EVALUATION OF SOLAR-POWERED ABSORPTION HEAT PUMP SYSTEMS FOR MULTIPURPOSE APPLICATIONS

BY

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A THESIS SUBMITTED TO THE POSTGRADUATE SCHOOL, AHMADU BELLO UNIVERSITY, IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF SCIENCE IN CHEMICAL ENGINEERING.

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DECLARATION

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

Mumah Sylvester Ndoyi

Date.....

CERTIFICATION

This project report entitled "Evaluation of Solar-Powered Absorption Heat Pump Systems for Multipurpose Applications" by Mumah Sylvester Ndoyi meets the regulation governing the award of Master of Science in Chemical Engineering of Ahmadu Bello University and is approved for its contribution to knowledge and literary pesentation.

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DEDICATION

DEDICATION

TO

THE MUMAH FAMILY, THAT LOVE SHOULD
BE THE ONLY WORD

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ABSTRACT

Theoretical, evaluations of ammonia - water heat pump systems for various applications and the construction and experimental evaluation of absorption heat pump for cooling is presented. The theoretical evaluations include; (i) theoretical evaluation of the various tracking modes as a means of solar energy augmentation for solar flat plate collectors, (ii) the evaluation of performance of solar fiat plate collectors as a energy source for absorption heat pump systems , (iii) application of Chebyshev polynomials for the evaluation of thermodynamic property data equations for ammonia-water mixtures for absorption heat pump systems and (iv) thermodynamic analysis of ammonia-water absorption heat pump systems operating in the following modes; (a) conventional (heating and cooling) (b) reversed and (c) experimental. A 5 kWatt experimental solar-powered absorption heat pump system was constructed using locally available materials and skills. The simulated performance results and experimental performance data for the absorption heat pump system for cooling showed great potential for solar cooling in Nigeria.

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LIST OF SYMBOLS

A	Area (m^2)
CP	Specific Heat J/kg^oC
COP	Coefficient of Performance (Dimensionless)
F'	Fin Efficiency (Dimensionless)
f"	Collector Efficiency Factor(Dimensionless)
f	Collector Flow Factor (Dimensionless)
F_R	Collector Heat Removal Factor (Dimensionless)
FR	Flow Ratio (Dimensionless)
G_{SC}	Solar Constant (W/m^2)
H	Daily Irradiation (W/m^2), Saturated Enthalpy (J/kg)
I	Hourly Irradiation (J/m^2)
K_T	Hourly Clearness Index (Dimensionless)
M	Flowrate (kg/s)
N	Number of Covers
n	Day Number
P	Pressure (Bars)
q	Enthalpy (J/kg)
Q	Energy Per Unit Area (J/m^2); Energy (J)
R_b	Ratio of beam Radiation on Tilted Plane to that in Plane of Measurement (Dimensionless)
R	Reflux Ratio (Dimensionless)

S	Absorbed Solar Energy Per Unit Area (J/m^2), Entropy $J/^oC$
S_T	Slope for Various Tracking Modes (Degrees)
T	Temperature (^oC, K)
TA	Approach Temperature (^oC)
U	Heat Transfer Coefficient W/m^2C
W	Work Input ((J/kg) Equivalence)
X	Concentration (Weight Percent)
α	Absorptance (Dimensionless)
β	Slope (Degrees)
δ	Declination (Degrees)
η	Efficiency (Dimensionless)
θ	Angle (Degrees)
ρ	Reflectance (Dimensionless)
σ	Stefan - Boltzmann Constant ($W/m^2\ K^4$)
ϕ	latitude Angle (Degrees)
ω	Hour Angle (Degrees)
W_s	Sunset (Sunrise) Angle (Degrees)
τ	Transmittance (Dimensionless)

Subscripts

a,aM,AM	Ambient
AAH	- Actual; Reversed heat Pump
AHA	Absorption Heat Pump Actual
ACH	Absorption heat pump, heating

Av	Average
AB	Absorber
AHMO	Absorption Heat Pump, Maximum Obtainable (Heating)
ACMO	Absorption Heat Pump, Maximum Obtainable (cooling)
AHA	Absorption Heat Pump Actual
ACMOE	Absorption Heat Pump, Maximum Obtainable - Expermental System (Cooling)
AHMOE	Absorption Heat Pump, Maximum Obtainable - Experimental System (Heating)
b	Beam, Bottom
c	Collector, Carnot, Cooling
CAH	Carnot - Reversed Absorption heat Pump
CO	Condenser
d	Diffuse
DIS	Disengager
EV	Evaporator
e	Edge
f	Fully
g	Ground
GE	Generator
H	Horizontal, Heating
i	Inlet

L	Total
m	Mean
Max	Maximum
Min	Minimum
o	Extraterrestial, Outlet
p	Plate
J	Tracking, on Tilted plane
t	Top
w	Water
z	Zenith

CHAPTER ONE

INTRODUCTION

1.1 PRIMARY ENERGY SOURCE

The primary source of all energy utilised on the earth, with a few minor exceptions - notably nuclear power, geothermal power and tidal power is radiant solar energy. Fossil fuels represent the accumulation of many million years of solar irradiance transformed by photosynthesis into vegetation. Fossil fuels are non - renewable energy resources. Solar energy is renewable, it cannot be depleted. It is clean, safe, abundantly available and a prime candidate to supplement existing energy sources.

Photosynthesis, the basic process for non-renewable energy resources formation can be represented thus,

$$CO_2 + H_2O \xrightarrow{\text{sunlight}} \text{Organic Material} + O_2$$

The organic material represents stored energy and the conversion efficiency lies in the range of 0.5 to 3 percent (Hall, 1977). This process, even though with its very low conversion efficiency has produced all the non-renewables upon which the world has relied for its energy

requirements. A better ultilisation can be achieved if a better method of conversion (higher efficiency and shorter reaction time for the process) is realised. At present, the world energy consumption is about $3x10^{20}J$, whereas the total solar energy received on the earth surface is approximately 10000 times this value (Hall, 1977). Thus solar energy if properly managed can be able to contribute to every day living confort.

Modern civilisation depends on energy. Thus energy is an important aspect in every development programme. The importance of energy differs for different regions and different countries. For many less developed countries. imported energy is essential for the country survival. For the developing or developed countries,imported energy may mean the difference between economic growth or recession, and possibly, a massive change in lifestyles. Oil exporting developing countries like Nigeria depend on hydrocarbons to build their economies.

1.2 THERMODYNAMICS AND ENERGY

The first law of thermodynamics states that energy is always conserved. While energy is conserved and hence cannot be used up, the usefulness of a quantity of energy does decrease through usage . Energy is degraded through usage , eventually to the point of zero usefulness. Energy is usually compared in terms of potential. The energy of low - utility resources such as exhaust heat of

a thermal power plant has lower potential than high utility sources such as petroleum.

Entropy is a negative measure associated with utility. An increase in entropy corresponds to a decrease in utility . For example the combustion of fossil fuels and the fissioning of nuclear fuels result in ultimate entropy increase. A fraction of the total energy consumed in used for direct heat and hence in the process is degraded to heat with an average temperature approximately ambient. The remaining fraction is used to produce mechanical work which eventually is dissipated as heat; also a degradation process.

Various systems exist where this process of downgrading can be reversed. A system capable of accomplishing this is the heat pump system. In such a process low grade energy (low temperature heat source) is upgraded to high grade energy (high temperature heat source) which can then be used. Solar energy is one of the alternatives for providing the low - temperature heat source.

1.3 SOLAR SYSTEM PERFORMANCE

The performance of solar system depend upon local climatic conditions which varies widely over the year and in different locations of the world. However, animals and materials or processes that have to derive benefit from solar energy systems feel comfortable or perform well

within a relatively limited range of values of air temperatures and humidities. Thus proper solar system performance studies must include the matching of solar radiation availability with space heating or cooling requirements.

1.4 SOLAR ENERGY CONVERSION TECHNIQUES

There are several ways in which solar energy can be converted to suitable energy forms for subsequent end use. They are:

(1) Low - temperature heat production for domestic, industrial and agricultural purpose through the use of thermal collectors (Thermal uses) (Hunn et al, 1977), (Klein et al,1977).

(2) Thermodynamic conversion. (Brinkworth, 1975).

(3) Photovoltaic and solar thermal electricity generation. (Moore, 1976).

(4) Biomass production and conversion into gaseous and liquid fuels (Leboff, 1976).

(5) Solar energy assisted hydrogen generation .

Only the first two will be considered based on their extensive research backing (Duffie and Beckman, 1980), (Wilbur, 1977).

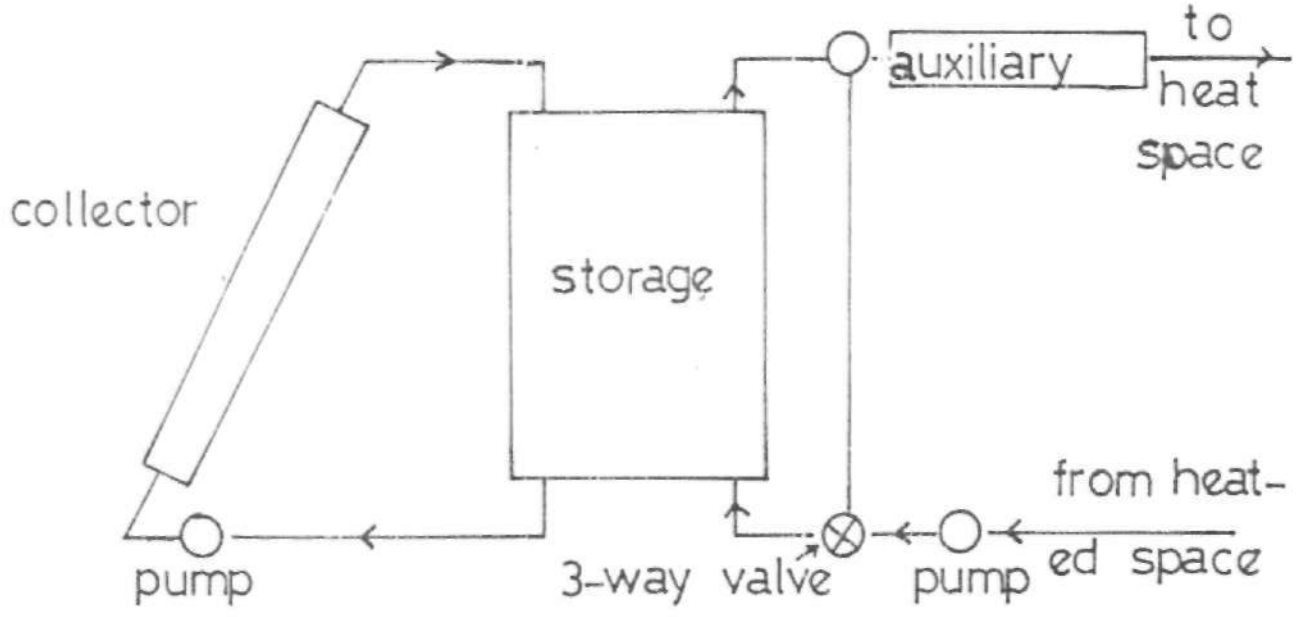


FIGURE 1.1 BASIC HOT LIQUID FLUID SYSTEM

1.4.1 Thermal Uses

Low temperature heat production in the range below 100^{o}C has so far found wider applications than other methods of solar energy conversion, because of its high conversion efficiency and its suitability to substitute for increasingly expensive oil and natural gas. Within the low temperature applications one can distinguish between those for fluid heating and those for space heating. Heating fluids involve using a collector, which basically consists of a dark absorber and a glass top. When the absorber surface is struck by sunlight, it warms up and heat is trapped by the collector. Fluid pumped through the hot collector is warmed and can then be used or stored until needed. Space heating directly with solar energy involves employing the traditional elements of building such as glazed areas and heavy walls to collect, store and distribute the heat by means of the natural processes of conduction, convection and radiation. For continuous operation only natural gas systems do not usually include storage facilities at the point of use. Usually an auxiliary source of heat is included since sunshine availability is at its lowest when heat demand is at its highest.

Figure 1.1 is a schematic of a basic water heating system with water tank storage and auxiliary energy source. The system allows independent control of the solar collector - storage part of the system on the one

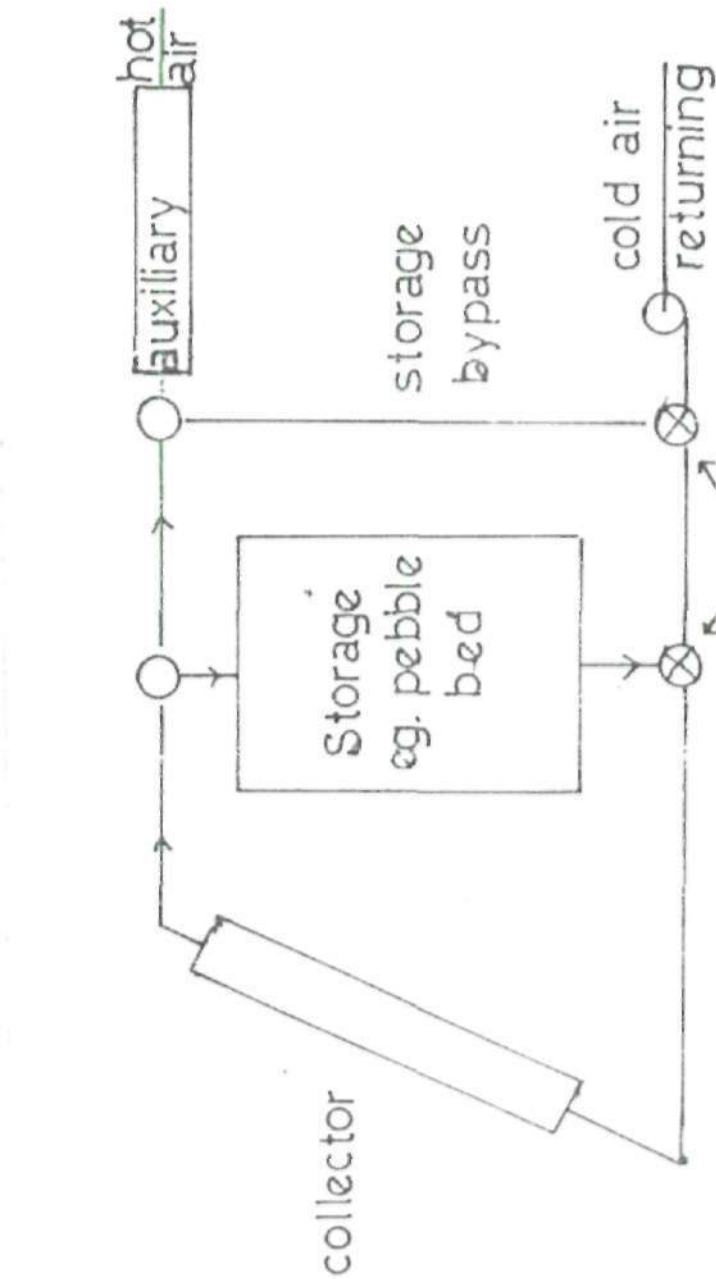


FIGURE 1.2 BASIC HOT AIR SYSTEM

hand and storage auxiliary - load part of the system on the other, as solar heated water can be added to storage at the same time that hot water is removed from storage to meet building loads.

Figure 1.2 shown a schematic of a basic heating system using air as the heat transfer fluid. As can be seen from the arrangement, it is not practical to combine the process of adding energy to and removing energy from storage at the same time.

One of the limitations of the direct thermal system so far described is the limited temperature range over which they operate, which is ambient temperature to the temperature which can be achieved by the collector. A lot of processes need temperature ranges above and below this range. At the industrial scale very high temperature of 300 ^{o} C and above are usually encountered, but flat plate collectors scarcely achieve exit temperature above 120^{o}C. In the tropics, ambient temperature rise above 35 ^{o} C in some areas, a temperature which is not condusive for human comfort. Agricultural materials need storage space sometimes below O ^{o} C. Temperatures higher than those normally achieved in flat plate collectors can be achieved using various solar augmentation methods which include, the use of concentrating devices and evacuated collectors. However, this means additional cost and usually a substantial increase in temperature is not normally achieved.

Lower temperature can be achieved by the addition of evaporative cooling system, but scarely less than 7 ^{o} C reduction in temperature is usually accomplished (Bugage, 1984), (Dubey, 1978) (Hay and Yellot, 1970).

Higher and lower temperature out of the range discussed above can be reliably achieved if the direct thermal system is coupled with a thermodynamic conversion system.

1.4.2 Thermodynamic Conversion

Thermodynamic changes in fluids induced by energy from the sun through direct thermal means can be used to achieve heating or cooling effect outside the range normally achieved by direct thermal source. However such fluids used must have certain intrinsic properties to make them applicable to such systems. Such systems usually operate in cycles so that the fluids are not wasted. Systems that use the solar heat to produce temperatures above the temperature of the solar heat are called solar heat pump systems. Those that use the solar heat to produce temperatures below ambient are usually called refrigerating systems. The cycles and principles in both systems are the same, and thus in this project, the word "heat pump" will be used to denote both.

Two principal types of both systems exist based on the type of drive for the systems. They are the vapour compression system and the absorption system.

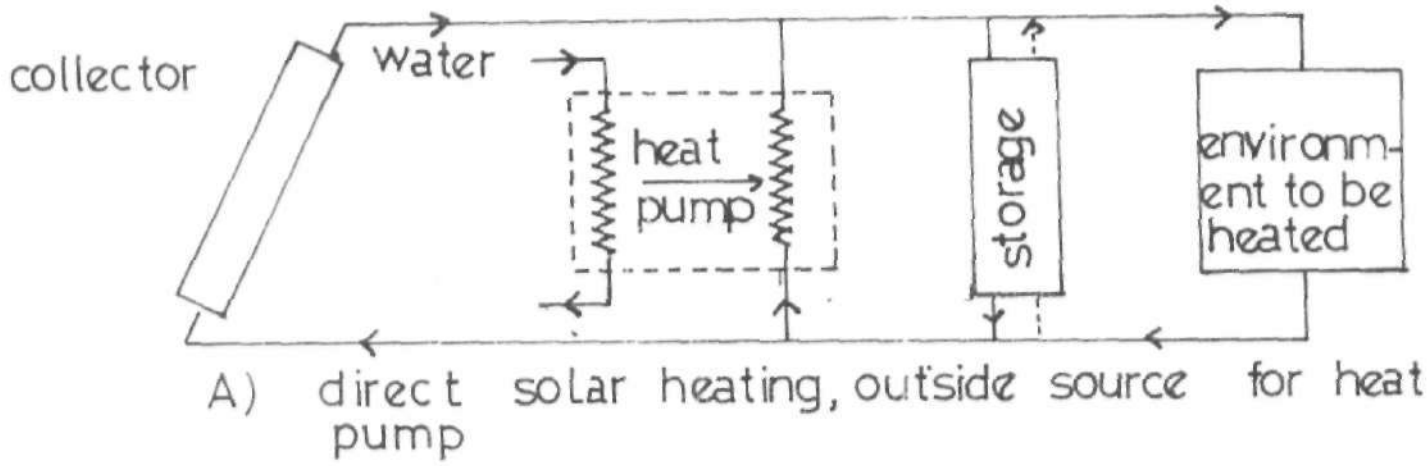
In the vapour compression system, the heating or cooling effect is produced with the help of a compressor. In the absorption system the mechanical process of compression performed by the compressor is replaced by a "thermodynamic pump", which includes a thermophysical reaction between the working fluid and a second medium known as an absorbent in which the working fluid is soluble (see Appendix A for details).

The vapour compression system requires a compressor which is normally very expensive and has a large power comsumption. The absorption system requires a pump which is not as expensive as a compressor and the power comsumption is low. Some absorption systems can perform satisfactorily with the exclusion of a mechanical pump (Reay and Machmicheal, 1979).

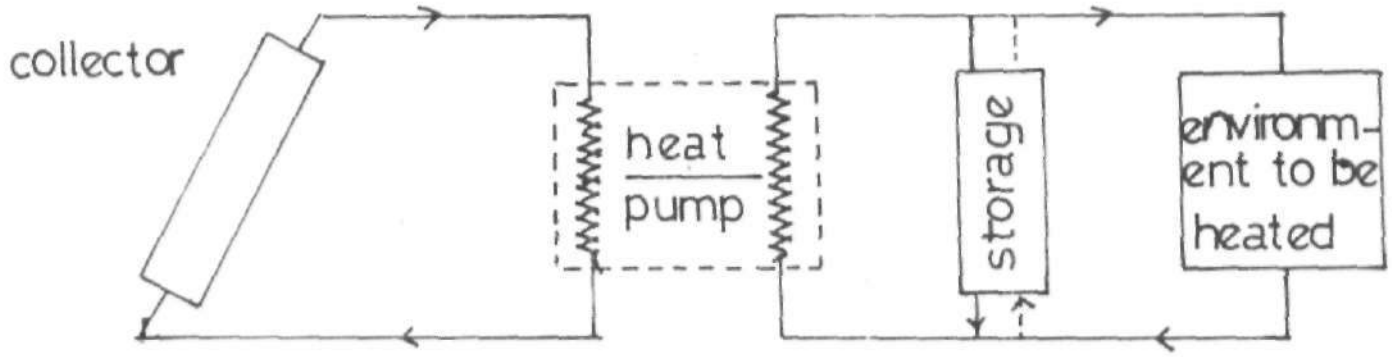
1.5 APPLICATION OF HEAT PUMPS

1.5.1 Heating

Heat pumps for heating purposes are normally applied where high temperatures are needed. Normal domestic heating systems using direct heating systems can effectively provide the heating needs in domestic residences. However such systems can not be effectively applied in industrial establishments. Thermal energy is by far the largest fraction of the total energy consumed in the industry. The thermal energy used in industrial processes can be broadly broken down into application of

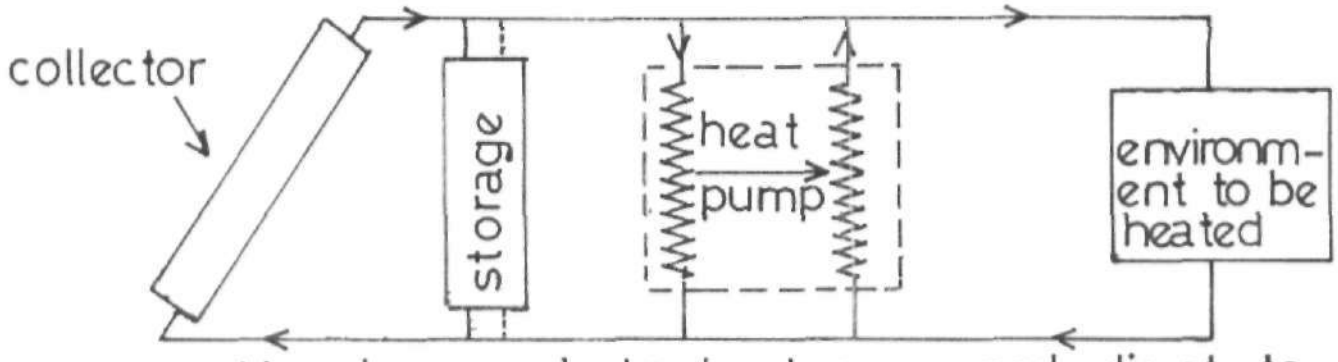


A) direct solar heating, outside source for heat pump

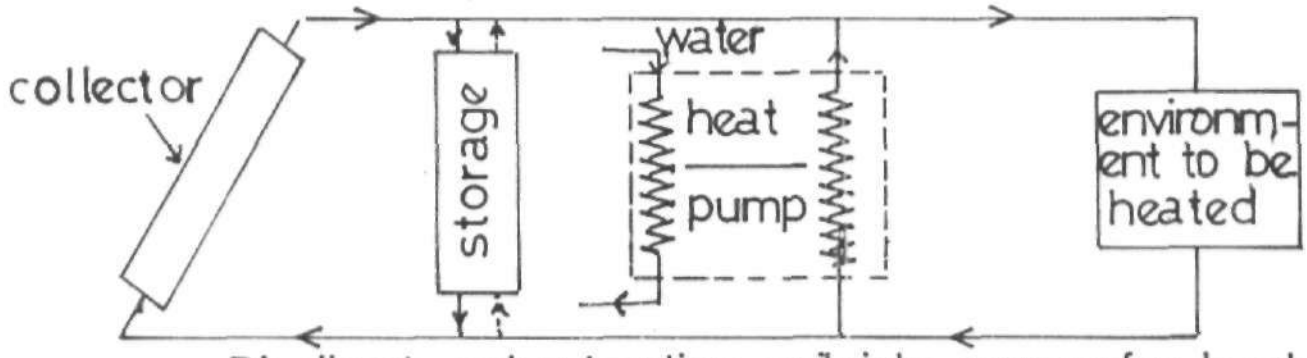


B) solar supply to heat pump only

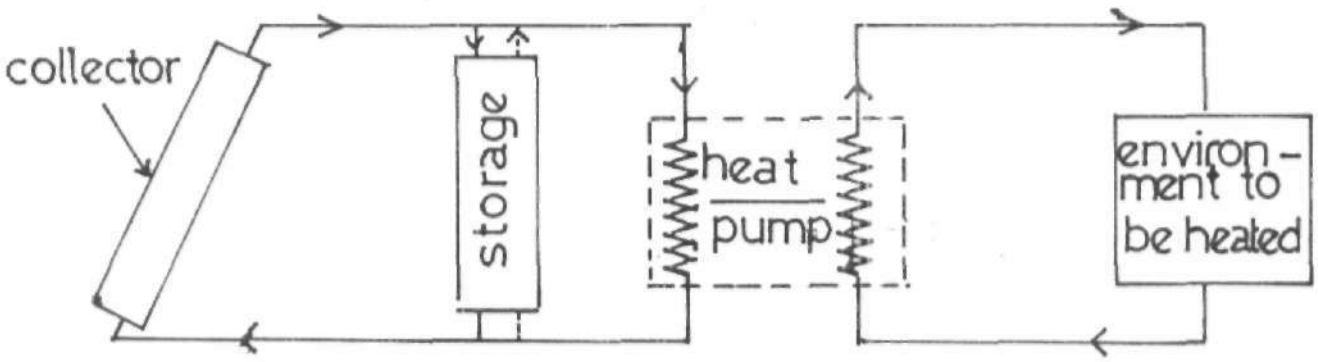
FIGURE1.3 SOLAR COLLECTOR HEAT PUMP SYSTEM(HIGH SIDE STORAGE)



A) solar supply to heat pump and direct to environment



B) direct solar heating, outside source for heat pump



C) solar supply to heat pump only

FIGURE 1.4 SOLAR COLLECTOR HEAT PUMP SYSTEMS(LOW SIDE STORAGE)

hot air, hot water, and steam. In terms of the temperature range, a very substantial amount of the thermal energy is used at temperatures below 300°C. For temperatures in this vicinity, it appears that solar technology in conjuntion with heat pump technology has developed sufficiently to allow the replacement of conventional fuels by solar energy. Particularly applicable in this case is the solar powered reversed heat pump systems which has the unique ability of raising the temperature of a working fluid above that of the input source. This it is apparent that the contribution by solar energy would mean a significant saving in the overall energy consumption system.

