A Project Report on

SOLAR WATER COOLER

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In

MECHANICAL ENGINEERING

Under guidance of

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CERTIFICATE

This is to certify that the project entitled as "**SOLAR WATER COOLER**" done by following students has been successfully completed under my supervision and guidance in partial fulfillment for the award of Bachelor of engineering degree in the Mechanical engineering discipline of Government College of engineering, Aurangabad.

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During this period, I found them hardworking, sincere and obedient.

I wish them good luck in all future endeavors.

Prof. K. S. Wasankar
Guide

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Chapter-1

Problem identification - Need of solar refrigeration

1.1 Introduction

In the Stone Age, the average human had at his disposal about 4,000 calories of energy per day which included energy invested in preparing tools, clothing, art and campfires along with the food. Today Americans (and inhabitants of Europe and Asia to a lesser extent) use on average 228,000 calories of energy per person per day, to feed not only their stomachs but also their cars, computers, refrigerators and televisions [1]. The fact that the average American uses 57 times more energy than the average Stone Age hunter-gatherer implies that energy is the soul of the modern world. The economic growth and technological advancement of every country depends on it. The quality of life of a common citizen is often thought to be reflected in the country's available energy.

The unprecedented growth of economy and population of India during the last few decades have caused the increase in demand for energy, making reliable energy one of the massive challenges for the 21st century.

In the modern world, fossil fuels are used for transportation, communication, industrial, educational and domestic purposes for heating, cooling, cooking, lighting and industrial as well as domestic appliances. However, concern grows daily over the negative impacts fossil fuels have on the environment, whether from their limited sources or from global warming through depletion of the ozone layer. Carbon dioxide is produced when fossil fuels are burned, causing an increase to the earth's temperature and a green-house

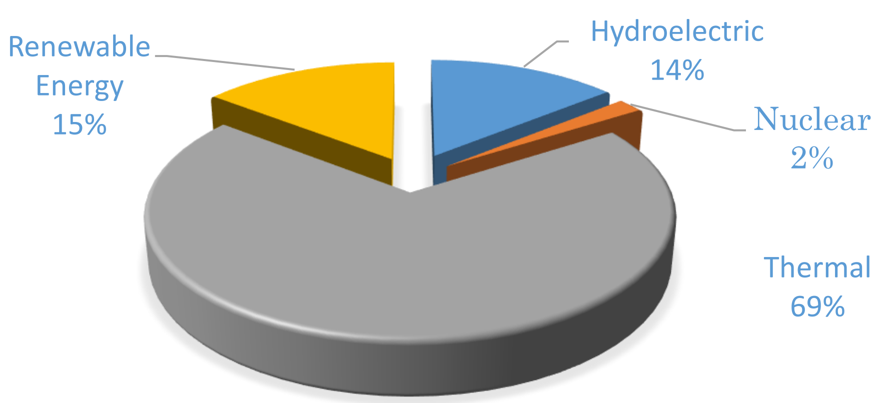


Figure 1.1 Sources of Electricity in India by Installed Capacity [2]

gas effect on the atmosphere's ozone layer. In the era where problems like ozone layer depletion and global warming are about to exert catastrophic effects, the fact that only 14.5% of electricity production in India uses renewable energy sources is alarming. The non-renewable energy sources, therefore, pose threat to the modern civilization.

The Vienna Convention for the Protection of the Ozone Layer (1985), the Kyoto Protocol on Global Warming (1998) and the five amendments of the Montreal Protocol (1987) all discussed the reduction of CFCs (chlorofluorocarbons) to protect the ozonosphere, but the situation continues to decline. According to the latest NASA investigation, the holes in the ozonosphere over the two poles currently occupy approximately 28,300,000 km^2, up from approximately 24,000,000 km^2 in 1994 [3]. As a consequence, the European Commission Regulation 2037/2000, implemented on 1 October 2000, works to control and schedule all the ozone depleting materials; all HCFCs (hydro-chlorofluorocarbons) will be prohibited by 2015 [4]. The issue remains of seeking an alternative to fossil fuels before they deplete and/or destroy the earth.

The difficulties involved in the acquisition and distribution of conventional energy sources in developing countries is due mainly to the poor road infrastructure, and the high cost of installation of hydroelectric dams and nuclear power stations have given rise to much interest in renewable energies in general and solar energy in particular.

In some developing countries, despite the efforts by most governments to meet the energy needs of the growing population, only a very little percentage of the population has access to conventional forms of energy. Because of these problems it is necessary to look for other sources of energy that are easy to harness and that do not require long distances for transportation; energy sources that are gentle to the environment, renewable and cheaper to exploit. The availability of these sources would create a lot of small-scale economic activities and generally improve the quality of life of the rural population. Harnessing solar energy and other forms of renewable energies seem to be the most economically viable for enclaved rural areas [5].

Good health and energy are interdependent factors, which determine to a great extent the progress of development of the rural sector. Thirty-eight countries in the world that have the highest infant mortality rate are those in which over 73 percent of the population live in the rural areas [6]. Renewable energy sources that could power some of the primary health equipment can considerably help in reducing infant mortality in rural areas, improve their health status, and improve on their intelligence. This in turn will have a major impact on the agricultural productivity of the rural population.

1.2 Solar energy

Solar energy is the result of electromagnetic radiation released from the Sun by the thermonuclear reactions occurring inside its core. Although all electromagnetic waves have the same general features, waves of different wavelength differ significantly in their behavior. The electromagnetic radiation encountered in practice covers a wide range of wavelengths and includes gamma rays, X-rays, ultraviolet radiation, visible light, infrared radiation, microwaves, and radio waves.

Different types of electromagnetic radiation are produced through various mechanisms. For example, gamma rays are produced by nuclear reactions, X-rays by the bombardment of metals with high-energy electrons, microwaves by special types of electron tubes such as klystrons and magnetrons, and radio waves by the excitation of some crystals or by the flow of alternating current through electric conductors.

Thermal radiation (or simply light) constitutes complete infrared and visible spectrum and a part of ultraviolet spectrum as shown in the following figure [7].

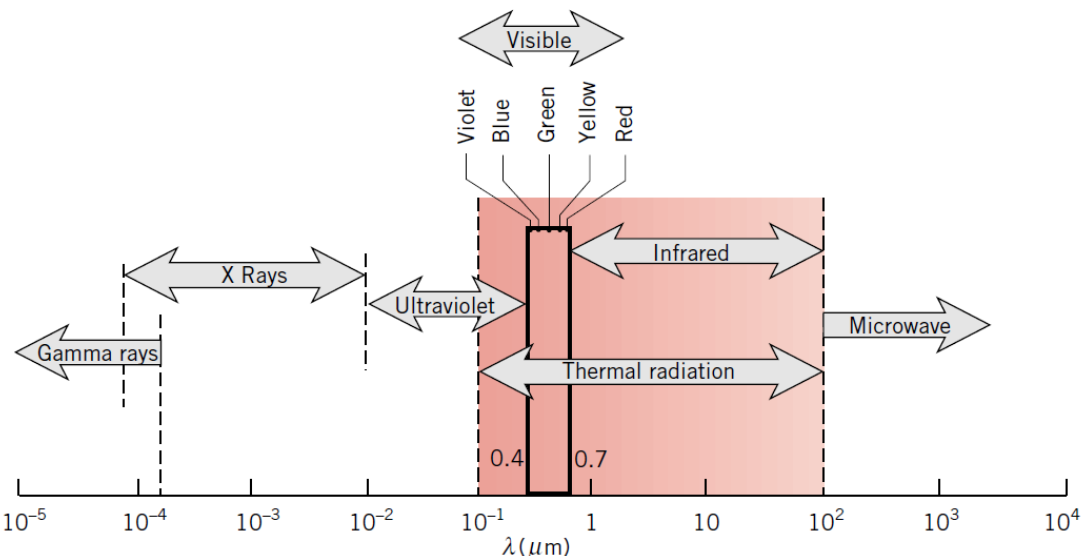


Figure 1.2 Electromagnetic spectrum

All of the energy resources on earth originate from the sun directly or indirectly, except for nuclear, tidal and geothermal energy. The sun actually transmits a vast amount of solar energy to the surface of the earth. The term "solar constant, S_c" signifies the radiation influx of solar energy. Calculated mean solar constant value is equal to 1368 W/m^2 [8]. Therefore, considering a global plane area of 1.275×10^{14} m^2 and the mean radius of the earth being approximately 6371 km, the total radiation transmitted to the earth is 1.74×10^{17} W.

1.3 Need of refrigeration system driven by solar energy

Keeping development up-till now in mind, along with photovoltaic systems, solar thermal energy has been used over the last few decades to meet the refrigeration needs for both domestic and industrial purposes. Solar powered refrigeration and air conditioning have been very attractive since the availability of sunshine and need for refrigeration both reach maximum levels in the same season. In India, refrigerators have the highest aspirational value of all consumer durables, with the exception of televisions. This accounts for the high growth rate of the refrigerator market. The refrigerator market has been growing at a rate of about 15% per year, while the consumer durables industry as a whole has grown at almost 8%. The size of the refrigerator market is estimated to be 3.5 to 4 million units approximately, valued at ₹50 billion [9]. The AC market has also been growing but the penetration of AC's in Indian household is limited due to negative factors like power shortage and poor quality of power. Therefore switching to alternative eco-friendly refrigeration systems harnessing solar energy adds new dimension to the world of refrigeration. These systems gives some amount of relief to the refrigeration world by making it independent of electric power supply and zero running cost.

1.4 Why solar water cooler?

On the one hand the size of the Indian refrigerator market is currently valued at ₹50 billion and on the other hand adequate supplies of drinking water and water for irrigation are still a fantasy in most of the rural parts of India. In many places supplies of water are found only at considerable depth below the surface. Remote locations generally do not have the infrastructure to provide an electrical grid to pump the water with electricity nor do they have the infrastructure to provide roads to bring in electrical generators or even the fuel for those generators. Therefore without an electrical grid or without generators to generate electricity, most rural areas do not have potable water nor do they have the refrigeration to keep medicine or foodstuffs from spoiling.

To soothe these problems to some extent, a refrigeration system which does not require electricity would be nothing less than a panacea. A refrigerator working on vapor absorption refrigeration cycle harnessing solar energy will help us fight following problems:

1. India is among the world leaders in agricultural production however much of our produce goes waste due to absence of proper cold storage facilities.

2. Milk produce is also adversely affected due to lack of refrigeration.

3. Cool drinking water is unavailable to the people in non-electrified villages.
4. Medical facilities are also adversely affected due to break in the cold chain as the medicines move from the production zone to the rural areas.

We have decided to work on design, fabrication and testing of a solar water cooler so that cool drinking water will be available to the people in non-electrified areas. Solar water cooler also has potential applications in urban areas where it can be used in schools and colleges, government offices, hospitals and any other institutes where cool drinking water is essentially required in large amounts. The roofs of these types of buildings can house the solar water cooler easily.

Chapter-2

Refrigeration

2.1 Qualitative overview of refrigeration systems

Refrigeration is a process of moving heat from low temperature reservoir to high temperature reservoir under controlled conditions with the intention of maintaining below atmospheric temperature.

Many refrigeration systems have been developed according to application and some of them are depicted in the Figure 2.1. Now we will discuss working principle of these refrigeration systems.

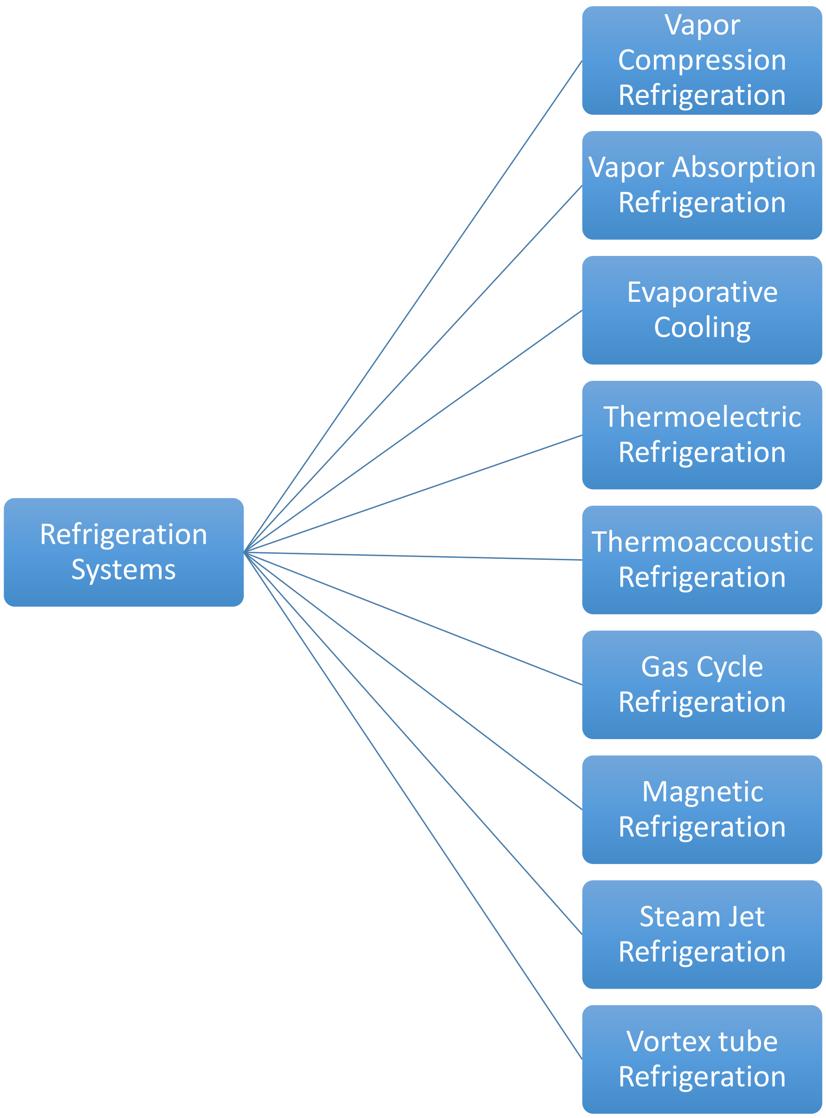


Figure 2.1 Different refrigeration systems

Vapor-Compression Refrigeration or **vapor-compression refrigeration system (VCRS)**, in which the refrigerant undergoes phase changes, is one of the many refrigeration cycles and is the most widely used method for air-conditioning of buildings and automobiles. It employs a compressor to give work input for desired refrigeration effect.

An absorption refrigerator working on **Vapor Absorption Refrigeration Cycle** is a refrigerator that uses a heat source (e.g., solar energy, a fossil-fueled flame, waste heat from factories, or district heating systems) to provide the energy needed to drive the cooling process. These systems use waste heat or low grade thermal energy for producing refrigeration effect.

An **evaporative cooler** (also swamp cooler, desert cooler and wet air cooler) is a device that cools air through the evaporation of water. Evaporative cooling differs from typical air conditioning systems which use vapor-compression or absorption refrigeration cycles. Evaporative cooling works by employing water's large enthalpy of vaporization. The temperature of dry air can be dropped significantly through the phase transition of liquid water to water vapor (evaporation), which can cool air using much less energy than refrigeration. In extremely dry climates, evaporative cooling of air has the added benefit of conditioning the air with more moisture for the comfort of building occupants.

Thermoelectric cooling uses the Peltier effect to create a heat flux between the junctions of two different types of materials. A Peltier cooler, heater, or thermoelectric heat pump is a solid-state active heat pump which transfers heat from one side of the device to the other, with consumption of electrical energy, depending on the direction of the current.

Thermoacoustic refrigeration systems operate by using sound waves and a non-flammable mixture of inert gas (helium, argon, air) or a mixture of gases in a resonator to produce cooling.

In **gas cycle refrigeration,** the working fluid is a gas that is compressed and expanded but doesn't change phase. Air is most often this working fluid. The gas cycle is less efficient than the vapor compression cycle because the gas cycle works on the reverse Brayton cycle instead of the reverse Rankine cycle. For the same cooling load, a gas refrigeration cycle needs a large mass flow rate and is bulky.

Magnetic refrigeration is a cooling technology based on the magneto-caloric effect. This technique can be used to attain extremely low temperatures, as well as the ranges used in common refrigerators. Compared to traditional gas-compression refrigeration, magnetic refrigeration is safer, quieter, and more compact, has a higher cooling efficiency, and is more environmentally friendly because it does not use harmful, ozone-depleting coolant gases.

Steam jet cooling uses a high-pressure jet of steam to cool water or other fluid media. Typical uses include industrial sites, where a suitable steam supply already exists for other purposes or, historically, for air conditioning on passenger trains which use steam for heating.

The **vortex tube**, also known as the **Ranque-Hilsch vortex tube**, is a mechanical device that separates a compressed gas into hot and cold streams. The air emerging from the "hot" end can reach temperatures of 200 °C (392 °F), and the air emerging from the "cold end" can reach −50 °C (−58 °F). It has no moving parts.

2.2 Solar refrigeration cycles

Starting from the inflow of solar energy, the refrigeration cycles which harness solar energy are broadly of two types: solar thermal collectors to heat or PV cells to generate electricity. Thermally driven cycles are absorption, adsorption, chemical reaction, desiccant cooling, ejector and the Rankine refrigeration cycles. PV driven cycles are the Sterling, thermo-electric and vapor compression refrigeration cycles. These cycles are shown in the Figure 2.2.

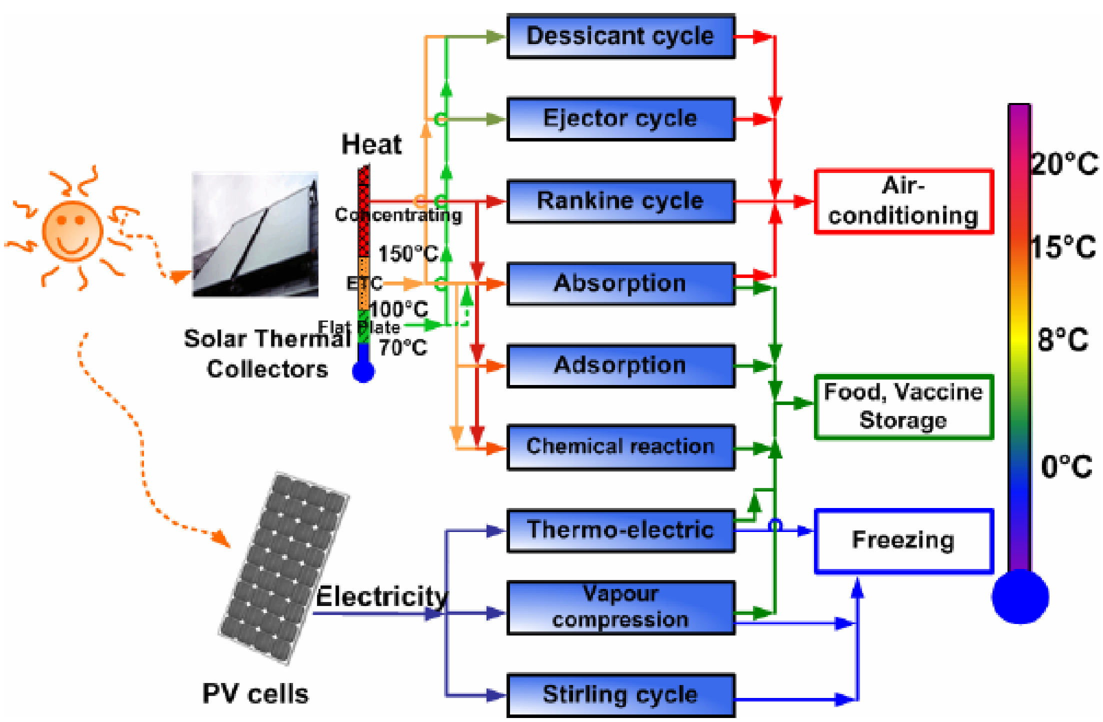


Figure 2.2 Different solar refrigeration cycles

Desiccant cooling is a combination of dehumidification and evaporative cooling processes. Moisture is removed from the air by a desiccant material, thus the latent heat load is removed. The heated air is then cooled in the heat exchanger wheel. The air temperature is further lowered and humidified by an evaporative cooling process. The desiccant material can be regenerated by heat.

The ejector in the **ejector cycle** is a thermally driven compressor that operates in a heat pump refrigeration cycle. A liquid pump is required to generate a pressure difference for the ejector heat pump to operate, but since liquid is being compressed, the amount of electricity required is relatively small. All other components in the heat pump circuit are conventional.

