

DESIGN AND CONSTRUCTION OF AN AIR-COOLED
SOLAR-POWERED REFRIGERATION SYSTEM

BY

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DECLARATION

I hereby declare that this thesis has been composed by myself and that it is a record of my own research work. It has not been accepted in any previous application for a higher degree. All quotations are distinguished by quotation marks and the sources of information are specifically acknowledged by means of references.



Emmanuel C. Nsofor.

30th October, 1982.

DEDICATION

Dedicated

To

Ngozi my wife, and all the members of my family,
for their help and understanding.

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I am sincerely grateful to the Almighty God for guiding me successfully through the whole programme.

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ABSTRACT

This project is one in the series of work in solar cooling at the Mechanical Engineering Department of Ahmadu Bello University, Zaria. It is also a major improvement on the previous work done in solar refrigeration. The results and experience are presented.

In the system, the solar absorber and the refrigeration system's ammonia generator are combined to form one unit. Design calculations based on the refrigeration load of the cold room which was built earlier were made and consequently, the various other components were constructed. Air cooling of the components by natural convection was adopted because of easier availability of air and to eliminate running costs due to the use of water.

The components were built and assembled at the department's workshop. (See PLATES I and II). Leakage tests were performed using compressed air and soap bubbles. The system was charged with ammonia gas and ammonia-water solution. However, the available pump used was not suitable for the suction of the required ammonia-water solution and the system's circulation, as it is not self-priming. Because of this, the vapour generated in the solar collector could not circulate to achieve continuous refrigeration. Intermittent refrigeration was achieved by the use of a vacuum pump.

The results obtained are given in this report. Also recommendations for further modification of the plant are presented.

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NOMENCLATURE

a	-	Specific surface of packing.
α	-	Absorptivity.
A	-	Area.
b	-	Distance, spacing.
β	-	Coefficient of volume expansion, solar altitude.
c	-	Specific heat.
C.O.P.	-	Coefficient of Performance.
d	-	Inside diameter.
D	-	Outside diameter.
δ	-	Declination.
Δ	-	Difference.
ϵ	-	Emissivity.
f	-	Cooling rate factor, friction factor, convection coefficient.
ϕ	-	Latitude.
g	-	Acceleration due to gravity.
G	-	Grashoff number.
h	-	Heat transfer coefficient, specific enthalpy, head loss.
H	-	Solar intensity, height.
k	-	Thermal conductivity.
L	-	Length, molar liquid flow rate.
m	-	Mass flow rate.
M	-	Molecular weight.

n - Number, day of the year.
 η - Efficiency.
 Nu - Nusselt number.
 P - Pressure, Perimeter.
 Pr - Prandtl number.
 q - Unit area heat transfer rate.
 Q - Heat flux.
 r - Inner radius.
 R - Outer radius.
 Re - Reynold's number.
 ρ - Density, reflectivity.
 S - Angle of tilt.
 σ - Stress, Stefan-Boltzmann constant.
 t - Temperature.
 T - Absolute temperature.
 θ - Angle, temperature change.
 τ - Transmissivity.
 U - Overall heat transfer coefficient.
 μ - Dynamic viscosity.
 v - Velocity.
 v_s - Average speed.
 V - Molar gas flow rate, Volume.
 $\dot{V}$ - Volume flow rate.
 W - Width, weight.
 ω - Hour angle.
 x - Mass concentration, mole fraction in solution,
 thickness.
 y - Mole fraction in vapour.

Subscripts

a	-	Air, ambient.
A	-	Absorber, ammonia.
b	-	Bottom.
B	-	Absorber plate.
c	-	Condenser, convective.
d	-	Diffuse, day, daily, discharge.
E	-	Evaporator, east, edge.
f	-	Fin, final, friction.
fg	-	Difference between saturated vapour value and saturated liquid value.
g	-	Glass, generator.
h	-	Hour, hoop.
i	-	Initial, inside surface.
k	-	Conductive.
l	-	Longitudinal.
L	-	Lost, liquid.
N	-	North.
o	-	Outside surface.
p	-	Packing, pump, product.
r	-	Radiative, rectifier.
s	-	Suction, saturated, shade, solar, second.
S	-	South.
t	-	Tube, top.
U	-	Useful.
V	-	Vapour.
w	-	Wall, wind, water.
W	-	West.

CHAPTER ONE

INTRODUCTION

Refrigeration is defined as the branch of science that deals with the process of reducing and maintaining the temperature of a space or material below the temperature of the surroundings. To accomplish this, heat must be removed from the body being refrigerated and transferred to another body whose temperature is below that of the refrigerated body. With advancement in science and technology, the role and functions of refrigeration and its applications have steadily become indispensable to the existence of the modern society.

1.1. REFRIGERATION IN THE SOCIETY

Refrigeration was developed as a method of food preservation and this is still its most important and largest use. At first, refrigeration was employed for the large scale transportation and the storage of foodstuff, but domestic refrigeration has grown steadily in the last forty years. Today, production of refrigerators of this type throughout the world constitutes the largest proportion of the refrigeration industry.

It is interesting to also note that the chloro-fluoro-carbons (freons) which were developed specifically as refrigerants have since been adapted to act

as blowing agents for some of the modern forms of thermal insulation.

Refrigeration In Industry: The use of refrigeration in industry is by no means new. In the chemical industry, ice has long been used in the production of certain dyestuffs for preventing development of high temperatures in exothermic reactions. Refrigeration is used for condensation of gases, for the recovery of highly volatile compounds, for the control of temperature in the production of many products. In the pharmaceutical industry, one application is that of freeze drying. Cold is used to reduce the water content of air in contact with the product being dried, so that water is transferred from the product, through the air, to the refrigerator condenser. Products that are not heat-stable are dehydrated in this way to give finished materials that are readily transferable and have a much longer "shelf life". This form of drying has been used extensively in the production of antibiotics, such as Penicillin. Refrigeration also plays its part in the metallurgical industry, where low temperatures are used in shrink-fitting liners into cylinders and similar operations. In certain large-scale civil engineering undertakings, such as the construction of dams, refrigeration has enabled work to go ahead more rapidly and effectively. It is used

for the consolidation of water-logged ground by freezing, which allows excavation or borings to be made without risk of either flow of the faulty ground or cave-in. Laboratories dealing with many different problems, rely on refrigeration as a servant. Biological laboratories, for example, and laboratories studying food preservation, use incubators for low temperature work. Industrial laboratories, such as are encountered in the aircraft industry, rely on refrigeration to simulate operating conditions for materials being tested for use in the field. The scope for refrigeration in industry is unending and the applications are many.

Refrigeration In Transport: The use of refrigerated transport has increased over the years, developing from the carriage of chilled meat to the present day transport of vast quantities of chilled and frozen foods, many of them in deep-frozen condition by land, sea and air.

Refrigerated air transport is limited to the supply of off-season fruit or vegetables when cost of purchase is high. Refrigerated transport by road and rail is extensively used: by road for local deliveries from depots to retailers, and by both road and rail for long-distance haulage from the location of initial freezing to final storage, often involving transportation from one country to another.

Refrigerated transport at sea involves the shipment of chilled and frozen foodstuffs in vast quantities. Holds are refrigerated by grids through which refrigerated brine circulates, or by air circulation, for which expansion machines are used. Modern passenger ships are invariably air-conditioned, since passengers now demand a higher degree of comfort than was expected even 20 years ago.

The fishing industry is also not left out. Large deep-sea trawlers now have cold storage rooms where cleaned fish can be kept in prime condition, permitting the ships to stay at sea for long periods at a time.

Comfort Cooling/Air-Conditioning: Air conditioning, though still a luxury in some parts of the world, is an essential in others. Industrial operations carried out under hot conditions, call for air-conditioning for the operatives. Works and offices in hot regions must also be air-conditioned. Even in semi-tropical and temperate zones of the world, the incidence of air conditioning is rapidly increasing. Large new buildings throughout the world incorporate partial or complete air conditioning. Modern trains in many countries are now regularly air-conditioned. Without adequate refrigeration, high-speed high-altitude air travel in reasonable comfort would be impossible.

The modern jet liners of today are all cooled by refrigeration. The field of comfort cooling is therefore one of considerable importance.

Miscellaneous Applications of Refrigeration: In addition to the many pre-dominating outlets for refrigeration, there are many minor fields in which refrigeration plays its part. In medicine, the preservation of tissue, bones and blood would not be possible for even a short time if it were not for the reliable refrigerators that are available. In the operating theatre, cooling is employed to reduce the metabolic rate of the patient and to enable operations to be carried out without the use of an anaesthetic.

In the world of sports, refrigeration also finds a place, being used for such purposes as maintaining tennis balls in perfect condition and, of course, in artificial ice rinks. Refrigerants are not used for cooling only; they have many other applications of interest to the refrigerator manufacturer and those connected with the industry. Refrigerant gases are used for charging vapour-pressure devices, operating thermostats for refrigerators and in addition, thermostats for many other applications, both in the home and the industry. The "freons" are used as blowing agents for plastic foams. The solvent properties of the higher boiling members of the freon family enable them to be used for specialized cleaning operations by virtue of their relative freedom from hazard, their stability and their non-corrosive character.

1.2. WHY IT IS NECESSARY TO APPLY SOLAR ENERGY TO REFRIGERATION

From the above discussion, we have been able to show how important refrigeration is to the functioning of the society. It is also, however, clear that refrigeration functioning depends on the available energy resources. Recent large price increases in oil and gas, reduced availability of fossil fuels, such as evidenced in the energy crises of the early seventies, and the rising awareness regarding environmental pollution issues, have led to a surge of interest in using the power of the sun. This is because solar energy represents the only totally non-polluting inexhaustible energy resources that can be utilized economically to supply man's energy needs for all time.

The unavoidable release of pollutants associated with the combustion of fossil fuels presently being widely used has been a source of concern to man for some time now. Coal and petroleum contain sulphur impurities which form sulphur oxides during combustion. At the high temperatures typical for the internal combustion engines, nitric oxides are produced. Incomplete combustion produces carbon monoxide and hydrocarbons. Even if these undesirable and biologically hazardous by-products are eliminated, the generation of carbon-dioxide is unavoidable. While

the atmosphere contains carbon dioxide (an essential component of life process), the effect of an increase in the atmospheric content is uncertain.

Nuclear energy has been seen by many as an ultimate energy solution apart from solar energy. With nuclear energy, there is, however, the inherent problem of the effects of radiation and radionuclides on biological materials. Of particular significance is the problem of the disposal of long-half-life waste products that need to be isolated from the environment for thousands of years. At present therefore, the nuclear future is far from certain. Unless an unforeseen major breakthrough occurs with respect to its safe waste disposal, even its most optimistic proponents do not expect nuclear energy to modify this century's energy picture.

Of all possible direct uses of the sun's energy, the one greeted around the world with the greatest enthusiasm is solar cooling. Although in some countries, electricity and fuel are presently abundant and cheap, solar cooling may eventually become the most widely used application of solar energy. In tropical countries, there is a special interest because everyone desires cooling devices but the ones now available are expensive and the cost of electricity is high. Solar house cooling has the advantage that usually it is needed most when the solar radiation is most intense,

and long storage of energy is not absolutely necessary. Present mechanical refrigeration and air cooling are expensive both in initial capital cost and in continued operating cost for electricity or fuel. Although capital cost for solar energy is high, the operating cost is much less than that for conventional cooling with electricity or fuel.

Air conditioning and refrigeration can only be afforded by the wealthy in many poor countries of the tropics but this social problem can be greatly solved by the development of solar cooling. Also in many tropical countries, milk and meat are not traditionally used as they ought to, because they cannot be preserved. It is also known that in some hot countries, about one quarter of the vegetable crop spoils because of lack of cooling facilities. Factories in some tropical community can be made to operate with a higher efficiency by applying solar energy for air conditioning. Also the health of millions of people can be improved by the development of solar cooling.

1.3. REFRIGERATION METHODS

Quite obviously, to produce refrigeration, by which is meant the removal of heat, there must be a flow of heat from the substance to be cooled. This heat flow or heat transfer, can only take place provided there is a cooling medium to which the heat can flow, at a

lower temperature than the substance being refrigerated. The rate of heat flow will depend on the temperature difference between the two substances, the surface areas of substances to and from which heat flows, and the resistance to flow of any heat insulation which may exist between the two substances to impede the flow.

Before the advent of modern refrigeration processes, water was kept cool by storing it in earthenware jugs so that the water could flow through the porous holes and evaporate. The evaporation carried away heat and cooled the water. Natural ice from lakes and rivers were often cut during the winter and stored in caves, straw-lined pits, and later in sawdust-insulated buildings to be used as required. The early Romans carried pack trains of snow from the Alps to Rome for cooling the Emperors' drinks. Though these methods of cooling all make use of natural phenomena, they were used to maintain a lower temperature in a space or product and may properly be called refrigeration.

The process of refrigeration may be produced in many ways, the most usual being:

- (a) The use of natural or manufactured ice;
- (b) The use of freezing mixtures;
- (c) Vacuum systems
- (d) Thermo-electric processes;
- (e) Adiabatic demagnetisation;

- (f) Mechanical vapour compression;
- (g) Vapour absorption.

Use of Natural or Manufactured Ice: A limited degree of refrigeration can be obtained by placing the substance to be cooled in close proximity to blocks of ice. Heat flows from the substance to the ice, thereby cooling the substance and melting the ice. Only moderate degrees of cooling can be effected by this system. One very common example of this method is the ice blocks used to display fish on fishmonger's slabs.

Dry ice (solid carbon dioxide) is used principally for the preservation of perishable goods in transit, and for any other application where the melting of ordinary ice would constitute an objection. The normal type of mobile ice cream container is the most usual example of this form of refrigeration. To produce dry ice, clean dry carbon dioxide gas is raised to a high pressure in a compressor, and after being freed from any traces of oil picked up in the compressor, is passed to a condenser where heat is taken from it. The gas consequently liquefies, and this liquid is allowed to expand suddenly through nozzles into a chamber. The rapid expansion causes the carbon dioxide to form snowflakes, which are compacted into blocks or sheets ready for use.

Use of Freezing Mixtures: Freezing mixtures or chemical refrigerators as they are sometimes called, work on a principle which involves the absorption of latent heat when two solids unite chemically to form a liquid. In the case of ice and salt, this is the heat required to change ice into water without any change in temperature. Freezing mixtures find no wide commercial application but are extensively used for laboratory work or other applications where the production of low temperatures is quickly required.

The mechanism of the working of the ice and salt mixture, for example, is that first, the ice melts, absorbing heat from its surroundings and a saline solution is formed which in turn will dissolve more ice. To transform the ice into water, latent heat is required, and if the mixture is thermally insulated from its surroundings, this heat can only come within the mixture itself. The water formed will in turn dissolve more salt and eventually a state of equilibrium will result in which the concentration remains constant and provided the system is insulated from all external heat application, a constant minimum temperature will also result. Some of the salts used in this type of refrigeration method are Calcium Chloride, Sodium Chloride, Ammonium Nitrate and Ammonium Sulphate.

Vacuum Systems: Water is unsuitable for use as a refrigerant in low temperature applications because of its low vapour pressure and high specific volume at low temperatures. However, at temperatures encountered in air-conditioning, it is possible to maintain the degree of vacuum required even though at 7°C , the vapour pressure is about 0.01 bar and the specific volume is about $129 \text{ m}^3/\text{kg}$. The large volume flow implied by this has led to the use of an ejector pump in place of the usual compressor for this type of plant.

Here, a steam is expanded in a nozzle to form a high speed jet at low pressure, which entrains the vapour from the evaporator. The combined stream from the mixing chamber is then diffused in the divergent part of the ejector pump and pumped to the condenser. A portion of the condensate is throttled into the evaporator and the remainder is pumped back to the steam boiler.

This type of system has no moving parts, but it has a very low efficiency. If waste steam is available from an existing boiler plant, its poor efficiency is of no consequence as the refrigeration plant itself is cheap to construct and operate.

Thermo-Electric Systems: Thermo-Electric systems are still in the development stage. These systems depend on the phenomenon known as the Peltier effect, which forms the basis of thermo-electric cooling. It

is found that when current is passed through a thermocouple whose junctions were initially at the same temperature, the junction temperatures will change. This is the Peltier effect.

With proper choice of materials, this Peltier effect can be utilized for refrigeration purposes. Normally, two dissimilar conductors are joined to form the thermocouple. The cold junction is attached to a metal plate or some form of heat-transfer surface. This surface is exposed to the space or substance to be refrigerated, and serves the same purpose as the evaporator of a vapour compression system. The hot junction is attached to some type of surface which would allow heat rejection to atmospheric air or some other medium. The surface performs the same function as the condenser of a vapour compression system. An external battery is used to provide for circulation of direct current (D.C.) through the circuit. The battery may be compared functionally to the compressor of a vapour compression system.

It has been found that semi-conductor materials are good thermo-electric materials whereas insulators are poor thermo-electric materials because of their small electrical conductivity. Metals also fail because of small thermo-electric power and large thermal conductivity. The best results are obtained with the semi-conductors which have property values midway between

Mechanical Vapour Compression System: The most widely used method for refrigeration is the method of the mechanical vapour compression refrigeration cycle. Here, electric or mechanical energy is used to operate a compressor that circulates a refrigerant vapour to an air-cooled or water-cooled condenser, from which the resulting liquid passes through an expansion valve to the evaporator or cooling unit. Evaporation of the refrigerant results in heat removal by the cooling unit in contact with the space. The cycle is completed by vapour returning to the compressor. Heat is thus removed from a high temperature source and discharged to the surrounding atmosphere or to cooling water by means of mechanical work input to the compressor, which raises the temperature of the refrigerant above the temperature of the heat sink. Mechanical vapour compression system is efficient but the energy input is shaft-work, which is high-grade energy and therefore expensive. The relatively large amount of work required in it is because in compression, the vapour undergoes a large change in specific volume.

Of all the various refrigeration methods, the vapour-absorption method has been found to be most suitable for direct utilization of solar energy. In the vapour-absorption system, only a small amount of work is needed because the pressure of the refrigerant can be raised without appreciably altering its volume.

Heat, a low grade energy is required for vapour generation. This heat is easily supplied by direct solar radiation.

1.4. THE VAPOUR-ABSORPTION REFRIGERATION CYCLE

The vapour-absorption cycle is similar to the mechanical vapour compression cycle in that it employs a volatile refrigerant, usually either ammonia or water, which alternately vaporises under low pressure in the evaporator by absorbing latent heat from the material being cooled and condenses under high pressure in the condenser by surrendering the latent heat to the condensing medium.

The principal difference between the absorption and vapour compression cycles is the compression work requirement. While the vapour compression cycle requires much energy to compress the vapour from lower pressure to the higher pressure, the absorption system compresses saturated liquid to the same higher pressure but the compression work required is either very small or negligible. In fact, the compression work is reduced to zero in the electrolux absorption system where the potential drive is provided only by the partial pressure principle. In addition, whereas the energy input required by the vapour compression cycle is supplied by the mechanical work of the compressor, the energy input in the absorption cycle is in the

form of heat supplied to the generator directly. The source of heat supplied to the generator is usually low pressure steam or hot water, although in some smaller systems the heat is usually supplied by the combustion of an appropriate fuel such as natural gas propane or kerosene, directly in the generator or by an electric resistance heater installed in the generator. As stated earlier, solar energy is employed as the direct source of heat to the generator in this project.

The basic cycle is illustrated in Fig. 1.1. The system consists of four basic components: an evaporator and an absorber which are located on the low-pressure side of the system, and a generator and a condenser, which are located on the high-pressure side of the system. Two working fluids are employed, a refrigerant and an absorbent. The flow cycle of the refrigerant is from the condenser to the evaporator through an expansion valve and then to the absorber, to the generator and back to the condenser, while the absorbent passes from the absorber to the generator and back to the absorber.

With reference to Fig. 1.1, the operating sequence is as follows. High pressure liquid refrigerant from the condenser passes into the evaporator through the expansion valve that reduces the pressure of the refrigerant to the low pressure existing in the evaporator. The liquid refrigerant vaporizes in the evapo-

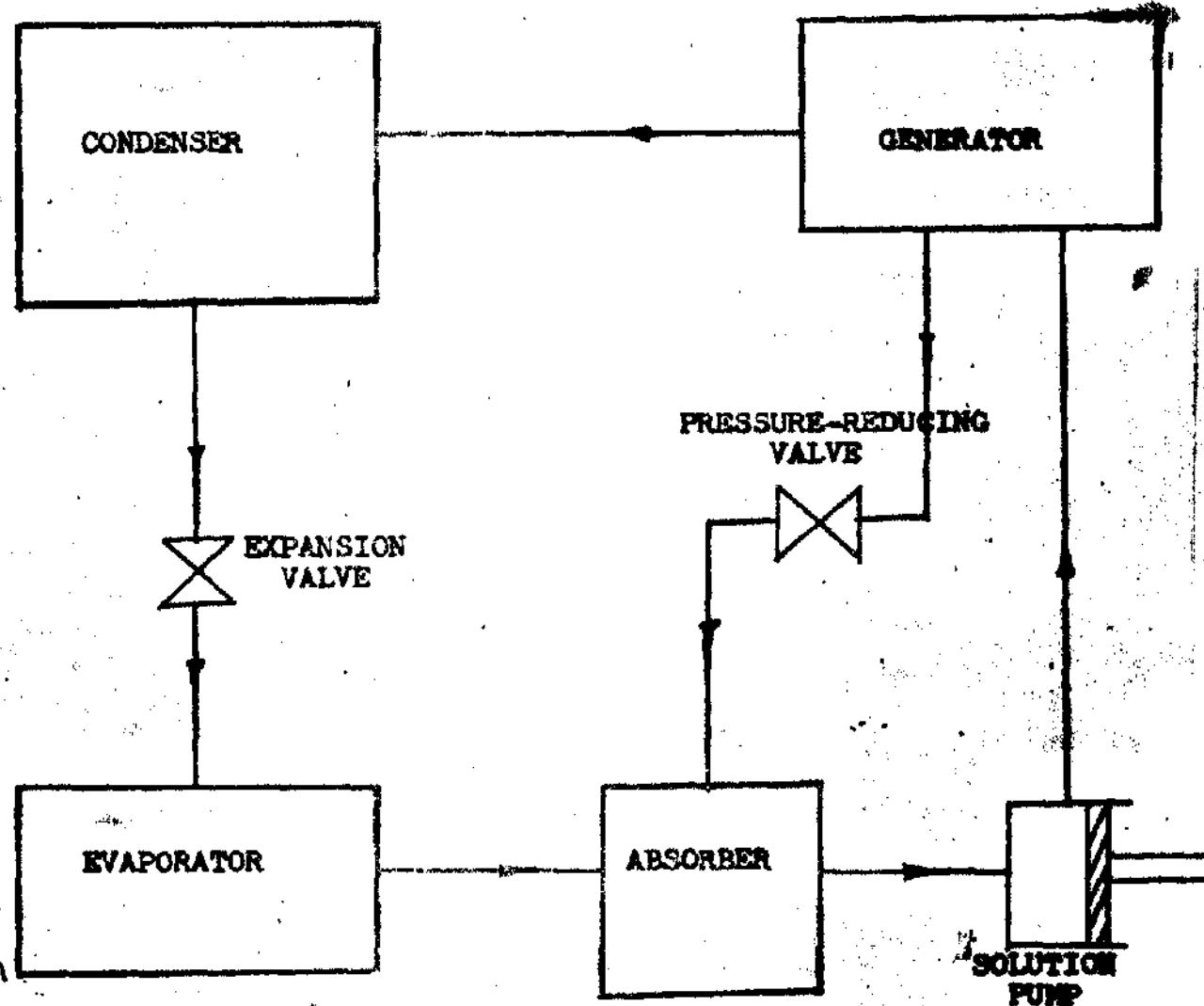


Fig. 1.1: The Basic Absorption Refrigeration Cycle.

rator by absorbing latent heat from the material being cooled and the resulting low pressure vapour then passes from the evaporator through an unrestricted passage to the absorber, where it is absorbed by, and goes into solution with, the absorbent.

