PROSPECTS OF SOLAR ICE MAKING

BY

PANKAJ DARA

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Department of Mechanical Engineering, Ahmadu Bello University, Zaria, Nigeria.

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NOMENCLATURE

P_c = Condenser pressure

P_e = Evaporator pressure

T_g = Generator temperature

t_c = Condenser temperature

t_e = Evaporator temperature

T_a = Absorber temperature

ξ_d = Refrigerant vapor concentration

ξ_r = Rich solution concentration

ξ_p = Poor solution concentration

f = Rich solution circulation

m_1 = Mass of NH_3 - NaSCN mixture

C_{pg} = Specific heat of mixture at generator temperature

L_{tg} = Latent heat of ammonia at evaporator temperature

C_{pa} = Specific heat of ammonia at evaporator temperature

H_T = Total heat absorbed by collector

U_L = Total heat loss coefficient

t_p = Metal plate temperature of collector

t_a = Ambient temperature

I_{Dt}, I_{dt} = Direct and difuse radiation intensity

$(\alpha\tau)_D$, $(\alpha\tau)_d$ = Effective transmissivity and absorptivity product for direct and diffuse radiation.

R_D, R_d = Orientation factors for direct and diffuse radiation

H & D = Instantaneous value of total and diffuse radiation on horizontal surface

θ_t = Angle of Incidence

θ_h = Zenith angle

I_t = Total Radiation Absorbed by collector

β = Angle of tilt of collector

ϕ = Latitude angle

ω = Hour angle

δ = Declination angle

$\overline{H}$ & $\overline{D}_h$ = Monthly average daily and hourly diffuse radiation on horizontal surface

H_o = Extraterrestial daily insolation received on horizontal surface

$\overline{K}_T$ = Average cloudiness index.

I_{sc} = Solar constant

D = Diameter of Headers.

d = Diameter of Pipes

h_r = Refrigerant side heat transfer coefficient.

h_w = Water side heat transfer coefficient

h_b = Brine side heat transfer coefficient

A_o & A_i = Outside & Inside area of coil.

K_q = Thermal conductivity of Insulation

$\bar{H}$ & $\bar{H}_h$ = Monthly average daily and hourly total radiation on horizontal surface.

L_t = Thickness of Insulation used

μ = Dynamic Viscosity

ν = Kinematic Viscosity

ρ = Density

K = Thermal conductivity.

ω_s = Sunrise Hour Angle

V = Velocity

σ° = Surface Tension at liquid vapor interface

l = Length of coil

C_{sf} = Constant

g_c = Gravitational accleration

$\dot{m}$ = Mass flow rate

q = Boiling Heat Flux

ΔT = Temperature difference

L_c = Latent heat of condensation

L_v = Latent heat of vaporization

r_2 = Outer Radius

r_1 = Inner Radius

q_r = Heat rejected in condenser.

N = Number of Glass covers

n = Day of the year

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CHAPTER 1

INTRODUCTION

1.1 Scope of Solar energy

In recent years the growing public awareness of fossil fuels and incresed costs of conventional energy has led to the search of new sources of energy such as solar, wind, tidal, geothermal etc. As a result of the energy crises in 1973, the utilization of solar energy has attracted much attention for many purposes, including refrigeration and air conditioning.

There is however, a need for solar powered refrigeration that does not arise from increasing oil prices and the more acute awareness of the limited reserves of fossil fuels. In many developing Countries the lack of infrastructure severly limits the application of refrigeration for storage of perishable food stuffs which is essential for developing the potential of increased production that exists in many areas. In such Countries solar powered refrigerating systems which are of simplg design and donot depend either on other energy sources or on the availbility of trained servicemen, may be attractive.

Nigeria, being one of the developing Countries and lying in the Tropics L(5-15°N) must try to harness this

source of energy. Solar energy is virtually free and is effectively inexhaustible. It does not involve any form of environmental pollution and can be collected at the point of application. Based on Nigerian meteorological data[1] the monthly average of global solar radiation incident in Nigeria is 13.4×10^4 KCal/m^2/month. Thus we receive about 161×10^4 KCal/m^2/year which gives the yearly incidence of solar energy of approximately 64.5×10^6 KWh/year. Even if a very small percentage of this energy could be harnessed we shall be able to meet the energy needs of this Country.

The use of solar energy for cooling purposes has two fold attraction.

(a) The demand for cooling and ice making is the greatest at the time of maximum solar intensity in contrast to the uses of solar energy for heating.

(b) The cooling for food preservation and for Comfort air conditioning is more important in hot regions than in cold regions. Refrigeration and air conditioning that may otherwise be condidered luxuries, may infact become neccessities for better working conditions.

It is thus not surprising that on the subject of solar

cooling, studies are being conducted in various research institutes all over the world.

1.2 Refrigeration Systems

The two refrigeration systems prevalent now are the vapor compression refrigeration system and the vapor absorption refrigeration system the basic difference between the two systems is that in the former there is a vapour compressor, whose function is to remove the vapor from the evaporator and to raise the temprature and pressure of the vapor to a point such that the vapor can be condensed with normally available condensing media, whereas in the latter heat energy is used instead of mechanical to make a change in conditions necessary to complete a refrigeration cycle.

The condenser, receiver and evaporator (cooling coil) are quite similar to those used in the compression system, but the compressor has been replaced by a heater and generator.

However, in this country vapour compression systems are more prevalent. they use only high grade energy such as electricity or mechanical work. Only in recent years has some work been started in various parts of the world where

the use of low grade energy such as solar energy waste steam (heat) is possible. Therefore the absorption system which utilizes heat energy directly, rather than electrical energy seems to be most suitable for use with solar energy.

Two approaches have been made to operate a solar absorption refrigeration system.

1.2.1 Continuous Absorption Refrigeration System

In case of a continuous system, normally, the heat supplied to the generator is provided directly by a gas flame or indirectly by some other heating media such as hot water, steam or some other fluid heated by solar energy. Solar energy being an intermittent heat source, it is necessary either to store for the time it is not available or to use any other electrical or thermal heating system in conjuction with a solar generator.

Further, the continuous system needs a liquid pump to increase the pressure of working fluid from absorber pressure to the generator pressure which requires some form of energy to run it.

1.2.2 Intermittent Absorption System

In the intermittent absorption system one cycle gets completed in 24 hours. Regeneration of refrigerant occurs

during the day and evaporation takes place during the night.

Solar energy is an intermittent heat source. The intermittent vapor absorption refrigeration system, therefore, affords a likely means of utilizing solar radiation directly to provide cooling. It does not require storage of the solar energy or any other electrical or thermal heating system due to its inherent intermittancy.

In this system the distillation of the refrigerant from the absorbent occurs during the regeneration stage of operation, and the refrigerant is condensed and stored in a condenser receiver. During the cooling portion of the cycle refrigerant from condenser goes to the evaporator due to the difference in pressure, where it is evaporated and then reabsorbed in generator/absorber. Because the generator absorber is a single unit, this sytems removes the use of any liquid pump.

Therefore for the same capacity the size of the intermittent system is more in comparison to the continuous refrigeration system. However, because of its inherent simplicity the intermittent system will be considered for our design.

1.3 Refrigerant - absorbent selection

The selection of a refrigerant absorbent is the most important aspect to get a good performance with the available climatic conditions. In order to obtain a reasonably high coefficient of performance one should look for a refrigerant absorbent combination for which

1. The absorption capacity of absorbent is large.
2. The specific heats of the absorbent and refrigerant are low.
3. The heat of reaction is small.
4. The vapor pressure of the absorbent at the generator temprature is low. Though various refrige rant - absorbent combinations have been studied in the past keeping in mind the above points, but none of the projects had so far been carried beyond the experimental stage nor had a systematic evaluation of the merits & demerits of various working substance for this specific purpose been made Only some attempts have been made to study the following refrigerant absorbant cominations by making experimental trials.

(a) Water - Ammonia (H_2O - NH_3)

(b) Sodium thiocynate - Ammonia ($NaSCN$ - NH_3)

(c) Littium Nitrate - Ammonia ($Li\ No_3$ - NH_3)

(d) Calcium chloride - Ammonia ($CaCl_2$ - NH_3)

(e) Stronthium chloride - Ammonia ($SrCl_2$ - NH_3)

The NaSCN - NH_3 system, in which absorbent is a solution of NaSCN in NH_3 with ammonia as the refrigerant, has been studied by Blytas and Daniels[2] and Sargent and Beckman[2]. This system appears to have good thermodynamic properties for cycles for ice making.

Chinnapa[2] and Swartman and Swaminathan have experimentally studied the operation of H_2O- NH_3 machine using flat plate collectors as energy supply. The NaSCN - NH_3 system seem to be suitable for refrigerating systems using low temprature heat source such as solar energy. This solution possesses many desireable properties. It is inexpensive, non explosive and non corrosive in Iron and Steel. It has relatively low vapor pressure & viscosity, and high thermal conductivity and heat of vaporization. Moreover because of its availability in the local premises this combination will be considered for our design. This work aims at establishing the feasibility of the use of sodium thiocyanate - ammonia combination in intermittent solar ice maker.

CHAPTER 2

REVIEW OF LITERATURE

In the last few decades use of solar energy for refrigeration has gained widespread attention in different parts of the world. Lots of work have been done on vapor absorption refrigeration system using solar energy and most of the earlier work is concentrated on the development of continuos vapor absorption system The continuos systems are now well understood but little work has been done on the development of intermittent systems. In recent years the intermittent system has become a subject for further exploration and thus considerable work is in progress at different research centres all over the world. Few of the works are listed below.

An experimental investigation[3] of the absorption and desorption of Ammonia and Calcium Chloride and strontium Chloride were carried out in Khartoum to establish suitable design parameters. To serve as a demonstration plant a small drinking water cooler with a $4m^2$ collector area was built. On a clear day an overall C.O.P. of 10% was obtained corresponding to an ice production of 6 kg/m^2 of collector area.

As a preliminary step in the above project the merits of various absorbents and refrigerants were examined. In the experiments carried out, for tempratures of - 10°C and below Ammonia was found to be unsurpassed as the refrigerant. Three liquid absorbents (H_2O, NaSCN, $LiNo_3$) and two solid absorbents ($CaCl_2$, $SrCl_2$), all with NH_3 as refrigerant were compared on the basis of a simple in termittent process. For the NH_3/H_2O system rectification of vapor was included whereas heat exchange between strong and weak solutions was not considered for any of the wet systems.

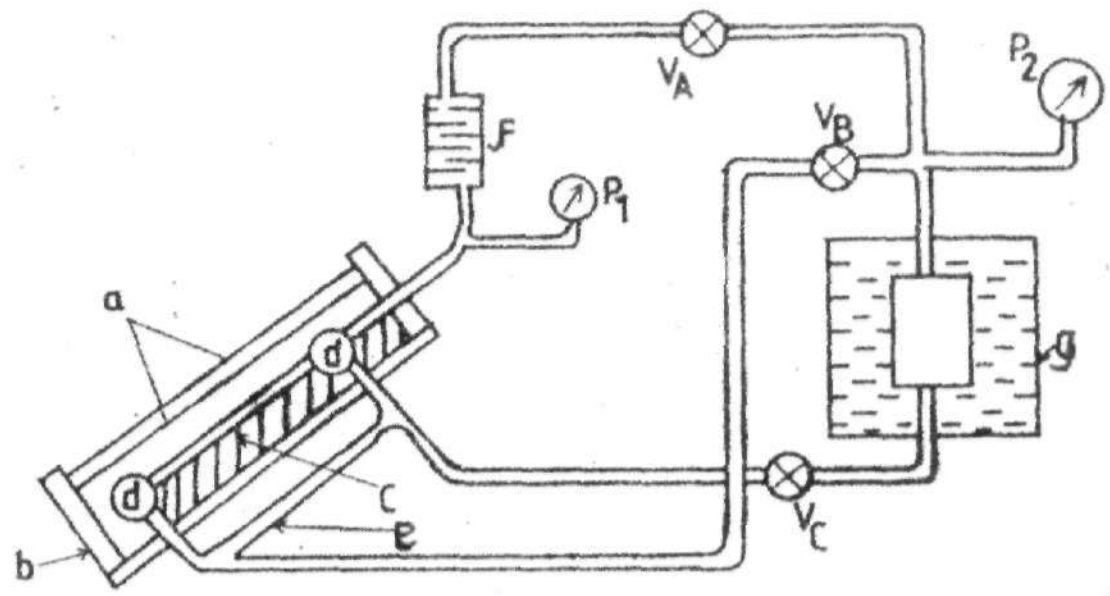
It was found that with a liquid absorbent the Coefficient of Performance of the process (defined as ratio of het amount of cooling to the total of energy supplied to the desorber) depends critically on the final temprature that can be reached during desorption. As the final temprature is raised more refrigerant is driven off the solution i.e. the concentration difference between the strong and weak solution increases.

As a consequence the amount of solution the heating of which to the desorption tempreture const ituets a significant loss - decreases and C.O.P. goes up.

In a Continuos cycle this loss may be reduced due to regenerative heat exchange between strong and weak solution, However for an intermittent svstem-and particularly one with out the solution pump - it is not alwavs possible and at the best difficult to arrange for a reasonablv effective heat exchange between the strong and weak solution. This is illustrated by the absorption cooling svstem of wet type shown below in Fig (2.1)

For a given final desorption temprature the lowest obtainable evaporation temprature (t_e) is determined by the minimum temprature (t_e) during absorbtion. The more the evaporation temprature (t_e) is lowered the smaller the concentration difference between strong and weak solution becomes and hence larger the amount of solution that is needed. This explains why C.O.P. drops off rapidlv below a certain evaporation temprature (t_e). For the same reason the wet system is sensitive to changes in the condensing temprature (t_c) These characteristics are illustrated in figures (2.2 & 2.3) for H_2O/NH_3 system. The LiBr/NH_3 & NaSCN/NH_3 system behave similarly, the C.O.P of the former being higher, that of the latter slightly lower than the value found for H_2O/NH_3 system.

TYPICAL LAYOUT OF A SMALL EXPERIMENTAL ABSORBTION COOLING SYSTEM OF 'WET TYPE', THE SYSTEM SHOWN WORKS ON NH_3 + H_2O

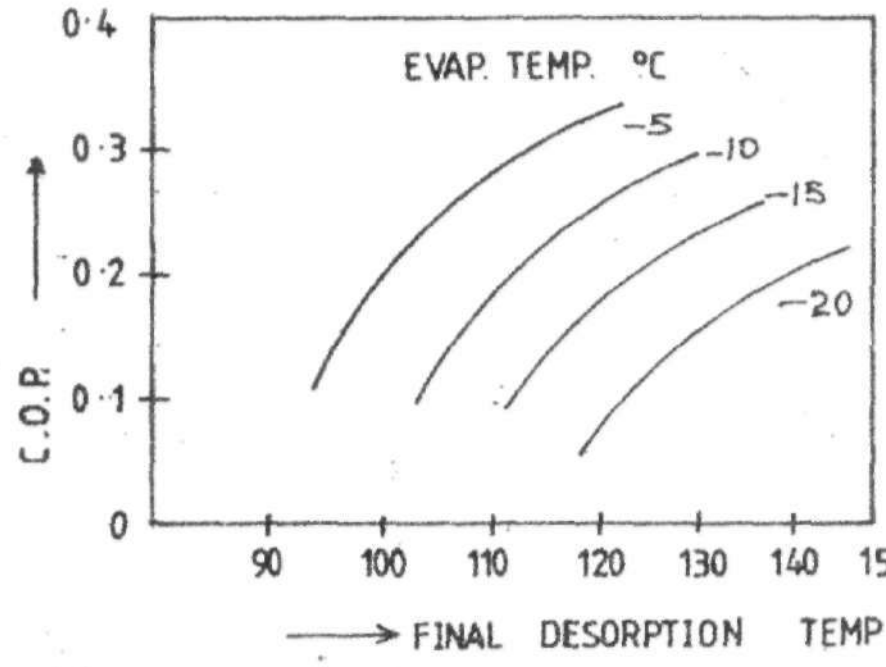


a) 3mm glass sheets
(b) Collector frame
(c) Glasswood insulation
(d) Lower and upper headers
(e) Return . line
(f) Rectifier
(g) Condenser cum evaporator

V_A V_B V_C VALVES

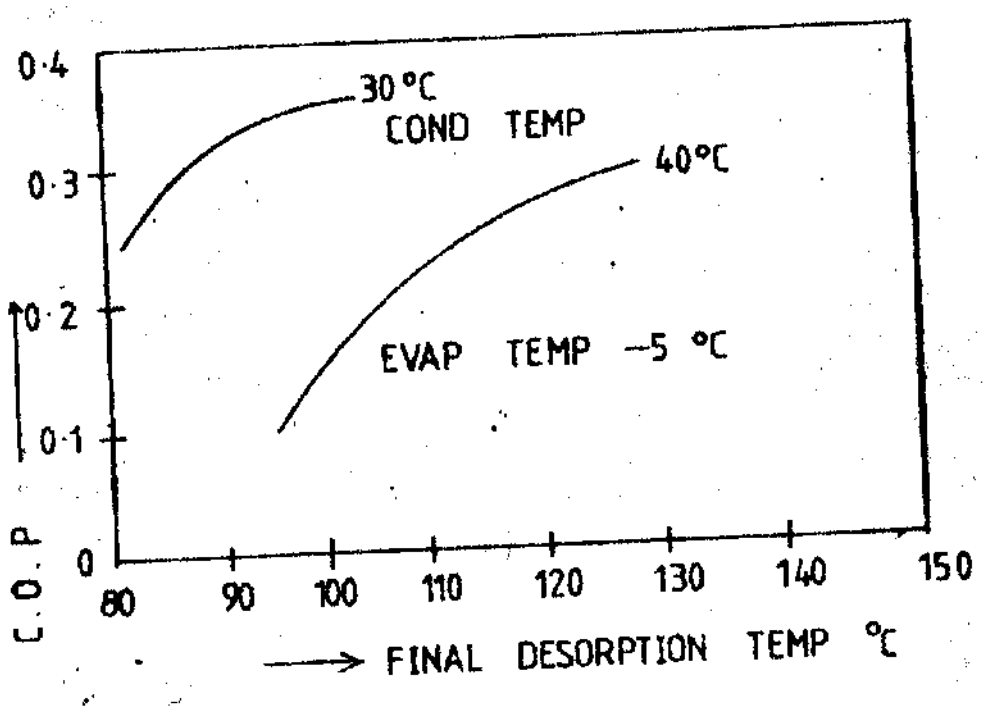
P_1 P_2 PRESSURE GAUGES

FIG (2.1) REF-4



C.O.P OF INTERMITTENT H_2O/NH_3 ABSORPTION COOLING SYSTEM. ABSORBER MASS CORRESPONDING TO 10 KG/KG NH_3 GENERATED, CONDENSER TEMP. 40°C, EVAPORATOR TEMP °C

FIG. (2.2). REF.4



C.O.P. OF INTERMITTENT H_2O/NH_3 ABSORPTION COOLING SYSTEM. ABSORBER MASS CORRESPONDING TO 10 KG/KG NH_3 GENERATED.

FIG. (2.3). REF 4

A larger concentration difference can be obtained without increasing the final desorption temprature by employing two stage desorption at the expense of however a lower Co-efficient of performance. This sort of a system has been used in Afganistan for a dairy project. With a solid absorbent the problems are rather different. Absorption and desorption occur in one or more steps at equilibrium tempratures which are solely determined by the pressure in the system. $CaCl_2$ absorbs a to tal of 8 moles of NH_3. The reaction taking place is as follows:-

$$CaCl_2 + 2NH_3 \rightleftharpoons CaCl_2\ 2NH_3 + L_{o2} \quad - \quad (i)$$

$$CaCl_2\ 2NH_3 + 2NH_3 \rightleftharpoons CaCl_2\ 4NH_3 + L_{2-4} \quad - \quad (ii)$$

$$CaCl_2\ 4NH_3 + 4NH_3 \rightleftharpoons CaCl_2\ 8NH_3 + L_{4-2} \quad - \quad (iii)$$

The two moles absorbed initially are so strongly bound that decomposition requires tempratures far in excess of those obtainable in a refrigerating system.

For a system $SrCl_2/NH_3$ the reactions are

$$SrCl_2 + NH_3 \qquad SrCl_2\ NH_3 + L_{o-1} \quad - \quad (iv)$$

$$SrCl_2\ NH_3 + 7NH_3 \qquad SrCl_2\ 8NH_3 + L_{1-8} \quad - \quad (v)$$

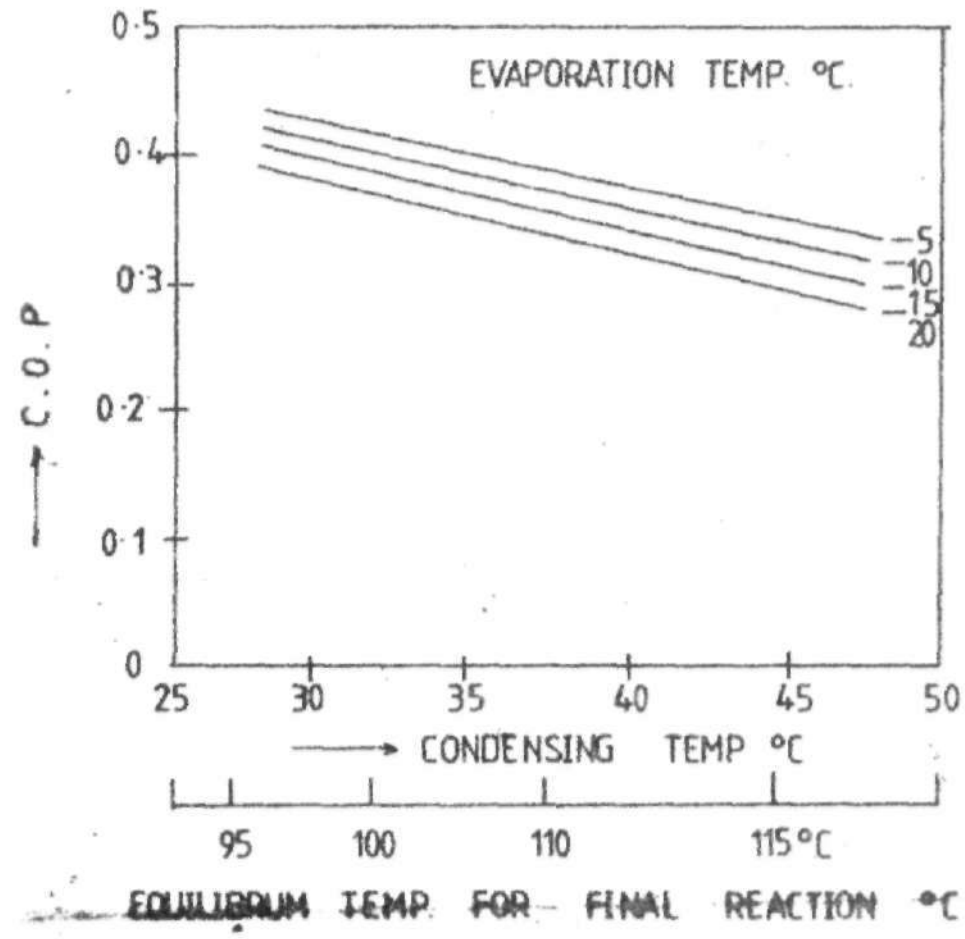
Again one mole of ammonia initially absorbed is very strongly bound. The equilibrium tempratures for the secondreaction

fall inbetween those for the system $CaCl_2/NH_3$.

The heats of reaction for (ii) (iii) (v) are nearly equal when referred to a unit mass of NH_3 and about twice the latent heat of evaporation. This means that C.O.P. for a refrigerating system will never exceed 0.5. It will in fact be somewhat lower because part of the energy supplied goes into heating the absorber from the absorption to the desorption temperature. On the other hand the C.O.P. of a solid absorption system is not very sensitive to changes either in the condensing temperature or the evaporation temperature. The problem here is whether sufficently high temperatures can be obtained during desorption and sufficent cooling provided during absorption.

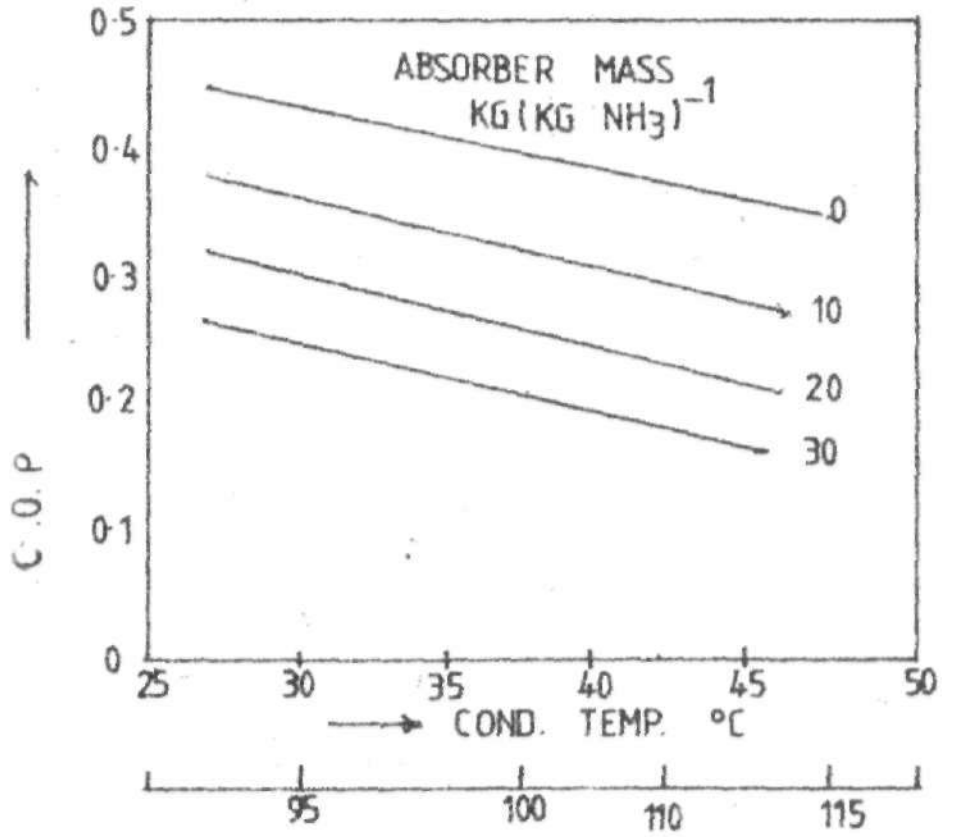
The performance of a $CaCl_2/NH_3$ system is shown in figures (2.4) and (2.5). From figure (2.4) which refers to a system with an absorber mass of 10 kg/kg of Ammonia generated it is seen that C O P lies consistently above that of the H_2O/NH_3 system and show only slight variation with condensing and evaporating temperatures. Figure (2.5) emphasizes the importance of a light absorber design.

