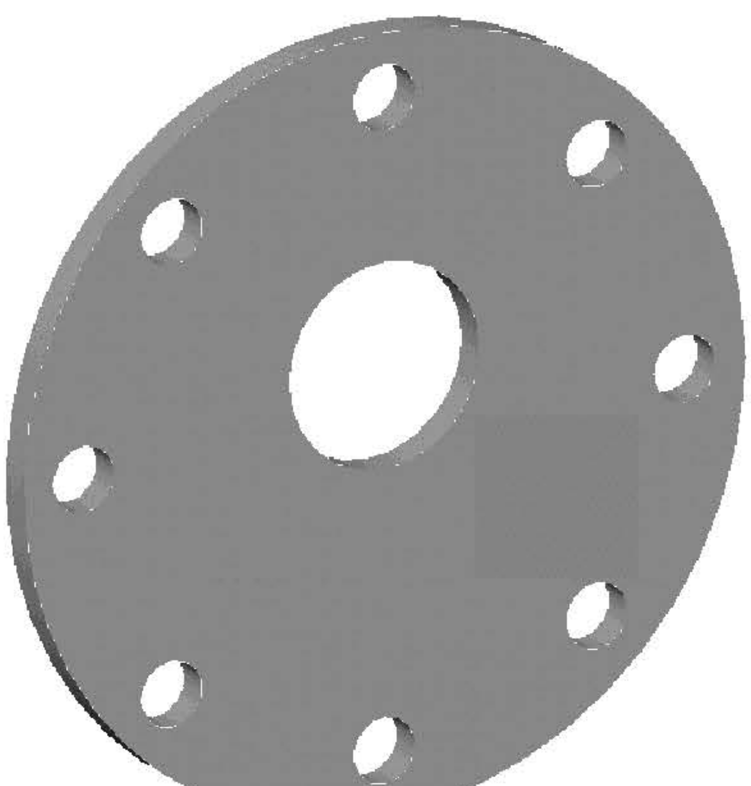
DRAWING NO :- SVAR-M3-DT09.DWG

REV. NO :- 01

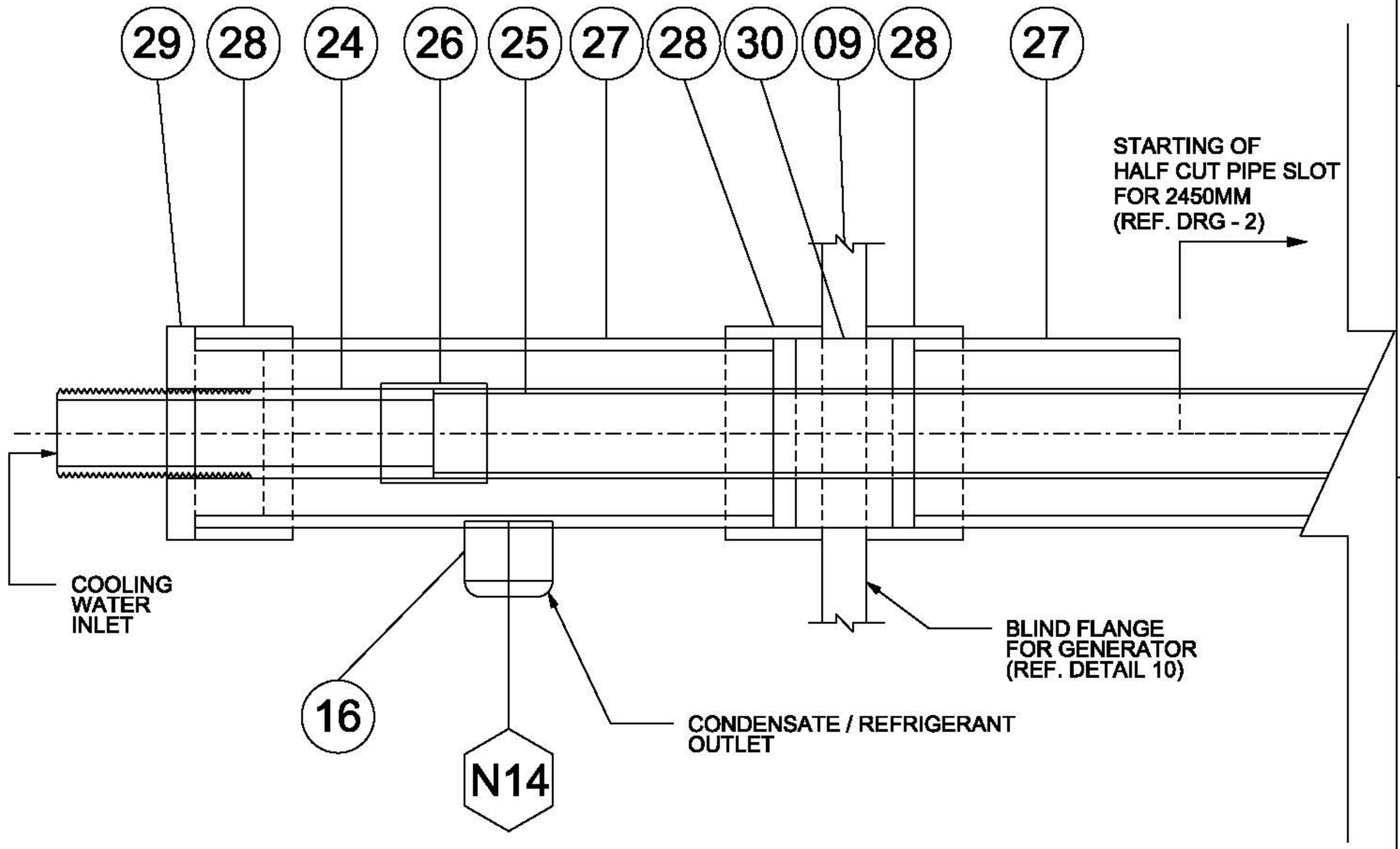


DETAIL 10

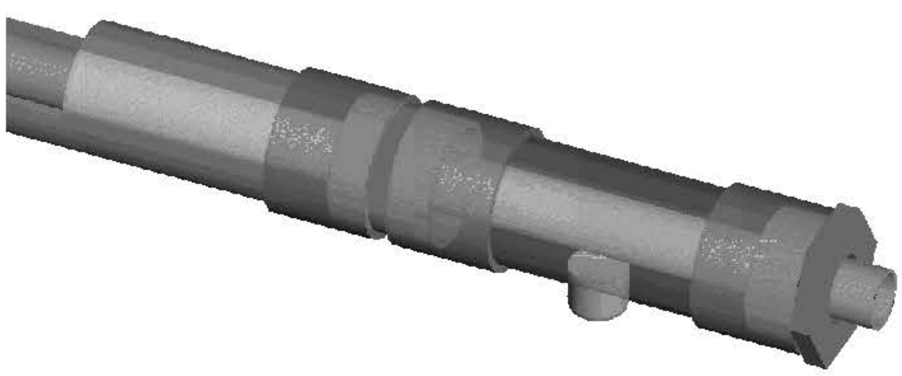
(4" BLIND FLANGE DETAILS)
(FOR GENERATOR)