A Rankine power cycle and a vapor compression refrigeration cycle can be put together as a **Duplex-Rankine system**. It uses the high-pressure vapor fluid to drive a turbine in the power cycle. Consequently, work from the turbine drives the compressor in the refrigeration cycle. The working fluid in the Rankine power cycle and the refrigeration cycle can be different.

The single effect **absorption cycle** can be either operated intermittently or continuously. An ammonia-water solution is used as the working fluid pair for low temperature applications and water-lithium bromide is used for air-conditioning applications.

In case of solar driven **thermoelectric refrigeration**, cooling is achieved by driving an electric current through the electric circuit containing junctions of different metals. This phenomenon is called the reversed Seebeck effect. There are no moving parts and no working fluid in this system.

Solar powered **compression refrigeration systems** employ the use of photovoltaic array to power compressor unit. Their principle of operation is same as that of standard vapor compression refrigeration systems.

Stirling cycle solar refrigerators operate on reversed Stirling cycle. This cycle consists of two isothermal and two constant volume processes.

2.3 Advantages and disadvantages of solar refrigeration cycles

With this information about different solar refrigeration cycles, we summarized their advantages and disadvantages as follows [10]:

1. Absorption systems

Advantages	Disadvantages
Only one moving part (pump) with the possibility of no moving parts for small systems (e.g. Platen-Munters cycle)	It cannot achieve a very low evaporating temperature in using
Possible to utilize low-temperature heat supply	The system is quite complicated and difficult for service

Table 2.01 Advantages and disadvantages of absorption systems

2. Adsorption systems

Advantages	Disadvantages
No moving parts (except valves)	The high weight and poor thermal conductivity of the adsorbent make it unsuitable to use for high capacities and can cause long-term problems
Low operating temperatures can be achieved	Low operating pressure requirement makes it difficult to achieve air-tight
	Very sensitive to low temperature especially the decreasing temperature during the night.
	It is an intermittent system

Table 2.02 Advantages and disadvantages of adsorption systems

3. Chemical reaction systems

Advantages	Disadvantages
No moving parts	Low COP
Low operating temperatures can be achieved	High weight of adsorbent, not suitable to use for high capacities
	The system design is complex especially due to the volume of the adsorber that changes when chemical reaction occurs
	Low operating pressure at a lower temperature, difficult to achieve air-tightness

Table 2.03 Advantages and disadvantages of chemical reaction systems

4. Desiccant Cooling Systems

Advantages	Disadvantages
Environmentally friendly, water is used as the working fluid	It is difficult to have good control of the system in a humid area
Can be integrated with a ventilation and heating system	It is not appropriate for areas where water is scarce
The driving temperature is quite low, thus possible to use low temperature solar collector	In case of the solid desiccant, the system requires maintenance due to moving parts in the rotor wheel of the solid desiccant system

In case of the liquid desiccant, the system doesn’t need a condenser because the refrigerant can be released to the atmosphere	In case of the liquid desiccant, the liquid sorbent can contaminate the supplied air
	Crystallization generally occurs in liquid desiccant systems due to poor process control

Table 2.04 Advantages and disadvantages of desiccant cooling systems

5. Ejector refrigeration system

Advantages	**Disadvantages**
A low temperature heat source can be utilized	Low COP
Low operating and installation cost	Complicated design of the ejector
	Difficult to operate in a wide range of ambient temperatures

Table 2.05 Advantages and disadvantages of Ejector refrigeration systems

6. Rankine cycles (Duplex Rankine cycle)

Advantages	**Disadvantages**
Suitable for high capacity systems	High installation cost
Suitable for integration into polygeneration systems (heat, electricity and refrigeration)	Regular maintenance required due to complications and many moving parts

Table 2.06 Advantages and disadvantages of Rankine cycles

7. Stirling/PV systems

Advantages	Disadvantages
Relatively high COP for high temperature lifts	High production costs
Can be used for cryogenic applications and is mechanically simpler than other applications for low temperature operations	Complexity in design
Environmentally friendly	
Mobile and light weight	

Table 2.07 Advantages and disadvantages of Stirling/PV systems

8. Thermoelectric/PV systems

Advantages	Disadvantages
No working fluid and no moving parts (except fans)	Low COPs
Quiet small size and light weight	Difficult to achieve a low refrigeration temperature
	Low reliability especially when the power supply is cut off
	Requires efficient heat sink in order to reject heat from the thermo-electric module

	Not suitable for large cooling load
	Induces thermal short circuiting when not operated

Table 2.08 Advantages and disadvantages of thermoelectric/PV systems

9. Vapor compression/PV systems

Advantages	**Disadvantages**
High COP	For a PV system, installation cost is high and requires battery for energy backup
Long term experience and widely available commercially	Can be noisy
Scalable from small to a large systems	

Table 2.09 Advantages and disadvantages of vapor compression/PV systems

2.4 Vapor Absorption Refrigeration System

These systems are particularly useful where low grade thermal energy or waste heat is available. As the name implies, absorption refrigeration systems involve the absorption of a *refrigerant* by a *transport medium*. The most widely used absorption refrigeration system is the ammonia–water system, where ammonia serves as the refrigerant and water serves as the transport medium.

Vapor absorption refrigeration is similar to compression refrigeration except compressor is replaced by dotted box as shown in Figure 2.3. Here is what happens in the box [11]:

Ammonia vapor leaves the evaporator and enters the absorber, where it dissolves and reacts with water to form $NH_3{\cdot}H_2O$. This is an exothermic reaction. The amount of NH_3 that can be dissolved in H_2O is inversely proportional to the temperature. Therefore, it is necessary to cool the absorber to maintain its temperature as low as possible, hence to maximize the amount of NH_3 dissolved in water. The liquid NH_3 - H_2O solution, which is rich in NH_3, is then pumped to the generator. Heat is transferred to the solution from a source to vaporize some of the solution. The vapor, which is rich in NH_3, passes through a rectifier, which separates the water and returns it to the generator. The high-pressure pure NH_3 vapor then continues its journey through the rest of the cycle. The hot NH_3 -H_2O solution, which is weak in NH_3, then passes through a regenerator, where it transfers some heat to the rich solution leaving the pump, and is throttled to the absorber pressure.

Because liquid is compressed instead of vapor, work input required is almost negligible in case of vapor absorption system.

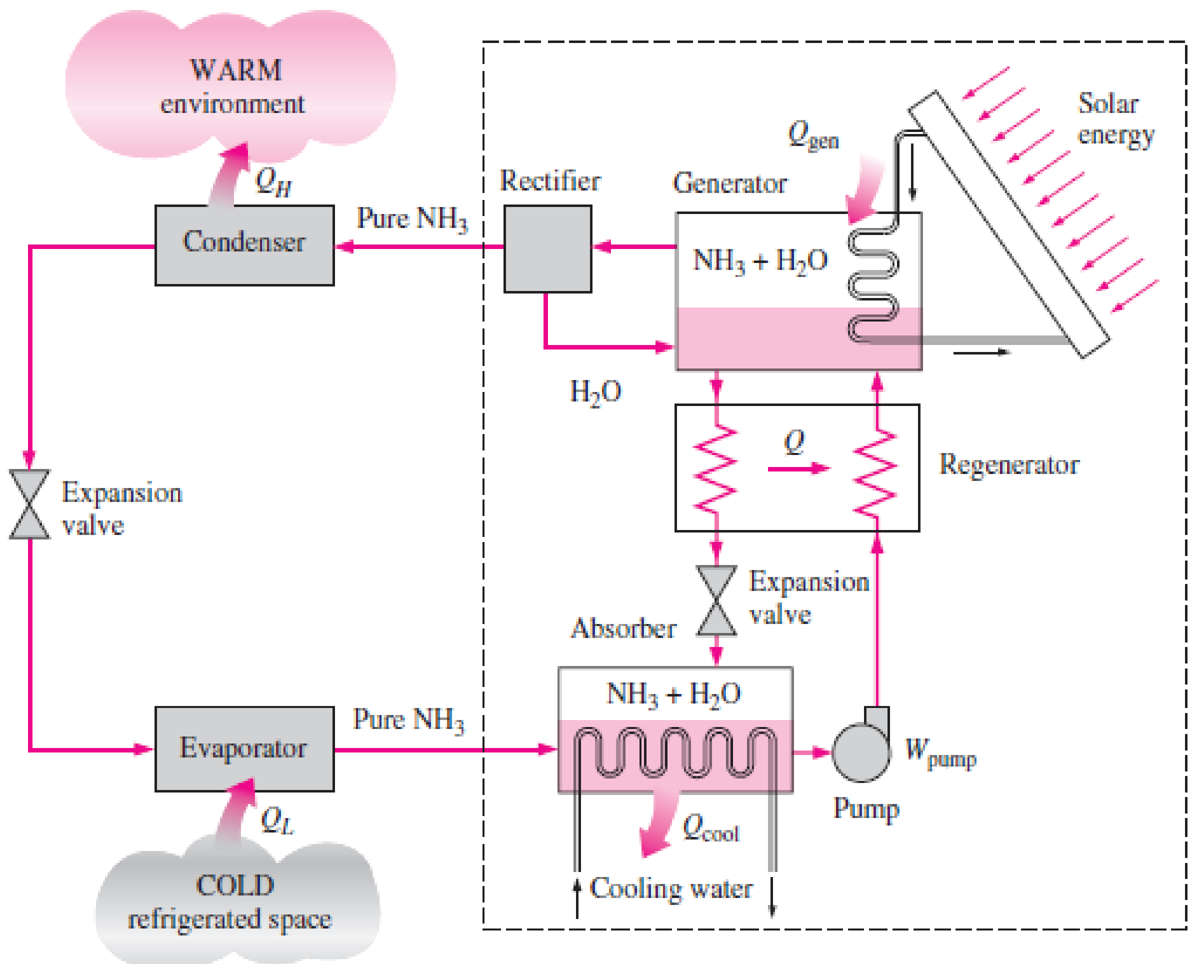


Figure 2.3 Schematic representation of Vapor Absorption Refrigeration System

2.4.1 Desirable properties of refrigerant-absorbent pair for VARS

Following are some desirable properties that must be possessed by refrigerant:

1. The refrigerant should exhibit high solubility with solution in the absorber.
2. There should be large difference in the boiling points of refrigerant and absorbent (greater than 200˚C), so that only refrigerant is boiled-off in the generator. This ensures that only pure refrigerant circulates through refrigerant circuit (condenser-expansion valve-evaporator) leading to iso-thermal heat transfer in evaporator and condenser.
3. It should exhibit small heat of mixing so that a high COP can be achieved. However, this requirement contradicts the first requirement. Hence, in practice a trade-off is required between solubility and heat of mixing.
4. The refrigerant-absorbent mixture should have high thermal conductivity and low viscosity for high performance.
5. It should not undergo crystallization or solidification inside the system.
6. The mixture should be safe, chemically stable, non-corrosive, and inexpensive and should be available easily.

Commonly, two refrigerant absorbent pairs are used: LiBr-H_2O and NH_3-H_2O. We have decided to work with aqua-ammonia as refrigerant absorbent pair due to following reasons:

H_2O-NH_3	**LiBr-H_2O**
Suitable for refrigeration applications	Suitable for air conditioning applications
Leakage is not possible because of higher saturation pressures (greater than atmospheric pressure)	Possibility of leakage due to low saturation pressures (in the order of kPa)
Cheaper and easily available	Costly when compared with ammonia
Freezing point of Ammonia is -77˚C therefore can be used to produce very low temperatures	Possibility of crystallization when operating near freezing point of water

Table 2.10 Comparison between H_2O-NH_3 and LiBr-H_2O

Chapter-3

Basic Design Aspects

With this discussion about necessity of solar powered refrigeration system and technology introduction of Vapor Absorption Refrigeration System (VARS), we can now redefine our project objective and proceed for design aspects.

3.1 Objective of Project

The objective of our project can be stated as follows:

> To design, fabricate and test a Water cooler of 05 liter volume based on Intermittent Vapor Absorption Refrigeration System (VARS) harnessing solar energy with aqua-ammonia refrigerant-absorber pair with an intention of achieving chilled water at temperature of 5°C within 6 hours.

3.2 Calculation of cooling load

The cooling load on refrigerating equipment seldom results from any one single source of heat. Rather, it is the summation of the heat which usually evolves from several different sources. Some of the more common sources of heat apart from the primary refrigeration load which supply the load on refrigerating equipment are:

1. Heat that leaks into the refrigerated space from the outside by conduction through the insulator.
2. Heat that enters the refrigerated space from the outside by direct radiation.

The importance of any one these heating sources with relation to the total cooling load on the equipment varies with the individual application. In proposed solar water cooler product load would be the major heat load.

The product load is made up of the heat which must be removed from the refrigerated product (water) in order to reduce the temperature of the product to the desired level. This heat removed from product is the required cooling load. The amount of heat given off by the product in cooling to space depend upon the temperature of the space and upon the weight, specific heat, and entering temperature of product. The heat gain from the product is computed by the following equation:

$$\dot{Q} = W \times C_p \times (T_1 - T_2)$$

Where, $\dot{Q}$ = Cooling load

W = Weight of products

C_p = Specific heat of products

T_1 = Entering temperature

T_2 = Space temperature of products

In case of solar water cooler, evidently the cooling product is water and specific heat of water is 4.18 kJ/kg-K. Capacity of this proposed solar refrigerator is 05 liters. Generally atmospheric temperature of water that is entering temperature of water at inlet of solar water cooler is 30 °C. The intended temperature of water at outlet is 05 °C. It is expected that this process of cooling must be completed in 6 hours.

$$\dot{Q} = W \times C_p \times (T_1 - T_2)$$

$$\dot{Q} = 05 \times 4.18 \times (30 - 05)$$

$$\dot{Q} = 522.5 \text{ kJ}$$

$$\text{Cooling load } (\dot{Q}_L) = \frac{522.5}{3600 \times 6}$$

$$\text{Cooling load } (\dot{Q}_L) = 0.02419 \text{ kW}$$

However, the cooling load just now we calculated is not just limited to cooling water. As it is evident from the table 3.1, milk and milk products, most vegetables and fruits have specific heat values around the specific heat of water (4.18 kJ/kg-K). Therefore, with slight modifications in the design of the evaporator, the solar water cooler can also be used for storing products mention in table 3.1.

Product	**Specific heat(kJ/kg-K)**
Water	4.18
Milk	3.93
Cream	3.77
Cabbage	3.94

Carrot	3.81
Lemon	3.81
Tomato	3.98
Apple	3.63
Grapes	3.60
Cucumber	4.09

Table 3.01 Specific heats of perishable products

3.3 Calculation of mass flow rate of ammonia

Amount of heat calculated above as cooling load is taken off by liquid ammonia and it gets converted into saturated vapor in evaporator space. While calculating, evaporator temperature is taken as 0 °C since a temperature difference is required for heat transfer to take between liquid ammonia and the water. Whereas condenser temperature is assumed to be 40 °C where the temperature of water in condenser shell would be around 30 to 35°C. This temperature difference facilitates heat transfer between condenser tubes and surrounding water, and eventually ammonia gets condensed to liquid state. With this data of temperatures of evaporator and condenser along with cooling load, we can determine mass flow rate of Ammonia vapor in evaporator. For this calculation, value of enthalpy change of ammonia in evaporator is required. This enthalpy change can be calculated from P-h chart of Ammonia shown in fig. 3.1.

We have,

Evaporator temperature (T_1) = 0 °C

Condenser temperature (T_3) = 40 °C

From P-h chart, we get the values of enthalpy as follows:

$T_1 = 0$ °C $h_1 = 1462.24$ kJ/kg

$T_3 = 40$ °C $h_4 = h_3 = 390.64$ kJ/kg

$\Delta h = h_1 - h_4$

$\Delta h = 1462.24 – 390.64$

$\Delta h = 1071.60$ kJ/kg

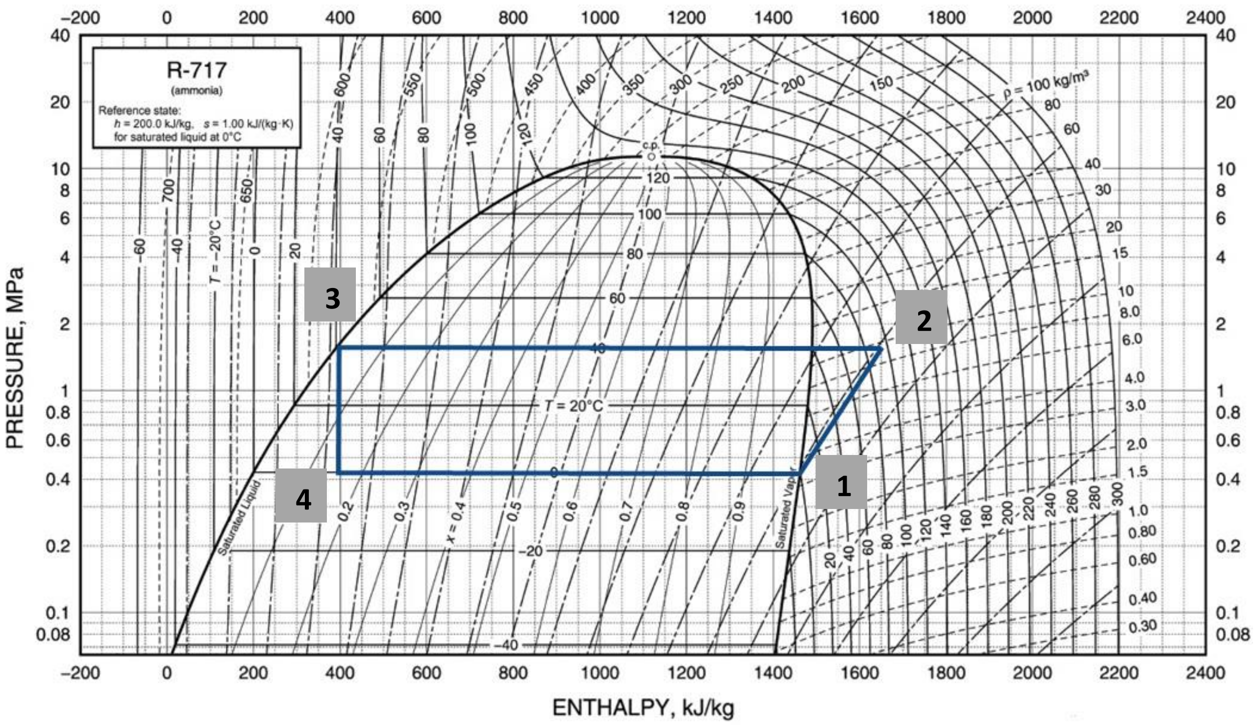


Figure 3.1 Pressure-enthalpy chart of R-717 (ammonia)

We have relation,

$$\dot{Q}_L = m \times \Delta h$$

Where, $\dot{Q}_L$ = Cooling load

$\dot{m}$ = Mass flow rate

Δh = Enthalpy change

Thus, we have,

$$\dot{Q}_L = \dot{m} \times \Delta h$$

$$0.02419 = \dot{m} \times 1071.60$$

$$\dot{m} = \frac{0.02419}{1071.60}$$

$$\dot{m} = 2.26 \times 10^{-5} \text{ kg/s}$$

$$\dot{m} = 81.26 \text{ gm/hr}$$

Therefore, if desired refrigeration effect is to be produced, mass flow rate of ammonia should be 81.26 gm/hr.

3.4 Calculation of required generator temperature

In above calculation we get required mass flow rate of ammonia in cycle. This much mass flow of ammonia must be produced in generator with expense of heat energy. Now next step is to calculate required temperature of generator so that ammonia vapor would produce in generator.

We have,

Evaporator temperature	= 0 °C	
Evaporator saturation pressure	= 4.2938 bar	[12]
Condenser temperature	= 40 °C	
Condenser saturation pressure	= 15.554 bar	[12]
Absorber temperature	= 35 °C	
Absorber saturation pressure	= 4.2938 bar	[12]

The objective is to produce an ammonia liquid of 0 °C temperature in the evaporator, the saturation vapor pressure of anhydrous ammonia at this temperature is 4.2938 bar. The temperature of the absorber is the atmospheric temperature which is assumed to be 35 °C. Thus in the absorber there is an aqua-ammonia mixture at temperature of 35 °C with the pressure of ammonia vapor at 4.2938 bar.