For a system that is powered by free energy the C.O.P



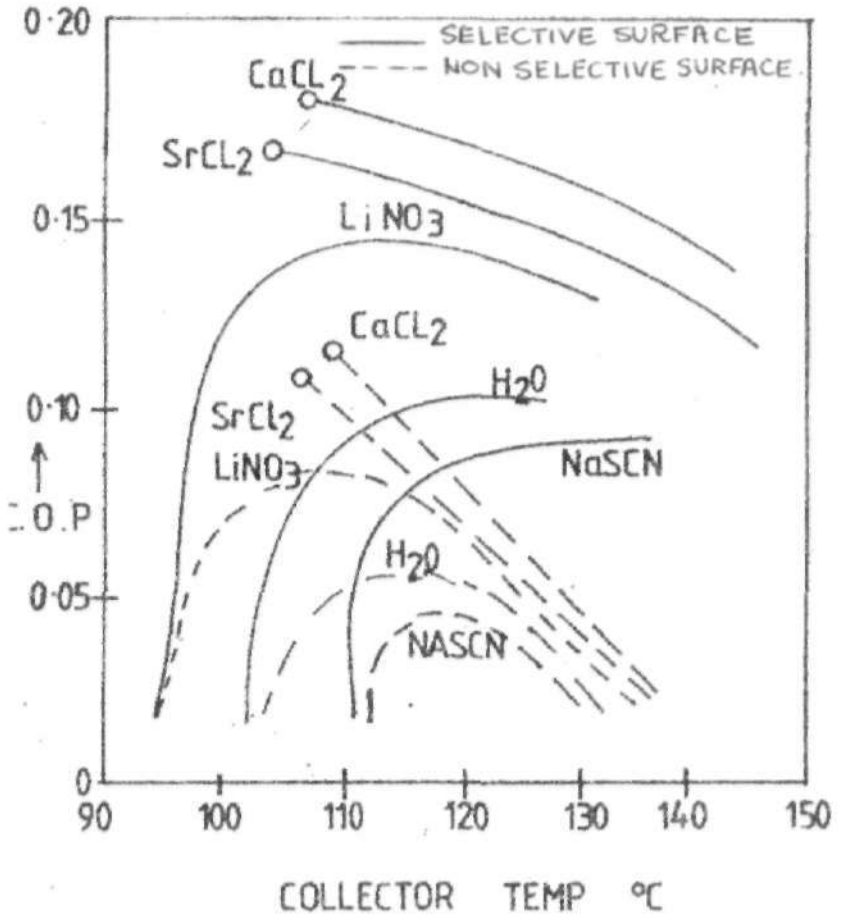
C.O.P FOR AN INTERMITTENT SOLID ABSORPTION $CaCl_2/NH_3$ COOLING SYSTEM. ABSORBER MASS CORRESPONDING TO 10 KG/KG AMMONIA GENERATED.

FIG (2.4). REF. 4



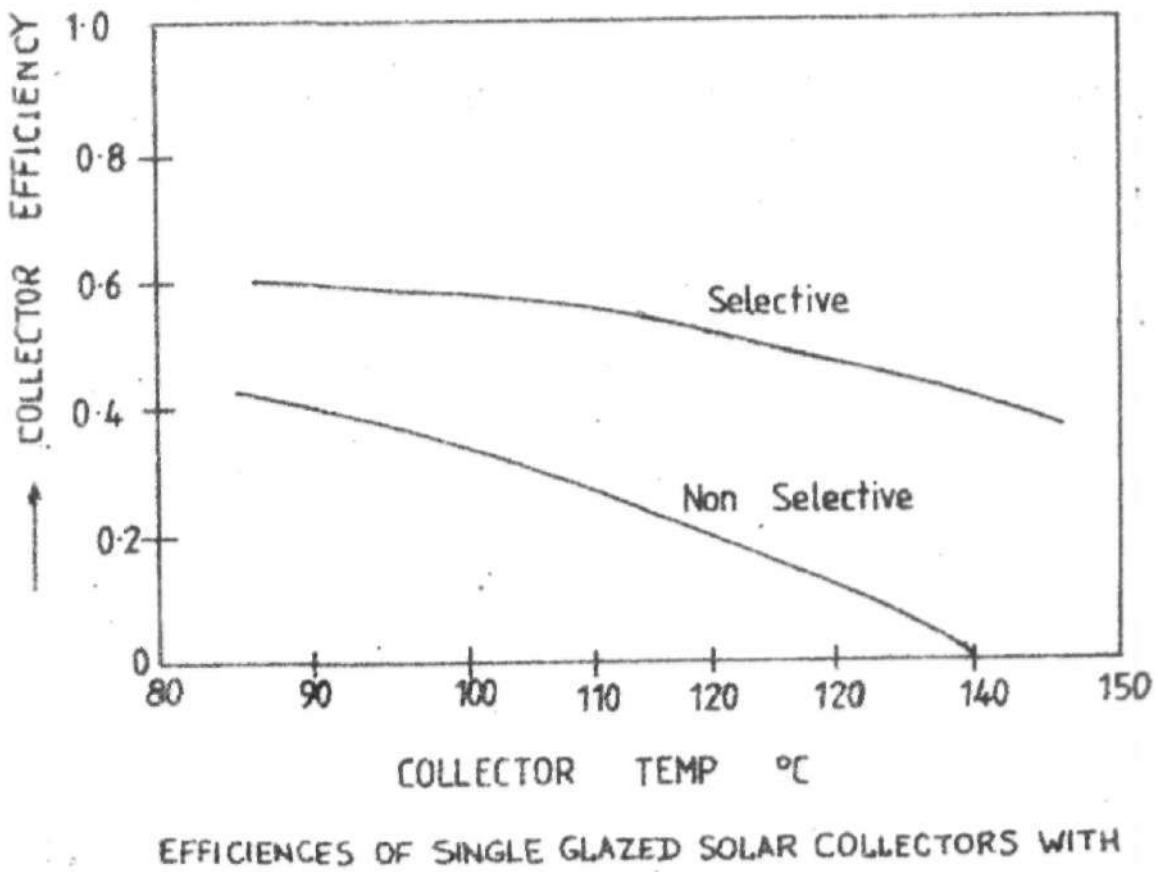
C.O.P. OF INTERMITTENT SOLID ABSORPTION $CaCl_2/NH_3$ SYSTEM WITH DIFFERENT ABSORBER MASS RATIOS.

FIG (2.5). REF 4.



TOTAL C.O.P. OF A SOLAR POWERED, INTERMITTENT ABSORPTION COOLING SYSTEM. EVAP. TEMP. -10°C, CONDENSER TEMP. 40°C. ABSORBER MASS CORRESPONDING TO 10 KG/KG AMMONIA GENERATED

FIG. (2.6). REF 4



EFFICIENCES OF SINGLE GLAZED SOLAR COLLECTORS WITH SELECTIVE COATING ($K = 0.96$, $\epsilon = 0.09$) AND ORDINARY BLACK SURFACE ($K = 0.96$). INSULATION THICKNESS 100 MM DIRECT RADIATION 890 $W m^{-2}$, DIFFUSE RADIATION 150 $W m^{-2}$. AMBIENT TEMPRATURE 31 °C.

FIG. (2.7). REF 4.

[illegible], non selective collector surface. The solar collector/absorber/desorber were intergrated into a single unit with the absorbent contained in horizontal tubes as shown in figure (2.8)

The diagram gives a schematic representation of a solid absorption refrigerating system. The combined solar collector/absorber/desorber consists of a number of horizontal tubes containing the absorbent connected by straight fins. The system goes through a full cycle every 24 hours with absorption and cooling during the night whereas desorption and condensation of Ammonia vapor takes place during 5-7 hours around noon. The liquid Ammonia is stored in the receiver at the bottom of water basin until the temprature of collector and hence the pressure in the system starts decreas ing. The check valve then closes and the liquid ammonia is then transferred to insulator receiver and surge drum. By continued cooling of the absorber the pressure is lowered further and evaporation of ammonia commences.

Design and performance of demonstration plant

The demonstration plant was designed for a net

SCHEMATIC REPRESENTATION OF A SOLID ABSORPTION REFRIGERATION SYSTEM (A SMALL DRINKING WATER COOLER)

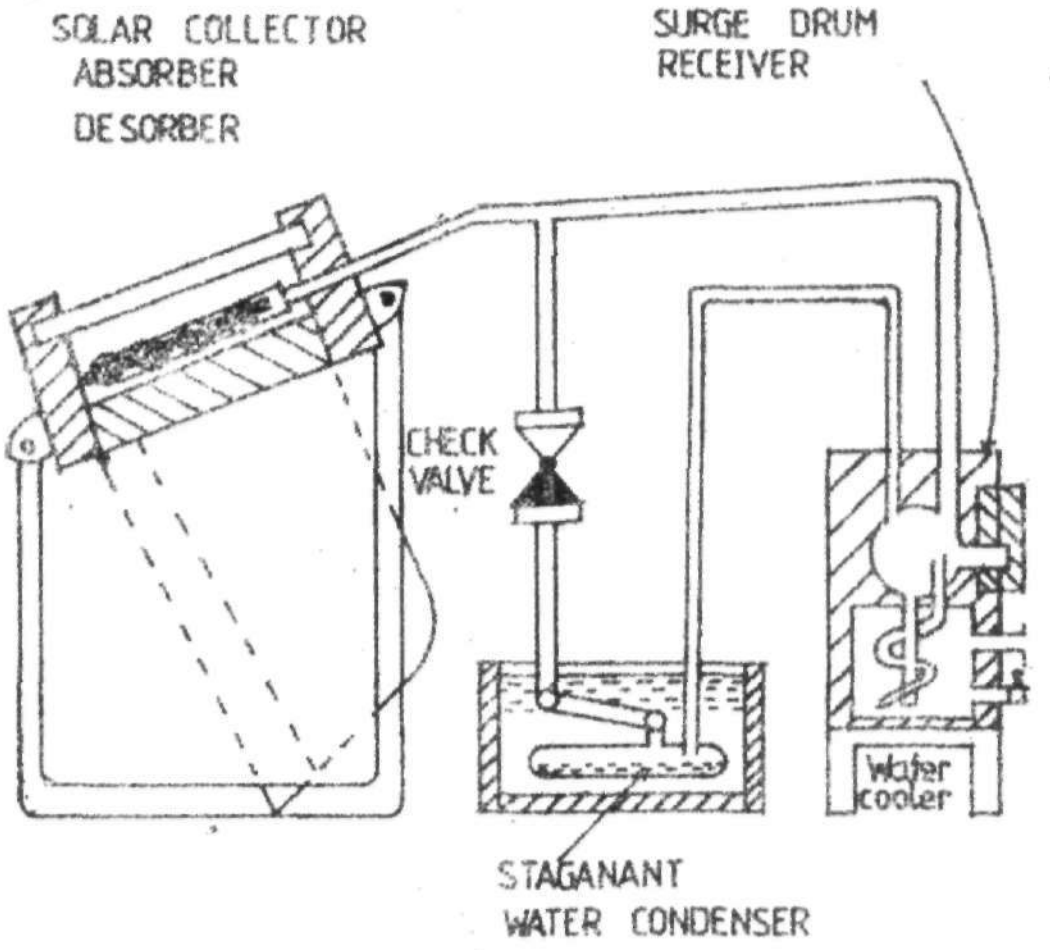
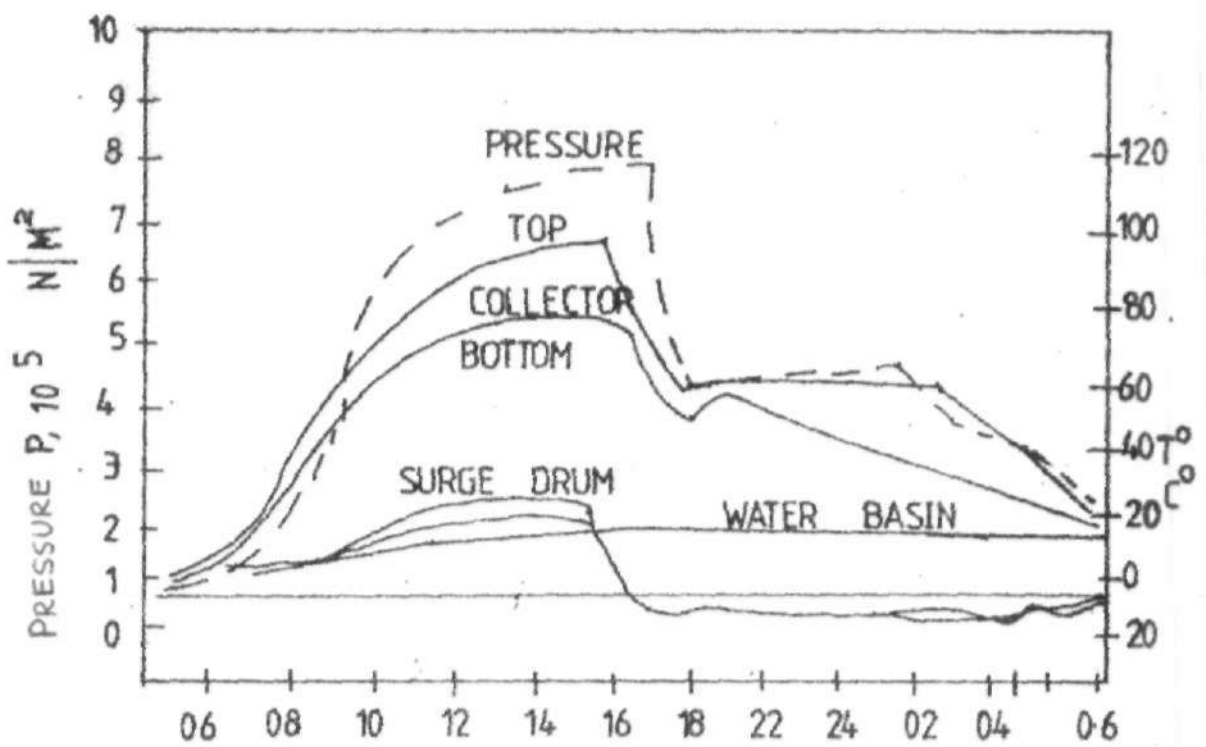


FIG. (2-8) REF. 4

The demonstration plant was provided with thermocouples for recording the temprature of collector plates (ten on each), of the cooling water and of the second receiver and water tank. The pressure in the tank was measured by a pressure transducer. A sight glass on the second receiver permitted the determination of Ammonia generated during the day. In addition the amount of ice frozen during the night could be determined directly by measuring the expansion accompanying the formation of ice. This was done by connecting the water in the tank to a 0.070mm plexiglass tube and observing the risen water level. Climatic data like air temprature, wind speed and direction were needed.

During tests the collectors were raised to an angle of 45° corresponding to maximum solar height in late August. On reasonably good days in August the Ice production was around 20 kg. However on a perfect day with clearsky 7.25 kg of Ammonia was generated & 24.5 kg of Ice was frozen. The tempratures and pressures recorded are shown in figure (2.9) and in Table (2.1) A summary of test results is shown.



Pressure and temperature as recorded during test on september. The back side insulation was closed at 08·30 and open again at 16·00. At 17·20 the insulation was closed in order to prevent Evap. Temp. from dropping below −10 °C.

Temperature at the bottom of surge drum corresponds to Evaporator Temp. (lower curve)

FIG. (2·9) REF 4

SUMMARY OF DATA FROM TEST (TABLE 2.1)

Amount of NH_3 generated	7.25 kg	
Entholpy difference (saturated vaporat-10°C & saturated liquid at 18°C	1169 kJ/kg	
Gross Amount of cooling	7.25x1169 =	8480 KJ
Cooling of surge drum	364 KJ	
Heat Infiltration	226 KJ	
Net amount of cooling	=	7890 KJ
Expansion due to freezing of ice	2.221	
Amount of ice frozen $\frac{2.221}{1/0.917-1}$	=	24.5 kg
Latent heat of fusion	333 KJ/kg	
Net amount of cooling	=	8160 KJ
Discrepancy $\frac{100(8160-7890)}{8160}$	=	3.3%
Mean Insolation (total) 8:30-16:00		762 W/m^2
Total Insolation 7.5x3600x762	=	83,300 KJ
C.O P = $\frac{8,025}{\text{Total } 83,300}$	=	0.096
Thermal Capacity of collector/absorber		100 KJ/°C
(Tfinal - Tinitial) mean = 96 - 27	=	69°C
Average heat of reaction		2300 KJ/kg
Sensible heat 100 x 69	=	6900 KJ

Total heat of reaction	7.25 x 2300	= 16700 KJ
Total amount of heat absorbed		23,600 KJ
(C.O.P) process	= $\frac{8.025}{23600}$	= 0.34
Average collector effeiciency	$\frac{23600}{83300}$	= 0.28

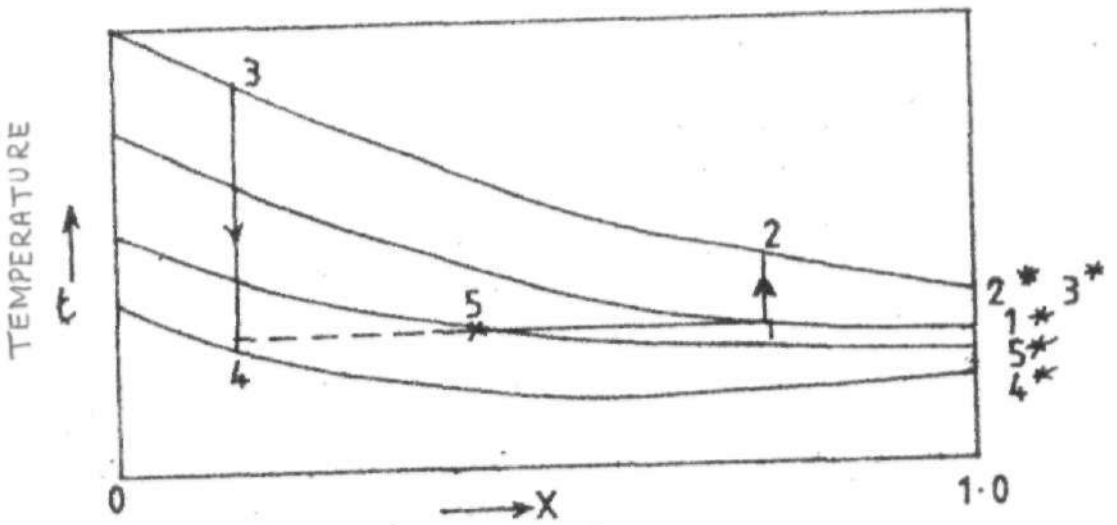
Results obtained from the test

An ice production of 6 kg/m^2 of collector surface was obtained corresponding to an overall C.O.P. of 0.10. This was found to be pretty satisfactory. The fact that ambient temprature is rather low does not invalidate the results because the equilibrium temprature during desorption goes down by the same amount as the condensing temprature so that the collector works under the same conditions at a higher ambient temprature.

Another experiment was conducted by GUPTA & VENKATESH (4) in which the performance of a NH_3-H_2O system was analysed. The intermittent solar refrigerator consisted of a conventional Tubular flat plate collector which served as the generator. The top header of the collector was comparitively large as compared to the bottom header to provide sufficent volume to hold Ammonia water mixture.

The cycle followed is as shown in figure (2.10). Solar radiation striking the generator raises the pressure and temprature as indicated by path 1-2 in figure (2.10) which shows the theoretical cycle At 2 the saturation pressure is equal to the saturation pressure of Ammonia Corresponding to condensing temprature.

Generation of Ammonia continues from 2 to 3 at constant pressure as solar heating is continued. The penerator vapors pass to a water cooled condenser through an air cooled rectifier and are condensed. State 3 the end of the generation process depends on the maximum attainable temprature in the flat plate collector. At this instant when the state of liquid ammonia in the condenser is at 2* 3* and the state of solution in the generator is at 3 the generator and condenser are isolated and the former is cooled to ambient conditions & solution reaches the static corresponding to 4. At 4 the solution concentration, temprature and pressure are low & hence the generator is ready to act as an absorber. The communication between condenser & generator is now restored. A fraction of liquid ammonia in the condenser at 2* 3* flashes



THEORETICAL CYCLE FOR THE SOLAR REFRIGERATOR WORKING ON $NH_3 - H_2O$

FIG (2.10). REF 3

adiabatically into the absorber to change solution state to 5.

As a consequence of flashing the temprature of the remaining quantity of liquid Ammonia in the condenser decreases to bring its state to 5*. The condenser is now ready to operate as an evaporator. From 5* to 1* the evaporator absorbs energy from the surroundings to provide refrigerating effect & the rising vapors are reabsorbed in the absorber unitl solution reaches initial state 1.

It is reasonable to assume in tropics the average temprature of cooling water & the night ambient temprature are around 30°C. Assuming the condensing temprature and the temprature of solution duing refrigeration to be 30°C analysis of the cycle yielded the following results.

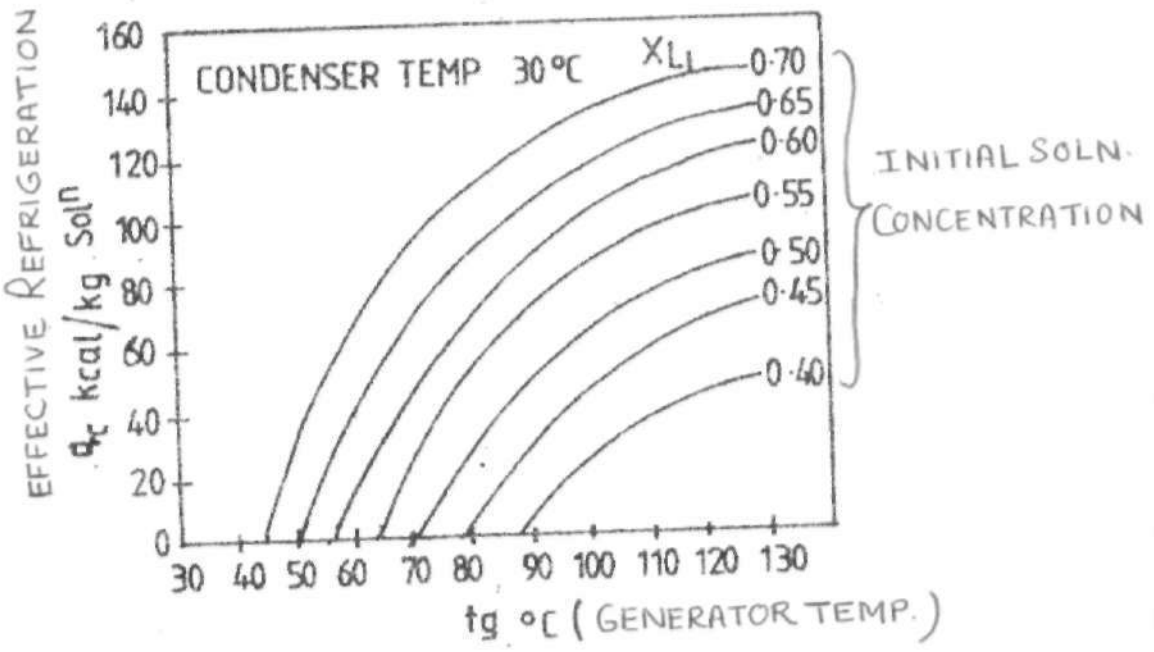
i. Energy absorbed during generation process and the amount of ammonia condensate increases with increase in initial solution concentration and maximum generator temprature

ii. The lowest temprature in evaporator during refrigeration, increases with increase in initial concentraction for a given maximum generator temprature.

iii. For a given initial solution concentration the lowest attainable evaporator temprature decreases with increase in maximum temprature during regeneration.