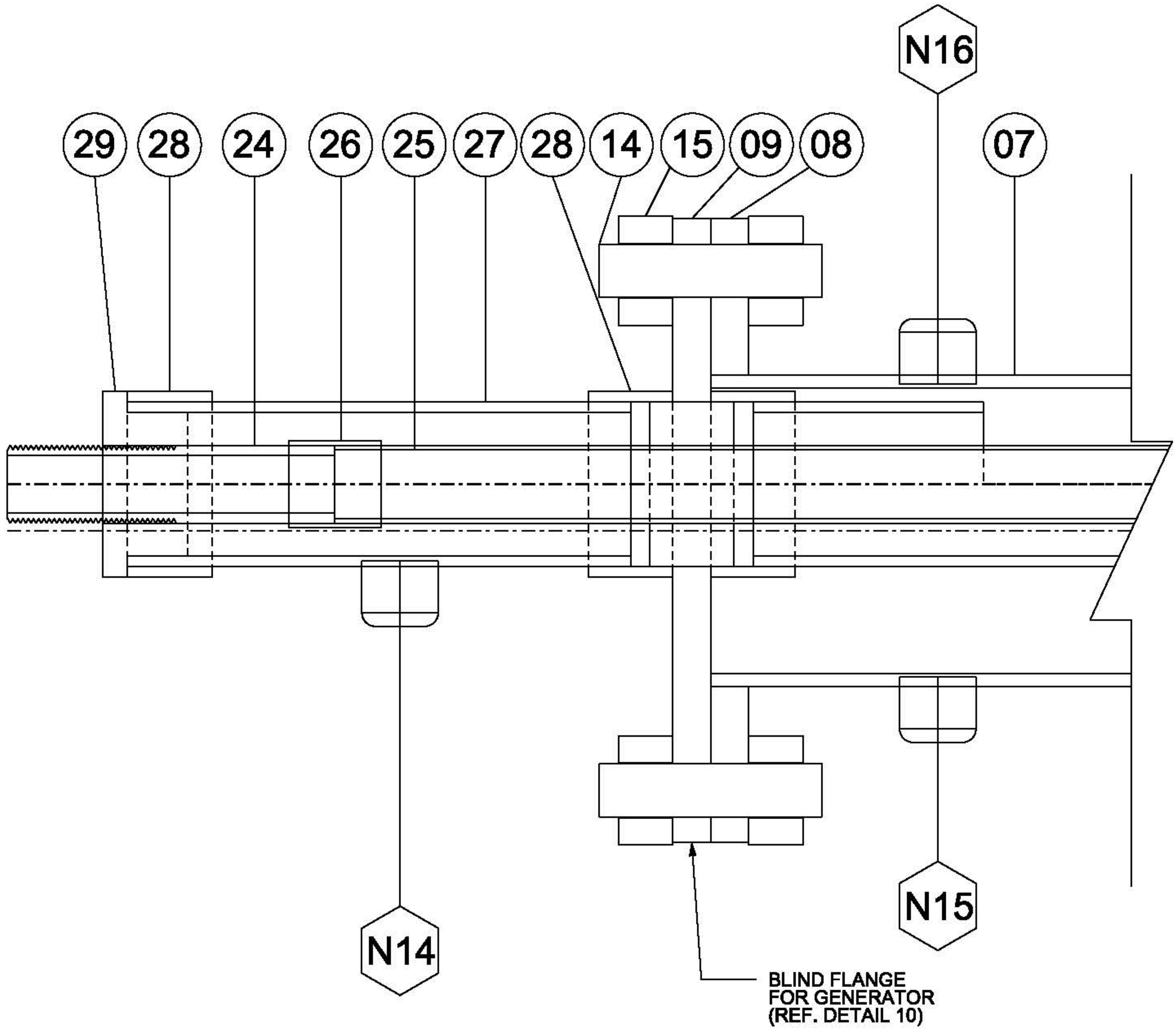
PROJECT :-	SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :-	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :-	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :-	HEAT EXCHANGER, CONDENSER AND GENERATOR ARRANGEMENT
DRAWING NO :-	SVAR-M3-DT10.DWG
REV. NO :-	01