This temperature and pressure in receiver is located on Concentration-Pressure-Temperature (CPT) chart for aqua-ammonia pair as point 1 in the fig. 3.2. Hence, from this chart the percentage concentration (C_1) is found to be 0.4791. Extrapolating this vertical line of concentration so as to cut the line of condenser pressure at point 2 as shown in Figure 3.2.

We have relation of mass fraction,

$$X_i = \frac{\text{Mass of Ammonia (Ma)}}{\text{Mass of Water (Mw)}} = \frac{17Ci}{18(1-Ci)}$$

$$X_i = \frac{17Ci}{18(1-Ci)}$$

$$X_1 = \frac{17\times 0.4791}{18\times(1-0.4791)} \qquad (\text{As } C_1 = 0.4791)$$

$$X_1 = 0.8686$$

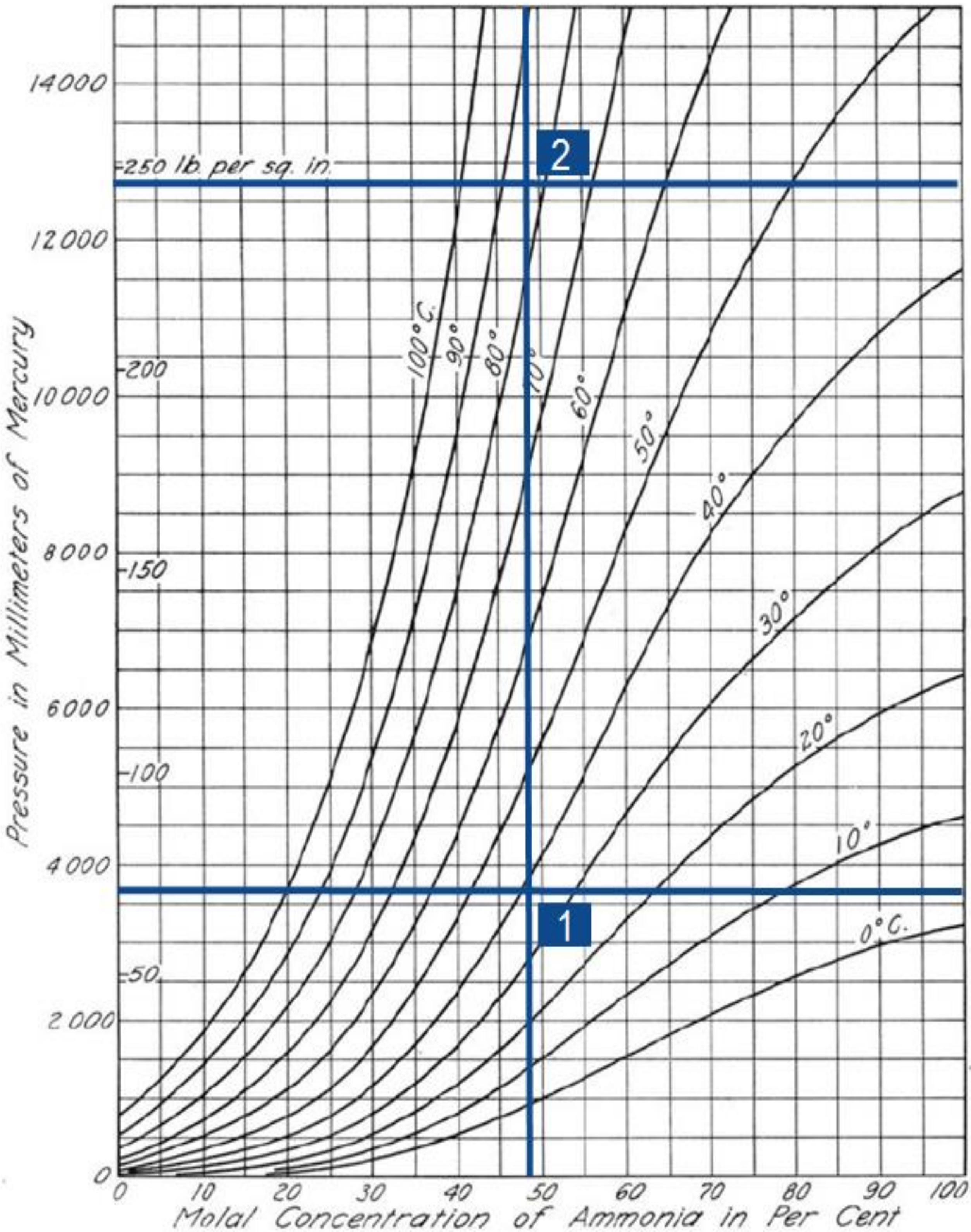


Figure 3.2 Concentration-Pressure-Temperature chart for aqua-ammonia pair

It means that if there is 1000gm of water in absorber then there would be 868.6gm of Ammonia. After the process of generation we want concentration of ammonia in solution left in absorber in such way that it would have produced required mass flow of ammonia that is $(81.26\ gm/hr) \times (6\ hr) \approx 500gm$ ammonia.

Therefore, mass of water per 500gm ammonia just before generation is as follows:

$$X_1 = \frac{m_a}{m_w} = \frac{500}{m_w} = 0.8683$$

So, m_w = 575gm.

Now, liberating this required amount of ammonia from a large volume of aqua-ammonia solution by varying its concentration by a small degree is easier as compared to liberating it from a small volume of aqua-ammonia solution by varying its concentration by a large degree. Hence, in order to reduce concentration difference, both m_a and m_w are multiplied by factor of 6. This simply means that total amount of the solution is increased in such a manner that 500gm ammonia will be generated even with relatively low generator temperature.

Therefore, mass of solution before generation:

$$m_{sol1} = m_{a1} + m_{w1} = 6\ (500 + 575) = 6450\ gm$$

In this solution, there is 3000gm ammonia and 3450gm water.

Mass of solution after generation:

$$m_{sol2} = m_{sol1} - m_{gen} = 6450 - 500 = 5950gm$$

Therefore, the mass fraction after generation would be 0.78734. With this mass fraction we can calculate percentage concentration of ammonia required as follows.

$$X_i = \frac{17Ci}{18(1-Ci)}$$

$$0.78734 = \frac{17 \times Ci}{18 \times (1-Ci)}$$

$$C_i = 0.4546$$

From CPT chart for aqua-ammonia solution, the line with this concentration of 0.4546 intersects with line of condenser pressure at point 3 as shown in Figure 3.3. This point 3 shows that minimum temperature required in generator is nearly 88 °C.

With taking this as base and considering feasibility we selected Fresnel lens to harness solar energy which is discussed in the following chapter.

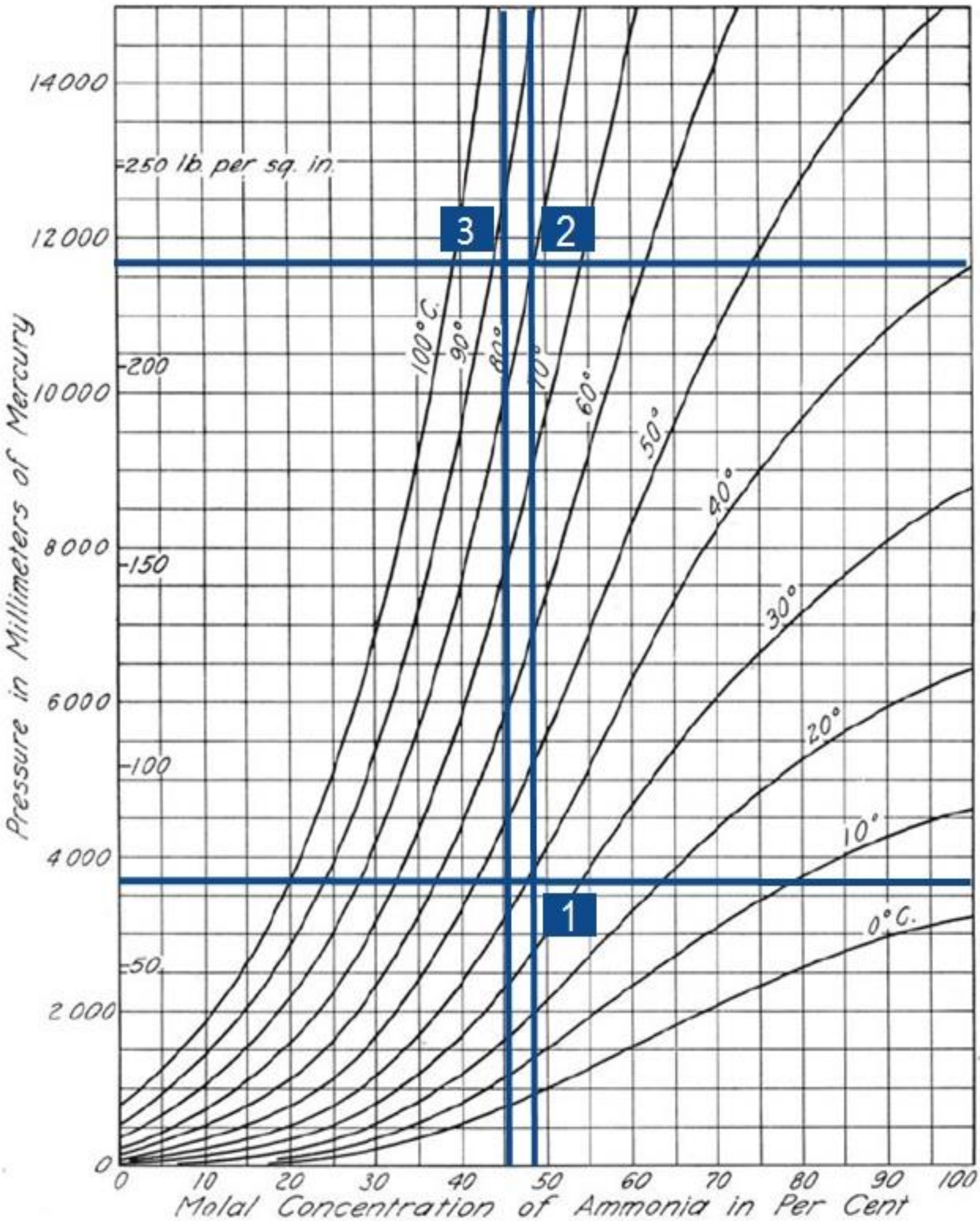


Figure 3.3 Concentration-Pressure-Temperature chart for aqua-ammonia pair

Chapter-4

Technologies to harness solar energy

In the previous chapter, we have seen that in order to achieve required mass flow rate of ammonia, the temperature of the generator needs to be equal to or above 88°C. Before finally choosing the Fresnel lens we studied following two different ways to achieve this temperature using solar technologies:

1. Solar water heater
2. Parabolic reflectors

4.1 Solar water heater

The solar energy incident on the absorber panel coated with selected coating transfers the heat to the riser pipes underneath the absorber panel. The water passing through the risers gets heated up and is delivered the storage tank. The re-circulation of the same water through absorber panel in the collector raises the temperature to 80 °C (Maximum) in a good sunny day. The total system with solar collector, storage tank and pipelines is called solar hot water system.

Water heated by solar energy has many domestic as well as industrial applications. Domestic applications includes bathing, washing, cleaning etc. whereas industrial applications comprises milk processing, sugar, pulp-paper, textile and pharmaceutical industries.

Figure 4.1 Solar water heater

Ironically, the solar water heater provides hot water of about 80°C on hot sunny days in summer when it is least needed for domestic purposes. Therefore attempts have been made to use the heat of hot water as input to generator of vapor absorption system. [13]

But we require the temperature of the generator to be 88°C which necessitates that the temperature of water from solar water heater to be greater than 88°C so that heat transfer takes place between water and the solution in generator. Therefore it seems apparent that for the parameters we have considered, solar water heater won't be useful as it can provide a maximum temperature of 80°C only.

Assuming the average temperature of water from solar water heater to be 65°C, our calculation show that a water cooler of 10 liter volume using heat from hot water of a solar water heater can work with condenser temperature of 35°C, absorber temperature of 30°C and it will be able be deliver cool water of about 15 °C.

4.2 Parabolic reflectors

A parabolic reflector is a reflective surface used to collect solar energy. Its shape is a part of circular paraboloid, that is, the surface generated by a parabola revolving around its axis.

Figure 4.2 Parabolic reflector

A parabolic trough which is straight in one direction and curved as parabola in the other two, lined with a polished metal mirror can also be used as solar thermal collector.

Figure 4.3 Parabolic trough reflector

Ample amount of research has been done on parabolic solar reflectors and temperatures as high as 200°C are accomplished on average sunny and cloud free days [14]. Thus, parabolic solar reflectors have been the most commonly used solar concentrators for solar refrigeration systems.

In our project, we are using *Fresnel lens* which have been under development since 1960s but are rarely used in solar refrigeration systems.

4.3 Fresnel lens solar reflectors

A Fresnel lens (pronounced *frey-NEL* or *FREZ-nal*) is a type of compact lens consisting of a series of concentric grooves etched into glass or plastic. They are named after Augustin-Jean Fresnel, (1788–1827), the French physicist who pioneered them in the early 19th century. They were originally developed for lighthouses.

While Augustin-Jean Fresnel was not the first to conceptualize a Fresnel lens, he was able to popularize it by integrating it into lighthouses. Since then, Fresnel lenses have been utilized in a variety of applications.

A solar concentrator made by using a Fresnel lens from such a TV can reach temperatures around 800°C [15].

Figure 4.4 Cape Meares (USA) Lighthouse showing Fresnel lens

Figure 4.5 Augustin-Jean Fresnel (1788–1827)

4.3.1 Theory of Fresnel lenses

The driving principle behind the conception of a Fresnel lens is that the direction of propagation of light does not change within a medium (unless scattered). Instead, light rays are only deviated at the surfaces of a medium. As a result, the bulk of the material in the center of a lens serves only to increase the amount of weight and absorption within the system.

To take advantage of this physical property, 18th century physicists began experimenting with the creation of what is known today as a Fresnel lens. At that time, grooves were cut into a piece of glass in order to create annular rings of a curved profile. This curved profile, when extruded, formed a conventional, curved lens – either spherical or aspherical.

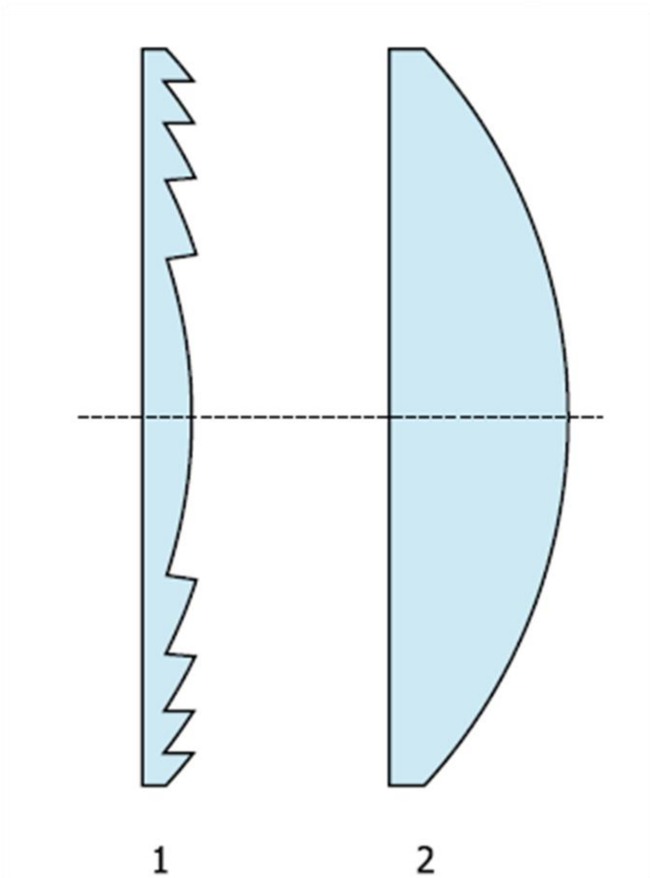


Figure 4.6 (1) Cross section of a spherical Fresnel lens (2) Cross section of a conventional spherical plano-convex lens of equivalent power

The direction of propagation of light does not change within a medium (unless scattered). Instead, light rays are only deviated at the surfaces of a medium.

As a result, the bulk of the material in the center of a lens serves no purpose.

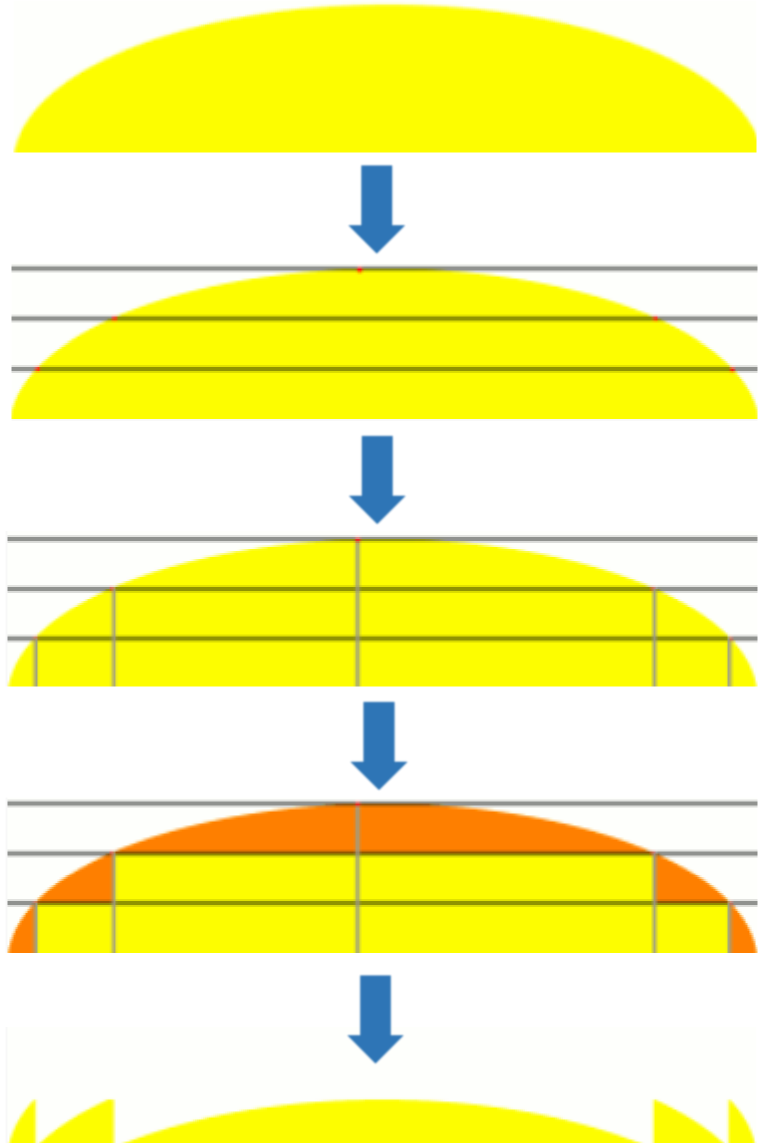


Figure 4.7 Plano-convex lens to Fresnel lens

Figure 4.8 Close-up view of a flat Fresnel lens shows concentric circles on the surface

4.3.2 Description

The Fresnel lens reduces the amount of material required compared to a conventional lens by dividing the lens into a set of concentric annular sections. An ideal Fresnel lens would have infinitely many such sections. In each section, the overall thickness is decreased compared to an equivalent simple lens. This effectively divides the continuous surface of a standard lens into a set of surfaces of the same curvature, with stepwise discontinuities between them.

In some lenses, the curved surfaces are replaced with flat surfaces, with a different angle in each section. Such a lens can be regarded as an array of prisms arranged in a circular fashion, with steeper prisms on the edges, and a flat or slightly convex center. In the first (and largest) Fresnel lenses, each section was actually a separate prism. 'Single-piece' Fresnel lenses were later produced, being used for automobile headlamps, brake, parking, and turn signal lenses, and so on. In modern times, computer controlled milling equipment (CNC) might be used to manufacture more complex lenses.