The refrigerant flows from the evaporator to the absorber because the vapour pressure of the absorbent-refrigerant solution in the absorber is lower than the vapour pressure of the refrigerant in the evaporator. It is the vapour pressure of the absorbent-refrigerant solution in the absorber that determines the pressure on the low-pressure side of the system and consequently the vaporizing temperature of the refrigerant in the evaporator.

As the refrigerant vapour from the evaporator is dissolved into the absorbent, the volume of the refrigerant is decreased and the heat of absorption is released. In order to maintain the temperature and vapour pressure of the absorbent solution at the required level, the heat released in the absorber, which is equal to the sum of the latent heat of condensation of the refrigerant vapour and the heat of dilution of the absorbent must be discarded to the surroundings, usually by air-cooling or water-cooling.

The refrigerant vapour dissolving into the absorbent solution increases the strength and vapour pressure of the solution and so, it is necessary to

continuously reconcentrate the solution in order to maintain the vapour pressure of the solution at a level low enough to provide the required lower temperature and pressure in the evaporator. The reconcentration is accomplished by continuously removing the strong absorbent solution from the absorber and recirculating it through the generator, where most of the refrigerant vapour is boiled off by the application of heat and the resulting weak solution is returned to the absorber to absorb more refrigerant vapour from the evaporator.

Since the absorber is on the low-pressure side of the system and the generator is on the high-pressure side, the strong solution must be pumped from the absorber to the generator and the weak solution returned to the absorber through a pressure reducing valve. While the pressure of the absorbent solution is increased from the low-side pressure to the high-side pressure as the solution is pumped from the absorber to the generator, no compression of the refrigerant occurs in the process since the compression of the refrigerant has already been accomplished in the absorber. Consequently, the power required by the solution pump is relatively small.

In the generator, the refrigerant is separated from the absorbent by heating the solution and vaporizing the refrigerant. The resulting high pressure

refrigerant vapour then passes to the condenser, where it is condensed by giving up latent heat to the condensing medium, after which it is ready for recirculation to the evaporator. The weak absorbent solution left in the generator is returned to the absorber, through the absorber return pipe. The relative strength of the weak solution is controlled by the amount of heat supplied to the generator.

An improvement in the system efficiency can be achieved by providing for a heat exchange between the strong solution going to the generator and the high temperature weak solution returning from the generator to the absorber. Since the temperature of the strong solution going to the generator is increased, while the temperature of the weak solution going to the absorber is decreased, the heat exchange results in a reduction in both the heat supplied to the generator and the cooling required by the absorber.

As we have seen, an absorption system makes use of the ability of one substance (the absorbent) to absorb relatively large volumes of the vapour of another substance, usually a liquid (the refrigerant). The absorbent has the ability to absorb large quantities of vapour when cold and give them up when heated. There are two main types of absorption system in use today:- (a) the ammonia-water system and (b) the water-

lithium bromide system. In the water-lithium bromide system, water serves as the refrigerant and a solution of lithium bromide salts in water serves as the absorbent. Because of easier availability of materials and data, the ammonia-water combination was adopted for the project. We shall therefore proceed to examine the ammonia-water system.

1.5. THE AMMONIA-WATER ABSORPTION SYSTEM

To be suitable for use in an absorption refrigeration system, there are certain criteria that the refrigerant-absorbent combination should meet. Obviously, the absorbent must have a strong affinity for the refrigerant vapour and the two must be mutually soluble over the desired range of operating conditions. It is desirable that the two fluids should be safe, stable and non-corrosive both individually and in combination. Ideally, the absorbent should have a low volatility so that the refrigerant vapour leaving the generator will contain little or no absorbent, and working pressures should be reasonably low to minimize equipment weight and leakage. The refrigerant should also have a reasonably high latent heat value so that the required refrigerant flow rate is not excessive.

The ammonia-water combination, wherein the ammonia is the refrigerant and water is the absorbent meets many

of these important criteria well. The water absorbent has a very strong affinity for the ammonia vapour, and the two are mutually soluble over a wide range of operating conditions. Both fluids are highly stable and are compatible with most materials found in refrigeration systems. One notable exception, however, is copper and its alloys which are not suitable for use in any ammonia system. The ammonia refrigerant has a high latent heat value but is slightly toxic, which limits its use in air conditioning applications, and operating pressures are relatively high.

A major problem with the ammonia-water system is the fact that the absorbent (water) is reasonably volatile, and so the refrigerant (ammonia) vapour leaving the generator will usually contain appreciable amounts of water vapour, which if allowed to pass through the condenser and go into the evaporator, will raise the evaporator temperature and reduce the refrigerating effect by carrying unvaporized refrigerant out of the evaporator. To reduce this problem and thereby improve the efficiency of the ammonia-water system, a rectifier is incorporated to the generator (Fig. 4.3). Its function is to remove the water vapour from the mixture leaving the generator before it reaches the condenser.

The rectifier is essentially a distillation column that is attached at the top of the generator. As the

ammonia and water vapours coming off the generator rise up through the rectifier, they are cooled and the water vapour, having the higher saturation temperature, condenses and drains back to the generator, while the ammonia vapour continues to rise and leaves at the top of the rectifier. The rectifier is also either air-cooled or water-cooled.

1.6. OTHER FEATURES OF THE ABSORPTION SYSTEM

The vapour-absorption system can be classified into two main groups:- (a) Continuous system and (b) Intermittent system.

In the continuous system, all the processes of evaporation, absorption, and condensation take place simultaneously. A solution pump is sometimes required between the absorber and generator for the circulation. The continuous system can be divided further into two: (i) vapour absorption with little pumping and (ii) vapour absorption with no pumping. The ammonia-water system already described is an example of the continuous system with little pumping. An example of the vapour absorption system with no pumping is the electrolux refrigeration or domestic absorption system. It employs ammonia and water in conjunction with hydrogen gas to balance the low and high side pressures. Because of the presence of the hydrogen and the partial pressure that it exerts on the low side of the system (evaporator and absorber), the partial pressure exerted by the ammonia

vapour in these points will be lower than that exerted by the ammonia vapour in the generator and condenser where no hydrogen gas is present. As a result, the ammonia can evaporate at a low pressure and temperature in the evaporator and at the same time condense at high pressure and temperature in the condenser.

As the name implies, intermittent systems are those whose cycles operate in two stages. The sources of energy or heat are usually non-continuous. No circulating pump is needed and hence electricity is not required. Generally, heat is supplied to the generator which also acts as the absorber in the first stage of the cycle and the resultant refrigerant vapour is condensed and stored in a properly insulated liquid receiver. In the second stage of the cycle when refrigeration is required, the refrigerant is evaporated at the evaporator and the resultant vapour flows to the absorber/generator to be absorbed by the absorbent. The evaporator is enclosed in a brine solution which is required to provide hold-over capacity. Control between the two stages is achieved by means of valves. The refrigerant normally used in this type of system is ammonia. The absorbents for ammonia with suitable thermodynamic and physical properties for refrigerating with solar energy are water, sodium thiocyanate (NaSCN) and Calcium Chloride. Sodium thiocyanate and Calcium Chloride are solid absorbers.

The major problem encountered with intermittent systems is the difficulty with the thermo-circulation.

1.7. REVIEW OF SOME OTHER SOLAR COOLING METHODS

1.7.1. Night Radiation Cooling:- Cooling can be produced by storing "Cold" from the cooler night air in some areas by radiation to the night sky. This is known as Night Radiation Cooling. Here, a collector that has a net radiation exchange to the night sky is employed. Water in such a collector, for example, will then be chilled below the ambient night temperature. The amount of cooling depends on the infra-red nature of the collector. For a significant effect to be obtained, such a collector cannot have glass covers. A collector with no cover can cool (as was demonstrated by Bliss with the University of Arizona house⁽³⁵⁾), but such a cover is not efficient because a night wind can warm the surface reducing its net cooling effect. Earlier studies show that only about 20% of the demand would be met in this manner. This means that the solar collector in the night radiation cooling mode would be of only marginal effectiveness in a hot climate. Therefore some additional means of cooling would be required for ideal comfort.

A nocturnal cooling storage system first tried in a desert in the South-West United States has been more recently tested and developed in Australia⁽³⁵⁾. The system consists of a large bed of rocks cooled by

drawing cool night air across them and exposing them to night sky radiation. During the day, warm inside air may be cooled by circulating it through the rock bed. Augmented cooling can be achieved by drawing the night air through a porous surface having a high emittance at low temperature and facing the night sky. Such a system operates best in a desert climate, where the night skies are clear, humidity is low, and the diurnal temperature variation may be 10°C . Night radiation cooling is a solar related system even though it is not solar operated.

1.7.2. Evaporative Cooling:- In a dry climate, cooling can be done effectively with an evaporative cooler. This type of cooler functions by evaporating water into the air-stream flowing into the space. The amount of cooling is dependent upon the humidity of the air before the evaporation stage and the maximum tolerable humidity after evaporation. When the entering air is already loaded with water vapour, the cooling will be slight and the humidity unbearably high after evaporation.

In a single stage evaporative cooler, a fan pulls outside air through a porous wet pad. The air-flow picks up some water vapour, in the process, absorbing latent heat of vaporization of water that has been evaporated. The result is that the air-flow now has a greater partial pressure of water vapour and a significantly higher relative humidity since the air tempera-

ture has also been lowered. At low humidity for the entering air, more weight of water vapour per kilogram of air is evaporated and a lower air temperature results for a given final humidity. At high humidity, less water is evaporated and the cooling effect is less, so the comfort index is poor.

The variant of the evaporative cooler known as a two-stage evaporative cooler avoids this problem by injecting the cooled but humid air into a heat exchanger rather than into the space. The air for the space to be cooled is then cooled by contact with the heat exchanger, thus adding no additional water burden to the inside air.

1.7.3. Munter Environmental Control Cooling

System (MEC): This system⁽¹⁸⁾ named after the inventor is also based on the principle of cooling of air when water is evaporated into it. It is the basis for the old swamp-box coolers of the desert South-West U.S.A. The addition that Munter introduced was to first dry the ambient air so that after evaporation, the humidity is restored to a comfortable level along with the cooled air. Its success depends very much on the efficient drying of the air and regeneration of the dryer. The basic component of the Munter drying wheel is a thin pancake made of rolled asbestos, corrugated so that air can flow through the small corrugations along the thin dimension of the pancake. The asbestos corrugations are covered with

lithium chloride which is hygroscopic and stable, absorbing moisture when cold and releasing it when heated.

1.7.4. Solar Mechanical Systems:- A solar cooling system that has received some attention in recent years couples a conventional air conditioning system, involving a vapour compression refrigerator, with a solar-powered prime mover. It may be done by converting solar energy into electricity by means of photovoltaic devices, then using the electricity in an electric motor to drive the vapour compressor. The efficiency of conversion of solar energy into electricity by photovoltaic means is in the order of 10%. This low efficiency, coupled with the efficiency of the mechanical system, results in low overall efficiencies.

The solar-powered prime mover can also be a Rankine engine. In this case, the combined system would have an overall efficiency of 17 - 23%, which makes it more attractive than the photovoltaic/electric motor system. The problems associated with both systems are basically the problems associated with converting mechanical energy from solar energy, to drive a compressor. A schematic representation of a simple Rankine Cycle Cooling System is shown in Fig.1.2.

Energy from the collector is stored, then transferred to a heat exchanger which, in turn, transfers energy to the heat engine. The heat engine drives a

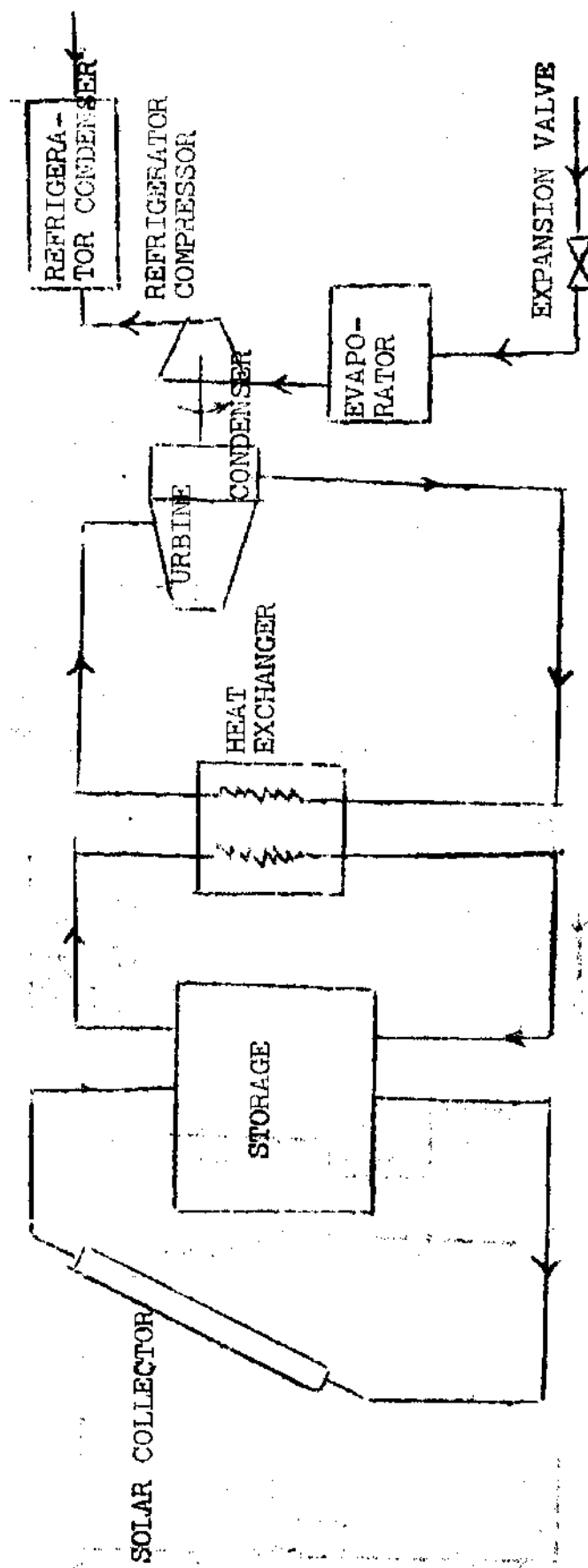


Fig. 1.2.: Rankine Cycle Solar Mechanical Cooling System

vapour compressor, finally producing a cooling effect at the evaporator. The efficiency of the solar collector decreases as the operating temperature increases, whereas the efficiency of the heat engine for the same system increases as the operating temperature increases, giving an optimum operating temperature for steady-state operation. The thermodynamic analysis of such a cycle is not straight-forward for unsteady operation. The prediction of component performance at off-design conditions is very difficult and the matching of all components into a complete system which optimizes the overall performance is currently being researched.

1.8. WHY VAPOUR-ABSORPTION, AMMONIA-WATER SYSTEM WAS ADOPTED FOR THE PROJECT

The different methods of obtaining solar cooling already described can be grouped into three classes: (a) Solar absorption cooling, (b) Solar mechanical systems, and (c) Solar related systems, which are those that operate with solar principles but are not solar powered. There are variations within each class and many have been described in the preceding sections.

For the first two classes (a) and (b), which are solar powered, the solar absorption cooling has been found to be more suitable for solar refrigeration. The solar mechanical system has a very low efficiency

because of the problem associated with converting solar energy to mechanical energy to drive a compressor. Since without high quality concentration, solar energy cannot heat a working fluid much above 100°C , the efficiency of converting this energy into work is low compared with its direct utilization as heat to evaporate the refrigerant. Hence the heat required to evaporate the refrigerant in the generator can be supplied directly by solar energy. Consequently, space cooling with flat plate collector seems to favour the use of an absorption process.

The ammonia-water combination was chosen because ammonia and water are more easily available than the other combinations. Also, property tables and charts for ammonia-water combination are available. As stated earlier, the water absorbent has a very strong affinity for the ammonia vapour, and the two are mutually soluble over the range of operating conditions considered. The ammonia refrigerant also has a high latent heat value.

1.9. THE AIM OF THE PROJECT

The purpose of the project is to examine various ways of using solar energy for refrigeration, choose an appropriate system, and then to design and construct the system. Emphasis was to be on simplicity and the use of easily available materials, as much as

possible and independence from the use of other sources of energy. Air cooling by natural convection was to be adopted instead of water-cooling because of easier availability of air for the cooling and simplicity. The solar absorption system is aimed at being used in the rural areas or areas with little or no electricity supply. The prospects in the utilization of the sun's energy depend, to a large extent, on its availability. Nigeria lies within the tropics where solar energy is abundant and so, it is hoped that the development of this system will go a long way in helping the rural community of this country to benefit from the uses of refrigeration, particularly in the area of food preservation.

1.10.CONCLUSIONS

This chapter introduces the reader to the basic refrigeration principles, methods, and application to society and industry. It also provides the reader with the aim of the project and the guiding principle of simplicity, knowing fully well the constraints such as unavailability of materials.

CHAPTER TWO

LITERATURE SURVEY OF SOME REPORTED STUDIES ON SOLAR COOLING SYSTEMS

Since recent large price increases in the conventional energy resources, reduced availability of fossil fuels, and the energy crisis of the early seventies, there has been a marked increase in solar energy research and development all over the world. Considerable work has been done on solar cooling. The review presented in this chapter shows how much has been done and level of successes achieved. In particular, this review is used as a justification for the present effort. The review is done under the following sub-headings:

1. Intermittent Absorption Cooling Systems;
2. Continuous Absorption Cooling Systems;
3. Evaporative and Other Special Studies and Continuous Systems.

2.1. INTERMITTENT ABSORPTION COOLING SYSTEMS

In 1958, Williams, D. A.⁽⁶⁰⁾, working with a group of Scientists, used a parabolic solar reflector to focus sunlight onto a steel vessel containing ammonia-water solution in a closed system. The reflector used was 4 ft. in diameter. The ammonia was driven out with the heat and condensed in a small vessel connected by a steel tube or flexible rubber

hose capable of withstanding high pressures. The pressure of the heated ammonia reached over 10 atmospheres and was condensed in an external vessel which was cooled in a pail of water. After the ammonia was driven out and condensed, the two-vessel system which weighed about 25 lb. was taken into the house and the small vessel of ammonia liquid was put into an insulated box. When removed from the focused sunlight, the water solution cooled and re-absorbed the ammonia. The evaporation of the liquid ammonia kept the insulated box cold. An exposure of 4 hours in the bright sun kept the refrigerator below 5°C during the 24-hour day.

Trombe and Foex⁽⁵⁴⁾ produced ice by heating ammonia water solution in a central tube at the focus of a large parabolic-cylindrical focusing collector during the day and allowing the ammonia that is driven out and liquefied to be vaporized and re-absorbed during the night.

In 1962, Chinnapa⁽⁹⁾ described an intermittent ammonia-water refrigerator in which he made small quantities of ice with a flat plate collector containing three glass plates 5 ft. by 3.5 ft. The coefficient of performance was about 0.06, operating in a hot climate. The maximum temperature of regeneration was 100°C with the flat plate collector and the temperature of the absorbing solution was 33 to 37°C . Earlier on

he had studied⁽⁸⁾ the efficiency of refrigeration using both the ammonia-lithium nitrate system and the ammonia-water system. The equipment consisted of a condenser-evaporator vessel, and an absorber-generator vessel. The heat source was oil bath and bunsen burner for higher temperatures. The actual condensation pressure was found to be higher and the actual weight of distillate obtained was less than expected from theory. The coefficient of performance for both systems were close to the theoretical values while the actual cooling per kilogram of refrigerant was 5 - 15% less than the theoretical value. The ammonia-lithium nitrate system gave a better coefficient of performance while the ammonia-water system gave a better refrigeration per unit mass of solution.

Blytas, G. C. et. al⁽⁶⁾ carried out an extensive investigation to find the salt with the best physico-chemical properties for intermittent solar cooling with ammonia. Ammonia nitrate, lithium nitrate, the alkali halides, and sodium and lithium thiocyanate were among the salts investigated theoretically and experimentally. The nitrates dissolved in liquid ammonia, have many desirable properties, but thermodynamically there is a chance that spontaneous reaction will produce nitrogen gas and steam with explosive violence. Preliminary tests failed to show explosion hazards, but the general use of concentrated ammonia

solutions of nitrates were still being investigated. Ammonia salts are highly soluble, but in solution, they are corrosive to iron containers, although they can be used in aluminium containers. Sodium thocyanate was finally chosen as the best salt to dissolve in ammonia for use in solar cooling refrigeration. It was found that the solutions have suitable thermodynamic properties with very high solubilities, low vapour pressures and high heats of vaporization. They are chemically stable and inert, inexpensive and can be used in iron vessels. They have high heat conductivities and low viscosities.

Farber⁽²²⁾ in reporting the University of Florida programme in solar refrigeration and air conditioning, mentioned the design of an intermittent absorption system to investigate the feasibility of using low temperature heat sources such as is obtainable with flat plate solar collectors. It was found that the ammonia-water system with high concentration (0.5 - 0.6) by weight could operate well within temperatures obtained with flat plate solar collectors. Other refrigerant-absorbent combinations investigated and found to be promising were methyl chloride-methyl acetate, sulphur dioxide-chloroform and ethyl flouride-ethyl iodide. The ammonia-water combination seemed to have the best properties, giving the highest refrigeration rate per kilogram. Water temperatures of as

low as 54°C could be used to heat the generator at the highest ammonia concentration. The coefficient of performance of about 0.6 was obtained.

Chung⁽¹¹⁾ reported the test of sodium thiocyanate-ammonia system for solar refrigeration in the laboratory. A 20 lb. steel cylinder containing 12 lb of solution with about equal weights of sodium thiocyanate and ammonia was placed at the focus of a 4 ft. parabolic solar cooker. The cylinder was connected by a flexible pressure hose to a smaller vessel immersed in water at room temperature for the liquefaction of the ammonia. After about 4 hours, much of the ammonia had been driven out and the cylinder was removed and brought to room temperature. The vessel of liquid ammonia was placed overnight in water contained in an insulated box, and 9 lb. of ice were produced. A second batch of ice could have been made with another set of vessels using the solar collector for a second 4-hour period of the day.

2.2. CONTINUOUS ABSORPTION COOLING SYSTEMS

Farber et al⁽²³⁾, reported the operation and performance of the continuous absorption refrigeration system, using ammonia and water. The system was operated by flat plate solar absorbers. Operating continuously over the greater part of the day, it produced 2.4 tons of refrigeration with peak loads of 3.4 tons of refrigeration. During steady state operation,

the effect over the heat input was 0.57. This could be raised to 0.72 with minor system modifications, namely heat exchanges between absorber and generator and the absorption system. Farber also presented a paper at the International Solar Energy Society Conference in Melbourne in 1970 in which he reported of the design and performance of a compact solar refrigeration system. The system consisted of a solar-operated ammonia-water cooler, using a flat plate collector. The temperature range of water supply to the generator was generally 60 - 93°C. The ammonia concentration in the absorber and generator were typically 58% and 39%. The operation was continuous and at varying rates, depending on the energy supply.

At the University of Nigeria, Nsukka, Ezeilo⁽²¹⁾ in 1978, compared the performance of a flat plate solar collector and a solar concentrator for supplying heat for generation in an adapted kerosene fired domestic electrolux refrigerator. The concentrator was a wooden parabola with small strips of reflecting mirror glued onto the forward surface while the flat plate collector was of a single glass cover type with blackened corrugated copper plate to which was soldered copper pipes. The flat plate collector was found to be more effective than the parabolic concentrator as it gave a temperature drop of about 14°C while the concentrator gave a temperature drop of only about 6°C below ambient.