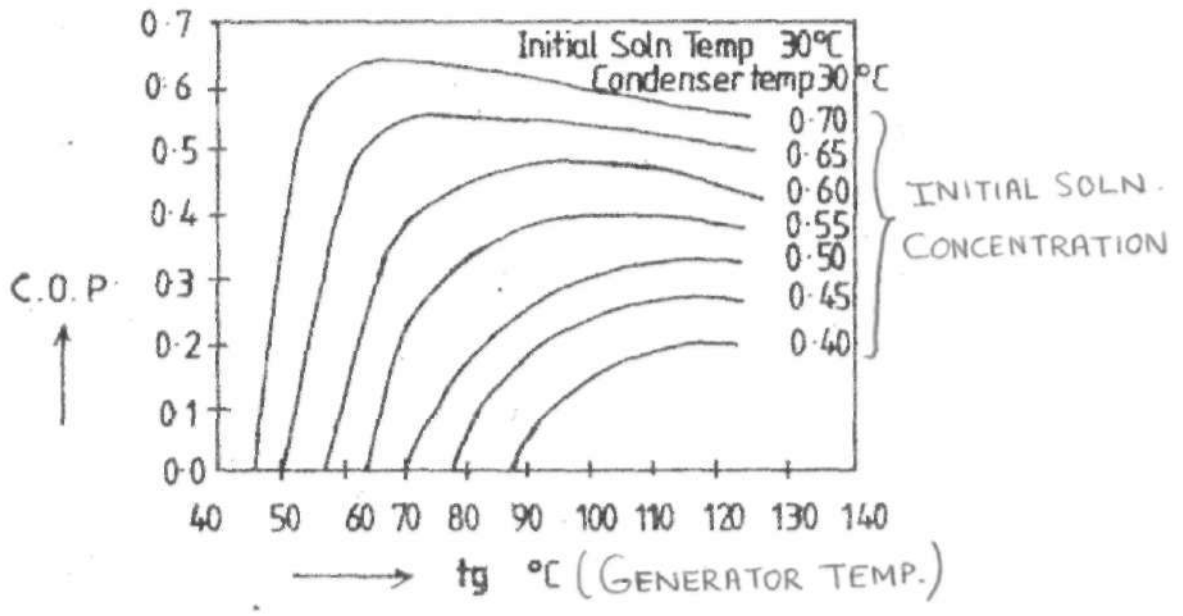
The flat plate collector with an exposed area of 1 m^2 consisted of 0.015m MS pipes spaced 0.1m apart with 22 gange aluminium sheet wrapped over them, .045m diameter lower header, 0.1m diameter upper header, 0.1m glass wool insulation & two glazings The condenser Cum evaporator was a cylindrical vessel of 0.15m diameter and 0.4m height. The evaporator was surrounded by an insulated jacket holding 10 kg of water the mass of condensate in condenser was measured by a level indicator. All connecting lines were of 0.015m MS pipes. Experiments were carried out for varidus concentrations of $NH_3 - H_2O$ solution between 0.48 - 0.65. During refrigeration the temprature of 10kg of water round the evaporator was measured. Maximum temprature attained during generation was 100°C. A higher initial concentration refrigeration continued for longer periods and at low concentrations temprature obtained during refrigeratinn were lower.

Figures (2.11), (2.12), (2.13) show the ratiation of refrigerating effect, the coefficient of Performance and the minimum evaporator temprature respectively as a function of generator temprature for various initial solution concentration The coefficeint of performance for a given initial solution concentration increases rapidly at first and then shows a decreasing trend with increase in generator temprature. Also for a given generator temprature the C.O.P. increases with increase in initial solution concentration. The reason for the above variation is relative change of refrigerating effect with evaporator temprature tends to increase with increase in initial solution concentration for a given generator temprature. As initial concentration increases the quantity of condensate also increases. Thus the amount of liquid ammonia available for effective refrigeration increases. As the adiabatic flashing of ammonia has to bring down the internal energy of a larger quantity of Ammonia the temprature obtained after flashing is high for high initial solution concentration. At a fixed initial solution concentration the minimum evaporator temprature decreases with increase in generator temprature.



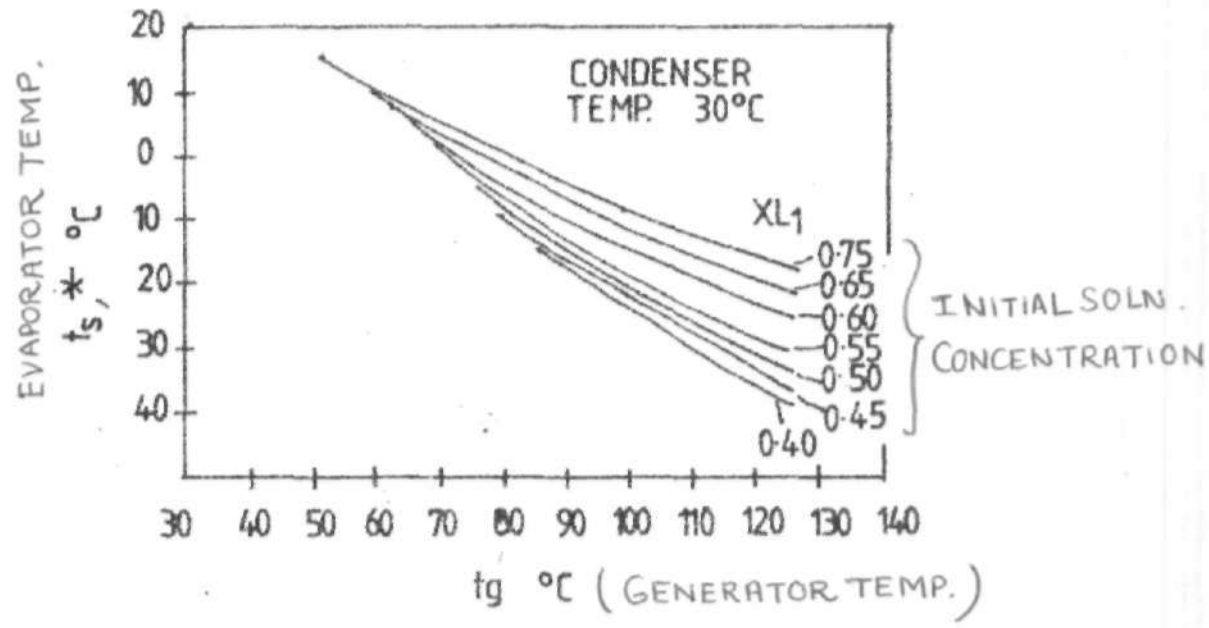
EFFECT OF GENERATION TEMPRATURE AND INITIAL SOLUTION CONCENTRATION ON EFFECTIVE REFRIGERATION

FIG.(2.11). REF 3.



EFFECT OF GENERATOR TEMPRATURE AND INITIAL SOLUTION CONCENTRATION ON COEFFICIENT OF PERFORMANCE.

FIG. (2.12). REF 3



EFFECT OF GENERATOR TEMPRATURE AND INITIAL SOLUTION CONCENTRATION ON MINIMUM EVAPORATOR TEMPRATURE .

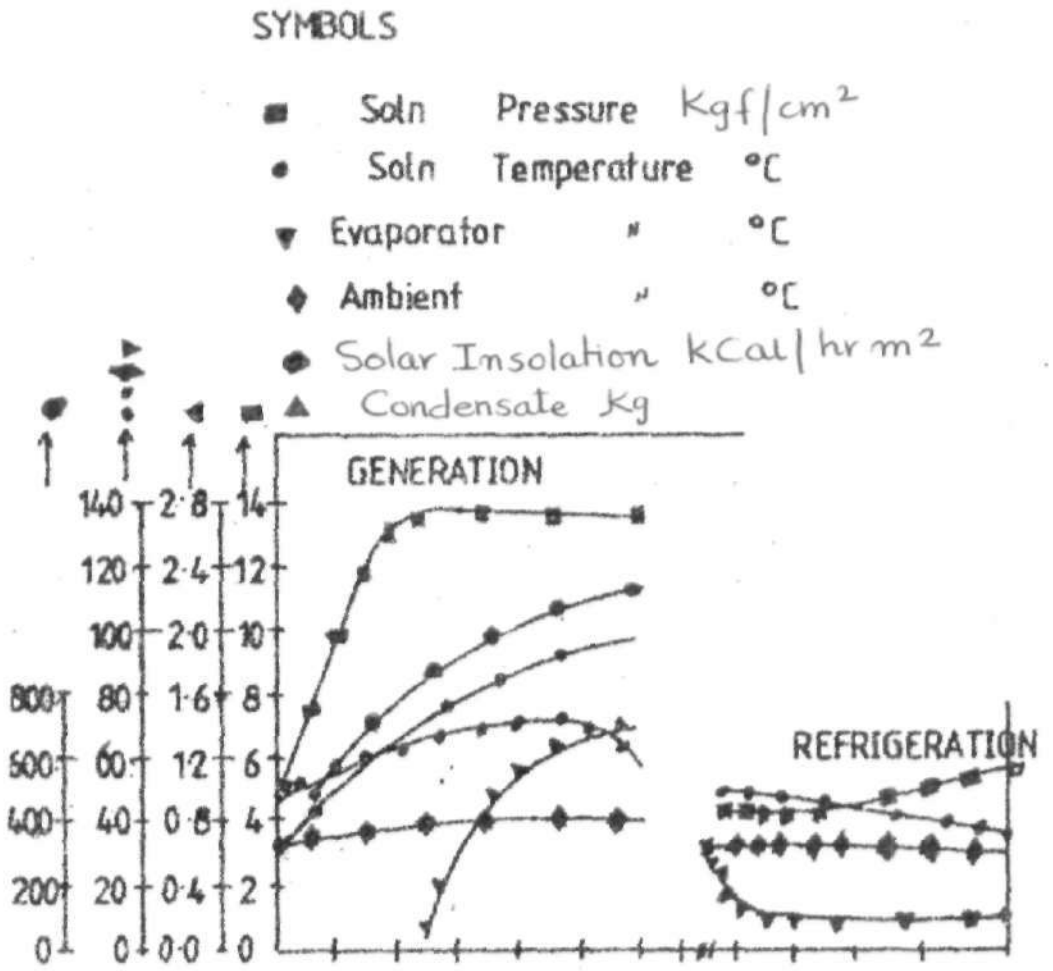
FIG. (2·13). REF 3.

As the generator temprature increases the pressure difference (P_2*_ P_4) responsible for adiabatic flashing of liquid ammonia increases causing more Ammonia to flash bringing evaporator tempratures to lower values.

Experiments were conducted with an initial solution concentration of 0.55 and results obtained are given in Figure (2.14). The initial mass of solution in the collector was 5.33 kg. The maximum generation temprature was 102°C and the amount of Ammonia Condensed was 1.54 kg. During refrigeration the temprature of 10 kg of water (initially at 30°C) around the evaporator was measured. The refrigeration continued for nearly 7 hours and the minimum temprature recorded was 7°C. The evaporator temprature was higher than the theoretically predicted valuve due to:

(i) During genration the system pressure was higher than the theoretical pressure which reduced the amount of Ammonia condensed.

(ii) During adiabatic flashing the solution temprature increased thereby increasing the system pressure that reduced the amount of Ammonia flashed.



PERFORMANCE CHARACTERISTICS OF SOLAR REFRIGERATOR.(INITIAL CONCENTRATION = 0·55 AND CONDENSING TEMPRATURE = 30°C)

FIG.(2.14). REF 3

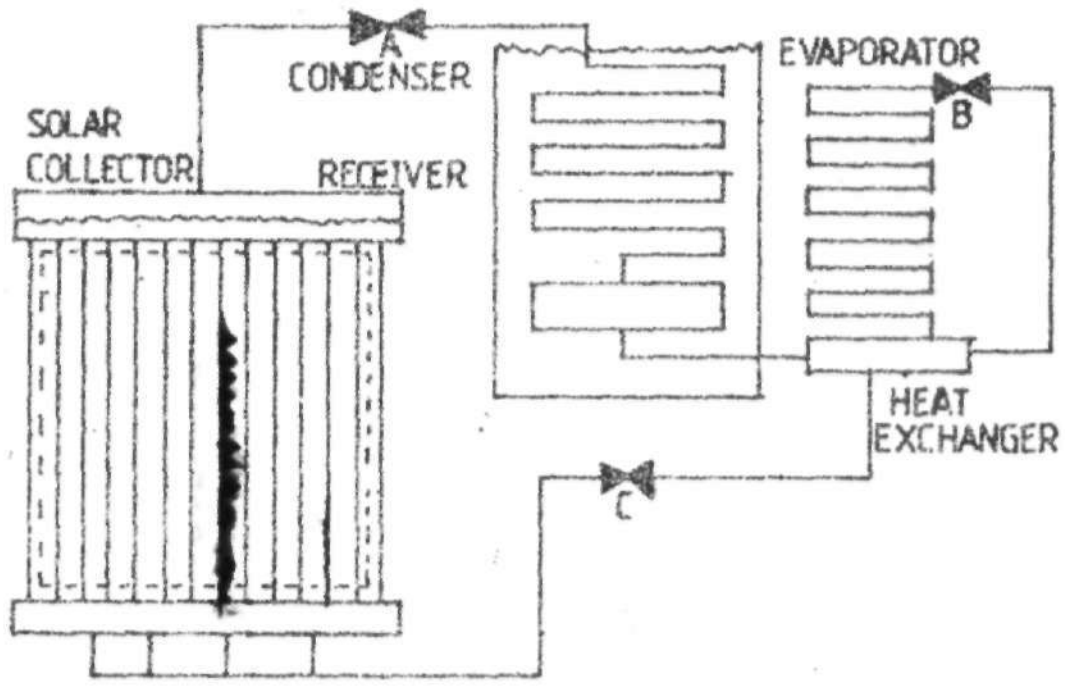
(iii) Although the water around the evaporator was surrounded by insulated jacket heat losses would still be there. More experiments[5] were conducted on a proto type solar powered refrigerator for Remote area for food preservation. The refrigerator was of intermittent nature and worked on $NH_3 - H_2O$ Combination. The new features of this system were

(i) dissipation of heat of absorption from flat plate collectors.

(ii) The addition of an expansion valve with a dry evaporator coil & heat exchanger.

(iii) The use of a mirror to enhance solar heating the system used is shown in Figure (2.15).

The system was constructed of welded steel pipe. The collector had fourteen 0.025m diameter risers 0.100m apart connecting a 0.050m feeder to a 0.100m header. The outermost risers were thermally insulated to provide return lines for circulation of thermosyphoning. copper sheets connecting the remaining risers and painted dull black for a collecting area of $1.44m^2$ which had one glass Cover and was tilted at 14^o to the horizontal facing south. A plane mirror was facing east on the western edge of the collector during the



FLOW DIAGRAM FOR AN INTERMITTENT
NH_3-H_2O ABSORPTION REFRIGERATOR FOR REMOTE AREAS

FIG 2·15 REF 5

morning and was transferred to the eastern side in the afternoon. It could be held fixed 90° - 100° to the collector or adjusted hourly to maximise its effect. The collector contained 18.7 kg of Ammonia water solution having 46% Ammonia by weight, a concentration chosen to give a refrigerating effect of 265°K with absorption at 303°K. The header was half filled with liquid whose level could be observed through glass windows. During the day valve A was opened, and B and C were closed. Ammonia vaporised by solar heating of the solution condensed in a condenser 5.2m long and was stored in a receiver immersed ina tank of water which was kept cool by exposure at night and by thermal In sulation in the day. At dusk valve A was closed and glass Cover of collector was removed. When the solution had cooled valves B and C were opened to produce refrigeration in the evaporator coil by evaporation of Ammonia which was reabsorbed into the solution through 4 inlets into the collector feeder. The heat of absorption escaped from the collector surface. The evaporator coil made of 0.013m diameter steel pipe 5.2m long was inmersed in a flat steel box 0.381mx0.413m x0.620m containing the water to be frozen.

Two steel food containers each of volume 250 x $10^{-6}m^3$ were attached to the ice box, one on each side. The counter flow double pipe heat exchanger was 0.38m long. Polystyrene foam 0.10m thick was used to insulate the assembly.

The results obtained in the above experiments were as follows : On a typical bright day in April the solar radiation falling directly on the collector was 28.2 MJ but was increased by a factor 1.36 using the mirror adjusted everyhour. The ambient temprature reached 310°K while the tank cooling water remained close to 301°K. The temprature and pressure of Ammonia water solution rose from 304°K & 310 Kpa to 373°K and 1210 Kpa in the afternoon and 3.16 kg of Ammonia were distilled since heat of generation calculated from standard data was 10.8 MJ the collector efficiency was 0.28.

During refrigeration which lasted 5 hours at night the absorption pressure increased from 170 Kpa - 340 Kpa giving evaporator temprature from 251°K to 269°K. Starting with 7.5 kg of water in ice box at 302°K 5.18 kg of ice were produced and overall C.O.P. was 0.0726. It was estimated that heat exchange increased the refrigeration effect by

3.4%. In the best run 3.47kg of Ammonia were distilled and 6.4 kg of Ice were produced. In further experiments 3.46 kg of fruit in food containers initially at 300°K were cooled to 283°K by dawn and were held below 288°K throughout the day. Williams et al[6] described a small refrigerator is which a parabolic solar reflector 1.2m in diameter focussed the sunlight onto a steel vessel containing Ammonia and water in a closed system. The ammonia was driven out with heat and condensed in a small vessel connected by a steel tube or a flexible rubber hose capable of withstanding high pressure. The pressure of heated Ammonia reached over 10.1 bar 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 11.3 kg was taken into the house and the small vessel of liquid Ammonia was put into an insulated box with a capacity of $0.062m^3$, When removed from focussed sunlight the water solution cooled and reabsorbed the Ammonia The evaporation of liquid Ammonia kept the insulated box cold. An exposure of 4 hours in the bright sun kept the refrigerator below 278°K during the 24 hour day.

The absorptivity α of the blackened receiver holding the Ammonia and water was about 0.95 and the reflectivity of the aluminized Mylar that lined the parabolic focussing collector was 0.7. The fraction $\alpha\tau$ of solar radiation striking the collector that was absorbed by the metal receiver was determined calorimetrically. This value was about 0.67. In an Ammonia water system with typical solar heating of 5591.5 KJ optical losses were 33 percent and heat losses into the metal container were about 3 percent. About 40 percent of the total solar radiation intercepted by the collector was picked up by the Ammonia water solution and 24 percent of this heat was lost to the surrounding by radiation and convection. The cooling ratio ranged upto 0.4 and solar heating ratio ϵ (heat absorbed by Ammonia and water in the generator divided by the direct radiation striking the solar collector) was also about 0.4. The overall performances E equal to $\eta\epsilon$ was 0.16. The solution contained 1.6 kg of Ammonia in 2 kg of water and maximum heating temprature was 402°K A special valve was developed for separting the water from liquefied Ammonia.

Trombe and Foex produced 6 kg of Ice per day is a solar refrigerator using a cylindroparabolic mirror of 1.5m^2 area, oriented east west and working on intermittent Ammonia cycle. The ammonia water were heated in a Central tube at the focus of the cylindro-parebolic mirror during the day and the Ammonia that was driven out and liquefied was allowed to be vaporized during the night.

Chinnapa described on intermittent Ammonia water refrigerator in which small quantities of ice were made with a flat plate collector containing 3 glass plates 1.52m x 1.60m.. The C.O.P. was about 0.06 operating in a hot climate. The maximum temprature of regeneration with the flat plate collector was 373^oK and temprature of absorbing solution was 306 - 310^oK.

The water ammonia system had the disadvantage that some of the water vaporized along with Ammonia and it was desired that at intervals water must be returned to the larger vessel containing Ammonia water solution. Hence it was desirable to eliminate the water which could be done by using very concentrated solutions of salts in liquid ammonia. The vapor pressure of Ammonia could be reduced

considerably at room temprature as the salts have negligible vapor pressure even at high tempratures. Extensive investigations were carried out by Blytas and Daniel's to find out the salt with best. Physio-chemical properties for solar cooling with Ammonia. Ammonium Nitrate lithium Nitrate, the alkali halides and Sodium thiocyanate were among the salts investigated theoretically and experimentally.

The nitrates dissolved in Ammonia have many desirable properties but thermodynamically there is a chance that a spontaneous reaction will produce nitrogen gas and steam with explosive violence.

Ammonia salts are highly soluble but in solution they are corrosive to Iron containers, although they can be used in Aluminium containers. Sodium thiocyanate was ultimately chosen as the best salt to dissolve in Ammonia for use in solar cooling and refrigeration. The solution had suitable thermodynamic properties.

The sodium thiocyanate - Ammonia system was tested in the laboratory for solar refrigeration by R. Chung.

A. 9.1 kg steel cylinder containing 5.4 kg of solution with about equal weghts of sodium thiocyanate and ammonia was

placed at the focus of a 1.2m parabolic solar cooker . The cylinder was connected by a flexible pressure hose to a small vessel immersed in water at room temprature for the liquefaction of Ammonia. After about 4 hours much of the Ammonia had been driven out and the cylinder was removed and brought to room tempr:ture the vessel of liquid ammonia was placed overnight in water contained in an insulated box and 4.1 kg of Ice were produced.

2.1 Justification of present project and approach

So far solar cooling using NH_3 - H_2O combination has been tried in Samaru. The NH_3 - NaSCN combination has so far not been tried, and this has inspired me to make an analysis of the system working on this combination.

A feasibility analysis of the NH_3 - NaSCN system has been reviewed & the Degassing Width, Charge, required per kg of Ammonia vapor, and Coefficient of performance at various Absorber, Generator, and Condenser temperature Combinations has been found out. A temperature Combination, keeping in mind, the climatic conditions of Samaru, and reasonable charge required for the system has been assumed for the design of various components.

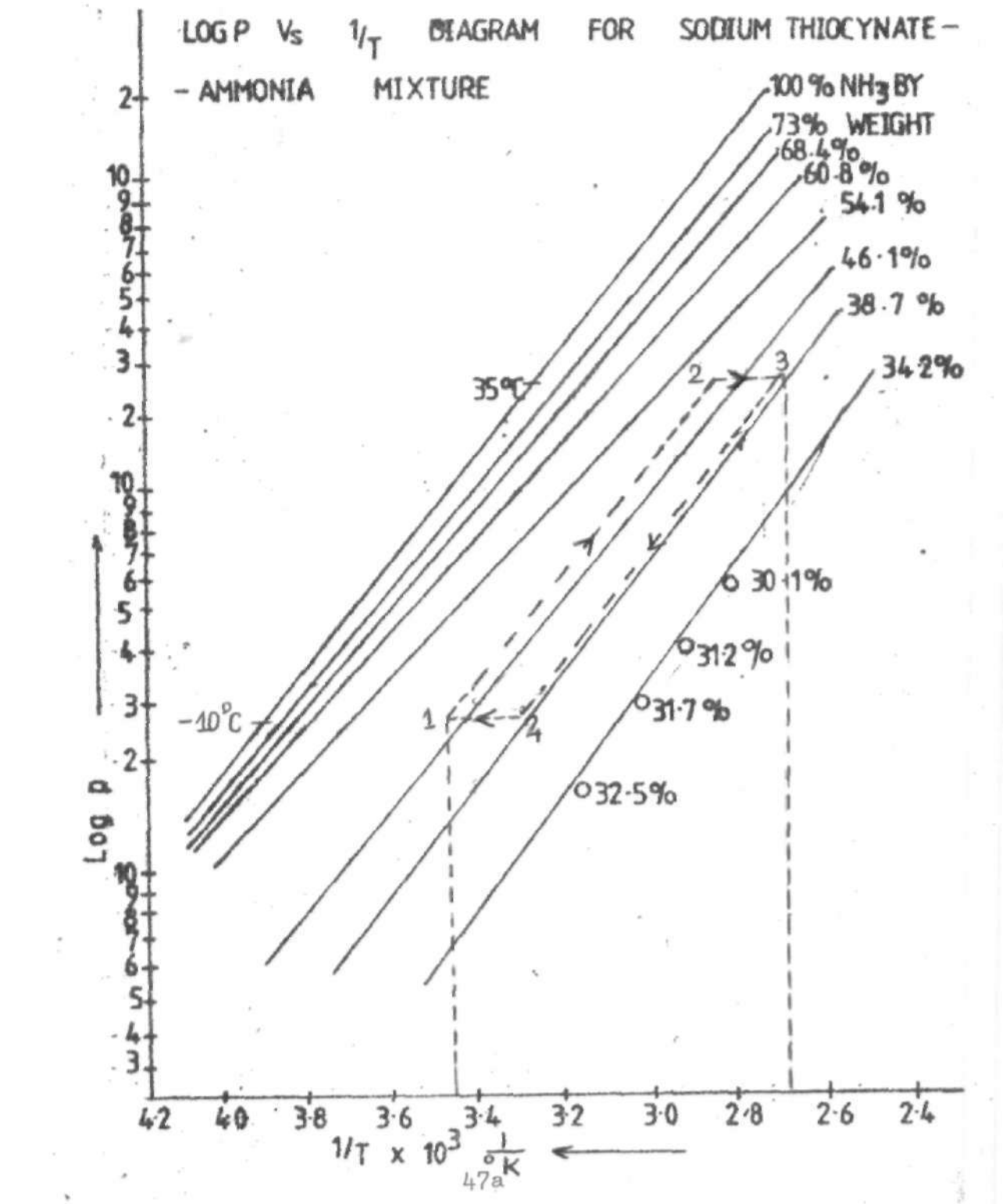
CHAPTER 3

FEASIBILITY ANALYSIS OF AMMONIA - SODIUM THIOCYANATE INTERMITTENT SYSTEM

3.1 Introduction

Feasibility analysis is just a theoretical analysis of the system to predict its performance at different working conditions and to select those conditions which will give a best performance in a particular region. For this particular system feasibility analysis has been done at various temperature combinations of generator absorber and condenser using log P Vs $\frac{1}{T}$ diagram for the intermittent cycle.

Fig (3.1 gives us the log P vs $\frac{1}{T}$ (i.e. Pressure vs temperature) diagram for the Ammonia - Sodium Thiocyanate system at different solution concentrations.(7) The temperatures have been plotted on the abcissa and the pressures on the ordinate. The inclined lines give us the various solution concentrations. The cycle followed is as shown in Figure (3.1). Experimental results have shown that a concentration of 49% Ammonia will yield the best generation of refrigerant vapors. Point 1 in the figure is the strong saturated solution state in the generator, The generator is initially at this condition. Assuming a solution concentration of 49% Ammonia and an initial generator temperature, point 1 is obtained.