There are two methods of operating the solar collector - heat pump combination. One is the arrangement having low-temperature storage, that is, solar heat is stored prior to its supply to the heat pump. Another method involves operation of the heat pump whenever solar heat is available, regardless of whether the environment to be heated needs heat at the time. The resulting higher - temperature heat can then be stored as heated water or any adequate fluid, sensible heat in gravel or as heat of fusion in chemicals. This storage can then be called upon as needed. Various configurations of the two methods are shown in Figure 1.3 , 1.4A, 1.4B and 1.4C.

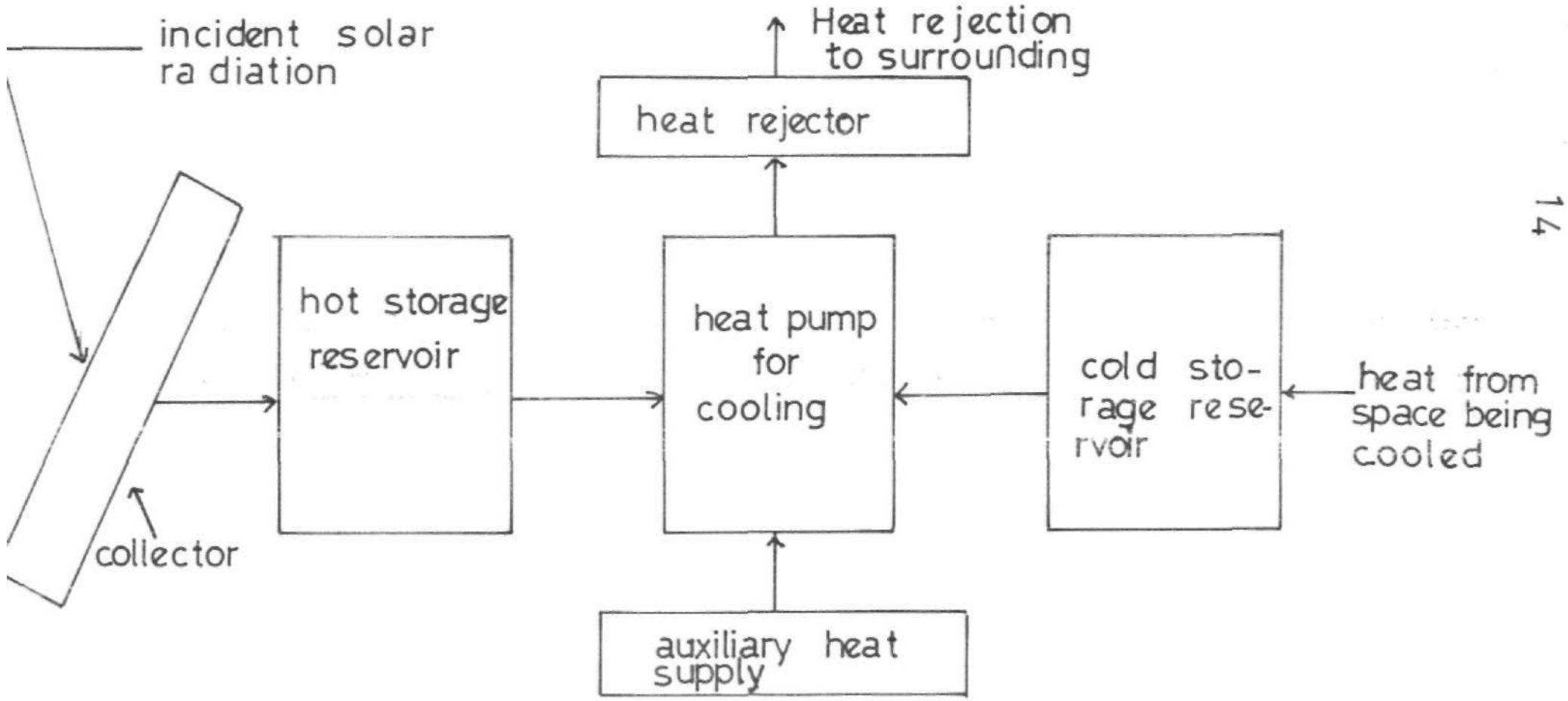


FIGURE 1.5 HEAT FLOW DIAGRAM FOR SOLAR-POWERED COOLING SYSTEM

1.5.2 Cooling

There is a considerable need in the development of solar - powered heat pump systems for cold storage applications particularly in rural areas. In Nigeria and other tropical countries, storage facilities for agricultural product, and other perishables are grossly inadequate to meet the present demand. A lot is lost due to lack of suitable storage facilities. Moreover food production centres are usually situated in remote places where electric power supply for running large heat pump plants are not available. The primary environmental problem is that of excessive heat most of the time which results in human discomfort. Thus cooling with solar energy, which is a free and renewable source and in any case is the cause of the problem requiring investigation naturally povide an effective solution to these problems.

A solar cooling system comprises a solar heating system and some type of cooling subsystem. An energy flow schematic for a solar - powered cooling system is shown in Figure 1.5. Energy is received from the sun by the collector and transferred through some high temperature energy storage reservoir to the cooling subsystem. This subsystem takes heat from a cold storage reservoir which in turn removes heat from the space to be cooled and the two streams of heat, from hot storage and from cold storage, are then passed out through a heat rejector to the environment. In the event the hot storage and/or cold

storage are depleted and the sun is not shining, energy may be supplied by an auxillary heat source to effect cooling.

Of the solar - powered heat pump systems for cooling alternatives, two stand out as most promising at the present state of development. They are (1) the absorption heat pump (Anon, 1972) and (2) a conventional heat engine driving a vapour compression heat pump (Anon, 1972). Other cooling concepts which employ the adsorption principle (Lunde 1974), (Rush, 1974), (Johnson, 1967) (Kopdzhanyan, 1967), the jet ejector (Anon, 1972), (Cavallini. 1967), (Kakabaev and Davletov, 1966) the Rankine - reverse Brayton cycle principle of operation (Anon, 1972) are insufficient to be competitive at the present time.

The fundamental limit on cooling system performance is set by the second law of thermodynamics. The system which will achieve the optimum performance, is a Carnot heat engine driving a Carnot heat pump system. The performance of such a system utilising infinitely conducting heat exchangers is determined by the temperature at which heat is supplied to the heat engine, the temperature of the space being cooled, and the available ambient temperature to which heat is being rejected.

1.6 RESEARCH OBJECTIVE

There is a limited base of research findings upon which the practical application of solar energy for heating and cooling can be based. With this is coupled the limitation of the capital that is needed to harness solar energy. With the rising cost of fuels, low incomes of the masses, and the inherent problems caused by these; deforestation with the consequent desertification of the land, lack of storage facilities for agricultural products and discomfort caused by intense heat, it is imperative that if things are to improve for the population, research should be geared and amplified towards a cheaper source of energy. Solar energy is not cheap but free and abundantly avaliable. Thus it is the aim of this project to carry out a theoretical evaluation of solar - powered heat pump systems which will act as a directory to what might be expected in the practical system. The absorption system is chosen because of its overall low running cost when compared with the vapour compression system.

The ammonia - water combination is chosen as the working fluid - absorbent combination because of its inherent advantages over other combinations when applied to solar-powered heat pump systems. The practical system will be constructed to operate in the cooling mode although there are plans to operate it in the heating mode in the future.

No storage facility will be included at this stage of

the project but it is expected to be included in future.

CHAPTER TWO

AUGMENTATION AND PERFORMANCE STUDIES FOR SOLAR FLAT PLATE COLLECTORS

2.1 INTRODUCTION

The key component in solar heating and cooling systems is the solar collector. Of this, the most widely used are the flat plate solar collectors. These collectors are suitable for low temperature application and usually are operated in a fixed position oriented in a north-south direction in the northern hemisphere tilted at an angle usually equal to latitude angle. The collector performance is usually expressed in terms of the useful energy gain, which in effect is the difference between the heat absorbed by the collector and losses from the collector. To date the major thrust in collector performance improvement has been in reducing the front and rear losses from the collector. Several methods have been proposed and used to achieve the front losses reduction viz:

a) The use of selective surfaces and heat reflecting glass (For example, black chrome) (Hollands, 1976).

b) The insertion of a honeycomb between the collector plate and the glass cover which

suppresses free convective heat loss. Most honeycombs used now are transparent to solar rays (for example, glass, plastic). Usualy the honeycomb material is opaque to long wave radiation, thereby giving suppression of the radiant loss as well, so that a selective surface may not be necessary.

c) The use of evacuated collectors. In evacuated collectors the free convective and gaseous conduction heat losses are eliminated. A selective surface (or heat reflecting glass) is used to reduce the radiative loss and the result is a collector with a very low heat loss coefficient and corressponding high efficiency.

d) The use of concentrating collectors. The philosophy behind the concentrating collector is to concentrate the solar rays onto a smaller area. Since the heat losses are proportional to the collector area, they are reduced, with a resultant high efficiency.

e) The use of V-Groove concentrators (Hollands, 1971), compound parabolic concentrator (CPC) (Winston, 1974), fresnel collectors (Nelson, 1975) and planar reflectors (Grassie, 1967).

Effective design and utilisation of solar collectors

depend to a large extent on the knowledge of the insolation characteristics in the region in which the collectors are to be located. In general, such characteristics are either obtained directly from measured horizontal solar radiation data collected over a long period of time, or in locations where measurement data are not available or insufficient, the characteristics are derived from other meteorological measurements by using empirical relationships established elsewhere. For Zaria, Jos, Sokoto, Kano and Kaduna locations measured solar radiation data exist (Arinze, 1984)

2.2 AUGMENTATION OF ENERGY COLLECTION IN FLAT PLATE SOLAR COLLECTOR BY SUN TRACKING FOR VARIOUS NORTHERN LOCATIONS.

A method that has been receiving attention in recent times for increasing the output of the flat plate solar collector is that of making the flat plate collector track the Sun (Drago, 1978). However tracking collectors will however entail problems with regard to fluid - pipe connections and hence the most appropriate fluid for such collectors appears to be air since slight leakages can be tolerated and pressures are low.

For the rays to be normal to the surface requires that the collector follows the sun both in the East-West and the north-south direction continuously. This type of

collector is known as the fully tracking flat plate collector. A resticted form of tracking, called semi-tracking, may however be obtained by constrainina the collector to rotate about a north-south axis,with the collector inclined to the horizontal at an appropriate slope. Two cases may be distinguished; one when the normal to the collector rotates at a uniform speed of 15^{o} per hour and the other when the normal to the collector is always in the solar azimuthal plane. The latter case, though better, requires that the collector be rotated at rotational speed which varies over the day. This may entails much instrumentation. The last type of semi-tracking is by constraining the collector to rotate about a east-west axis, which itself is inclined to the horizontal at an appropriate slope usually 0^{o}. This is usually done on a monthly basis.

The various types of solar tracking methods will be investigated for five northern location in Nigeria namely; Zaria, Jos, Kano, Kaduna and Sokoto.

2.2.1 Analysis of Tracking Collectors.

Flat plate collectors absorb beam radiation, diffuse radiation and solar radiation reflected from the ground. In order to calculate the maximum absorbed radiation for tracking flat plate collectors, we need to relate the radiation incident on the collector to the total radiation

on a horizontal surface. This is normally represented by the angular correction factor R. The zenith angle of the sun and the angle of incidence of the beam radiation for a flat plate collector can be calculated using the relationships given by Duffie and Beckman (1980). For a fully tracking flat plate collector, the slope of the collector must be equal to the complement of the Sun's altitude angle, and the collector azimuth angle must be equal to the Sun's azimuth angle. For the fully tracking collector, the slope is given by (Duffie and Beckman, 1980).

$$S_T = \cos^{-1} (\sin \delta \sin \phi + \cos \delta \cos \phi \cos \omega) \quad (2.1)$$

The surface arimuth angle is given by (Duffie and Beckman,1980)

$$V_f = \sin^{-1} (\cos \delta \sin \omega / \sin S_T) \quad (2.2)$$

The angle of incidence for the tilted tracking collector is given by (Duffie and Beckman, 1980)

$$\begin{aligned} \text{Cos } \theta_f = {} & \text{Sin } \delta \sin \phi \cos S_T - \text{Sin } \delta \cos \phi \sin S_f \cos V_f \\ & + \cos \delta \cos \phi \cos S_T \cos \omega + \\ & \cos \delta \sin \phi \text{ Sin } S_T \text{ Cos } V_f \text{ Cos } \omega + \\ & \cos \delta \sin S_T \sin V_f \text{ Sin } \omega \end{aligned} \quad (2.3)$$

The angle of incidence for the horizontal surface is obtained by substituting $S_T = 0$ in Equation (3)

$$\cos \theta_z = \text{Sin } \delta \text{ Sin } \phi + \cos \delta \cos \phi \cos \omega \quad (2.4)$$

The angular correction for hourly radiation on the horizontal surface to radiation on a tilted fully tracking flat plate collector is given by

$$R_b = \text{Cos } \theta_f / \text{Cos } \theta_z$$

The procedure outlined by Lui and Jordan (1963) is used to evaluate the conversion factor for diffuse radiation and the reflected radiation from the surroundings. In this method the assumption made is that the diffuse solar radiation is uniformly distributed over the Sky. In a method proposed by Puri et al (1980), the Sky diffuse solar radiation is divided into isotropic and non-isotropic components depending on the cloud cover. Variations of calculated results from the two methods are very small. The method of Liu and Jordan is used in this project.

A surface tilted at slope β from the horizontal has a view factor to the sky given by $(1 + \cos \beta)/2$. The surface has a view factor to the ground of $(1 - \cos \beta)/2$ and if the surrounding has a reflectance of ρ for the total solar radiation, the reflected radiation from the surrounding on the surface from the solar radiation is $(I_d + I_b) \rho (1 - \cos \beta)/2$. Combining the three components, the total solar

radiation on a tilted surface is given by

$$I_T = I_b R_b + I_d(1+\cos\beta)/2 + (I_b+I_d)\rho(1-\cos\beta)/2 \quad \ldots\ldots\ldots \quad (2.6)$$

For no Snow cover Lui and Jordan suggested a ground reflectance of 0.2

To calculate the diffuse component from the daily total radiation, the Collares-Pereira and Rabl correlation (Duffie and Beckman, 1980) is used. An equations representing this correlation is

$$\frac{H_d}{H} \begin{cases} = 0.99 & \text{For } K_T \leq 0.17 \\ = 1.188 - 2.272K_T \times 9.473K_T^2 & \\ & \text{For } 0.17 < K_T < 0.75 \\ - 22.865\ K_T^3 \times 14.648K_T^4 & \\ = -0.54K_T \times 0.632 & \text{For } 0.75 < K_T < 0.8 \\ = 0.2 & \text{for } K_T \geq 0.80 \end{cases}$$

(2.7)

There hourly total radiation is calculated from the Collares - Pereira and Rabl (1979) correlation.

$$r_t = I_T / H = (a+b\cos\omega)\ \frac{\cos\omega - \cos\omega_s}{\sin\omega_s - (2\pi\omega_s/360)\cos\omega_s} \quad (2.3)$$

where

$$a = 0.409 \times 0.5016 \sin(W_s - 60)$$

$$b = 0.6609 - 0.4767 \sin(W_s - 60)$$

$$K_T = H/H_o \quad (2.9)$$

The extraterrestrial radiation on the horizontal surface H_o is given by

$$H_o = \frac{24\times3600}{\Pi} G_{sc} \left|1+0.033 \cos\right| \frac{(360n)}{365} \times$$

$$\text{Cos } \phi \cos \delta \text{ Sin } \omega_s \frac{+2\Pi\omega_s}{360} \text{ Sin } \phi \text{Sin } \delta \qquad (2.10)$$

The Sun-set hour angle ω_s = is given by

$$\text{Cos } \omega_s = \tan \phi \tan \delta \qquad (2.11)$$

and the declination of the Sun is given by

$$\delta = 23.45 \text{ Sin } (360 \frac{284+n)}{365} \qquad (2.12)$$

The hourly diffuse radiation is calculated from the relationship

$$r_d = I_d/H_d = \frac{\Pi}{24} \frac{\text{Cos } \omega - \text{Cos } \omega_s}{\text{Sin } \omega_s - (2\Pi\omega_s/360) \text{ Cos } \omega_s} \qquad (2.13)$$

The monthly average daily radiation on a horizontal surface for the northern locations of Zaria, Jos, Sokoto, Kano, and Kaduna has been presented by Arinze (1984) and tabulated on Table 2.1

TABLE 2.1 SOLAR RADIATION DATA FOR DIFFERENT NORTHERN NIGERIA LUCATIONS

Month	Monthly Average Daily Horizontal Radiation H (MJ/m^2 day)				
	Zaria	Kaduna	Kano	Sokoto	Jos
January	23.89	24.79	25.32	25.70	25.05
Febrary	25.17	25.85	27.55	26.86	26.70
March	25.69	25.49	28.67	27.00	25.45
April	25.68	25.17	29.18	27.86	25.00
May	24.33	24.49	28.09	26.08	21.45
June	23.15	23.91	27.21	28.10	21.38
July	26.67	19.91	25.05	22.85	18.90
August	20.82	20.10	26.41	21.00	18.90
September	23.25	23.45	28.06	24.76	22.21
October	24.11	23.85	27.06	27.42	23.17
November	23.63	24.56	25.34	26.21	25.61
December	22.60	21.59	24.02	24.53	25.07

A computer programme was written using Equations (2.1) to (2.12) to calculate the total solar radiation on the tracking solar flat plate collectors. This was carried out for all months of the year.

TABLE 2.2 SLOPE FOR MAXIMUM ENERGY GAIN FOR ZARIA

(TO NEAREST 2.5°)

Month	Fixed Collector	Variable Speed Semi-Tracking	Constant Speed Semi-Tracking
January	45	57.5	57.5
February	30	50	50
March	15	45	42.5
April	-2.5	40	35
May	- 15	32.5	25.0
June	-17.5	25	20
July	- 15	25	17.5
August	-7.5	30	22.5
September	7.5	40	37.5
October	25.0	47.5	45.0
November	37.5	52.5	52.5
December	40.0	55	55

TABLE 2.3 SLOPE FOR MAXIMUM ENERGY GAIN FOR KANO LOCATION (TO NEAREST 2.5^{O})

Month	Fixed Collector	Variable Speed Semi-Tracking	Constant Speed Semi-Tracking
January	45	57.5	57.5
February	32.5	55	52.5
March	15	45	42.5
April	- 2.5	42.5	35.0
May	-15	37.5	27.5
June	-20	32.5	25
July	-17.5	32.5	25
August	- 7.5	40.0	32.5
September	7.5	45	40
October	2.5	47.5	47.5
November	3.75	52.5	52.5
December	42.5	55	52.5

TABLE 2.4 SLOPES FOR MAXIMUM ENERGY GAIN FOR KADUNA LOCATION (TO NEAREST 2.5°)

Month	Fixed Collector	Variable Speed Semi-Tracking	Fixed Speed Semi Tracking
January	42.5	57.5	57.5
February	30.0	50.0	47.5
March	12.5	45	42.5
April	- 2.5	40	32.5
May	-15.0	32.5	25.0
June	-20.0	27.5	20.0
July	-15.0	22.5	15.0
August	- 7.5	27.5	20.0
September	5	40	35.0
October	22.5	47.5	45.0
November	35.0	52.5	50.0
December	40.0	55	52.5

TABLE 2.5 SLOPES FOR MAXIMUM ENERGY GAIN FOR JOS LOCATION (TO NEARST 2.5^{O})

Month	Fixed Collector	Variable Speed Semi-Tracking	Constant Speed Semi-Tracking
January	42.5	57.5	55.
February	27.5	47.5	47.5
March	12.5	45	40.0
April	- 5	40	32.5
May	-12.5	27.5	20.0
June	-17.5	22.5	15.0
July	-12.5	20.0	12.5
August	- 5.0	25.0	17.5
September	5.0	37.5	32.5
October	22.5	45	45
November	35.0	50	50
December	37.5	50	50

TABLE 2.6 SLOPES FOR MAXIMUM ENERGY GAIN FOR SOKOTO LOCATION (TO NEAREST 2.5^{c})

Month	Fixed Collector	Variable speed Semi-Tracking	Fixed Speed Semi Tracking
January	45.0	57.5	57.5
February	35	55.0	55.0
March	15	45.0	42.5
April	- 2.5	42.5	37.5
May	-12.5	35.0	27.5
June	-20.0	32.5	25.0
July	- 7.5	30.0	22.5
August	- 5.0	32.5	25.0
September	10.0	42.5	40.0
October	25.0	47.5	47.5
November	37.5	52.5	52.5
December	42.5	55	52.5

TABLE 2.7 SLOPES MAXIMUM USEFUL ENERGY GAIN FOR VARIOUS LOCATIONS.

Location	Optimum Slope For Yearly Operation		
	Fixed collector	Variable Speed semi-Tracking	Constant Speed Semi-Tracking
Zaria	15	45	40
Kano	16.5	45	42.5
Sokoto	17.5	50	45
Jos	17.5	42.5	40
Kaduna	15	45	40

TABLE 2.8 TOTAL ANNUAL ENERGY FOR FULLY TRACKING COLLECTOR AND FOR THE VARIOUS LOCATIONS

Location	Total Annual Energy For Fully Tracking Collector GJ/m^2
Zaria	10.455
Kano	12.922
Sokoto	13.096
Jos	10.688
Kaduna	10.691

TABLE 2.9 TOTAL ANNUAL ENERGY FOR DIFFERENT ORIENTATIONS AND FOR VARIOUS LOCATIONS.

	Total Annual Energy GJ/m^2					
Location	fixed Position		Variable Speed Semi-Tracking		Fixed Speed Semi-Tracking	
	Fixed At Optimum Monthly Slope	Fixed At Optimum Slope for The Year	Fixed At Optimum Monthly Slope	Fixed At Optimum slope for The Year	Fixed At Optimum Monthly slope	Fixed At Optimum Slope for The Year
Zaria	8.495	7.932	10.226	10.075	9.85	9.703
Kano	9.673	8.990	12.05	11.890	11.589	11.390
Sokoto	9.208	8.684	11.459	11.324	11.09	10.90
Jos	8.218	7.811	9.823	9.680	9.555	9.326
Kaduna	8.3949	7.912	10.150	10.075	9.841	9.703

TABLE 2.10 PERCENTAGE INCREASE OVER FIXED POSITION COLLECTOR FOR DIFFERENT LOCATIONS.

	Percentage Increase Over Fixed Position Collector Fixed At Optimum slope Of The Year					
Location	Fixed Collector Monthly Variation	Fully Tracking	Variable Monthly Slope Variation	Speed Fixed forThe Year	Fixed Monthly Slope Variation	Speed Fixed For The Year
Zaria	7	31.8	29	27.0	24.18	22
Kano	7.6	43.7	34.03	32.25	29.0	26
Sokoto	6.0	39.2	31.9	30.4	27.7	25.5
Jos	5.2	36.8	25.7	23.92	22.3	19.6
Kaduna	6.1	35.0	28.28	27.34	24.4	22.6

For the variable speed Semi - tracking flat plate collector, the surface azimuth angle is calculated using Equation (2.2) for a given fixed slope. For the constant

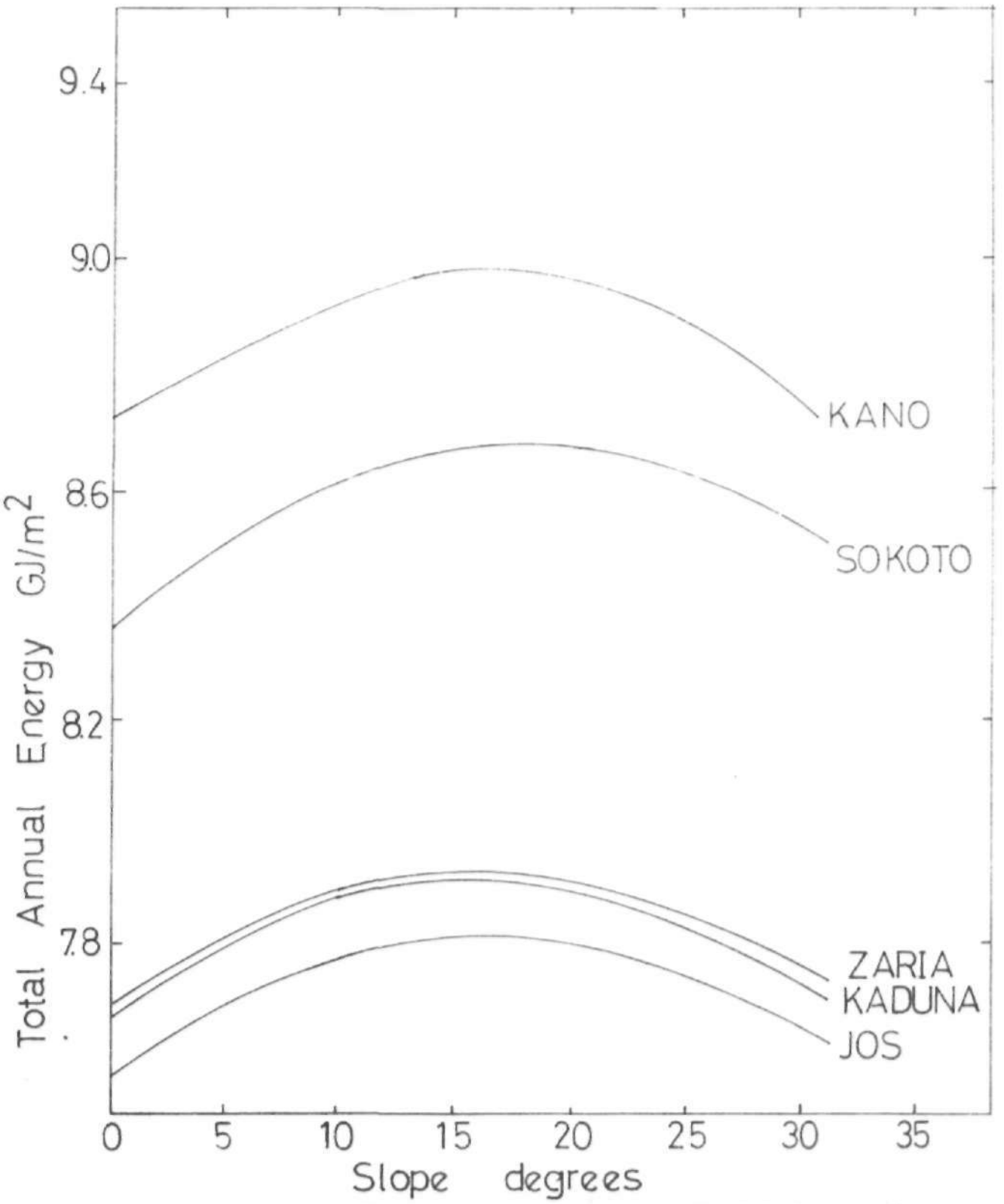


FIGURE 2.1 TOTAL ANNUAL ENERGY AS A FUNCTION OF SLOPE FOR FIXED POSITION

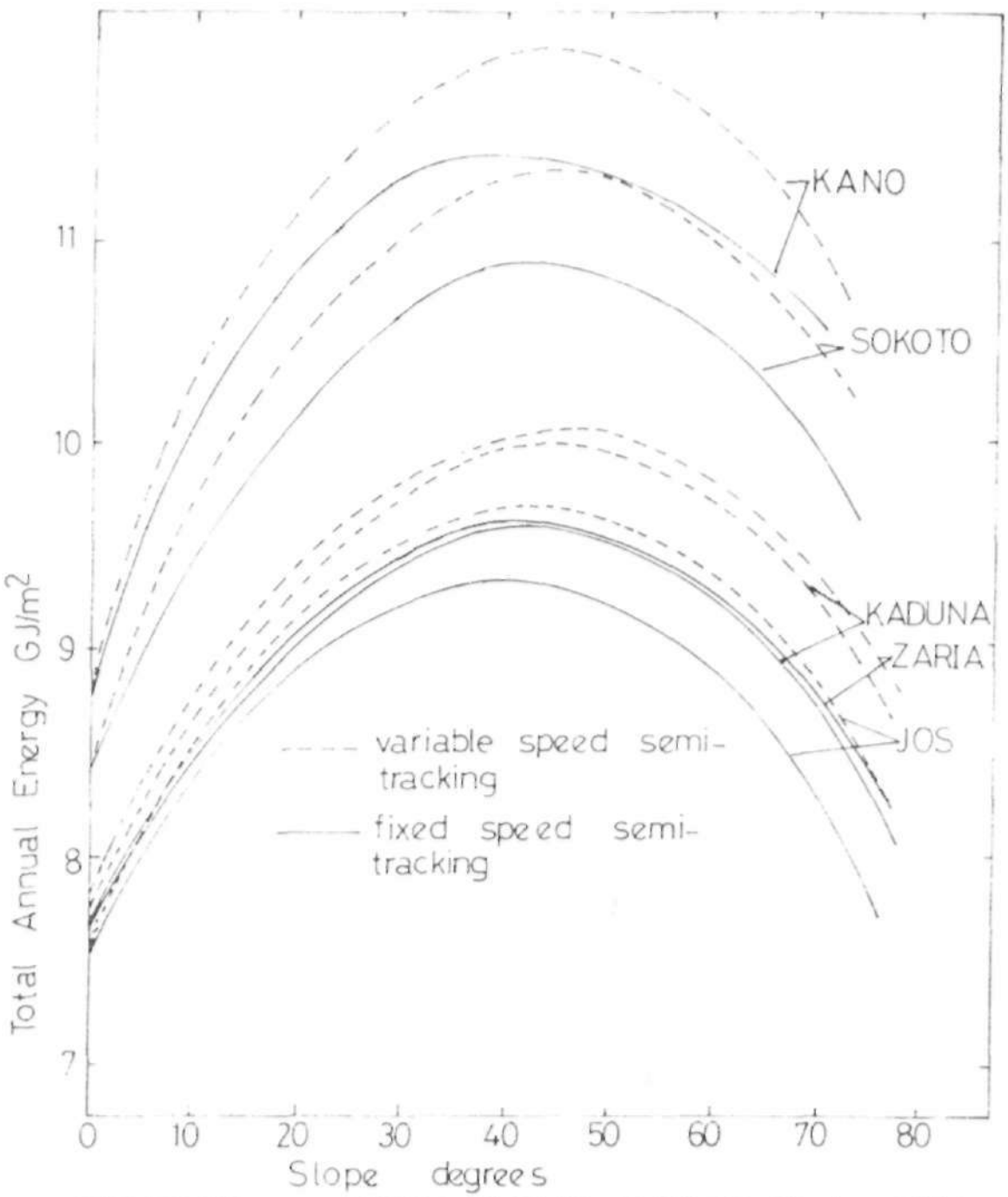


FIGURE 2.2 TOTAL ANNUAL ENERGY AS A FUNCTION OF SLOPE FOR VARIOUS TRACKING MODES

speed semi-tracking flat plate collector the surface azimuth angle is equal to the hour angle.