At the International Solar Energy Congress in Los Angeles in 1975, Anderson, H.⁽²⁾ reported the analysis of a solar-powered butane vapour jet pump (ejector) air conditioning system and compared its performance with solar-powered absorption and Rankine cycle/compressor air conditioning systems. The vapour jet system was shown to be nearly as efficient as a Rankine cycle/compressor system and operates at lower supply temperatures than absorption systems. Anderson suggested that its low cost, low maintenance and simplicity warrant further studies into its solar energy applications.

Chinnapa⁽¹⁰⁾ devised aqua-ammonia vapour absorption cycles employing resorption and multi-staging with up to 90% improvement in coefficient of performance when compared with the conventional cycle. Using known values for flat plate solar collector heat losses, the performance of the solar collector-refrigerator combination was assessed for a year using insulation and ambient temperature data for George Town, Guyana and Colombo, Ceylon. When the performance of the proposed cycles was compared to that of the conventional cycle, it was found that the reduction in solar collector cost more than offset the increased cost of the refrigerator plant.

Experiments made by Chung et al⁽¹²⁾ at the Solar Energy Laboratory, University of Wisconsin, Madison,

Lithium Bromide-Water absorption cooler with energy input from flat plate collector operated with overall coefficient of performance (cooling to incident solar energy) at 0.15 for 6-hour operating periods. The cooler cycle C.O.P. was about 0.6 with hot water supply at above 180 F with collector operation at 25% efficiency. The cooler was found to have characteristics desirable for solar operation and feasibility of combination was demonstrated.

A proto-type system⁽¹⁷⁾ was designed which performs off-peak air conditioning using thermal energy storage. The condensing unit was sized for continuous operation on the design day and draws 50% as much power as a conventional system with identical air side characteristics. Since the thermal energy storage unit was re-charged during the cool night hours, the same room comfort conditions could be maintained at a savings of 6% in the total amount of energy consumed. The unit operated by evaporating the refrigerant through the use of thermal storage, and allowing it to evaporate a second time at 5°C. The air was precooled by the first evaporator and dehumidified and cooled by the second evaporator. When an engineering demonstration of this proto-type was built, it was found to perform in accordance with the analysis. The operation successfully demonstrated a practical method of reducing electric utility peak summer demand.

single-stage lithium bromide-water absorption air conditioner heated from a flat plate solar collector were compared theoretically to those for an ammonia-water system. The lithium bromide-water system was selected as the preferred one. Double-stage absorption systems with their improved performance were shown theoretically to require generator temperatures that are too great to make them attractive for use with flat plate collectors. Dual series-connected systems which require no cooling tower for heat rejection were shown by analysis to have a low coefficient of performance. Systems utilizing refrigerant storage and a heat rejection buffer between a cooling tower and the absorber and condenser were shown to require smaller cooling towers than conventional units. Operation with an air heat exchanger rather than the cooling tower in such a system was shown to yield acceptable system performance with a small reduction in the fraction of the cooling load, which can be met by solar energy.

2.3. EVAPORATIVE AND OTHER SPECIAL STUDIES AND CONTINUOUS SYSTEMS

In 1968, Hay and Yellot⁽²⁸⁾ constructed and operated a naturally air conditioned building in Phoenix, Arizona. The building involved air conditioning system using solar radiation as heat source and atmosphere as heat sink. Novel features included movable insulated

roof which permits water ponds, in thermal contact with metallic ceiling to absorb and retain solar heat in winter and to dissipate summer heat gain by radiation, convection, and evaporation to night sky, with ambient temperatures ranging from sub-freezing to 46°C . The system developed and maintained room temperature between 20°C and 28°C throughout the year without supplementary heating or cooling.

Hay²⁹⁾ reported in 1975 that roof ponds used in the 1930's for cooling buildings, have now, in combination with movable insulation, been developed to be adequate for economic heating and cooling of buildings with light to medium thermal loads. A roof/ceiling pond integrated into the structural design served as roof and ceiling. It functions as a low-cost solar heat collector, heat dissipator, thermal storage unit and as a means of radiant heat transfer for temperature control in underlying or remote rooms.

Bar-Cohen and Ranbach⁽⁴⁾ reported studies of theoretical operational characteristics and limits of a closed cycle, liquid cooling skyward radiator operating in Beer-Sheva, Israel. The studies were by numerical solution of an idealized governing relation. Significant parametric trends, including variation in systems thermal capacity, convective parameters and initial radiator temperature were explored to determine appropriate guidelines for future designs. The results

established the general feasibility of nocturnal radiator thermal control of structure in Beer-Sheva, Israel. Furthermore, evaluation of the optimum thermal storage capacity suggests that effective skyward radiator cooling in Beer-Sheva may require considerably shallower water ponds than used in similar systems in Phoenix, Arizona.

At the International Conference on Heliotechnique and Development in November 1975, Bartoli et al⁽⁵⁾ presented a paper discussing development in the area of natural radiative cooling. The design and operation of two proto-types were described. In the original design, the selective radiator consisted of a polished aluminium sheet coated with a polymetric material. The improved design used selective surfaces with a low emissivity outside the wavelength window of interest. Research areas which may further improve the design were identified.

Hogg⁽³⁰⁾ reported of a successful operation of a switched-bed regenerative cooling system in Australia. It used the evaporation of water in the discharge air to chill the rocks in a switched-bed rock-filled recuperator. The power consumption per square feet of floor area was 0.75 - 1.0 watt instead of 8.8 watts for mechanical refrigeration.

In the interest of eliminating heat loss, More⁽³⁹⁾ made a theoretical study of the various possible configurations which could serve the dual role of a

solar collector and an ammonia generator. The theoretical analysis of the combined solar collector-generator and the results of series of experimental investigations indicated the following advantages resulting from combining the two units into one:- (a) The undesirable heat loss that occurs between solar collector and separate generator of previous solar refrigeration systems is eliminated in the combined collector-generator. (b) The flat plate type solar collector-generator can generate substantial amounts of ammonia even on days when only diffuse solar energy is available. (c) The solar collector-generator developed can generate the equivalent of 30 lbs. of ice per day with a medium size collector area. (d) The combined collector-generator has a relatively short warm-up time.

Prigmore and Barber⁽⁴²⁾ also reported on the development of a demonstration package supplying residential cooling via a solar heated Rankine Cycle. The 3-ton air conditioning system employs a solar collector to warm flowing water which provides input heat to a low temperature organic (R-113) Rankine cycle. Expansion through a high speed turbine-speed reducer drives an available R-12 refrigeration compressor and 3600 rpm motor generator. With a water-cooled R-113 condenser providing a condensing temperature of 35°C and a turbine efficiency design goal of 80%, Rankine Cycle efficiency is 11.5%. An 85% efficient

CHAPTER THREE

SOLAR ENERGY

Solar energy reaches the earth's surface as short-wave electromagnetic radiation in the wavelength band between 0.3 and 3.0 μ m. Its peak spectral sensitivity occurs at 0.48 μ m. The total solar radiation intensity on a horizontal surface at sea level varies from zero at sunrise and sunset to a noon maximum which can reach 1070 W/m² on clear summer days. This endless source of energy, despite its variability in magnitude and direction, can be used in three major processes:

1. Heliothermal, in which the sun's radiation is absorbed and converted into heat which can then be used for many purposes, such as evaporating sea-water to produce salt, or distilling it into drinkable water; domestic hot-water supplies; house heating by warm air or hot water; cooling by absorption refrigeration; cooking; generating electricity by vapour cycles and thermoelectric process; attaining temperatures as high as 3600°C in solar furnaces.
2. Heliochemical, in which the shorter wavelengths can cause chemical reactions, sustain growth of plants and animals, convert carbon dioxide to oxygen by photosynthesis, and cause degradation and fading of fabrics, plastics and paint.

3. Helioelectrical, in which part of the energy between 0.33 and 1.3 μm can be converted directly into electricity by photovoltaic cells. Silicon solar batteries have become the standard power sources for communication satellites, orbiting laboratories, and space probes. Their use for terrestrial power generation is currently under intensive study, with primary emphasis upon cost reduction.

3.1. SOLAR-RADIATION INTENSITY

In space at the average earth-sun distance, 93 million miles (150 million km), solar radiation intensity on a surface normal to the sun's rays is 1350 W/m². This quantity, called the solar constant (I_{sc}), undergoes small periodic variations which affect primarily the short-wave portion of the spectrum. Since the earth-sun distance varies throughout the year, the intensity beyond the earth's atmosphere (I_0) also varies. The great seasonal variations in terrestrial solar-radiation intensity are due to the earth's tilted axis, which causes the solar declination (δ) (i.e. the angle between the earth's equatorial plane and the earth-sun line) to change from 0° on March 21 and September 21 to - 23.5° on December 21 and + 23.5° on June 21.

In passing through the earth's atmosphere, the sun's radiation is partially and selectively absorbed, scattered and reflected by water vapour and ozone, air

molecules, natural dust, clouds, and man-made pollutants. Some of the scattered and reflected energy, reaches the earth as diffuse or sky radiation (I_d). The intensity of the direct normal radiation (I_{DN}) depends upon the clarity and the amount of precipitable moisture in the atmosphere and the length of the atmospheric path, which is determined by the solar altitude β and expressed in terms of the air mass.

The total irradiation (I_t) reaching a terrestrial surface is the sum of the direct, diffuse, and reflected components, i.e.

$$I_t = I_d + I_r + I_{DN} \cos \theta \quad \text{-----}(3.1)$$

where θ is the incident angle between the sun's rays and a line perpendicular to the receiving surface.

I_r is the short-wave radiation reflected from adjacent surfaces.

3.2. SOLAR-OPTICAL PROPERTIES OF TRANSPARENT MATERIALS

When solar radiation with total intensity

I_t falls on a transparent material, part of the energy is reflected (q_R), part is absorbed (q_A) and the remainder is transmitted (q_T). At any instant,

$$I_t = q_R + q_A + q_T = I_t (\rho + \alpha + \tau) \quad \text{-----}(3.2)$$

The sum of the solar-optical properties ρ , α , and τ , must equal unity, but the individual values depend on

the incident angle, and wave-length of the radiation, the composition of the material, and the nature of any coatings which may be applied to the surfaces.

For uncoated single-strength clear window glass, solar transmissivity at normal incidence ($\theta = 0^\circ$) is approximately 0.9, but the transmissivity for long wave thermal radiation ($5\mu\text{m}$) is virtually zero. Thus glass acts as a "heat trap" by admitting solar radiation readily but retaining most of the heat produced by the absorbed sunshine. This "green-house effect" which is also exhibited but to a lesser degree by some plastic films is the basis for most helio-thermal processes and solar flat plate collectors specifically. Heat absorbing glass, which absorbs more than 50% of the incident solar radiation, is widely used by architects to reduce the heat and glare admitted through unshaded windows. Reflective coatings also have been developed to serve similar purposes.

For all types of glass, transmissivity falls and reflectivity rises as θ exceeds about 30° . Absorptivity increases somewhat owing to the increased path length and then drops off sharply toward zero as θ exceeds 60° .

3.3. ABSORPTIVITY AND EMISSIVITY OF OPAQUE SURFACES

Opaque materials absorb or reflect all the incident sunshine. The absorptivity (α) for solar radiation and the emissivity (ϵ) for long wave radiation

at the temperature of the receiving surface are particularly important in heliotechnology. For a grey body, α and ϵ are equal and do not change with wavelength. Most real surfaces have reflectivities and absorptivities which vary with wavelength. Aluminium foil has a consistently low absorptivity and high reflectivity over the entire spectrum from 0.25 to 25 μ m, while black paint has a high absorptivity and low reflectivity. This explains why the surfaces of solar flat plate collectors are painted black while those of parabolic collectors are lined with aluminium foil. White paint, however, has low short-wave (solar) absorptivity, but beyond 3 μ m its absorptivity and reflectivity are virtually the same as for black paint.

Solar collectors require a high $\frac{\alpha}{\epsilon}$ ratio, while surfaces which should remain cool, such as roof-tops or space vehicles should have low ratios, since their objective usually is to absorb as little solar radiation and emit as much long-wave radiation as possible. Special surface treatments have been developed for which the ratio $\frac{\alpha}{\epsilon}$ is above 7.0, making them suitable for solar collectors; others with ratios as low as 0.15 are useful as heat rejectors for space applications.

3.4. FLAT PLATE COLLECTORS

Direct diffuse and reflected solar radiation can be collected and converted into heat by flat plate

collectors such as that used for this project (Fig.4.9). These generally use blackened metal plates which are finned, tubed, or otherwise provided with passages through which water, air, or other fluids may flow and be heated to temperatures as much as $(55 - 86^{\circ}\text{C})$ above the ambient air. The loss from the back of the collector plate can be minimized by the use of adequate insulation. The radiation component of the loss from the upper surface can be reduced by using selective-reflectance coatings with high $\frac{\alpha}{\epsilon}$ ratios and by using covers which are transparent to solar radiation but opaque to long-wave emissions (e.g. glass). Both convection and radiation can be reduced by the use of honeycomb structures in the air space between the cover and the plate.

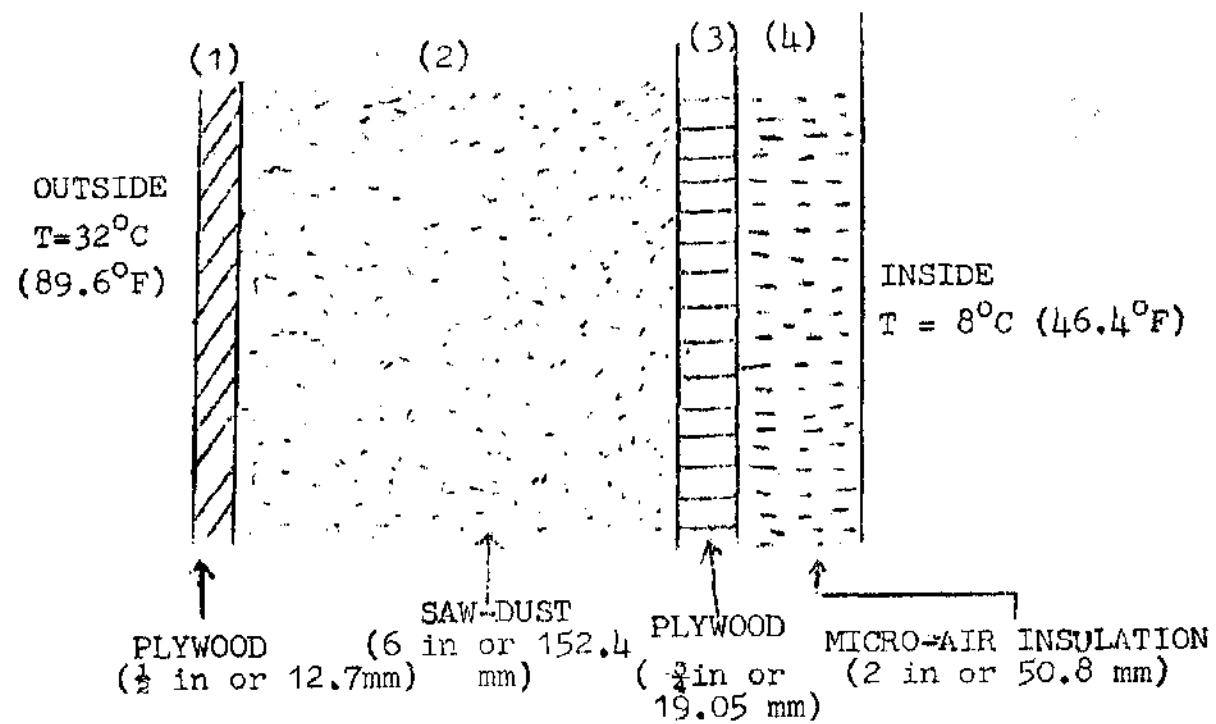


FIG. 4.1.: CONFIGURATION OF THE COLD-ROOM WALL CONSTRUCTION

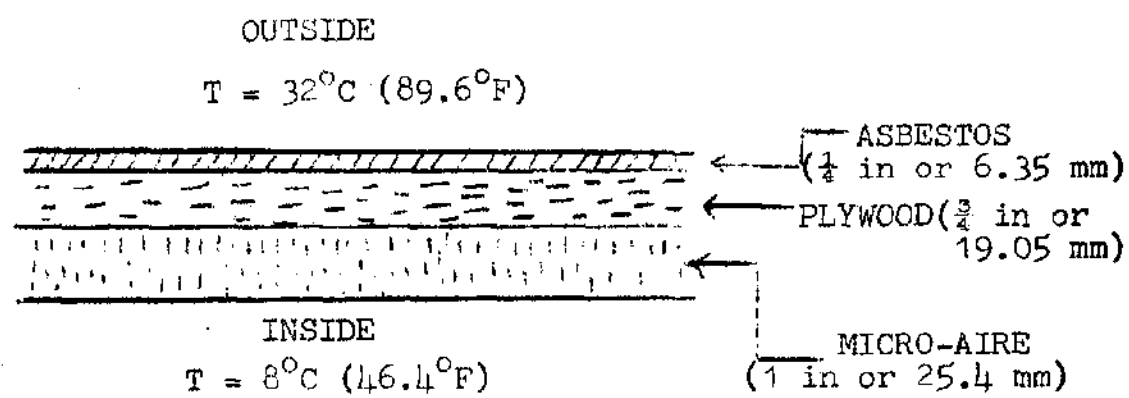


FIG. 4.2.: CONFIGURATION OF THE COLD-ROOM ROOF CONSTRUCTION

Therefore equation (4.2) yields

$$\begin{aligned}\frac{1}{U} &= \frac{1}{4} + \frac{1}{1.6} + \frac{0.5}{0.45} + \frac{1}{1.07} + \frac{2}{12 \times 0.2} + \frac{1}{4} \\ &= 4.002\end{aligned}$$

$$\therefore U = 0.25 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F.}$$

Heat gain through the 4 walls is given by

$$Q_{\text{wall}} = Q_N + Q_W + Q_S + Q_E \text{ -----(4.3)}$$

Making allowance for solar radiation assuming unpainted wood as type of surface for the walls we obtain the relevant correction factors from⁽¹⁶⁾. Consequently, for each of the 4 walls we apply equation (4.1) and obtain

$$\begin{aligned}Q_N &= \frac{68.5 \times 56}{144} \times 0.25 \left[(89.6 - 46.4) + 0 \right] \\ &= 287.7 \text{ Btu/hr.}\end{aligned}$$

$$\begin{aligned}Q_S &= \frac{68.5 \times 56}{144} \times 0.25 \left[(89.6 - 46.4) + 4 \right] \\ &= 314.3 \text{ Btu/hr.}\end{aligned}$$

$$\begin{aligned}Q_E &= \frac{68.5 \times 56}{144} \times 0.25 \times \left[(89.6 - 46.4) + 6 \right] \\ &= 307.7 \text{ Btu/hr.}\end{aligned}$$

$$\begin{aligned}Q_W &= \frac{68.5 \times 56}{144} \times 0.25 \times \left[(89.6 - 46.4) + 6 \right] \\ &= 307.7 \text{ Btu/hr.}\end{aligned}$$

Hence wall heat gain

$$\begin{aligned}Q_{\text{wall}} &= 287.7 + 314.3 + 307.7 + 307.7 \\ &= 1217.4 \text{ Btu/hr.} \\ &= \underline{\underline{0.357 \text{ kW.}}}\end{aligned}$$

4.2.2. Heat Gain Through the Roof: When the top of the cold room is exposed to the outdoors, then for purposes of heat gain, the ceiling is treated as an outdoor wall⁽¹⁶⁾. The configuration of the roof construction is as shown in Figure 4.2. Applying equation (4.2) to this case, we have

$$\frac{1}{U} = \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_i} \text{ -----(4.4)}$$

From Figure 4.2, we obtain

$$x_1 = \frac{1}{4} \text{ in}$$

$$x_2 = \frac{3}{4} \text{ in}$$

$$x_3 = 1 \text{ in.}$$

From tables⁽¹⁶⁾ we obtain the various values of f and k .

The thermal conductivity for plywood $\frac{3}{4}$ in is 1.07

Btu/hr ft² °F. Therefore equation 4.4 yields

$$\begin{aligned} \frac{1}{U} &= \frac{1}{4} + \frac{0.25}{6.5} + \frac{1}{1.07} + \frac{1}{12 \times 0.2} + \frac{1}{4} \\ &= 1.8897 \end{aligned}$$

$$\therefore U = 0.53 \text{ Btu/hr ft}^2 \text{ °F.}$$

Making allowance for solar radiation assuming light coloured surface as the nature of the roof then the heat gain through the roof is

$$\begin{aligned} &= \frac{56 \times 56}{144} \times 0.53 [(89.6 - 46.4) + 9] \\ &= 602.5 \text{ Btu/hr} \\ &= \underline{\underline{0.177 \text{ kW.}}} \end{aligned}$$

4.2.3. Heat Gain Through The Floor: The floor construction is of concrete slab of thickness 5 cm, corkboard slab of thickness 5 cm and concrete finish of thickness 5 cm. We apply equation (4.2)⁽¹⁶⁾ to this situation and obtain

$$\frac{1}{U} = \frac{1}{f_1} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3}$$

If we convert the data obtained from Reference 16 and apply to this case, we have

$$\frac{1}{U} = \left[\frac{1}{68.2} + \frac{5 \times 10^{-2}}{17.3} + \frac{5 \times 10^{-2}}{0.403} + \frac{5 \times 10^{-2}}{17.3} \right] \times 10^{-4}$$

which gives $U = 6.9 \times 10^{-4} \text{ kW/m}^2 \text{ K}$

If we take ground temperature to be 25°C , we have the heat gain through the floor as

$$\begin{aligned} &= 1.42 \times 1.42 \times 6.9 \times 10^{-4} (25 - 8) \\ &= 236.5 \times 10^{-4} \text{ kW} \\ &= \underline{\underline{0.024 \text{ kW}}} \end{aligned}$$

4.2.4. Usage Load: When coolers are used for general-purpose storage, the product load is frequently unknown and /or varies somewhat from day to day, so that it is not possible to compute the product load with any real accuracy as in this case. Hence we apply the method of Usage Load or Usage heat gain⁽¹⁶⁾ which involves the use of load factors which have been determined by experience. This is computed by the equation: Usage Load/24 hours = Interior Volume of Cold room x Usage Factor. -----(4.5)

From the dimensions of the cold room (Section 4.1) we obtain:

$$\begin{aligned}\text{Interior Volume} &= \frac{56}{12} \times \frac{56}{12} \times \frac{68.5}{12} \\ &= 124.315 \text{ cu.ft.}\end{aligned}$$

Temperature difference between the inside and the outside of the cooler = $89.6 - 42.8 = 46.8^{\circ}\text{F}$.