4.3.3 Types of Fresnel lens

There are two main types of Fresnel lens: imaging and non-imaging. Imaging Fresnel lenses use segments with curved cross-sections while non-imaging lenses have segments with flat cross-sections. They are further classified as follows:

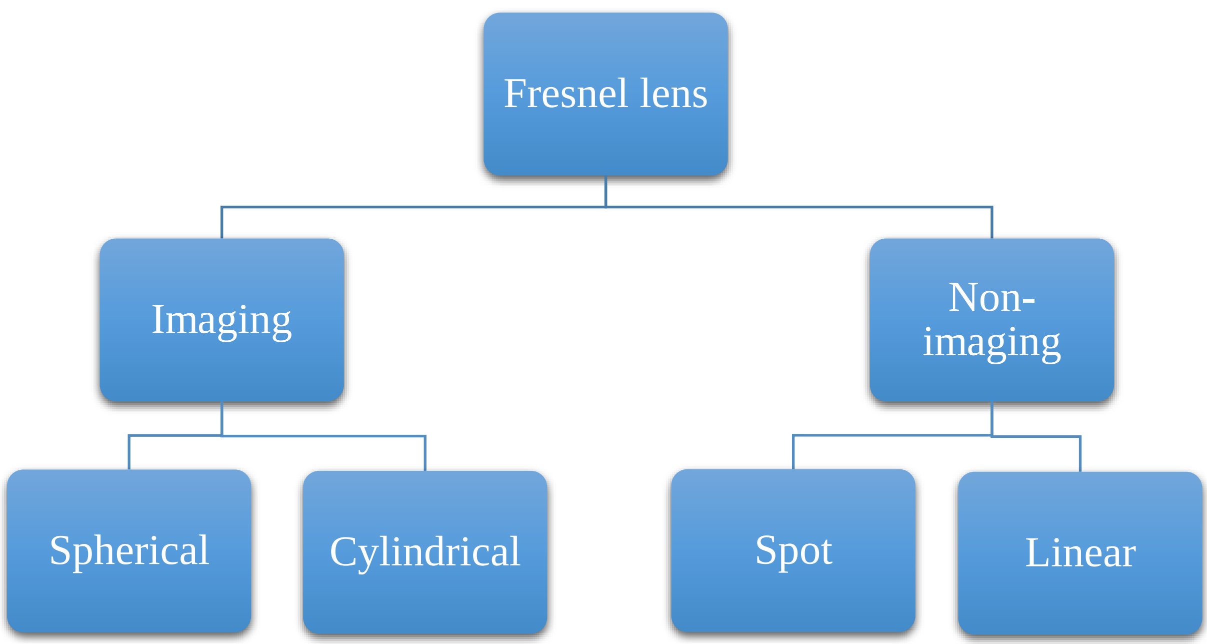


Figure 4.9 Types of Fresnel lens

An *Imaging Spherical* Fresnel lens is equivalent to a simple spherical lens, using ring-shaped segments that are each a portion of a sphere, that all focus light on a single point.

An *Imaging cylindrical* Fresnel lens is equivalent to a simple cylindrical lens, using straight segments with circular cross-section, focusing light on a single line.

A *Non-imaging spot* Fresnel lens uses ring-shaped segments with cross sections that are straight lines rather than circular arcs and focuses light on a spot.

A *Non-imaging linear* Fresnel lens uses straight segments whose cross sections are straight lines rather than arcs. These lenses focus light into a narrow band.

4.3.4 Manufacturing

The first Fresnel lenses were made by tediously grinding and polishing glass by hand. Eventually, molten glass was poured into molds, but it was only with the development of optical-quality plastics and injection-molding technology in the 20th century that the use of Fresnel lenses in many industrial and commercial applications became practical.

Fresnel lenses can be manufactured from a variety of substrates. They are manufactured from acrylic to polycarbonate to vinyl, depending on the desired wavelength of operation. Acrylic is the most common substrate due to its high transmittance in the visible and ultraviolet (UV) regions, but polycarbonate is the substrate of choice in harsh environments due to its resistance to impact and high temperature. However, polymethylmethacrylate (PMMA) which is a light-weight, clear, and stable polymer with optical characteristics nearly same as that of glass, serves as a suitable material for the manufacturing of Fresnel lenses. PMMA is resistant to sunlight, remains thermally stable up to at least 800C, its special transmissivity matches the solar spectrum, and its index of refraction is 1.49, which is very close to that of glass. Consequently, most Fresnel lens designers of concentrated solar energy applications choose PMMA for their lenses because of its high optical quality combined with less costly manufacturing technologies. Modern plastics, new molding techniques, and computer-controlled diamond turning machines have improved the quality of Fresnel lenses and have opened new horizons for the design of Fresnel lenses for solar energy concentration applications

Fresnel lenses can be pressure-molded, injection-molded, cut, or extruded from a variety of plastics and the production costs for large outputs are considerably low.

4.3.5 Acquisition of Fresnel lens

Fresnel lenses are used in Rear-projection TVs (RPTV) of size as large as 100 inch. Rear-projection television is a type of large-screen television display technology which uses a projector to create a small image or video from a video signal and magnify this image onto a viewable screen. The projector uses a bright beam of light and *a lens system* to project the image to a much larger size. The aforementioned lens system essentially uses a Fresnel lens.

Figure 4.10 Rear-projection TV

The advent and evolution of LCD (Liquid Crystal Display), LED (Light-Emitting Diode) and OLED (Organic Light-Emitting Diode) television technologies has made Rear-projection TVs obsolete after latter part of the preceding decade. This fact makes it relatively easy to procure a Rear-projection TV to secure the Fresnel lens used in it.

With a little bit of effort, we were able to get hold of a Fresnel lens from a 52" Philips Rear-projection TV from local market at affordable price.

4.3.6 Fabrication of Solar Concentrator

Before following the design of our solar concentrator, it is indispensable to comprehend relative motion of sun around the earth.

During the course of a year the Earth completes one orbit around the Sun. This is seen on Earth as the Sun moving against the background of stars, around an imaginary circle which we call the *ecliptic*. This defines the plane in which the Earth and most of the other planets orbit around the Sun over a year.

The *celestial equator* is an imaginary projection of the Earth's equator onto the sky. As the Sun moves on its apparent track along the ecliptic, it can be seen for half a year above the equator (northern summer); and for half a year below the equator (northern winter). So the Sun seems to cross the equator twice a year.

At the times when the Sun is crossing the celestial equator, day and night are of nearly equal length at all latitudes. So we call these dates the *equinoxe*s, which means 'equal night'. The times when the Sun is at its furthest from the celestial equator are called the summer and winter *solstice*s. These occur at midsummer and midwinter.

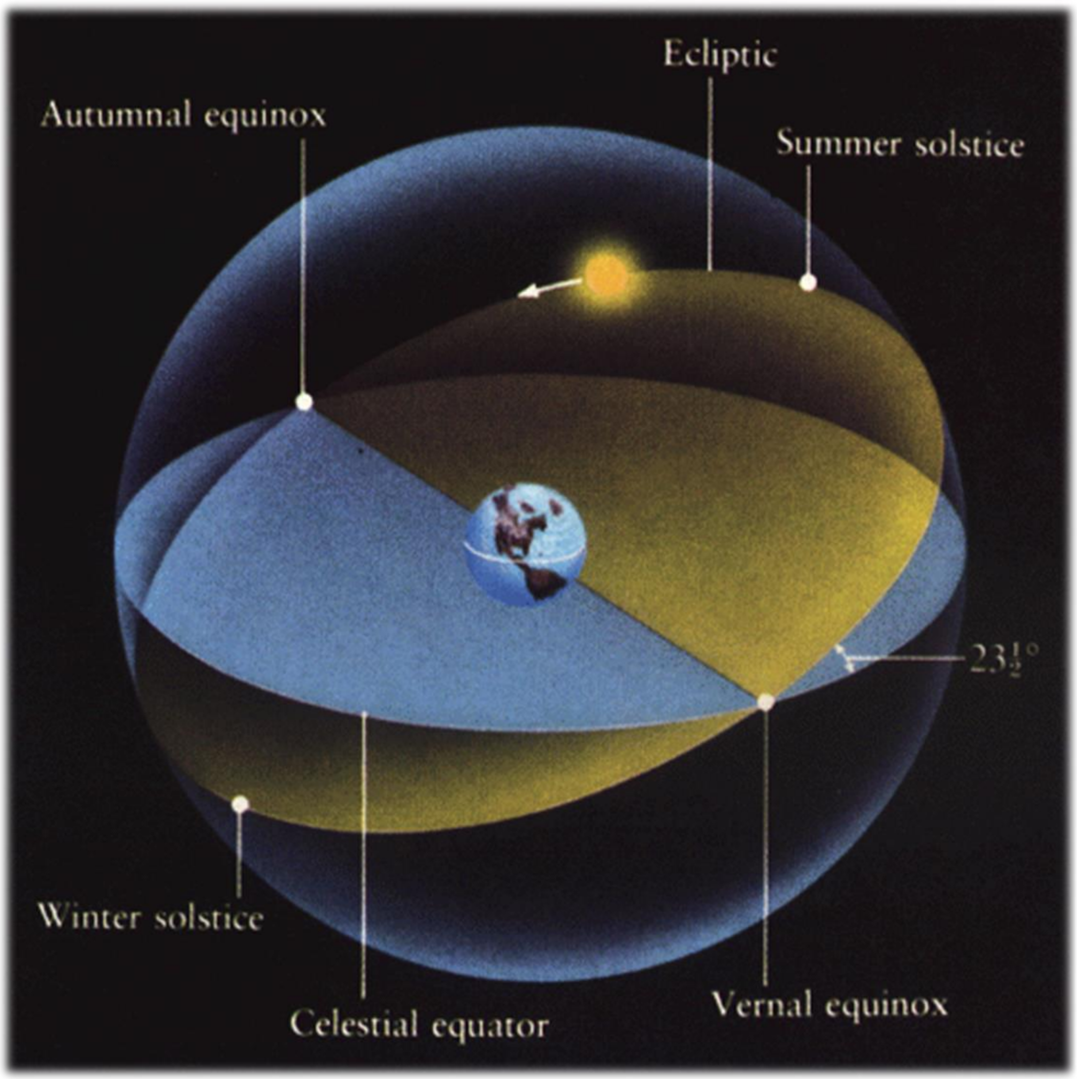


Figure 4.11 Celestial sphere

This complex relative motion between the earth and the sun caused due to earth's axial tilt results in shifting of day-to-day locus of sun along the year. The Figure 4.12 shows locus of sun in northern hemisphere on different dates.

The designed solar concentrator is able to adapt to these changes in the perceived path of the sun.

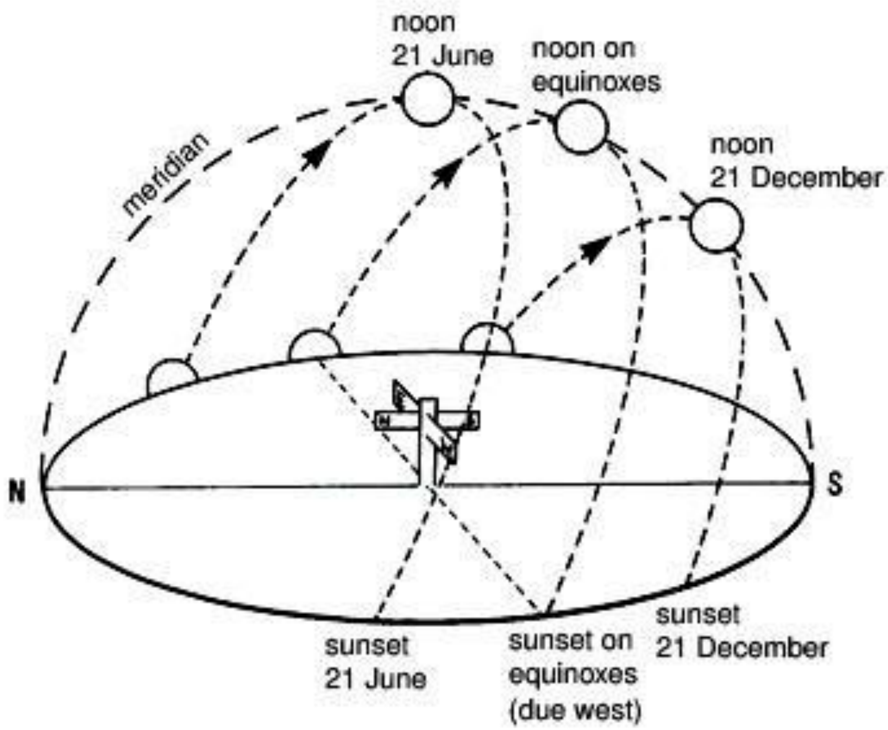


Figure 4.12 Locus of sun in northern hemisphere

We have employed a two-frame design for the solar concentrator as shown in the following figure.

The frame and the supporting structure are made from deodar cedar wood which provides good strength as well as low weight. The inner frame is attached to the outer frame using a dowel pin and a ball bearing and so is outer frame to the structure.

The rotation of the outer frame about axis number 1 in Figure 4.13 is used to realign the lens according to the change in position of the sun throughout one day.

The rotation of the inner frame about axis number 2 is used to realign the lens according to the change in locus of sun as shown in Figure 4.12.

To hold the inner and the outer frame in required position, we have used a simple locking mechanism as shown in Figure 4.14.

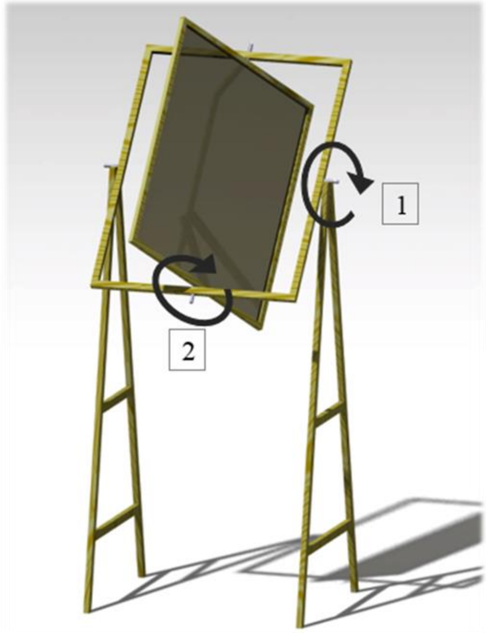


Figure 4.13 Fresnel lens solar concentrator

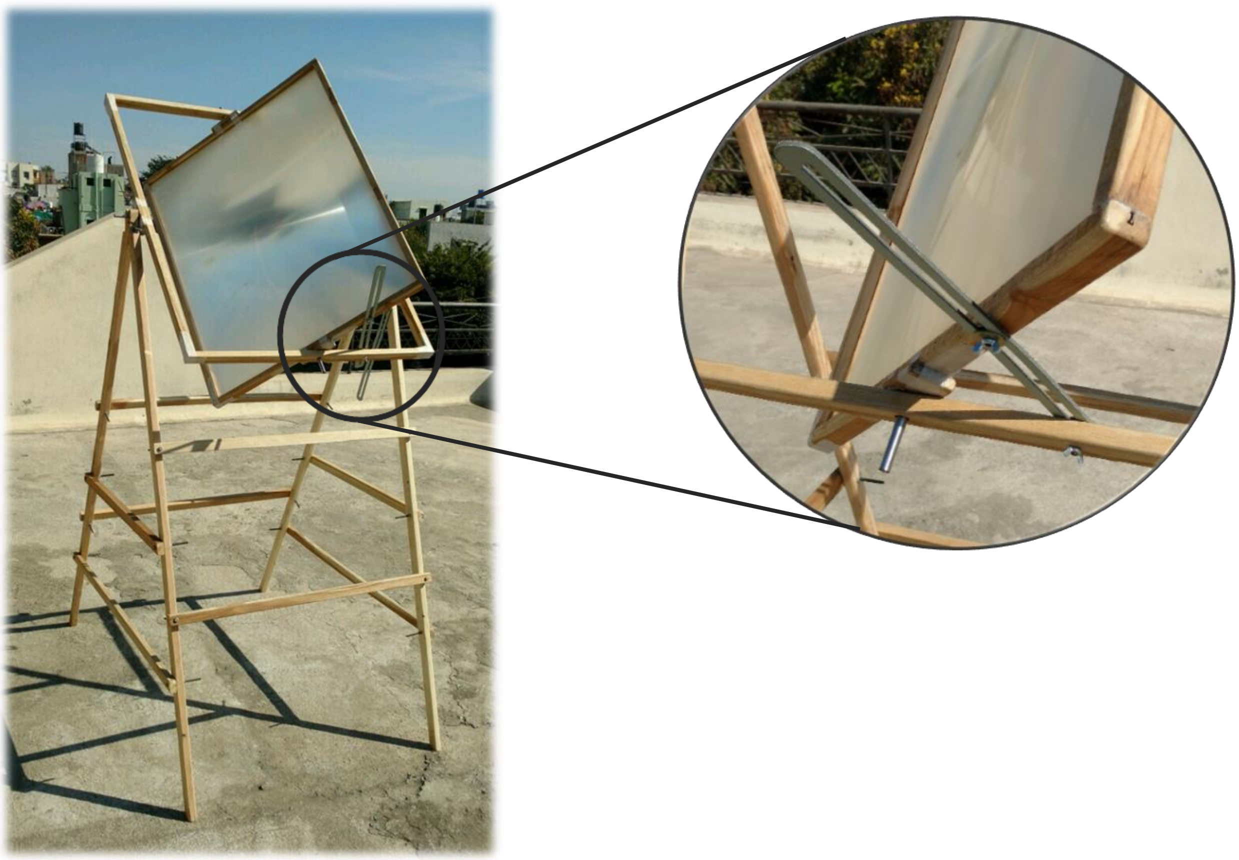


Figure 4.14 Actual solar concentrator with enlarged view of locking mechanism

After the fabrication of the solar concentrator, we carried out a series of test runs by keeping a water filled container of known volume with an intention to calculate amount of solar radiation collected. This amount in Watt which is summarized below is nothing but the heat input to the generator.

Sr. No.	Date	Temperature rise (Celsius)	Time (Second)	Power (Watt)
1	11/02/2017	32.0	1800	218.9
2	19/02/2017	34.0	1800	226.3
3	27/02/2017	35.5	1800	231.9
4	02/03/2017	37.0	1800	237.5
5	14/03/2017	40.0	1800	248.6

Average Power	**232.6 Watt**

Table 4.1 Results of test runs of solar concentrator

In the next chapter we will see the design and fabrication of the evaporator and the condenser.

Chapter-5

Condenser and Evaporator

5.1 Introduction

Condensers and evaporators are basically heat exchangers in which the refrigerant undergoes a phase change. Proper design and selection of condensers and evaporators is very important for satisfactory performance of any refrigeration system. Since both condensers and evaporators are essentially heat exchangers, they have many things in common as far as the design of these components is concerned. However, differences exist in the case of heat transfer. In condenser, the refrigerant vapor condenses by rejecting heat to an external fluid, which acts as a heat sink. Normally, the external fluid does not undergo any phase change. In evaporator, the liquid refrigerant evaporates by extracting heat from an external fluid (low temperature heat source). The external fluid may not undergo phase change.

5.2 Design of condenser

As mentioned earlier, the refrigerant enters the condenser and then is condensed by rejecting heat to an external medium. While designing condenser for this application, it is assumed that nearly saturated ammonia vapor would enter in the condenser. And indeed it would be nearly saturated and not super-heated because there is limited heat input in heater. The refrigerant may leave the condenser as a saturated or a sub-cooled liquid, depending upon the temperature of the external medium and design of the condenser. In this case we designed condenser in such way that there would only be saturated liquid at the end of condenser. Taking these aspects into consideration we design the water cooled condenser.

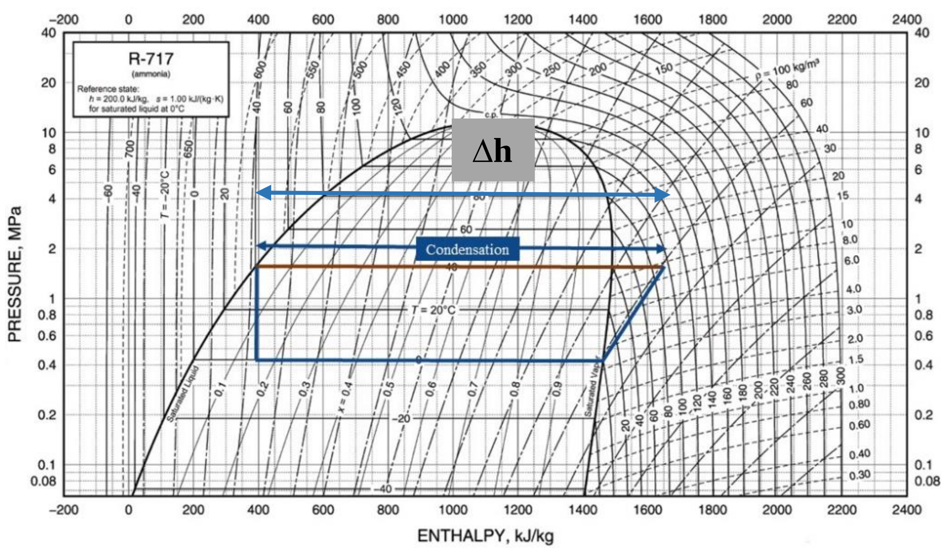


Figure 5.1 Condensation process on P-h chart

The amount of heat to be rejected by the refrigerant to an external medium can be calculated from the enthalpy change for condensation from figure 5.1 and mass flow rate of ammonia.