The programme is presented in Appendix (B) Table 2.2 to 2.6 shows the slopes for maximum useful energy gain for the various months. Table 2.7 and Figure 2.1 shows the optimum slopes for yearly operation for the various locations. For the fixed collector, it can be seen that though not to a high percentage, it varies from the normally accepted latitude angles. This variation can be attributed to the harmattan winds which, with its high dust intensity, greatly reduces the received solar radiation, and thus the clearness index for some months. The variable speed semi - tracking and constant speed semi - tracking collectors optimum slope appear to bear no relation with latitude angle.

Table 2.8 and 2.9 and Figure 2.2 show the total annual energy gain for different orientations for the various locations, whilst Table 2.10 shows the percentage increase achieved by tracking the Sun. It can be seen that by fully tracking the sun more than 35 percent increase in solar radiation is achieved. Variable speed Semi- tracking and constant speed semi- tracking result in an increase of 30 and 25 percent respectively of solar radiation absorbed. Monthly tracking for fixed collectors results in an increase of about 6 percent of absorbed solar radiation.

Thus it can be seen that significant energy gains

from flat plate collectors may be achieved by operating them in the various sun tracking modes. However, tracking involves additional costs and will have to be wieghed against the additional costs in the decision as to whether to track or not.

2.3 PREDICTING THE PERFORMANCE OF THE SOLAR FLAT PLATE COLLECTOR.

Performance predictions for solar -powered cooling system requires a good knowledge of the daily insolation, temperatures, heat loss coefficients as well as some relevant climatological data for specific reqions. For accurate predictions these data must be collected for long periods of time so as to be sufficiently representive.

A detailed analysis of a system usually requires that parameters such as insolation and heat loss coefficients be evaluated on an hourly basis from hourly weather data using a large, high speed computer. Although this approach gives good results, it has several shortcomings. In addition to being very costly and time consuming, the system preformance results apply only to the period described by the input data. True long time performance can only be perdicted by analysing about 10 to 20 years of data or by providing long term average input weather data.

Several methods have been developed for detail system analysis. Liu and Jordan (1963) described a method of predicting the long term average performance of flat

plate collectors. Although they have developed an effective means of generating average weather data, the method requires many calculations on a large high speed computers. Barnaby and Wilcox (1975) coupled a degree-day heating load to the Liu and Jordan technique. The method is however very complicated and requires a high speed computer to perform its numerous calculations. Another method described by Klein et al (1975) and extended by Stickford (1976) shows that the overall system performance can be predicted using average weather data and average system characteristics. However such averaging resulted in a 15 percent variation of the predicted energy collected.

In this project, the only performance data averaged are the insolation, (averaged over 5 years (Arinze, 1984) (a reasonably good estimate)) and the effective transmittance -absorptance product.

The daily average collector transmittance - absorptance product has been evaluated through out the year for latitudes of from 10 to 60 degrees and for a collector with one and two cover plates (Stickford, 1976) For all latitudes and collector tilt angles, the variation of the transmittance - absorptance product throughout the year was less than 4 percent. Thus a constant transmittance - absorptance product will not be very erroneous for performance studies.

The purpose of the performance studies is to predict the collector outlet fluid temperature for various times

of the day, the reason being to know when the system when acting as heating source for an absorption heat pump system would be ineffective. Since the system is expected to be operational throughout the year, it is imperative to choose the smallest insolation for the year as an input parameter. For Zaria location, this occurs in July (Arinze, 1984). Thus, parameter used in the simulation as those for the mean day of the month of July.
From such simulations, it will be possible to predict the performance of the heat pump system, know when to add various solar augmentation methods to increase energy output or when other sources of heating should be applied.

2.3.1 Performance Prediction.

The purpose of this calculation is to obtain the collector fluid outlet temperature so as to determine whether a substantial temperature rise will be achieved between the absorber and the collector outlet. A temperature decrease or low temperature rise will indicate which hours of the day would have its collector energy augmented. It would also indicate which concentration and circulation rate are best to operate the system.

The calculation of the collector outlet fluid temperature requires a knowledge of the absorbed solar radiation on an hourly basis S. This is given by

$$S = (\tau\alpha)_{ave} \; I_T \tag{2.13}$$

The method of calculating I_T has already been outlined in Section 2.2.

The outlet collector temperature is given by the expression (Duffie and Beckman, 1980).

$$T_{fo} = (T_{f,i} - T_a - S/U_L)\, e^{-(A_c U_L F'/MC_p)} \tag{2.14}$$

U_L is given by

$$U_L = U_t + U_b + U_e \tag{2.14}$$

U_t is calculated from the Klein correlation (Duffie and Beckman, 1980) represented by

$$U_t = \left[\frac{N}{\frac{C}{T_{p,m}} \left[\frac{(T_{p,m}-T_a)}{(N+f)} \right]^e} + \frac{1}{h_w} \right]^{-1} + \frac{\sigma (T_{p,m}+T_a)(T_{p,m}^2+T_a^2)}{(\varepsilon_p+0.00591Nh_w)^{-1} + \frac{2N+f-1+0.133\varepsilon_p}{\varepsilon_g} - N} \qquad (2.15)$$

where N = number of glass covers;

$f = (1+0.089h_w - 0.1166h_w\varepsilon_p)(1+0.07866N)$

$C = 520(1 - 0.000051\beta^2)$ For $0^o < \beta < 70^o$.

For $70^o < \beta < 90^o$, use $\beta = 70^o$

$e = 0.43 (1 - 100/T_{p,m})$

β = Collector tilt (degrees)

ε_g = emittance of glass (0.88)

ε_p = emittance of plate

T_a = ambient temperature (K)

$T_{p,m}$ = mean plate temperature (K)

h_w = wind heat tranfer coefficient $(W/_m{}^2C)$

The collector overall loss coefficient, U_L is the sum of the top, bottom and edge loss coefficients

$$U_L = U_t + U_b + U_e \tag{2.16}$$

where $U_b = K/L$

$$U_e = \frac{(UA)_{edge}}{A_c}$$

The mean plate temperature is given by the expression (Duffie and Beckman, 1980)

$$T_{p,m} = T_{f,i} + \frac{Q_u/A_c}{U_L F_R}(1 - F_R) \tag{2.17}$$

where $Q_u = A_c(S - U_L(T_{p,m} - T_a)$

$$F_R = \frac{M}{A_c U_L}\, 1 - e^{-(A_c U_L F'/MC_p)}$$

$$F' = (1/U_L)/W \left[(1/(U_L(D+(W-D)F) + 1/C_B + 1/\pi D_i h_{f,i}\right]$$

Equations (2.17) and (2.15) are solved in an iterative manner. First an estimate of the mean plate temperature is made from which U_L is calculated. With approximate values of F_R, F' and Q_u, a new mean plate temperature is obtained from Equation (2.17) and used to

find a new value of the top loss coefficient. With the new values of U_L, F_R and F' are calculated and the process is repeated until the guessed temperature of the plate is equal to calculated temperature.

The daily minimum and maximum temperatures are used to calculate the ambient temperature for various times of the day. The method adopted in this simulatation assumes that the maximum temperature occurs at 13 hours local time, the minimum at 6 hours local time, and the average at 18 hours local time. In between this time the ambient temperature is assumed to follow a sinusiodal variation. Thus from 6 hours to 13 hours local time, the ambient temperature is given by the relation.

$$T_a = T_{min} + (T_{max}-T_{min})\,\text{Sin}\ \pi/2\,\frac{(t-60)}{8} \qquad (2.18)$$

From 13 hours local time to 18 hours local time,

$$T_a = T_{av}+(T_{max}-T_{av})\,\text{Sin}\pi/2\,\frac{(t-13)}{6} \qquad (2.19)$$

The mean fluid temperature is needed to calculate the specific heat capacity of the fluid flowing in the tubes.

The mean fluid temperature is given by (Duffie and Beckman, 1980).

$$T_{f,m} = T_{f,1} + \frac{Q_u/A_c(1-F'')}{U_L F_R} \qquad (2.20)$$

where $F'' = F_R/F'$

The specific heat capacity of the ammonia - water mixture is approximated to that of water which is given by the relationship

$$C_p = 4186.8(A+BxT+CxT^2+DxT^3) \qquad (2.21)$$

where A = 0.5741

B = 2.825×10^{-3}

C = -8.371×10^{-6}

D = 8.609×10^{-9}

The following are values of various parameters and constants used in the simulation,

Refractive index = 1.526

Ground reflectance = 0.2

Absorptance - transmittance product = 0.85

Collector area A_c = 17.0 m^2

Plate thermal conductivity= 430 W/m^oC

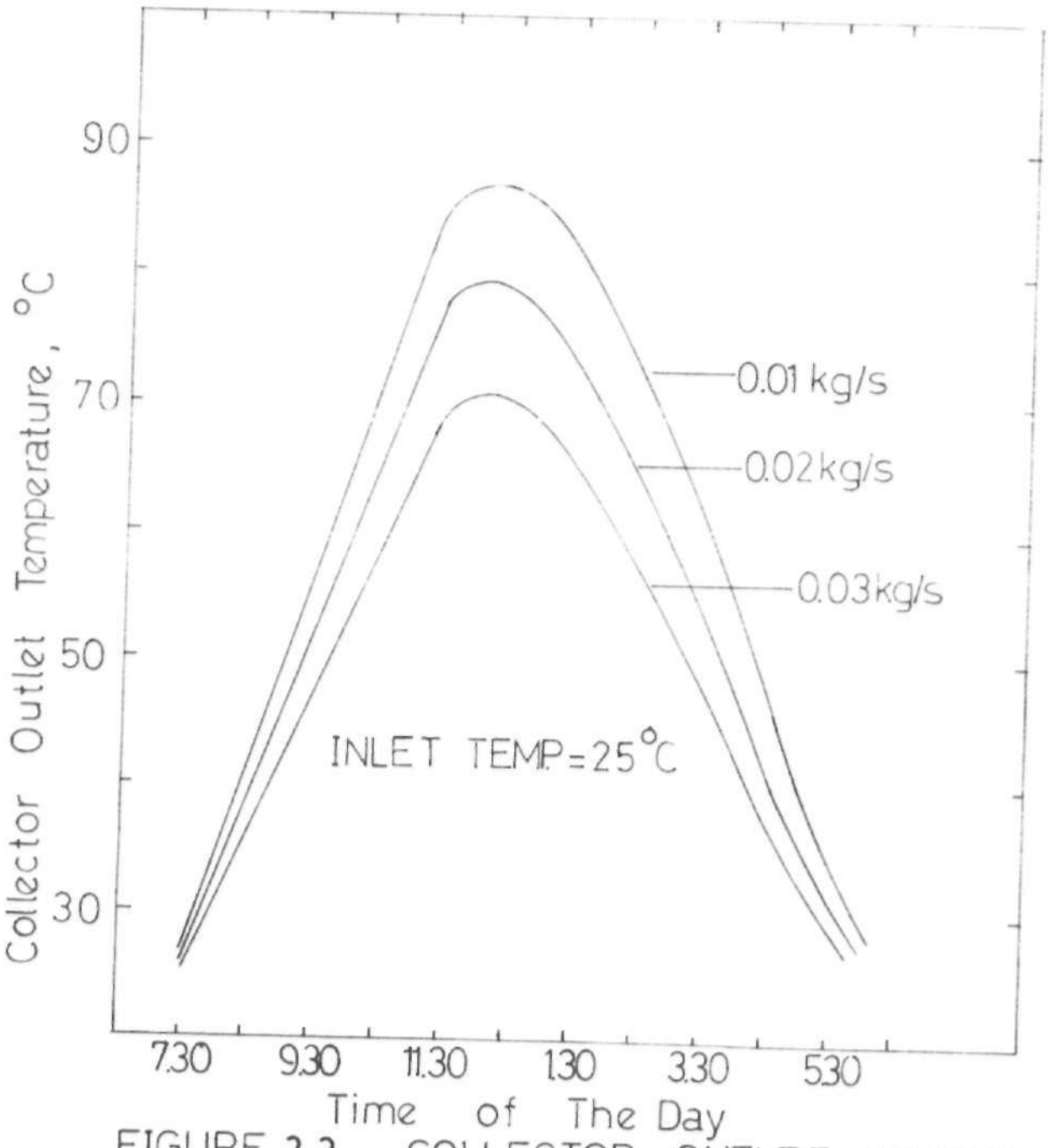


FIGURE 2.3 COLLECTOR OUTLET TEMPERATURE AGAINST TIME OF THE DAY

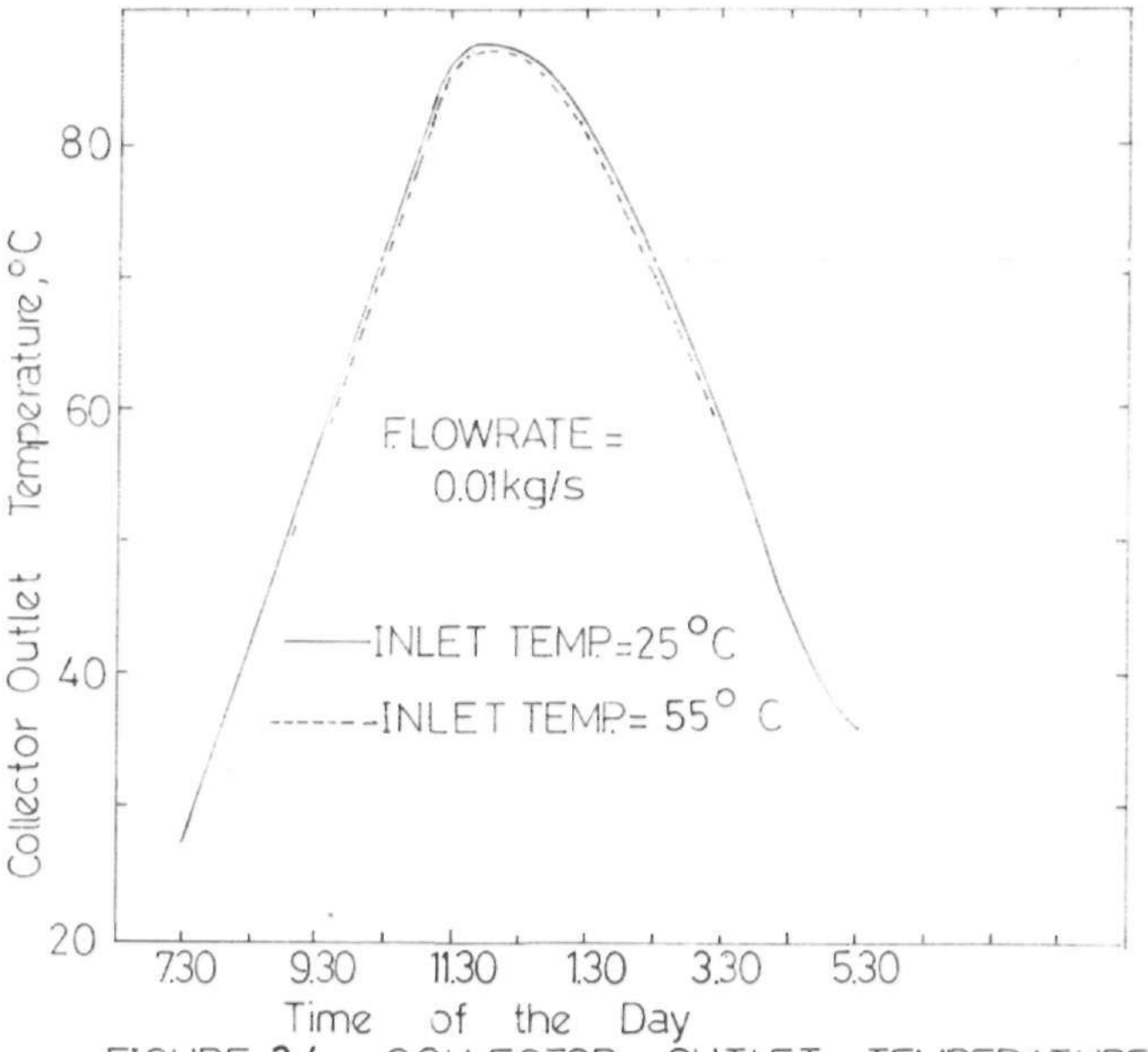


FIGURE 2.4 COLLECTOR OUTLET TEMPERATURE AGAINST TIME OF THE DAY

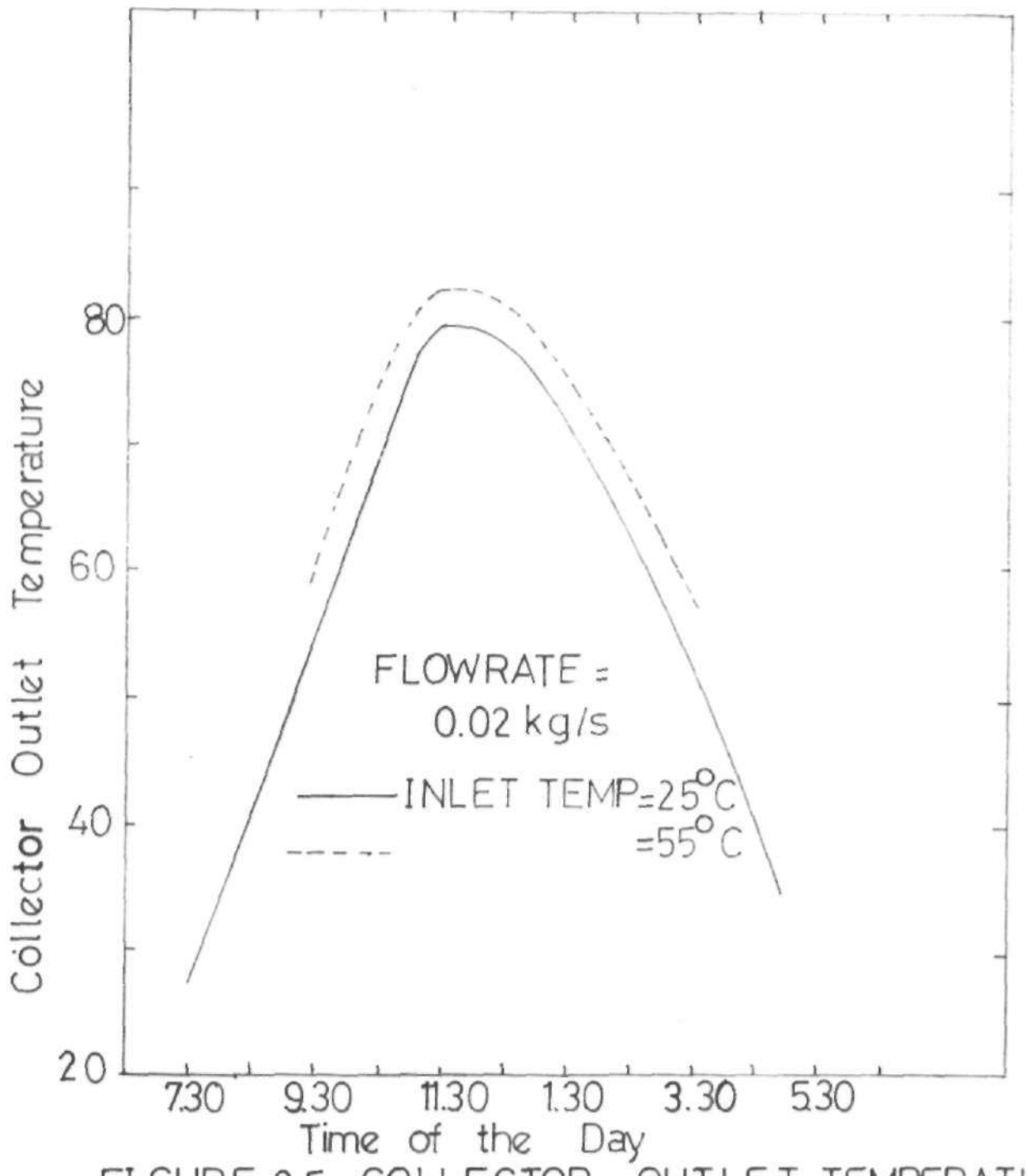


FIGURE 2.5 COLLECTOR OUTLET TEMPERATURE AGAINST TIME OF THE DAY

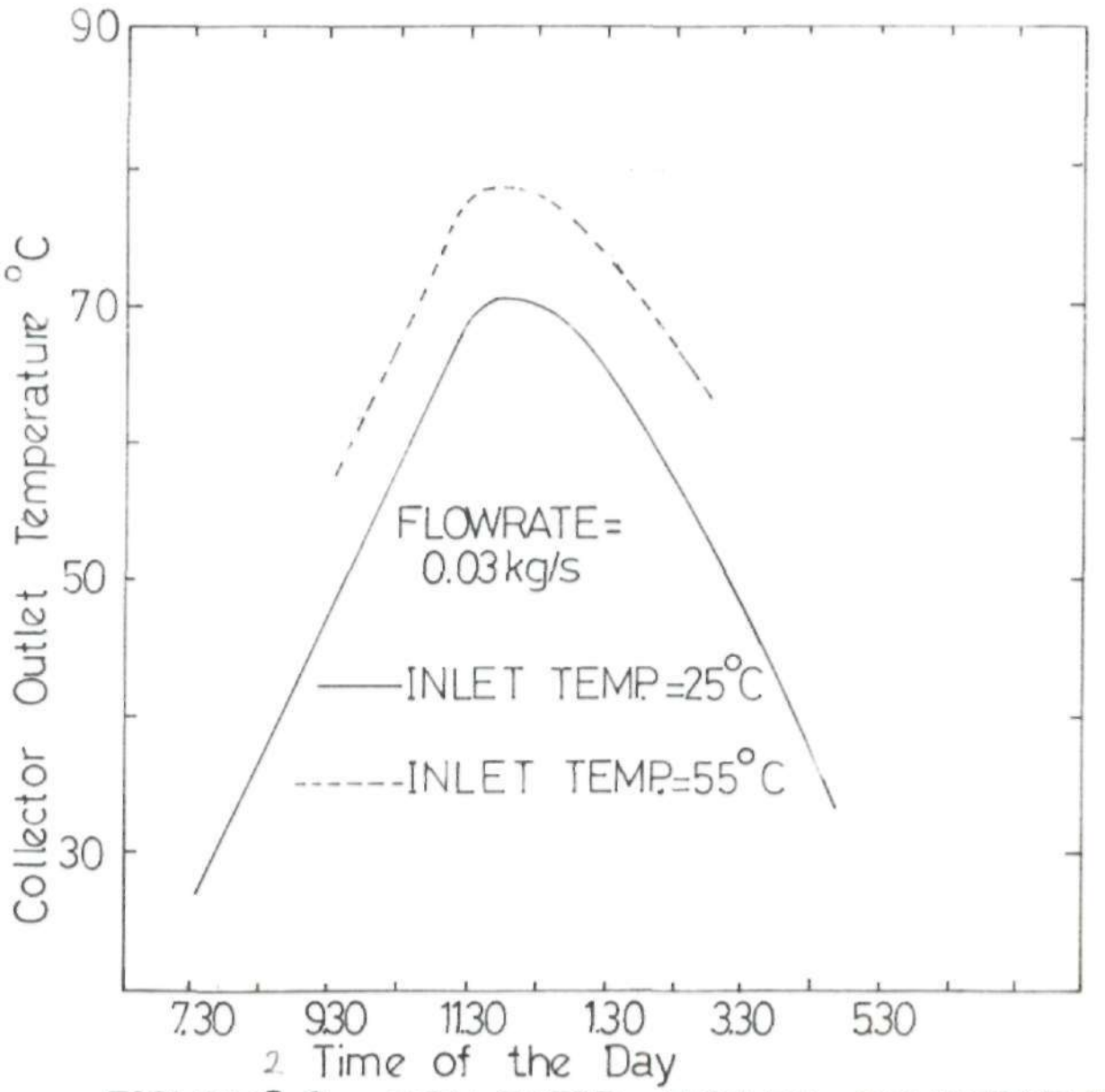


FIGURE 2.6 COLLECTOR OUTLET TEMPERATURE AGAINST TIME OF THE DAY

Plate thickness = 0.05m

Tube internal diameter = 0.0157m

Tube external diameter = 0.02134m

Tube spacing = 0.05m

Heat transfer coefficient = 300 $w/m^{2o}C$

Minimum and Maximum Temperature = 19.7 and 28.5^{o}C

The fluid flowrate was varied from 0.01 kg/s to 0.03 kg/s and inlet temperature from 25^{o}C to 55^{o}C. A computer program was written to carry out the simulation. It is presented in Appendix (C)

2.3.2 Results analysis

The mean hourly collector outlet fluid temperatures for three different working fluid flowrates and extreme inlet temperatures for the collector of 25^{o}C and 55^{o}C, are shown in Figure 2.3, to 2.6. From the figures, the outlet temperatures for both cases of inlet temperatures increase up to about 13 hours local time and then start to fall. This can be explained by the increase in insolation values and ambient temperatures for the time concerned. Comparing the four plots, it is seen that for a fixed collector configuration, inlet temperature and weather, the collector outlet temperature increase as the flowrate decreases. However the inlet temperature has a significant effect on the various flowrates. For very low flowrates the inlet fluid temperature has no significant

effect on the outlet temperature as can be seen on Figure 2.4. In fact, there is a slight decrease in outlet temperature for an increase in inlet temperature for the flowrate of 0.01 kg/s. This can be explained by the increase in energy losses when the inlet temperature increases. Also there is a reduction of useful energy gain when flowrate is decreased.