From Table 4.1 obtained from Reference 16 and shown attached, Usage factor for Average Usage = 78.46. The table was extracted by R. J. Dossat from ASHRAE data Book, Fundamentals Volume, 1972 Edition. Therefore equation (4.5) gives

$$\begin{aligned}\text{Usage Load} &= \frac{124.315 \times 78.46}{24} \text{ Btu/hr.} \\ &= 406.4 \text{ Btu/hr} \\ &= \underline{\underline{0.1191 \text{ kW}}}\end{aligned}$$

4.2.5. Total Cooling Load: The total cooling load is the sum of the Wall Gain Load, Heat Gain Through the Roof, Floor and the Usage Load. Hence

$$\begin{aligned}Q_{\text{TOTAL}} &= 0.357 + 0.177 + 0.024 + 0.1191 \text{ kW} \\ &= 0.6671 \text{ kW.}\end{aligned}$$

Applying a safety factor of 10%⁽¹⁶⁾ we obtain

$$\text{Total Cooling Load} = \underline{\underline{0.7338 \text{ kW}}} \text{ or } 2503.82 \text{ Btu/hr.}$$

4.3. THERMODYNAMIC PROPERTIES AND FLOW RATES

We shall use the "Enthalpy Concentration Chart for Ammonia-Water Solution"⁽⁵²⁾ and refer to state points in Figure 4.3. We shall assume that there are

TABLE 4-1 Usage Heat Gain (REF. 16)

Volume cu ft	Service*	Temperature difference (ambient temp minus storage room temp), F deg										
		1	40	50	55	60	65	70	75	80	90	100
20	Average	4.68	187	234	258	281	305	328	351	374	421	468
	Heavy	5.51	229	276	303	331	358	386	413	441	496	551
30	Average	3.30	132	165	182	198	215	231	248	264	297	330
	Heavy	4.56	182	228	251	274	297	319	342	365	410	456
50	Average	2.28	91	114	126	137	148	160	171	182	205	228
	Heavy	3.55	142	177	196	213	231	249	267	284	320	355
75	Average	1.85	74	93	102	111	120	130	139	148	167	185
	Heavy	2.88	115	144	158	173	188	202	216	230	259	288
100	Average	1.61	64	81	84	97	105	113	121	129	145	161
	Heavy	2.52	101	126	139	151	164	176	189	202	227	252
200	Average	1.38	55	69	76	83	90	97	103	110	124	138
	Heavy	2.22	90	111	122	133	144	155	166	178	200	222
300	Average	1.30	52.0	65	71.5	78	84.5	91	97.5	104	117	130
	Heavy	2.08	83.2	104	114	125	135	146	156	166	187	208
400	Average	1.24	49.6	62	68.2	74.4	80.6	86.8	93	99.2	112	124
	Heavy	1.96	78.4	98	108	118	128	137	147	157	176	196
500	Average	1.21	48.4	60.5	66.6	72.6	78.7	84.7	90.7	96.8	109	121
	Heavy	1.87	74.8	93.5	103	112	122	131	140	150	168	187
600	Average	1.17	46.8	58.5	64	70	76	82	88	94	105	117
	Heavy	1.85	74.0	92.5	102	111	120	130	139	148	167	185
800	Average	1.11	44.4	55.5	61.1	66.6	72.2	77.7	83.3	88.8	100	111
	Heavy	1.76	70.4	88.0	96.8	106	115	123	132	141	158	176
1,000	Average	1.10	44.0	55.0	60.5	66	71.5	77	82.5	88	99	110
	Heavy	1.67	66.8	83.5	91.9	100	108	117	125	134	150	167
1,200	Average	.995	39.8	49.8	54.7	59.7	64.7	69.7	74.7	79.6	89.6	99.5
	Heavy	1.58	63.2	79.0	86.9	94.8	103	111	119	126	142	158
1,500	Average	.920	36.8	46.0	50.6	55.2	59.8	64.4	69	73.6	82.8	92
	Heavy	1.50	60.0	75.0	82.5	90.0	97.5	105	113	120	135	150
2,000	Average	.835	33.4	41.8	45.9	50.1	54.3	58.5	62.7	66.8	75.2	83.5
	Long storage	.775	31.0	38.8	42.6	46.5	50.4	54.3	58.1	62	69.8	77.5
3,000	Average	.750	30.0	37.5	41.3	45.0	48.8	52.5	56.2	60.0	67.5	75.0
	Long storage	.576	23.0	28.8	31.7	34.6	37.3	40.3	43.2	46.1	51.8	57.6
5,000	Long storage	.403	16.1	20.3	22.2	24.2	26.2	28.2	30.2	32.2	36.3	40.3
7,500	Long storage	.305	12.2	15.3	16.8	18.5	19.8	21.4	22.9	24.4	27.5	30.5
10,000	Long storage	.240	9.6	12.0	13.2	14.4	15.6	16.8	18.0	19.2	21.6	24.0
20,000	Long storage	.187	7.48	9.35	10.3	11.2	12.2	13.1	14.0	15.0	16.8	18.7
	Long storage	.178	7.12	8.90	9.79	10.7	11.6	12.5	13.4	14.2	16.0	17.8
50,000	Long storage	.176	7.04	8.80	9.68	10.6	11.5	12.3	13.2	14.1	15.8	17.6
	Long storage	.173	6.93	8.65	9.52	10.4	11.2	12.1	13.0	13.8	15.6	17.3

* For average and heavy service, product load is based on product entering at F deg above the refrigerator temperature; for long storage the entering temperature is approximately equal to the refrigerator temperature. Where the product load is unusual, do not use this table.

From ASHRAE Data Book, Fundamentals Volume 1972 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

no heat and pressure losses in components and lines. The required properties as obtained from the chart and calculations are tabulated in Table 4.2. We shall also assume equilibrium (saturated) conditions for the state points 3, 5, 6, 9, 10 and 2.

Using the Low side and the High side pressures, we can establish the pressures of the state points immediately. The points 3, 6, 10 are equilibrium points and so we can locate them on the chart since we know their temperatures and pressures. For state point 9, we should, however, note that even though we are assuming it to be saturated vapour for simplicity, it is actually superheated vapour. After locating states 3, 6, 10, we immediately, by inspection and induction, get the concentrations (x) for the remaining states 1 through 10. Consequently, we are able to get the enthalpies for states 2, 3, 5, 6, 9 and 10, as well as their temperatures.

Heat balance for the evaporator gives

$$Q = m_1 (h_2 - h_1)$$

where Q is the heat load for the cold room and $h_1 = h_{10}$ because of throttling.

$$\therefore \text{Mass flow rate at state 1 } m_1 = \frac{Q}{h_2 - h_1}$$

$$\text{i.e. } m_1 = \frac{2503.82}{720 - 130}$$

$$\begin{aligned} \therefore m_1 &= m_2 = m_9 = m_{10} = 4.24 \text{ lb/hr} = 0.07 \text{ lb/min} \\ &= 5.35 \times 10^{-4} \text{ kg/s.} \end{aligned}$$

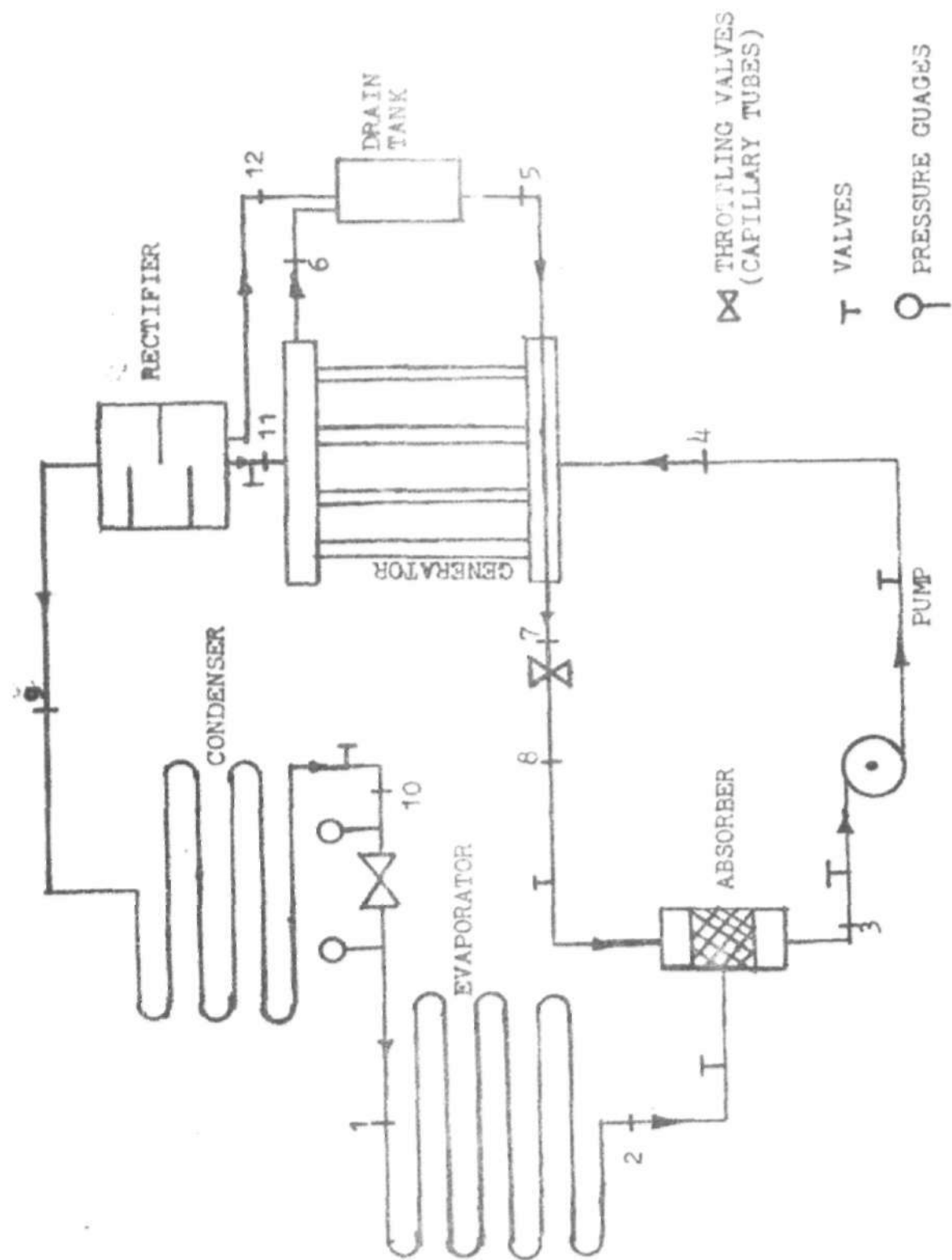


FIG.4.4.3.: THE AQUA-AMMONIA REFRIGERATION CYCLE

For the absorber, we have

$$m_2 + m_8 = m_3 \text{ -----(4.6)}$$

$$m_2 x_2 + m_8 x_8 = m_3 x_3 \text{ -----(4.7)}$$

Substituting values, equation (4.6) becomes

$$4.24 + m_8 = m_3$$

while equation (4.7) becomes

$$(4.24)(0.99) + 0.53 m_8 = 0.57 m_3$$

Solving the two equations, we obtain

$$m_3 = 43.49 \text{ lb/hr} = 0.725 \text{ lb/min} = 54.8 \times 10^{-4} \text{ kg/s}$$

$$m_8 = 39.25 \text{ lb/hr} = 0.654 \text{ lb/min} = 49.4 \times 10^{-4} \text{ kg/s}$$

Therefore $m_3 = m_4 = m_5 = 0.725 \text{ lb/min}$ or $54.8 \times 10^{-4} \text{ kg/s}$

and $m_8 = m_6 = m_7 = 0.654 \text{ lb/min}$ or $49.4 \times 10^{-4} \text{ kg/s}$

Thus all the flow rates of the system are known.

For the pump, we have

$$h_4 = h_3 + \frac{(P_4 - P_3)V_3}{J} \text{ -----(4.8)}$$

where V_3 is the specific volume of the liquid solution, P is pressure and J is the Joules Mechanical Equivalent of heat. The specific volume V_3 may be found from the empirical relation⁽⁵²⁾.

$$V_3 = (1 - x_3)V_{H_2O} + 0.85 x_3 V_{NH_3}$$

Using steam and ammonia tables, we find

$$\begin{aligned} V_3 &= (1 - 0.57)(0.016) + (0.85)(0.57)(0.026) \\ &= 0.019 \text{ ft}^3/\text{lb.} \end{aligned}$$

Hence equation (4.8) becomes

$$\begin{aligned} h_4 &= 0 + \frac{(201.4 - 77.6)(0.019) \times 144}{778} \\ &= 0.44 \text{ Btu/lb.} = 1.02 \text{ kJ/kg} \end{aligned}$$

For the heat exchanger,

$$m_4 h_4 + m_6 h_6 = m_5 h_5 + m_7 h_7 \text{ -----(4.9)}$$

$$\text{But } m_4 = m_5$$

$$\text{and } m_6 = m_7$$

Then from equation (4.9), we obtain

$$\begin{aligned} h_7 &= \frac{m_4 h_4 + m_6 h_6 - m_4 h_5}{m_6} \\ &= \frac{(0.725)(0.44) + (0.654)(73) - (0.725)(66)}{0.654} \\ &= 0.32 \text{ Btu/lb.} \end{aligned}$$

The process 7 to 8 is a throttling process. Hence

$$h_7 = h_8 = 0.32 \text{ Btu/lb.}$$

Using x_7 , h_7 , x_8 , h_8 , we can locate the points 7 and 8 on the chart.

4.4. PIPE LINE SIZING

Refrigerant Pipe Lines are supposed to be designed bearing mainly three points in mind, (a) Pressure drop, (b) Oil return, and (c) Oil and liquid trapping. In this project, we are not concerned with compressors and lubrication and so we shall design the pipe line on the basis of pressure drop.

TABLE 4.2.: THERMODYNAMIC PROPERTIES AND FLOW RATES

STATE POINT	PRESSURE (P) (Psia)	TEMPERATURE t ($^{\circ}$ F)	CONCENTRATION x (lb NH_3 /lb mix) also (kg NH_3 /kg mix)	ENTHALPY h (Btu/lb mix)	FLOW RATE (lb mix/min)
1	77.6(5.35 bar)	42.8(6 $^{\circ}$ C)	0.99	130(302.4 kJ/kg)	0.07(5.3x10 $^{-4}$ kg/s)
2	77.6(")	48(8.9 $^{\circ}$ C)	0.99	720(1674 ")	0.07(")
3	77.6(")	93.2(34 $^{\circ}$ C)	0.57	0(0 ")	0.725(54.8x10 $^{-4}$ kg/s)
4	201.4(13.89 bar)	87(30.6 $^{\circ}$ C)	0.57	0.44(1.02 ")	0.725(")
5	201.4(")	146(63.3 $^{\circ}$ C)	0.57	66(153.5 ")	0.725(")
6	201.4(")	158(70 $^{\circ}$ C)	0.53	73(169.8 ")	0.654(49.4x10 $^{-4}$ kg/s)
7	201.4(")	88(31.1 $^{\circ}$ C)	0.53	0.32(0.74 ")	0.654(")
8	77.6(5.35 bar)	79(26.0 $^{\circ}$ C)	0.53	0.32(0.74 ")	0.654(")
9	201.4(13.89 bar)	122(50 $^{\circ}$ C)	0.99	732(1702 ")	0.07(5.3x10 $^{-4}$ kg/s)
10	201.4(")	96.8(36 $^{\circ}$ C)	0.99	130(302.4 ")	0.07(")

TABLE 4.3: ENERGY BALANCE FOR THE SYSTEM

COMPONENT	CALCULATIONS	ENERGY (watts)	
		GAINS	LOSSES
Evaporator	Total Cooling Load as Calculated	733.8	
Pump	$W_p = m_1 (h_4 - h_3)$	0.54	
Condenser	$Q_C = m_9 (h_9 - h_{10})$		741.79
Absorber	$Q_A = m_2 h_2 + m_8 h_8 - m_3 h_3$		890.66
Generator/Rectifier	$Q_G - Q_R = m_9 h_9 + m_6 h_6 - m_5 h_5$	899.07	
		1663.41	1632.45

Suction Line:

The suction line should be sized such that the pressure drop will give a maximum temperature drop of ⁽³³⁾ 1°F per 100 ft. When converted to S.I. Units, this gives a maximum temperature drop of 0.555°C per 30.46 m of pipe length. The evaporator temperature is 6°C and so, this should be the suction temperature. From thermodynamic tables of ammonia properties, we obtain that for

$$t = 6^{\circ}\text{C}, \quad P_s = 5.35 \text{ bar}$$

$$\text{and for } t = 4^{\circ}\text{C}, \quad P_s = 4.975 \text{ bar}$$

$$\text{Therefore, for } \Delta t = 2^{\circ}\text{C} \quad \Delta P_s = 0.375 \text{ bar.}$$

$$\text{Hence for } \Delta t = 0.555^{\circ}\text{C},$$

$$\Delta P = \frac{0.375 \times 0.555}{2} \text{ bar.}$$

This means that the desired pressure drop per unit length of piping should be

$$\left(\frac{\Delta P}{L}\right)_s \leq \frac{0.375 \times 0.555 \times 10^5}{2 \times 30.46} \text{ N/m}^3$$

$$\text{or } \left(\frac{\Delta P}{L}\right)_s \leq 341.6 \text{ N/m}^3$$

Assuming that we have pure ammonia at the suction line, then the specific volume $V_s = 0.2351 \text{ m}^3/\text{kg}$

$$\begin{aligned} \therefore \text{Volume flow rate } \dot{V}_s &= \dot{m}_s V_s \\ &= 5.3 \times 10^{-4} \times 0.235 \text{ m}^3/\text{s} \\ &= 12.4 \times 10^{-5} \text{ m}^3/\text{s} \end{aligned}$$

$$\text{Now } \dot{V}_s = \frac{\pi d_s^2}{4} V_s \text{ ----- (4.10)}$$

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$$\therefore \bar{V}_s = \frac{4 \dot{V}_s}{\pi d_s^2}$$

Let us assume that for the pipe flow, the Euler equation holds since density changes can be considered small.

$$\therefore h_f = \frac{\Delta P}{\rho_s g} + \frac{\Delta (\bar{V}_s^2)}{2g} + \Delta Z$$

Neglecting the kinetic and potential energy terms as they will be small, we have

$$h_f = \frac{\Delta P}{\rho_s g} \text{ ----- (4.11)}$$

We can also apply the Darcy-Weisbach formula⁽⁵⁾ to obtain the pressure drop in the pipe flow. Hence

$$h_f = \frac{f_s L \bar{V}_s^2}{d_s \times 2g} \text{ ----- (4.12)}$$

where f_s is the friction factor and L is the length of the pipe.

Combining equations (4.11) and (4.12), we have

$$\left(\frac{\Delta P}{L}\right)_s = \frac{f_s \rho_s \bar{V}_s^2}{2 d_s} \text{ ----- (4.13)}$$

We note that $\left(\frac{\Delta P}{L}\right)_s \leq 341.6$

$$\text{i.e. } \frac{f_s \rho_s}{2 d_s} \left[\frac{4 \dot{V}_s}{\pi d_s^2} \right]^2 \leq 341.6$$

$$\text{or } d_s^5 \geq \frac{16 f_s \rho_s \dot{V}_s^2}{2 \times 341.6 \times \pi^2} \text{ ----- (4.14)}$$

Assuming $f_s = 0.02$ for commercial rough pipe

$$\rho_s = \frac{1}{V_s} = \frac{1}{0.2351} = 4.254 \text{ kg/m}^3$$

$$\therefore d_s^5 \geq \frac{16 \times 0.02 \times 4.254 \times (12.4 \times 10^{-5})^2}{2 \times 341.6 \times \pi^2}$$

$$\text{or } d_s \geq 4.995 \text{ mm } \underline{0.196} \text{ in.}$$

In view of the available pipe diameters in the market and standard sizes, a pipe of internal diameter $\frac{1}{4}$ in. is recommended.

Check For Pipe Strength:- The maximum allowable working pressures for steel pipes of nominal size $\frac{1}{4}$ in range from⁽³³⁾ 1000 psig to 3000 psig for both internal and external refrigeration duty. Hence the pipe size is adequate for our required operation as our maximum pressure is 201.4 psia.

Discharge Line:- For the discharge line also, it should be sized such that the pressure drop will give a maximum temperature drop of⁽³³⁾ 1°F per 100 ft or 0.555°C per 30.46 m of pipe length. The conditions at the condenser inlet should be considered as the conditions at the discharge line. The properties of ammonia from thermodynamic tables give for

$$t = 36^\circ\text{C}, \quad P_s = 13.89 \text{ bar}$$

$$\text{and for } t = 34^\circ\text{C}, \quad P_s = 13.11 \text{ bar.}$$

$$\text{Therefore, for } \Delta t = 2^\circ\text{C}, \Delta P_s = 0.78 \text{ bar}$$

$$\therefore \text{ for } \Delta t = 0.555^\circ\text{C}, \Delta P = \frac{0.78 \times 0.555}{2}$$

Therefore the desired pressure drop per unit length of piping is $\left(\frac{\Delta P}{L}\right)_d \leq \frac{0.78 \times 0.555}{2 \times 30.46} \times 10^5 \text{ N/m}^3$

$$\text{or } \left(\frac{\Delta P}{L}\right)_d \leq 710.6 \text{ N/m}^3$$

We can similarly deduce from equation (4.14) that

$$d_d^5 \geq \frac{16 f_d \mathcal{P}_d \dot{V}_d^2}{2 \times 710.6 \times \pi^2}$$

$$\therefore \frac{d_d^5}{d_s^5} = \frac{f_d}{f_s} \times \frac{\mathcal{P}_d}{\mathcal{P}_s} \times \frac{\dot{V}_d^2}{\dot{V}_s^2} \times \frac{341.6}{710.6} \text{ ----- (4.15)}$$

Now from tables $V_d = 0.0931 \text{ m}^3/\text{kg}$

$$\begin{aligned} \therefore \dot{V}_d &= mV_d = 5.3 \times 10^{-4} \times 0.0931 \text{ m}^3/\text{s} \\ &= 4.9 \times 10^{-5} \text{ m}^3/\text{s} \end{aligned}$$

Assuming that $f_d = f_s = 0.02$, then equation (4.15) gives

$$\begin{aligned} \frac{d_d^5}{d_s^5} &= \frac{10.74}{4.25} \times \frac{(4.9 \times 10^{-5})^2}{(12.4 \times 10^{-5})^2} \times \frac{341.6}{710.6} \\ &= 0.19 \end{aligned}$$

$$\therefore \frac{d_d}{d_s} = (0.19)^{0.2} = 0.717$$

$$\begin{aligned} \therefore d_d &= 0.717 \times 4.995 \text{ mm} \\ &= 3.58 \text{ mm} \underline{\underline{ \geq }} \frac{1}{8}'' \end{aligned}$$

Recommendation:- The available pipe sizes in the market show that a nominal size of $\frac{1}{8}''$ is available. Also, the maximum allowable working pressures for steel pipes of nominal size $\frac{1}{8}''$ range from⁽³³⁾ 1500 psig to 400 psig for both internal and external refrigeration duty. Hence the pipe size chosen will withstand the pressures that will be involved during the operation. It is envisaged that the condenser will be the bare coil in-air type and air-cooled. This size of pipe can be easily

worked to get the shape required in view of the considerably large number of bends that are expected for the air-cooled condenser.

Liquid Lines:- Liquid lines generally do not contribute so much pressure drop as their fittings. In this case, we envisage little or no fittings as the tendency is to reduce the chances of leakage. A point to note, however, in the construction is the problem of vapour locks. Just as air locks can stop flow in water pipe lines, so can vapour collecting at the highest point in an upwards loop stop the flow in a liquid refrigerant line. Heat input to the line should be avoided by insulation if the line is to pass through hot areas. Vapour formation through pressure drop is to be prevented by avoiding upward loops in the liquid line.

By inspection and reference to piping handbook, (33) the liquid line from condenser to the receiver tank as well as the liquid line from receiver tank to the expansion valve is to be $\frac{1}{8}$ " internal diameter. After the expansion, the line is to be $\frac{1}{4}$ " on to the evaporator.

4.5. PUMP HORSE POWER

If we assume a mechanical efficiency of 75% (33) and refer to Figure 4.3, the pump horse power is given by

$$\text{H.P.} = \frac{m_3 (h_4 - h_3)}{42.4 \eta_m} \quad \text{where the units are in imperial}$$

$$\text{i.e. H.P.} = \frac{0.725 \times 0.44}{42.4 \times 0.75} = \underline{\underline{0.01 \text{ horse power}}}$$

In metric units, we find that the power required by the pump is given by

$$\begin{aligned}
 P &= \frac{m_3 (h_4 - h_3)}{\eta} \\
 &= \frac{54.8 \times 10^{-4} \times 1.02}{0.75} \text{ kW} \\
 &= 7.45 \times 10^{-3} \text{ kW} \\
 \text{or} &= \underline{\underline{7.45 \text{ W}}}
 \end{aligned}$$

Mass flow rate through the pump = 0.725 lb/min or 54.8×10^{-4} kg/s.