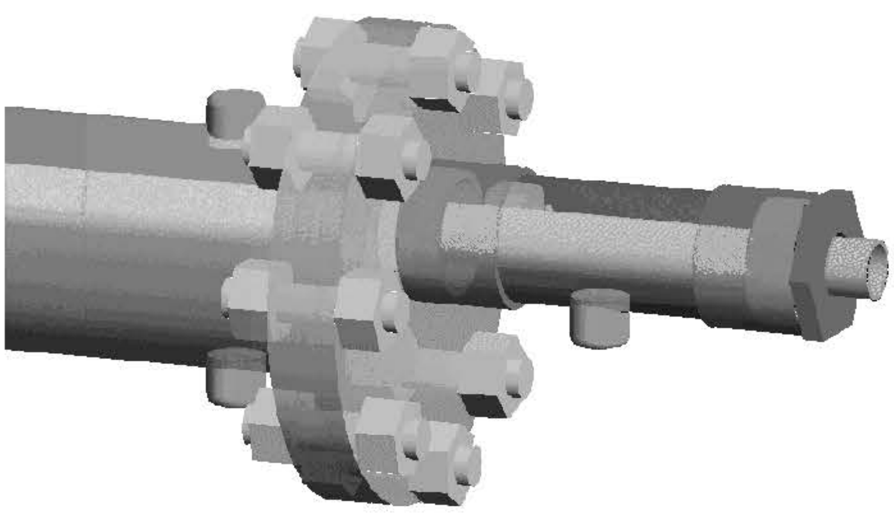
DETAIL 11



PROJECT :-	SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM
STUDENT :-	ABHIJIT. S. MUSALE. (Roll No : 11MMET08)
COLLAGE :- NIRMA UNIVERSITY	INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY (Guide : Prof. N.K.SHAH)
INDUSTRY :- ERDA	ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA) (Guide : Dr. V. SHRINET)
DRAWING TITLE :-	HEAT EXCHANGER, CONDENSER AND GENERATOR ARRANGEMENT
DRAWING NO :- SVAR-M3-DT11.DWG	REV. NO :- 01



DETAIL 12



PROJECT :-

SOLAR AND FUEL BASED VAPOR ABSORPTION REFRIGERATION SYSTEM

STUDENT :-

ABHIJIT. S. MUSALE.

(Roll No : 11MMET08)

COLLAGE :-

INSTITUTE OF TECHNOLOGY, NIRMA UNIVERSITY

(Guide : Prof. N.K.SHAH)

INDUSTRY :-

ELECTRICAL RESEARCH AND DEVELOPMENT AGENCY (ERDA)

(Guide : Dr. V. SHRINET)