Generator temperature for the solar - powered ammonia - water system are expected to vary from 60^{o}C to about 120^{o}C for reliable cooling to be achieved. For a condenser temperature of 30^{o}C, the pressure is about 11.4 bars which is also equals to the generator pressure. For various weight percent of ammonia in water of 30, 40 and 50, the corresponding mixture temperatures are 107^{o}C, 86^{o}C and 68^{o}C respectively. Assuming an absorber temperature of 30^{o}C (collector inlet temperature) Table 2.11 shows the time of the day when such temperatures can be achieved. From Table 2.11 it can be seen that there is room for manipulation of the flowrate when the generator concentration is above 50 weight percent. The manipulation of flowrate is an important criterion because, there is, an optimum level at which the disengager can be operated so that working fluid liquid should not flow into the rectifier. The flowrate must be decrease when this level is exceeded. This, however, critically affects the flow ratio (This is essentially the ratio of the mass flowrate of solution in the circuit linking the generator and the

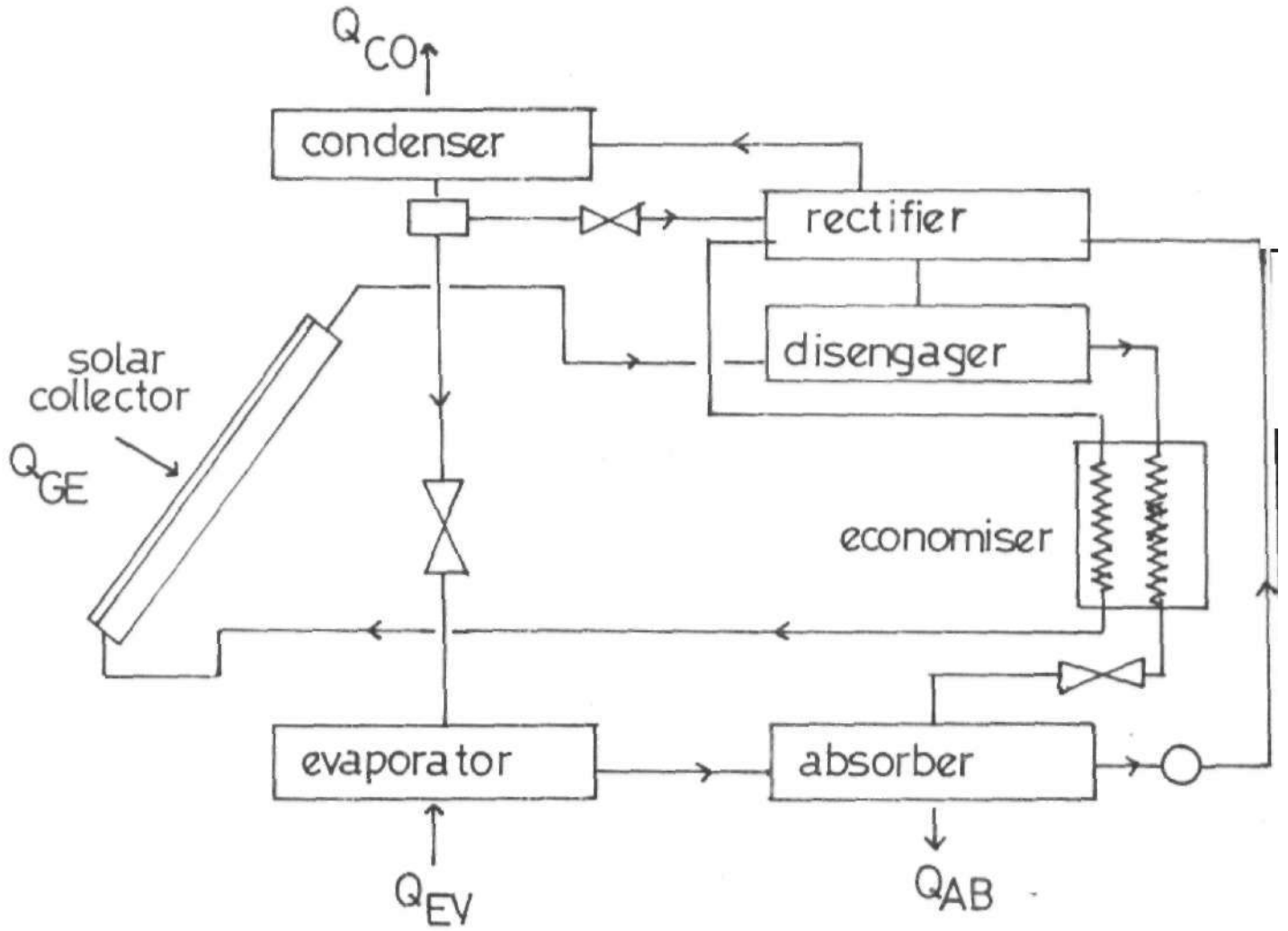


FIGURE 2.7 SOLAR POWERED ABSORPTION HEAT PUMP SYSTEM

absorber to the mass flowrate of pure working fliud in the circuit linking the condenser and evaporator). Figure 2.7 shows a schematic of an absorption heat pump system showing the various circuits. From the table it can be seen that effective solar heating can be achieved only between 10.30 local time and 14.30 local time, for a flowrate of 0.01 kg/s and an absorber concentration of 50 weight percent. Lower concentrations and higher flowrates result in poor cooling.

TABLE 2.11 THE TIMES OF THE DAY WHEN EFFECTIVE COOLING CAN BE ACHIEVED

Flowrate kg/s / WT. %	0.01	0.02	0.03
50	10.30-14.30	10.30-14.30	11.30-13.00
40	11.30- 13.30	/	/
30	/	/	/

CHAPTER THREE

HEAT PUMP SYSTEMS

3.1 INTRODUCTION

The operation of a heat pump cycle results in the transfer of heat energy from a medium to another medium at a relatively higher temperature. The operation requires the input of a quantity of energy which may be, in its final, intermediate or primary forms. This prime energy may be supplied directly as in solar- powered heat pumps or through another thermodynamic cycle. [A review of thermodynamic cycles is presented by **Haywood** (1980)]. Thermodynamic cycle combinations include; Brayton power cycle-Heat pump cycle, Stirling power cycle -Heat pump cycle, and Rankine power cycle - Heat pump cycle. Of these the one most commonly exploited in the Rankine power cycle - Heat pump cycle.

Heat pump systems can be broadly divided into two types based on the mode the prime mover effects the temperature lift. They are:

(1) Vapour compression heat pumps[5] in which the prime energy causes a compressor to effect vapour compression resulting in a corresponding saturation temperature lift.

(2) Sorption heat pumps where the lift is achieved via one of the absorption or adsorption processes.

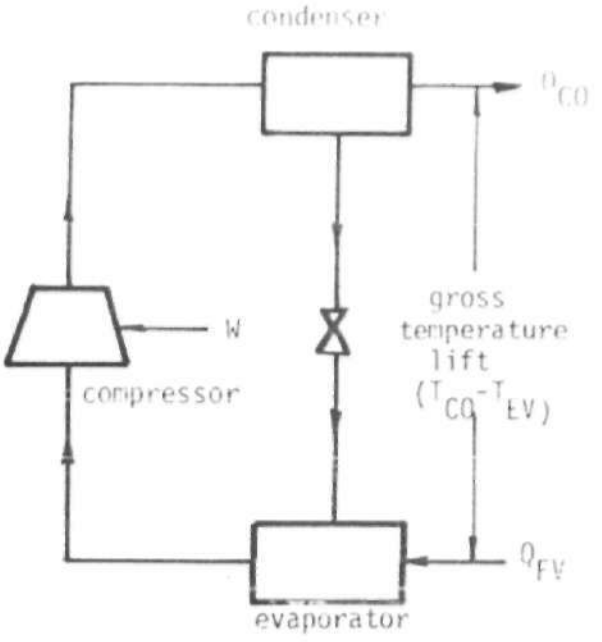


FIGURE 3.1 MECHANICAL VAPOUR COMPRESSION HEAT PUMP
(ADAPTED FROM ADEFILA (1983))

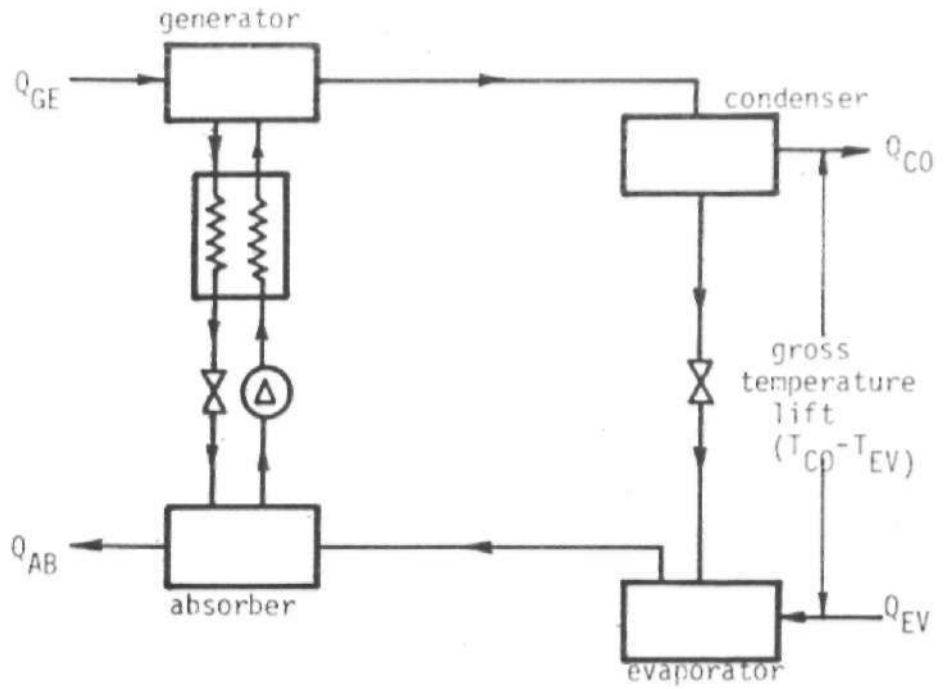


FIGURE 3.2 CONVENTIONAL ABSORPTION HEAT PUMP
(ADAPTED FROM ADEFILA (1983))

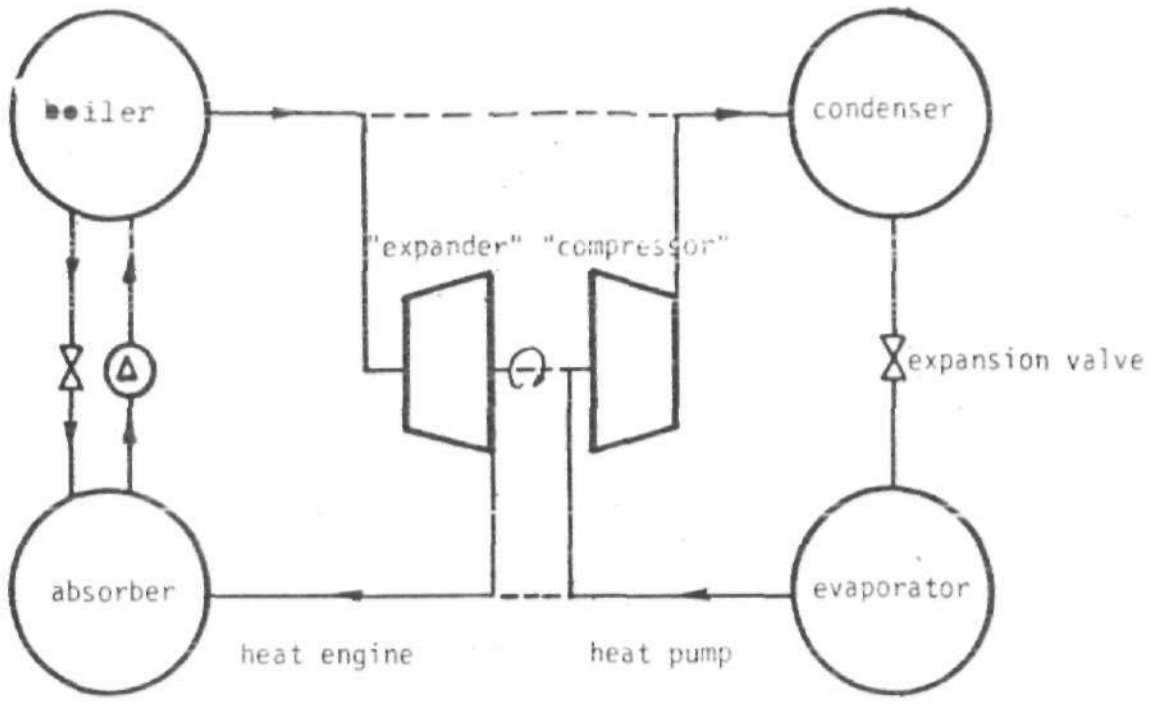


FIGURE 3.3 HEAT ENGINE AND HEAT PUMP COMPONENTS OF A CONTINUOUS ABSORPTION HEAT PUMP SYSTEM (ADAPTED FROM ADEFILA (1983))

Both cases are reviewed in this chapter.

3.2 VAPOUR COMPRESSION HEAT PUMP

This system consists basically of condenser and evaporator heat exchangers, a compressor, an expansion valve and a homogeneous working fluid. The system can be represented as shown in Figure 3.1.

The processes that go on in the system is as follows: The working fluid is compressed using the compressor to produce a high temperature, high pressure working fluid, which is condensed when it is passed through the condenser and heat is rejected from it. The condensed working fluid is expanded through an expansion valve into an evaporator where vaporisation takes place and heat is drawn from either the cooled storage reservoir or directly from the space to be cooled. Once the vaporisation process has occurred, the vapour flows to the compressor for recompression and the cycle is repeated.

Various ways exist to run the compressor. It may be mechanically driven using an electric motor or it may be connected to a power cycle. In the power cycle configuration, a Rankine cycle heat engine which receives energy from a waste heat source or from the sun through a collector converts this energy into mechanical work and then uses this mechanical work to drive the conventional vapour compression heat pump system. This is shown schematically in Figure 3.3. Heat is supplied to the

boiler of this system from the heat source or from heat storage. Vaporisation of the working fluid takes place in the boiler. The vapour is separated, if necessary, from the liquid and is then passed to a turbine (expander) where mechanical work is extracted from it. Depending on the working fluid involved, the fluid may then be passed through a regenerator where it rejects some of its energy before it flows to the condenser where it is liquefied. The condensed liquid is pumped through the regenerator back into the boiler. The regenerator serves to improve system efficiency by tranferring available energy from the turbine exhaust fluid to the boiler feed. Mechanical work obtained from the heat energy system is then used to drive a compressor.

Selection of working fluids for the power cycle and heat pump cycle has for many years been receiving considerable attention. Water has been a common Rankine cycle power fluid. However for lower boiler temperatures in solar- powered systems, water is less desirable because it results to lower system efficiencies. Several criteria must be considered in power cycle working fluid selection. These include:

(1) The shape of the dew line on the temperature - entropy diagram..

(2) Thermal stability of the fluid at the operating temperature.

(3) The vapour pressure of the fluid at the

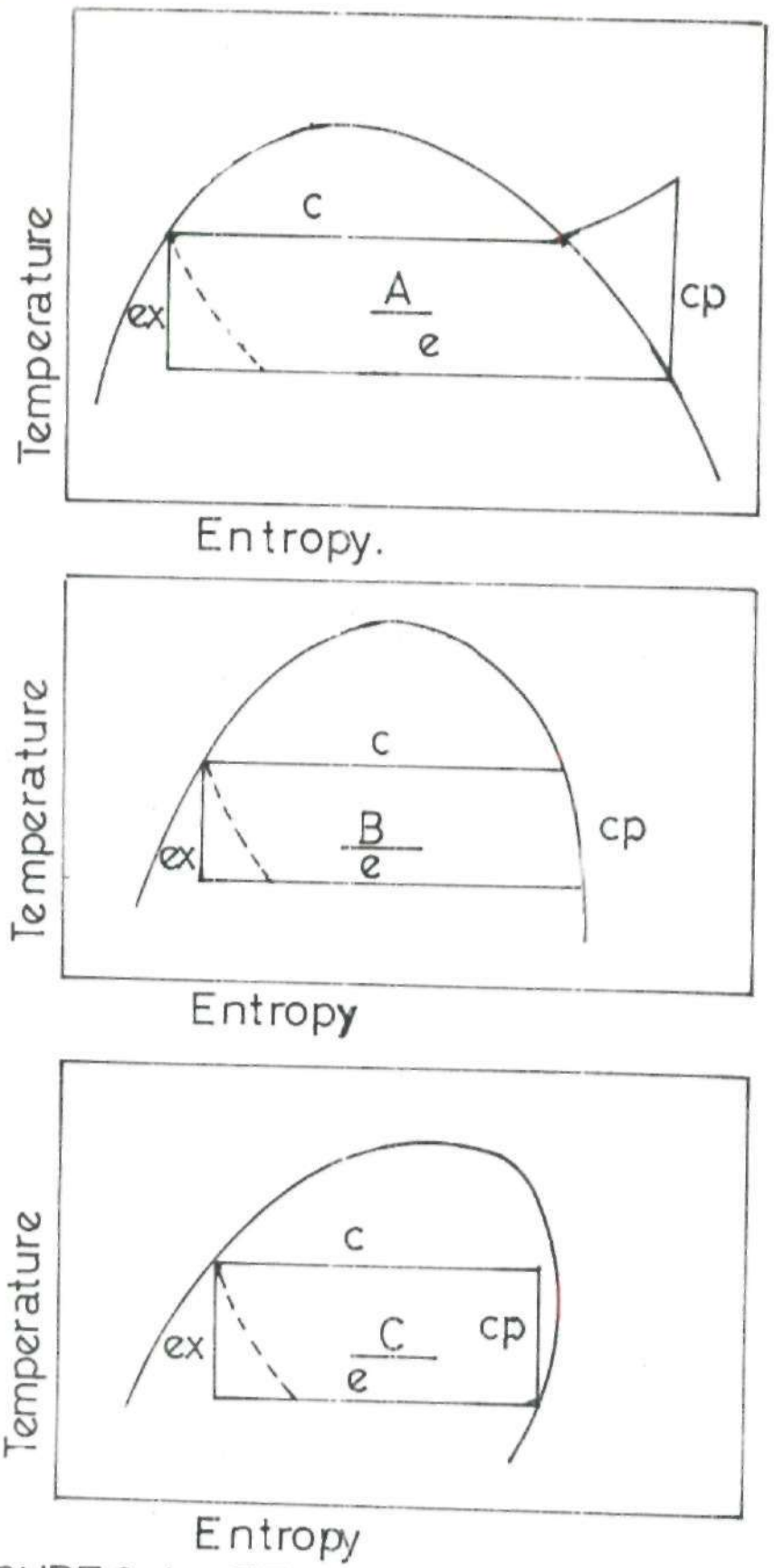


FIGURE 3.4 TEMPERATURE AGAINST ENTROPY FOR THREE KINDS OF FLUIDS; c = condensation, ex = expansion, cp = compression, e = evaporation

heat rejection temperature.

(4) The molecular weight of the working fluid.

The first criterion is very important. This can be understood by comparing the various diagrams in Figure 3.4. Figure 3.4a shows the envelope for a wetting fluid such as water. When a fluid like this is expanded isentropically (in general it is regarded as adiabatic, shown by dotted lines.) in a turbine from a saturated state, liquid droplets form which decrease turbine efficiency and at the same time can results in turbine erosion damage. Figure 3.4 b shows a two phase envelope for a so-called drying fluid. When a saturated vapour of this class is expanded isentropically in a turbine, the expanded fluid, existing in the turbine is in the superheated state . Because this fluid contains energy which could be used to improve cycle efficiency, it is necessary to install a regenerator (desuperheater) to extract and transfer the energy to the feed going to the boiler. Figure 3.4 c show a vapour - liquid dome for an ideal working fluid; a fluid which is referred to as isentropic. When such a working fluid is expanded through a turbine from saturated vapour conditions at the entrance, it remains near saturation throughout the expansion. As a result, there is no turbine damage or loss in efficiiency associated with the process and no

regenerative heat exchanger is required.

Generally the above criterion can be analysed by examination of the Mollier diagrams. However, for many chemicals, such diagrams are not available. It has been shown by Tabor and Bronicki (1965) that the slope can be approximated, assuming ideal gases by the equation

$$\frac{\partial T}{\partial s} = \frac{T}{Cp^{v} - \frac{DH^{v}}{T}} \qquad (3.1)$$

For low temperature solar- powered systems, the second criterion (Thermal stability) is generally not a significant problem with most working fluid.

A high vapour pressure at the heat rejection temperature insures a sufficiently high density that the heat transfer surface area required in the condenser is small, and therefore a low condenser cost.

High molecular weight working fluids insures a reasonable low spouting velocity for the turbine which facilitates the use of a low turbine tip speed and hence a turbine which operates at acceptably low stress level (Wilbur and Karaki, 1977).

A low Prandtl number reflects the desirability of having high convective heat transfer coefficients, high thermal conductivities and low viscosities in the working fluid. The high heat transfer coefficient of course

results in smaller heat transfer components and low viscosity results in lower pressure drops and hence lower pumping requirements.

There is need for other criteria such as low flammability, low toxicity and low cost.

Studies (Sargent and Teagan, 1973) (Pugmore and Barber, 1974) show that various Freon products(For example,R113 and R-114) can be used as working fluids for the power cycle and for the heat pump cycle in solar cooling applications.

3.3 THE SORPTION HEAT PUMP SYSTEM

Basically two types of sorption heat pump systems exist which can either be closed or opened. They are the absorption and adsorption heat pump systems. In convention absorption and adsorption heat pump systems such as those employing lithium bromide - water or ammonia-water and silica gel - water respectively, both the absorbent or adsorbent and working fluid circulate in a closed cycle and there is no physical contact with the environment to be heated or cooled. In the open cycle only the absorbent or adsorbent is recirculated and the working fluid usually water is continually evaporated to the atmosphere and must therefore be continuously added to the system. The condenser and generator of the conventional closed cycle are not necessary, so there is a substantial reduction in equipment requirements. An

important combination that has been tested with promising result is the Lithium chloride - water absorption open heat pump system (Leboeuf and Lof, 1980). It is necessary to note that in an absorption heat pump system a working fluid combines with an absorbent in the liquid state whilst in the adsorption heat pump, the working fluid combines with the adsorbent in the solid state.

3.3.1 Absorption Closed Heat Pump Systems

Of the several heat pump system alternatives which might be considered for solar cooling applications, one of the most attractive is the absorption system where heat is supplied to the system and the cooling or heating effect is achieved directly with no significant amount of rotating machinery. This system is attractive not only because of its simplicity but also because of it's demonstrated perfomance when used in solar powered applications (Chung et al, 1963), (Eisenstadt et al, 1960), (Duffie and Sheridan, 1965), (Sheridan, 1972), (Swartman et al, 1973) and (Ward et al, 1973). The fluid for the system is a solution containing a working fluid and an absorbent which has a strong chemical affinity for each other. Such a system can be represented as shown in Figure 3.2. If the chemical affinity results in large temperature change, then the system is called a chemical heat pump. Thus a chemical heat pump is a device which through reversible chemical reaction, uses the

thermodyamic availability of a high temperature heat input to extract from a low temperature source. A variety of working fluids with either liquid absorbents or solid adsorbents can be utilised for chemical heat pump. Possibilities include (Mclinden and Klein, 1980) $CaCl_2$ - CH_3OH, NH_4NO_3 - NH_3, H_2SO_4 - H_2O and NaSCN-NH_3. The process that goes on in the conventional absorption heat pump system is as follows: Heat is transferred to the solution of working fluid and absorbent contained in the generator and as a result, working fluid is vaporised from the mixture, leaving a solution having a low working fluid concentration behind. The liberated vapour is free to flow to the condenser where heat is removed from it to bring about its liquifaction. The generator and the condenser together comprise the high pressure section of the system. Liquid working fluid which is accumulated in the condenser is available for expansion from this high pressure portion of the system into the low pressure evaporator, wherein vaporisation of the working fluid takes place and the cooling effect is achieved. Once the working fluid has been vaporised in the evaporator and has removed heat from cold storage or the space being cooled, it is discharged to the absorber. In the absorber, the vapour recombines with the mixture coming from the generator, which is weak in working fluid concentration. Since this recombination reaction is exothermic, heat must be removed from the absorber to maintain its temperature

at a sufficiently low value to assure the desired high chemical affinity between the working fluid and the solution. The resulting absorber solution which is rich in working fluid is collected in the absorber and pumped back to the generator to maintain the generator solution level and concentration. The pump maintains the desired pressure difference in the system.

The requirement to circulate working fluid - weak solution continuously from the high temperature generator to the low temperature absorber and working fluid - strong solution in the opposite direction necessitates the installation of an economiser. The economiser is a simple counter - flow heat exchanger which minimises the heat losses associated with the fluid transfer between these two components. Without the economiser, the heat load on the collector and the heat rejection load associated with the absorber would both be increased with a consequent reduction in the coefficient of performance of the system.