Specific volume (as calculated in Section 4.3)

$$\begin{aligned}
 &= 0.019 \text{ ft}^3/\text{lb} \\
 &= 11.86 \times 10^{-4} \text{ m}^3/\text{kg} \\
 \therefore \text{Volume flow rate} &= 0.065 \times 10^{-4} \text{ m}^3/\text{s}.
 \end{aligned}$$

4.6. EXPANSION VALVE SELECTION

From Figure 4.3, we deduce that 2 number expansion valves are required; one between state points 10 and 1, and the other between state points 7 and 8.

$$\begin{aligned}
 \text{The rated pressure drop (continuous is } 13.89 - \\
 13.89 - 5.35 &= \underline{\underline{8.54 \text{ bar}}} \\
 \text{or } 201.4 - 77.6 &= \underline{\underline{123.8 \text{ psi}}}.
 \end{aligned}$$

4.7. DESIGN OF THE EVAPORATOR

The evaporator is the bare coil-in-air type. Heat transfer in the cold room is by natural convection of the air in the room. The heat transfer across the

evaporator must be equal to the total cooling load calculated for the cold room (Section 2.5), i.e.

$$Q_E = 0.7338 \text{ kW} \text{ -----(4.16)}$$

If we refer to Figure 4.4, which is a cross-section through the evaporator coil and note that t_R is the temperature of the cold room,

$$t_E = 6^\circ\text{C}, \quad t_R = 8^\circ\text{C}$$

$$d = \frac{1}{4} \text{ in.} = 6.35 \times 10^{-3} \text{ m}$$

$$\therefore r = \frac{d}{2} = \frac{1}{8} \text{ in.} = 3.175 \times 10^{-3} \text{ m}$$

$$D = \frac{3}{8} \text{ in.} = 9.53 \times 10^{-3} \text{ m}$$

$$\therefore R = \frac{D}{2} = \frac{3}{16} \text{ in.} = 4.76 \times 10^{-3} \text{ m}$$

The equation for heat transfer by conduction across the walls of the evaporator is given by

$$Q_E = 2\pi r L h_i (t_i - t_E) = \frac{2\pi L (t_R - t_i)}{\frac{\ln \frac{R}{r}}{K} + \frac{1}{Rh_o}} \text{ -----(4.17)}$$

where L is the length of the coil. To be able to calculate h_i , we note that the Chato equation (4.5,7) for evaporation inside a tube is

$$Nu = 0.555 \left[\frac{E \rho_v (\rho_L - \rho_v) d^3 h_{fg}}{\mu_v K_v \Delta T} \right]^{\frac{1}{4}} \text{ -----(4.18)}$$

where $\Delta T = t_i - t_E$. Hence,

$$Nu = \frac{h_i d}{K_v}$$

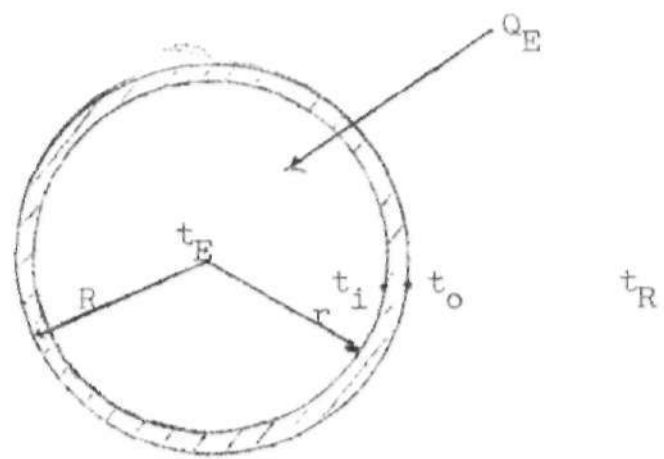


FIG. 4.4.: CROSS-SECTION THROUGH THE EVAPORATOR COIL

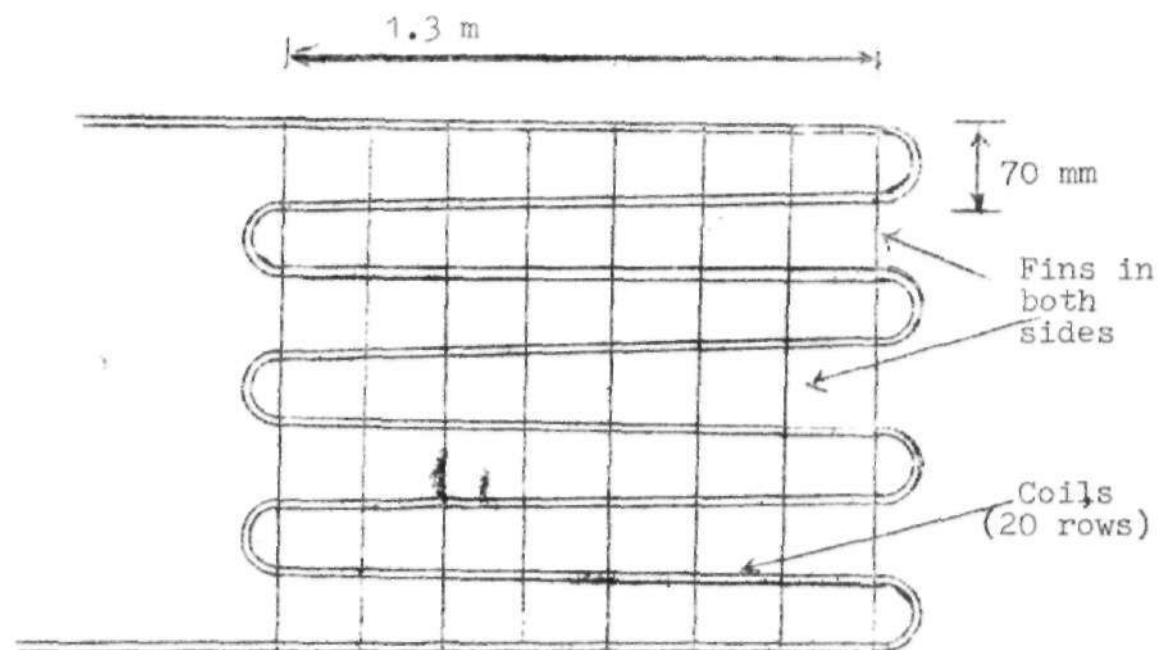


FIG.4.5.: THE EVAPORATOR WITH FINS

70.A

$$= 0.555 \left[\frac{9.81 \times 0.782 (629 - 0.782) (6.35 \times 10^{-3})^{1/4} (1234)}{(9.56 \times 10^{-6}) (22.7 \times 10^{-6}) (t_i - t_E)} \right]^{1/4}$$

$$\therefore Nu = \frac{6.35 \times 10^{-3} h_i}{22.7 \times 10^{-6}} = 160.6 \left[\frac{1}{t_i - t_E} \right]^{1/4} \text{ kW/m}^2 \text{ K}$$

$$\text{i.e. } h_i = 0.574 \left[\frac{1}{t_i - t_E} \right]^{1/4} \text{ kW/m}^2 \text{ K} \text{ ----- (4.19)}$$

To calculate h_o , within the cold room, we note that for free convection

$$Gr = \frac{g D^3 \rho_{\text{air}}^2 \Delta T}{\mu_{\text{air}}^2 T} \text{ ----- (4.20)}$$

We expect the resistance of the air film to be much higher than the sum of those due to the evaporating refrigerant and the tube wall. Hence, much of the temperature drop between the refrigerant and the room will take place in the air film. Assuming $\Delta T = 2^\circ\text{C}$, then equation 4.20 gives

$$Gr = \frac{9.81 \times (9.53 \times 10^{-3})^3 \times (1.258)^2 \times 2}{(1.754 \times 10^{-5})^2 \times 281}$$

$$= 310.99$$

From thermodynamic tables⁽³⁾ we obtain the value of the Prandtl number Pr as $= 0.712$

$$\text{Therefore, } Gr \times Pr = 310.99 \times 0.712$$

$$= 221.4$$

The Elenbas formula⁽⁵⁷⁾ for heat transfer on horizontal cylinders under free convection for this range of

$$Gr \times Pr \text{ is } Gr \times Pr = 235(Nu)^3 e^{-6/Nu} \text{ ----- (4.21)}$$

$$\therefore (Nu)^3 e^{-6/Nu} = \frac{221.4}{235}$$

$$= 0.94$$

or $\frac{(Nu)^3}{e^{6/Nu}} = 0.94$

Trial and error gives $Nu \approx 2.3$

Therefore

$$\frac{h_o D}{K_{air}} = 2.3$$

and

$$h_o = \frac{K_{air} \times 2.3}{D}$$

$$= \frac{2.475 \times 10^{-5} \times 2.3}{9.53 \times 10^{-3}}$$

$$= 0.0059 \text{ kW/m}^2 \text{ K}$$

From equation (4.17), we deduce

$$rh_1 (t_i - t_E) = \frac{t_R - t_1}{\frac{\ln \frac{R}{r}}{K} + \frac{1}{Rh_o}}$$

Substituting values and taking h_1 from equation (4.19),

we have

$$3.175 \times 10^{-3} \left[0.574 \left(\frac{1}{t_i - 6} \right)^{\frac{1}{4}} \right] (t_i - 6)$$

$$= \frac{8 - t_i}{\frac{\ln \frac{4.76}{3.175}}{0.0435} + \frac{1}{(4.76 \times 10^{-3})(0.0059)}}$$

Trial and error gives $t_i \approx 6.2^\circ \text{C}$.

To calculate the value of the length of pipe required we refer to equation (4.17) and deduce that

$$L = \frac{Q}{2\pi r h_i (t_i - t_E)} \quad \text{----- (4.22)}$$

If we substitute the value of h_i from equation (4.19) we have

$$\begin{aligned} L &= \frac{Q}{2\pi r (t_i - t_E) \times 0.574 \left(\frac{1}{t_i - t_E} \right)^{\frac{1}{4}}} \\ &= \frac{Q}{2\pi r \times 0.574 (t_i - t_E)^{\frac{5}{4}}} \\ &= \frac{0.7338}{2 \times \pi \times 3.175 \times 10^{-3} \times 0.574 \times (0.2)^{\frac{5}{4}}} \\ &= 214.2 \text{ m} \end{aligned}$$

Use of Fins:- The use of fins is necessary to increase the heat transfer area because due to the very long length of bare tube calculated for free convection heat transfer, a shorter length of tube has to be used in view of the dimensions of the cold room. For easier construction, the configuration shown in Figure 4.5 is adopted. The fins are to be tacked on both sides of the coils.

Taking the dimensions of the cold room into consideration, a bank of bare coil of 20 rows is recommended such that each row is 1.3 m. This gives a new length of about $20 \times 1.3 = 26$ m. The fins recommended are steel wires or rods of diameter 3 mm. The spacing between rows is 70 mm and this gives the length of each fin as 1.33 m.

For the required heat transfer to take place with the shorter length of coil and fins, the old area must be equal to the new heat transfer area, (3) i.e.,

$$\tilde{h} D_E L_E = \tilde{h}' D_f L_f n + \tilde{h} D_E L_{\text{new}} \text{-----(4.23)}$$

where n is the number of fins required.

L_E and D_E apply to the evaporator without fins.

L_{new} is the new shorter length to which fins are to be attached and the subscript f applies to fins.

Therefore,

$$\begin{aligned} n &= \frac{\tilde{h} D_E (L_E - L_{\text{new}})}{\tilde{h}' D_f L_f} \\ &= \frac{2 \times 14.76 \times 10^{-3} (214.2 - 26)}{3 \times 10^{-3} \times 1.33} \\ &= 449 \text{ fins.} \end{aligned}$$

This means that we shall have 225 fins on both sides of the bank. This gives spacing between fins as

$$\begin{aligned} &\frac{1300}{225} \\ &\div 6 \text{ mm.} \end{aligned}$$

Hence a spacing of 6 mm between the fins is recommended.

4.8. DESIGN OF THE CONDENSER

The condenser is to be the bare coil-in-air type. Heat transfer for cooling the condenser is by free convection of the ambient air. Across the walls of the condenser, heat transfer is given by

$$\begin{aligned} Q_c &= m_9 (h_9 - h_{10}) \\ &= 741.8 \text{ W or } 0.7418 \text{ kW (See Table 4.3).} \end{aligned}$$

From the initial design data (Section 4.1), we note that

$$t_c = 36^\circ\text{C}$$

$$t_a \text{ (Ambient temperature)} = 32^\circ\text{C}$$

If we refer to Figure 4.6 which is a cross-section through the condenser coil,

$$r = \frac{1}{16} \text{ in} = 1.6 \times 10^{-3} \text{ ,}$$

$$R = \frac{D}{2} = 3.175 \times 10^{-3} \text{ m}$$

We shall assume that all heat transferred is between the condensing ammonia and the ambient air which we are using as the cooling medium. Also, the heat transfer is across the tube walls.

The heat transfer equation for this situation is given by

$$Q_c = 2\pi r L h_i (t_c - t_i) = \frac{2\pi L (t_i - t_o)}{\ln \frac{R}{r} + \frac{1}{\frac{K}{R} + \frac{1}{R h_o}}} \text{ -----(4.24)}$$

where L is the length of the condenser tube.

For condensation inside a cylindrical tube, we note that after a comprehensive work and experimental investigations, Akers, Deans and Crosser⁽⁷⁾ reported that for both horizontal and vertical tubes, the following correlations hold:

$$\text{For } Re_G < 5 \times 10^4, Nu_d = 5.03 Re_G^{\frac{1}{3}} Pr^{\frac{1}{3}} \text{ -----(4.25)}$$

$$\text{For } Re_G > 5 \times 10^4, Nu_d = 0.0265 Re_G^{0.8} Pr^{\frac{1}{3}} \text{ ---(4.26)}$$

where $Re_G = \frac{G d}{\mu}$

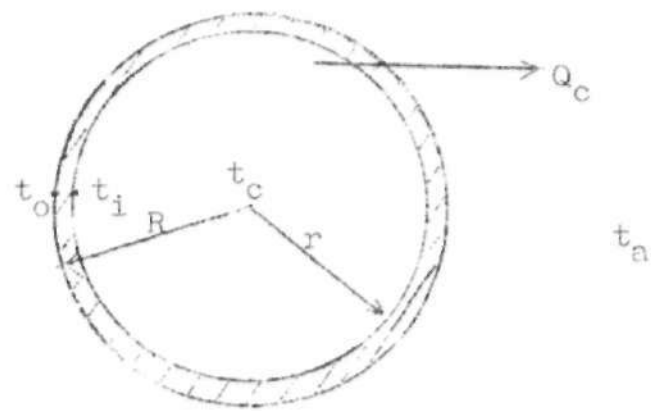


FIG. 4.6.: CROSS-SECTION THROUGH THE CONDENSER COIL

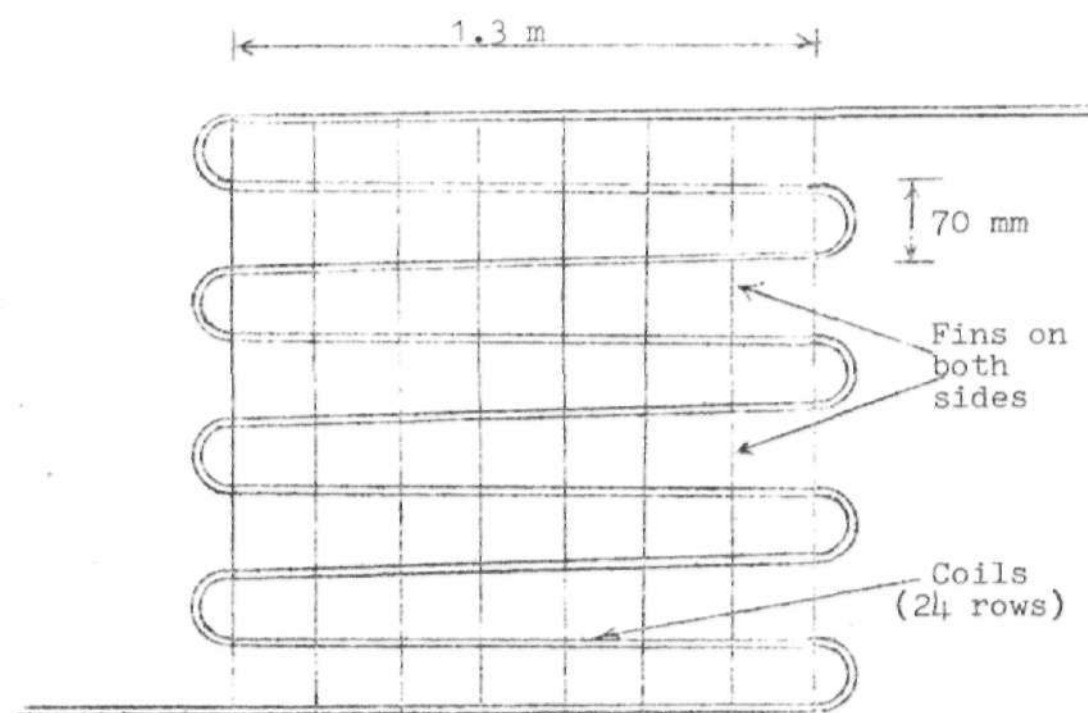


FIG. 4.7.: THE CONDENSER WITH FINS

For these correlations, G , the equivalent mass velocity is defined as:

$$G = G_L + G_V \left(\frac{\rho_L}{\rho_V} \right)^{\frac{1}{2}} \text{-----(4.27)}$$

G_L and G_V are the mass velocities (mass flow rate per unit cross-section area) based on the full pipe or tube area of the liquid and vapour phases respectively.

Hence from equation (4.27), we have

$$G = \frac{m_L}{A} + \frac{m_V}{A} \left(\frac{\rho_L}{\rho_V} \right)^{\frac{1}{2}}$$

From Table 1, we obtain $m_V = m_9 = 5.3 \times 10^{-4} \text{ kg/s}$
and $m_L = m_{10} = 5.3 \times 10^{-4} \text{ kg/s}$.

Therefore,

$$\begin{aligned} G &= \frac{5.3 \times 10^{-4}}{\frac{\pi}{4} (3.175 \times 10^{-3})^2} \left[1 + \left(\frac{\rho_L}{\rho_V} \right)^{\frac{1}{2}} \right] \\ &= 66.915 \left[1 + (600 \times 0.0931)^{\frac{1}{2}} \right] \\ &= 66.915 \times 8.474 \\ &= 567 \text{ kg/s m}^2 \end{aligned}$$

Consequently,

$$\begin{aligned} Re_G &= \frac{G d}{\mu} \\ &= \frac{567 \times 3.175 \times 10^{-3}}{119 \times 10^{-6}} \\ &= 1.513 \times 10^4 \end{aligned}$$

which is less than 5×10^4 . We therefore apply equation (4.25) and deduce that

$$Nu = \frac{h_i d}{K} = 5.03 Re_G^{\frac{1}{3}} Pr^{\frac{1}{3}}$$

$$\therefore \frac{h_i d}{K} = 5.03(1.513 \times 10^4)^{\frac{1}{3}} (Pr)^{\frac{1}{3}} \text{----- (4.28)}$$

Now

$$\begin{aligned} Pr &= \frac{\mu_{c,p}}{K} \\ &= \frac{119 \times 10^{-6} \times 4.86}{511 \times 10^{-6}} \\ &= 1.132 \end{aligned}$$

Hence equation (4.28) gives

$$\begin{aligned} h_i &= \frac{K}{d} \times 5.03(1.513 \times 10^4)^{\frac{1}{3}} (1.132)^{\frac{1}{3}} \\ &= \frac{511 \times 10^{-6}}{3.175 \times 10^{-3}} \times 5.03(1.513 \times 10^4)^{\frac{1}{3}} (1.132)^{\frac{1}{3}} \\ &= 20.87 \text{ kW/m}^2 \text{ K.} \end{aligned}$$

To calculate h_o , we consider free convection of the air on the outside of the condenser and note that:

$$Gr = \frac{g D^3 \rho_{air}^2 \Delta T}{\mu_{air}^2 T_{\infty}}$$

If the ambient air at 32°C is used to cool the condenser, we shall assume that its temperature at exit after the cooling should be about 34°C . With this, we expect the outside wall temperature to be about 35°C . Hence we have ΔT estimated to be 3°C , i.e. $(35 - 32)^\circ\text{C}$.

Therefore,

$$\begin{aligned} Gr &= \frac{9.81 \times (6.35 \times 10^{-3})^2 \times (1.159)^2 \times 3}{(1.869 \times 10^{-5})^2 \times 3.05} \\ &= 81.975 \end{aligned}$$

From thermodynamic tables of properties, we obtain

$$Pr = 0.706.$$

$$\therefore Gr \times Pr = 57.87$$

The Elenbas formula⁽⁵⁷⁾ for heat transfer on horizontal cylinders under free convection for this range of $Gr \times Pr$ is

$$Gr \times Pr = 235(Nu)^3 e^{-6/Nu}$$

$$\therefore \frac{(Nu)^3}{e^{-6/Nu}} = \frac{57.87}{235}$$

$$= 0.25$$

Trial and error gives $Nu \doteq 1.85$

Therefore,

$$\frac{h_o D}{K_{air}} = 1.85$$

$$\text{or } h_o = \frac{K_{air}}{D} \times 1.85$$

$$= \frac{2.662 \times 10^{-5} \times 1.85}{6.35 \times 10^{-3}}$$

$$= 0.0078 \text{ kW/m}^2 \text{ K.}$$

From equation (4.24), we have

$$rh_i(t_o - t_i) = \frac{t_i - t_o}{\frac{\ln \frac{R}{r}}{K} + \frac{1}{Rh_o}}$$

Substituting values, we have

$$1.6 \times 10^{-3} \times 20.87(36 - t_i)$$

$$= \frac{1}{\frac{\ln \frac{3.175}{1.6}}{0.0435} + \frac{1}{(3.175 \times 10^{-3})(0.0078)}}$$

$$\text{which gives } t_i = 35.965^\circ\text{C}$$

Therefore from equation (4.24), we obtain the length of tube or pipe required as

$$\begin{aligned}
 L &= \frac{Q}{2 \tilde{h} r h_1 (t_c - t_1)} \\
 &= \frac{0.7338}{2 \tilde{h} \times 1.6 \times 10^{-3} \times 20.87 \times (36 - 35.965)} \\
 &= \underline{\underline{99.9 \text{ m}}} \quad \text{or} \quad \doteq 100 \text{ m.}
 \end{aligned}$$

Use of Fins:- As in the case of the evaporator, this is necessary to increase the heat transfer area, because due to the very long length of bare tube calculated for free convection heat transfer, a shorter length of tube has to be used in view of available space. For easier construction, the configuration shown in Figure 4.7 is adopted. The fins are to be tacked on both sides of the coils.

A bank of bare coil of 24 rows is recommended such that each row is 1.3 m long. This gives a new length of about $24 \times 1.3 = 31.2 \text{ m}$. The fins recommended are steel rods or wire of diameter 3 mm. The spacing between rows is 70 mm and this gives the length of each fin as 1.61 m.

For the required heat transfer to take place with the shorter length of tube recommended and fins, the old area must be equal to the new heat transfer area, i.e.

$$\tilde{h} D_c L_c = \tilde{h} D_f L_f n + \tilde{h} D_c L_{\text{new}} \text{ ----- (4.29)}$$

where n is the number of fins required,

L_c and D_c apply to the condenser without fins

L_{new} is the new shorter length to which fins are to be attached and the subscript f applies to fins.

Therefore,

$$\begin{aligned} n &= \frac{\eta_{D_c}(L_c - L_{new})}{\eta_{D_f} L_f} \\ &= \frac{6.35 \times 10^{-3} (100 - 31.2)}{3 \times 10^{-3} \times 1.61} \\ &= 91 \text{ fins.} \end{aligned}$$

This implies that we shall have 46 fins on both sides of the bank. Addition of more fins will not, however, have an adverse effect on the condenser and so a spacing of 20 mm for the fins is recommended. At 20 mm spacing of the fins, we should have on each side

$$\frac{1300}{20} = 65 \text{ fins.}$$

This makes a total of 130 fins.