47a

FIG. (3·1) REF 7.

Heat is added in the generator from 1 - 2 till vapor pressure developed in the generator equals the condenser saturation pressure corressponding to condenser temperature. This in fact is the operating pressure for the system. Generation of Ammonia continues from 2 - 3 at constmnt pressure as solar heating is continued. State 3 the end of the generation process corresponds to the maximum attainable temperature in the generator.

During the generation period 2 - 3, the generator vapors pass to a water cooled condenser, are condensed and stored in the receiver. At 3 the state of liquid Ammonia in the condenser is different from the state of solution in the generator. At this instant the generator and condenser are isolated and former is cooled down to Ambient conditions. Corresponding to point 4. At 4 the solution concentration, pressure and temperature are low enough and hence the generator is ready to act as an absorber. The communication between the generator and condenser is now restorted. A fraction of liquid Ammonia in the condenser flashes into the absorber and changes the solution state. Evaporation starts, the evaporator absorbs energy from the surrounding to provide refrigerating effect during process 4 - 1 and the rising vapors are reabsorbed in the absorber until solution reaches initial state 1.

The cycle followed refers to the flow diagram shown on the back page.

The points obtained refer to the following temperature combination.

Maximum Generator temperature	=	$100^{\circ}C$
Condenser temperature	=	$35^{\circ}C$
Evaporator temperature	=	$-10^{\circ}C$
Absorber temperature	=	$20^{\circ}C$

From log P vs $\frac{1}{T}$ diagram, concentration, pressure and temperature at different stages is known to us. With the help of those feasibility Analysis has been done as follows:

3.2 Degassing Width

Degassing width is the difference between rich and poor solution concentration in generator at given temperature conditions. Degassing width at different temperature combinations has been calculated and its variation with

condenser temperature for various generator and absorber temperature is shown in figures (3.2), (3.3) & (3.4). The same results are also given in Table 3.1. The calculation for one temperature combination is shown below:-

Generator temperature = 100°C

Absorber temperature = 20°C

Condenser temperature = 35°C

Evaporator temperature = -10°C

Saturation Pressure Corresponding to condenser temperature (35°C)

= 135 N/cm² = 13.5 bar

Saturation pressure corresponding to evaporator temperature (-10°C) = 29.1 N./cm² = 2.91 bar

From Log P vs 1/T diagram for sodium Thiocyanate system we have Rich solution concentration (ξ_r) = 0.491

Poor solution concentration (ξ_p) = 0.4094

Degassing width = 0.491 - 0.4094 = 0.0816

3.3 Rich Solution Circulation

Rich solution circulation is the amount of solution to be charged for each kilogramme of ammonia distilled from the generator . The variation of charge required for each kilogramme of Ammonia distilled from generator, with

VARIATION OF DEGASSING WIDTH WITH CONDENSER AND GENERATOR TEMPERATURE

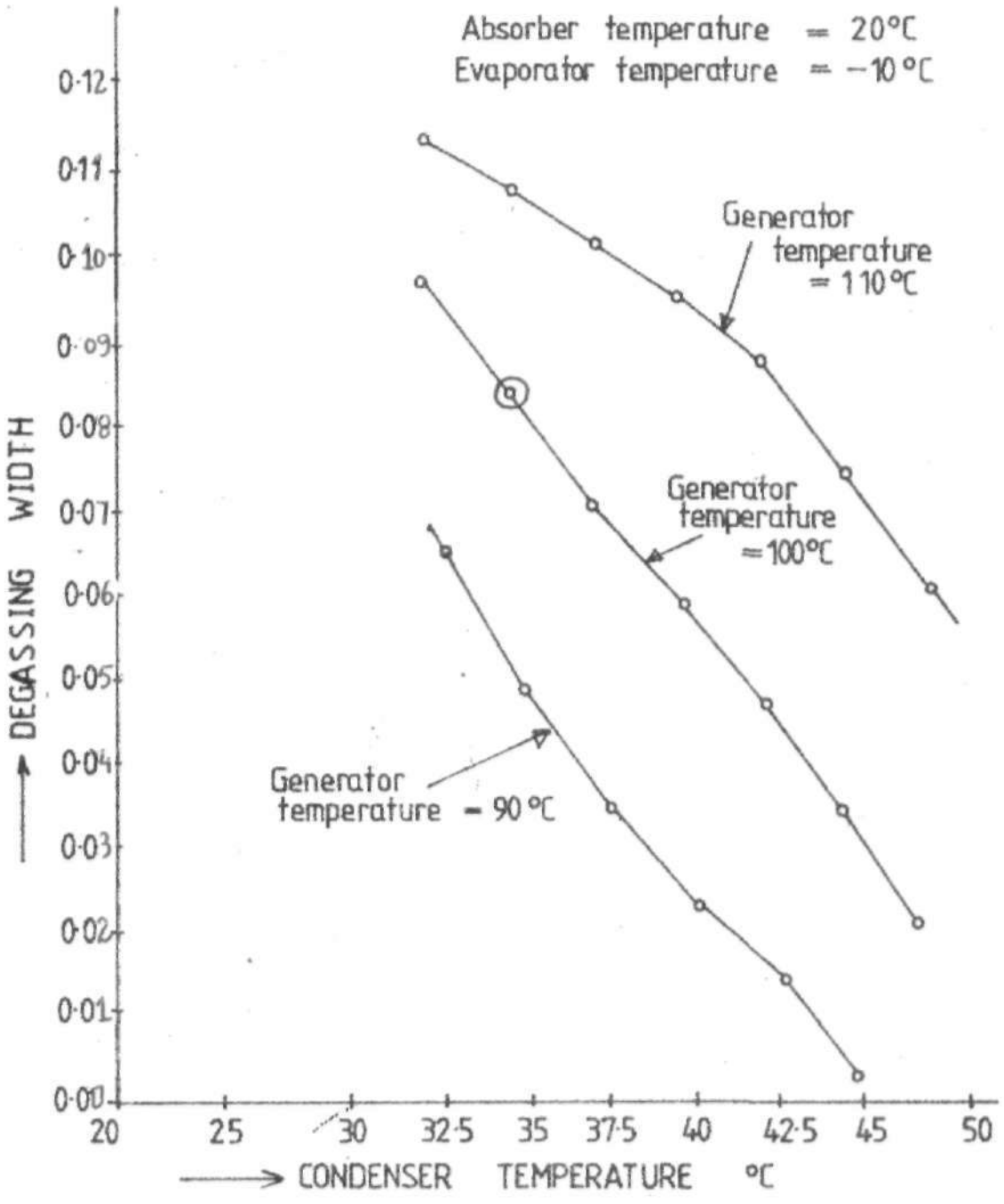


FIG (3·2)

VARIATION OF DEGASSING WIDTH WITH CONDENSER AND GENERATOR TEMPERATURE

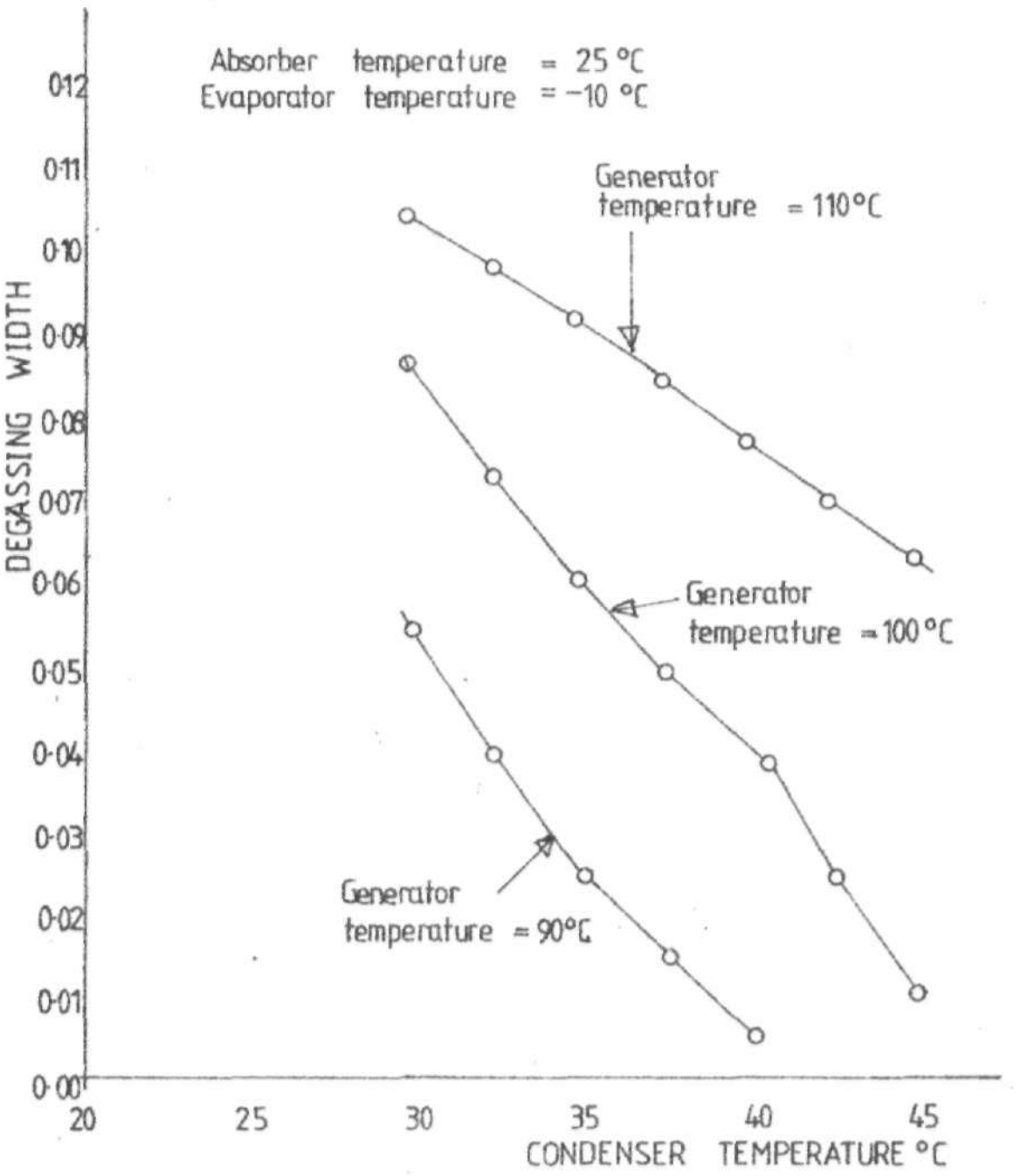


FIG (3·3)

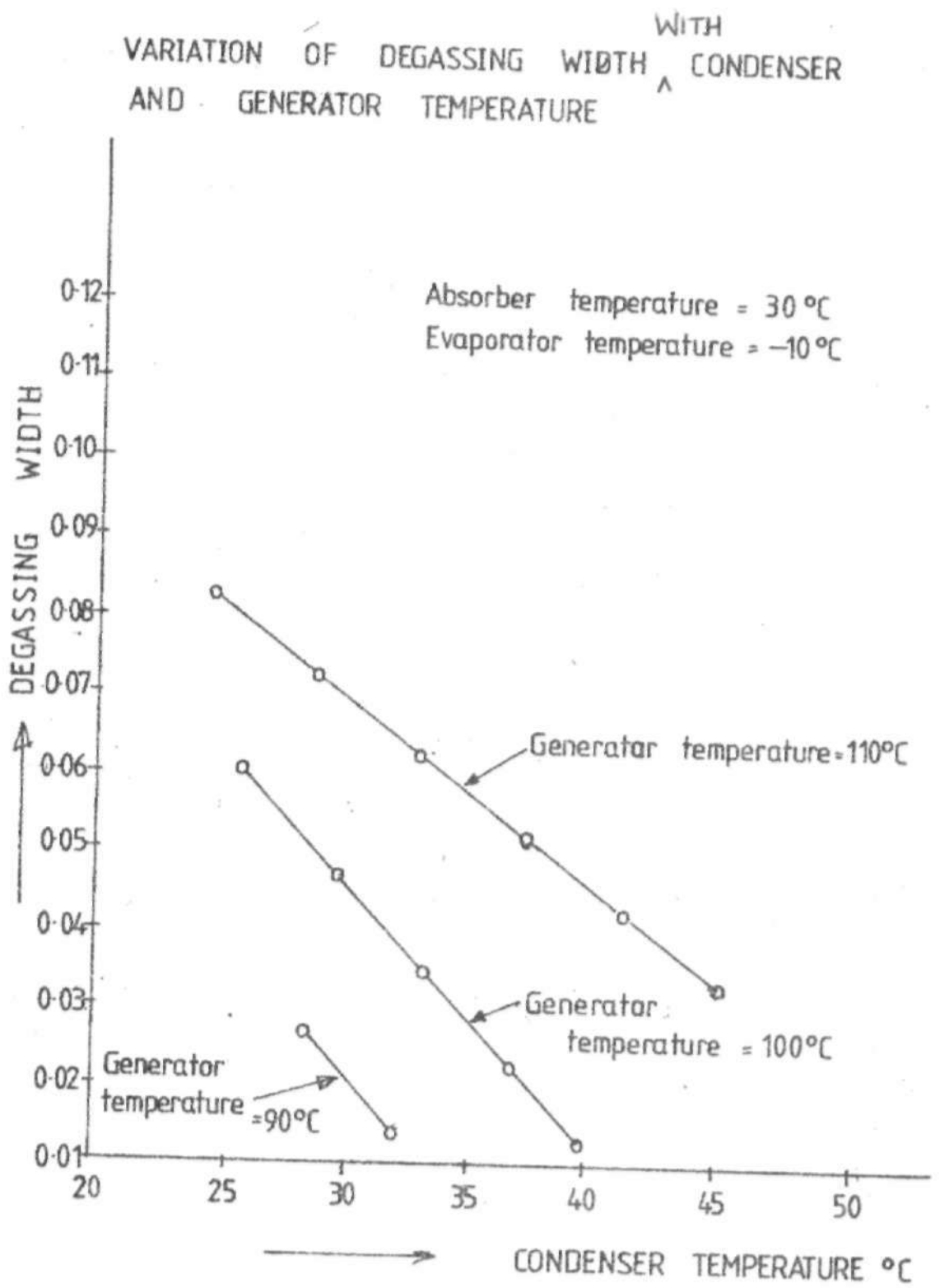

VARIATION OF DEGASSING WIDTH WITH CONDENSER AND GENERATOR TEMPERATURE
Absorber temperature = 30 °C
Evaporator temperature = –10 °C
DEGASSING WIDTH
0·12
0·11
0·10
0·09
0·08
0·07
0·06
0·05
0·04
0·03
0·02
0·01
Generator temperature = 110°C
Generator temperature = 100°C
Generator temperature = 90°C
20
25
30
35
40
45
50
CONDENSER TEMPERATURE °C

FIG (3-4)

DEGASSING WIDTH AT DIFFERENT GENERATOR ABSORBER AND CONDENSER TEMPRATURE COMBINATIONS. EVAPORATOR TEMPRATURE = 10°C

Condenser Temprature °C	Generator Temprature	ABSORBER TEMPRATURE = 20°C			ABSORBER TEMPRATURE = 25°C			ABSORBER TEMPRATURE = 30°C		
		90°C	100°C	110°C	90°C	100°C	110°C	90°C	100°C	110°C
30		0.0755	0.1080	0.1255	0.0545	0.0870	0.1055	0.0283	0.0608	0.0793
32.5		0.0598	0.0948	0.1192	0.0397	0.0738	0.0987	0.0135	0.0476	0.0725
35		0.0456	0.0816	0.113	0.0249	0.0606	0.920	-0.0013	0.0344	0.0658
37.5		0.0340	0.070	0.1055	0.0149	0.0494	0.0848	-0.0112	0.0232	0.0586
40		0.0260	0.0593	0.0987	0.0050	0.0382	0.0777	-0.0212	0.0121	0.0515
425		-0.0129	0.0457	0.0845	-0.0081	0.0241	0.0703	-0.0343	0.0020	0.0441
45		-0.0002	0.031	0.0702	-0.0212	0.0100	0.0630	-0.0474	-0.0162	0.0368

TABLE (3.1)

COMMENT:- The temperature combinations in which degassing is negative are not feasible.

condenser temprature for various absorber and generator tempratures is shown in figures 3.5, 3.6 and 3.7. A sample calculation for the same temprature combination as for degassing width is shown below:-

$$\text{Charge required (f)} = \frac{f_d - f_p}{f_r - f_p} \qquad \text{(3.1)}$$

$$= \frac{1.0 - 0.4094}{0.491 - 0.4094}$$

$$= \frac{0.5906}{0.0816}$$

= 7.25 kg/kg of Ammonia Vapor.

3.4 Coefficient of Performance

Co-efficient of performance is the ratio of refrigeration effect produced from unit amount of the refrigerant vapor to generator heat required for the generation of same amount of refrigerant vapor.

Generator heat required is the sum of sensible heat, required to bring the charge from night temprature to the generator temprature at which regrigerant will start vaporizing and latent heat of ammonia at generator temprature.

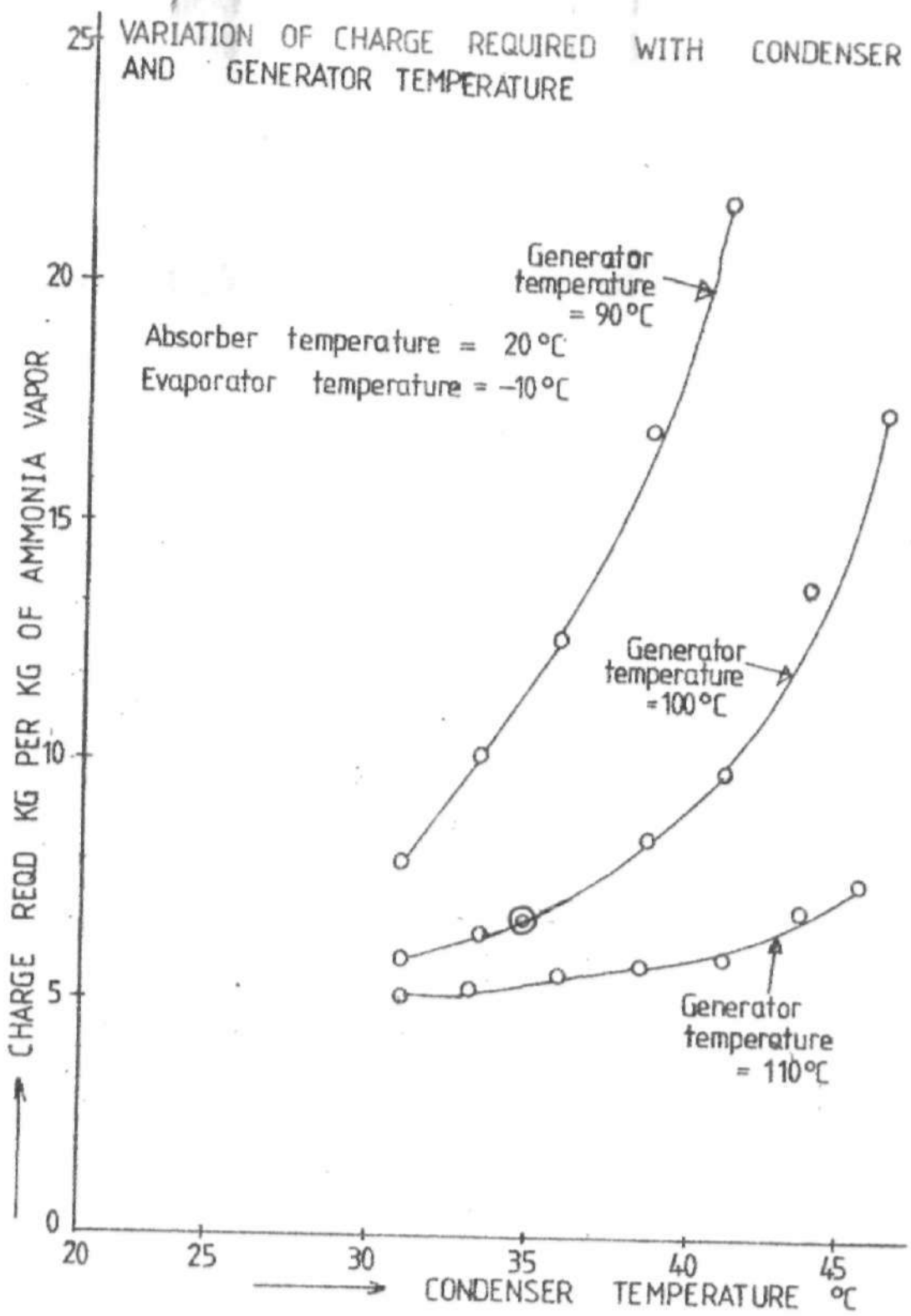

VARIATION OF CHARGE REQUIRED WITH CONDENSER AND GENERATOR TEMPERATURE
CHARGE REQD KG PER KG OF AMMONIA VAPOR
0
5
10
15
20
25
Generator temperature = 90 °C
Absorber temperature = 20 °C
Evaporator temperature = −10 °C
Generator temperature = 100 °C
Generator temperature = 110 °C
20
25
30
35
40
45
CONDENSER TEMPERATURE °C

FIG. (3·5)

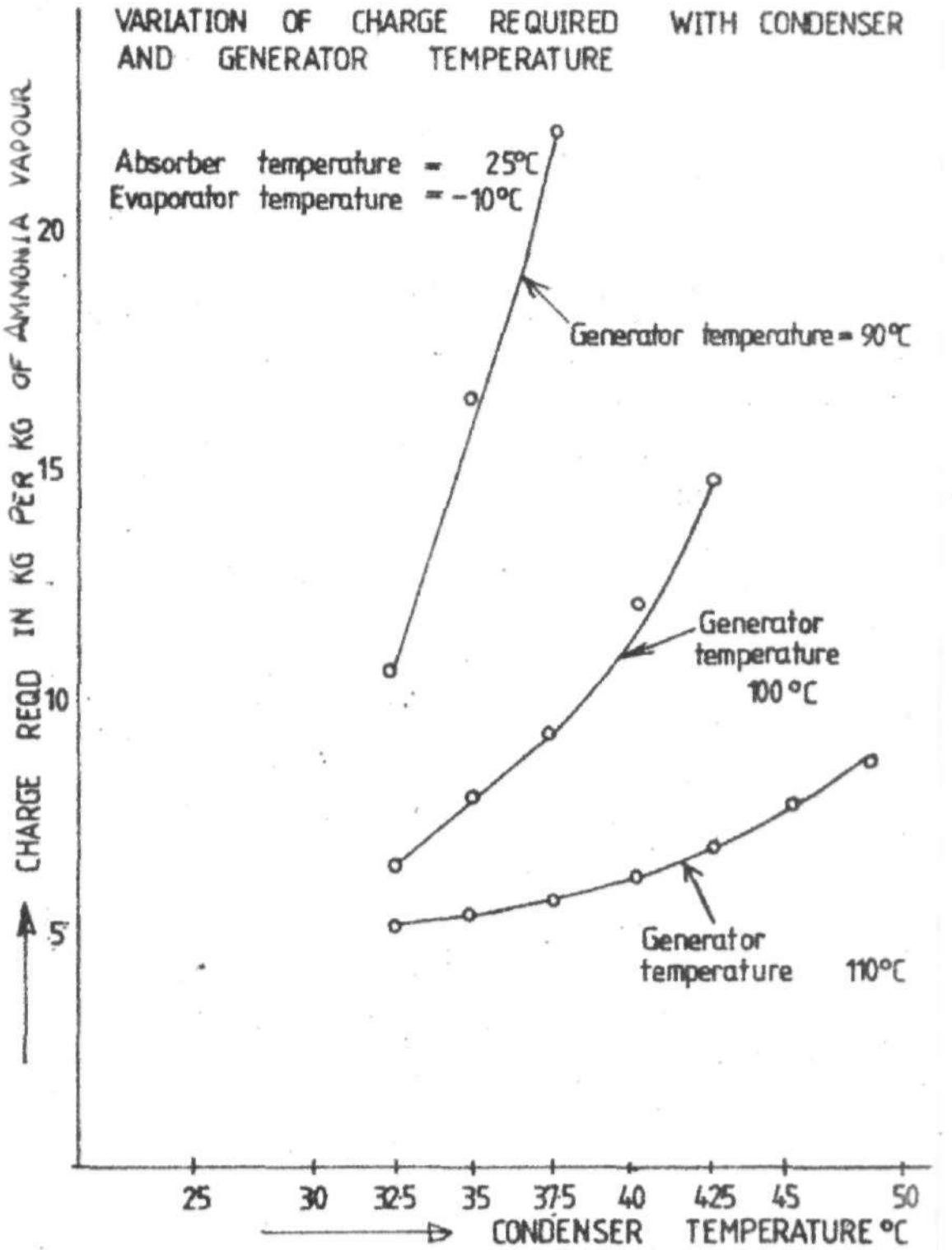

VARIATION OF CHARGE REQUIRED WITH CONDENSER AND GENERATOR TEMPERATURE
Absorber temperature = 25°C
Evaporator temperature = -10°C
Generator temperature = 90°C
Generator temperature 100°C
Generator temperature 110°C
CHARGE REQD IN KG PER KG OF AMMONIA VAPOUR
20
15
10
5
25
30
32·5
35
37·5
40
42·5
45
50
CONDENSER TEMPERATURE °C

FIG (3·6)