3.3.2 *Working Fluid Characteristics*

The working fluid combination used in absorption systems is very important. The working fluid characteristics that are desired are as follows: (Buffington, 1949), (Adefila, 1983)

(1) The ratio of the latent heat of vaporisation of the working fluid to the heat required to

separate the working fluid from the absorbent and produce working fluid vapour should be as near to unity as possible. Typical values for this ratio are between 0.7 and 0.9 (Wilbur and Karaki, 1977)

(2) The solubility of the working fluid in the absorbent should be extra-ordinarily high at the absorber temperature and at the pressure corresponding to the evaporator temperature. If the solubility is not high, excessive amount of solution will have to be circulated between the generator and absorber. As a result internal heat losses can become excessive and system performance can be poor.

(3) The viscosity of the solution should be low to facilitate rapid absorption in the absorber and to minimise pumping power requirements.

(4) The diffusion, mixing and heat transfer characteristics of the solution should facilitate rapid absorption of working fluid through the boundary layer of the solution.

(5) A high proportion of the working fluid content of the working fluid- rich solution should be recoverable upon distillation to minimise the amount of solution that must be circulated.

(6) The absorbent should be non-volatile or at least much less volatile than the working fluid

to avoid or minimise carry-over of the absorbent into the condenser. The presence of significant absorbent carry-over necessitates the installation of a rectification system between the generator and condenser.

(7) The solidification temperature of any liquid solutions which may exist in the system should be lower than any temperature which can be encountered during operation to avoid clogging of system pipings. This criterion becomes a problem in systems which use salts as the absorbent of the mixture. In such systems, the solution may be cooled at the outlet from the recuperator to the point where solidification can occur. This phenomenon is most likely under transient operating conditions.

(8) At the prevailing heat rejection temperature and the evaporator temperature, the required generator temperature must be sufficiently low that operation from flat plate collectors at reasonable collector efficiencies can be achieved. If this condition is satisfied, temperatures in the absorption unit will generally be sufficiently low that corrosion, decomposition and other irreversible processes in the heat pump system will not be significant. Corrosion or decomposition if they do occur can

produce gaseous products which interfere with the absorption and condensation processes and reduce the cooling capacity of the unit.

9) The vapour pressure for the mixtures should be lower than those predicted by Raoult's law.

10) The pressure of the absorber and evaporator should be above atmospheric so that contaminants such as air and water vapour should not leak into the system if it is not properly sealed.

11) The pressure difference between the high and low pressure side should not be very large because that will necessitate the installation of an expensive pump.

There is no one set of absorbent/working fluid combination which posseses all of the above mentioned characteristics. Although ammonia - sodium thiocyanate has been mentioned by many researchers as a mixture with great advantages for solar - powered heat pump system, ammonia-water combination has been chosen because of availability. It also has the outstanding advantage of being able to operate the systems' generator with a high source temperature as low as 65^{o} C. Such a source is available from solar collection/storage subsystem, utilising low cost flat plate collectors. Since this subsystem represents a substantial portion of the total system cost, it is essential that the components required for collection and storage of solar energy be mininised.

The lithium bromide - water absorption system requires source temperature in excess of 85^{o}C for efficient operation (Ward and Lof, 1975). The collection systems require to produce these temperatures are very expensive as they usually involve the use of focusing collectors. The solubility of ammonia in water is such that a wide range of operating temperature are possible. It is even possible to use the ammonia - water system for producing temperatures below 0^{o}C.

The most obvious disadvantage of the ammonia - water system is the toxicity of ammonia. Classification rating of ammonia as a working fluid (ASHRAE, 1972) reports the relative safety classification for ammonia, indicating that ANSI B91 - 1971 safety code group 2 (Group 1 is least hazardous and Group 3 most hazardous) and in the Underwriter's laboratories classification Group 2 (vapours which are lethal or produce serious injuries with exposures of 1/2 to 1 percent concentration for 1/2 hour). Thus ammonia heat pump equipment is generally located outside the conditioned space and chilled or some other suitable heat transfer media is used to provide cooling to the space for human comfort.

For storage facilities in an aerated environment, when human or animal contact is rare , such as in the rural areas, such transfer media may not be necessary. However the technology of ammonia equipment is well

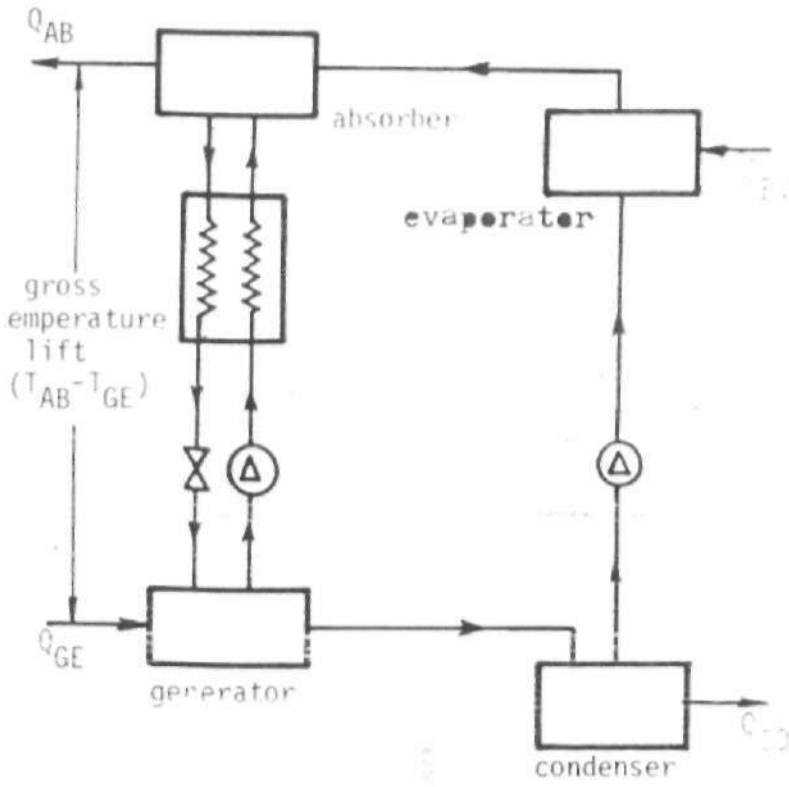


FIGURE 3.5 REVERSED ABSORPTION HEAT PUMP (ADAPTED FROM ADEFILA (1983))

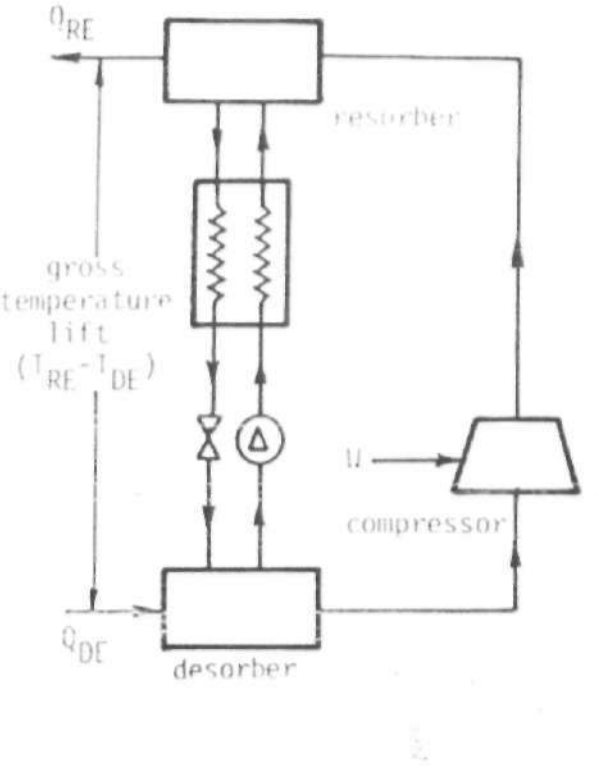


FIGURE 3.6 RESORPTION HEAT PUMP

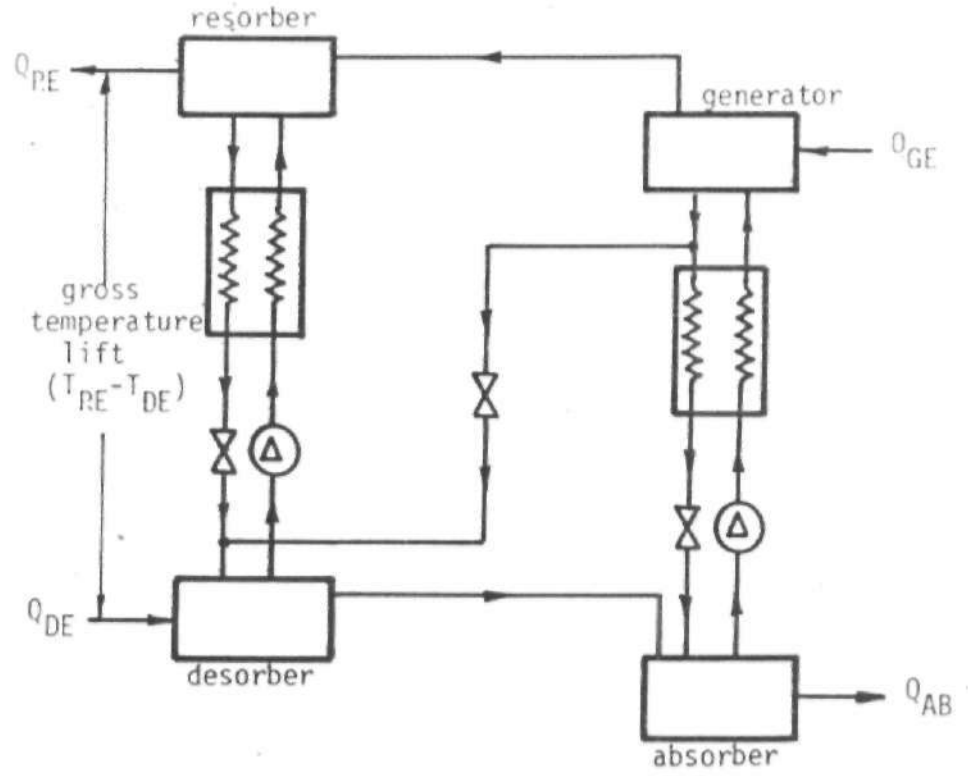


FIGURE 3.7 ABSORPTION-RESORPTION HEAT PUMP (ADAPTED FROM ADEFILA (1983))

established. Therefore its toxicity and corrosive properties offer no serious barriers to its use in solar-powered heat pump systems.

3.4 DERIVATIVES OF THE VAPOUR COMPRESSION AND THE ABSORPTION HEAT PUMPS

Various derivatives of the vapour compression and the absorption heat pump have been documentated (Adefila, 1983). They include the reversed absorption heat pump, the resorption heat pump, the absorption - resorption heat pump and the chemical heat pump already mentioned before. Illustrations of the reversed absorption heat pump, the resorption heat pump and the absorption - resorption heat pump are presented in Figure 3.5, 3.6 and 3.7 respectively.

In the reversed absorption heat pump or absorption heat transformer, the relative pressure levels in the heat exchangers are reversed when compared with the conventional absorption heat pump system described before. The direction of flow between the generator - absorber loop is reversed and between the condenser and evaporator the throttling valve is replaced by a pump.

The resorption heat pump solution circuit is the same as that of the reversed absorption heat pump in the concentration and temperature levels. However the pure working fluid circuit is completed through a compressor.

Compressed vapour is then absorbed by working fluid - poor solution in the resorber. The working fluid - rich solution from the resorber passes through an expansion valve to the desorber where it takes in heat to generate working fluid vapour. Working fluid - poor solution is pumped into the resorber and the working fluid vapour passes to the compressor to complete the cycle.

The absorption - resorption heat pump has the generator - absorber solution circuit similar to that of the conventional absorption heat pump system in conjunction with the equivalent of the resorber - desorber solution circuit of the resorption heat pump system.

3.5 INTERMITTENT ABSORPTION SYSTEMS

The systems so far described are continuous systems. Absorption heat pump system can also be operated intermittently.

In a typical intermittent system, the generator is heated by the heat source. The working fluid is driven off and condenses in the condenser, which is cooled by air or water. At the end of the regeneration, the condenser is placed in the space to be cooled; it now functions as the evaporator. The generator is remqved from the heat source and cooled; it now functions as the absorber. The vapour pressure over the solution decreases and the working fluid in the evaporator vaporises and is reabsorbed in the

absorber to the mass flowrate of pure working fliud in the circuit linking the condenser and evaporator). Figure 2.7 shows a schematic of an absorption heat pump system showing the various circuits. From the table it can be seen that effective solar heating can be achieved only between 10.30 local time and 14.30 local time, for a flowrate of 0.01 kg/s and an absorber concentration of 50 weight percent. Lower concentrations and higher flowrates result to poor cooling.

TABLE 2.11 THE TIMES OF THE DAY WHEN EFFECTIVE COOLING CAN BE ACHIEVED

Flowrate kg/s	0.01	0.02	0.03
WT. %			
50	10.30-14.30	10.30-14.30	11.30-13.00
40	11.30- 13.30	/	/
30	/	/	/

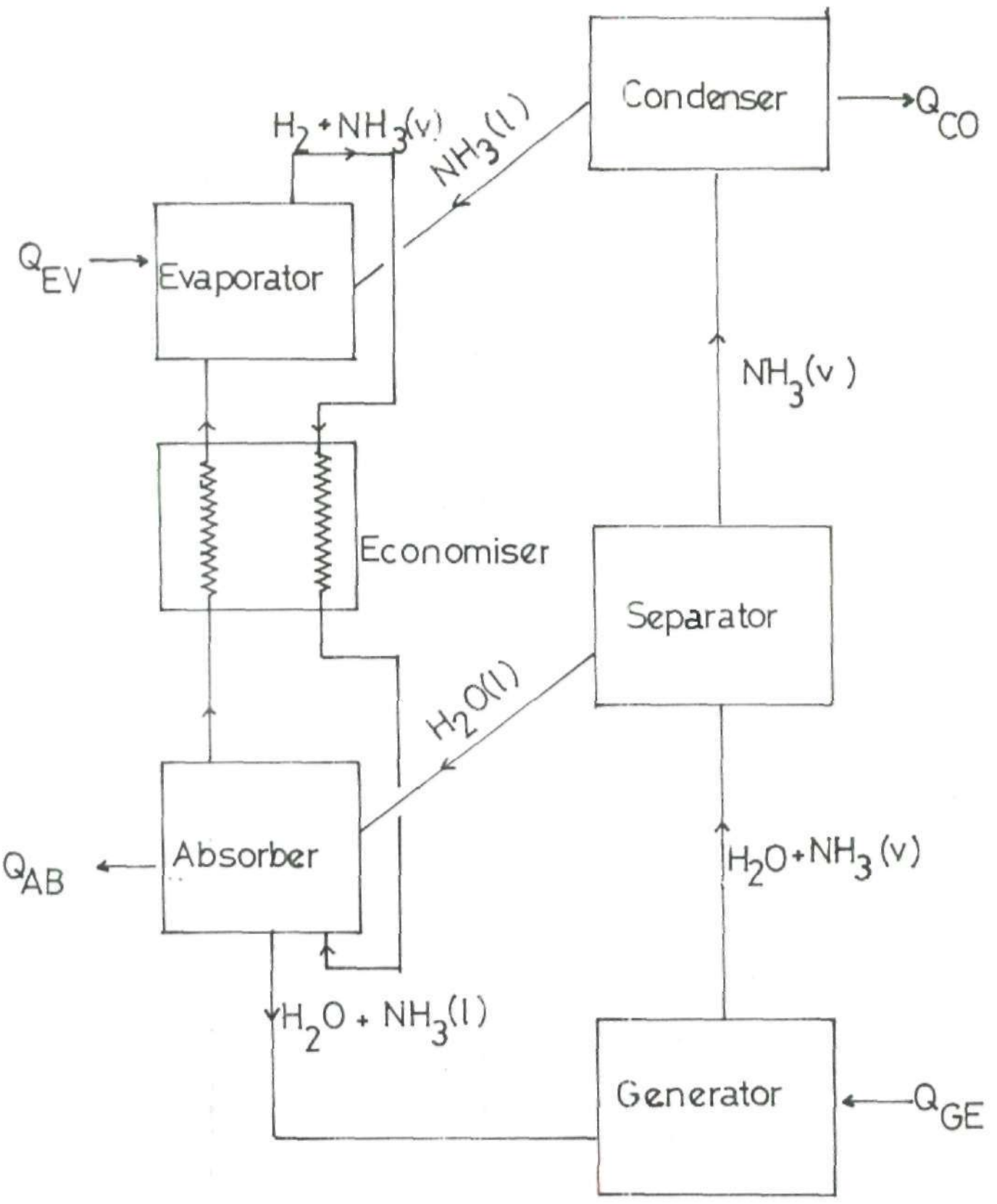


FIGURE 3.8 AMMONIA-WATER- HYDROGEN VAPOUR-LIFT ABSORPTION SYSTEM; l = liquid v = vapour.

solution.

3.6 VAPOUR-LIFT ABSORPTION HEAT PUMP SYSTEM

The commonest form of the vapour - lift absorption heat pump system in the one employing the three fluid combination of ammonia, hydrogen and water. The system is shown schematically in Figure 3.8. Evaporation of the ammonia is achieved by mixing the liquid with hydrogen gas which is trapped in one part of the cycle. Because the ammonia liquid then evaporates to exert only a partial pressure, it evaporates as if the pressure was actually reduced, that is at a lower temperature. This neatly eliminates the need for any moving parts. In the generator only water and ammonia exist, whilst in the evaporator and condenser only ammonia and hydrogen come in contact. The three fluid come in contact in the absorber.

3.7 ADVANTAGES AND DISADVANTAGES OF ABSORPTION HEAT PUMP OVER VAPOUR COMPRESSION HEAT PUMP

The advantages are:

(1) They are inherently simple and potentially highly reliable. The solution pump and heat transfer circulation equipment are the only

moving parts. No compressor is needed.

(2) They are highly efficient in the heating mode due to possible heat recovery from onsite combustion or from solar energy as supplementary heat to that provided by the absorption cycle condenser itself.

(3) They have lower energy requirements than vapour compression heat pumps.

(4) They use working fluid - absorbent mixtures, which are chemically non - reactive with atmosphere ozone.

(5) There is easier coupling of absorption cycle to solar thermal sources.

The following are the substantial disadvantages:

(1) Comparatively large heat exchangers area (at attendant high first cost) are required to realise acceptable COPs.

(2) The "plumbing" of absorption equipment are apparently more complex.

3.8 ADSORPTION HEAT PUMP SYTEM

The adsorption heat pump systems are not well exploited despite their potentials. It also exist in the closed and opened configurations. In the closed cycle, a solid hygroscopic material such as silica gel which readily adsorbs working fluid vapour is placed in the adsorber. The application of heat will release the adsorbed vapour. To make the process continuous, two adsorbers are provided to operate alternatively with a condenser and an evaporator. While one adsorber is operating, the other is being reactivated.

The open cycle is still very far behind in development. Dunkle (1965) proposed a system which is based on the use of rotary desiccant wheels for air dehumidification.

CHAPTER FOUR

THERMODYNAMIC PROPERTY DATA EQUATIONS FOR AMMONIA - WATER MIXTURES

4.1 INTRODUCTION

The study of ammonia - water solution systems for application in absorption heat pump and other system employing other thermodynamic cycle is an important step necessary for an optimum design and system characterisation. This is obvious since what principally happens in such systems are changes in the equilibrium conditions in terms of pressure enthalpy, concentration and temperature in the various units that make up the system.

Though considerable amount of data (Macriss et al, 1964) have been compiled on ammonia - water mixtures, most of it are in tabular or chart form and this does not make it handy in the computer-aided design and analysis of thermodynamic units that use this solution mixture. It is therefore necessary to transform these tabulated data into appropriate equations for ready use by designers.

A number of attempts have been made in this direction notably by Jain and Gable (1971). The equations were conceived for use in air-cooled air conditioning systems for which the pressure ranges of interest were 17 to 24

bar and 3.4 to 5.5 bar corresponding to ammonia temperature ranges of 43 to 55°C and - 5 to 7°C respectively. Application of these equations to absorption heat pump systems where ranges outside those considered are normally encountered is limited. A more embracing representation is thus needed.

4.2 EVALUATION OF THE POLYNOMIALS

An extremely important role in the approximate computation of a function F (x) is played by its expansion in a series of the polynomial P_n (x)

$$F(x) = \sum_{j=0}^{L} A_j P_j(x) \qquad \ldots\ldots(1)$$

this expansion gives in a number of cases, good expansion to F (x) in the sense of the uniform norm. This holds particularly for expansions in Chebyshev polynomial which are set of orthogonal polynomials on the internal(-1,1)

The Chebyshev polynomial of the first kind can be written as

$$P_n(x) = \text{Cos } (n \text{ arcCos } x) \qquad \ldots\ldots(2)$$

The series in Equation (1) can be extended to two independent variables lead to the double Chebyshev series.

$$F(x) = \sum_{i=0}^{K} \sum_{j=0}^{L} A_{ij} P_j(x_1) P_i(x_2) \qquad \ldots\ldots(3)$$

Since the values of x_1 and x_2 must be in the range (-1,1), a transformation from their original values is necessary.

Thus we have

$$x_1 = (2X_1 - (X_1max + X_1min))/(X_1max - X_1min). \quad (4)$$

$$x_2 = (2X_2 - (X_2max + X_1min))/(X_2max - X_2min) \quad (5)$$

These are the values that are used in the evaluation of the polynomials.

To calculate the coefficients of the double Chebyshev series of the first kind, use was made of the Numerical Algorithms Group (NAG) subroutines. The standard convention followed by this subroutine is that coefficients in the expansion in equation (3) which has either i or j zero, are written as 0.5 A_{ij} instead of simply A_{ij} and the coefficient with both i and j = zero is written as $0.25A_{oo}$. It should be noted the maximum and minimum values are the end-points of the ranges and when using a particular set of coefficients only values within such ranges should be applied.

THE CHEBYSHEV COEFFICIENTS OF THE FITS

The ranges of the (IGT) data considered were subdivided into sections for which sets of coefficients were obtained. This was done for the following reasons:

(i) limitation of data points which may be handled by the (NAG) subroutines

(ii) limitation of accuracy with increasing data points

and (iii) the user may extract the range of data required.

Fixed valued values for the maximum and minimum used for each set of coefficients to put it in a form readily handled by a digital computer. Thus there are possibilities of calculating values outside concentration ranges not presented in the (IGT) data.

4.31 Saturated liquid enthalpy of water-ammonia mixtures as a function of saturation temperature and saturated solution concentration (H_L - T - X) .

The series can be written as

$$H = \sum_{i=o}^{K} \sum_{j=o}^{L} A_{ij} P_i(x) P_j(t)$$

A) T_{min} = 222.04 K

P_{max} = 244.26 K

X_{min} = 20.0 wt. percent

X_{max} = 100.0 wt. percent

The Chebyshev coefficients of the fit are

-1.3216E + 03	9.4969E + 01	-3.1666E - 02	4.3314E - 01
1.5740E + 02	3.2014E + 01	2.4325E - 01	-3.1984E - 01
1.6094E + 02	-1.3050E+ 00	-2.2632E - 02	3.1497E - 01
-2.3146E + 01	1.4216E + 00	1.3909 01	-3.8417E - 01
-7.4279E - 02	-8.6024E - 01	8.2936E - 02	1.7467E - 01

Maximum percentage error = $\pm$ 0.98 percent.

Representing the above coefficients thus

$$\begin{array}{cccc} A_{00} & A_{01} & A_{02} & A_{03} \\ A_{10} & A_{11} & A_{12} & A_{13} \\ A_{20} & A_{21} & A_{22} & A_{23} \\ A_{30} & A_{31} & A_{32} & A_{33} \\ A_{40} & A_{41} & A_{42} & A_{43} \end{array}$$

Transforming the original values of the concentration and temperature we have

$$x = (2.X - (X_{max} + X_{min}))/(X_{max} - X_{min}) \qquad (7)$$

$$t = (2T - (T_{max} + T_{min}))/(T_{max} - T_{min})$$

Changing them into Chebyshev polynomial terms of the first kind, we have

$$x_1 = \text{Cos}(\text{arc Cos } x) = P_1(x) = x$$

$$x_2 = \text{Cos}(2.0 \text{ arc Cos } x) = P_2(x) = 2x^2 - 1$$

$$x_3 = \text{Cos}(3.0 \text{ arc Cosx}) = P_3(x) = 4x^3 - 3x$$

$$x_4 = \text{Cos}(4.0 \text{ arc Cos } x) = P_4(x) = 8x^4 - 8x^2 + 1$$

The transformed temperature value is similarly expressed. Other terms which are encountered in this paper are expressed thus:

$$x_5 = \text{Cos}(5.0 \text{ arc Cos } x) = P_5(x) = 16x^5 - 20x^3 + 5x$$

$$x_6 = \text{Cos}(6.0 \text{ arc Cos } x) = P_6(x) = 32x^6 - 48x^4 + 18x^2 - 1.$$

The expansion of equation (6) can be written, following the convention explained before as

$$A = 0.25*A_{oo} + 0.5*A_{01}*t_1 + 0.5* A_{02}*t_2 + 0.5* A_{03}*t_3 \quad .$$

$$B = x_1*[0.5*A_{10} + A_{11}*t_1 + A_{12}*t_2 + A_{13}*t_3]$$

$$C = x_2*[0.5* A_{20} + A_{21}*t_1 + A_{22}*t_2 + A_{23}*t_3]$$

$$D = x_3*[0.5*A_{30} + A_{31}*t_1 + A_{32}*t_2 + A_{33}*t_3]$$

$$E = x_4*[0.5*A_{40} + A_{41}*t_1 + A_{42}*t_2 + A_{43}*t_3]$$

The saturated liquid enthalpy is thus

$$A = A + B + C + D + E$$

The other group of coefficients are similarly expanded.