4.9. DESIGN OF THE GENERATOR

The generator and the solar collector form a unit. Figure 4.8 is extracted from Figure 4.3.

Referring to Figure 4.8, we make the following description of the state points:

State Point 9 : Rectified ammonia vapour.

" " 11 : Ammonia vapour before rectification.

" " 12 : Condensate (water).

" " 5 : Strong ammonia-water solution.

" " 6 : Weak ammonia-water solution.

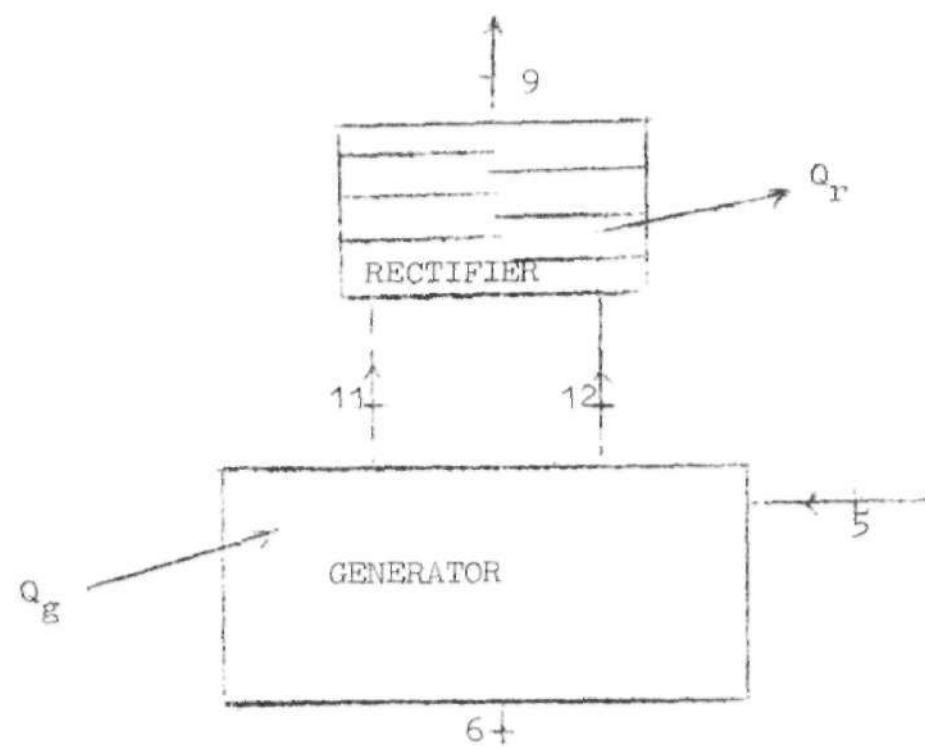


FIG. 4.8.: FLUID FLOW INTO AND OUT OF THE GENERATOR AND RECTIFIER

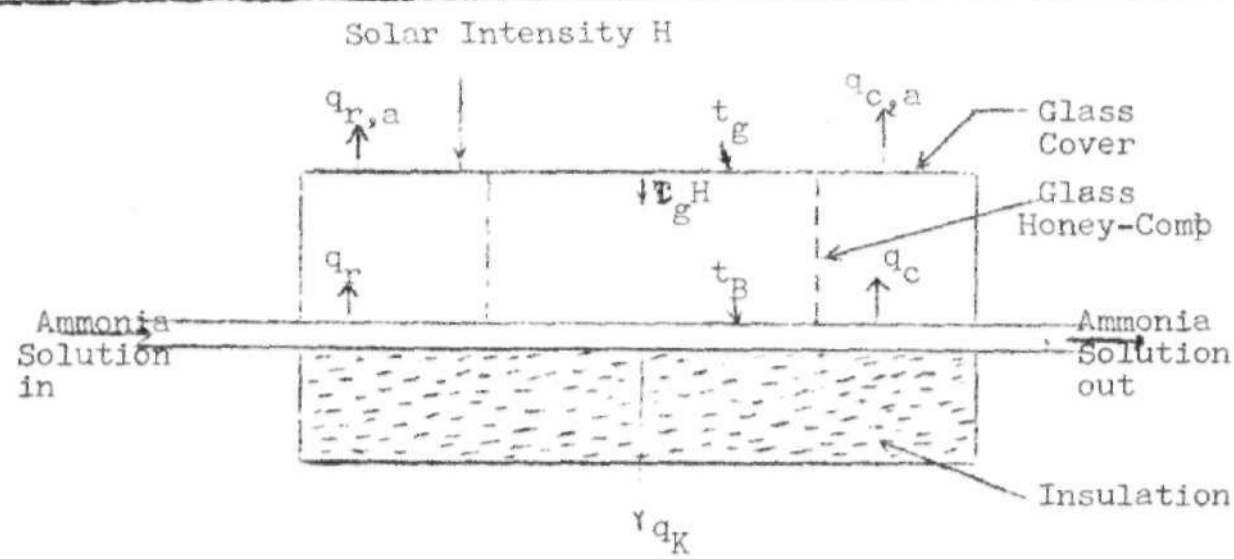


FIG. 4.9.: SECTION THROUGH THE COLLECTOR/GENERATOR

The condensate is liquid in equilibrium with vapour going out of the rectifier and so it may be considered as saturated liquid at 50°C (122°F) and 13.89 bar (201.4 psi). Using the Enthalpy concentration chart for ammonia-water solution we obtain

$$x_{12} = 0.5 \text{ and } h_{12} = 80 \text{ Btu/lb}$$

$$\text{or } h_{12} = 186.1 \text{ kJ/kg.}$$

State (11) can be considered as saturated vapour at 13.89 bar and 70°C (158°F). Hence, using the Enthalpy concentration chart also we obtain:

$$x_{11} = 0.98 \text{ and}$$

$$h_{11} = 670 \text{ Btu/lb or } 1558.5 \text{ kJ/kg.}$$

Considering mass balance in the rectifier, we have

$$m_9 + m_{12} = m_{11} \text{ -----(4.30)}$$

$$\text{and } m_9 x_9 + m_{12} x_{12} = m_{11} x_{11} \text{ -----(4.31)}$$

Eliminating m_{11} from (4.30) and (4.31), we obtain

$$\begin{aligned} m_{12} &= \frac{m_9(x_{11} - x_9)}{x_{12} - x_{11}} \\ &= \frac{5.3 \times 10^{-4}(0.98 - 0.99)}{(0.5 - 0.98)} \\ &= 0.11 \times 10^{-4} \text{ kg/s.} \end{aligned}$$

Substituting in equation (4.30), we have

$$\begin{aligned} m_{11} &= m_9 + m_{12} = 0.11 \times 10^{-4} + 5.3 \times 10^{-4} \\ &= 5.41 \times 10^{-4} \text{ kg/s.} \end{aligned}$$

4.9.1. Heat Required at the Generator:- Referring to Figure 4.8, the heat required at the generator

$$Q_g = m_{11}h_{11} + m_6h_6 - m_{12}h_{12} - m_5h_5$$

$$\begin{aligned}
&= (5.41 \times 10^{-4} \times 1558.5) + (49.4 \times 10^{-4} \times 169.8) \\
&- (0.11 \times 10^{-4} \times 186.1) - (54.8 \times 10^{-4} \times 153.5) \text{ kW} \\
&= 0.8387 \text{ kW}
\end{aligned}$$

i.e. Heat required from generator = 838.7 W.

Both the generator and the solar collector as stated earlier form a unit. The pipes carrying the ammonia-water solution are attached to the solar collector plate and the two ends of the pipes are connected by pipe headers. The flat plate has a single glass cover. The multi-cell flat plate solar collector is recommended because it has the advantage of using both beam and diffuse solar radiation not requiring orientation towards the sun. This involves the placement of a honeycomb structure between the absorber plate and cover glass to reduce natural convection and radiative heat losses from the hot absorber.

4.9.2. Temperature of the Glass Cover (T_g):- This is required in order to compute the radiative and convective heat loss coefficients h_r and h_c respectively. We shall make use of the following data among others:
Ambient temperature $t_a = 32^\circ\text{C}$ $\therefore T_a = 305 \text{ K}$
Solar absorber plate temperature $t_B = 70^\circ\text{C}$ $\therefore T_B = 343 \text{ K}$
Angle of tilt of the collector, $S = 11^\circ 11' = 11.2^\circ$

Referring to Figure 4.9, the heat loss from the absorber plate to the glass cover will be equal to the heat lost from the glass to the surroundings under steady state if we neglect edge losses. If we assume

that the absorber plate and glass cover are grey opaque bodies for the wave-lengths of thermal radiation under consideration and that the sky is a black body at ambient temperature, we shall have the heat transfer equation

$$q_{L,top} = h_c(T_B - T_g) + \frac{\sigma(T_B^4 - T_g^4)}{\frac{1}{\epsilon_B} + \frac{1}{\epsilon_g} - 1}$$

$$= h_{c,a}(T_g - T_a) + \epsilon_g \sigma(T_g^4 - T_a^4) \text{ ----- (4.32)}$$

where ϵ is emissivity, and σ is Stefan-Boltzmann Constant.

For free convection of air between parallel, tilted plates, Dropkins and Somerscales⁽¹⁸⁾ recommended

$$Nu = \frac{h_c L}{K_a} = \left[0.06 + 0.17 \left(\frac{S}{90} \right) \right] Gr^{\frac{1}{3}} \text{ ----- (4.33)}$$

$$\text{i.e. } \frac{h_c L}{K_a} = \left[0.06 + 0.17 \left(\frac{S}{90} \right) \right] \left[\frac{\epsilon_g \theta^2 \rho_a^2 L^3}{\mu_a^2 T} \right]^{\frac{1}{3}}$$

where L is length,

$$\theta = T_B - T_g$$

$$\text{and } T = \frac{T_B + T_g}{2}$$

$$\therefore h_c = K_a \left[0.06 + 0.17 \left(\frac{S}{90} \right) \right] \left[\frac{2g(T_B - T_g) \rho_a^2}{\mu_a^2 (T_B + T_g)} \right]^{\frac{1}{3}} \text{ --- (4.34)}$$

We shall assume, for convenience, air properties at a temperature between that of the absorber plate and the ambient air, $\rho_a = 1.086 \text{ kg/m}^3$, $K_a = 2.816 \times 10^{-5} \text{ kW/m K}$ and $\mu_a = 1.962 \times 10^{-5} \text{ kg/m s}$.

Therefore, substituting in equation (4.34), we obtain

$$h_c = (2.816 \times 10^{-5} \left[0.06 - 0.17 \left(\frac{11.2}{90} \right) \right]$$

$$\left[\frac{2 \times 9.81 \times (1.086)^2}{(1.962 \times 10^{-5})^2} \right]^{\frac{1}{3}} \left[\frac{T_B - T_g}{T_B + T_g} \right]^{\frac{1}{3}}$$

$$\text{or } h_c = 4.20 \times 10^{-3} \left[\frac{T_B - T_g}{T_B + T_g} \right]^{\frac{1}{3}} \text{ kW/m}^2 \text{ K} \quad \text{---(4.35)}$$

For radiative heat transfer between absorber plate and glass we have

$$h_r = \frac{\epsilon_B^2 + \epsilon_g^2 (T_B + T_g)}{\frac{1}{\epsilon_B} + \frac{1}{\epsilon_g} - 1} \quad \text{---(4.36)}$$

From (6, 12, 13), we obtain

$$\epsilon_B = 0.9$$

$$\epsilon_g = 0.9$$

and $\sigma = 56.7 \times 10^{-12} \text{ kW/m}^2 \text{ K}^4$ then

$$h_r = \frac{56.7 \times 10^{-12} (T_B^2 + T_g^2) (T_B + T_g)}{\frac{1}{0.9} + \frac{1}{0.9} - 1}$$

$$\text{or } h_r = 46.4 \times 10^{-12} (T_B^2 + T_g^2) (T_B + T_g) \quad \text{---(4.37)}$$

For convective heat loss from glass to ambient air, McAdams equation⁽¹⁸⁾ for heat loss from flat plates exposed to outside winds is given by

$$h_{c,a} = (5.7 + 3.8 V_w) \times 10^{-3} \text{ kW/m}^2 \text{ K}$$

If we take the velocity of the wind to be 5 m/s, then

$$h_{c,a} = 24.7 \times 10^{-3} \text{ kW/m}^2 \text{ K} \quad \text{---(4.38)}$$

For radiative heat loss from cover plate

$$h_{r,a} = \epsilon_g (T_g^2 + T_a^2)(T_g + T_a)$$

$$= 0.9 \times 56.7 \times 10^{-12} (T_g^2 + T_a^2)(T_g + T_a)$$

$$\text{or } h_{r,a} = 51.03 \times 10^{-12} (T_g^2 + T_a^2)(T_g + T_a) \text{---(4.39)}$$

Hence by substitution, equation (4.32) becomes

$$4.2 \times 10^{-3} \left[\frac{T_B - T_g}{T_B + T_g} \right]^{\frac{1}{3}} (T_B - T_g) + 4.64 \times 10^{-12} (T_B^4 - T_g^4)$$

$$= 24.7 \times 10^{-3} (T_g - T_o) + 51.03 \times 10^{-12} (T_g^4 - T_a^4) \text{----(4.40)}$$

We remember that

$$t_B = 70^\circ\text{C} \quad \therefore T_B = 343 \text{ K}$$

$$\text{and } t_a = 32^\circ\text{C} \quad \therefore T_a = 305 \text{ K}$$

Solving equation (4.40) by method of trial and error,

$$\text{we obtain } \underline{T_g = 310 \text{ K or } t_g = 37^\circ\text{C}}$$

Therefore, from equation (4.35),

$$h_c = 4.20 \times 10^{-3} \left[\frac{343 - 310}{343 + 310} \right]^{\frac{1}{3}}$$

$$\text{or } h_c = 1.553 \times 10^{-3} \text{ kW/m}^2 \text{ K.}$$

Also from equation (4.37),

$$h_r = 46.4 \times 10^{-12} (T_B^2 + T_g^2)(T_B + T_g)$$

$$\text{or } h_r = 46.4 \times 10^{-12} (343^2 + 310^2)(343 + 310)$$

$$= 6.48 \times 10^{-3} \text{ kW/m}^2 \text{ K}$$

From equation (4.39), radiative heat loss coefficient from glass cover to sky, $h_{r,a}$, is given by

$$\begin{aligned}
h_{r,a} &= 51.03 \times 10^{-12} (T_g^2 + T_a^2)(T_g + T_a) \\
&= 51.03 \times 10^{-12} (310^2 + 305^2)(310 + 305) \\
&= 5.85 \times 10^{-3} \text{ kW/m}^2 \text{ K}
\end{aligned}$$

4.9.3. Coefficient of Conductive Loss Through Back Insulation, h_I :- We shall assume that the resistance to heat flow by conduction is due only to the insulation. Now

$$h_I = \frac{k}{x}$$

where x is the insulation thickness.

The insulation thickness is 5 cm or 50 mm and the thermal conductivity $k = 0.403 \times 10^{-4} \text{ kW/m K}$

Hence,

$$\begin{aligned}
h_I &= \frac{0.403 \times 10^{-4}}{50 \times 10^{-3}} \\
&= 8.06 \times 10^{-4} \text{ kW/m}^2 \text{ K.}
\end{aligned}$$

4.9.4. Overall Heat Loss Coefficient, U_L

$$\begin{aligned}
U_L &= \left[\frac{1}{h_c + h_r} + \frac{1}{h_{c,a} + h_{r,a}} \right]^{-1} \\
&+ h_I \left[\frac{T_B - T_{a,s}}{T_B - T_a} \right] \text{-----(4.41)}
\end{aligned}$$

$T_{a,s}$ is the ambient shade temperature and it is equal to 26°C. Hence, $T_{a,s} = 299 \text{ K}$

$$U_L = \left[\frac{1}{(1.553 + 6.48) \times 10^{-2}} + \frac{1}{(24.7 + 5.35) \times 10^{-3}} \right]$$

$$= 8.05 \times 10^{-4} \times \left[\frac{343}{343} - \frac{299}{305} \right]$$

$$= 6.416 \times 10^{-3} \text{ W/m}^2 \text{ K}$$

4.9.5. Useful Heat Collected From Solar Radiation. q_u :- For the tube-sheet arrangement of the collector, we can consider the region between the centre-line of two adjacent tubes as a classical fin problem. With this idea, it can be shown⁽⁴⁸⁾ that the useful gain of the collector per tube per unit length in the direction of flow is

$$q_u = \left[(b - D_o)F + D_o \right] \left[\alpha_{B,E} \tau_{E,S,H,S,T} - U_L (T_E - T_E) \right] W/m \quad (4.42)$$

D_o is the outside diameter of the tube = 1/2 in or 0.016 m.

Before we can calculate the useful heat q_u collected from solar radiation, we shall calculate the fin efficiency F and the optimum spacing b of the tubes.

4.9.6. Fin Efficiency (F):- The fin efficiency F is defined⁽¹⁸⁾ as

$$F = \frac{\tanh \left[\frac{m(b - D_o)}{2} \right]}{\frac{m(b - D_o)}{2}} \quad (4.43)$$

where $m^2 = \frac{U_L}{k_B x_B}$

x_B the thickness of the absorber plate = 0.5 mm and

$$k_B = 0.0435 \text{ kW/m K}$$

therefore,

$$m = \sqrt{\frac{6.416 \times 10^{-3}}{0.0435 \times 5 \times 10^{-4}}}$$

$$= 17.23 \text{ /m}$$

Therefore, from equation (4.43),

$$F = \frac{\tanh \left[\frac{17.23(b - 0.016)}{2} \right]}{\left[\frac{17.23(b - 0.016)}{2} \right]} \text{-----(4.44)}$$

To be able to determine F by evaluating equation (4.44), we shall determine the optimum spacing b.

4.9.7. Optimum Fin Spacing (b):- If we refer to equation (4.43) above, we note that for maximum efficiency of the fin, $\frac{dF}{dD_o} = 0$.

Now

$$F = \frac{\tanh \left[\frac{m(b - D_o)}{2} \right]}{\frac{m(b - D_o)}{2}}$$

$$\frac{dF}{dD_o} = \frac{-\frac{m^2(b - D_o)}{4} \text{sech}^2 \frac{m(b - D_o)}{2} + \frac{m}{2} \tanh \frac{m(b - D_o)}{2}}{\left[\frac{m(b - D_o)}{2} \right]^2}$$

The denominator is always positive and finite. Hence

for $\frac{dF}{dD_o} = 0$, we have

$$\frac{m}{2} \tanh \frac{m(b - D_o)}{2} = \frac{m^2(b - D_o)}{4} \text{sech}^2 \frac{m(b - D_o)}{2}$$

which gives

$$\tanh \frac{m(b-D_o)}{2} = \frac{m}{2} \frac{(b-D_o)}{2} \operatorname{sech}^2 \frac{m(b-D_o)}{2} \text{ ----- (4.45)}$$

$$\text{Let } x = \frac{m(b-D_o)}{2}$$

then equation (4.45) reduces to

$$\tanh x = \frac{x}{2} \operatorname{sech}^2 x$$

By plotting the graphs of $y = \tanh x$ and $y = \frac{x}{2} \operatorname{sech}^2 x$ on the same axis we find the point of intersection to be at $x = 0.724$

Hence

$$\frac{m(b-D_o)}{2} = 0.724$$

$$\therefore b = \frac{2 \times 0.724}{17.23} + 0.016 = 0.1$$

Therefore, equation (4.44) gives Fin efficiency

$$F = \frac{\tanh \left[\frac{17.23(0.1 - 0.016)}{2} \right]}{\frac{17.23(0.1 - 0.016)}{2}} = 0.856$$

Consequently, we can proceed towards calculating the useful heat q_u collected from solar radiation.

From data⁽³⁾ of monthly means of total radiation we deduce that in Zaria, the mean daily total radiation on a horizontal surface for the month of April may be taken as $H_{\text{daily}} = 25,128 \text{ kJ/m}^2$ per day.

Using the approximate equation of Cooper⁽¹⁸⁾ the declination, i.e. angular position of the sun at solar

noon with respect to the plane of the equator is

$$\delta = 23.45 \sin \left[360 \left(\frac{284 + n}{366} \right) \right] \text{ ----- (4.46)}$$

where n is the day of the year. If we design for a day in the middle of April, say April 15, we have

$$\begin{aligned} \delta &= 23.45 \sin \left[360 \left(\frac{284 + 106}{366} \right) \right] \\ &= 9.4^\circ \end{aligned}$$

We are using the month of April because data obtained from the MANDILAS Air Conditioning Division, Lagos, show that the design month for Zaria is April. The latitude ϕ is 11.2° . Therefore the length of day for April 15 in hour is given by⁽¹⁸⁾

$$\begin{aligned} \tau_d &= \frac{2}{15} \cos^{-1} (-\tan \phi \tan \delta) \\ &= \frac{2}{15} \cos^{-1} [-\tan(11.2^\circ) \tan(9.4^\circ)] \\ &= 11.75 \text{ hours.} \end{aligned}$$

Considering the hour between 1200 hours and 1300 hours, i.e. mean of $\frac{1}{2}$ hour from solar noon, then from Figure 4.10, copied from ⁽¹⁸⁾, the ratio of hourly total to daily total radiation is 0.143. Therefore, estimated radiation is

$$\begin{aligned} H_h &= 0.143 H_{\text{daily}} \\ &= 0.143 \times 25,128 \text{ kJ/m}^2 \text{ hr} \\ &= 3593.3 \text{ kJ/m}^2 \text{ hr.} \end{aligned}$$

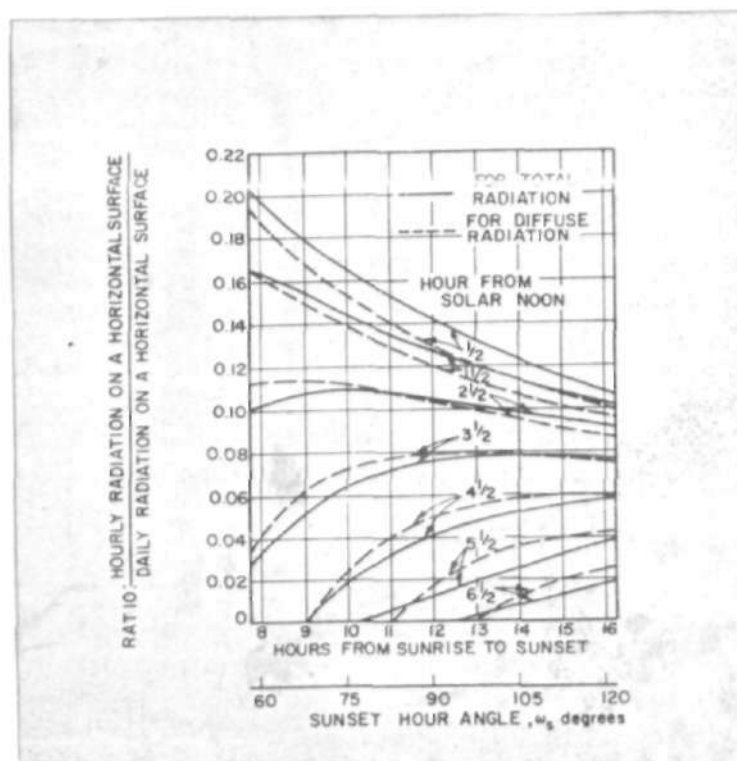


FIG. 4.10.: GRAPH FOR THE ESTIMATION OF HOURLY AND DAILY RADIATION ON A HORIZONTAL SURFACE (REF. 18)

For uniform radiation, we have

Radiation per second, $H_s = 0.998 \text{ kW/m}^2$ or 998 W/m^2 .