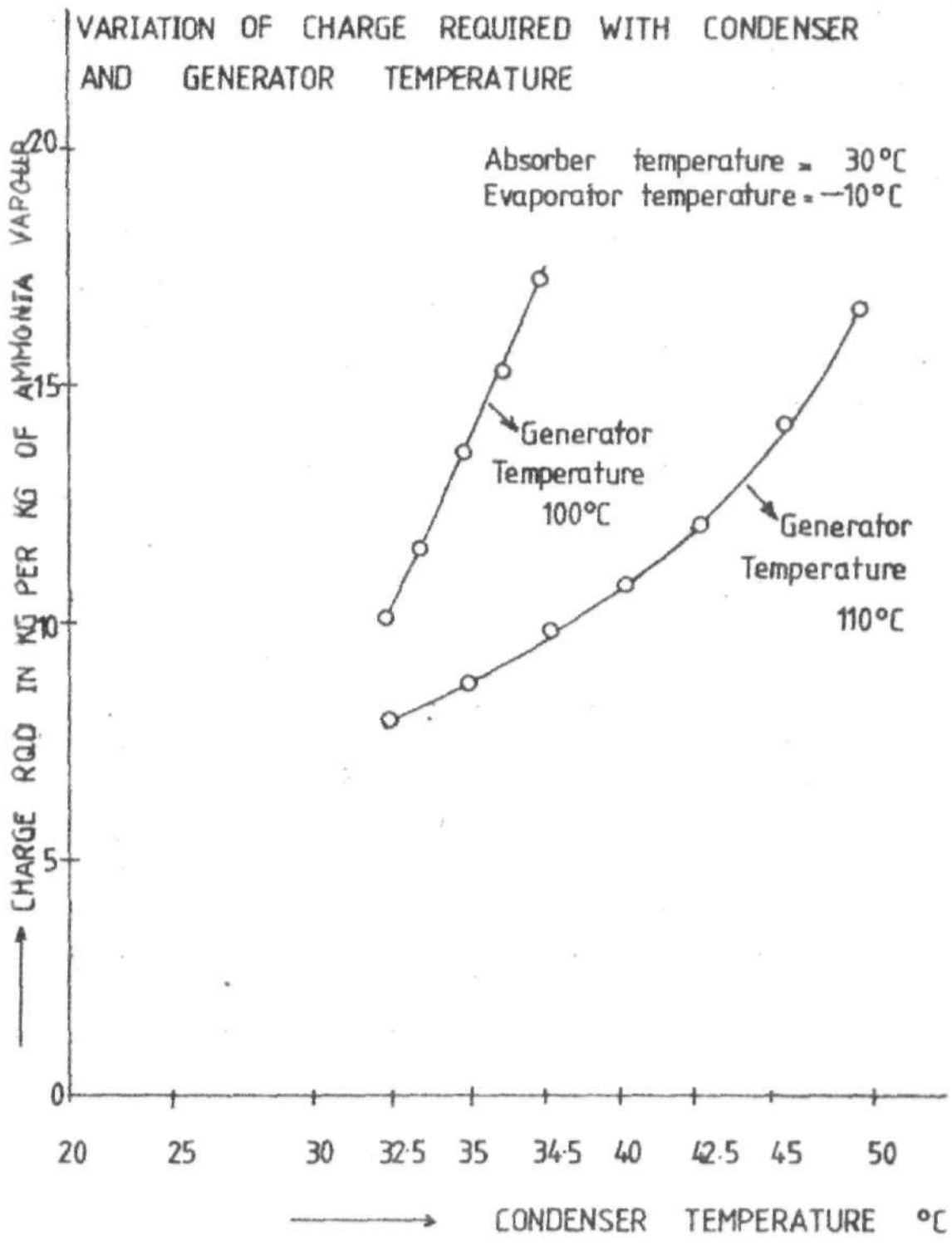

VARIATION OF CHARGE REQUIRED WITH CONDENSER AND GENERATOR TEMPERATURE
Absorber temperature = 30°C
Evaporator temperature = −10°C
Generator Temperature 100°C
Generator Temperature 110°C
CHARGE REQD IN KG PER KG OF AMMONIA VAPOUR
0
5
10
15
20
20
25
30
32·5
35
34·5
40
42·5
45
50
CONDENSER TEMPERATURE °C

FIG (3·7)

Generator heat = Sensible heat of the mixture

+ Latent heat of Ammonia per kg of vapor

at generator temprature.

$$= m_1 C_{pg} (T_g - T_a) + L_{tg} \quad \text{——————} (3.2)$$

Refrigeration effect produced is the sum of latent heat of ammonia at evaporator temperature minus the loss of cooling to bring liquid ammonia from condenser temperature to evaporator temprature.

$$\text{Refrigeration effect} = L_{te} - C_{pa}(t_c - t_e) \quad \text{——————} (3.3)$$

Coefficient of performance at different absorber, generator and condenser temperature combinations has been calculated and its variation with condenser temperature for various generator and absorber temperatures is shown in the figures (3.8, 3.9 & 3.10). The same results are also shown in Table (3.2).A sample calculation for the same temperature combination as for degassing width is shown below.

Generator heat = $m_1 C_{pg} (T_g - T_a) + L_{t_g}$

= 7.25 x 0.68 (100-20) + 267.5

= 394 + 267.5

= 661.5 K Cal/kg of NH_3 vapor.

VARIATION OF COEFFICIENT PERFORMANCE WITH CONDENSER TEMPERATURE

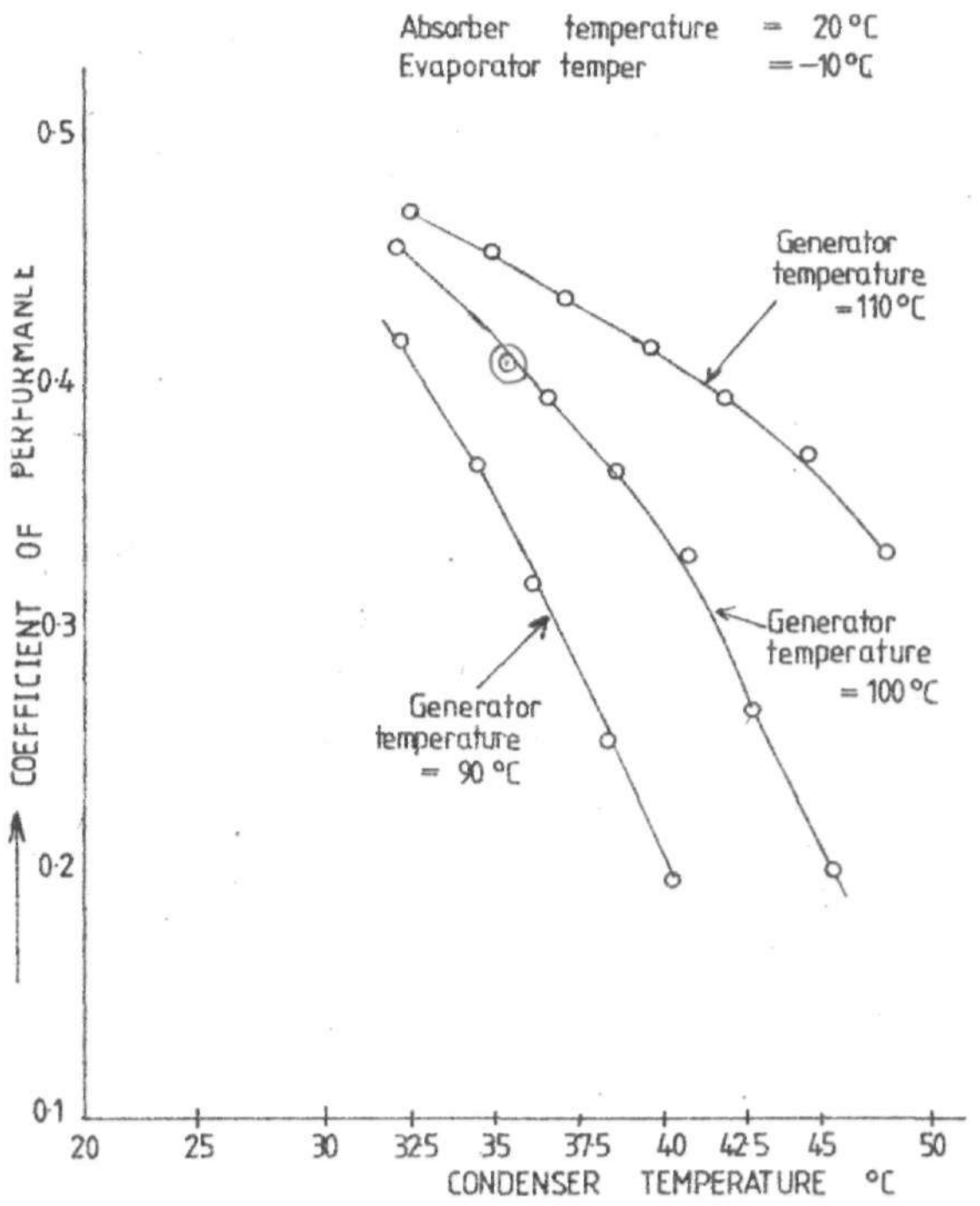


FIG (3·8)

VARIATION OF COEFFICIENT OF PERFORMANCE WITH CONDENSER TEMPERATURE

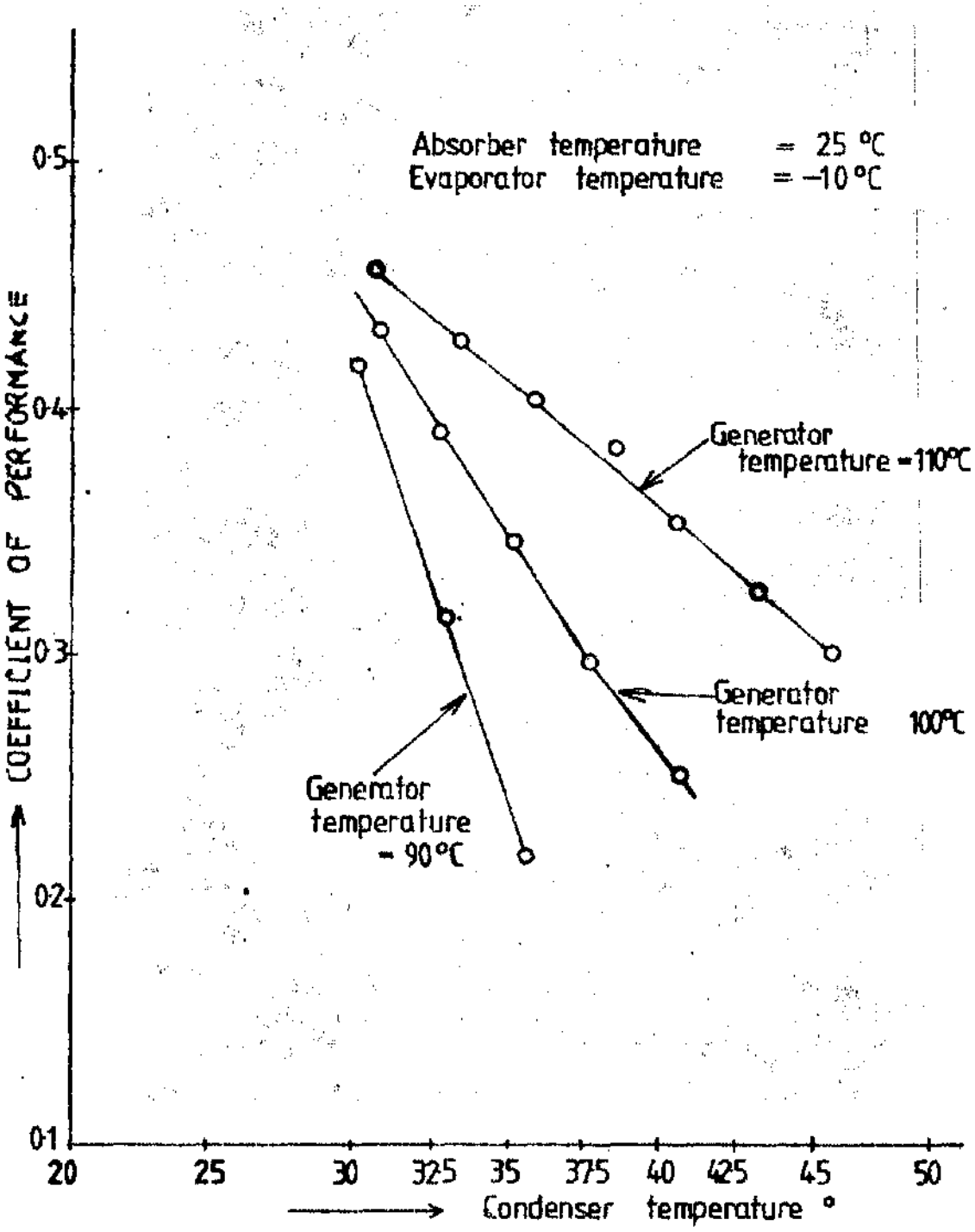


FIG (3·9)

VARIATION OF COEFFICIENT OF PERFORMANCE WITH CONDENSER AND GENERATOR TEMPERATURE

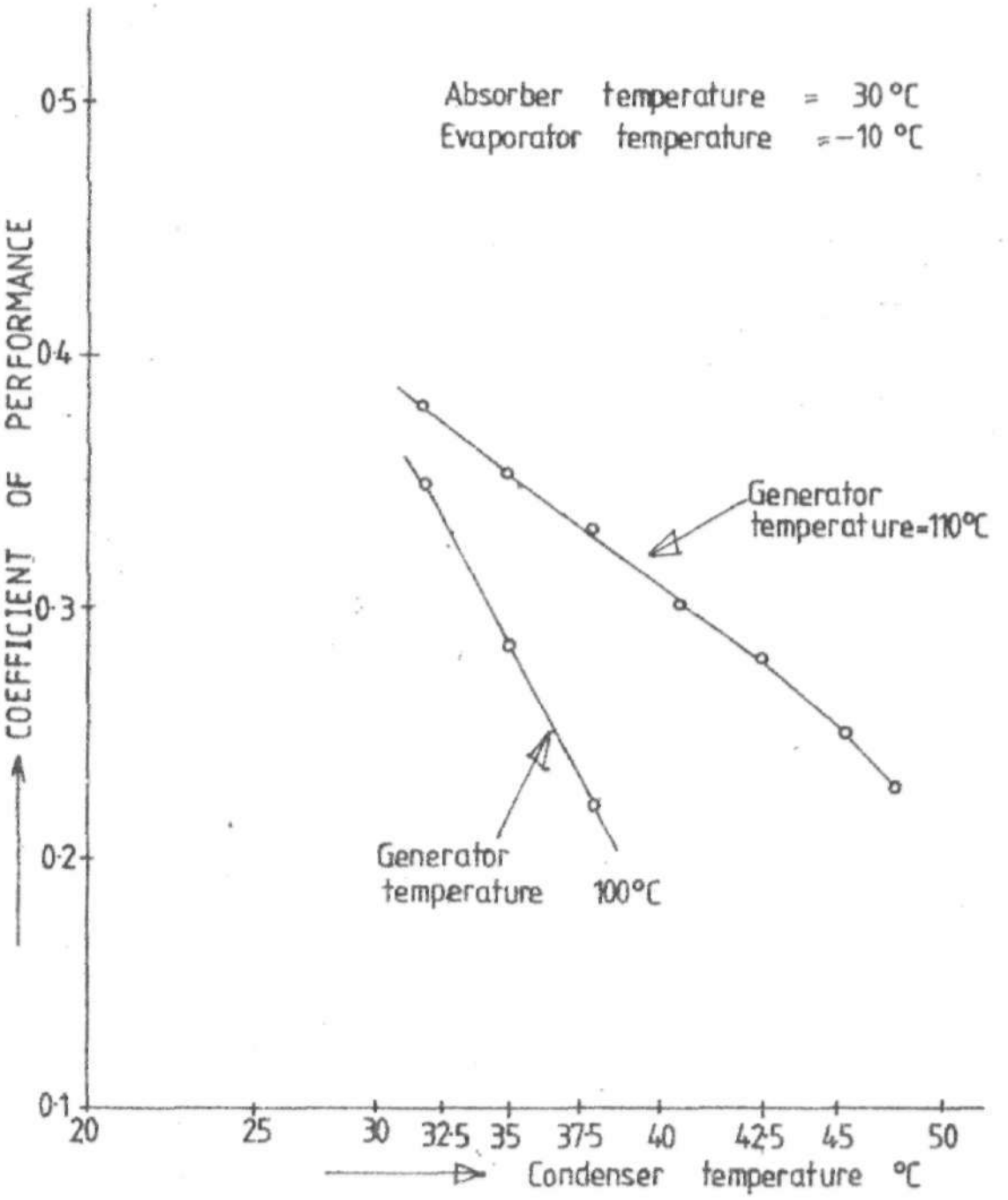


FIG. (3·10)

COEFFICIENT OF PERFORMANCE AT DIFFERENT GENERATOR, ABSORBER AND CONDENSER TEMPRATURE COMBINATIONS

EVAPORATOR TEMPRATURE = -10°C

Condenser Temprature °C	Generator Temprature	ABSORBER TEMPRATURE = 20°C			ABSORBER TEMPRATURE = 25°C			ABSORBER TEMPRATURE = 30°C		
		90°C	100°C	110°C	90°C	100°C	110°C	90°C	100°C	110°C
30		0.420	0.460	0.470	0.420	0.438	0.460	0.280	0.350	0.380
32.5		0.370	0.430	0.455	0.3175	0.394	0.435	-	0.285	0.355
35		0.320	0.400	0.440	0.215	0.350	0.410	-	0.220	0.330
37.5		0.260	0.370	0.420	-	0.3025	0.3875	-	-	0.305
40		0.200	0.340	0.400	-	0.255	0.365	-	-	0.280
42.5		-	0.270	0.3675	-	-	0.3375	-	-	0.2475
45		-	0.200	0.335	-	-	0.310	-	-	0.215

TABLE (3.2)

Refrigeration effect = $L_{te} - C_{pa}(t_g - t_e)$

$= 310 - 1.135\left(35 - (-10)\right)$

$= 310 - 51.07$

$= 258.93$ KCal/kg of NH_3 vapor

Coefficient of Performance = $\frac{\text{Refrigeration effect}}{\text{Generator Heat}}$

$= \frac{258.93}{661.5} = 0.398$

As per the feasibility analysis of the system at different condenser and absorber temprature combinations, keeping in mind the climatic conditions of Zaria and also for reasonable charge required for the system the following temprature have been assumed for design.

Generator temprature = $100^{o}C$

Condenser temprature = $35^{o}C$

Absorber temprature = $20^{o}C$

Evaporator temprature = $-10^{o}C$

Corresponding to the above temprature combination.

(i) Coefficient of Performance of system = 0.398

(ii) Rich solution circulation = 7.25 kg/kg of NH_3 vapor.

(iii) Rich solution circulation = 0.491

(iv) Refrigeration effect/kg of NH_3 vapor = 258.93 KCal.

CHAPTER 4

THERMAL DESIGN OF SYSTEM COMPONENTS

4.1 Introduction

In intermittent solar ice making plant there are only three Components.

(i) Solar collector cum generator/absorber

(ii) Condenser

(iii) Evaporator

This chapter presents the thermal design of the system components e.g. collector cum generator absorber for generation of required vapors, Condenser with flow of cooling water and evaporator.

4.2 Design of Collector Cum generator / Absorber

Collector Cum generator / absorber is a single unit which works as a generator in day time when the radiations fall on it and works as an absorber during the night when the temprature of the absorber falls down. There are two types of collectors that can be used for collection of Solar energy.

(a) Flat plate collectors

(b) Concentrating Collectors

Flat plate collector is the simplest means to collect solar energy for the system that requires thermal energy at

comparitively lower temprature. It consists of a flat metallic plate painted black on the surface facing the sun, insulated on the bottom side to reduce heat losses and covered from the top with transparent glass plates. The flat plate collectors have advantages over concentrating type of collectors.

(i) Flat plate collectors can be easily fabricated in any designed shape.

(ii) They collect both direct and diffuse radiation.

(iii) Their orientation is not critical.

(iv) Their initial and maintainence cost is low.

(v) They can be used as part of roof.

After careful study of characteristics and economy of both type of collectors, the flat plate collector is found to be more suitable for the intermittent ice making plant in which heat energy is required comparitively at a lower temprature.

4.2.1 CALCULATION FOR TOTAL RADIATION

The useful energy available from the solar collector is given by

$$Q_u = H_T \alpha \tau - U_L (t_p - t_a) \quad - \qquad (4.1)$$

The first term represents the total radiation absorbed by the collector in tilted position and second term represents overall heat losses from the collector.

Total energy absorbed by the collector is the sum of two components i.e. direct radiation intensity (I_{Dt}) and diffuse radiation intensity (I_{dt})

These components are given by the following expressions (Ref 2)

$$I_{Dt} = (\alpha\tau)_D R_D (H - D) \quad - \quad (4.2)$$

$$I_{dt} = (\alpha\tau)_d R_d (D) \quad - \quad (4.3)$$

The values of orientation factors R_D & R_d are given by

$$R_D = \frac{\cos \theta_t}{\cos \theta_h} \quad - \quad (4.4)$$

$$R_d = \frac{1 + \cos\beta}{2} \quad - \quad (4.5)$$

$$\text{Here } \cos t = \cos(\phi - \beta)\cos\delta\cos\omega + \sin(\phi - \beta)\sin\delta \quad (4.6)$$

$$\cos\theta_h = \sin\delta\sin\phi + \cos\delta\cos\phi\cos\omega \quad - \quad (4.7)$$

For year round performance angle of tilt of collector is generally taken equal to latitude of the place.(2)

Location of Samaru on the map of Nigeria = $11^\circ 11'N = 11.2^\circ N$.

Since Nigeria lies in the Northern hemisphere the flat plate Collector is tilted up from the horizontal and made to face south, in order to maximise soler irradiance.

Taking angle of tilt equal to the Latitude of the place, we have,

Angle of tilt of collector (β) = 1.0 x 11.2 = 11.2°

From radiation data for Samaru we see that the hottest month of the year is April. According to KOWAL AND KNABE (1) the maximum solar radiation falling on Samaru is in April. Since the declination angle varies daily, we assume April 8 as the date for calculation of declination angle.

Declination angle (δ) = $23.45 \text{ Sin } \left\{360 \times \frac{n + 284}{365}\right\}$

n = day of the year

Number of days till April 8 = 31 + 28 + 31 + 8 = 98

$$\delta = 23.45 \text{ Sin } \left\{360 \times \frac{98 + 284}{365}\right\}$$

$$= 23.45 \text{ Sin } \left(\frac{360 \times 382}{365}\right)$$

$$= 23.45 \text{ Sin } 376.76^\circ$$

$$= 23.45 \times 0.2883 = 6.8^\circ$$

Latitude angle = 11.2°N

Hour angle (ω) = 0 (at solar noon)

On Substituting these values in equation (4.6) and (4.7) we have,

$$\text{Cos}\theta_t = \text{Cos}(11.2^\circ - 11.2^\circ) \text{ Cos } 6.8^\circ \text{Cos } 0^\circ + \text{Sin } (11.2^\circ - 11.2^\circ) \text{ Sin } 6.8^\circ$$

$$= \text{Cos}(0^\circ) \text{ Cos } 6.8^\circ \text{ Cos } 0^\circ + \text{Sin } (0^\circ) \text{ Sin } 6.8^\circ$$

$$\text{Cos}\,\theta_t = 1 \text{ x } 1 \text{ x } 0.9925 + 0 \text{ x } 0.118$$

$$= 0.9925 + 0$$

$$\text{Cos}\,\theta_t = 0.9925$$

$$t = \text{Cos}^{-1}\, 0.9925 = 7.02^{\circ}$$

$$\text{Cos}\,\theta_h = \text{Cos } 6.8^{\circ} \text{ Cos } 11.2^{\circ} \text{ Cos } 0^{\circ} + \text{Sin } 6.8^{\circ} \text{ Sin } 11.2^{\circ}$$

$$\text{Cos}\,\theta_h = 0.9929 \text{ x } 1 \text{ x } 0.980 + 0.1184 \text{ x } 0.1942$$

$$= 0.9730 + 0.02299$$

$$= 0.996$$

On substituting the values of $\text{Cos}\,\theta_t$ and $\text{Cos}\,\theta_h$ in equation (4.4) and (4.5) we have,

$$R_D = \frac{0.9925}{0.9960} = 0.9964$$

$$R_d = \frac{1 + \text{Cos } 11.2^{\circ}}{2} = \frac{1 + 0.9725}{2} = 0.986$$

Effective absorptivity and transmissivity product for direct radiation is given by

$$(\alpha\tau)_D = 0.95 - \frac{0.115}{\text{Cos}\,\theta_t} \qquad - \quad (4.8)$$

$$= 0.95 - \frac{0.115}{0.992}$$

$$= 0.95 - 0.116 = 0.834$$

It is assumed that the diffuse radiations falling on the collector has an angle of 60°. So the value of effective absorptivity and transmissivity product for diffuse radiation (R_d) is taken as

$$(\alpha\tau)_d = 0.72 \qquad - \quad (4.9)$$

On substituting the values of equations (4.4), (4.5), (4.8) and (4.9) in equations (4.2) and (4.3) and adding for average hourly values of total radiation per unit area absorbed by the collector surface.

$$I_t = \bar{H}_h \left[(\alpha\tau)_D \frac{\cos\theta_t}{\cos\theta_h} \left\{ 1 - \frac{\bar{D}_h}{\bar{H}_h} \right\} + (\alpha\tau)_d \left\{ \frac{(1+\cos\beta)}{2} \right\} \frac{\bar{D}_h}{\bar{H}_h} \right]$$

$$= \bar{H}_h \left[\left\{ 0.834 \times \frac{0.992}{0.996} \right\} \left\{ 1 - \frac{\bar{D}_h}{\bar{H}_h} \right\} + 0.72 \times \left\{ \frac{1+\cos 13.45^\circ}{2} \right\} \times \frac{\bar{D}_h}{\bar{H}_h} \right]$$

$$= \bar{H}_h \times \left[0.83 \times \left\{ 1 - \frac{\bar{D}_h}{\bar{H}_h} \right\} + 0.72 \times 0.986 \times \frac{\bar{D}_h}{\bar{H}_h} \right]$$

$$= \bar{H}_h \times \left[0.83 \times \left\{ 1 - \frac{\bar{D}_h}{\bar{H}_h} \right\} + 0.71 \times \frac{\bar{D}_h}{\bar{H}_h} \right] \quad - \quad (4.10)$$

The ratio of monthly average total radiation ($\bar{H}$) on the horizontal surface to the extra-terrestial daily insolation (H_o) received on the horizontal surface is known as average cloudiness index $\bar{K}_T$.