B) T_{min} = 244.26 K

T_{max} = 277.59 K

X_{min} = 0.0 wt. percent

X_{max} = 100.0 wt. percent

The Chebyshev coefficients of the fit are:

-6.6586E + 02	1.2289E+02	1.6225E + 01	-5.5491 + 00	1.7900E + 00
-3.4341E + 01	3.9441E + 01	-2.5658E + 01	1.2577E + 01	-4.8478E + 00
2.8283E + 02	-2.7887E+ 01	1.9848E + 01	-9.6408E + 00	3.6936E + 00
-2.8241E + 01	1.8012E + 01	-1.2881E + 01	5.8303E + 00	-2.3708E + 00
-8.9504E + 00	-9.2031E + 00	6.5372E + 00	-2.5547E + 00	1.1633E + 00
2.9213E + 00	3.1212E + 00	-2.3136E + 00	5.2531E - 01	-3.2884E - 01

Maximum error $\pm$ 0.692 percent

C) T_{min} = 277.59 K

T_{max} = 355.37 K

X_{min} = 0.0 wt. percent

X_{max} = 100.0 wt. percent

The Chebyshev coefficients of the fit are:

3.2255E + 02	3.6289E + 02	3.3806E +00	-7.2415E-01	1.4402E-01	6.6832E-02
3.4532E + 01	1.4225E + 01	1.6401E +00	8.9253E-02	-7.0264E-02	4.8604E-02
2.4860E + 02	-2.9560E + 00	-4.1666E-03	1.6996E-01	-3.2549E-01	-3.8759E-01
-8.7244E + 00	1.4846E - 00	1.9850E-02	-5.0182E-01	-3.3891E-01	-3.7317E-01
-1.9291E + 01	-1.4657E + 00	-6.5413E-01	-5.4658E-01	-5.9881E-01	-5.1416E-01
4.6776E + 00	-4.9093E-01	-7.9984E-01	-5.4673E-01	-2.6951E-01	-3.1059E-01
1.4003E + 00	4.0202E-01	-4.5985E-01	-4.6205-01	-6.0398-01	9.9959E-02

Maximum error = $\pm$ 0.772 percent

D) T_{min} = 355.37 K

T_{max} = 394.26 K

X_{min} = 0.0 wt. percent

X_{max} = 70.0 wt. percent

The Chebyshev coefficients of the fit are:

1.2758E + 03	2.4429E + 02	5.2142E + 01	2.9296E + 01	8.2448E + 00
-8.7811E + 01	6.5408E + 01	4.8115E + 01	2.7352E + 01	7.8816E + 00
1.6829E + 02	4.3925E + 01	3.79036E+01	2.2064 + 01	6.4675E + 00
4.7818E + 01	2.8686E +01	2.4066E + 01	1.4303E + 01	4.1339E + 00
5.2834E + 00	1.3162E + 01	1.2173E + 01	7.4215E + 00	2.0025E + 00
8.5591E - 01	3.4482E + 00	3.7938E + 00	1.7389E + 00	-1.3170E - 01

Maximum Error = $\pm$ 0.692 percent

E) T_{min} = 394.26 K

T_{max} = 433.15 K

X_{min} = 0.0 wt.percent

X_{max} = 50.0 wt.percent.

The Chebyshev coefficients of the fit are:

1.9090E + 03	1.7852 + 02	4.1156E + 00
-2.0197E + 02	5.6062E + 00	3.6370E + 00
3.2849E + 01	-1.9626E + 00	2.5930E + 00
9.5341E + 00	-6.7737E - 01	9.7004E - 01

Maximum Error = 0.572 percent.

4.3.2 Boiling Points of Water-Ammonia mixtures as a function of Saturation Pressure and Saturated Solution concentration (T-P-X).

The series can be written as

$$T = \sum_{i=0}^{K} \sum_{j=0}^{L} A_{ij} P_i(x) P_j(p)$$

P_{min} = 0.81655 Bars

P_{max} = 2.044 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

1.1808E + 03	2.2756E + 01	-2.1393E + 00
-1.3296E + 02	-2.0565E + 00	2.0270E - 01
-3.7784E + 00	1.8045E - 01	-1.5058E - 02
-1.7086E + 00	1.1242E - 01	-2.4552E - 02

Maximum error = $\pm$ 0.731 percent

A) P_{min} = 2.044 Bars

P_{max} = 10.0207 Bars

X_{min} = 0.0 wt. percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

0	1.3432E+03	5.0573E+01	-7.4615E+00	1.5989E+00	-4.4155E-01
1	-1.4677E+02	-4.4559E+00	7.0187E-01	-1.0395E-01	2.7736E-02
2	3.8944E+01	2.9585E-01	-2.0679E-02	8.4342E-03	-2.4076E-02
3	-7.7446E-01	2.8930E-01	-3.0539E-02	-1.9785E-02	2.6029E-02
4	-1.0967E+00	-4.3997E-01	-2.7643E-02	3.8438E-02	9.3406E-03
5	-1.2267E+00	3.5413E-01	-3.2161E-02	-2.1582E-02	-21347E-02

Maximum error $\pm$ 0.625

B) P_{min} = 10.0207 Bars

P_{max} = 20.414 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.00 wt.percent

The Chebyshev coefficients of the fit are:

1.4940E+ 03	2.8658E + 01	-2.1132E + 00	2.1205E - 01	-8.1942E - 03
-1.5924E + 02	-1.9905E + 00	3.6542E - 01	-5.9377E - 02	-2.6529E - 02
3.9428E + 01	-4.8887E-02	-2.0674E-02	1.2702E - 02	3.5557E - 03
-1.4843E - 01	-4.0882E-02	-8.9779E - 02	2.0667E - 02	1.8525E - 02
-2.2580E + 00	-2.7257E-02	1.0244E - 01	-2.9635E - 02	-8.9233E - 03
-2.9753E - 01	2.1228E - 02	-1.3925E - 01	2.6453E - 02	1.2928E - 02

Maximum Error = $\pm$ 0.487 percent

C) P_{min} = 20.414 Bars

P_{max} = 27.218 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

1.5745E + 03	1.3224E + 01	-2.3889E - 01	7.6243E - 02	-20605E - 02
-1.6455E + 02	-9.0433E - 01	5.6693E - 02	1.4544E - 02	-3.9139E - 03
-3,8984E + 01	-1.2025E - 01	6.9276E - 02	4.0042E - 03	-6.5407E - 03
-4.3264E - 01	-1.5508E - 02	6.7470E - 02	8.4691E - 03	-2.0490E - 02
-2.0433E + 00	9.8179E - 02	-5.9511E - 03	-2.0067E - 02	3.4546E - 04
-7.4846E -01	-5.0549E - 02	4.4410E - 02	-1.5876E- 02	9.1042E - 03

Maximum Error = $\pm$ 0.677 percent

D) P_{min} = 27.218 Bars

P_{max} = 34.023 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev Coefficients of the fit are:

1.6237E + 03	1.1459E + 01	-1.8412E - 01	-2.1184E-02	5.9552-02
-16772E + 02	-7.7490E- 01	-1.1069E - 02	-8.6803E - 03	-2.3327E - 02
3.8734E + 01	-8.1386E - 02	-4.2450E - 03	-2.9479E - 03	-1.0981E-02
-7.5713E - 02	1.3211E - 01	-1.3167E - 02	9.3962E - 03	-4.6226E - 03
-1.8076E + 00	-4.5050E - 02	-5.1957E - 02	2.3560E - 02	-1.4394F - 04
-6.9600E - 01	6.6907E - 02	2.5790E - 02	-5.9850E-03	-1.9396E - 03

Maximum Error = $\pm$ 0.718 percent.

4.3.3 Saturated Vapour Pressure of water-Ammonia mixtures as a function of Saturation temperature and Saturated liquid composition (P-T-X).

The series can be written as

$$P = \sum_{i=o}^{K} \sum_{j=o}^{L} A_{ij} P_{i(x)} P_{j(t)}$$

A) T_{min} = 222.04K

T_{max} = 272.04K

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

3.2448E + 00	1.5855E + 00	3.9354E - 01	2.9709E - 02	6.1097E-04
1.9682E + 00	9.9792E - 01	1.8535E - 01	3.5790E - 02	6.7818E-04
2.6682 E - 01	9.3827E - 02	4.2163E - 02	-9.5208E - 03	-3.7834E-04
-2.3900E - 01	-9.5458E - 02	-3.5255E - 02	5.3927E - 03	1.8212E-03
-3.3559 - 02	-1.3301E - 02	1.3301E - 02	1.0906E - 03	-2.0879E-03
5.6255E - 02	2.5078E - 02	-1.9766E - 04	-1.2476E - 04	2.4761E-03

Maximum Error = 0.017 per cent

B) T_{min} = 272.04K

T_{max} = 366.48K

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

4.1695E + 01	2.2234 + 01	5.3339E + 00	7.9489E - 01	1.4056 - 01	1.0985E-01
2.5041E + 01	1.2902E+00	2.855E + 00	3.6478E- 01	6.6891E - 02	6.6399E-02
3.6006E + 00	1.8077E +00	4.5798E - 01	2.2088E- 01	1.0704E - 01	8.8219E-02
-1.6133E + 00	-4.3590E-01	2.5530E - 01	3.2605E - 01	1.7448E - 01	1.0163E-01
3.9136E - 01	6.3426E -01	5.3495E - 01	4.1737E - 01	2.1193E - 01	1.2323E-01
1.0303E - 00	7.3074E-01	4.3854E - 01	3.0260E - 01	1.6554E - 01	7.9300E-02
3.3085E-01	2.8291E-01	2.2625E - 01	1.7384E - 01	9.3184E - 02	4.0563E-02

Maximum Error ± 4.013 percent.

C) T_{min} = 366.48 K

T_{max} = 433.15 K

X_{min} = 0.0 wt.percent

X_{max} = 70.0 wt.percent

The Chebyshev coefficients of the fit are:

1.3104E + 02	3.9958E + 01	3.9193E + 00
7.5969E + 01	2.1914E + 01	1.7746E + 00
1.7146E + 01	4.9447E + 00	4.6471E - 02
7.6566E - 01	5.7828E - 01	-2.7343E - 01

Maximum Error = ± 2.082 percent

4.3.4 Saturated Enthalpy of vapour of water - Ammonia mixtures as a function of saturation temperature and solution concentration H_v-T-x

The Series can be written as

$$H = \sum_{i=o}^{K} \sum_{j=o}^{L} A_{ij} P_i(X) P_j(t)$$

A) T_{min} = 222.04 K

T_{max} = 272.04 K

X_{min} = 10.0 wt

X_{max} = 100.0 wt

The Chebyshev coefficients of the fit are:

5.0128E + 03	1.1457E + 02	5.6897E + 00	-2.5324E + 00
-7.2308E + 01	-3.1350E + 01	-7.8275E + 00	2.8550E + 00
3.1638E + 01	1.7072E + 01	5.0754E + 00	-1.9750E + 00
-1.7719E + 01	-1.1221E + 01	-3.4124E + 00	1.6933E + 00
1.0248E + 01	6.9621E + 00	2.3052E + 00	-1,1045E + 00

Maximum Error = 1.23 percent.

B) T_{min} = 272.04 K

T_{max} = 366.48 K

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev Coefficients of the fit are:

6.1992E + 03	2.4716E + 02	3.3732E + 01	5.3471E + 01	1.7703E + 01	2.1241E+01
-8.0644E + 02	-9.9521E + 01	-1.7520E + 01	-1.8913E + 01	-3.0324E + 00	-1.0184E+01
5.3423E + 02	4.5009E + 01	2.4045E + 01	3.4128E +01	1.0801E + 01	1.3807E+01
-3.7555E + 02	-7.0265E + 00	-5.4134E + 00	-1.5606E + 01	-2.9876E + 00	-7.5637E+00
2.5739E + 02	2.4532E + 01	2.5047E + 01	2.8200E + 01	9.4392E + 00	1.2847E+01
-1.3096E + 02	5.8971E + 00	2.4314E + 00	-8.8593E + 00	2.5503E + 00	-5.6148+00
7.7035E + 01	9.8784E + 00	9.3588E + 00	1.7609E + 01	3.7885E + 00	8.1593E+00

Maximum Error = ± 1.73 percent

C) T_{min} = 366.48 K

T_{max} = 433.15 K

X_{min} = 0.0 wt.percent

X_{max} = 70 wt.percent

The Chebyshev coefficients of the fit are:

5.8265E + 03	-6.0326E + 02	-9.7316E + 01
-2.1256E + 03	-6.4875E + 02	-7.4702E + 01
-1.9946E + 02	-3.7572E + 02	-1.3820E +01
-5.5356E + 02	-9.2065E + 01	2.9696E + 01

Maximum Error = ± 1.356 percent.

4.3.5 Saturated Liquid Enthalpy of Water-Ammonia mixtures as a function of Saturation Pressure and Saturated liquid composition (H-P-X)

The Series can be written as

$$H = \sum_{i=0}^{K} \sum_{j=0}^{L} A_{ij} P_i(x) P_j(P)$$

A) P_{min} = 0.81655 Bars

P_{max} = 6.80848 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

3.2525E + 02	2.5706E + 02	-4.6352E + 01	1.4439E + 01	-4.5987E + 00
-6.0294E + 02	-2.0171E + 01	3.7090 + 00	-1.6979E + 00	3.9685E - 01
4.6512E+ 02	-2.1192E + 00	-2.6258E +00	3.1265E - 01	-2.3284E - 01
-1.1011E + 00	3.8417E + 00	-9.9586E - 01	2.8395E - 01	2.4992 - 02
-2.6428E + 01	-1.4649E + 00	3.0351E - 01	-7.1726E - 01	1.4498E - 01
-1.2699E + 00	6.1954E - 01	7.1192E - 01	4.0625E - 01	-9.6715E - 02

Maximum Error - ±0.767 percent

B) P_{min} = 6.8048 Bars

P_{max} = 13.609 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

1.0302E + 03	1.2349E + 02	-7.3496E + 00	3.4768E - 01	1.9758E - 01
-6.5555E + 02	-7.3533E + 00	-7.2908E - 02	-4.6616E - 02	8.8522E - 03
3.9925E + 02	-8.2259E - 01	-2.5459E - 02	1.4107E - 01	-1.9658E - 02
6.2130E + 00	4.6398 - 01	2.0011E - 01	7.5590E - 02	5.5231E - 02
-3.1963E + 01	-1.4215E + 00	-4.1321E - 02	2.2022E - 01	-1.2723E - 01
3.6534E + 00	8.8211E-01	1.9801E - 01	1.3624E - 01	-4.7432E - 02

Maximum Error = ± 0.978 percent.

C) P_{min} = 6.8048 Bars

P_{max} = 17.011 Bars

X_{min} = 50.0 wt.percent

X_{max} = 90.0 wt.percent.

The Chebyshev coefficientsof the fit are:

1.7326E + 00	4.3523E - 01	-1.5428E - 04	-4.8561E - 04
-1.0972E + 00	-2.6241E - 01	-1.0997E - 0.3	3.9816E - 04
4.4516E - 01	1.0318E - 01	3.9169E - 03	5.0312E - 03
-1.7375E - 01	-3.7716E - 02	-4.4397E - 03	-1.0980E - 03
6.0762E - 02	1.0568E - 02	2.2412E - 03	-2.1486E - 03

Maximum Error = $\pm$ 2.034 percent

D) P_{min} = 17.011 Bars

P_{max} = 27.218 Bars

X_{min} = 10.0 wt.percent

X_{max} = 20.0 wt.percent

The Chebyshev coefficients of the fit are:

8.4643E + 01	3.3098E + 00	-8.922OE - 02	1.3087E - 02
-5.0412E + 01	-1.1951E + 00	2.0089E - 02	2.5629E - 03
1.7012E + 01	4.4519E - 02	6.6964E - 03	-1.0070E - 02
-3.5300E + 00	1.2305E - 01	1.2014E - 02	4.5085 - 03
-2.2372E - 01	-1.4930E - 02	-3.3482E - 03	-3.3217E - 03

Maximum Error = $\pm$ 0.035 percent.

C) P_{min} = 13.609 Bars

P_{max} = 20.414 Bars

X_{min} = 0.0wt.percent

X_{max} = 100.0wt.percent

The Chebyshev coefficients of the fit are:

1.4279E + 03	8.0266E + 01	-3.1292E + 00	1.6884E - 01	-1.1254E - 01	-5.3468E - 02
-6.7480E + 02	-2.6592E + 00	6.2775E - 01	1.0462E - 02	6.7449E - 03	1.4341E - 01
3.9548E + 02	-2.1661E - 01	1.7830E - 01	1.1860E - 01	1.5553E - 01	3.1625E - 02
8.5521E + 00	5.4976E - 02	-3.1610E - 01	1.6259E - 02	-4.2885E - 03	-5.3280E - 02
-3.1332E + 01	1.4354E + 00	-2.0157E - 01	6.9942E - 03	-7.1022E - 02	-2.7154E - 02
4.7475E + 00	-7.0992E - 01	-2.5777E-01	-3.0238E - 02	-4.0910E - 03	-9.6515E - 02
5.2364E + 00	2.9082E - 02	1.1096E - 01	-1.1158E - 01	-5.9581E - 02	7.05316 - 03

Maximum Error = $\pm$ 0.96 percent

D) P_{min} = 20.414 Bars

P_{max} = 27.218 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

1.7140E + 03	6.2537E + 01	-1.2585E + 00	5.6442E - 01	-2.9888E - 01
-6.8360E + 02	-2.4989E + 00	-2.4176E - 01	3.0344E - 01	3.1456E - 01
3.9569E + 02	-3.7273E - 01	4.2217E - 01	1.3688E - 01	6.2714E - 02
7.8079E + 00	-1.1654E - 01	1.5774E - 01	-4.7343E - 02	-1.1990E - 01
-2.8574E + 01	7.1527E - 01	2.0110E - 01	-4.2414E - 01	-1.6799E - 01
5.8208E +02	-5.9439E - 01	4.7178E - 01	-2.2485E - 01	1.0589E - 01

Maximum error = 0.252 percent

4.3.6 Enthalpy of Saturated vapour of water-ammonia mixtures as a function of Saturation Pressure and saturated liquid composition (H_v-P-X)

The Series can be written as

$$H = \sum_{i=o}^{K} \sum_{j=o}^{L} A_{ij} P_i(x) P_j(p)$$

A) P_{min} = 0.81655 Bars

P_{max} = 6.8048 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

6.3265E + 03	1.2023E + 02	-2.2185E + 01	-2.7236E + 00	-2.8451E + 00
-1.0518E + 03	-3.4415E + 01	6.4881E + 00	-2.2626E + 00	7.5917E - 01
6.3542E + 02	-4.7632E - 01	-1.4656E + 00	4.1690E - 01	5.0363E - 02
-3.4019E + 02	1.3475E + 01	-2.1800E + 00	3.7572E - 01	1.6365E - 01
1.7089E + 02	-1.8912E + 01	2.8806E + 00	-9.4758E - 01	4.2408E - 01
-7.7869E + 01	1.3568E + 01	-2.7900E + 00	1.3542E + 00	-3.9454E - 01

Maximum Error = ± 1.8 percent

B) P_{min} = 6.8048 Bars

P_{max} = 13.609 Bars

X_{min} = 0.0 wt.percent

X_{max} = 1.00.0 wt.percent

The Chebyshev coefficients of the fit are:

6.6378E + 03	4.8285E+ 01	-4.3626E + 01	6.1399E - 01	1.3816E - 01
-1.1478E + 03	-1.7627E + 01	6.8694E - 01	-1.4043E - 01	8.6286E - 02
6.2614E + 02	-2.8124E + 00	2.8024E - 01	-8.0999E - 02	-1.0639E - 01
-3.0055E + 02	7.6677E + 00	-2.8903E - 01	-2.5974E - 02	8.4168E - 03
1.1776E + 02	-9.2941E + 00	6.5644E - 01	-5.6373E - 02	4.3966E - 02
-4.0844E + 01	6.7731E + 00	2.5970E - 02	9.9376E - 03	2.0703E - 02

Maximum Error = ± 1.28 percent

C) P_{min} = 13.609 Bars

P_{max} = 20.414 Bars

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of fit are:

6.7779E + 03	2.4116E + 01	-1.0851E + 00	4.3635E - 02	-9.3198E - 02
-1.2096E + 03	-1.3111E + 01	7.9144E - 01	6.8590E - 02	8.9306E - 02
6.1586E + 02	-2.4007E + 00	-1.3346E - 01	-1.2180E - 01	4.2400E - 03
-2.7378E + 02	5.6938E + 00	-4.5095E - 01	-2.2670E - 02	-6.7064E - 02
8.9034E + 01	-5.4526E + 00	3.3573E - 01	2.4573E - 02	2.2284E - 02
-1.6189E - 01	5.1768E + 00	-2.6500E - 01	6.6251E - 03	-6.1282E - 02

Maximum Error = ± 1.30 percent

D) P_{min} = 20.414 Bar

P_{max} = 27.218 Bar

X_{min} = 0.0 wt.percent

X_{max} = 100.0 wt.percent

The Chebyshev coefficients of the fit are:

6.8612E + 03	1.8172E + 01	-4.3053E - 01	6.2135E - 02	7.6284E - 02
-1.2541E + 03	-9.7743E + 00	1.7468E - 01	-1.6694E - 02	1.1330E - 01
6.0523E + 02	-2.7316E + 00	2.0343E - 02	1.0279E - 04	3.1993E - 02
-2.5456E + 02	4.1140E + 00	-2.6077E - 01	7.8706E - 02	5.2067E - 02
6.9034E + 01	-4.8301E + 00	-4.3557E - 02	-1.2062E - 01	-3.6739E - 02
2.0969E + 00	4.2868E + 00	-1.8550E - 02	-2.8709E - 02	-4.1407E - 02

Maximum Error = ± 1.32 percent

4.3.7 The amount of water in the saturated vapour of water-ammonia mixtures as a function of saturation pressure and saturated liquid conposition (C_w-P-X)

The series be written as

$$C_w = \sum_{i=o}^{K} \sum_{j=o}^{L} A_{ij} P_i(x) P_j(p)$$

A) P_{min} = 0.81655 Bars

P_{max} = 6.8048 Bars

X_{min} = 10.0 wt.percent

X_{max} = 50.0 wt.percent

The Chebyshev coefficients of the fit are:

4.4322E + 01	6.4803E + 00	-8.3306E - 01	3.0797E - 01
-3.1957E + 01	-3.6908E + 00	6.2418E - 01	-2.2386E - 01
1.3859E + 01	9.1089E - 01	-1.8953E - 01	9.9931E - 02
-4.1426E + 00	-6.8178E - 02	1.5439E - 02	-1.0640E - 02
8.1411E - 01	-7.4762E - 02	-8.1719E - 03	-2.1686E - 02

Maximum Error = ± 4.23 percent

B) P_{min} = 6.8048 Bars

P_{max} = 17.011 Bars

X_{min} = 10.0 wt.percent

X_{max} = 50.0 wt.percent

The Chebyshev coefficients of the fit are:

6.7780E + 01	5.4233 + 00	-4.3107E - 01	8.7446E - 04
-4.3722E + 01	-2.3697E + 00	2.6328E - 01	-5.7061E - 03
1.6273E + 01	3.5613E - 01	-4.9534E - 02	6.9108E - 03
-3.9853E + 00	1.3173E - 01	-2.1862E - 02	-7.5818E - 04
1.8745E - 01	-1.9827E - 01	3.5271E - 02	4.5584E - 04

Maximum error = ± 1.829 percent.

E) P_{min} = 17.011 Bars

P_{max} = 27.218 Bars

X_{min} = 50.0 wt.percent

X_{max} = 90.0 wt.percent

The Chebyshew coefficients of the fit are:

3.4699E + 00	4.4585E - 01	1.2187E - 02	-1.2318E - 03
-2.1525E + 00	-2.6951E - 01	-5.2767E - 03	5.0131E - 04
8.5657E - 01	9.1178E - 02	-8.7337E - 03	-2.1337E - 03
-3.2608E - 01	-2.5345E - 02	9.7675E - 03	-5.3357E - 03
9.7875E - 02	3.4599E - 03	-2.0249E - 03	2.9406E - 03

Maximum Error = $\pm$ 0.069 percent

Relevant equation for pure ammonia

Enthalpy of Ammonia Vapour (H_V)

$$H_V = 819.27971 - 7.149116E - 05P(T - 45652.165) + 11.6804322T$$

It fits accurately for pressure range 0.8 to 20.4 bar with a regression coefficient of 0.999.

Saturation pressure of ammonia as a function temperature (P-T).

$$P = \exp(11.2926) - 2600.6341/T - 25395.295/T^2)$$

It fits accurately for pressure range 0.8 to 20.414 bar with a regression coefficient of 0.998.

Enthalpy of Ammonia liquid

$$H_L = 1344.1126 + 4.9162453T$$

It fits accurately for temperature range 250K to 350 K with a regression coefficient of 0.978.

CHAPTER FIVE

THERMODYNAMIC ANALYSIS OF AMMONIA - WATER ABSORPTION HEAT PUMP SYSTEMS

5.1 INTRODUCTION

Most data representations available on ammonia-water absorption heat pump systems express the variation of the coefficient of performance of the systems as a functions of the components' temperatures. However, variation of some other factors such as recirculation ratio, reflux ratio and approach temperature of weak solution to absorbers also affect performances predictions. Some authors (Heppenstall, 1983) considered the approach temperature criterion in terms of heat exchanger effectiveness but this is not easily visualisable.

Most performance predictions assume, equal absorber and condenser temperatures. For the case of conventional absorption heat pump systems and equal generator and evaporator for the case of reversed absorption system. These cases are practical. For a proper characterisation of the above systems, variations in these parameters should be taken into account. On this thermodynamic analysis have been carried out for three variants of absorption system namely;

(1) Conventional absorption heat pump.

(2) Reversed absorption heat pump.

(3) Experimental conventional heat pump (The inclusion of some components like the rectifier makes the analysis of this system slightly different from the conventional systems in (1).

In the analyses, the following ranges of temperatures were assumed:

Conventional and Experimental System;

evaporator	- 20°C to 0.0°C
condenser	25°C to 45°C
absorber	25°C to 45°C
generator	60°C to 120°C

Reversed System;

evaporator	25°C to 45°C
condenser	- 15.0°C to 30°C
absorber	30°C to Maximum attainable
generator	25°C to 45°C

The reflux ratio, that is, the ratio of the mass flowrate of working fluid that returns to the rectifier from the condenser in the experimental system to that which continues to the evaporation, is varied from 0.1 to 0.3.