$\dot{Q}_c = \dot{m}_r \times \Delta h$

$\dot{Q}_c = (2.278 \times 10^{-5}) \times 1250$

$\dot{Q}_c = 28.47$ W

The condenser we are using is a simple shell and tube type condenser with refrigerant flowing through the tubes (Stainless Steel 304, 7.5mm inner radius, 10mm outer radius) and still water in a cuboid shaped shell. Therefore the heat gained by the still water can be represented as follows:

$$\dot{Q}_c = h \times A \times \Delta T \qquad \text{Eqn. (5.1)}$$

Where, h = Convective heat transfer coefficient, $\frac{W}{m^2.k}$

A= Surface area of tube, m^2

ΔT = Temperature rise of water, ℃

5.2.1 Calculation of convective heat transfer coefficient

The convection heat transfer coefficient h is not a property of the fluid. It is an experimentally determined parameter whose value depends on all the variables influencing convection such as the surface geometry, the nature of fluid motion, the properties of the fluid, and the bulk fluid velocity. Various empirical relations have been derived by the scientists to calculate the convective heat transfer coefficient.

Heat transfer coefficient is given by:

$$h = \frac{Nu \times k}{L_c} \qquad \text{Eqn. (5.2)}$$

Where, Nu = Nusselt number

k = Thermal conductivity of water, $\frac{W}{m.k}$

L_c = Characteristic length of tube (In this case, Diameter), m

The tubes are to be positioned in horizontal plane in the cuboid container. Therefore, for the given condition of a horizontal cylindrical tubes, Nusselt number is given as:

$$Nu = \left\{ 0.6 + \frac{0.387 Ra_D^{1/6}}{\left[1 + \left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}} \right\}^2$$ [16] Eqn. (5.3)

Where, Pr = Prandtl number

Ra = Rayleigh number

= Grashof number × Prandtl number

And Rayleigh number is given by:

$$Ra = \frac{g \times \beta \times \Delta T \times L_c{}^3}{\vartheta^2} \times Pr$$ Eqn. (5.4)

Where, g = Gravitational acceleration, $\frac{m}{s^2}$

= 9.8 $\frac{m}{s^2}$

β = Coefficient of volume expansion of fluid (here water), $\frac{1}{K}$

= 3.03×10^{-4} $\frac{1}{K}$ at 30°C

ΔT = Temperature difference between tube surface and water

= 10°C

L_c = Characteristic length of tube (Diameter), m

=10×10^{-3} m

ϑ = Kinematic viscosity of the fluid, $\frac{m^2}{s}$

= 8.01×10^{-6}

Pr = Prandtl number

= 5.43

And k_{water} = 0.6154 $\frac{W}{m.k}$

By putting these values in Eq. (5.4), we get:

$$Ra = \frac{9.8 \times 3.03 \times 10^{-4} \times 10 \times (10^{-2})^3}{(8.01 \times 10^{-6})^2} \times 5.43$$

$$Ra = 2.5 \times 10^5 \qquad \text{(a)}$$

By putting this value of Rayleigh number in Eqn. (5.3), we get:

$$Nu = \left\{ 0.6 + \frac{0.387(2.5 \times 10^5)^{1/6}}{\left[1 + \left(\frac{0.559}{5.43}\right)^{9/16}\right]^{8/27}} \right\}^2$$

$$Nu = 11.943 \qquad \text{(b)}$$

By putting this value of Nusselt number, L_c and k_{water} in Eqn. (5.2), we get:

$$h = \frac{11.943 \times 0.6154}{10^{-2}}$$

$$h = 735.019 \frac{W}{m^2.k} \qquad \text{(c)}$$

5.2.2 Calculation of length of condenser tube

From Eqn. (5.1) and above value of convective heat transfer coefficient, we get:

$$L_{condender} = \frac{Qc}{h \times \pi \times D \times \Delta T}$$

$$L_{condender} = \frac{28.47}{735.019 \times \pi \times 10^{-2} \times 5}$$

$$L_{condenser} = 0.2466 \text{ m}$$

This means that a 0.2466m long tube immersed in water will be sufficient to reject the required amount of heat to condense the ammonia refrigerant.

The cuboid shaped container may lose heat to the atmosphere thereby reducing the efficiency of the condenser. Consequentially, it is necessary to provide insulation to the condenser. We have used polypropylene plastic container as a condenser since thermal conductivity of polypropylene plastic is much less than most metals. Another advantage of polypropylene container is that it is relatively cheap and easily available in a range of varying dimensions in local market.

5.2.3 Calculation of dimensions of condenser

The plane of condenser coil immersed in water is horizontal. Though the cooling water is not circulated by an external agency like pump, the heat transfer taking place in the enclosed space of condenser is complicated by the fact that the fluid in the enclosure, in general, does not remain stationary. In an enclosure, the fluid adjacent to the hotter surface rises and the fluid adjacent to the cooler one falls, setting off a rotational motion within the enclosure that enhances heat transfer through the enclosure. Typical flow pattern in horizontal rectangular enclosures is shown in following figure.

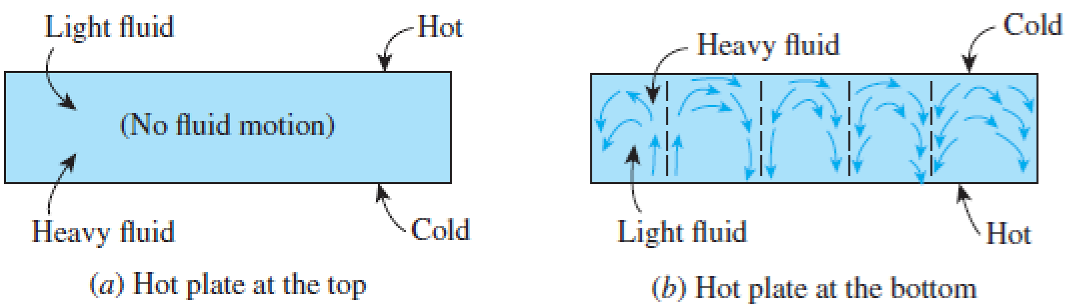
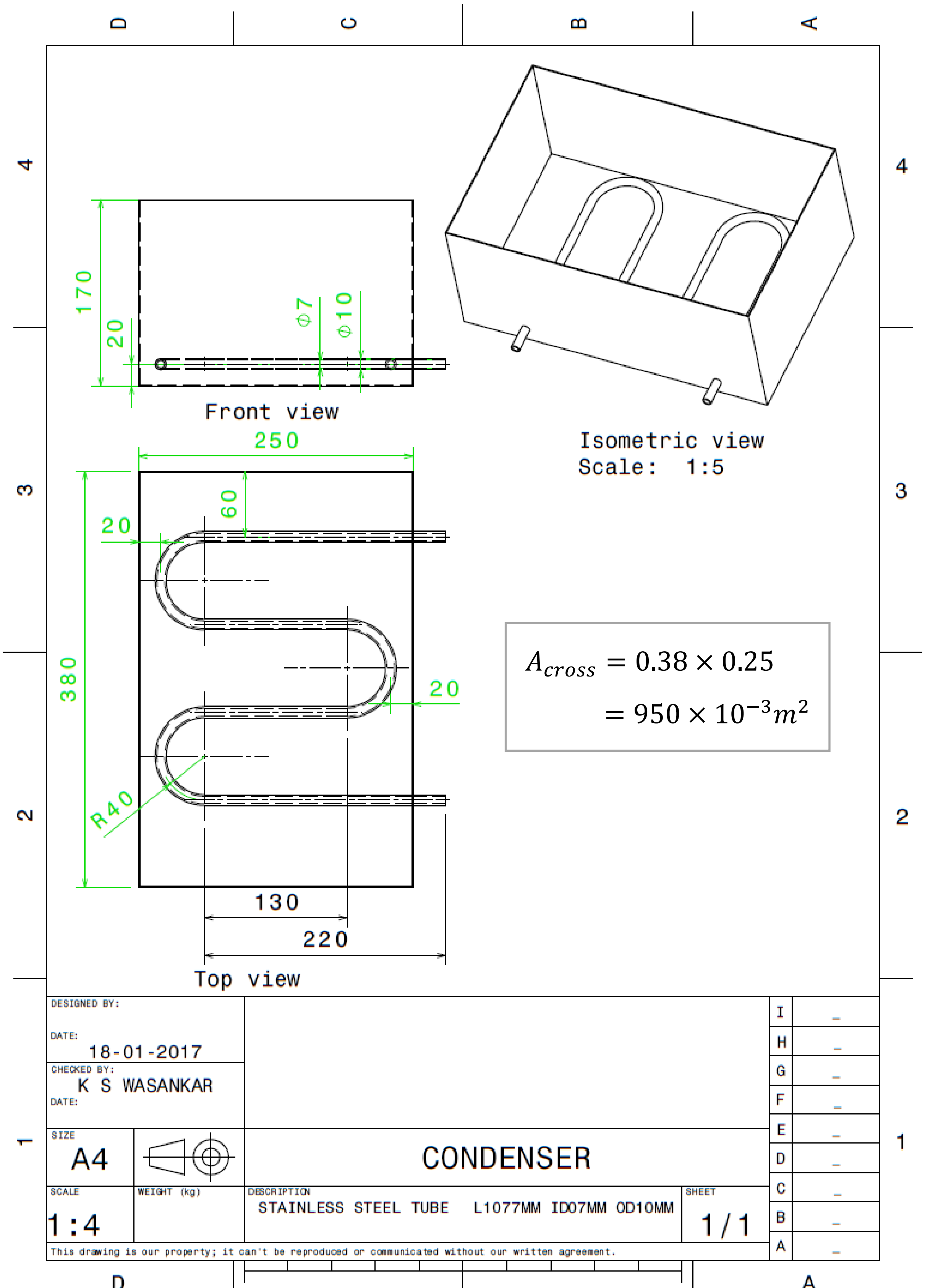


Figure 5.2 Convective currents in a horizontal enclosure

The characteristics of heat transfer through a horizontal enclosure depend on whether the hotter plate is at the top or at the bottom, as shown in above figure. When the hotter plate is at the top, no convection currents develop in the enclosure, since the lighter fluid is always on top of the heavier fluid. Heat transfer in this case is by pure conduction, and we have $Nu = 1$. When the hotter plate is at the bottom, the heavier fluid will be on top of the lighter fluid, and there will be a tendency for the lighter fluid to topple the heavier fluid and rise to the top, where it comes in contact with the cooler plate and cools down. Until that happens, however, heat transfer is still by pure conduction and Nu = 1. When $Ra_L > 1708$, the buoyant force overcomes the fluid resistance and initiates natural convection currents. The value of *Ra* (from (a)) for the particular conditions of our condenser is much higher than 1708, hence convection currents will unquestionably take place.

Since the refrigerant in the tube is hotter than the surrounding cooling water and by considering the plane of condenser coil as a hot surface, from above discussion it is clear that *the condenser coils are to be placed at as much depth as possible in the condenser container*. The same is achieved in our design which is shown in Figure 5.3.

Previously it has been commented that convective fluid currents are established in enclosures with two surfaces with different temperatures. The heat transfer taking place due to this natural convection between water layers should be greater than or equal to the heat transfer taking place in between condenser coil and water.



$$A_{cross} = 0.38 \times 0.25$$
$$= 950 \times 10^{-3} m^2$$

Figure 5.3 Design of condenser

When the Nusselt number is known, the rate of heat transfer through the enclosure is determined from:

$$\dot{Q} = h \times A_s \times \Delta T$$

$$= \frac{k \times Nu \times As \times \Delta T}{L_c} \qquad [17] \qquad \text{Eqn. (5.5)}$$

Globe and Dropkin (1959) obtained following empirical correlation for horizontal enclosures:

$$Nu = 0.069 Ra^{1/3} \times Pr^{0.074} \qquad [18] \qquad \text{Eqn. (5.6)}$$

We know that Rayleigh number is given by:

$$Ra = \frac{g \times \beta \times \Delta T \times {L_c}^3}{\vartheta^2} \times Pr$$

Above properties of water are to be calculated at average temperature. Thus, we have, at 35°C:

$$\beta = 3.45 \times 10^{-4} \frac{1}{K}$$

$$\vartheta = 5.865 \times 10^{-7}, \frac{m^2}{s}$$

$$Pr = 4.885$$

$$k_{water} = 0.623 \frac{W}{m.k}$$

By putting above values in equation of Rayleigh number, we get:

$$Ra = 4.80 \times 10^{11} \times L_c^3$$

Thus, from Eqn. (5.6), we get:

$$Nu = 602.37 \times L_c$$

As mentioned before, the heat transfer taking place due to this natural convection between water layers should be greater than or equal to the heat transfer taking place in between condenser coil and water. Thus, from above value of Nusselt number and Eqn. (5.5), we have:

$$Q_c = \frac{0.623 \times 602.37 \times L_C \times A_S \times 10}{L_C}$$

Thus, we have:

$$A_s = \frac{28.47}{3752.76}$$

$$A_s = 7.586 \times 10^{-3}\ \text{m}^2$$

This suggests that as the cross-sectional area of the condenser is 950×10^{-3} m^2 (from Figure 5.3) is greater than 7.586×10^{-3} m^2, heat transfer will take place indubitably.

5.2.4 Heat transfer through tube walls due to conduction

It is necessary that the quantity of heat to be rejected to the heat sink is conducted through the tube walls. Conduction heat transfer through a long cylindrical pipe is given by:

$$Q_{conduction} = \frac{2\pi \times L \times k \times \Delta T}{\ln\left(\frac{r_2}{r_1}\right)} \qquad [19] \qquad \text{Eqn. (5.7)}$$

Where,

L = Length of the pipe, m

= 0.2466m

k = Thermal conductivity of pipe material, $\frac{W}{m.k}$

= 16.3 $\frac{W}{m.k}$

r_2 = Outer radius

= 10mm

r_1 = Inner radius

= 7mm

By putting above values in Eqn. (5.7), we get:

$$Q_{conduction} = \frac{2\pi \times 0.2466 \times 16.3 \times 10}{\ln\left(\frac{10}{7}\right)}$$

= 708.08 W

It means that the pipe we have used is capable of conducting 708.08 W let alone the required 28.47 W.

5.3 Design of evaporator

The refrigerant leaves the condenser as saturated liquid. The temperature of this ammonia liquid is near atmospheric temperature. For refrigeration effect to take place in the evaporator, the temperature of the refrigerant must be less than the intended temperature to be achieved which in our case is 5℃. This temperature drop of the refrigerant from 40℃ to 0℃ is achieved in the expansion device. The selection and design of expansion device is discussed in detail in next chapter.

Similar to the condenser, the evaporator designed by us is shell and tube type. The evaporation process is shown in the following figure.

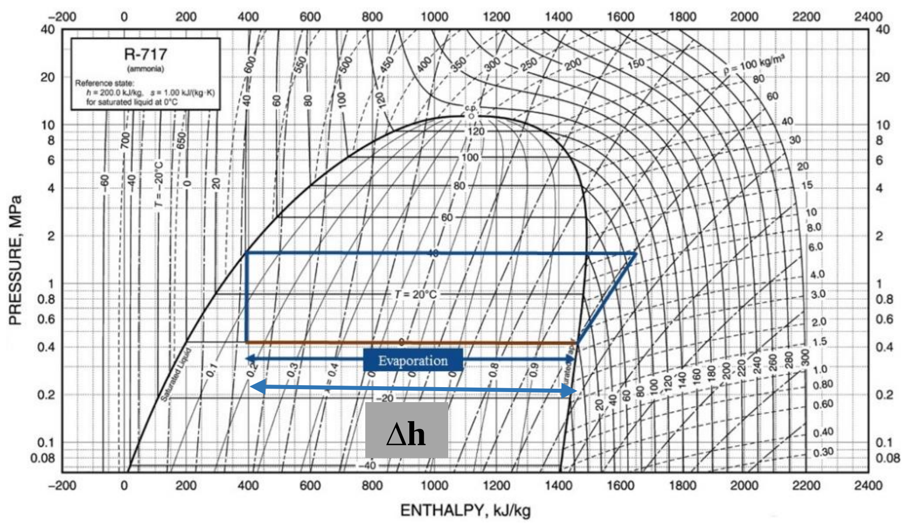


Figure 5.4 Evaporation process on P-h chart

The amount of heat to be absorbed by the refrigerant from the drinking water can be calculated from the enthalpy change for evaporation from figure 5.4 and mass flow rate of ammonia.

$\dot{Q}_e = \dot{m}_r \times \Delta h$

$\dot{Q}_e = (2.278 \times 10^{-5}) \times 1071.60$

$\dot{Q}_e = 24.18$ W

As mentioned earlier, the evaporator we are using is a simple shell and tube type condenser with refrigerant flowing through the tubes (which has same specification as that of the condenser tube) and still water in a cuboid shaped shell. Therefore the heat gained by the refrigerant can be represented as follows:

$$\dot{Q}_e = h \times A \times \Delta T \qquad \text{Eqn. (5.8)}$$

Where, h = Convective heat transfer coefficient, $\frac{W}{m^2.k}$

A= Surface area of tube, m^2

ΔT = Temperature rise of water, ℃

5.3.1 Calculation of convective heat transfer coefficient

Analogous to the discussion in the condenser design, we know that heat transfer coefficient is given by:

$$h = \frac{Nu \times k}{L_c} \qquad \text{Eqn. (5.9)}$$

Where, Nu = Nusselt number

k = Thermal conductivity of water, $\frac{W}{m.k}$

L_c = Characteristic length of tube (In this case, Diameter), m

The tubes are to be positioned in horizontal plane in the cuboid container. Therefore, for the given condition of a horizontal cylindrical tubes, Nusselt number is given as:

$$Nu = \left\{0.6 + \frac{0.387 {Ra_D}^{1/6}}{\left[1 + \left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2 \qquad \text{Eqn. (5.10)}$$

Where, Pr = Prandtl number

Ra = Rayleigh number

= Grashof number × Prandtl number

And Rayleigh number is given by:

$$Ra = \frac{g \times \beta \times \Delta T \times {L_c}^3}{\vartheta^2} \times Pr \qquad \text{Eqn. (5.11)}$$

Where, g = Gravitational acceleration, $\frac{m}{s^2}$

$= 9.8 \frac{m}{s^2}$

β = Coefficient of volume expansion of fluid (here water), $\frac{1}{K}$

= $1.6\times10^{-5}\frac{1}{K}$ at 5℃

ΔT = Temperature difference between tube surface and water

= 5℃

L_c = Characteristic length of tube (Diameter), m

$=10\times10^{-3}\,m$

ϑ = Kinematic viscosity of the fluid, $\frac{m^2}{s}$

= 1.79×10^{-6}

Pr = Prandtl number

= 11.57

And k_{water} = $0.579\ \frac{W}{m.k}$

By putting these values in Eq. (5.11), we get:

$$Ra = \frac{9.8\times1.6\times10^{-5}\ \times5\times(10^{-2})^3}{(1.79\times10^{-6})^2}\times11.57$$

$$Ra = 2824.707 \tag{d}$$

By putting this value of Rayleigh number in Eqn. (5.10), we get:

$$Nu = \left\{0.6+\frac{0.387(2824.707)^{1/6}}{\left[1+\left(\frac{0.559}{11.57}\right)^{9/16}\right]^{8/27}}\right\}^2$$

$$Nu = 3.9389 \tag{e}$$

By putting this value of Nusselt number, L_c and k_{water} in Eqn. (5.9), we get:

$$h = \frac{3.938\times0.579}{10^{-2}} = 228.063\ \frac{W}{m^2.k} \tag{f}$$

5.3.2 Calculation of length of evaporator tube

From Eqn. (5.8) and above value of convective heat transfer coefficient, we get:

$$L_{evaporator} = \frac{Q_e}{h \times \pi \times D \times \Delta T}$$

$$L_{evaporator} = \frac{24.40}{228.063 \times \pi \times 10^{-2} \times 5}$$

$$L_{evaporator} = 0.6699 \text{ m}$$

This means that a 0.6699m long tube immersed in water will be sufficient to absorb the required amount of heat to cool the intended 5 liter drinking water.