The extra-terrestial daily insolation received on a horizontal surface is given by

$$H_o = \frac{24}{\pi} I_{sc} \left\{ \left\{ 1 + 0.033 \cos \left(\frac{360n}{365} \right) \right\} \times \left(\cos\phi \cos\delta \sin\omega_s + \frac{2\pi\omega_s}{360} \sin\phi \sin\delta \right) \right\}$$

The sunrise hour angle is given by

$$\text{Cos } \omega_s = - \tan \phi \tan \delta$$

$$= - \tan 11.2^{\circ} \tan 6.8^{\circ}$$

$$= - 0.198 \times 0.119 = - 0.0236$$

$$\omega_s = \text{Cos}^{-1} (- 0.0236) = 91.3^{\circ}$$

$$H_o = \frac{24}{3.14} \times \frac{1.94 \times 10000 \times 60}{1000}$$

$$\times \left\{ \left(1 + 0\ 033 \text{ Cos } \left(\frac{360 \times 98}{365} \right) \right) \left(\text{Cos } 11.2^{\circ} \text{Cos } 6.8^{\circ} \text{ Sin } 91.3^{\circ} \right. \right.$$
$$\left. \left. 3\ 14 \times 91.3 \times \text{Sin } 11.2^{\circ} \text{ Sin } 6.8^{\circ} \right) \right\}$$

$$H_o = 8896.8 \times \left\{ \left\{ 1 - 0.0038 \right\} \left(0.981 \times 0.993 \times 0.999 \right. \right.$$
$$\left. \left. + 1.59 \times 0.194 \times 0.118 \right) \right\}$$

$$H_o = 8896.8 \times (0.996 \times 1.009)$$

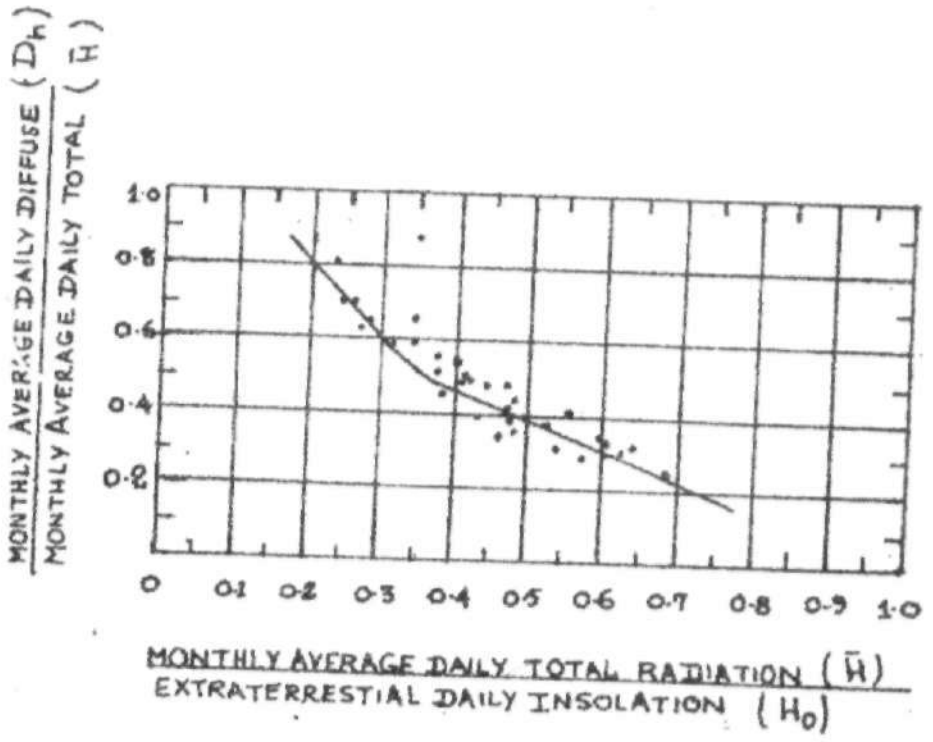
$$= 8941 \text{ KCal/m}^2\text{/ day}$$

Monthly average total radiation on horizontal surface in Samaru ($\bar{H}$) is 457 Cal/cm^2/day

$$\bar{H} = \frac{457 \times 10000}{1000} = 4570 \text{ KCal/m}^2\text{/day}$$

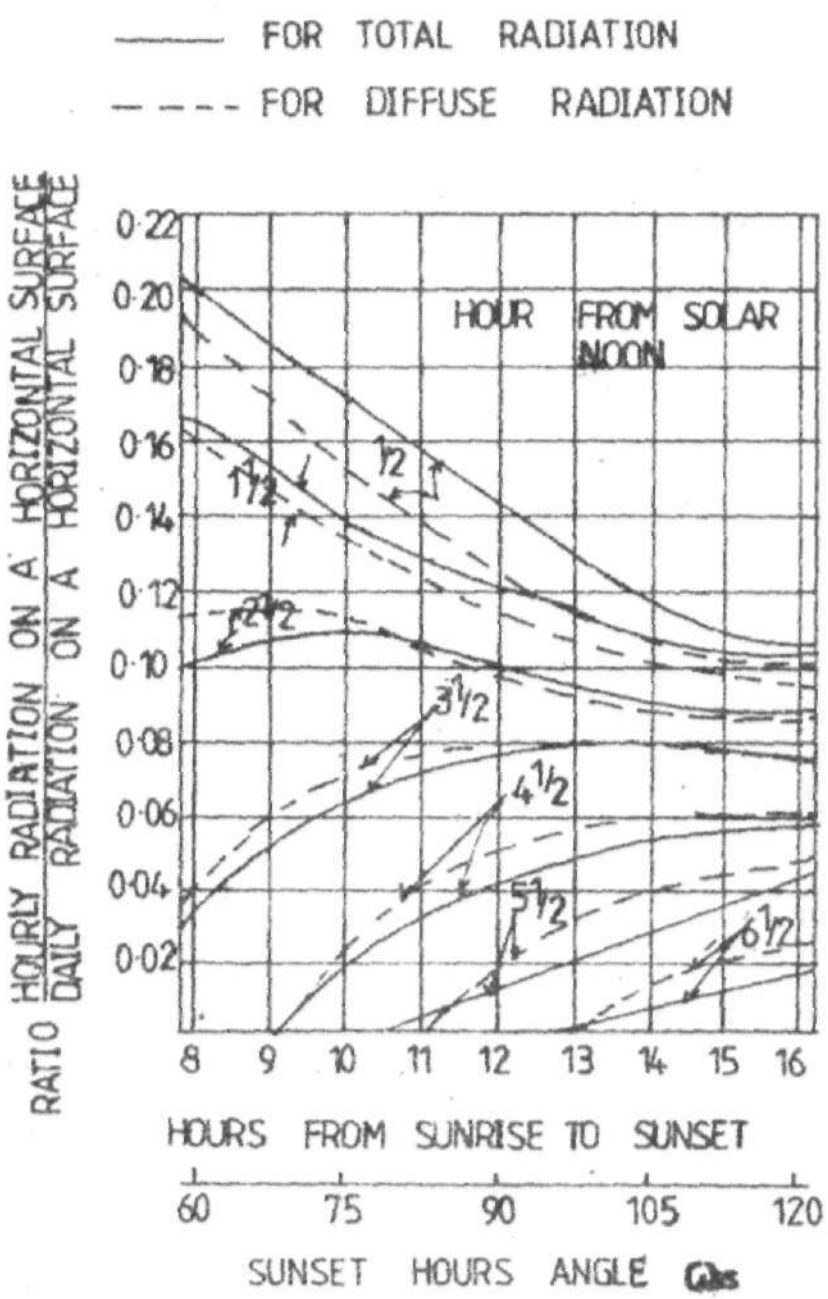
Hence average cloudiness index for Samaru is

$$\bar{K}_T = \frac{\bar{H}}{H_o} = \frac{4570}{8941} = 0.51$$



RATIO OF MONTHLY AVERAGES OF DAILY DIFFUSE TO DAILY TOTAL RADIATION AS A FUNCTION OF AVERAGE MONTHLY CLOUDINESS INDEX.

FIG 4.1 REF2



RELATIONSHIP BETWEEN DAILY RADIATION
AND HOURLY RADIATION ON A HORIZONTAL SURFACE.

FIG 4·2 REF 2.

From figure (4.1) Ref (2)

$$\bar{K}_T = \frac{\bar{H}}{\bar{H}_o}$$

Now at $\bar{K}_T$ = 0.51

$$\frac{\bar{D}_h}{\bar{H}} = 0.37 \quad \text{(Refer figure (4.1))}$$

$$\bar{D}_h = 0.37 \times 4570$$

The day length that is the number of hours from sunrise to sunset is given by

$$T_d = \frac{2}{15} \cos^{-1}(-\tan\phi \tan\delta)$$

$$= \frac{2}{15} \cos^{-1}(-\tan 11.2 \tan 6.8)$$

$$= \frac{2}{15} \cos^{-1}(-0.198 \times 0.119)$$

$$= \frac{2}{15} \cos^{-1}(-0.0236)$$

$$= \frac{2}{15} \times 91.5 = 12.2 \text{ hrs (approximately)}$$

For Sunset hour angle of 91.35° and day length of 12.2 hours and time 1300 hours i.e. one hour from solar noon we have from figure (4.2) Ref. (2)

$$r_t\left\{\frac{\text{Hourly Total Radiation}}{\text{Daily Total Radiation}}\right\} = 0.1375$$

$$r_d\left\{\frac{\text{Hourly Diffuse Radiation}}{\text{Daily Diffuse Radiation}}\right\} = 0.1260$$

Now ratio of monthly average hourly diffuse radiation to monthly average hourly total radiation.

$$\frac{\bar{D}_h}{\bar{H}_h} = \frac{r_d}{r_t}\left(\frac{\bar{D}_h}{\bar{H}}\right)$$

$$= \frac{0.126}{0.1375} \times 0.37 = 0.34$$

and Monthly average hourly total radiation on horizontal surface.

$$\bar{H}_h = r_t \bar{H} = 0.1375 \times 4570$$

$$= 628.4 \text{ KCal/hr m}^2$$

On Substituting all the values in equation (4.10) we have total radiation absorbed.

$$I_t = 628.4 \left\{0.83\ (1-0.34) + 0.71 \times 0.34\right\}$$

$$= 628.4 \left\{0.83 \times 0.66 + 0.71 \times 0.34\right\}$$

$$= 628.4 \left\{0.548 + 0.241\right\} = 496 \text{ KCal/hr m}^2$$

The above value for total radiation absorbed should be reduced to account the reduction in transmittance due to accumulation of dirt on glass cover and due to shading of collector by the side walls supporting the glass covers.

Duffie and Beckman[2] recommended that for design purposes without extensive test dirt correction factor may be taken as D = 0.98 and Whiller recommeds that shading correction factor should be 0.97 at noon. It varies for

other angle of Incidence.

Total radiation absorbed by collector

$I_t = 496 \times 0.98 \times 0.97$

$= 471.5$ KCal/hr m^2

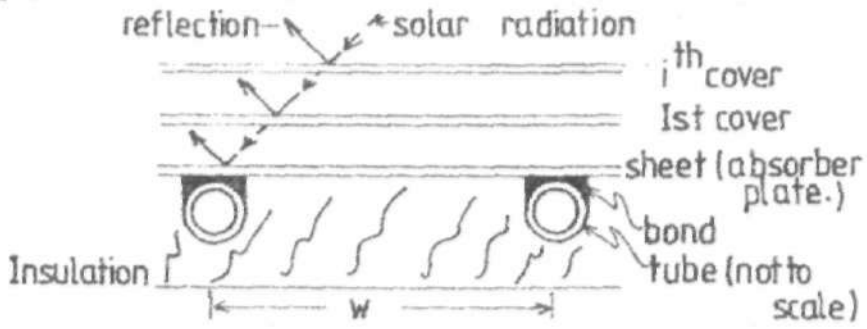
GENERAL DESCRIPTION OF A FLAT PLATE COLLECTOR

The important part of a typical flat plate collector are

(a) The black solar energy absorbing surface with means of transfering the absorbed energy to the fluid.

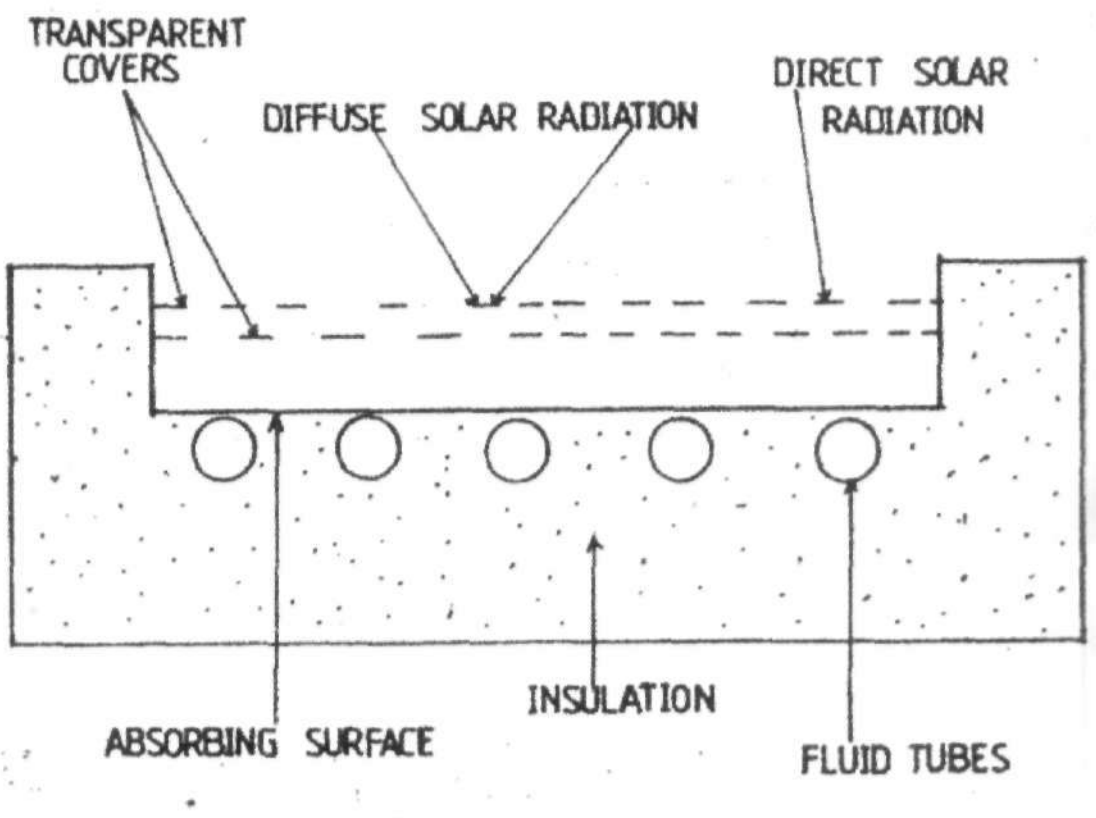
(b) Envelopes transparent to solar radiation over the solar absorber surface which reduces the convection and Radiation losses to the atmosphere.

(c) Back Insulation to reduce conduction losses as the geometry of the system permits. A general flat plate collector is shown in figure (4.3) The most commonly used configuration for flat plate collector is the fin & tube type, as shown below.



Sheet and tube solar collector

FIG (4·4) REF 2



BASIC FLAT PLATE COLLECTOR

FIG. (4.3) REF. 2

4.2.2 CALCULATION OF OVERALL HEAT LOSSES

Heat losses from the collector take place from the top bottom and edges. Losses from the top and bottom can be calculated easily. Calculation of losses from the edges is a complex problem Generally for large collector areas edge losses are negligible but for smaller collectors they are taken as a very small percentage of the bottom losses, (almost negligible).

Heat Loss from the top of the Collector

Top heat losses from the collector metal plate at any given temprature can be calculated (since the absorption of solar Radiation in the glass plate is small,, $\overline{U}$ top is not directly influenced by solar radiation. Variables determining the upward heat flux are

(a) Temprature of absorbing surface.

(b) Temprature of outer air and sky.

(c) Number of glass plates and spacing.

(d) Tilt of glass plates from the horizontal.

(e) Wind velocity over the top plate.

On the above basis Hottel and Woertz[2] developed an emperical relation which was modified by Klien. The relation developed was for an angle of tilt of 45^{o}, and is shown below.

$$U_t = \left(\frac{N}{(344/t_p)(t_p - t_a)/N + f)^{0.31}} + \frac{1}{h_w} \right)^{-1} + \frac{\sigma (t_p + t_a)(t_p^2 + t_a^2)}{\left(\varepsilon_p + 0.0425N (1-\varepsilon_p) \right)^{-1} + (2N + f - 1)/\varepsilon_g} - N \quad (4.11)$$

A relation which relates the outside heat transfer coefficient or wind heat transfer coefficient to the wind velocity was given by McAdams.[8] The heat transfer Coefficient was in $W/m^2\ ^oC$ whereas the wind speed was in. m/s the relation is shown below.

$$h = 5.7 + 3.8\ V$$

Taking wind speed as 3 m/s in Samaru we have.

$$h \text{ wind} = 5.7 + 3.8 \times 3$$

$$= 5.7 + 11.4 = 17.1\ W/m^2\ ^oC$$

Assuming double glass covers plate emmisivity ε_p as 0.95, Glass emittance ε_g as 0.88 and average plate temperature of 65 ^oC, Boltzmans Constant = 5.65 $W/m^2/^oK^4$

$$f = (1.0 - 0.04\ h + 5 \times 10^{-4}\ h^2)\ (1 + 0.058N)$$

$$= \left\{1.0 - 0.04 \times 17.1 + 5 \times 10^{-4} \times (17.1)^2\right\}(1 + 0.058 \times 2)$$

$$= 0.51$$

$$t_a = 0.7\ t_{max} + 0.3\ t_{min}.$$

$$= 0.7 \times 34.9 + 0.3 \times 22.1$$

$$= 24.4 + 6.6 = 31^oC.$$

Substituting these values in equation (4.11)

$$U_{top} = \left\{ \frac{2}{\frac{344}{338}\left(\frac{338-304}{2+0.51}\right)0.31} + \frac{1}{17.1} \right\}^{-1}$$

$$+ \frac{5.65 \times 10^{-8}(338+304)\left((338)^2 + (304)^2\right)}{\left\{0.95+0.0425x2x(1-0.95)\right\}^{-1} + \left(\frac{2x2+0.51-1}{0.88}\right)^{-2}}$$

$$= \left(\frac{2}{2.28} + \frac{1}{17.1}\right)^{-1} + \frac{5.65x10^{-8}x642x206660}{1.047 + 3.988 - 2}$$

$$U_{top} = 1.069+3.35 = 4.10 \text{ w/m}^2\ ^{o}C$$

For anyother angle of tilt the following expression given by Klein (1973) is used. $\frac{U_{t(\beta)}}{U_{t(45)}} = 1-(\beta-45)(0.00259-0.00144\varepsilon_p)$

$$\frac{U_{t(\beta)}}{U_{t(45)}} = 1-(11.2-45)(0.00259-0.00144 \times 0.95)$$

$$\frac{U_{t(\beta)}}{U_{t(45)}} = 1-(-33.8)(0.00259-.00136) \times \frac{3.14}{180}$$

$$= 1-(-33.8)(00123) \times \frac{3.14}{180}$$

$$= 1-(-.038806) \times \frac{3.14}{180}$$

$$= 1-(-.0006772)$$

$$= 1 + 0.0006772$$

$$= 1.0006772 = 1.00068$$

$$U_{t}(11.2^{o}) = 1.00068 \times 4.10$$

$$= 4.102 \text{ W/m}^2\ ^{o}C$$

Heat losses from the top per unit of collector area

$$= U_{top}\,(t_p - t_a)$$
$$= 4.1\,(65 - 31)$$
$$= 4.10 \times 34 = 139.4\ W/m^2$$

Heat loss from the bottom of collector

Losses at the bottom are by conduction through the insulation and then convection and radiation from the lower surface of the insulation to the environment. The resistance due to insulation is generally much larger than the resistance due to convention and radiation therefore the latter is neglected and the bottom loss coefficient is taken as follows:

$$U_{bottom} = \frac{K_q}{L_t} \quad - \quad (4.13)$$

K_q = 0.037 $W/m^{\circ}K$ for glass wool insulation

L_t = 5 cms = 0.0508 m, $U_{bottom} = \frac{0.037}{0.0508} = 0.728\ W/m^2\ ^{\circ}K.$

Heat losses from the bottom = 0.728 x (65-31)

= 0.728x34 = 24.75 W/m^2

Total losses = 139.4 + 24 . 75 = 164.15 W/m^2 = 141.01 KCal/hr m^2

Useful energy available from collector per m^2 of collector area = 471.5 - 141.01 = **330.5** KCal/hr m^2

4.2.3 CALCULATION OF COLLECTOR AREA.

Let the quantity of ice required be 10kg. Assuming mass of Ice equal to 10kg we can now proceed to calculate the amount of ammonia vapor required per day by making a heat balance.

Let us assume initial temprature of water = 26 ^{o}C

Now water at 26 ^{o}C first comes to 0 ^{o}C and then freezes into ice. Heat is therefore given up by water.

$$m (t_w - t) + L_i . m$$

= Refrigerating effect x Ammonia vapor required per day.

10 x (26 - 0) + 10 x 80 = Refrigerating effect x Ammonia vapor required per day.

{Latent heat of fusion of ice is 80 kCal/kg}

Now refrigerating effect = 258.93 KCal/kg of NH_3 vapor

Let x be the Amount of Ammonia vapor required per day.

$$10 \times (26-0) + 10 \times 80 = 258.93\ x$$
$$10 \times 26 + 10 \times 80 = 258.93\ x$$
$$10 \times 106 = 258.93\ x$$
$$x = \frac{10 \times 106}{258.93}$$
$$= 4.10 \text{ kg / day}$$

Total charge required = 7.25 x 4.10

= 29.68 kg

1 ton refrigeration = 3.516 kW

= 3516 Watts

Also 1.163 Watts = 1 KCal/hr

3516 watts = $\frac{3516}{1.163}$ = 3023.21 KCal/hr

1 ton refrigeration = 3023.21 KCal/hr

= 3023.21 x 24 KCal/day

Let y be the refrigeration Capacity of machine.

$$\frac{y \times 3023.2 \times 24}{258.93} = 4.1$$

$$y = \frac{4.1 \times 258.93}{3023.2 \times 24} = 0.01463 \text{ tons}$$

$$y = \frac{1}{68} \text{ tons}$$

Useful Energy available from collector per m^2 of collector area = 471.5 - 141.01 = 330.49 KCal/hr m^2

Total tonnage required = $\frac{3023.21}{68}$ = 44.5 KCal/hr

Coefficient of Performance of system = 0.398.

Energy required from collector = $\frac{44.5}{0.398}$ = 144.5 KCal/hr

Duration of bright sunshine = 6 hours.

Area of collector required = $\frac{144.5 \times 24}{330.5 \times 6}$ = 1.75 m^2

To be on the safe side we take the area as 1.8 m^2 ie 1.8 m x 1 m.

4.2.4 Next step is the selection of collector plate-tubes and headers. There are two important factors before this selection.

(i) Strength of tubes and headers

(ii) Size of the tubes and headers should be such that it can accomodate the whole of the charge required for 1/68 ton capacity.

Since copper is non-corrosive, has high thermal conductivity and can be easily fabricated & coated, we consider it, as the material for the absorber plate in the design of our collector. Keeping in mind the above factors we have selected, steel tubes of the following specifications:

Diameter of tubes = 1.5" = 3.8 cms

Thickness of tube = $\frac{1}{8}$" = 0.32 cms

Thickness of absorber plate = $\frac{1}{16}$" = 0.159 cms.

Recommended spacing between the tubes = 15 cms.

In this way 11 tubes can be arranged in the collector

Size of the headers has to be determined on the basis of volume required for the charge. Total charge required for $\frac{1}{68}$ ton capacity = 29.7 kg. Ammonia vapor required = 4.1 kg

Density of mixture at 49% NH_3 = 0.9 kg/litre. Volume required in collector = $\frac{29.7}{0.9}$ = 33 litres

Let the diameter of header = D

We condider that only 50 cm of tube length will be filled by the mixture.

So $0.5 \times 11 \times \frac{11}{4} \times (0.038)^2 + 2 \times 1.8 \times \frac{11}{4} D^2 = 0.033$

$6.237 \times 10^{-3} + 2.83 D^2 = 0.033$, $2.83 D^2 = 0.033 - 0.0062$

$2.83 D^2 = 2.68 \times 10^{-2}$, $D^2 = 0.9469 \times 10^{-2}$, $D = 9.730 \times 10^{-2}$ m

or D = 0.0973 m = 9.73 cms. **D = 9.73 cms**

The detailed drawing of the collector assembly is shown in Fig. (4.5) at the end of the thesis.