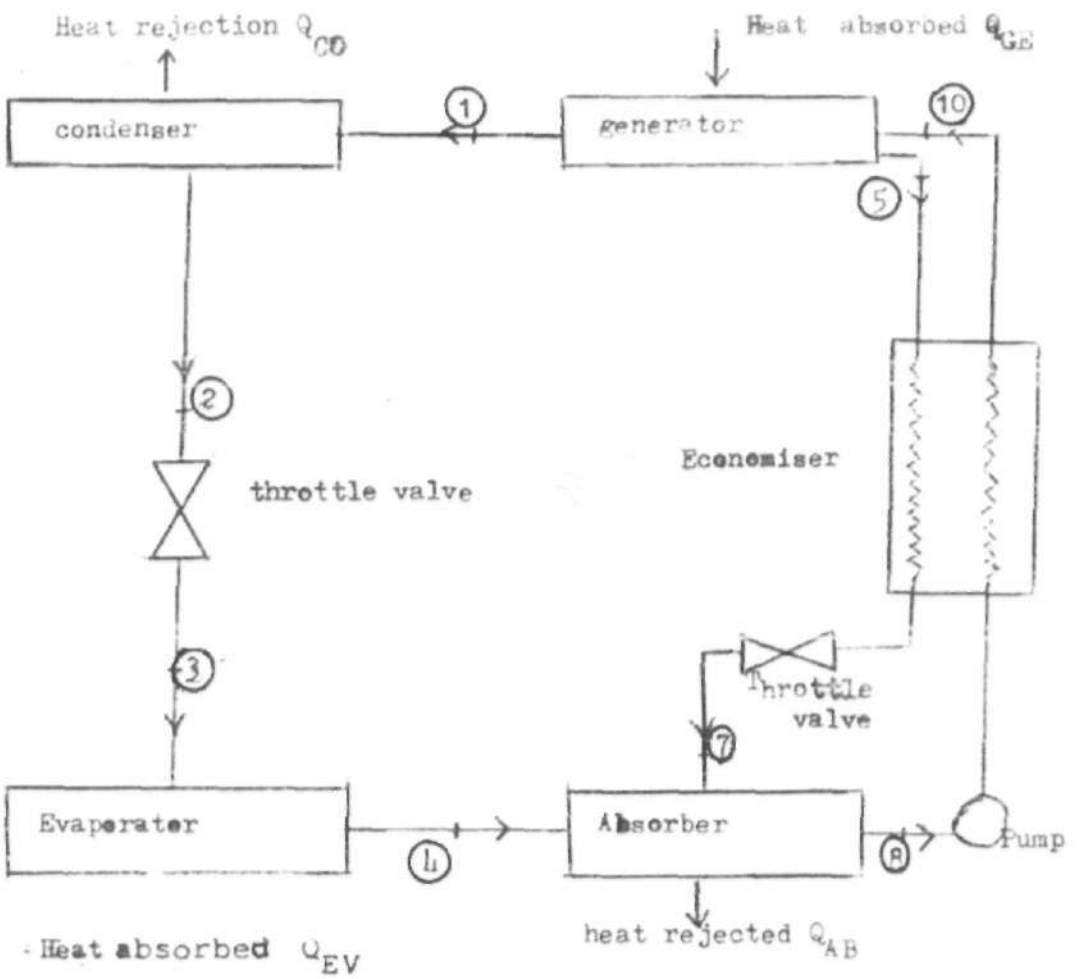


FIGURE 5.1 SCHEMATIC CIRCUIT FOR A CONVENTIONAL HEAT PUMP SYSTEM

5.2. *CONVENTIONAL ABSORPTION HEAT PUMP SYSTEM*

5.2.1. Theory

The absorption heat pump system is shown schematically in Figure 5.1. In the ammonia - water heat pump system, water is the absorbent with which it forms a liquid solution when it is mixed with sufficient ammonia working fluid. The solubility of ammonia in water is higher the lower the temperature of the mixture. Referring to Figure 5.1, between state points 4 and 8, vapour is passed into the absorber and dissolved in the ammonia-water mixture forming a solution which is "strong" in ammonia content. Heat is rejected from the absorber to the surroundings as the vapour is absorbed by the mixture. Cooling arrangements can be designed to serve the condenser and absorber simultaneously, but practical experience have shown that they are hardly at the same temperature. The liquid solution is then pumped up to the generator's and condenser's pressure and heated in two stages; first in a heat exchanger (economiser) which recoups some of the heat in the weak solution returning from the generator, and then in the generator itself. This causes the working fluid to boil off from the mixture, absorbing the latent heat of vaporation required from the environment at high temperature. Thus relatively high grade heat must be supplied to the generator to cause this to occur. The pure vapour then passes to the condenser.

The overall work input is small but the cycle is predominately motivated by the supply of heat to the generator.

5.2.2 Thermodynamic Considerations:

5.2.2.1 Carnot coefficient of performance:

An ideal coefficient of performance for an absorption heat pump system may be estimated by considering the system to be composed of two thermodynamically reversible machines which when taken together, perform the function of the absorption plant; (a) a reversible heat engine which receives a quantity of heat Q_{GE} at T_{GE} (Figure 5.1) (the generator's temperature) and rejects heat Q_{AB} at T_{AB} (the absorber temperature) whilst producing a quantity of work W_{GEAB}: The Carnot efficiency of this heat engine is given by

$$\eta = \frac{W_{GEAB}}{Q_{GE}} = \frac{T_{GE}-T_{AB}}{T_{GE}} \qquad (5.1)$$

and (b) a reversible heat pump which receives a quantity of eat Q_{EV} at T_{EV} (the evaporator temperature) and rejects heat at T_{AB} whilst absorbing a quantity of work W_{EVAB}. The coefficient of performance of this heat pump is given by (if cooling is the aim)

$$(COP)_c = -\frac{Q_{EV}}{W_{EVAB}} = \frac{T_{EV}}{T_{AB}-T_{EV}} \qquad (5.2)$$

The $(COP)_C$ of the combined plant acting as a heat pump can be defined as

$$(COP)_C = \frac{\text{Cooling load}}{\text{heat supply}} = \frac{Q_{EV}}{Q_{GE}} \qquad (5.3)$$

Substituting Q_{EV} and Q_{GE} from Equations (5.2) and (5.3), this $(COP)_C$ becomes

$$(COP)_C = - \frac{W_{EVAB}\ T_{EV}(T_{GE}-T_{AB})}{W_{GEAB}\ T_{GE}(T_{AB}-T_{EV})} \qquad (5.4)$$

and since no net work is actually performed,

$$W_{GEAB} = - W_{EVAB}$$

and so

$$(COP)_C = \frac{T_{EV}(T_{GE}-T_{AB})}{T_{GE}(T_{AB}-T_{EV})} \qquad (5.5)$$

The system does not contravene the first law of thermodynamics and consequently the heating coefficient of performance $(COP)_H$ can be expressed as

$$(COP)_H = (COP)_C + 1 \qquad (5.6)$$

These are the maximum theoretical values for the

coefficient of performance of any absorption heat pump:

These efficiency calculations are based on the first law of thermodynamics. With the application of the second law of thermodynamics the expression for the COP_C will be that expressed in Equation (5.4). This is because any process that actually occurs can do so only with some irreversibilities.

5.2.2.2. Maximum obtainable coefficient of performance.

The Carnot coefficient of performance gives the maximum value of efficiency for any heat pump. It takes into effect only the temperature levels in the system. It does not consider the working fluid, working fluid/absorbent mixture intrinsic properties and hard wares contributions to inefficiencies.

The maximum obtainable coefficient of performance evaluation is based on the Rankine cycle. Its definition is based on the validity of certain assumptions on the behaviour of the hardware and the fluid in the system. These assumptions can be summaried as follows: (Adefila, 1983), (Wilbur, 1970)

(1) Steady state and equilibrium conditions are assumed in the various

components of the system.

(2) The high side pressure of the system, that is the pressure in the condenser and generator is determined by the saturation pressure of the working fluid in the condenser.

(3) The low side pressure of the system, that is the pressure in the absorber and evaporator is determined by the saturation pressure in the evaporator.

(4) No temperature variation exist in the various components of the system.

(5) There is no change in enthalpy as liquid flow across valves or pumps.

(6) Heat exchange with the environment and pressure drops due to friction are negligible.

The mathematical relationship between the maximum obtainable coefficient of performance $(COP)_{AHMO}$ is found through mass and energy balances. The balances over the various components shown in Figure 5.1 are given by the

following equations. All mass flowrate M will be expressed as a fraction of the pure working fluid flowrate denoted by m. The flow ratio can be expressed in terms of either the mass flowrate of solution from the generator to the absorber M_{GE} or the mass flowrate of solution from the absorber to the generator M_{AB}

In the first case

$$(FR)_1 = \frac{M_{GE}}{M_{WF}}$$

which can also be written in terms of concentration as

$$(FR)_1 = \frac{1-X_{AB}}{X_{AB}-X_{GE}} \qquad (5.7)$$

In the second case,

$$(FR)_2 = \frac{M_{AB}}{M_{WF}}$$

which can also be written in terms of concentration as

$$(FR)_2 = \frac{1-X_{GE}}{X_{AB}-X_{GE}} \qquad (5.8)$$

In this project the second definition is used to express the flow ratio . The energy and mass balances assumes first law of thermodynamics.

$$Q_{EV} + Q_{GE} = Q_{AB} + Q_{CO} \tag{5.9}$$

$$M_1 = M_2 = M_3 = M_4 = M \tag{5.10}$$

$$M(H_1 - H_2) = Q_{CO} \tag{5.11}$$

$$H_1 - H_2 = q_{CO} \tag{5.12}$$

$$M(H_4 - H_3) = Q_{EV} \tag{5.13}$$

$$(H_4 - H_3) = q_{EV} \tag{5.14}$$

$$M + M_7 = M_8 \tag{5.15}$$

$$M_{8/M} = m_8 = F_R \tag{5.16}$$

$$m_7 = (FR) - 1 \tag{5.18}$$

$$- M_8 H_8 + MH_4 + M_7 H_7 = Q_{AB} \tag{5.19}$$

$$- (FR)H_8 + H_4 + ((FR)-1)H_7 = q_{AB} \tag{5.20}$$

$$m_{10} = m_5 + 1 \tag{5.21}$$

$$m_{10} = m_8 \tag{5.22}$$

$$m_5 = (FR) - 1 \tag{5.23}$$

$$m_{10} = m_8 = (FR) \tag{5.24}$$

$$- M_{10} H_{10} + M_5 H_5 + M_1 H_1 = Q_{GE} \tag{5.25}$$

$$- (FR)H_{10} + (FR)-1)H_5 + H_1 = q_{GE} \tag{5.26}$$

$$(FR)(H_5 - H_{10}) + H_1 - H_5 = q_{GE} \tag{5.27}$$

The coefficient of performance (COP) for the conventional absorption heat pump system for heating purposes is defined by the expression

$$(COP)_{AHMO} = \frac{Q_{AB} + Q_{CO}}{Q_{GE}} \tag{5.28}$$

Substituting

$$(COP)_{AHMO} = \frac{(FR)(H_8-H_4)+((FR)-1)H_7+H_1-H_2}{(FR)(H_5-H_{10})+H_1-H_5} \quad (5.29)$$

$$= \frac{H_4-H_7+(FR)(H_7-H_8)+H_1-H_2}{(FR)(H_5-H_{10})+H_1-H_5} \quad (5.30)$$

For cooling purpose the coefficient of performance is expressed as

$$(COP)_{ACMO} = \frac{Q_{EV}}{Q_{GE}} \quad (5.31)$$

$$= \frac{H_4-H_3}{FR(H_5-H_{10})+H_1-H_5} \quad (5.32)$$

Since the first law of thermodynamics is obeyed, then

$$1+ (COP)_{ACMO} = (COP)_{AHMO} \quad (5.33)$$

This can be proved thus, with reference to Figure 5.1

$$\frac{1+H_4-H_3}{(FR)(H_5-H_{10})+H_1-H_5} = \frac{H_4-H_7+(FR)(H_7-H_8)+H_1-H_2}{(FR)(H_5-H_{10})+H_1-H_5} \quad (5.34)$$

$$\Rightarrow \quad \frac{(FR)\ (H_5-H_{10}) - H_1 - H_5 + H_4 - H_3}{(FR)\ (H_5 - H_{10}) + H_1 - H_5} =$$

$$\frac{H_4 - H_7 + (FR)\ (H_7 - H_8) + H_1 - H_2}{(FR)\ (H_5 - H_{10}) + H_1 - H_5}$$

(5.35)

$$\Rightarrow (FR)\ (H_5- H_{10}) + H_1-H_5+H_4-H_3 = H_4-H_7+(FR)(H_7-H_8)+H_1-H_2$$

(5.36)

$$\Rightarrow \quad H_5(\ (FR)\ -1)-(FR)H_{10}-H_5 = (FR)(H_7-H_8)-H_2-H_7 \quad (5.37)$$

$$M_5H_5+M_8H_8 = M_{10}H_{10} +M_7H_7 \quad (5.38)$$

$$\Rightarrow \quad H_{10} = (1-1/(FR))\ (H_5 - H_7) + H_8 \quad (5.39)$$

Substituting Equation (5.39) in Equation (5.37) we have

$$H_5(FR)-H_5-H_5(FR) - (FR)H_7+H_7 - (FR)\ H_8-H_3$$

$$= \quad (FR)\ H_7-(FR)H_8-H_2-H_7 \quad (5.40)$$

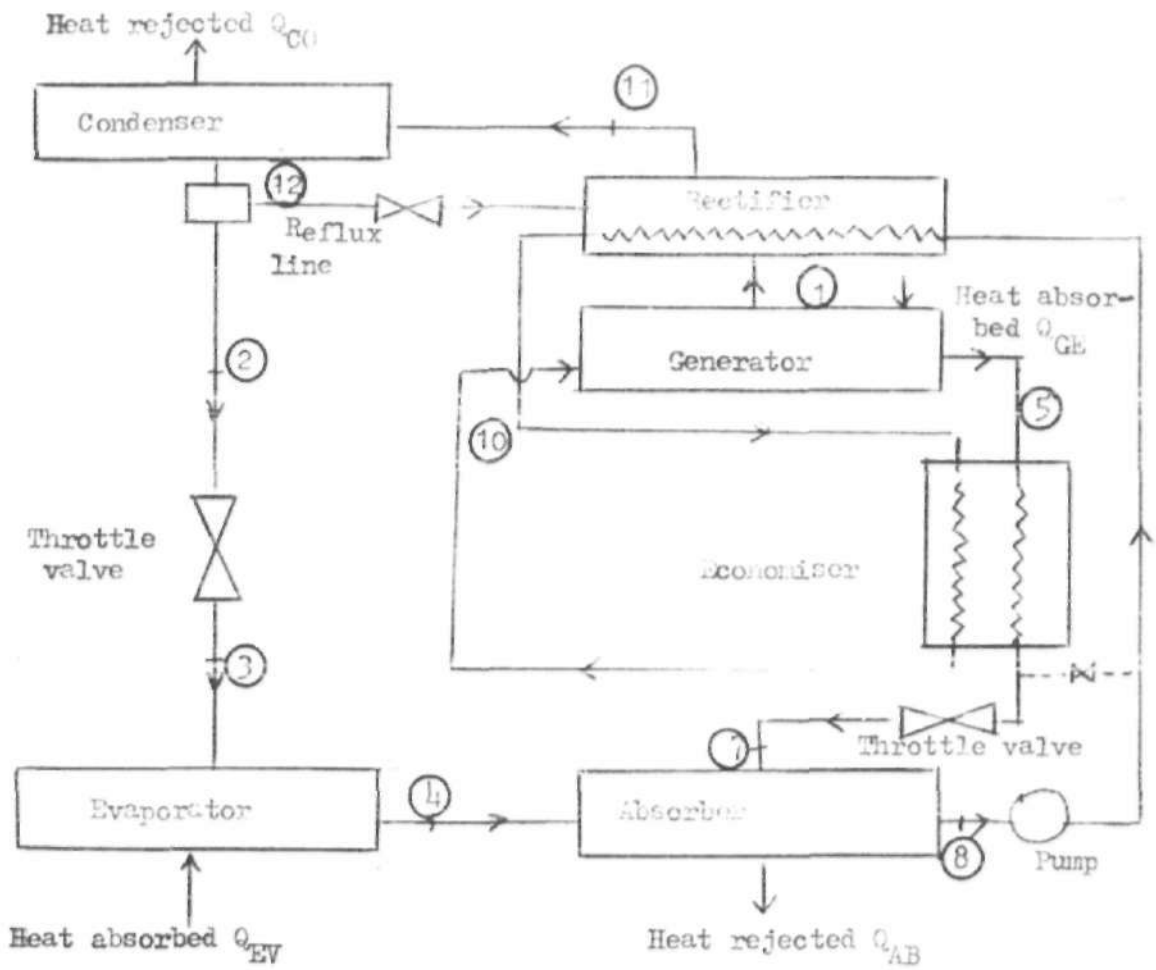


FIGURE 5.2 SCHEMATIC CIRCUIT OF EXPERIMENTAL ABSORPTION HEAT PUMP SYSTEM

$$\Longrightarrow H_3 = H_2 \qquad (5.41)$$

which is one of the assumptions for the derivation of the $(COP)_{AHMO}$ equation.

The values of $(COP)_H$ and $(COP)_C$ are always higher than $(COP)_{AHMO}$ and $(COP)_{ACMO}$ respectively for the same temperature levels. This is because there are some inherent inefficiencies which arise because of the working fluid and working fluid/absorbent mixtures which are not taken into account when $(COP)_H$ and $(COP)_C$ are calculated.

5.2.2.3. The practical coefficient of performance $(COP)_{AHA}$.

The practical coefficient of performance $(COP)_{AHA}$ is obtained from actual values of Q_{AB}, Q_{EV}, Q_{CO} and Q_{GE} recorded from the experimental system. This value are less than $(COP)_{AHMO}$ or $(COP)_H$ because it takes into account inefficiencies which arise because of the working fluid and the hard ware of the equipment.

5.3. EXPERIMENTAL ABSORPTION HEAT PUMP SYSTEM

The presence of the rectifier and a secondary circuit different from the conventional system entials that a different thermodynamic analysis be carried out for the experimental system. Figure 5.2. shows a schematic

diagram of the experimental system. To carry out the heat and mass balances across the equipment, the collector, disengager and rectifier have been merged and considered as a single component.

Taking heat balance over this component (with reference to Figure 5.2), we have

$$Q_{GE} = M_{11}H_{11}+M_7H_7-M_2H_2-M_{12}H_{12}-M_8H_8 \qquad \ldots(5.42)$$

but $M_{12} = RM_2$ and $M_{11} = (R+1)M_2$ ... (5.43)

based on pure working fluid flowrate

$$q_{GE} = \frac{(R+1)H_{11}+M_7H_7-H_2-RH_{12}+M_8H_8}{M_2} \qquad \ldots(5.44)$$

$M_8/M_2 = (FR) = m_8$

$M_{12}+M_{11}+M_7 = M_8$

or

$-RM_2+(+1)M_2+M_7 = M_8$

or

$-R+R+1+M_7 = (FR)$

or

$M_7 = (FR)-1$

Thus

$$q_{GE} = (R+1)H_{11}+((FR)-1)H_7-H_2-RH_2-FRH_8 \qquad \ldots(5.45)$$

$$\Rightarrow q_{GE} = (FR)(H_7-H_8)+R(H_{11}-H_2)+H_{11}-H_7-H_2 \qquad \ldots (5.46)$$

- Taking a heat balance across the absorber

$$M_4H_4+M_7H_7-M_8H_8 = Q_{AB} \qquad \ldots (5.46)$$

or

$$H_4+(FR)-1)H_7-(FR) = q_{AB} \tag{5.47}$$

- Taking a heat balance across the condenser

$$M_{11}H_{11}-M_2H_2-M_{12}H_2 = Q_{CO} \tag{5.48}$$

$$\Rightarrow (R+1)H_{11}-H_2-RH_2 = q_{CO} \tag{5.49}$$

- Taking a heat balance across the evaporator

$$M_4H_4-M_3H_3 = Q_{EV} \tag{5.50}$$

$$\Rightarrow H_4-H_3 = q_{EV} \tag{5.51}$$

The maximum obtainable Coefficient of performance for the experiemental system is expressed by the equation, (for cooling purposes)

$$(COP)_{ACMDE} = \frac{Q_{EV}}{Q_{GE}} = \frac{H_4-H_3}{(FR)(H_7-H_8)+R(H_{11}-H_2)+H_{11}-H_7-H_2} \tag{5.52}$$

for heating purposes

$$(COP)_{AHMDE} = \frac{Q_{AB}+Q_{CO}}{Q_{GE}} \tag{5.53}$$

$$= \frac{H_4+((FR)-1)H_7-(FR)+(R+1)H_{11}-H_2-RH_2}{(FR)(H_7-H_8)+R(H_{11}-H_2)+H_{11}-H_2)+H_{11}-H_7-H_2} \tag{5.54}$$

$$= \frac{H_4+(FR)(H_7-1)-H_7-R(H_{11}-H_2)+H_{11}-H_2}{(FR)(H_7-H_8)+R(H_{11}-H_2)+H_{11}-H_7-H_2} \tag{5.55}$$

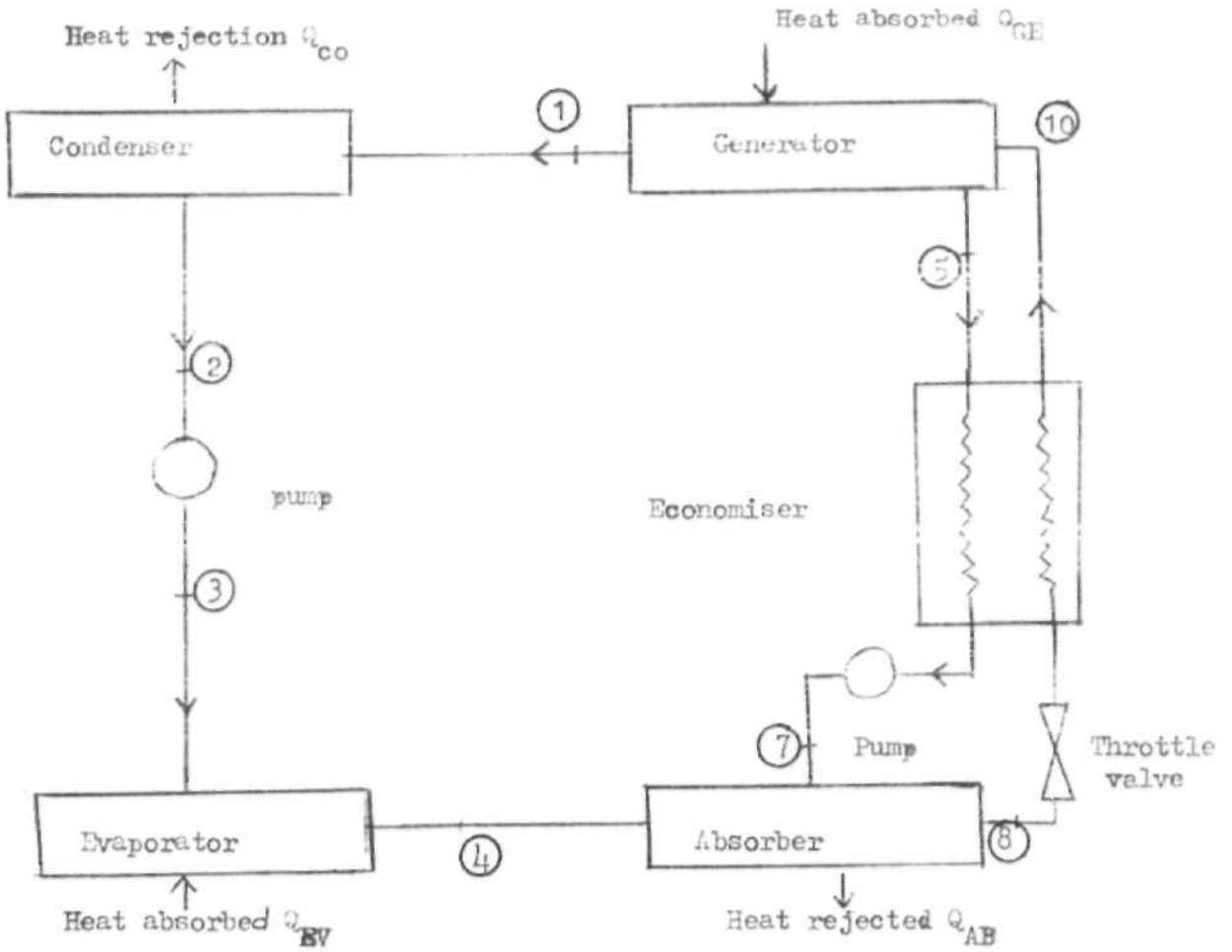


FIGURE 5.3 SCHEMATIC CIRCUIT FOR A REVERSED ABSORPTION HEAT PUMP SYSTEM

The presence of the reflux leads to a reduction in the cooling effect in the evaporator and thus values of $(COP)_{AHMOE}$ or $(COP)_{ACMOE}$ are lower than values of $(COP)_{AHMO}$ or $(COP)_{ACMO}$ respectively. However its inclusion in the experimental system ensures the reduction of carry-over of water into the condenser and a corresponding increase in the cooling effect in the evaporator. Thus there is always an optimum value for the reflux ratio which will balance these two effects and thus lead to an optimum experimental value for the coefficient of performance.

5.4. REVERSED ABSORPTION HEAT PUMP SYSTEM

The conventional absorption heat pump system and the reversed absorption heat pump system are similar with respect to components of equipment. They differ, however, in relative temperature and pressure levels of these components and also their arrangement. The reversed absorption heat pump system or heat transformer is shown schematically in Figure 5.3. Thermal heat, either from waste heat or from solar collector is taken up at a relatively low temperature T_{GE} by generator. The working fluid vapour generated by this effect is condensed in the condenser at a temperature T_{CO}. The liquefied working fluid leaving the condenser is pumped to the higher pressure region where it is evaporated by the thermal

waste heat or solar energy from collectors. The evaporated working fluid is then absorbed in the absorber at a higher temperature T_{AB}.

This system has the unique capability of raising the temperature of a working fluid above that of the input energy.

The Carnot coefficient of performance for the reversed heat pump system can be expressed as (Siddig-Mohammed et al, 1983)

$$(COP)_{CAH} = \frac{(1-\frac{T_{CO}}{T_{GE}})}{} \frac{(T_{AB})}{T_{AB}-T_{CO}} \qquad (5.56)$$

The actual coefficient of performance of the system is expressed as

$$(COP)_{AAH} = \frac{Q_{AB}}{Q_{GE}+Q_{EV}} \qquad (5.57)$$

The heat and material balances of the system are similar to that of the conventional absorption heat pump system.

The maximum obtainable coefficient of performance can be expressed by substituting Equation (5.20), (5.27) and (5.14) into Equation (5.57).

$$(COP)_{ARHMG} = \frac{(FR-1)+H_4-FRH_8}{(FR-1)H_5-FRH_{10}+H_1+H_4-H_2} \qquad (5.58)$$

Therefore the maximum obtainable coefficient of performance for the reversed heat pump system must be less than 1.0 (Siddig Mohammed et al, 1983). Practically, values hardly exceed 0.5.

The characteristics of the conventional and reversed absorption heat pump systems are highlighted in Table 5.1, for comparision.

TABLE 5.1 COMPARISON OF REVERSED AND CONVENTIONAL ABSORPTION HEAT PUMP CHARACTERISTICS.