Based on the same logic employed in selection of material of condenser, we have used polypropylene plastic container as an evaporator.

5.3.3 Calculation of dimensions of evaporator

We reminisce from our discussion of calculation of condenser dimensions (Section 5.2.3) that in an enclosure, the fluid adjacent to the hotter surface rises and the fluid adjacent to the cooler one falls, setting off a rotational motion within the enclosure that enhances heat transfer through the enclosure.

Since the refrigerant in the tube is cooler than the surrounding water which to be cooled and by considering the plane of evaporator coil as a cold surface, we can observe that *the evaporator coils are to be placed at as much height as possible in the evaporator container*. The same is achieved in our design which is shown in Figure 5.5.

We know that convective fluid currents are established in enclosures with two surfaces with different temperatures. As discussed previously the heat transfer taking place due to this natural convection between water layers should be greater than or equal to the heat transfer taking place in between evaporator coil and water.

When the Nusselt number is known, the rate of heat transfer through the enclosure is determined from:

$$\dot{Q} = h \times A_s \times \Delta T$$

$$= \frac{k \times Nu \times A_s \times \Delta T}{L_c} \quad \text{Eqn. (5.12)}$$

Where, $$Nu = 0.069 Ra^{1/3} \times Pr^{0.074} \quad \text{Eqn. (5.13)}$$

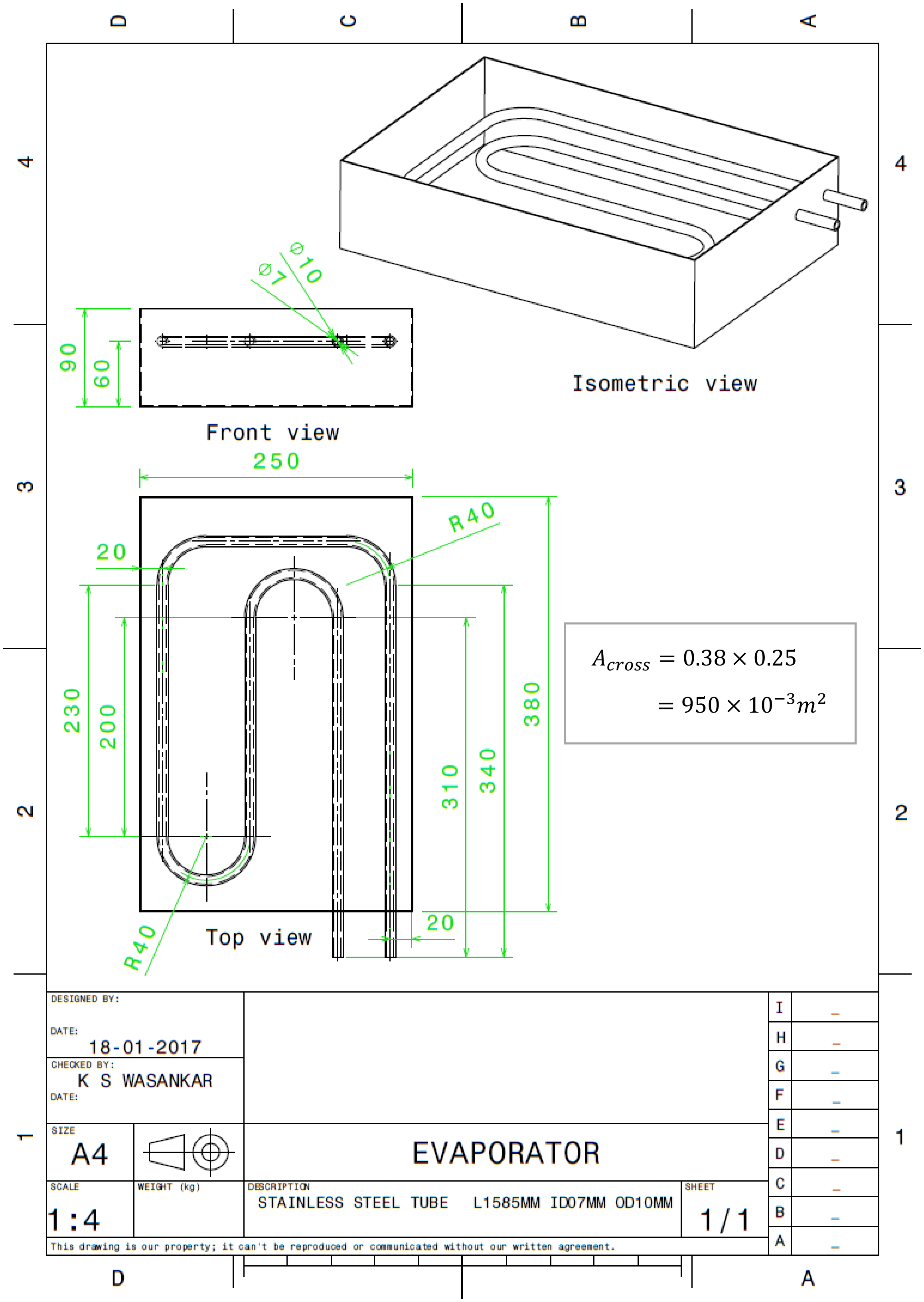


Figure 5.5 Design of evaporator

We know that Rayleigh number is given by:

$$Ra = \frac{g\times\beta\times\Delta T\times{L_C}^3}{\vartheta^2}\times Pr$$

Where, β = 1.6×10^{-5} $\frac{1}{K}$

ϑ = 1.79×10^{-6}, $\frac{m^2}{s}$

Pr = 11.57

k_{water} = 0.579 $\frac{W}{m.k}$

By putting above values in equation of Rayleigh number, we get:

$$Ra = 2.84\times10^9\times L_c^3$$

Thus, from Eqn. (5.13), we get:

$$Nu = 108.84 \times L_c$$

As mentioned before, the heat transfer taking place due to this natural convection between water layers should be greater than or equal to the heat transfer taking place in between evaporator coil and water. Thus, from above value of Nusselt number and Eqn. (5.12), we have:

$$Q_e = \frac{0.579\times108.84\times L_C\times A_S\times5}{L_C}$$

Thus, we have:

$$A_s = \frac{24.40}{315.09}$$

$$A_s = 7.744 \times 10^{-2}\ \text{m}^2$$

This suggests that as the cross-sectional area of the condenser is 95×10^{-2} m^2 (from Figure 5.5) is greater than 7.744×10^{-2} m^2, heat transfer will take place indubitably.

Chapter-6

Expansion Device and Generator

6.1 Introduction

An expansion device is another basic component of a refrigeration system. The basic functions of an expansion device used in refrigeration systems are to:

1. Reduce pressure from condenser pressure to evaporator pressure
2. Regulate the refrigerant flow from the high-pressure liquid line into the evaporator at a rate equal to the evaporation rate in the evaporator.

The expansion devices used in refrigeration systems can be divided into fixed opening type or variable opening type. As the name implies, in fixed opening type the flow area remains fixed, while in variable opening type the flow area changes with changing mass flow rates. There are basically seven types of refrigerant expansion devices. These are:

1. Hand Expansion Valve
2. Capillary Tube
3. Orifice
4. Automatic Expansion Valve
5. Thermostatic Expansion Valve
6. Float type Expansion Valve
7. Electronic Expansion Valve

Out of these types, Capillary tube and orifice belong to the fixed opening type, while the rest belong to the variable opening type. Application of these expansion devices changes according to a load on the refrigeration system, nature of a load (constant or variable), and required pressure drop cross the device.

In case of solar powered water cooler, load on cooler is kept moderately low and constant in addition to this the pressure difference is not so high. Capillary tube is also economical to use and it doesn't have any moving part. Taking all these aspect into consideration, we choose capillary tube as expansion device for our solar powered water cooler. We have taken capillary of Stainless Steel as ammonia in an aqua ammonia solution reacts with copper.

6.2 Capillary tube

A capillary tube is a long, narrow tube of constant diameter. Typical tube diameters of refrigerant capillary tubes range from 0.5 mm to 3 mm and the length ranges from 1.0 m to 6 m. The pressure reduction in a capillary tube occurs due to the following two factors:

1. The refrigerant has to overcome the frictional resistance offered by tube walls. This leads to some pressure drop.

2. The liquid refrigerant flashes (evaporates) into mixture of liquid and vapor as its pressure reduces. The density of vapors is less than that of the liquid. Hence, the average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter (hence area) being constant, the velocity of refrigerant increases since, $m = \rho VA$. The increase in velocity or acceleration of the refrigerant also requires pressure drop.

6.2.1 Design and selection of capillary tube

There are analytical and graphical methods to select the capillary tube. The fine-tuning of the length is finally done by *cut-and-try* method. A tube longer than the design (calculated) value is installed with the expected result that evaporating temperature will be lower than expected. The tube is shortened until the desired balance point is achieved. If a single system is to be designed then tube of slightly shorter length than the design length is chosen. The tube will usually result in higher temperature than the design value. The tube is pinched at a few spots to obtain the required pressure and temperature.

As mentioned above, pressure drop in capillary tube is due to two causes: frictional pressure drop and momentum pressure drop. The cumulative pressure drop must be equal to the difference in pressure at the two ends of the tube. The mass flow through the capillary tube will adjust according to the pressure drop through the tube which is difference between evaporator and condenser pressure. For given state, pressure drop is directly proportional to the length and inversely proportional to the bore diameter of the tube.

A number of different combinations of length and diameter are possible for capillary tube. However, once a capillary tube has been selected, it will be suitable only for the design pressure drop and flow. It cannot be used for varying condenser and evaporator pressures.

While making the selection, a capillary tube of given bore D is selected. Step decrements in pressure are then assumed and the corresponding required increments of length are calculated. These increments are then totaled and complete length of capillary tube can be calculated for a given pressure drop. Method used for this selection is as described below.

Flow in the capillary tube is actually compressible, three-dimensional and two-phase flow with heat transfer and thermodynamic meta-stable state at the inlet of the tube. However, it is assumed to be steady, one-dimensional and in single phase or a homogenous mixture. One dimensional flow means that the velocity does not change in the radial direction of the tube. Homogeneous means annular flow or plug flow model etc. are not considered for the two-phase flow. Figure below shows a small section of a vertical capillary tube with momentum and pressure at two ends of an elemental control volume.

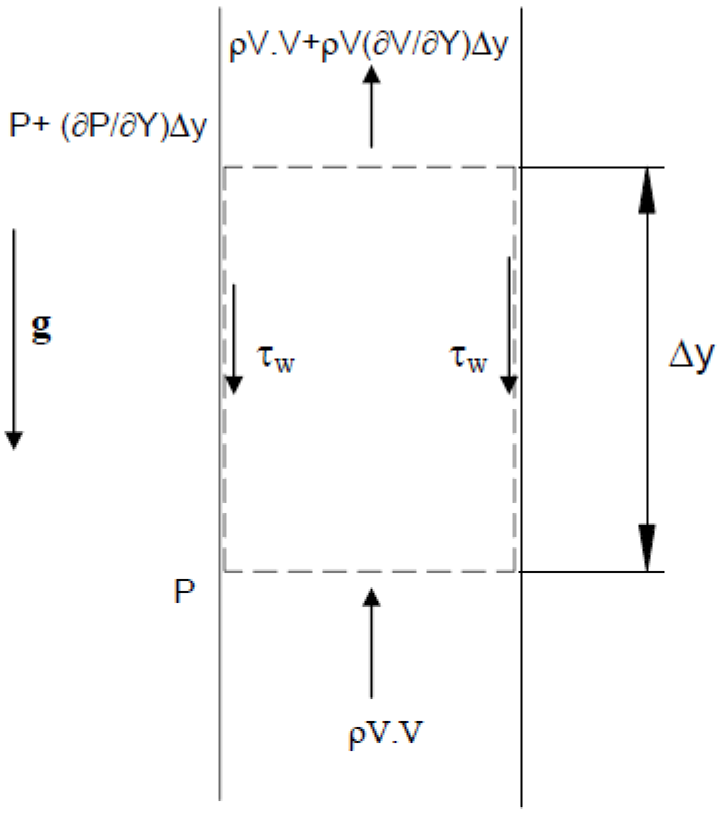


Figure 6.1 A small section of a capillary tube considered for analysis

From mass conservation, we have:

$$\rho VA + \frac{\partial(\rho V)}{\partial y}\Delta yA - \rho VA = 0 \qquad \text{Eqn. (6.1)}$$

$$\frac{\partial(\rho V)}{\partial y} = 0$$

From momentum conservation, we have:

The momentum theorem is applied to the control volume. According to this:

[Momentum] $_{out}$ – [Momentum] $_{in}$ = Total forces on control volume

$$\pi R^2\left(\rho VV + \rho V\frac{\partial V}{\partial y}\Delta y\right) - \pi R^2(\rho VV) = -\pi R^2\frac{\partial p}{\partial y}\Delta \mathrm{y} - \rho_{avg}\mathrm{g}\pi R^2\Delta y - 2\pi R\Delta y\tau_w$$

Eqn. (6.2)

At the face y+Δy, Taylor series expansion has been used for pressure and momentum and only the first order terms have been retained. The second order terms with second derivatives and higher order terms have been neglected. If the above equation is divided by $\pi R^2 \Delta y$ and limit $\Delta y \rightarrow 0$ is taken; then all the higher order terms will tend to zero.

Also, ρ_{avg} will tend to ρ since the control volume will shrink to the bottom face of the control volume where ρ is defined. Further, neglecting the effect of gravity, which is very small, we obtain:

$$\rho V \frac{\partial V}{\partial y} = -\frac{\partial p}{\partial y} - 2\frac{\tau_w}{R} \qquad \text{Eqn. (6.3)}$$

The wall shear stress may be written in terms of friction factor. In fluid flow through pipes the pressure decreases due to shear stress. This will be referred to as frictional pressure drop and a subscript 'f' will be used with it and it will be written in terms of friction factor. The Darcy's friction factor is for fully developed flow in a pipe. In fully developed flow the velocity does not change in the flow direction. In case of a capillary tube it increases along the length. Still it is good approximation to approximate the shear stress term by friction factor. For fully developed flow the left hand side of above equation is zero, hence the wall shear stress may be obtained from the following equation:

$$\tau_w = R\frac{\Delta p_f}{2\Delta y} \qquad \text{Eqn. (6.4)}$$

The frictional pressure drop is defined as:

$$\Delta p_f = \rho f \frac{\Delta L}{D}\frac{V^2}{2} \qquad \text{Eqn. (6.5)}$$

Substituting Eqn. (6.5) in Eqn. (6.4), we get:

$$\tau_w = \rho f \frac{V^2}{8} \qquad \text{Eqn. (6.6)}$$

Substituting for τ_w in Eqn. (6.3) we have:

$$\rho V \frac{\partial V}{\partial y} = -\frac{\partial p}{\partial y} - 2\frac{\rho f V^2}{R} \qquad \text{Eqn. (6.7)}$$

Mass conservation Eqn. (6.1) indicates that the product ρV is constant in the tube. In fact it is called mass velocity and is denoted by G which is written as:

$$G = \rho V$$

We have mass flow rate,

$$\dot{m} = \left[\frac{\pi D^2}{4}\right]\rho V$$

$$\therefore \rho V = \frac{m}{A}$$

$$= G \qquad \text{Eqn. (6.8)}$$

Hence Eqn. (6.7) is rewritten as follows:

$$\rho V \frac{\partial V}{\partial y} = -\frac{\partial p}{\partial y} - 2\frac{FvG}{R} \qquad \text{Eqn. (6.9)}$$

In this equation the term on the left hand side is the acceleration of fluid. The first term on the right hand side is the pressure drop required to accelerate the fluid and to overcome the frictional resistance. The second term on the right hand side is the frictional force acting on the tube wall. The friction factor depends upon the flow Reynolds number and the wall roughness for the fully developed flow. For the developing flow it is function of distance along the tube also in addition to Reynolds number. The flow accelerates along the tube due to vapor formation, as a result, the Reynolds number increases along the tube. The velocity and Reynolds number vary in a complex manner along the tube and these are coupled together. Hence, an exact solution of Eqn. (6.9) is not possible. To a good approximation the integral of product f V, that is, $\int f$Vdy can be calculated by assuming average value of the product fV over a small length ΔL of the capillary tube.

Accordingly, integrating Equation (6.9) over a small length ΔL of the capillary tube we obtain:

$$G\Delta V = -\Delta p - [fV]_{mean}\Delta L \left[\frac{G}{2D}\right] \qquad \text{Eqn. (6.10)}$$

$$\Delta p = G\Delta V + [fV]_{mean}\Delta L \left[\frac{G}{2D}\right] \qquad \text{Eqn. (6.11)}$$

Where, $\Delta V = V_{i+1} - V_i$

And $\Delta p = p_{i+1} - p_i$

Here, Δp is negative since $p_i > p_{i+1}$.

Eqn. (6.11) may be expressed as follows:

$$\Delta p = \Delta p_{acceleration} + \Delta p_f$$

This means that total pressure drop over a length ΔL is the sum of that required for acceleration and that required to overcome frictional resistance.

For laminar flow the effect of wall roughness is negligible and friction factor is given by:

$$f = \frac{64}{Re}$$

For turbulent flow the friction factor increases with increase in roughness ratio. Moody's chart gives the variation of friction factor with Reynolds numbers for various roughness ratios. A number of empirical expressions are also available for friction factor in standard books on Fluid Mechanics. One such expression for the smooth pipe, known as Blasius Correlation is as follows:

$$f = 0.3164\ Re^{-0.25} \approx 0.32\ Re^{-0.25} \quad : \text{for } Re < 10^5 \qquad \text{Eqn. (6.12)}$$

The solution procedure for Eqn. (6.11) as suggested by Hopkins, Copper and Brisken [20] is as follows:

The condenser and evaporator temperatures T_c and T_e, the refrigerant and its mass flow rate are usually specified and the length and bore of capillary tube are required. Eqn. (6.11) is valid for a small length of the tube. Hence, the tube is divided into small lengths ΔL_i such that across each incremental length a temperature drop Δt_i of say 1 or 2 degrees takes place depending upon the accuracy of calculation required. The length of the tube ΔL_i for temperature to drop by say, 5℃ is found from Eqn. (6.11). The temperature base is taken for calculations instead of pressure base since the refrigerant properties are available on the basis of temperature.

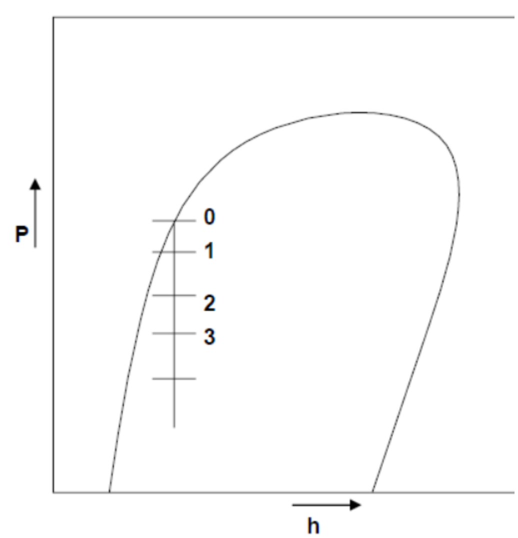


Figure 6.2 Step-wise calculation procedure for capillary tube length on p-h diagram

1. Assume an appropriate diameter D for the tube. At condenser exit and inlet to capillary tube point "0" shown in Figure 6.2, say the state is saturated liquid state hence, $v_o = v_f$, $h_o = h_f$, $\mu_o = \mu_f$ and $\dot{m}$ is known from thermodynamic cycle calculation for the given cooling capacity.

$$\therefore Re = \frac{4m}{\pi D \mu}$$

$$G = \frac{m}{A} = \rho V = \frac{V}{v}$$

The constants in Eqn. (6.11), G, $\frac{G}{2D}$ and $\frac{4m}{\pi D}$ which are required for solution are then calculated.