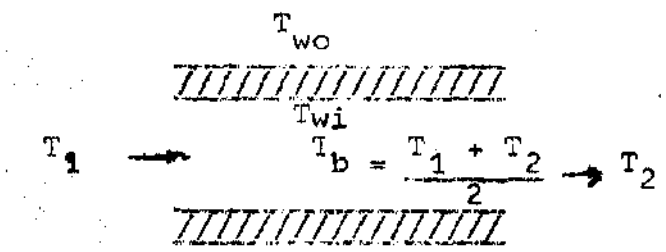
4.3 DESIGN OF CONDENSER

This is an important component of the refrigeration system which needs more Consideration in design and construction. The condenser removes heat from the refrigerant carried from evaporator and added by generator in vapor absorption system and converts the vapor refrigerant into liquid refrigerant. It is simply a heat exchanger in which heat transfar takes place from high temprature vapor to low temprature air or water which is used as a cooling medium. There are different types of condensers used in refrigeration systems. In the present system a water cooled condenser has been designed.

4.3.1 Design Of Condenser Coil

The Condenser coil surface area is calculated on the basis of complete Condensation during the regeneration period. For its design 2.54 cm nominal diameter tube has been taken and water side heat transfer coefficient and refrigerant side heat transfer coefficient have been calculated as follows.

Water side or Tube Inside Heat Transfer Coefficient



Latent heat of condensation of Ammonia = 313 KCal/Kg.

Mass flow rate of Ammonia vapor = 4.10 Kg/day

Total heat removed in condenser = 313 x 4.1

= 1283.3 KCal/day.

Let cooling water enter at a temprature of 26^{o} C

Total heat rejected in condenser (q_r) = 1283.3 KCal/day

Assuming condensation to take place from 10 a.m to 4 p.m. i.e. 6 hours we have $q_r = \frac{1283.3}{6} = 213.9$ KCal/hr

In order to ensure a heat balance this heat rejected by the vapor has to be taken up by the cooling water.

According to the heat balance equation

$q_r = \dot{m} C_p \Delta T = \dot{m} C_p (T_2 - T_1)$

Now mass flow rate of water is given by $\dot{m} = \rho A.V$

Density of water at 26^{o} C = 999.08 Kg/m^3

Let us assume the velocity of wate as 0.1 m/s

$\dot{m} = 999.08 \times \frac{\pi}{4} \times (0.026)^2 \times 0.1 = 0.053$ Kg/sec.

Substituting this value of $\dot{m}$ in the heat balance eq^{n}

Specific heat of water at 26° C = 1 KCal/Kg $^{\circ}$C

$\therefore$ $213.9 = 0.053 \times 3600 \times 1 \times (T_2 - 26)$

or $T_2 = 26 + 1.12 = 27.12^{\circ}$ C

$\therefore$ Bulk temprature of water = $\frac{27.12 + 26}{2} = 26.56^{\circ}$ C

Assuming a wall temprature of 32° C we can now calculate the heat transfer coefficient on the water and later check whether our assumed value is correct or not.

Evaluating properties of water at $\frac{26.56 + 32}{2} = 29.3^{\circ}$C

ρ = 999.02 kg/m^3

C_p = 1 KCal/kg$^{\circ}$C

ν = 0.0847×10^{-5} m^2/s

K = 0.6114/1.163 = 0.525 KCal/hr m$^{\circ}$C

Pr = 5.774

Now Reynolds Number (Re) = $\frac{Vd}{\nu} = \frac{0.1 \times 0.026}{0.0847 \times 10^{-5}} = 3069$

This value of Reynolds Number lies in the Turbulent range

Hence we use the Dittus Boelter Equation,

$Nu = 0.023 (Re)^{0.8}(Pr)^{0.4} = 0.023(3069)^{0.8}(5.77)^{0.4}$

$Nu = 0.023 \times 616.02 \times 2.01 = 28.5$

Now $Nu = \frac{h_w \cdot d}{K}$ or $h_w = \frac{Nu \times K}{d} = \frac{28.5 \times 0.525}{0.026}$

or h_w = 575.5 KCal/hr m^2 $^{\circ}$C

Refrigerant side or Tube outside Heat transfer coefficient

The following equation is used to calculate the heat transfer coefficient on the refrigerant side.

$$h_r = 0.725 \left(\frac{\rho_l^2 g L_c K_l^3}{\mu_l d_o (T_s - T_w)} \right)^{1/4} \quad (4.14) \text{——— (Ref 11)}$$

The above equation has been derived for calculation in British units in the case of one tube. Hence we will first calculate the value of h_r in British units and then can convert the result obtained in S.I units. Properties of Ammonia evaluated at $\frac{35 + 32}{2}$ i.e. 33.5°C or 91.4°F are

K_l = 0.29 BTU/hr ft°F

ρ_l = 36.88 lbm/ft^3

g = 4.17 x 10^8 ft/hr^2

L_c = 486.7 BTU/lbm

μ_l = 0.504 lbm/hr ft

Substituting this values in equation 4.14 we have

$$h_r = 0.725 \left(\frac{(36.88)^2 \times 4.17 \times 10^8 \times 486.7 \times (0.29)^3}{0.504 \times 0.108 \times 7.2} \right)^{1/4}$$

$$= 0.725\ (2035.8) = 1476 \text{ BTU/hr ft}^2\ ^\circ\text{F}$$

$$= \frac{1476 \times 0.252 \times 9}{5 \times 0.0929} = 7206.8 \text{ KCal/hr m}^2\ ^\circ\text{C}$$

4.3.2 CALCULATION FOR SIZE OF RECEIVER

In the condenser, the receiver has been provided to collect the Condensed Ammonia. Size of the receiver has been calculated as follows:-

Weight of Ammonia vapours required per day = 4.1 kg

Density of Ammonia at 35°C temperature = 588 kg/m^3

∴ Volume of receiver required to keep this Ammonia

$$= \frac{4.1}{588} = 6.97 \times 10^{-3}\ m^3$$

Taking diameter of receiver as 12.7 cms and material as steel, the length of receiver will be

$$\frac{\pi}{4} D^2 L = 6.97 \times 10^{-3}$$

$$L = \frac{6.97 \times 10^{-3} \times 4}{\pi \times (0.127)^2}$$

$$L = \underline{0.5 \text{ meters}}$$

The detailed drawing of the condenser assembly is shown in Fig. (4.6) at the end of the thesis.

We will now calculate the overall heat transfer coefficient which is given by

$$\frac{1}{U} = \frac{1}{h_r} + \frac{1}{h_w} \cdot \frac{A_o}{A_i} + \frac{r_2}{K} \cdot \ln \frac{r_2}{r_1}$$

$$= \frac{1}{7206.8} + \frac{1}{575.5} \cdot \frac{0.033}{0.026} + \frac{0.033}{40.9} \ln \cdot \frac{0.033}{0.026}$$

$$= 10^{-4} (1.387 + 17.3 + 1.923) = 20.61 \times 10^{-4}$$

or $U = \frac{10^4}{20.61} = 485.4$ KCal/hr m^2 ^oC

Since $q = U.A.\Delta T = U.\pi. d_o.l.(T_s - T_b)$

$\therefore$ $213.9 = 485.4 \times 3.14 \times 0.033 \times l \times (35 - 26.56$

or $l = 0.5$ meters

We now check back for the wall temprature and see if our assumed value is correct or not.

Since $h_r.A_o.(T_s - T_w) = h_w.A_i.(T_w - 26.56)$

$7206.8 \times 0.033 \times (35 - T_w = 575.5 \times 0.026 \times (T_w - 26.56)$

or $556.32 - 15.89\ T_w = T_w - 26.56$

$582.88 = 16.89\ T_w$ or $T_w = \underline{32.5^oC}$

This value of T_w is almost equal to our previously assumed value of T_w. Hence we need not calculate again using hit and trial.

4.4 DESIGN OF EVAPORATOR

Evaporator is a part of refrigeration system in which refrigerant is vaporized to produce refrigeration. In the system being designed here, brine is used as a secondary refrigerant (25 percent common salt in water which has a freezing point below 15°F). The liquid refrigerant which comes from the condenser first collects in evaporator receiver, then it is passed to the evaporator coil where it gets vaporized & produces cooling. The refrigerant vapor which forms in the evaporator gets absorbed by the absorbent in collector cum absorber.

4.4.1 DESIGN OF EVAPORATOR COIL

In evaporator coil heat is transferred from brine to the refrigerant boiling at the evaporator temprature. The heat transfer coefficient of refrigerant side is considerably higher compare with heat transfer coefficient of the brine side. The total amount of heat transfer in evaporator is given by $Q = U_A(t_b - t_s) = h_b\ A_o\ (t_b - t_w) = h_r\ A_i\ (t_w - t_s)$
Brine and refrigerant side heat transfer Coefficient are required to be calculated for calculating the area required for evaporator coil for complete heat transfer to take place from refrigerent to the brine solution.

REFRIGERANT SIDE HEAT TRANSFER COEFFICIENT

The following equation has been developed to calculate the refrigerant side heat transfer coefficient.

$$\frac{C_p\,(t_w - t_{sat})}{Pr^{1.7} \times L_v} = C_{sf}\left\{\frac{q/A_1}{\mu_l L_v}\sqrt{\frac{g_c \cdot \sigma^*}{g(\rho_l - \rho_v)}}\right\}^{1/3} \qquad (4.\overline{15})$$

Saturation temperatire of

liquid ammonia = $-10^{\circ}C$

Temperature of brine varies from - 5 to - $8^{\circ}C$

Let wall temperature = - $8^{\circ}C$

Properties of ammonia at Saturation temperature

C_p = 1.09 KCal/Kg $^{\circ}C$

L_v = 309.4 KCal/Kg

ρ_l = 652 Kg/m^3

ρ_v = 2.4 Kg/m^3

μ_l = 0.89 Kg/m-hr

K = 0.467 K Cal/hr-m $^{\circ}C$

g_c = 1.275×10^8 Kgm/Kgf hr^2

C_{sf} = 0.013 , σ^* = 23.9×10^{-4} kgf/m

Prandtl Number (Pr) = $\frac{C_p \cdot \mu}{K}$

= $\frac{1.09 \times 0.89}{0.467}$ = 2.1

On substituting the above values we have

$$\frac{1.09 \times (-8-(-10)}{(2.1)^{1.7} \times 309.4} = 0.013 \left\{\frac{q/A_1}{0.89 \times 309.4} \sqrt{\frac{23.9 \times 10^{-4}}{(652 - 2.4)}}\right\}^{1/3}$$

$$\text{or} \quad \frac{1.09 \times 2}{3.53 \times 309.4} = 0.013 \left\{\frac{q/A_i}{276} \quad 3.68 \times 10^{-6}\right\}^{1/3}$$

$$\text{or} \quad 1.996 = 0.013 \times 1000 \left\{\frac{q/A_i}{276} \times 1.92 \times 10^{-3}\right\}^{1/3}$$

$$\text{or} \quad \left\{\frac{1.996}{13}\right\} = \frac{q/A_i}{276} \times 1.92 \times 10^{-3}$$

$$\text{or} \quad (0.1535)^3 = q/A_i \times 6.96 \times 10^{-6}$$

$$q/A_i = \frac{0.0036 \times 10^6}{6.96} = 517 \text{ K Cal/hr-m}^2$$

Brine Side Heat Transfer Coefficient

Brine side heat transfer coefficient is calculated using following equation

$$Nu = 0.53 \ (Gr \ Pr)^{1/4} \quad - \quad (1.15)$$

The properties for the above eqn have been seen at the mean bulk temperature of brine which is calculated as below

$$\text{L. M. T. D.} = \frac{(t_s - t_{b1}) - (t_s - t_{b2})}{\ln \left\{\frac{t_s - t_{b1}}{t_s - t_{b2}}\right\}}$$

$$= \frac{(-10-(-8)) - (-10-(-5))}{\ln \left\{\frac{-10-(-8)}{-10-(-5)}\right\}}$$

$$-\frac{2+5}{\ln \frac{-2}{-5}} = \frac{3}{\ln 0.4} = \frac{3}{0.916}$$

$$= -3.28^{\circ}C$$

$$\text{Bulk temperature} = \frac{10 - 3.28}{2} = -\frac{13.28}{2} = -6.64 \ ^{\circ}C$$

Properties of brine at temperature of - 6.64 ^{o}C

at 24% concentration are

μ = 15.25 kg/m hr

C_p = 0.785 KCal/kg ^{o}C

K = 0.358 KCal/hr m ^{o}C

ρ = 1194 kg/m^3

Prandtl Number (Pr) = $\frac{C_p}{K}$

= $\frac{15.25 \times 0.785}{0.358}$

= 33.4

Grashoffs Number = $\frac{\beta d^3 g \Delta t \rho^2}{\mu^2}$

= $\frac{(0.033)^3 \times 9.81 \times (1194)^2 \times 3.75 \times 10^{-3} \times 2}{233}$

$\frac{271 \times 36 \times 10^{-6} \times 9.81 \times 13 \times 10^{6} \times 3.75 \times 10^{-3} \times 10^{6} \times 1.42}{233}$

= 28.4×10^{6}

Nu = $\frac{h_b d_i}{K}$ = 0.53 (Gr Pr) $^{1/4}$

$\frac{h_b d_i}{K}$ = 0.53 $(33.4 \times 28.4 \times 10^{6})^{1/4}$

$h_b \times \frac{0.026}{0.358}$ = $0.53(33.4 \times 28.4 \times 10^{6})^{1/4}$

h_b = $\frac{.358}{.026} \times 0.53 (33.4 \times 28.4 \times 10^{6})^{1/4}$

h_b = 1280.5 KCal/hr m^2 ^{o}C

$$\frac{q}{A_o} = h_b\ (-6.64 + 8) = 1280.5 \times 1.36$$

$$= 1741 \text{ KCal/hr } m^2$$

The heat transfer from refrigerant to pipe wall is the same as heat transfer from pipe wall to brine.

$$\frac{q}{A_o} = \frac{q}{A_i} \times \frac{A_i}{A_o} = 517 \times \frac{0.026}{0.033}$$

$$= 408 \text{ KCal/hr } m^2$$

The above value of q/A_o does not agree with the value calculated before. So we need a second trial for wall temperature. Let wall temperature $= -7.4\ ^{\circ}C$

Again substituting the values in equation (4.15) we have

$$\frac{1.09 \times (-7.4 - (-10))}{(2.1)^{1.7} \times 309.4} = 0.013 \left\{ \frac{q/A_i}{0.89 \times 309.4} \frac{23.9 \times 10^{-4}}{(652 - 2.4)} \right\}^{1/3}$$

$$\frac{1.09 \times 2.6}{3.53 \times 309.4} = 0.013 \left\{ \frac{q/A_i}{276} \frac{23.9 \times 10^{-4}}{(652 - 2.4)} \right\}^{1/3}$$

$$\frac{2.6}{13.} = \left\{ \frac{\frac{q}{A_i}}{276} \times 1.92 \times 10^{-3} \right\}^{1/3}$$

Cubing both sides we have $\left\{\frac{2.6}{13}\right\}^3 = \frac{\frac{q}{A_i}}{276} \times 1.92 \times 10^{-3}$

$$\text{or } \frac{q}{A_i} = \frac{(0.2)^3 \times 276}{1.92 \times 10^{-3}}$$

$$= \frac{8 \times 10^{-3} \times 10^3 \times 276}{1.92}$$

$$\frac{q}{A_i} = 1150 \text{ KCal/hr } m^2$$

$$\frac{q}{A_o} = \frac{q}{A_i} \times \frac{A_i}{A_o} = 1150 \times \frac{0.026}{0.033} = 906 \text{ KCal/hr } m^2$$

By the other method we have $\frac{q}{A_o} = h_b \ (-6.64 + 7.4)$

$$= 1280.5 \times 0.76$$

$$= 973.2 \text{ KCal/hr } m^2$$

Since both the values seem to be nearest to each other the wall temperature can be taken as

$$t_{wall} = -7.4 \ ^{o}C$$

In the above q is nothing but the refrigerating effect that takes place in the evaporator.

$$q = 258.93 \times 4.10 = 1061.6 \text{ KCal/day}$$

Assuming evaporation to take place for 6 hours i.e. from 10 p.m. to 4 a.m. we have

$$q = \frac{1061.6}{6} = h_b A_o (t_b - t_w)$$

$$\text{or} \quad 1280.5 \times A_o \times (-6.64 + 7.4) = \frac{1061.6}{6}$$

$$A_o = \frac{1061.6}{6 \times 1280.5 \times 0.76} = 0.18 \ m^2$$

$$\text{Now } A_o = \pi d_o l$$

$$0.18 = \pi \times 0.033 \times l$$

$$\text{or } l = \frac{0.18}{\pi \times 0.033} = \underline{1.75 \text{ meters}}$$

Volume of each can = $\frac{10869.5}{3}$

= 3623.2 cm^3

Taking diameter of cans as 12.7 cms we have

$$\frac{\pi}{4} \times D^2 \times h = 3623.2$$

$$\frac{\pi}{4} \times (12.7)^2 \times h = 3623.2$$

$$h = \frac{3623.2 \times 4}{\pi \times (12.7)^2}$$

$$h = 28.6 \text{ cms}$$

We make three cylindrical Ice cans of diameter 12.7 cans and height 28 6 cms each.

4.4.4 CALCULATION OF INSULATION THICKNESS

Thickness of insulation should at least be equal to 50mm of Mineral wool.

Thickness of mineral wool = (X_1) = 50 mm

Let thickness of glasswool Insulation = (X_2) mm

Thermal Conductivity of mineral wool (K_{mw})= 0.027 KCal/hr m ^{o}C

Thermal Conductivity of glasswool = 0.031 KCal/hr m ^{o}C

$$\frac{0.027}{50} = \frac{0.031}{X_2}$$

or $X_2 = \frac{50 \times 0.031}{0.027}$ = 6 cm = 0.06 meters

The detailed drawing of the evaporator assembly is shown in Fig. (4.7) at the end of the thesis.

4.5 SUMMARY OF DESIGN

The designed solar collector is made up of eleven 3.8 cm diameter, 100 cm long steel tubes, with two Mild steel headers each of diameter 9.7 cms and length 176 cms. Both ends of the tubes are welded in the headers after cutting holes in them to get proper connections. The ends of the headers are closed by 1 cm thick mild steel discs welded to it. A copper sheet of 0.159 cm thickness is welded in the upper face of the collector tubes.

The collector sheet has been painted with Commercial black in order to absorb solar radiations. A pipe line of 3.8 cm diameter has been provided on the upper header for charging purposes. Two more connections of 2.54 cm diameter have been given from the upper and lower header to connect the collector or generator/absorber with the condenser and evaporator respectively. The whole collector assembly has been kept in a frame made of 2.54 cm thick plywood at the bottom of which 5 cm thick glasswool insulation is provided to reduce bottom losses. The upper face is covered by 4 mm thick double glass plates fixed in the plywood frame.

The spacing between the two glass plates and the metal plate to the inner plate are 2.5 cm and 5 cm respectively. The glass plates can be kept open with the help of angle Irons. The collector has been placed in an angle iron stand with a slope of 11.2° with the horizontal.

A double tube water cooled condenser has been designed. The outer tube has been connected to the collector whereas water from the city mains flows through the inner tube. A 02.54cm nominal diameter, 0.5 m long steel tube has been designed for the purpose. One 2.54 cm diameter pipeline has been taken out from the outer tube to connect it to the evaporator receiver.

The evaporator receiver is made from 12.7 cm diameter, 0.5 m long Mild Steel pipe with ends closed by 1 cm thick Mild Steel discs. Evaporator coil of 2.54 cm diameter, 1.75meters length, steel pipe has been designed. The coil is bended in such a fashion that it may come around the three ice cans made of the designed size. One end of the coil is connected with the receiver and the other with the collector.

A brine tank 68 cm x 60 cm x 30 cm has been made. The receiver is kept on the brine tank with the help of an Angle Iron Stand. The brine tank is housed in a 2.54 cm thick wooden box of size 80 cm x 72 cm x 38 cm. A glasswool insulation of 6 cms is provided in the space between the wooden box and the brine tank.

A top has been cut in the wooden box to take out the ice cans.

Three cylindrical ice cans of diameter 12.7 cms and length 28.6 cms have been designed. The cans are made of Galvanised Iron Sheet.

The whole evaporator assembly is kept on an angle Iron Stand. Stop Here

The system assembly/flow diagram is shown in Fig. (4.8) at the end.

CHAPTER 5

GENERAL REMARKS

5.1 Fabrication

It was intended that the system designed above be fabricated and tested so that the results obtained could be compared with those obtained by other people. Unfortunately this could not be carried out, firstly because of lack of time and secondly due to shortage of materials required for the fabrication.

5.2 System Assembly

Assembly of the Intermittent System is nothing but the joining of the three Components ie. the collector cum generator/absorber, Condenser and evaporator. Connections can be done with a 2.54 cm diameter welded steel pipe as generating pressures are quite high. One end of the collector can be connected to the outer tube of the Double tube water cooled Condenser. The outer tube can be made of a suitable diameter so that it can act as a receiver. The other end of the outer tube can be connected to the evaporator receiver. Water flows in through one end of the inner tube and leaves through the other thereby condensing the refrigerant on the

inside of the outer tube. One end of the evaporator coil can be connected to the evaporator receiver and the other to the generator/absorber. The valves can be provided in each connection to open or close the line.

Pressure gauges can be provided at the generator exit to measure the pressure during regeneration and refrigeration. Thermometer pockets can also be provided at the exit of the generator, condenser and evaporator to measure the temperatures. A charging line could be provided at one end of the top header to charge the mixture into the system.

5.3 Charging of Ammonia-Sodium Thiocyanate in the System

Till to date no specific procedure has been given in the literature for charging. If the system is fabricated the following procedure could be adopted for the charging of the mixture in the system.

First of all the air is removed from the system by a vacuum pump. A required quantity of Sodium Thiocyanate is then put in a separate cylinder, modified for the purpose and air is removed from the cylinder. After making a fairly complete vacuum, ammonia is charged into the cylinder as per the quantity required.

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The cylinder is then closed and well shaken to ensure complete solution of all salt. During the process the cylinder gets heated up, and its pressure rises. It has to be cooled down by keeping it in ice cold water so that the heat liberated during mixing is removed and the pressure decreases. This way the mixture of Ammonia-Sodium Thiocyanate can be made in a separate cylinder. Now to charge this mixture into the generator the cylinder is put at a higher level than the generator, to make a gravity flow. Charging can be done early in the morning when the cylinder pressure is considerably higher than the generator pressure. The complete charge is therefore transferred to the generator from the cylinder first due to pressure difference followed by gravity flow.

5.4 PRESSURE TESTING OF THE SYSTEM

If fabricated, the pressure testing of the system could be done in two phases. First of all the system components can be tested separately to a pressure about 1½ the designed pressure using Carbon dioxide gas. Then in the second phase the system components can be tested again for the same pressure. Further to check minor leakages the system can be left under the same pressure for about 2 days. Finally vacuum can also be performed to avoid any leakage of air into the system.

5.5 Conclusions

Theoretically it can be seen that the co-efficient of performance of a system operating on liquid absorbent depends critically on the final temperature reached during generation. As the final generation temperature is increased more and more refrigerant is driven off, and the concentration difference between the strong and weak solution increases. As a result, the amount of solution, the heating of which to the generation temperature constitutes a significant loss decreases and C.O.P. goes up.

The system using liquid absorbent is also sensitive to changes on condensing temperature. For a given final generation temperature the coefficient of performance decreases rapidly with increase in condensing temperature. This is because, as the condensing temperature increase the concentration difference between the strong and weak solution decreases and larger the amount of solution that is needed.

5.6 Suggestions for further work

The following suggestions should be incorporated in the course of time, to enhance the efficiency of the solar collector by conducting tests on an experimental model of the designed system.

1. A flat (planar) reflective surface (sheet metal or foiled surfaces) may be placed near the flat plate collector in order to achieve solar thermal concentration. Both direct and diffuse sunlight can be concentrated in this way. This can add significantly to the amount of sunlight striking the collector, and in some cases may even double the amount of sunlight. However since both the reflector and collector will be mounted in a fixed position the concentrating effect is significant only at certain times. The appropriateness of using such reflectors depends on location and material used.

2. One may try various concentrations of the Ammonia-Sodium Thiocyanate mixture and see which gives better results.

3. Selective surfaces may be used which have a high absorptance for solar radiation and low emittance for long wave radiation (for details see APPENDIX 1 and REF. 2 Pages (98 - 103).

4. To suppress Convention honeycombs may be used (FOR DETAILS SEE APPENDIX 1).

APPENDICES

APPENDIX 1

IMPROVEMENT OF PERFORMANCE OF FLAT PLATE COLLECTORS

There are a number of potential ways to improve the performance of a flat plate collector, but each exacts a cost. To determine whether a given way of improving performance is cost effective we need to determine to what extent it improves the efficiency of energy extraction. The ways to reduce losses and increase efficiency of extraction of solar heat are:

1. Reduce Convection losses by
   - (a) Inverting the absorber and other geometrical means.
   - (b) Using honey combs.
   - (c) Reducing the temprature difference between surfaces.
2. Reduce radiation losses by using.
   - (a) Selective absorbers.
   - (b) Selective windows
   - (c) Both Combined

Each of these methods is described in brief below.

1 (a) Convection occurs when a lower surface is hotter than free air and where buoyancy effects can move the heated air upward. If the heated surface faces down, for example, convection cannot occur because the air layers are stable up against the heated surface. The loss is then by Conduction alone (inthe absence of wind to move the heated air).