The collector surface is to be tilted. The factor for the conversion of radiation falling on a horizontal surface to that falling on a tilted surface⁽¹⁸⁾ is

$$R = \frac{H_{sT}}{H_s} \\ = \frac{H_n \cos \theta_T}{H_n \cos \theta_Z}$$

where T applies to tilt and Z to Zenith.

It can further be shown⁽¹⁸⁾ that

$$R = \frac{\cos \theta_T}{\cos \theta_Z} = \frac{\cos(\theta-s)\cos\delta\cos\omega + \sin(\theta-s)\sin\delta}{\cos\theta\cos\delta\cos\omega + \sin\theta\sin\delta} \quad \text{---(4.47)}$$

where ω is the hour angle.

If we assume that the day under consideration is clear, then R is about the same for direct radiation. Now

$$\theta = s = 11.2^\circ$$

$$\delta = 9.4^\circ$$

$$\omega = -7.5^\circ$$

Then substituting in equation(4.47), we obtain

$$R = \frac{\cos 0^\circ \cos 9.4^\circ \cos(-7.5^\circ) + \sin 0^\circ \sin 9.4^\circ}{\cos 11.2^\circ \cos 9.4^\circ \cos(-7.5^\circ) + \sin 11.2^\circ \sin 9.4^\circ} \\ = 0.987$$

Hence, the radiation that will fall on the tilted

collector is $H_{s,T} = R H_s = 0.987 \times 998 \text{ W/m}^2$

$$= 985 \text{ W/m}^2$$

Taking $\tau_{g,s} = 0.87$

and $\alpha_{B,s} = 0.9$,

then equation (4.42) can now be evaluated to give the useful gain of the solar collector per tube, per unit length in the direction of flow as

$$\begin{aligned} q_u &= \left[(0.1 - 0.016) \times 0.856 + 0.016 \right] \\ &\quad \left[0.9 \times 0.87 \times 985 - 6.416(343 - 310) \right] \times 10^{-3} \text{ kW} \\ &= 49.18 \times 10^{-3} \text{ kW} \\ &= 49.18 \text{ W/m} \end{aligned}$$

Since the length of the fin arrangement, $b = 0.1 \text{ m}$

then

$$\begin{aligned} q_u &= \frac{49.18}{0.1} \\ &= \underline{\underline{491.8 \text{ W/m}^2}} \end{aligned}$$

4.9.8. Area of Collector Plate:- Heat flow

through the generator is given by

$$Q_g = A_B \times q_u$$

Therefore,

$$\begin{aligned} A_B &= \frac{Q_g}{q_u} \\ &= \frac{838.7}{491.8} \\ &= 1.705 \text{ m}^2 \end{aligned}$$

4.9.9. Recommendation For The Generator:- The

standard glass size available in the workshop is 4 ft by 4 ft, i.e. (1.22 m by 1.22 m). Two of this is recommended to be used with an absorber plate width

of about 1.22 m. The thickness of the absorber plate is 0.5 mm. With this recommendation, a collection area of about 2.8 m^2 can be obtained depending on the other components of the collector. As a result of low efficiency expected due to fabrication, etc., this collection area should be adequate. 10 number $\frac{1}{2}$ inch pipes parallel to the length of the plate and connected by 2 inches headers are also recommended. The spacing between the pipes as calculated should be 10 cm. The absorber plate is normally painted dull black.

Glass honeycomb should be mounted between the flat plate and the cover glass of the collector. The glass honeycomb is made of a rectangular commercial glass, square array, with axes perpendicular to the absorber. The clearance between the cover glass and honeycomb should be 5 mm. The collector housing is a box made of plywood, having the usual back and edge insulation. For the insulation, two layers of 25 mm thick micro-aire should be used as it is the insulation available in the workshop. The collector is to be oriented to face South and it should be tilted at about 11.2° .

4.10. DESIGN OF THE RECTIFIER

The rectifier is to be a cylindrical column with rectangular fins welded or soldered on the outside for ambient air cooling. There should be a perforated

plate arrangement to improve turbulence and hence heat transfer in the rectifier as well as increase the contact area between the condensate and the outgoing vapour.

Figure 4.11 shows a cross-section through the rectifier column.

The heat that is removed from the rectifier Q_r is given by $Q_r = \text{Heat removed at the Generator/Rectifier (Table 4.3)} - \text{Heat required for generation (already calculated)}$

$$\begin{aligned} \text{i.e. } Q_r &= 899.07 - 838.7 \\ &= 60.37 \text{ W.} \end{aligned}$$

Referring to Figure 4.11,

$$Q_r = U.A.(t_b - t_a) \text{ -----(4.48)}$$

where A is the outside surface area and U is the overall heat transfer coefficient based on the outside surface area. Now

$$d = 6 \text{ ins} = 0.15 \text{ m}$$

$$\therefore r = \frac{d}{2} = 3 \text{ ins} = 0.075 \text{ m.}$$

The thickness of the 6 ins diameter tube is 5 mm.

Hence $R = 0.008 \text{ m}$, and

$$D = 0.16 \text{ m.}$$

For simplicity, we shall assume that the vapour in the rectifier is pure ammonia. Vapour at entry is at a temperature of 70°C while at exit, it is at a temperature of 50°C . The bulk temperature, t_b , is given by

$$t_b = \frac{70 + 50}{2} = 60^\circ\text{C.}$$

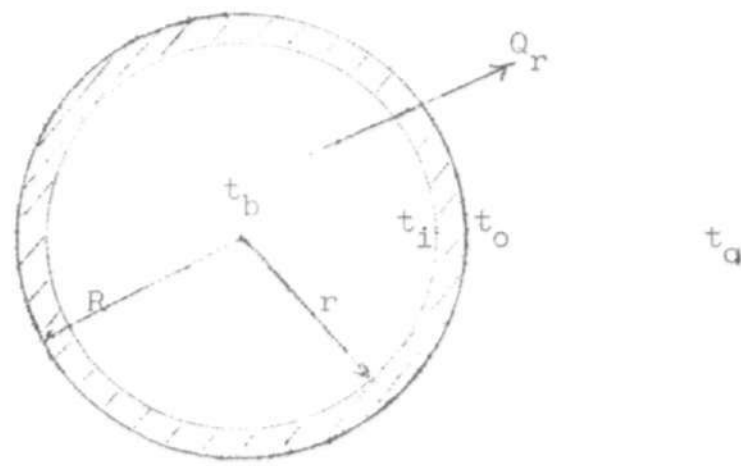


FIG. 4.11.: CROSS-SECTION THROUGH THE RECTIFIER COLUMN

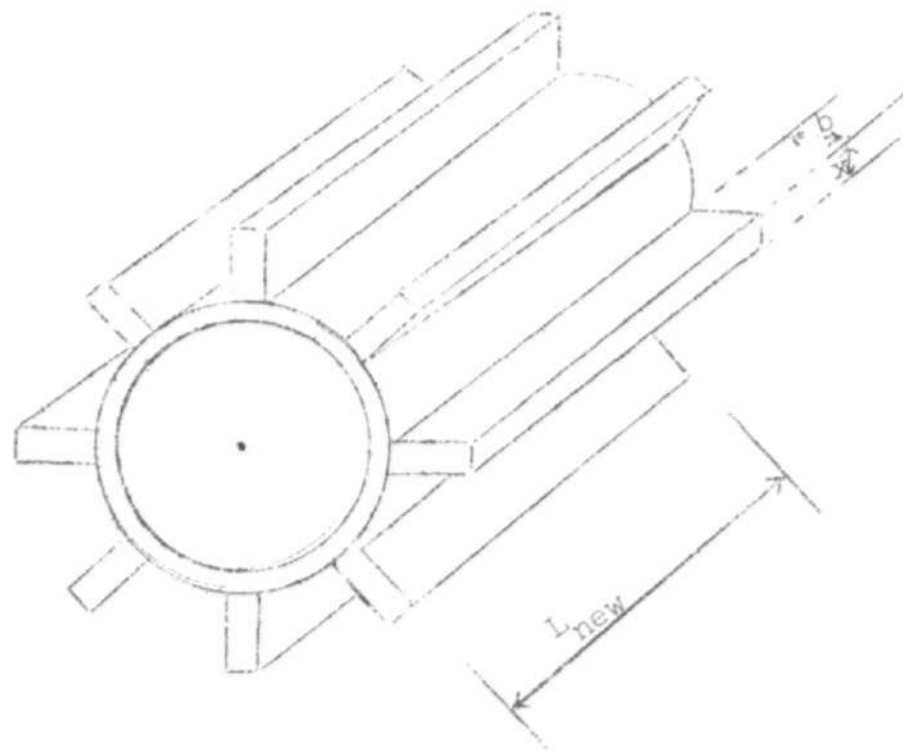


FIG. 4.12.: THE RECTIFIER WITH FINS

Using this bulk temperature, we obtain the following properties of ammonia from thermodynamic tables

$$\rho = 22.9 \text{ kg/m}^3$$

$$\mu = 11.5 \times 10^{-6} \text{ kg/m s}$$

$$K = 27.9 \times 10^{-6} \text{ kW/m K}$$

$$\text{and } C_p = 2.086 \text{ kJ/kg K}$$

Therefore,

$$\begin{aligned} \text{Pr} &= \frac{\mu C_p}{K} = \frac{2.086 \times 11.5 \times 10^{-6}}{27.9 \times 10^{-6}} \\ &= 0.86 \end{aligned}$$

Assuming a vapour flow velocity 0.1 m/s, the Reynolds number of the flow is thus

$$\begin{aligned} \text{Re} &= \frac{\rho v d}{\mu} \\ &= \frac{22.9 \times 0.1 \times 0.15}{11.5 \times 10^{-6}} \\ &= 29.87 \times 10^3 \end{aligned}$$

Thus the flow is turbulent. We are interested in turbulent flow because for flow inside tubes, turbulent flow results in a larger heat transfer than laminar flow.

For cooling of fluids inside tubes, McAdams^(7,57) suggests the following empirical relation based on the bulk fluid temperature

$$\text{Nu} = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.3} \text{ ----- (4.49)}$$

Therefore,

$$\frac{h_i d}{K} = 0.023 (29.7 \times 10^3)^{0.8} (0.86)^{0.3}$$

where h_i is the film coefficient at the inside surface.

$$\begin{aligned} \therefore h_i &= 0.023(29.7 \times 10^3)^{0.8}(0.86)^{0.3} \times \frac{K}{d} \\ &= 0.023(29.7 \times 10^3)^{0.8}(0.86)^{0.3} \times \frac{27.9 \times 10^{-6}}{0.15} \\ &= 0.098 \text{ kW/m}^2 \text{ K} \end{aligned}$$

We note that from the Fourier's heat conduction equation for this situation, U is given by

$$\frac{1}{U} = \frac{R}{rh_i} + \frac{R \ln(\frac{R}{r})}{K} + \frac{1}{h_o}$$

or $U = \frac{1}{\frac{R}{rh_i} + \frac{R \ln(\frac{R}{r})}{K} + \frac{1}{h_o}} \text{ -----(4.50)}$

To be able to calculate h_o we note that for cooling of fluids inside vertical tubes by natural convection, McAdams^(7,57) suggests the following empirical relation for turbulent flow

$$h_o = 0.19 \theta^{0.33} \text{ Btu/hr-ft}^2 \text{ } ^\circ\text{F} \text{ -----(4.51)}$$

where θ is the temperature difference between the wall and the surrounding air, i.e.

$$\theta = t_o - t_a \text{ -----(4.52)}$$

If we take a temperature drop of 2°C across the tube wall then $T_o = 58^\circ\text{C}$ (136.4°F). Hence

$$\begin{aligned} \theta = t_o - t_a &= 136.4 - 89.6 \\ &= 46.8^\circ\text{F} \quad (t_a = 32^\circ\text{C} = 89.6^\circ\text{F}). \end{aligned}$$

Therefore equation (4.51) gives

$$\begin{aligned} h_o &= 0.19(46.8)^{0.33} \text{ Btu/hr-ft}^2 \text{ } ^\circ\text{F} \\ &= 0.676 \text{ Btu/hr-ft}^2 \text{ } ^\circ\text{F} \\ &= 0.0039 \text{ kW/m}^2 \text{ K} \end{aligned}$$

Substituting values in equation (4.50), we obtain

$$U = \frac{1}{\frac{0.08}{0.075 \times 0.098} + \frac{0.08 \ln(\frac{0.08}{0.075})}{0.0435} + \frac{1}{0.0039}}$$

The thermal conductivity K of the steel material to be used is 0.0435 kW/m K.

∴ From equation (4.53), we obtain

$$U = 0.00365 \text{ kW/m}^2 \text{ K.}$$

4.10.1. Length of The Rectifier Column:- From equation (4.48), we have

$$\begin{aligned} A &= \frac{Q}{U(t_b - t_a)} \\ &= \frac{60.37 \times 10^{-3}}{0.00365 \times 28} \\ &= 0.59 \text{ m}^2 \end{aligned}$$

If L is the length of the rectifier column, then

$$\begin{aligned} A &= \pi DL \\ \text{or } L &= \frac{A}{\pi D} \\ &= \frac{0.59}{\pi \times 0.016} \\ &= 1.174 \text{ m.} \end{aligned}$$

4.10.2. Use of Fins:- Because of the apparently long length of the rectifier column, the use of fins to supplement the heat transfer area of a shorter length of tube is considered.

For simplicity, we shall assume a fin effectiveness of 1, i.e. that the whole fin surface is at a constant temperature. Also, we shall assume that no contact resistance will exist at the fin base so that the fin base temperature is the same as the temperature on the outer wall of the tube. We shall employ straight fins of the configuration shown in Figure 4.12. The thickness should be uniform.

For the required heat transfer to take place with a shorter length of tube, the old area must equal the new heat transfer area, i.e.

$$\tilde{h} D_r L_r = \tilde{h} D_r L_{\text{new}} + n(2 L_{\text{new}} b + x_f L_{\text{new}}) \quad \text{---(4.54)}$$

where n is the number of fins required

L_r and D_r apply to the rectifier without fins,

L_{new} is the new shorter length to which fins are attached,

subscript f applies to fins

x is thickness and

b is the breadth of the fins. (See Figure 4.12)

Taking the new length to be 0.4 m and the number of fins to be 8 and also since x_f will be very small, we can re-write equation (4.54) to be

$$\tilde{h} D_r L_r = \tilde{h} D_r L_{\text{new}} + 2 n b L_{\text{new}}$$

which gives

$$\begin{aligned} b &= \frac{\tilde{h} D_r (L_r - L_{\text{new}})}{2 n L_{\text{new}}} \\ &= \frac{\tilde{h} \times 0.16 (1.174 - 0.4)}{2 \times 8 \times 0.4} \\ &= 0.06 \quad \text{or} \quad \underline{\underline{6 \text{ cm}}} \end{aligned}$$

Now, for fin introduction to improve the heat transfer from a surface, we must have⁽³¹⁾ that

$$\frac{2 K_f}{h_f A_f} > 1$$

$$\text{or } \frac{2 K_f}{h_f L_f x_f} > 1$$

$$\text{i.e. } x_f < \frac{2 K_f}{h_f L_f}$$

$$\text{Now } K_f = 0.0435 \text{ kW/m K}$$

$$\text{and } h_f = 0.0039 \text{ kW/m}^2 \text{ K}$$

Therefore,

$$x_f < \frac{2 \times 0.0435}{0.0039 \times 0.4}$$

$$< 55.8 \text{ m.}$$

If we consider the available steel metal sheets to be used for fins in the workshop, then we can conclude that they are adequate.

4.11. DESIGN OF THE ABSORBER

The absorber is to be of the vertical column type such that the vapour from the evaporator enters near the bottom of the column while the weak solution from the generator enters at the top. A perforated plate (spray nozzle or other atomizing devices) should exist in the top of the column through which the weak solution is introduced in the form of droplets to make contact with the gas from the bottom in a counter-current motion. The perforated plate (spray nozzle) is supposed to break the

liquid into a large number of small drops; providing the interfacial surface across which diffusional transfer takes place. Raschig glass packings, crushed stone or ceramic slabs formed into grids are introduced in the absorber to provide a larger interfacial area for the vapour and weak solution to improve absorption. The weak solution flows down by gravity over the surface of the packing material to make contact with the ammonia vapour coming up from the bottom.

4.11.1. The Operating Curve and Equilibrium Curve For the Absorber:- To be able to draw these curves, we make the following definitions.

(a) Mole fraction of ammonia in the vapour (y) is

$$y = \frac{\text{Partial Pressure (P) of ammonia}}{\text{Total Pressure (P}_T\text{)}} \quad \text{(Given in solubility Tables) ---(4.55)}$$

From the data for the project (Section 4.1),

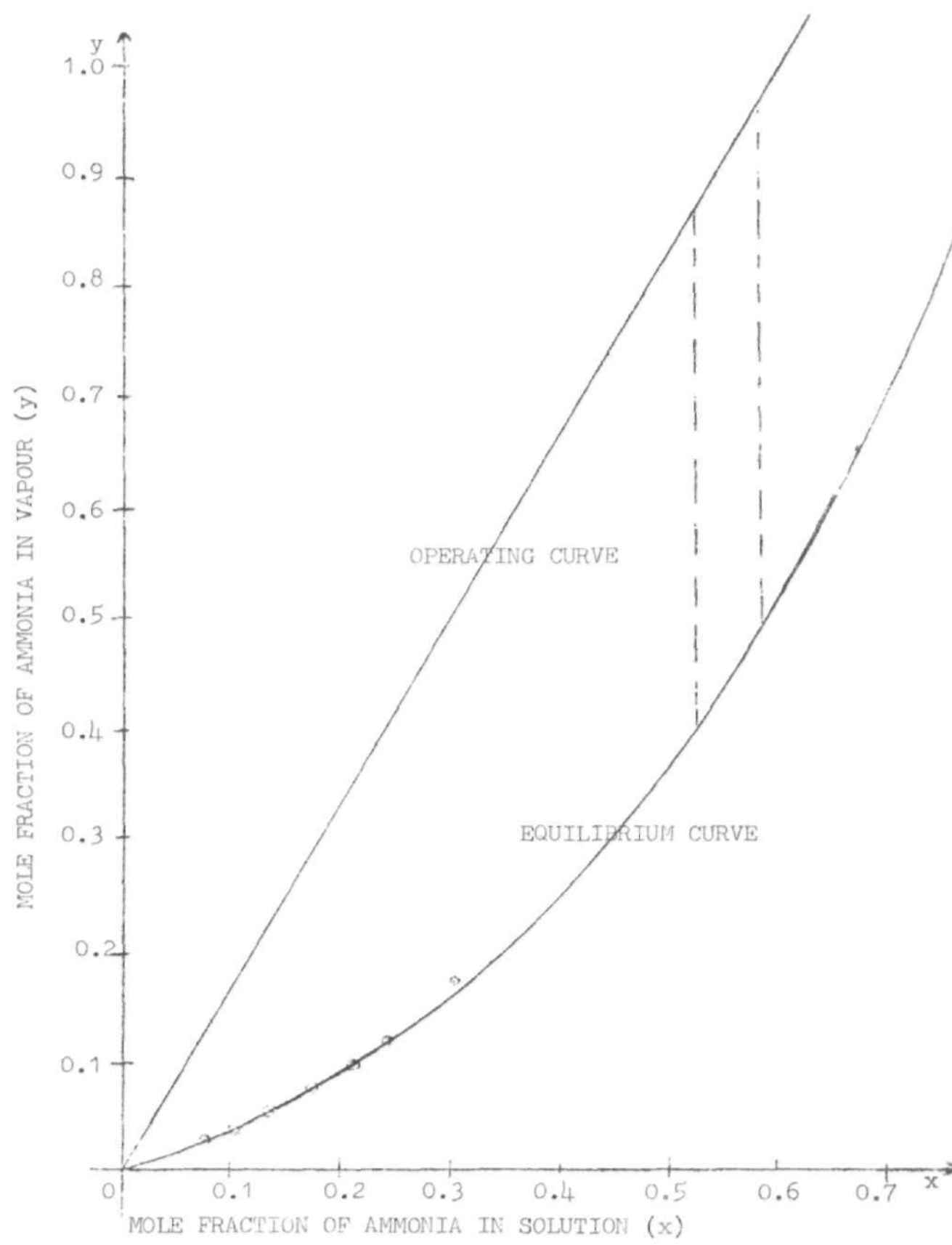
$$P_T = 5.35 \text{ bar or } 5.35 \times 769 \text{ mm Hg}$$

(b) Mole fraction of the ammonia in the solution(x) is

$$\begin{aligned} x &= \frac{\frac{m_A}{17}}{\frac{m_A}{17} + \frac{m_w}{18}} \\ &= \frac{\frac{C}{17}}{\frac{C}{17} + \frac{100}{18}} \quad \text{-----(4.56)} \end{aligned}$$

where m_A is mass fraction for the ammonia and m_w is the mass fraction for water. C is the mass of ammonia per 100 masses of water in solution.

FIG. 4.13.: OPERATING AND EQUILIBRIUM CURVES FOR THE
ABSORBER



100.A

We can take $y = 0$ for $x = 0$ since there is always vapour in contact with the liquid. Using this point and the point ($x_b = 0.532$, $y_b = 0.991$) and assuming for simplification that the operating curve is a straight line, we can obtain the operating curve.

4.11.2. Flooding Velocity:- The material used for packing in the absorber has a fixed void volume for liquid passage. As the rate of liquid flow is increased, the voids are filled with liquid and so the cross-sectional area available for gas flow is reduced. For a given liquid flow rate, the vapour flow rate which will cause flooding of the packed column can be found from pressure drop charts^(25,62) of

$$\frac{V_b^2 f_p \left[10^3 \mu_L \right]^{0.1}}{9810 \rho_v} \text{ versus } \frac{L_b}{V_b} \left(\frac{\rho_v}{\rho_{L_b} - \rho_{v_b}} \right)^{0.5}$$

(See Figure 4.15)

Since the vapour entering the absorber is almost pure ammonia, we can obtain from tables

$$\rho_v = 4.25 \text{ kg/m}^3$$

ρ_v is evaluated at the evaporator temperature of 6°C .
Molar gas flow rate at the bottom $V_b = 5.3 \times 10^{-4} \text{ kg/s}$.
Molar liquid flow rate from the bottom of the absorber

$$L_b = 54.8 \times 10^{-4} \text{ kg/s.}$$

(See Table 1).