In thermodynamic analysis of capillary tube, isenthalpic expansion is the common assumption. However, expansion takes place according to Fanno-line flow as specified in above diagram. Thus, enthalpy does not remain constant since with pressure drop, the volume increases and an increase in kinetic energy is obtained from a decrease in enthalpy. Nevertheless, in the first few steps of pressure drop, there is not much difference between isenthalpic and Fanno-line flow.

2. Determine the quality at the end of the decrement assuming isenthalpic flow. Then at point 1 at pressure p_1.

$$x_1 = \frac{h_k - h_{f1}}{h_{fg1}}$$

3. Determine the specific volume.

$$v_1 = v_{f1} + x_1(v_{g1} - v_{f1})$$

Assuming that viscosity of mixture can be taken as weighted sum of viscosity of saturated liquid and vapor we get:

$$\mu_1 = x_1 \mu_{1g} + \mu_{1f}(1 - x_1)$$

$$Re_1 = \frac{4m}{\pi D \mu}$$

$$G v_1 = V_1$$

$$f_1 = 0.32\ \mathrm{Re}^{-0.25}$$

$$\Delta V = V_0 - V_1$$

$$\Delta p = p_1 - p_o$$

$$[fV]_{mean} = \frac{f_o V_o + f_o V_1}{2}$$

Hence from Eqn. (6.11), the incremental length of capillary tube for the first step, ΔL_1, is:

$$\Delta L_1 = \frac{-\Delta p - G\Delta V}{\left[\frac{G}{2D}\right](fV)_{mean}}$$

4. For the next section, i = 2: $t_2 = t_1 - \Delta t_2$, find the saturation pressure p_2 at t_2. The saturation properties v_{2f}, v_{2g}, h_{2f}, h_{2g} and μ_{2f} and μ_{2g} are obtained at temperature t_2.

5. Assuming the enthalpy to remain constant, that is $h_2 = h_1 = h_o$, the quality x_2 is found and steps 4 and 5 are repeated to find the incremental length ΔL_2.

Steps 4 and 5 are repeated for all the intervals up to evaporator temperature and all the incremental lengths are summed up to find the total length of the capillary tube.

It is observed from Eqn. (6.11) that the total pressure drop is the sum of pressure drops due to acceleration that is, $\Delta p_{acceleration} = G\Delta V$ and the pressure drop due to friction, that is:

$$\Delta p_f = \left[\frac{G}{2D}\right] \times [fV]_{mean} \times \Delta L$$

It may so happen under some conditions that after a few steps of calculation, the total pressure drop required for a segment may become less than the pressure drop required for acceleration alone, $\Delta p < \Delta p_{accelեartion}$. The increment length ΔL for this segment will turn out to be negative which has no meaning. This condition occurs when the velocity of refrigerant has reached the velocity of sound (sonic velocity). This condition is called choked flow condition. The velocity of fluid cannot exceed the velocity of sound in a tube of constant diameter; hence the calculation cannot proceed any further. The flow is said to be choked-flow and the mass flow rate through the tube has reached its maximum value for the selected tube diameter. For a capillary tube of constant diameter, choked flow condition represents the minimum suction pressure that can be achieved. If further pressure drop is required a tube of larger diameter should be chosen in which the velocity of sound occurs at larger length.

6.2.2 Calculation of capillary tube length

Assuming that inner diameter of capillary tube is 0.6mm, we have:

Area of cross-section (A) = 2.826×10^{-7} m^2

Mass flow rate ($\dot{m}$) = 2.278×10^{-5} kg/sec

$v_o = v_f = 0.0017\ 27$ m^3/kg

$h_o = h_f = 371.93$ kJ/kg

$\mu_o = \mu_f = 0.0001219$ N.sec/m^2

$$G = \frac{m}{A} = \frac{2.278 \times 10^{-4}}{2.826 \times 10^{-7}}$$

$$G = 778.485\ kg/m^2.s$$

Considering incremental difference of 5˚C, parameters such as p (pressure), v_f (specific volume), v_g (specific volume) h_{fg} (specific enthalpy), μ_f (Specific viscosity), μ_g (Specific viscosity), h_f (Specific enthalpy) and h_{fg} (Specific enthalpy) are taken from standard tables.

After this parameters such as X_i (dryness fraction), μ_i (Specific viscosity), v_i (Specific volume), V_i (Velocity of fluid in tube), ΔV (Change in velocity), Δp (Change in pressure), Re_i (Reynolds Number), f_i, f_v (mean) and ΔL (incremental length) are calculated and summarized in following tables:

Sr.	T	P	µf	µg	vf	vg	hf	hfg	hf-ho	X1
Unit	oC	Pa	N.sec/m2	N.sec/m2	m3/kg	m3/kg	kJ/kg	kJ/kg	kJ/kg	
1	0	429586	0.00019	9.41E-06	0.0016	0.2898	181.20	1263.25	190.73	0.15
2	5	515862	0.000179	9.59E-06	0.0016	0.2436	204.46	1245.10	167.47	0.13
3	10	615103	0.00017	9.78E-06	0.0016	0.2060	227.72	1226.50	144.21	0.12
4	15	728276	0.000161	9.98E-06	0.0016	0.1751	251.44	1207.19	120.49	0.10
5	20	857241	0.000152	1.02E-06	0.0016	0.1496	275.16	1187.44	96.77	0.08
6	25	1002760	0.000144	1.04E-06	0.0017	0.1284	298.90	1166.94	73.03	0.06
7	30	1166896	0.000136	1.06E-06	0.0017	0.1107	323.08	1145.79	48.85	0.04
8	35	1350345	0.000129	1.08E-06	0.0017	0.0960	347.50	1123.93	24.43	0.02
9	40	1554483	0.000122	1.1E-06	0.0017	0.0834	371.93	1101.37		

Sr.	T	µi	vi	Vi=viG	ΔV=Vo-Vi	Δp=pi-po	Rei	fi	fv(mean)	ΔL
Unit	oC	N.sec/m2	m3/kg	m/sec	m/sec	Pa				m
1	0	0.0001624	0.0451	35.10	33.7539	-86276	2876.29	0.0437	1.3422	0.07
2	5	0.0001566	0.0341	26.57	25.23055	-99241	2983.46	0.0433	1.0034	0.12
3	10	0.0001509	0.0256	19.96	18.61255	-113173	3095.42	0.0429	0.7415	0.21
4	15	0.0001456	0.0189	14.74	13.39694	-128965	3208.79	0.0425	0.5377	0.34
5	20	0.0001396	0.0137	10.66	9.319918	-145519	3345.83	0.0421	0.3801	0.56
6	25	0.0001349	0.0096	7.47	6.123284	-164136	3463.39	0.0417	0.2577	0.95
7	30	0.0001303	0.0063	4.93	3.583297	-183449	3583.57	0.0414	0.1618	1.72
8	35	0.000126	0.0038	2.92	1.577708	-204138	3706.37	0.0410	0.0873	3.58
9	40	0.0001219	0.0017	1.34			3831.76	0.0407		**7.56**

Table 6.1 Calculation of capillary tube length

It is clear from last column of above table that incremental ΔL is positive in all cases and it means that capillary tube of 0.6 mm ID can be used for this application. On summing all incremental lengths, we get total length of capillary tube as 7.56 meter. However due to fabrication limitations, we chose capillary tube of 0.8 mm ID and to compensate this effect, length of nearly 10 meter is taken. While fabricating, we initially took 15 meter long capillary tube and using *cut-and-try* method length is reduced till the desired effect is achieved.

6.3 Generator

Any refrigeration cycle requires some work input since we have to move heat from low potential (low temperature) to high potential (high temperature). As per Clausius statement of Second law of Thermodynamics, *"It is impossible to construct a device that* *operates in a cycle and produces no effect other than the transfer of heat from a lower-temperature body to a higher-temperature body."*

In different types of refrigeration systems, different devices are used for this purpose. However, they serve one common purpose: pumping heat from low temperature reservoir (evaporator) to high temperature reservoir (condenser) or increasing pressure of the refrigerant from evaporator pressure to condenser pressure.

In vapor absorption system, due to the use of absorbent-refrigerant solution, two processes are to be performed on the absorbed solution:

1. Increasing pressure of the refrigerant from absorber pressure to generator pressure.
2. Removal of refrigerant from refrigerant-absorbent solution.

In continuous vapor absorption system, pump is used for increasing the pressure and generator is used for removal of refrigerant from absorbent by supplying heat to the solution. However, in *intermittent* vapor absorption system, pump is not used and hence generator increases pressure as well as removes refrigerant from the solution. Thus, generator can be said to be analogous to compressor in vapor compression refrigeration cycle (VARS). Generator requires heat energy to operate and this heat energy can be waste heat or solar heat. In this work, solar heat was used as energy source and Fresnel lens was used to focus sunlight onto the generator.

6.3.1 Design of Generator

The design of generator includes determination of required generation temperature, solution concentrations, volume and shape of generator so that it can withstand high temperature and pressure to which it is subjected.

Prerequisites of generator vessel are as follows:

1. As we already know, generator is placed at the focal point of Fresnel lens. So it is imperative that size of generator must be at least equal to size of focal region.
2. Moreover, volume of generator vessel must be at least equal to total volume of aqua-ammonia solution.
3. Generator vessel is at higher temperature (88˚C) than atmospheric temperature. So, generator vessel must be designed to curtail heat losses to the atmosphere.
4. Metal selected for this vessel must not react with aqua-ammonia solution.
5. The container must withstand higher pressures without compromising its structural integrity.

By taking into account all above factors, cylindrical vessel of material stainless steel SS304 was selected.

We have discussed the calculation of required generation temperature using Concentration-Pressure-Temperature chart for aqua-ammonia pair in Section 3.3. The computations showed that minimum temperature required in generator is nearly 88°C in order to attain the requisite mass flow rate of ammonia which is 81.26 gm/hr or 2.26 × $10^{-5} kg/s$.

6.3.1.1 Volume of Generator

Now, before selecting amount of ammonia and water in aqua-ammonia solution, it is mandatory to reckon the amount heat collected by Fresnel lens for heating the solution. A series of test runs of Fresnel lens solar concentrator were conducted and the results are represented in Table 4.1. This value was obtained as 132.6 W. Therefore, it is evident that the quantities of ammonia and water are to be determined in such a way that the solution would consume less than or equal to 132.6 W to undergo generation of ammonia at requisite mass flow rate of 81.26gm/hr.

Accordingly, 81.26gm/hr ammonia must be generated within 6 hours of daylight on which intensity of solar radiations is sufficient enough for the generation process to take place. This means $81.26 \times 6 \approx 500gm$ ammonia must be generated per cycle.

Therefore, mass of water per 500gm ammonia just before generation is as follows:

$$X_1 = \frac{m_a}{m_w} = \frac{500}{m_w} = 0.8683$$

So, m_w = 575gm.

Now, removing all 500gm of ammonia at 15.554 bar will require very high generation temperature which is a bit problematic to accomplish by using Fresnel lens. However, decreasing difference between the concentrations before and after generation process will help to depress generation temperature. Hence, in order to reduce concentration difference, both m_a and m_w are multiplied by factor of 6. This simply means that total amount of the solution is increased in such a manner that 500gm ammonia will be generated even with relatively low generator temperature.

Therefore, mass of solution before generation:

$$m_{sol1} = m_{a1} + m_{w1} = 6\,(500 + 575) = 6450\ gm$$

In this solution, there is 3000gm ammonia and 3450gm water.

After generation (Minimum volume in the generator):

$$m_{sol2} = m_{sol1} - m_{gen} = 6450 - 500 = 5950gm$$

Now, volume of generator vessel is given by:

$$V = \frac{mass\ of\ solution}{minimum\ density\ of\ solution}$$

Since density of aqua-ammonia solution decreases with increase in temperature and concentration, minimum density is used in the above equation. Following graph shows density variation of aqua-ammonia solution with concentration at different temperatures.

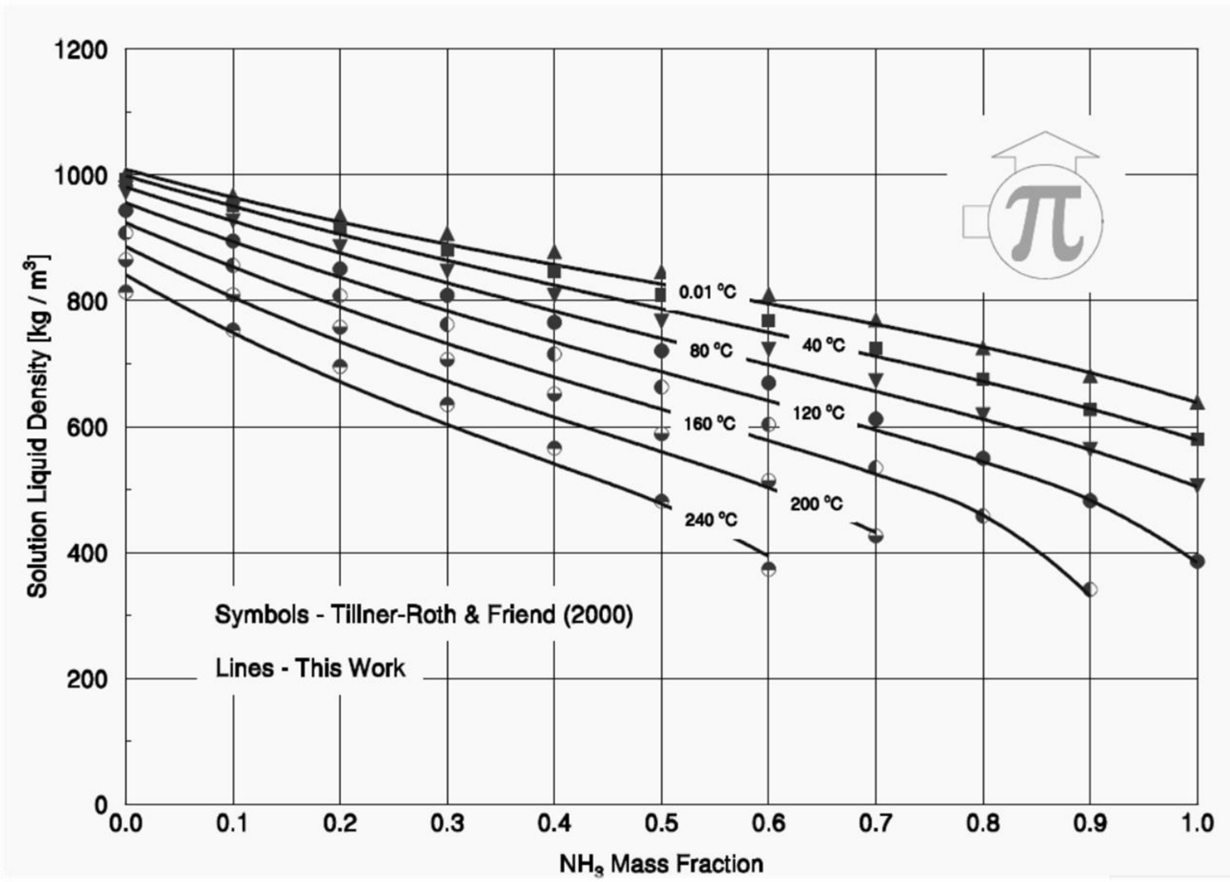


Figure 6.3 Density variation of aqua-ammonia solution with mass fraction at different temperatures

It is apparent from above figure that density of aqua-ammonia solution decreases with increase in its temperatures and ammonia concentration. Therefore, it will be lowest at highest temperature and highest concentration in refrigeration cycle.

First, it is to be analyzed at what condition the density is minimum.

1. Point of highest concentration: Absorbed ammonia before generation.

At this point, mass ratio:

$$X_1 = 0.8683$$

Hence, mass fraction:

$$\varphi_1 = \frac{X_1}{1+X_1} = \frac{0.8683}{1+0.8683} = 0.4647$$

Temperature at this point:

$$T_1 = 35^\circ C$$

And, density:

$$\rho_1 = 810 \text{ kg/m}^3$$

2. Point of highest temperature: Aqua-ammonia solution after generation.

At this point, mass ratio:

$$X_2 = 0.7246$$

Hence, mass fraction:

$$\varphi_2 = \frac{X_2}{1+X_2} = \frac{0.7246}{1+0.7246} = 0.4201$$

Temperature at this point:

$$T_2 = 91^\circ C$$

And density:

$$\rho_2 = 768 \text{ kg/m}^3$$

Now, minimum density:

$$\begin{aligned}\rho_{min} &= \min\{\rho_1, \rho_2\} \\ &= \min\{768,810\} \\ &= 768\ kg/m^3\end{aligned}$$

$$\therefore \quad \begin{aligned} V &= \frac{6.45}{768} \\ &= 8.398 \times 10^{-3}\ m^3 \\ &= 8.398\ lit \\ &= 8398\ cm^3\end{aligned}$$

Therefore the volume of the generator is finalized as 8.398 liters.

6.3.1.2 Shape of Generator

The shape of the generator is predominantly governed by the heat losses to the atmosphere while ease of fabrication is another plausible parameter. Considering the fact that the rate of heat transfer is directly proportional to surface area for a given volume, spherical vessel is the unparalleled choice. But the fabrication of a spherical vessel of desired volume is exasperating and expensive. Consequently, we opted for next best shape in this regard i.e. a cylinder. Furthermore, in cylinders, minimizing the surface area will also minimize the heat losses from the generator.

Radius for minimum surface area can be obtained by solving following equation:

$$\frac{dA}{dR} = 0 \qquad \text{Eqn. (6.13)}$$

Where, $A = \text{Surface area of cylinder}$

$= 2\pi R(H + R)$

$\text{R} = Radius\ of\ cylinder$

$\text{H} = Height\ of\ cylinder$

Relation between radius and height can be obtained by following equation:

Volume, $V = \pi R^2 H$

$\therefore \quad H = \frac{V}{\pi R^2} \qquad$ Eqn. (6.14)

Substituting this in Eqn. (6.13), we get:

$$A = 2\pi R(\frac{V}{\pi R^2} + R)$$

Differentiating this equation with respect to R, we have:

$$\frac{dA}{dR} = \frac{d(2\pi R(\frac{V}{\pi R^2} + R)}{dR} = 0$$

$\therefore \quad \frac{-2V}{R^2} + 4\pi R = 0$

Or, $R^3 = \frac{V}{2\pi} = \frac{8398}{2\pi}$

This gives, $R = 110.15\ mm$

$\therefore \quad H = \frac{V}{\pi R^2} = \frac{8398}{\pi \times 11.015^2} = 220.32\ mm$

$\therefore$ A cylinder of radius 110.15mm and height 220.32mm is optimum. (a)

6.3.1.3 Heat loss from the proposed generator (a)

Generator will lose majority of heat by convection and radiation heat transfer. Combined heat transfer equation may be written as:

$$Q_{lost} = Q_{rad} + Q_{conv}$$

$$Q_{lost} = A\varepsilon\sigma(T^4 - T_0^4) + hA(T - T_0) \qquad \text{Eqn. (6.15)}$$

Where, T= Generator temperature

= 91˚C

= 362 K

T_0 = Atmospheric temperature

= 35˚C

= 308 K

A = Surface area of generator

= 2πR (H+R)

= 2π×11.015(11.015+22.030)

= 0.2289 m^2

ε = Emissivity of black color painted stainless steel

= 0.92

σ = Stefan's radiation constant

= $5.67\times10^{-8} W/m^2K^4$

h = convection heat transfer coefficient for still air

= 25 W/m^2K

Substituting these values in Eqn. (6.15), we get:

$$Q_{lost} = 0.2289\big(0.92 \times 5.67 \times 10^{-8} \times (362^4 - 308^4) + 25 \times (362 - 308)\big)$$

$$= 426.607\ W \qquad \text{(b)}$$

It means this amount of heat is lost by the aqua-ammonia solution in the generator vessel.