One can thus place the collector absorber and window facing downward and use a mirror to reflect sunlight up into the collector. This method although does eliminate convection but it suffers from the additional reflectivity loss by the required mirror.

Geometrical suppression of convection may be useful in cases where the reflection loss is small compared to the convective loss or where a mirror already is used to concentrate light on the absorber.

Convection loss is also a function of the shape of the heated suface and the tilt of a flat plate surface. The largest convection loss is that from a horizontal flat plate. For vertical surfaces the loss is about 0.8 that of a horizontal surface.

1 (b) Suppression of convection can also take place through the use of honey combs. When a vertical channel is made small it has been found that vertical convection does not begin until a certain finite temprature difference between the top and bottom of the channel is exceeded. Acertain minimum Rayleigh Number must be exceed for the onset of convection, where Rayleigh Number is defined as

$$Ra = qL^3 \rho^2 g\gamma C_p/\mu K^2 \chi$$

where

q = heat flux per unit time.

L = Characteristic dimension, the width of the channel.

y = Volumetric expansion of the fluid.

Cp = Secific heat of the fluid at Constant pressure.

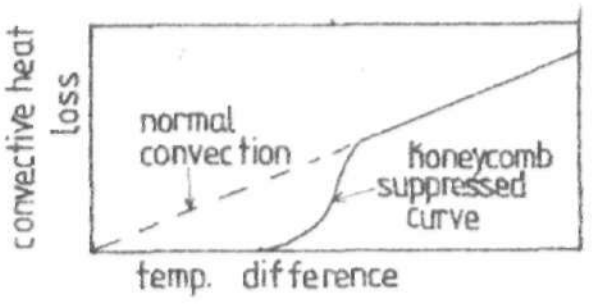
μ = dynamic viscosity.

k = Thermal Coefficient of Conductivity.

x = Length of the channel.

The theoretical minimum value for the onset of Convection is when Grasshoffor Rayleigh number is 1707: The presence of small channels of the mesh effectively increases the value of Gr or Ra for the surface from 1707 to 200,00. Suppression of convection would only be effective for small temprature differences between the top and bottom of the honey comb. If the absorber temprature rose beyond a certain minimum, then the value of q would cause the Ra value to exceed the limit and convection would begin.

The expected variation of the convective heat loss with increasing temprature difference for a honey combs is shown in Figure (A-1). Convection normally grows approximately linearly with temprature $T^{5/4}$ and is shown by dashed line. When the Rayleigh number is sufficently low, and when either the cell spacing of temprature difference is small the convection will be fully suppressed. When Rayleigh Number reaches about 2000, the convection begins and it rises rapidly to reach the normal convection level.



Schematic representation of the effect of convection suppression with honeycombs as a function of increasing temperature difference. The temp. differature at the knee of the curve depends upon the size of the opening of the honeycomb cell.

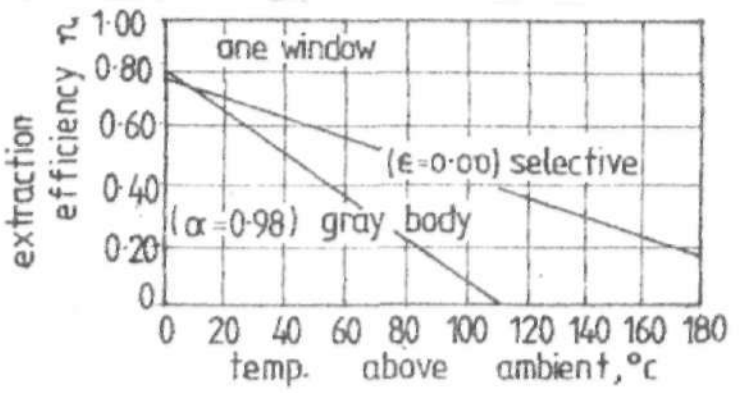
FIG (A-1·1) REF 9

1 (c) The addition of a selective absorber adversly affects the convective exchange between the selective surface and the nearest cover window. The temprature difference is increased and hence the Convection is increased because the window is at a lower temprature than with a black body absorber. The reason for this is simple; less energy is flowing out through the windows. This effect indicates that we need someway of heating the cover next to the selective surface one way of accomplishing this further tuning of the collector would be to inhibit some of the TIR (Thermal Infra Red) emission upward from the inner cover. In other wards one would like to have a selective transmitting

surface placed on top of the cover to raise its temprature and decrease the convective heat exchange with absorber. One might term this addition a selective coating to control convection loss by reducing the temprature difference between surfaces.

2 (a) To explore the effect of selective absorbers, let us assume that there is no emssion from such a surface. In reality there will be at most a few percent emission, perhaps as much as 10%, but to see the best performance we assume no emission. We will, however take into account that selective surfaces donot have as good an absorptance as $\alpha = 0.98$. As a realistic value for the absorptance we will assume the selective surface is a good one and has an absorptance of $\alpha = 0.90$.

Single Window Collector:- The change from a 0.98 gray body to a 0.90 selective absorber also provides a distinct advantage for a collector having a single cover window. The slightly lowered absorptance of the selective surface lowers performance only for small temprature rises above ambient. The change in efficiency caused by the addition of a selective absorber is as shown in Figure (A-1.2)



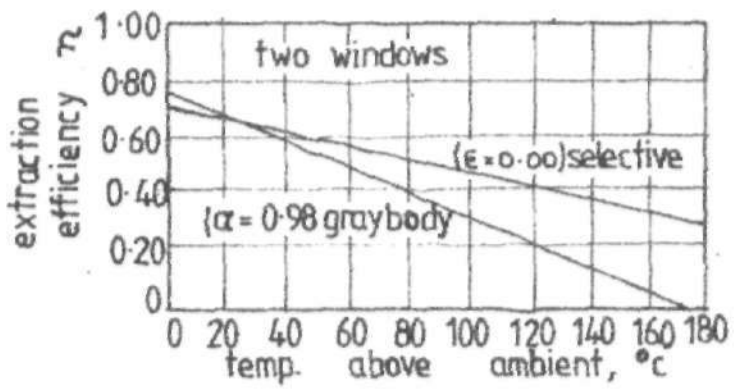
Change in performance for a collector having one window cover when the absorber is changed from an $\epsilon = \alpha = 0.98$ gray body absorber to an $\epsilon = 0.00$ selective absorber having $\alpha = 0.90$, for 10°C ambient temperature.

FIG (A-1·2), REF. 9

Double Window Collector: The addition of a selective absorber to a double cover flat plate collector improves performance as shown in Figure (A-1·3).

Performance is slightly poor at small temprature rises because of lower absorptance of the selective surface but is better at higher temprature rises. The effect when the absorber is not fully selective is between these two curves: a coating with emmittance of 0 ·0 is approximately 10% of the way to the blackbody curve from the selective curve shown.

Because a selective coating adds a significant cost to the absorber, one must operate the collector in a region where the selective curve lies significantly above the black body curve.

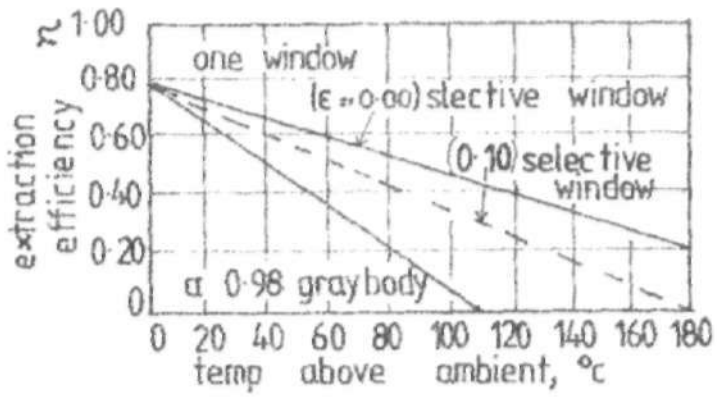


Change in efficiency of extraction η for a fluid collector having two cover windows in going from a 0·98 gray body absorber to a perfect selective absorber.

FIG (A-1·3) REF. 9

2 (b) The simplest case of a selective window is represented by a single window plus an absorber. Let us place a selective coating on the window facing the absorber. Since the window now has high reflectance for TIR, the thermal infrared loss from the absorber will be reflected back to the absorber. Figure (A-1·4) shows how this reflective surface modifies performance, compared to the case of no selective surfaces and a selective absorber. The best transmitting

selective coating is indium oxide plus 7% tin: oxide. The performance Curve for this coating on the inner face of the window is shown by dashed line in Figure (A-14).



Behavior of a collector having a 0·98 gray body absorber with a perfect TIR reflecting coating on the inner face of the window and a 0·90 TIR reflecting coating of indium oxide and tin oxide.

FIG (A-1·4) REF. 9

2 (c) Selective windows give the largest gain when the absorber is a black body but they still add useful performance when the absorber is a perfect selective surface. The role of selective windows has generally been restricted to that of acting as a reflector of Thermal infra red emitted from the absorber, returning that energy otherwise lost into the window. There is a second role of selective windows, that of preventing TIR emission from the upper

sides of window thus controlling the convection between surfaces. Materials like tin oxide (SnO_2) or indium Oxide ($In_2 O_3$) singly or in combination, sometimes with traces of antimony (Sb) can be used.

APPENDIX 2

COST FACTORS INVOLVED IN SOLAR COLLECTOR SYSTEMS

A-2.1 COST LIMITS

Before going in for mass production of solar collectors it is necessary to look into the economics of solar energy conversion. An important factor in further judgements is the unstability of energy Costs with traditional fuels which are driven up by the growing shortages of petroleum. The cost limits may change as fuel costs escalate. One must be cautious when making the simplistic statement that "as the cost of traditional fuel rises, solar energy will become more economic." The basic reason for caution is that the energy costs are quickly reflected in the cost of every produce manufactured with that energy, and in the labour costs incurred through the escalation of wages driven by the need of the worker for a larger wage to permit him to afford the more expensive energy. Recently the cost of basic materials like steel, glass, aluminium has risen about the same rate as the average cost of energy used in the industrial community. The reduction of the cost of solar energy conversion systems is urgent. Design, materials and manufacturing methods all have potentials for cost reductions.

One answer to economic solar energy is to either "do it yourself" and avoid the normal operations of the market place or find new points of leverage for cost reduction at the factory so that solar energy systems can reach the consumer in return for an acceptable expenditure of his annual income.

It should be noted that once solar energy systems are installed "the energy equivalent" cost of the solar power is stabilized while other energy costs could rise by large amounts during the lifetime of the solar collector. This fact encourages us that solar energy can have a significant stabilizing effect on the escalation of energy costs, once a reasonable fraction of any demand section in the economy has elected the solar option. There are two meaningful ways to compare costs. The first is to compare the cost of solar means with the traditional cost for the product. The second way is to compare the cost on a common basis of cost per unit of alternative energy.

A-2.2 SOLAR COLLECTOR COSTS

It is generally conceded that solar energy systems yield useful energy at costs above those of most alternative energy sources. To be effective in the national energy future, solar energy will need to become more competitive on a broader basis, but this will require cost reductions. If we assume a continuation of the present inflationary trend, then apparent prices of solar collector systems will certainly rise. The problem is how to normalize future apparent prices to a common basis.

There is so little commercial experience at hand that predicting future solar collector dosts is difficult. In the absence of an adequate record two techniques can be applied.

(i) The Delphi Process

(ii) Cost Sensitivity Analysis

A-2.2.1 (i) The Delphi Process: The Delphi process is a simple - chnique of surprising accuracy in some situations. It involves the preparation of a set of questions defining precisely what aspect of the future is being assessed. These questions are submitted as a questionnaire to a bread selection of people who are knowledgeable about one or more aspects of the subject, asking for their best guess. For various reasons only a limited

number of people asked will respond. Some refuse to guess. Others know much about the topic but think the questions are inadequately phrased, pointless or unanswerable. The respondents will generally be those who have a definate opinion, either optimistic or pessimistic. The Delphi process is probably most successful when mass influence has a direct influence on the subject being explored. In the case of solar energy costs, this process may be less reliable, the guessing being more a hope than a reality.

A-2.2.2 SENSITIVITY ANALYSIS METHOD:

The second way of estimating future costs is the Sensitivity analysis method. In it one takes the present costs for the existing designs and estimates within the cost breakdown where significant changes could occur. This method is more firmly based on fact but it still involves considerable uncertainity in what changes might occur that could affect the cost elements. In the case of solar energy one is dealing on the whole with a technology the elements of which are quite conventional insofar as thermal applications are concerned. In the estimation of future costs there are several factors influencing costs upward and downward, the interplay between these factors determining the final cost. Volume of production and sales are one of the major factors in downward

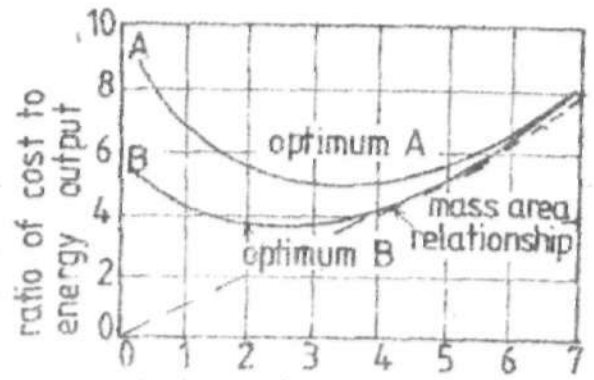
adjustment of cost and prices. The problem then becomes one of estimating the growth of volume. In case of many new products the market does not develop rapidly even though the technology does. People wait for others to test the technology and find out whether it is attractive and reliable. When the technology is finally accepted as being economic and desirable, the volume can rapidly build to higher levels.

In the more distant future a price rise may be forced by the fact that material costs will rise more rapidly than personal income owing to scarcity of natural resources and of fossil energy to process these resources into solar collector systems. The magnitude of the rise is dependent upon the type of materials required in each specific application and the amount of energy required to produce these materials.

A-2-3 DEPENDENCE OF COST OF UNIT SIZE

The cost of solar collectors relative to energy output is dependant on the size of the collector. Generally the ratio of output to the cost will increase as the size decreases, down to a limit where incidental costs having different scaling laws prevent further reduction . This scaling dependence for collectors is basically because the output is proportional to the surface area of collector whereas the cost is proportional to the mass of the collector which in turn is proportional to its volume. It is therefore readily seen than when a given collector is scaled upward in all dimensions, the volume of the material increases as the cube of the relative increase in linear dimensions, while the collector surface area increases as the square. The cost of other items such as labor in assembly, and generally are constant or linear functions. When the collector is made small, there will be an increase in the number of collectors required, to yield a given output. In this case the effect of the non-scaling factors is to increase the cost per unit output because of increase in number of modules. The result is that each collector type will show a curve like those shown in Fig. (A-2-1). Here Curve B differs from Curve A in having a minimum ratio of cost to energy output at a smaller

aperture size, illustrating that different designs will have different optimum aperture sizes. A minimum cost per unit energy output generally will be obtained. The optimum size is determined by analysing the individual cost elements for the design under study and then applying appropriate scaling laws for each component of the final system cost.



DEPENDENCE ON COLLECTOR APERTURE OF THE RATIO OF COST TO ENERGY OUTPUT FOR TWO HYPOTHETICAL DESIGNS, ILLUSTRATING THE EXISTENCE OF AN OPTIMUM SIZE COLLECTOR. DESIGN'B' DIFFERS FROM DESIGN'A'IN HAVING A MINIMUM RATIO OF COST TO ENERGY OUTPUT AT A SMALLER APERTURE SIZE

FIG. (A-24) , REF. (9)

APPENDIX 3

Derivation of Equation (3.1) on Page 54 for Charge required.

Let the solution be 1 kg

Let the rich solution concentration has fraction f_r as NH_3

Let the poor solution concentration has fraction f_p as NH_3

Let A kg of NH_3 to leave when the rich solution changes to poor

	Solution	NH_3
Rich	1 kg	f_r
Poor	1-A	f_p

$\therefore$ Amount of NH_3 in poor solution = $f_p(1 - A)$

Amount of NH_3 in rich solution = f_r

Amount of NH_3 left = A = $f_r - f_p (1 - A)$

$= (f_r - f_p) + f_p \cdot A$

$\therefore$ $A = (f_r - f_p) + f_p \cdot A$

$(A - f_p \cdot A) = f_r - f_p$

$A (1 - f_p) = f_r - f_p$

$$A = \frac{f_r - f_p}{1 - f_p} \left\{ \frac{\text{Amount of } NH_3}{1 \text{ kg of solution}} \right\}$$

Amount of solution/kg of NH_3 liberated = $\dfrac{1 - f_p}{f_r - f_p}$

APPENDIX 4

Derivation of equation (4.6) and (4.7) on Page 66 for θ_t and θ_h

ϕ = Latitude (North Positive)

δ = declination (angular position of the sum at solar noon with respect to the plane of the equator). (north positive)

β = the ange between the horizontal and plane (i.e. the slope)

γ = the surface azimuth angle, i.e. the deviation of the normal to the surface from the local meridian. Zero point being due south, east positive and west negative

ω = hour angle, solar noon being zero.

θ = angle of incidense of beam radiation, the angle being measured between the beam and normal to the plane.

The relation between θ and the other angles is given by.

Various angles involved are shown in Figs (A-4-1), (A-4-2) and (A-4-3)

$$\begin{aligned}\cos\theta = {} & \sin\delta \sin\phi \cos\beta - \sin\delta \cos\phi \sin\beta \cos\gamma \\ & + \cos\delta \cos\phi \cos\beta \cos\omega \\ & + \cos\delta \sin\phi \sin\beta \cos\gamma \cos\omega \\ & + \cos\delta \sin\beta \sin\gamma \sin\omega\end{aligned}$$

For fixed flat plate collectors which face the equator the last term drops out. For horizontal surfaces where $\beta = 0^\circ$ only the first and third terms remain.

$$\therefore \quad \cos\theta = \sin\delta \sin\phi \cos\beta + \cos\delta \cos\phi \cos\beta \cos\omega \quad (x)$$

For angle of Incidence on surfaces sloped to the north or south, surfaces with slope β to the north or south have the same angular relationship to beam radiation as a horizontal surface with an artificial latitude of $(\phi - \beta)$

At $= \beta$ 0° $\quad \text{Cos}\,\beta = 1$

$\therefore$ Equation (x) becomes

$$\text{Cos}\,\theta = \text{Sin}\,\phi\ \text{Sin}\,\delta + \text{Cos}\,\delta\ \text{Cos}\,\phi\ \text{Cos}\,\omega \quad - \quad (y)$$

Now Angle of incidence on sloped surfaces $\text{Cos}\,\theta_T$ is given by

$$\text{Cos}\,\theta_T = \text{Sin}\,(\phi - \beta)\ \text{Sin}\,\delta + \text{Cos}\,(\phi - \beta)\,\text{Cos}\,\delta\ \text{Cos}\,\omega$$

The zenith angle $\text{Cos}\,\theta_h$ is then given by equation (y) i.e.

$$\text{Cos}\,\theta_h = \text{Sin}\,\phi\ \text{Sin}\,\delta + \text{Cos}\,\delta\ \text{Cos}\,\phi\ \text{Cos}\,\omega$$

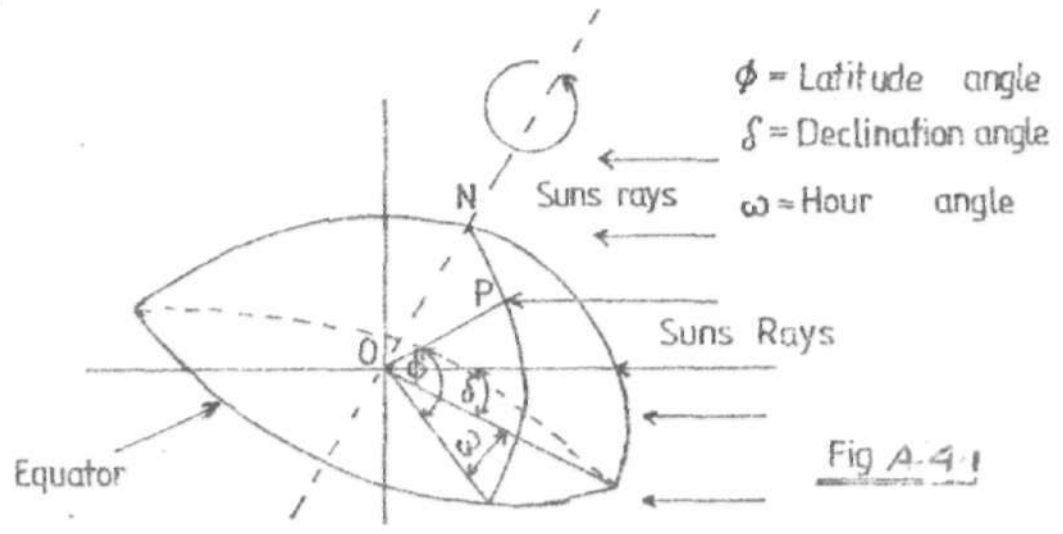


Fig A-4.1

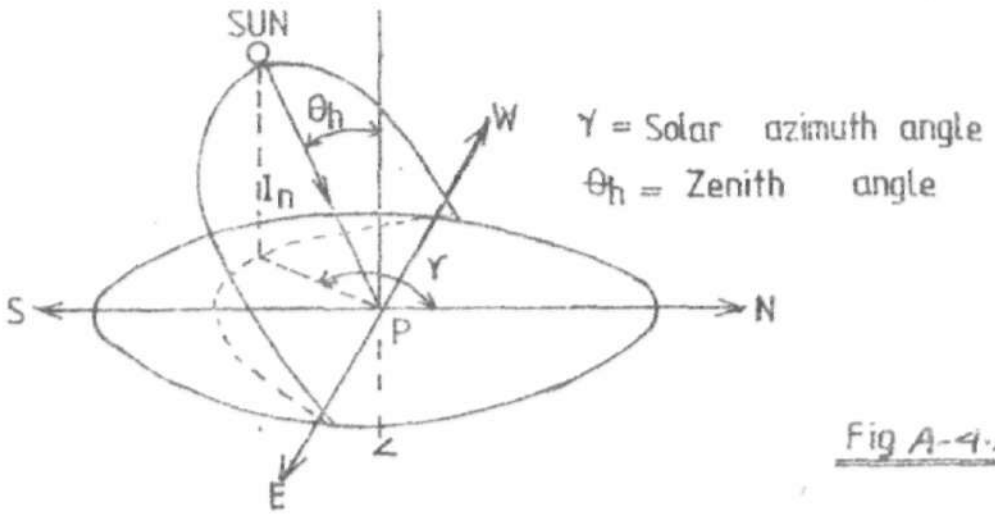


Fig A-4.2

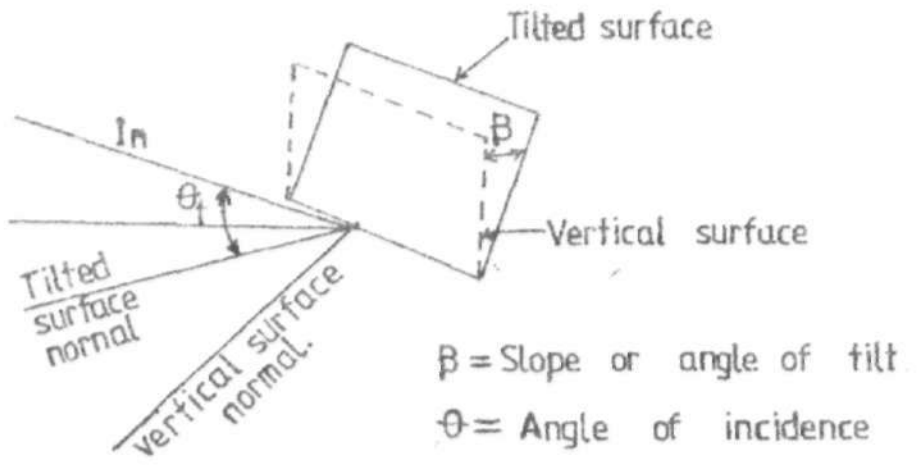


Fig A-4.3

APPENDIX 5

Derivation of total energy Absorbed by the collector (I_t). Eqn 4.10. Page 69

Total energy absorbed by the collector is the sum of two components i.e. direct radiation Intensity (I_{Dt}) and diffuse radiation intensity (I_{dt})

i.e. $I_t = I_{Dt} + I_{dt}$

The direct radiation intensity is given by the expression.

$$I_{Dt} = (\tau\alpha)_D \quad R_D \quad (H - D)$$

The diffuse radiation intensity is given by the expression

$$I_{dt} = (\tau\alpha)_d \quad R_d \, (D)$$

$$\therefore \quad I_t = (\tau\alpha)_D \quad R_D \, (H - D) + (\tau\alpha)_d \, R_d \, (D) \quad - (2)$$

R_D and R_d are the orientation factors for direct and diffuse radiation respectively, and are given by:

$$R_D = \frac{\cos \theta_t}{\cos \theta_h}$$

$$R_d = \frac{1 + \cos \beta}{2}$$

Substituting in equation (2) we have

$$I_t = (\tau\alpha)_D \cdot \frac{\cos \theta_t}{\cos \theta_h} (H - D) + (\tau\alpha)_d \cdot \frac{1 + \cos \beta}{2} \quad (D)$$

Adding for average hourly values of total radiation per unit area absorbed by the collector we have

$$I_t = \bar{H}_h \left\{ (\tau\alpha)_D \quad \frac{\cos \theta_t}{\cos \theta_h} \left\{ 1 - \frac{\bar{D}_h}{\bar{H}_h} \right\} + (\tau\alpha)_d \cdot \left\{ \frac{1 + \cos \beta}{2} \right\} \cdot \frac{\bar{D}_h}{\bar{H}_h} \right\}$$

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