The Density of the liquid can be obtained from the formula

$$\begin{aligned} \text{Specific Volume} &= 0.85 x_b V_{\text{NH}_3} + (1-x_b) V_{\text{H}_2\text{O}} \text{ --- (4.57)} \\ &= 0.85(0.57)(0.0016) + (0.43)(0.001) \\ &= 0.001205 \text{ m}^3/\text{kg.} \end{aligned}$$

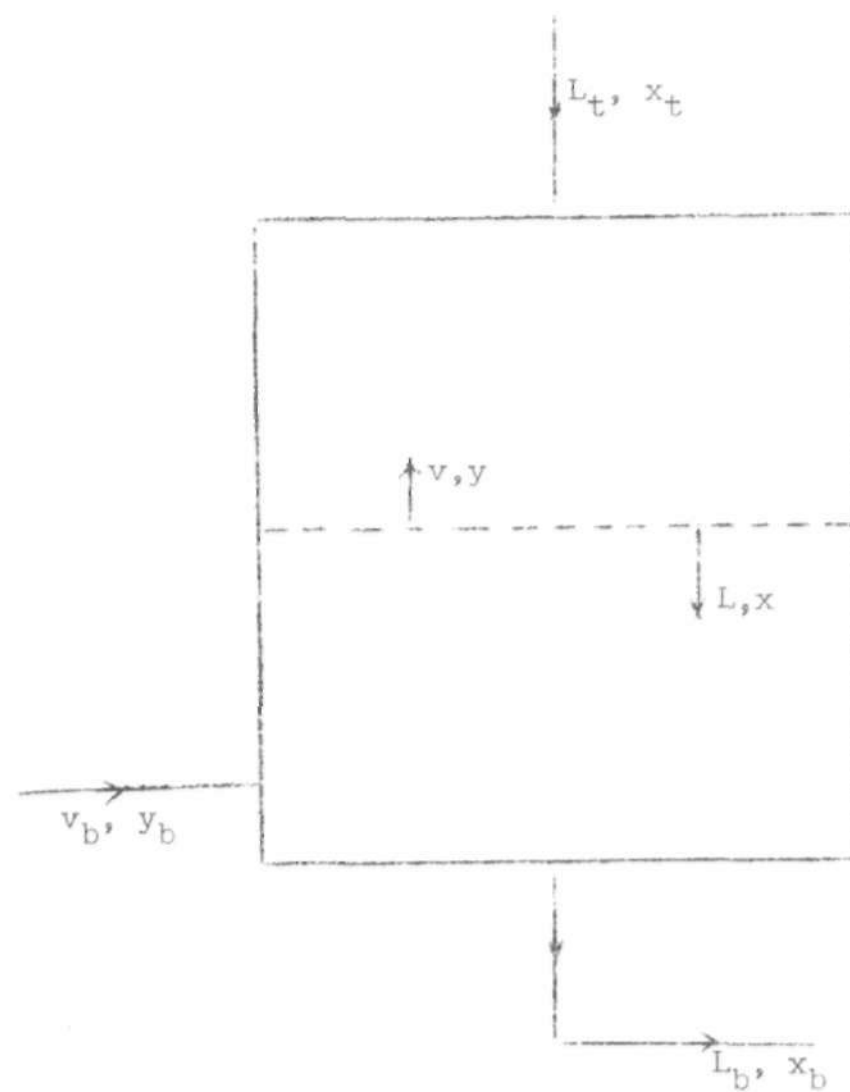
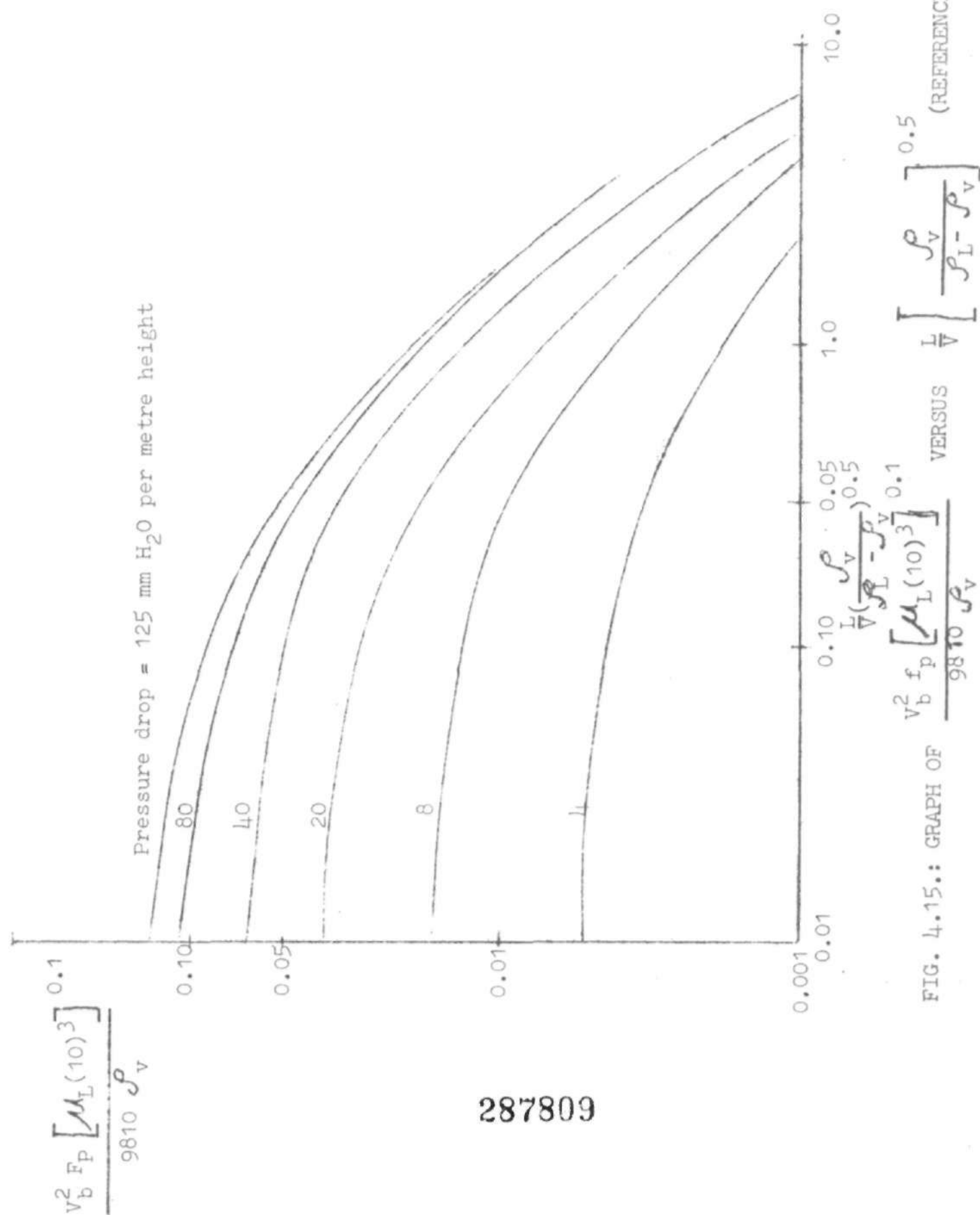


FIG. 4.14.: FLOW OF LIQUID AND VAPOUR INTO, IN AND OUT OF THE ABSORBER



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$$\therefore \rho_{L_b} = 830 \text{ kg/m}^3$$

Therefore,

$$\begin{aligned} \frac{L_b}{V_b} \left[\frac{\mathcal{P}_{V_b}}{\mathcal{P}_{L_b} - \mathcal{P}_{V_b}} \right]^{0.5} &= \frac{54.8}{5.3} \left[\frac{4.25}{830 - 4.25} \right]^{0.5} \\ &= 0.742 \end{aligned}$$

From the graph of

$$\frac{V_b^2 f_p (10^3 \mu_L)^{0.1}}{9810 \mathcal{P}_{V_b}} \text{ versus } \frac{L_b}{V_b} \left[\frac{\mathcal{P}_{V_b}}{\mathcal{P}_{L_b} - \mathcal{P}_{V_b}} \right]^{0.5}$$

(Figure 4.15)

we find that for the available minimum pressure drop curve of 4 mm H₂O per m height,

$$\frac{V_b^2 f_p (10^3 \mu_L)^{0.1}}{9810 \mathcal{P}_{V_b}} = 0.0023 \text{ ----- (4.58)}$$

Assuming that the packings in the absorber will be of ceramic Rashnic ring packing of size $\frac{1}{2}$ in, then taking the necessary data^(61,62),

$$\text{Packing factor, } f_p = 1900 \text{ /m.}$$

We are using data for Rashnic ring packing instead of Rashnic glass packing which is the available packing because of unavailability of data for it.

Taking viscosity of solution as that of water at absorber temperature of 34°C, then

$$\mu_{L_b} = 733.8 \times 10^{-6} \text{ kg/m s}$$

Hence, equation (4.58) becomes

$$\frac{V_b^2 \times 1900(10^3 \times 733.8 \times 10^{-6})^{0.1}}{9810 \times 4.25} = 0.0023$$

$$\therefore V_b^2 = 0.0521$$

$$\text{or } V_b = 0.23 \text{ kg/s m}^2$$

To avoid flooding we take

$$V_b = 13\% \text{ of } 0.23 \text{ kg/s m}^2$$

$$\text{or } V_b = 0.03 \text{ kg/s m}^2$$

4.11.3. Diameter of the Absorber Column:- The cross-sectional area A_A of the absorber column to give the design flow rate is given by

$$A_A = \frac{\pi D_A^2}{4} = \frac{\text{Vapour Flow Rate}}{\text{Design Vapour Flow Rate per m}^2}$$

$$\text{i.e. } \frac{\pi D_A^2}{4} = \frac{5.3 \times 10^{-4}}{0.03}$$

Therefore

$$D_A = \sqrt{\frac{4 \times 5.3 \times 10^{-4}}{\pi \times 0.03}} = 0.15 \text{ m or 6 ins.}$$

4.11.4. Height of the Absorber:- We find^(25,40,41) that a convenient method of obtaining the height of a counter-current absorption column or plate column is based on the concept of the "height of transfer unit" which was introduced originally by Chilton and Colburn.

The equation used for calculating the height of

the absorption packing is given by

$$H_A = H_{OG} \times N_{OG} \text{ -----(4.59)}$$

where H_{OG} is the Height of Transfer unit and

N_{OG} is the Number of Transfer units.

Number of Transfer Units, N_{OG}

This is defined as

$$N_{OG} = \int_{y_t}^{y_b} \frac{(1-y)_{lm} dy}{(1-y)(y^* - y)} \text{ -----(4.60)}$$

where y is the mole fraction in vapour

y^* is the equilibrium value for y ,

lm denotes logarithmic-mean difference and so we define

$$(1-y)_{lm} = \frac{(1-y^*) - (1-y)}{\ln\left(\frac{1-y^*}{1-y}\right)} \text{ -----(4.61)}$$

We can evaluate N_{OG} from the equilibrium and operating curves (Figure 4.13). First we obtain the limits of integration

$$\begin{array}{ll} x_t = 0.532 & y_t = 0.89 \\ x_b = 0.59 & y_b = 0.991 \end{array}$$

Consequently, from the earlier definitions, we obtain

Table 4.5 (Page 105)

The area under the curve of $-\frac{(1-y)_{lm}}{(1-y)(y^* - y)}$ versus y (Fig. 4.16) between the limits y_t and y_b is approximately 1.025. This is found by method of counting squares. Therefore,

$$N_{OG} = 1.025.$$

FIG. 4.16.: GRAPH OF $-\frac{(1-y)(y^*)_{lm}}{(1-y)(y^*-y)}$ Vs y

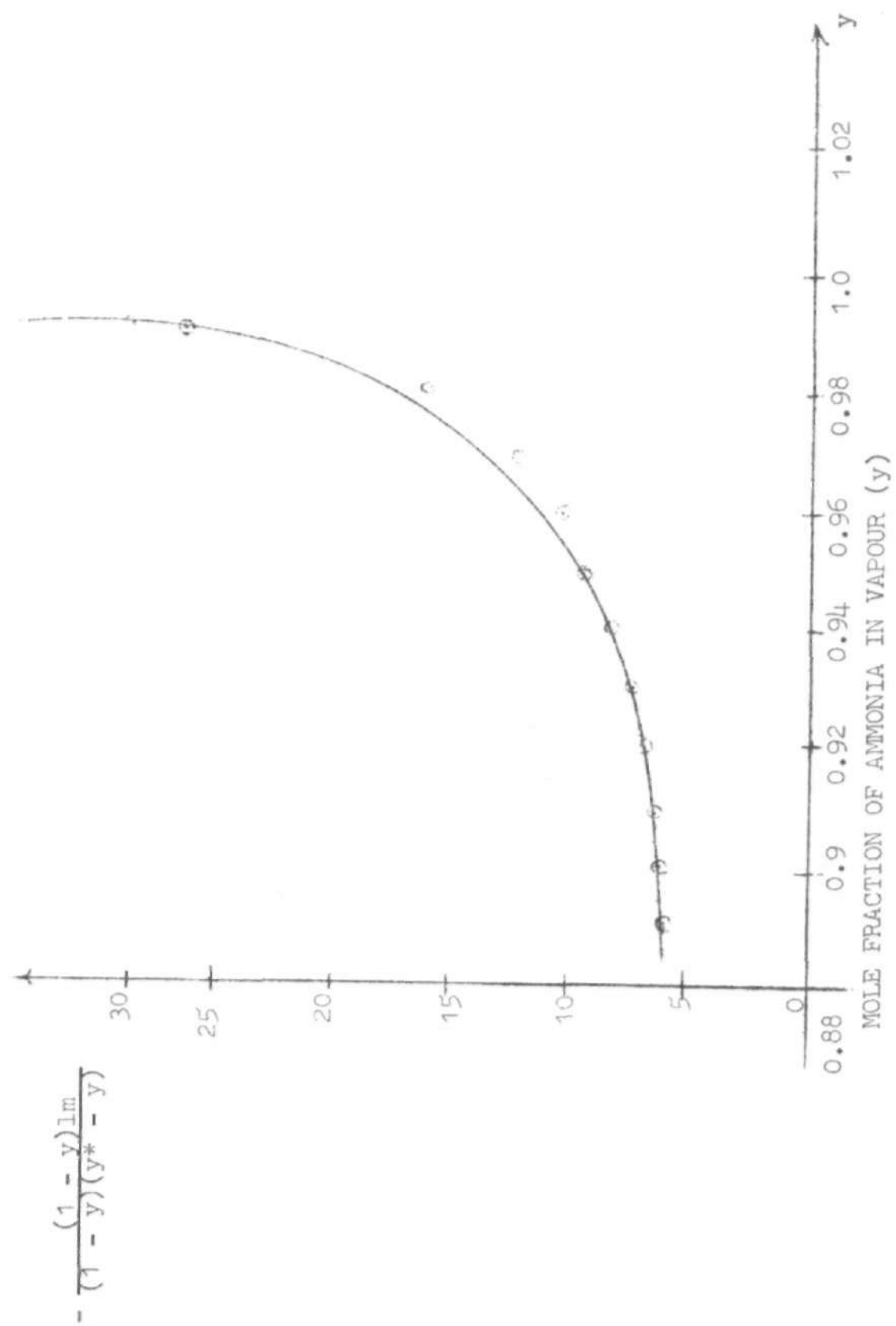


TABLE 4.5.: VALUES OF $\frac{(1-y) \ln}{(1-y)(y^*-y)}$ BETWEEN THE LIMITS OF INTEGRATION y_t and y_b

y	y^*	$1-y$	$1-y^*$	$(1-y) \ln$	y^*-y	$\frac{(1-y) \ln}{(1-y)(y^*-y)}$
0.89	0.468	0.11	0.532	0.268	-0.422	- 5.77
0.9	0.48	0.1	0.52	0.255	-0.42	- 6.07
0.91	0.488	0.09	0.512	0.243	-0.422	- 6.4
0.92	0.495	0.08	0.505	0.231	-0.425	- 6.79
0.93	0.505	0.07	0.495	0.217	-0.42	- 7.29
0.94	0.52	0.06	0.48	0.202	-0.42	- 8.02
0.95	0.53	0.05	0.47	0.187	-0.42	- 8.9
0.96	0.535	0.04	0.465	0.173	-0.425	-10.18
0.97	0.543	0.03	0.457	0.157	-0.427	-12.26
0.98	0.55	0.02	0.45	0.138	-0.43	-16.05
0.99	0.56	0.01	0.44	0.114	-0.43	-26.51
0.991	0.562	0.009	0.438	0.110	-0.429	-28.49

Height of Transfer Units, H_{OG}

This is given ^(25,40,45) as

$$H_{OG} = \frac{V_A}{K_V a A_A (1-y) \ln} \text{-----(4.62)}$$

where

V_A is the rate of mass transfer for the absorber

a is the specific surface packing

A_A is the cross-sectional area of the absorber, and

K_V is the overall mass transfer coefficient for the vapour.

It is also known ^(25,40) that

$$V_A = K_V (y - y^*) \text{-----(4.63)}$$

V_A which is the rate of mass transfer for the absorber is the rate of absorption of ammonia gas. Since all the ammonia that enters is assumed absorbed, then

$$\begin{aligned} V_A &= \frac{5.3 \times 10^{-4}}{(0.991 \times 17) + (0.009 \times 18)} \\ &= 0.312 \times 10^{-4} \text{ kg mol/s} \\ &= 17.66 \times 10^{-4} \text{ kg/mol s m}^2 \end{aligned}$$

The mean value of $(y - y^*)$, the driving force is obtained from Table 4.5 and this is equal to 0.424.

Hence, mean K_V

$$\begin{aligned} &= \frac{17.66 \times 10^{-4}}{0.424} \\ &= 41.65 \times 10^{-4} \text{ kg mole/s m}^2 \end{aligned}$$

From data on Rashnic ring packings^(61,62) $a = 367 \text{ /m}$

From Table 4.5 the mean value of $(1 - y)_{lm}$ is = 0.191.

Hence from equation (4.62), we obtain

$$\begin{aligned} H_{OG} &= \frac{17.66 \times 10^{-4}}{41.65 \times 10^{-4} \times 367 \times \frac{\pi}{4} (0.15)^2 \times 0.191} \\ &= 0.342. \end{aligned}$$

To obtain the height of the absorber, H_A , we substitute in equation (4.59) and have

$$\begin{aligned} H_A &= N_{OG} \times H_{OG} \\ &= 1.025 \times 0.342 \\ &= 0.351 \text{ m.} \end{aligned}$$

Because of the necessary clearances, etc., we take the height of the absorber to be 0.4 m.

4.11.5. Cooling of the Absorber:- The absorber will be a cylindrical column of internal diameter 0.15 m (6 ins) as calculated and from available data, the thickness will be 5 mm. It may have rectangular or circular fins welded or soldered on the outside for air cooling. If we refer to Table 4.3, the heat that will be required to be removed from the absorber by the ambient air for cooling is given by

$$\begin{aligned} Q_A &= m h_2 + m_8 h_8 - m_3 h_3 \\ &= 890.66 \text{ W} \end{aligned}$$

or 0.891 kW.

Figure 4.17 illustrates a cross-section through the packed absorber column. The heat transfer equation for this situation can be written as

$$Q_A = U \times A \times (t_A - t_a) \text{ -----(4.64)}$$

where A is the outside surface area,

U is the overall heat-transfer coefficient based on the outside surface area.

$$d = 0.15 \text{ m or 6 ins.}$$

$$r = \frac{d}{2} = 0.075 \text{ m.}$$

Since the thickness of the tube is 5 mm, then

$$D = 0.16 \text{ m, and}$$

$$R = 0.08 \text{ m.}$$

The inside of the absorber will be packed with Rashnic glass and we shall assume that the uniform absorber temperature of 34°C will be maintained. Hence heat transfer to the ambient cooling air will be of

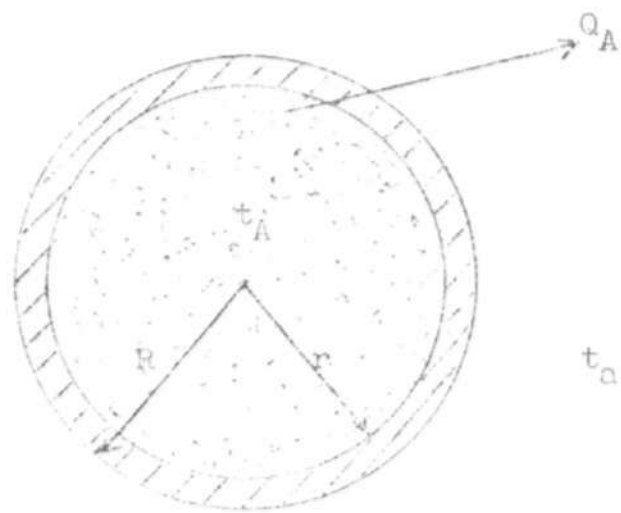


FIG. 4.17.: CROSS-SECTION THROUGH THE PACKED ABSORBER
COLUMN

the form of radial steady conduction through the wall of the tube. Therefore,

$$Q_A = \frac{2\pi L K (t_A - t_a)}{\ln\left(\frac{R}{r}\right)}$$

Hence the required length of the absorber for this heat transfer to the ambient air

$$\begin{aligned} L &= \frac{Q_A \ln \frac{R}{r}}{2\pi K (t_A - t_a)} \\ &= \frac{0.891 \ln\left(\frac{0.08}{0.075}\right)}{2\pi \times 0.0435 \times (34 - 32)} \\ &= 0.105 \text{ m} \end{aligned}$$

The packed height of the absorber has already been calculated to be 0.4 m. The height of 0.4 m will therefore be adopted as the design height of the absorber since it is the longer of the two. Also, there is no need for the use of fins as this height is adequate enough for the air cooling.

4.11.6. Conclusion:- This chapter gives the data on which the design of the system was based. It also presents the design of the various components of the system as well as the basis on which other necessary equipment were selected.

CHAPTER FIVE

RESULTS, CONCLUSIONS AND RECOMMENDATIONS

5.1. CHARGING PROCEDURE

Two charging valves were each installed on pipelines connected to the suction and the discharge sides of the pump. They were used for the charging of the system. The system was first evacuated by connecting the vacuum pump to the valve on the discharge side of the pump while the valve on the suction side charging line was closed. After evacuation, the valve was closed and the vacuum pump was disconnected. Using the vacuum created in the system, the ammonia solution was then sucked into it by means of the charging line on the suction side of the pump until the system pressure reached atmospheric. Just before the suction action stopped, the pump was started so as to complete the feeding of the required amount of ammonia solution into the system.

5.2. OBSERVATION

It was observed that the pump used in the system could not suck the ammonia solution because it required priming which was not possible because of the prevailing nature of the project. As a result, more ammonia solution could not be got into the system than that due to the vacuum action. The amount charged was only about half of that required. To increase the

concentration of the ammonia solution, some ammonia gas was fed in from the gas cylinder.

It was also observed that since the pump is not self-priming, it could also not perform its role of circulation of refrigerant in the system. Moreover, the ammonia attacked the pump seal and some other parts of the pump and caused it to leak after about 18 hours. All efforts made to get the proper type of pump failed.

Because of the failure of the pump to perform, the ammonia vapour generated in the collector-generator could not circulate to the condenser and evaporator to do the required refrigeration. However, when the ammonia solution was discharged and a vacuum pump used to force the vapour trapped at the top header of the solar collector and the rectifier to move to the condenser and consequently to the evaporator, considerable refrigeration was achieved. The result obtained for such refrigeration on the second day of operation is given below, on page 111.

A much lower temperature of the evaporator surface was obtained the previous day when the pump had not started to leak. The surface of the pipe was cooled to 18°C which is the dew point of the air around it in the cold room as condensation was observed on the surfaces. The ambient temperature then was about 34°C . Leakage from the pump as a result of ammonia action reduced the efficiency of the vacuum pump operation, thus accounting for the poor result reported in Table 5.1, obtained on the second day.

TABLE 5.1: RESULT

TIME (A.M.)	AMBIENT TEMPERATURE(°C)	TEMP. OF THE SURFACE OF THE EVAPORATOR (°C)	COLD ROOM TEMPERATURE (°C)
9.30	29	28	28.5
10.15	31	28	28.5
10.30 Vac. Pump started	32	27.5	28
10.45	33.5	25.5	27.5
11.00	34.5	25	27
11.15	35	24	27
11.30	36	24.5	27
11.45	35.5	25	27
12.00 Vac. Pump stopped	35.5	25.5	27

5.3. CONCLUSION

The aim of the project can be said to have been substantially achieved. The system is considerably simple and the materials used are easily available and can be fabricated with available equipment. Refrigeration was achieved although it was intermittent as a result of the failure of the available pump to perform properly and inability to obtain the right type of pump. However, with the installation of a self-priming pump whose material make-up is ammonia-resistant, and the incorporation of the suggestions given for further work, a lower temperature and better refrigeration would be obtained.

5.4. RECOMMENDATIONS FOR FURTHER WORK

Care should be taken in the selection of the pump to make sure that it can withstand the pressure



PLATE I: THE SYSTEM AND COLD-ROOM



PLATE II: THE AUTHOR ADJUSTS A VALVE IN THE SYSTEM

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