Heat loss by the solution to the atmosphere can also be written as:

$$Q_{lost} = m \times c \times \Delta T \qquad \text{Eqn. (6.16)}$$

Where, ΔT = Decrease in temperature of aqua-ammonia solution

c = Specific heat capacity

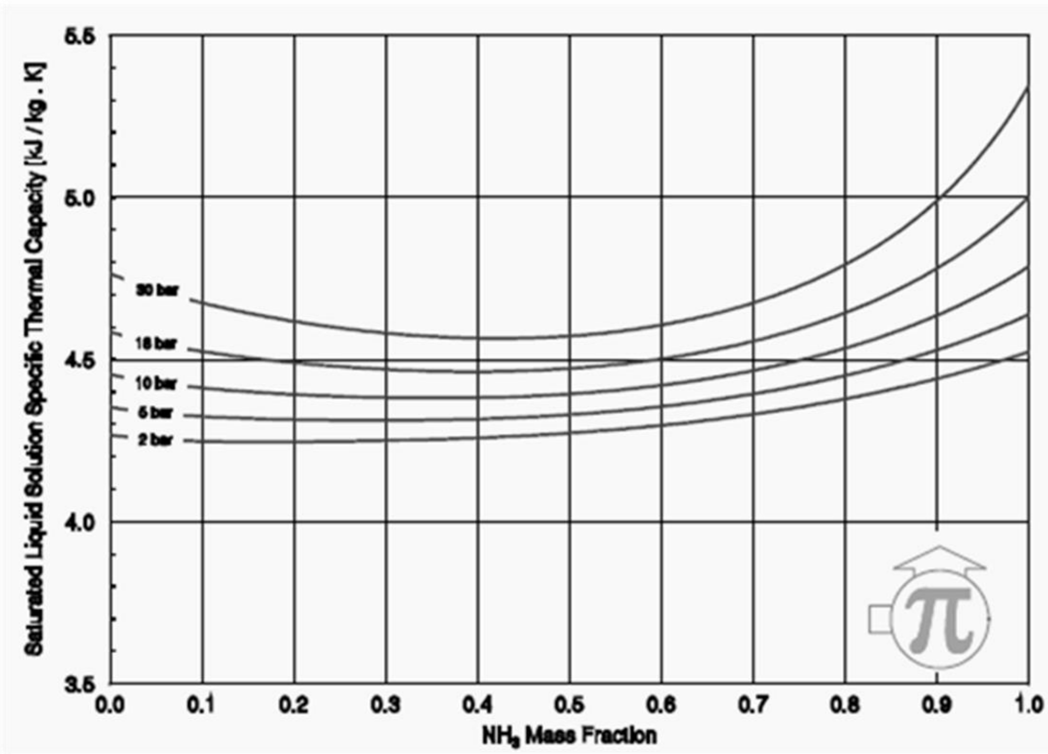


Figure 6.4 Variation of specific heat capacity of aqua-ammonia solution with mass fraction

From above figure:

For mass fraction, ϕ_1 = 0.4647

c_1 = 4.45kJ/kgK

= 4450 J/kgK

For mass fraction, ϕ_2 = 0.4201

c_2 = 4.42kJ/kgK

= 4420 J/kgK

Therefore, average specific heat:

$$c = \frac{c_1 + c_2}{2}$$

$$= \frac{4450+4420}{2}$$

$$c = 4435 \ \frac{J}{kgK}$$

∴ From value of c, (b) and Eqn. (6.16), we have:

$$426.607 = 2.78 \times 10^{-5} \times 4435 \times \Delta T$$

This gives,

$$\Delta T = \frac{426.607}{2.78\times10^{-5}\times4435}$$

$$= 3460.107 \ K$$

A temperature drop of 3460.107 is practically impossible. Instead, the temperature of the solution will decrease continuously until it becomes equal to atmospheric temperature. This means that heat gained by ammonia will be lost to atmosphere and no ammonia will be generated at all. Patently, it is imperative to design the generator to avoid such drastic temperature fall. In order to avoid large temperature losses, generator shape was changed as depicted in figure given below:

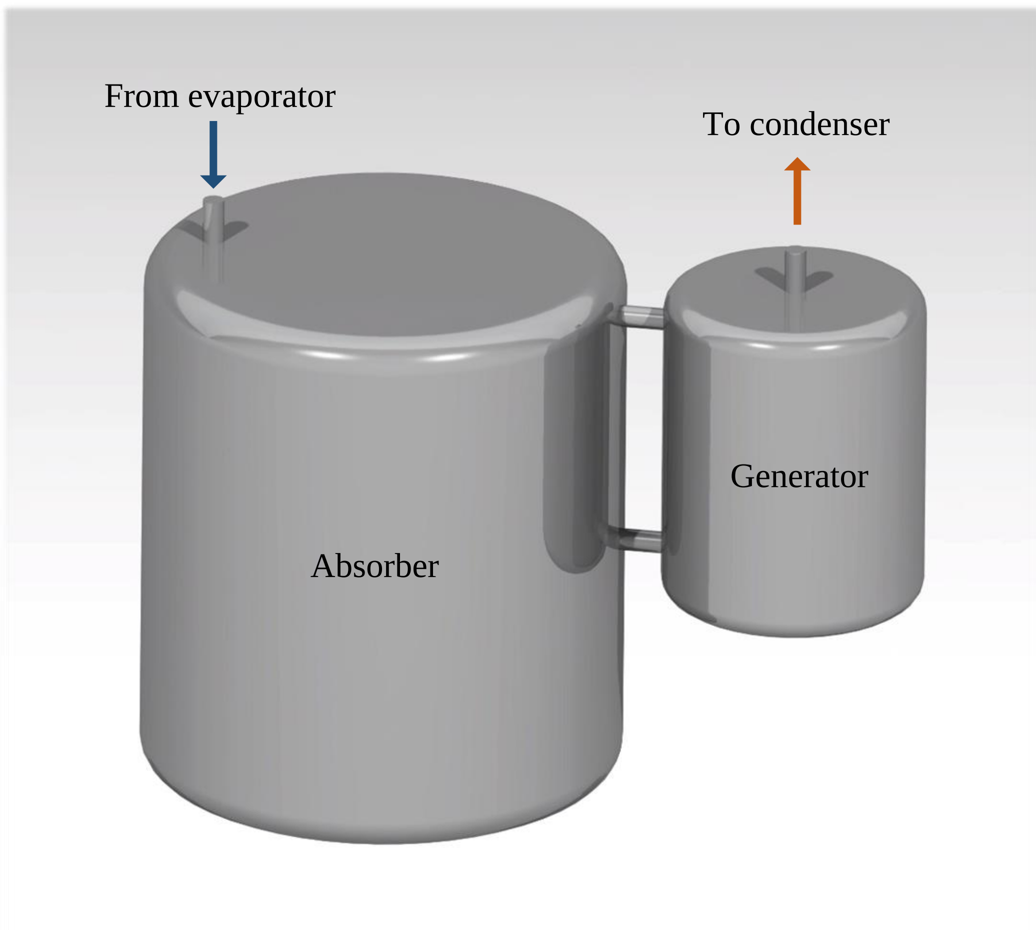


Figure 6.5 Absorber and generator

As we can see, generator is made of two stainless steel vessels connected by two stainless steel pipes. Smaller vessel is to be put at the focal point of Fresnel lens and this vessel will be referred to as *generator*. While larger vessel is to be attached to the evaporator where ammonia will absorbed in low concentration aqua-ammonia solution. Hence it is named as *absorber*.

Reasons for selecting this configuration for generator are as follows:

1. The actual size of focal region is very small as compared to theoretical diameter (From (a)). Thus, only small portion of the generator is actually heated. By choosing small container for the purpose of heating aqua-ammonia solution, the generation process will be appreciably improved by making effective use of solar radiations focused.
2. Since the absorber is not at the focus, it can be well insulated to impede heat losses.
3. The Two tubes connecting the generator and the absorber abet mass transfer and instigate convection currents within the solution. When solution inside generator is heated, it becomes ammonia deficient by liberating ammonia gas. The increase in temperature and decrease in concentration of the solution increase its density causing it to flow back into the absorber through lower tube. While upper tube carries ammonia rich, low density solution from absorber to generator.

6.3.2 Corroboration of the design of Generator

Most of the vital requirements of the generator mentioned in the section 6.3.1 are fulfilled by the design discussed until now. Before finalizing this design of the generator, it is essential to make sure following two things:

1. The heat supplied by the Fresnel lens solar concentrator is sufficient for the generation process to take place.
2. The generator withstands higher pressures without compromising its structural integrity.

In next section, we will verify above two conditions.

6.3.2.1 First condition

It is now clear that 6.45 kg aqua-ammonia solution with mole fraction 0.4790 at temperature 35˚C and pressure 4.295 bar is to be heated to 5.45 kg solution with mole fraction 0.4548 at temperature 88˚C and pressure 15.544 bar.

Before reviewing the calculations, it is crucial to understand the physical nature of generation process. From analytical point of view, two changes are taking place inside generator:

1. Generation of anhydrous ammonia at desired pressure in which ammonia separates from aqua-ammonia solution.
2. Change in concentration and pressure of solution without phase change in which the solution of 0.4790 molar concentration and 4.295 bar changes to 0.4548 molar concentration at 15.544 bar pressure.

By above consideration, enthalpy change can be expressed as:

$$\Delta H_{req} = m_{sol} \times h_{sol} + m_a \times h_a$$

Where, m_{sol} = Mass of solution

= 6.45 kg

m_a = Mass of anhydrous ammonia generated

= 0.5 kg

h_{sol} = Specific enthalpy required to raise pressure (from 4.295 bar to 15.544 bar) and reduce concentration (from 0.4790 molar to 0.4548 molar)

= 300 kJ/kg

h_a = Specific enthalpy required to raise pressure of anhydrous ammonia (from 4.295 bar to 15.544 bar)

= 1100 kJ/kg

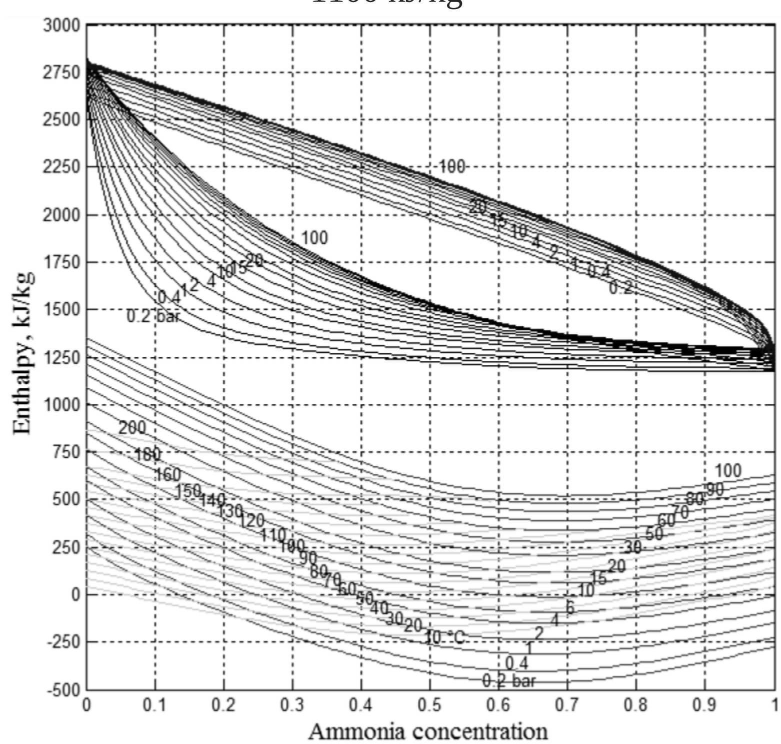


Figure 6.6 Pressure-temperature-concentration-enthalpy chart for aqua-ammonia solution

Therefore, total enthalpy change:

$$\Delta H_{req} = m_{sol} \times h_{sol} + m_a \times h_a$$

$$= 6.45 \times 300 + 0.5 \times 1100$$

$$= 2485\ kJ$$

This enthalpy change is to be achieved within 6 hours. Therefore, required heat transfer rate can be written as:

$$Q_{gen} = \frac{\Delta H_{req}}{t}$$

$$= \frac{2485}{6 \times 3600} \frac{kJ}{s}$$

$$Q_{gen} = 115\ W$$

And from Table 4.1, we know that heat supplied by Fresnel lens solar concentrator is 132.6 W which is greater than the required generation heat of 115W calculated above. Therefore, the first condition is satisfied.

6.3.2.2 Second condition

Generator vessel is a thin cylindrical pressure vessel and it is subjected to hoop stresses due to large pressure created by ammonia generated inside.

Hoop stresses induced in thin cylindrical pressure vessel are given by:

$$\sigma_H = \frac{pd}{2t}$$

Where, σ_H = Hoop Stress

p = Pressure, MPa

= 15.544 bar

= 1.5544 MPa

d = Internal diameter of pressure vessel, m

= 112 mm

t = thickness of pressure vessel, m

= 0.5 mm

$\therefore$ $\sigma_H = 174.0928$ (c)

Yield strength of stainless steel,

$$\sigma_{yt} = 215 \text{ MPa}$$

Factor of safety, FOS = 1.1

Allowable stress, $$\sigma_a = \frac{\sigma_{yt}}{FOS} = \frac{215}{1.1} = 195.4545\ MPa \quad \text{(d)}$$

From (c) and (d):

$$\sigma_H < \sigma_a$$

Hence, the generator is capable of withstanding the given pressure without compromising its structural integrity.

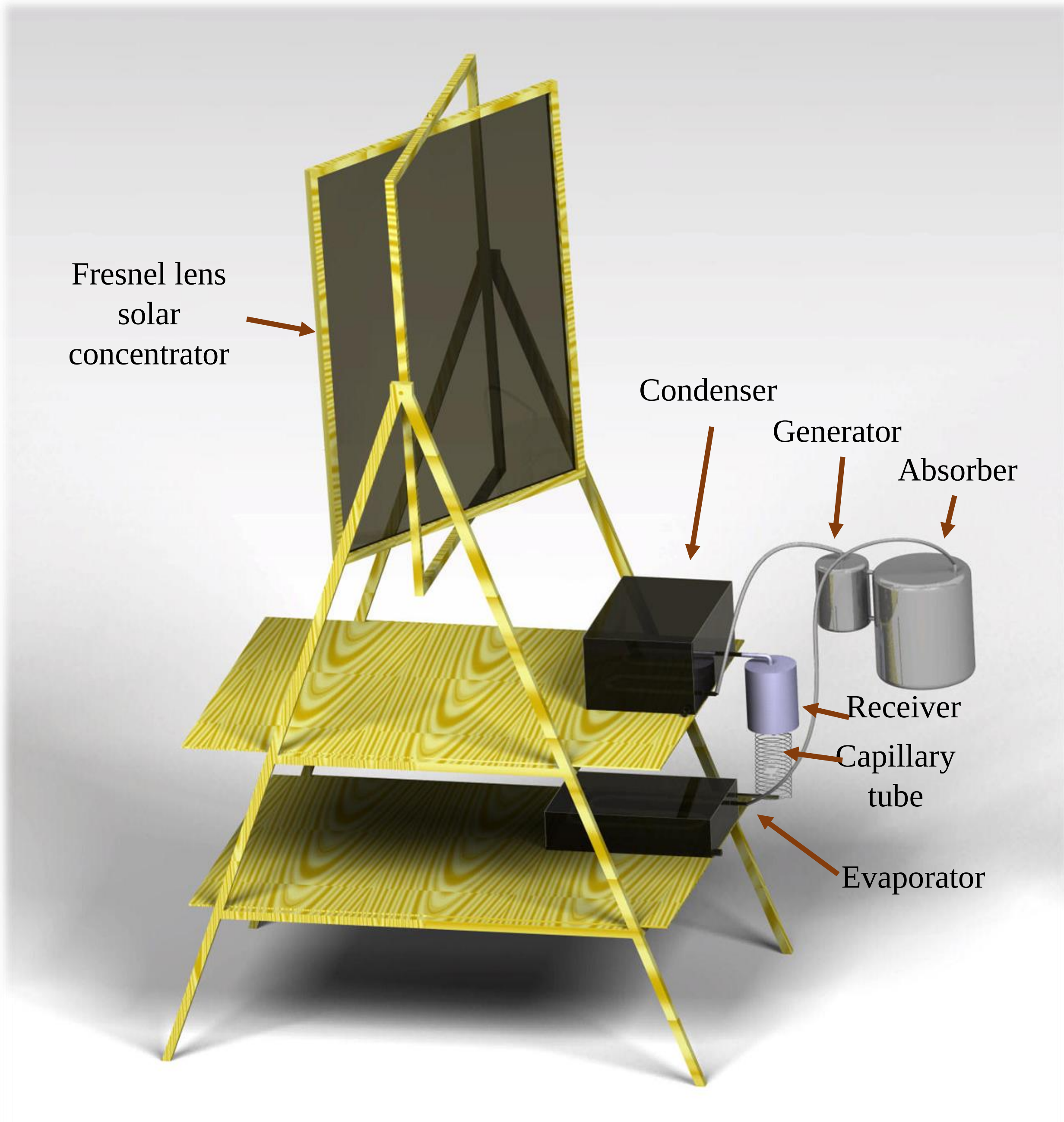


Figure A Design of the Assembly

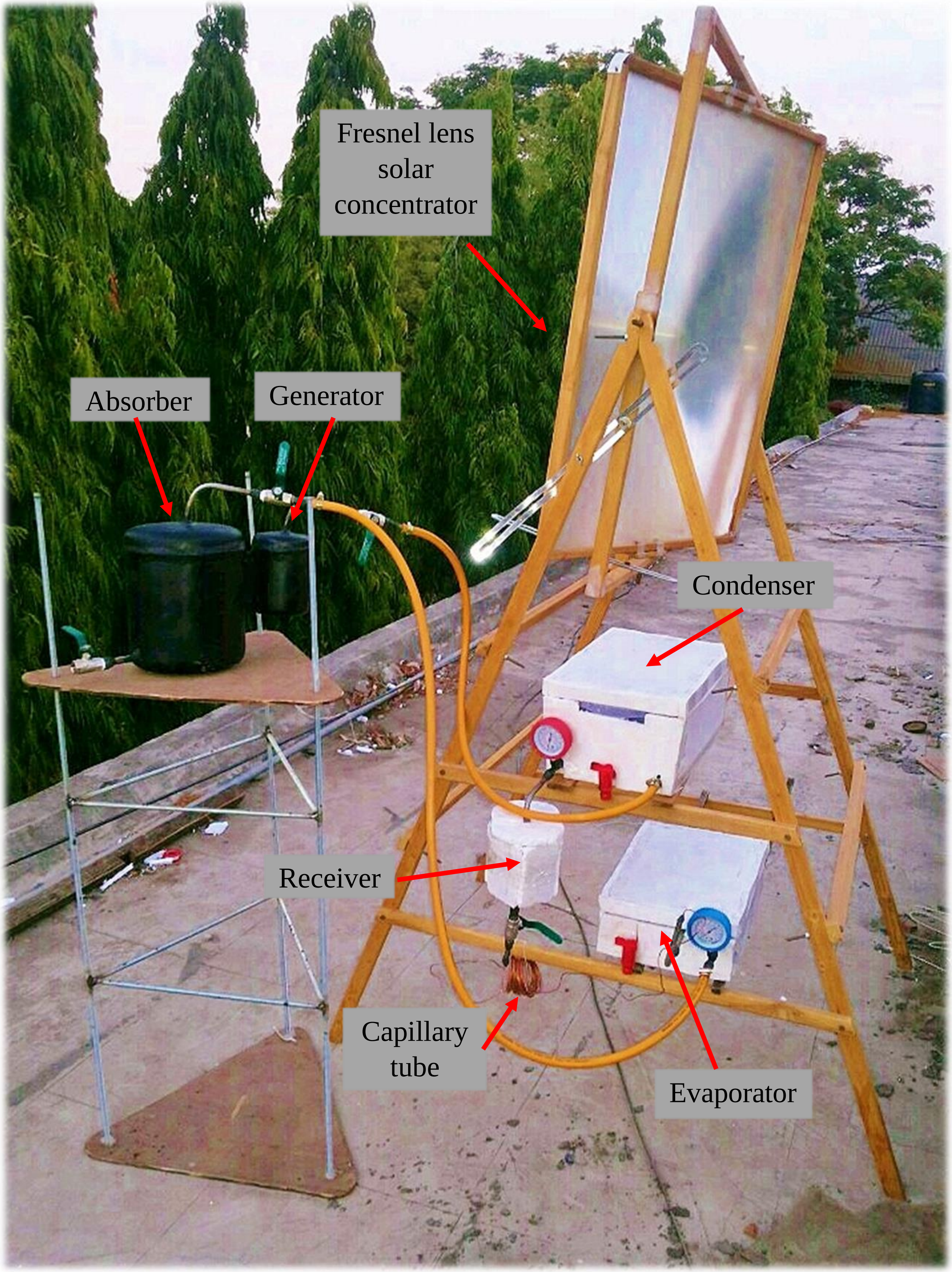


Figure B Completed Assembly

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