DRAWING TITLE :-

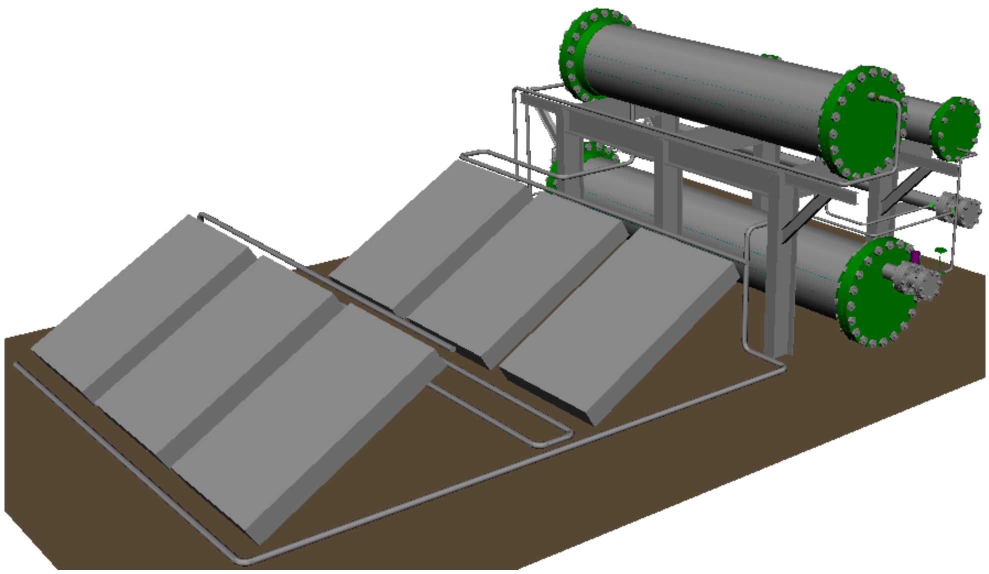
HEAT EXCHANGER, CONDENSER AND GENERATOR ARRANGEMENT

DRAWING NO :- SVAR-M3-DT12.DWG

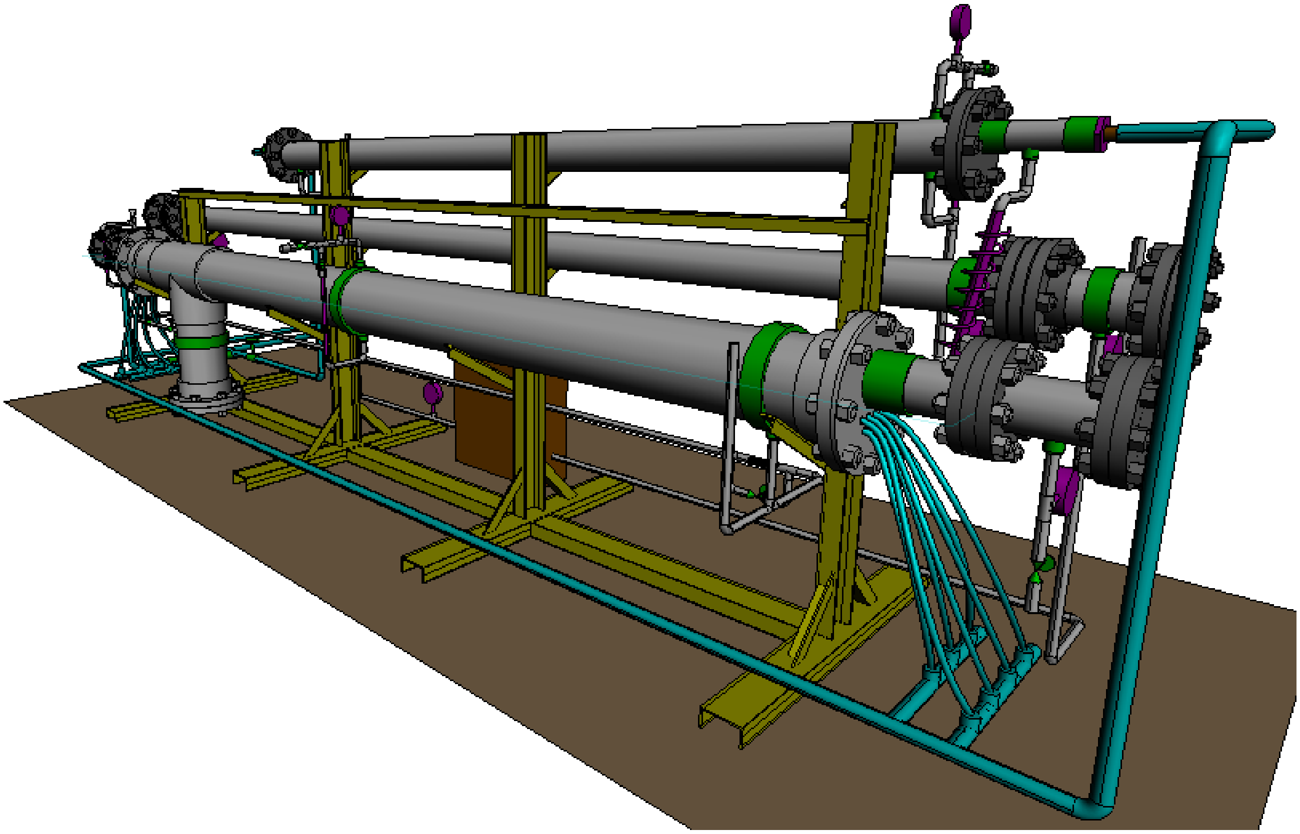
REV. NO :- 01

Appendix C

Images



Above is an image of the 3D model of solar operated VAR system, prepared in CAD software which can be found in attached CD.



Above is the 3D model of Fuel operated VAR system prepared in CAD software. Bellow is a photograph of actual prototype constructed at ERDA. More images and 3D model can be found in attached CD.