	Reversed System	Conventional System
Purpose	Raising the temperature of working fluid above heat source	Lowering the temperature below heat source.
Temperature level	$T_{AB}>T_{GE}=T_{EV}>T_{CO}$	$T_{GE}>T_{CO}=T_{AB}>T_{EV}$
Pressure level	$P_{AB}=P_{EV}>P_{GE}=P_{CO}$	$P_{GE}=P_{CO}>P_{AB}=P_{EV}$
Heat absorption (working fluid)	At generator and condenser	At generator and condenser

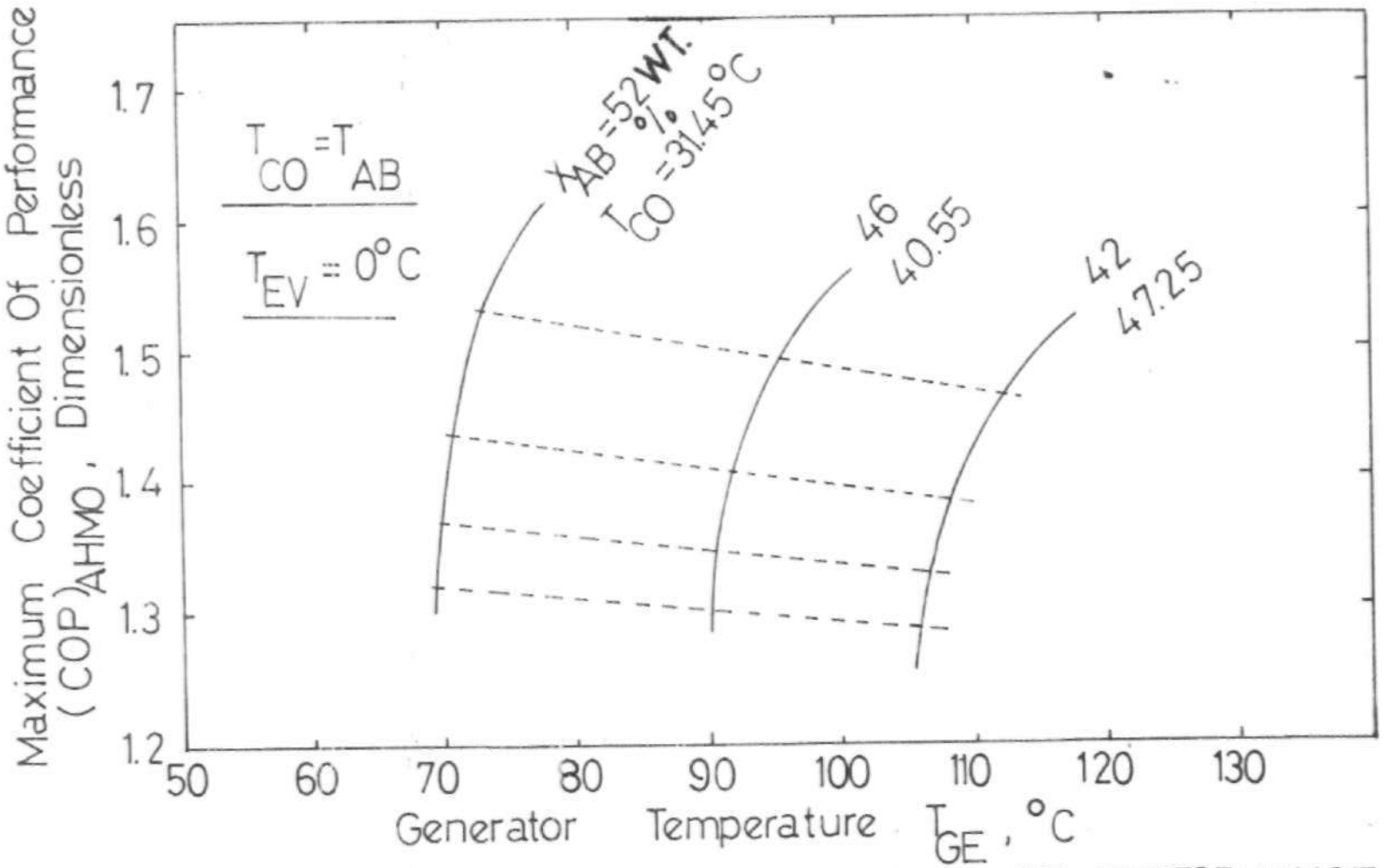


FIGURE 5.4 MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GENERATOR TEMPERATURE WITH ABSORBER WEIGHT PERCENT AND CONDENSER TEMPERATURE AS PARAMETERS

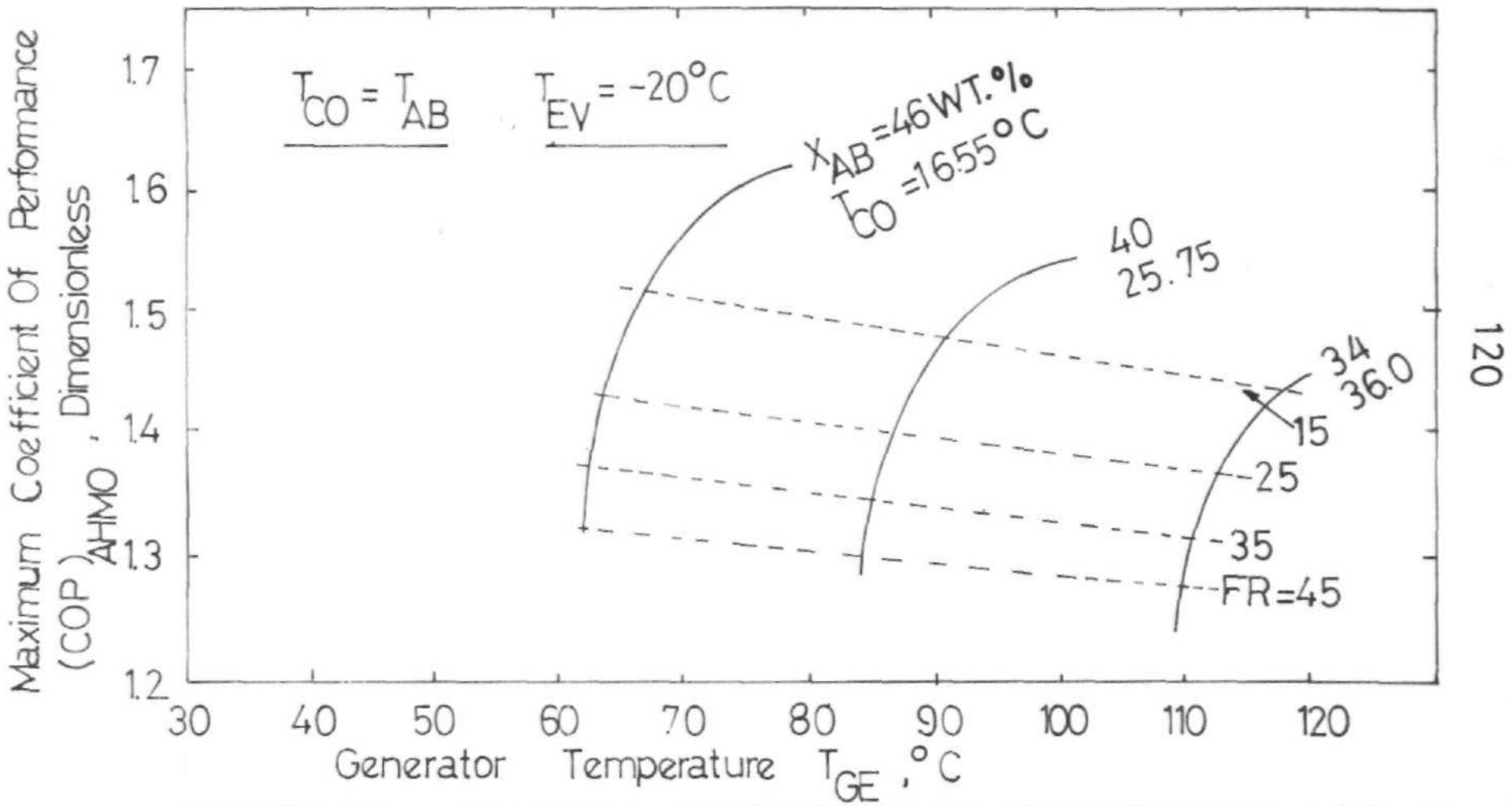


FIGURE 5.6 MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GENERATOR TEMPERATURE WITH RECIRCULATION RATIO CONDENSER TEMPERATURE AND ABSORBER WEIGHT PERCENT AS PARAMETERS

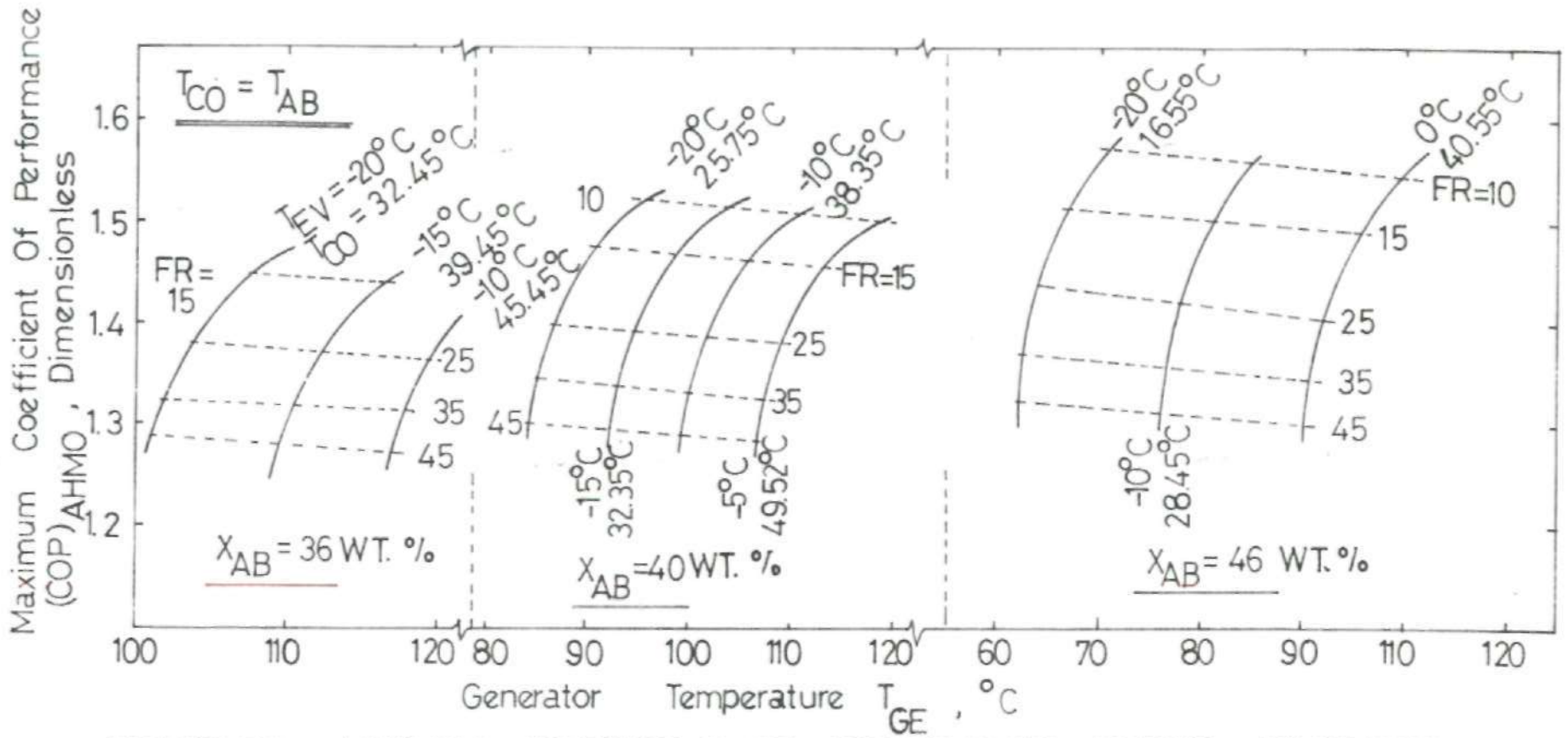


FIGURE 5.7 MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GENERATOR TEMPERATURE WITH EVAPORATOR TEMPERATURE, RECIRCULATION RATIO, CONDENSER TEMPERATURE AS PARAMETERS

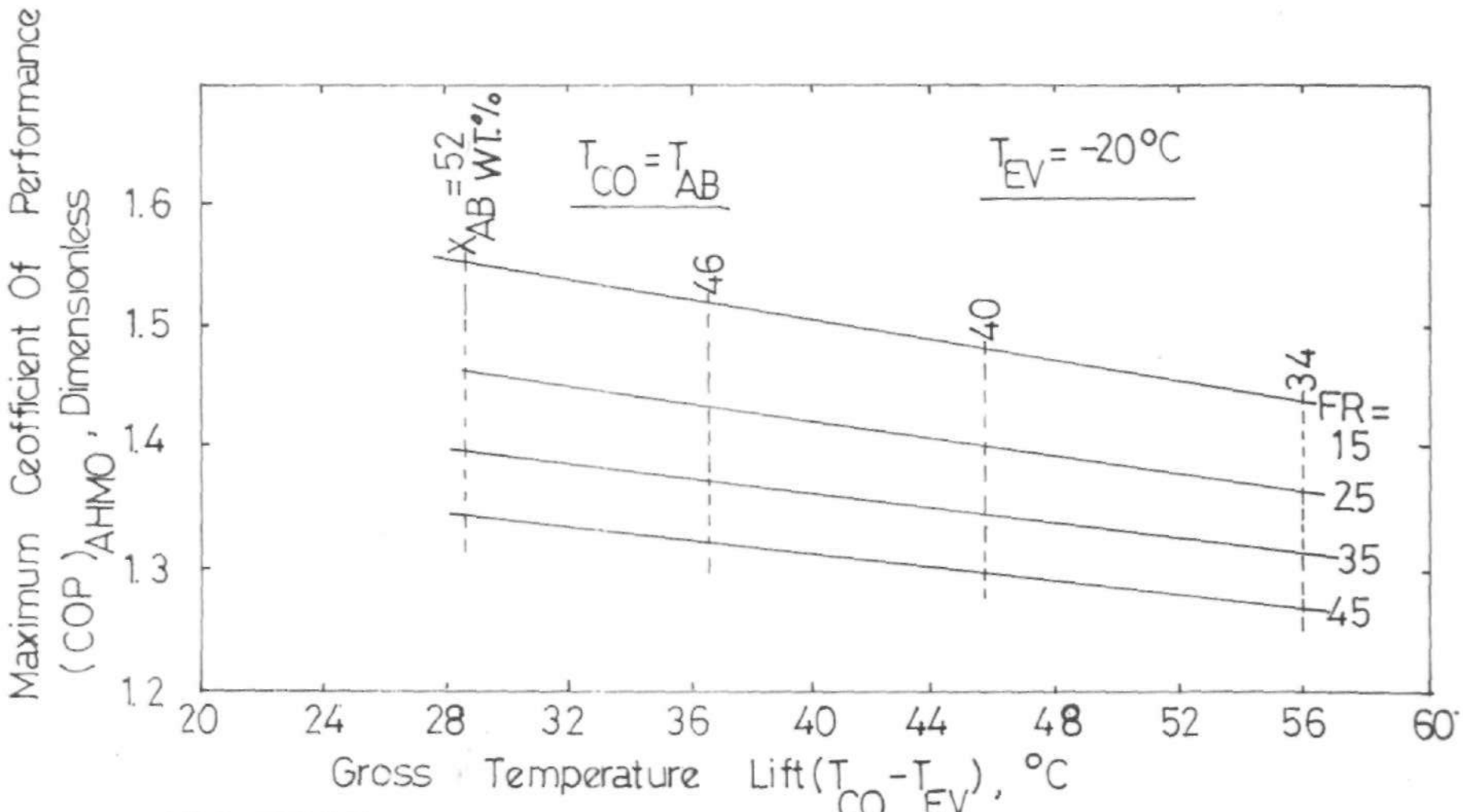


FIGURE 5.8C MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GROSS TEMPERATURE LIFT WITH RECIRCULATION RATIO AND ABSORBER WEIGHT PERCENT AS PARAMETERS

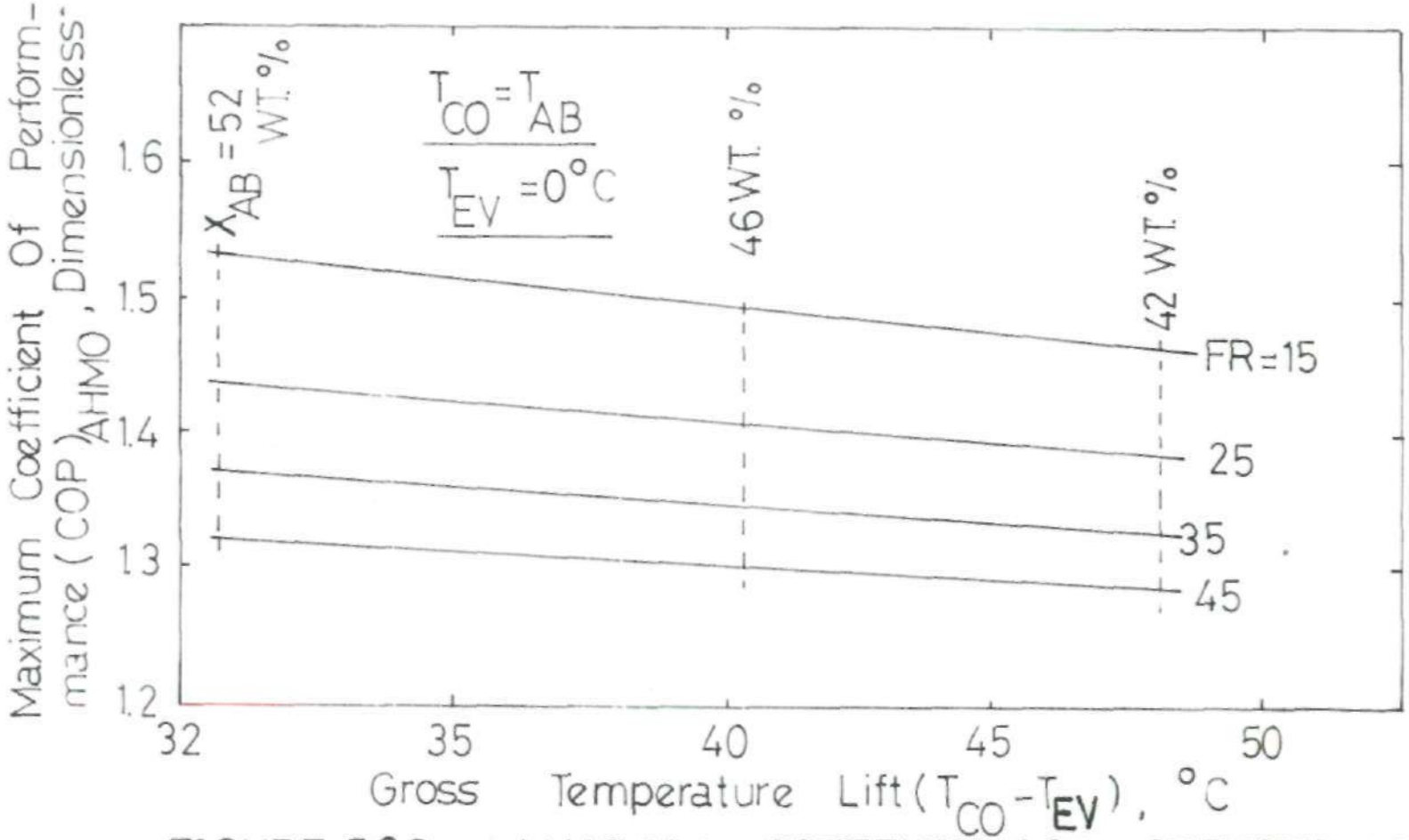


FIGURE 5.8C MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GROSS TEMPERATURE LIFT WITH ABSORBER WEIGHT PERCENT AND EVAPORATOR TEMPERATURE AS PARAMETERS

Table 5.1 continue;

Heat rejection (working fluid)	At absorber and condenser	At absorber and condenser
Component position	Between condenser and evaporator we have a pump	Between condenser and evaporator, a throttle valve exists
COP (Heating)	Greater than 1	Less than 1

5.5. COMPUTER SIMULATIONS

Computer programmes have been written to calculate $(COP)_{AHMO}$, $(COP)_{AHMOE}$ and $(COP)_{ARHMO}$. The relationships among the parameters were obtained from Chapter Four. The programmes are written in FORTRAN 4. A listing of the programmes are presented in Appendix (E)

5.6 DERIVED DATA ANALYSIS

5.6.1. Conventional Absorption Heat Pump System.

Figures 5.4 to 5.8. show the influence of the concentration of ammonia in water for the absorber, X_{AB} and condenser temperature T_{CO} on the plot of maximum

obtainable coefficient of performance against generator temperature, T_{GE}. The plots are produced assuming equal absorber and condenser temperatures. Observation of the figures show that the coefficient of performance is very sensitive to the generator temperature. For example, at an evaporator temperature of $0^{o}C$ and a X_{AB}/ T_{co} combination of 52 weight percent/$31.45^{o}C$, when the generator temperature is $70^{o}C$ the coefficient of performance is 1.32. With a $3^{o}C$ increase of the generator temperature, the coefficient of performance is 1.53, an increase of about 17 percent Such rapid increases can be visualised on the other plots. This suggests a need for a proper generator temperature control during operation.

For this purpose, the simulated outlet temperatures of the collector listed in Chapter Two is very handy. It is apparent from the plots that once a lower limit for the generator temperature is set, values of coefficient of performance become generally insensitive for higher values of generator temperature. Table 5.2 shows lower limits of Generator temperature for various ammonia weight percent in water for the absorber and condenser temperature combinations.

TABLE 5.2. GENERATOR TEMPERATURE LIMITS (LOWER) FOR VARIOUS ABSORBER CONCENTRATION/CONDENSER TEMPERATURE COMBINATIONS.

T_{EV}	X_{AB} / T_{CO} Combination	Lowest T_{GE} Limit For Adequate Performance
$0^{o}C$	52 Wt.% / $31.5^{o}C$	$80^{o}C$
	46 Wt.% / $40.55^{o}C$	$107^{o}C$
	42 Wt.% / $47.25^{o}C$	$125^{o}C$
$-10^{o}C$	52 wt.% / $19.65^{o}C$	$70^{o}C$
	46 wt.% / $28.45^{o}C$	$96^{o}C$
	40 wt.% / $38.37^{o}C$	$115^{o}C$
	36 wt.% / $45.45^{o}C$	$130^{o}C$
$-20^{o}C$	46 wt.% / $16.55^{o}C$	$80^{o}C$
	40 wt.% / $25.75^{o}C$	$105^{o}C$
	34 wt.% / $36.^{o}C$	$125^{o}C$

It is apparent from Table 5.2 that for lower evaporator temperatures, lower condenser temperatures are needed if we wish to operate at lower generator temperature. Most available cooling media are at temperatures above $20^{o}C$ and thus it will be difficult to operate at condenser

temperatures below 25^{o}C.

Thus, this leaves us wish the only alternative of operating at higher generator temperatures and lower absorber concentrations. Various solar augmentation methods have been presented in Chapter Two. Application of these solar augmentation methods will substantially increase the generator temperature.

From the various figures referred to above, it can be seen that the coefficient of performance is only weakly proportional to flow ratio and for sets of combination of X_{AB}/T_{AB}, directly and inversely proportional respectively. For an evaporator temperature of 0^{o}C and an absorber concentration and condenser temperature combination of 52 weight percent and 31.43^{o}C, increasing the flow ratio from 15 to 25 leads to a decrease of the coefficient of performance by only 6%. This percentage decrease is approximately maintained for other X_{AB}/T_{CO} combinations.

Figure 5.7 shows that, for a given absorber concentration, lower condenser temperature/evaporator temperature combinations leads to higher coefficient of performance. Furthermore, a given set of values of $(T_{CO} = T_{AB})/T_{GE}$ yield higher values of coefficient of performance at a penalty of lower evaporator temperatures.

Summarising the forgone observations from the analysis, indicate that a generator with fine temperature control and efficient absorber and condenser heat exchangers are required. The basic assumption is that the

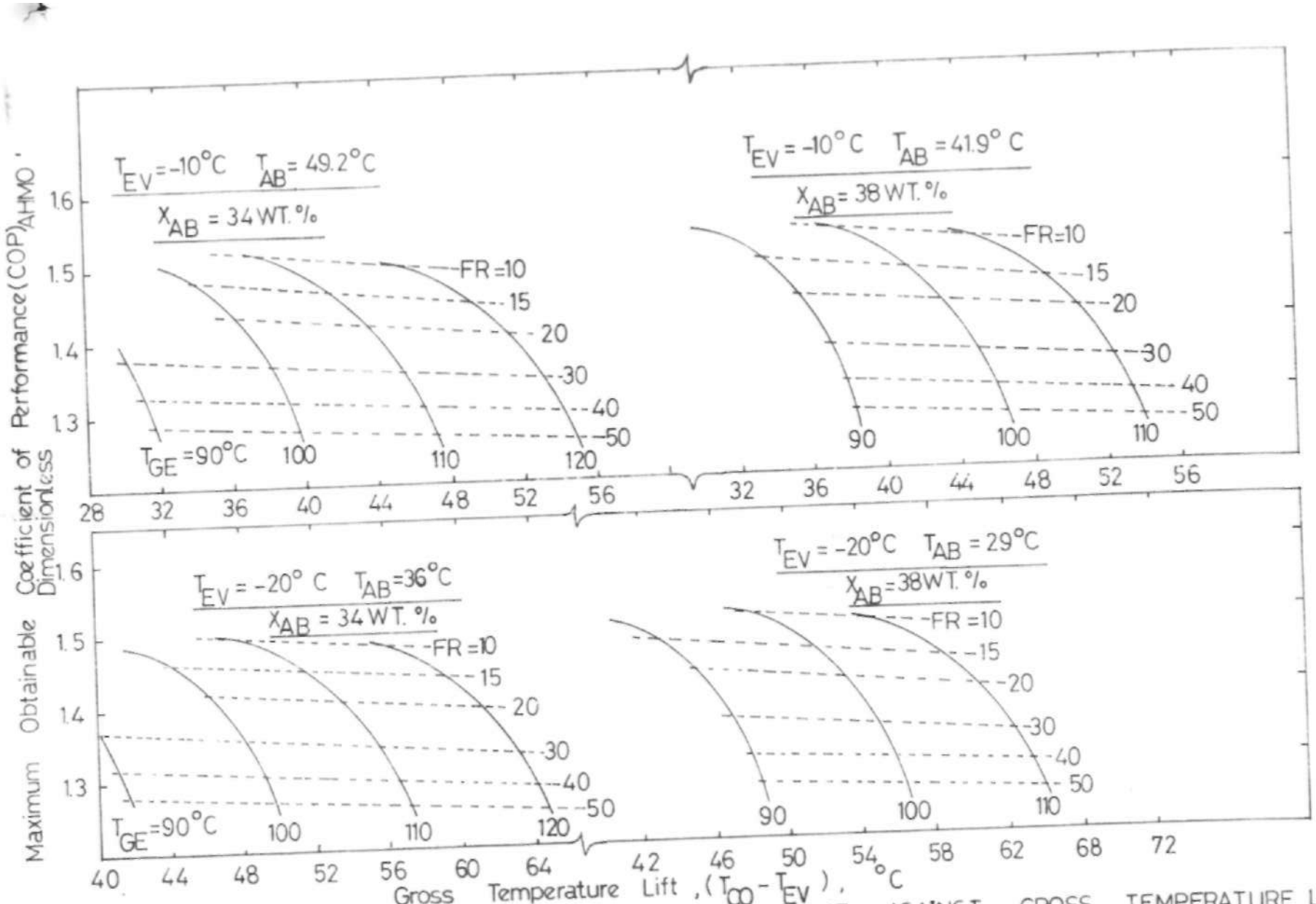


FIGURE 5.9 MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GROSS TEMPERATURE LIFT FOR VARIOUS ABSORBER CONCENTRATIONS AND EVAPORATOR TEMPERATURES

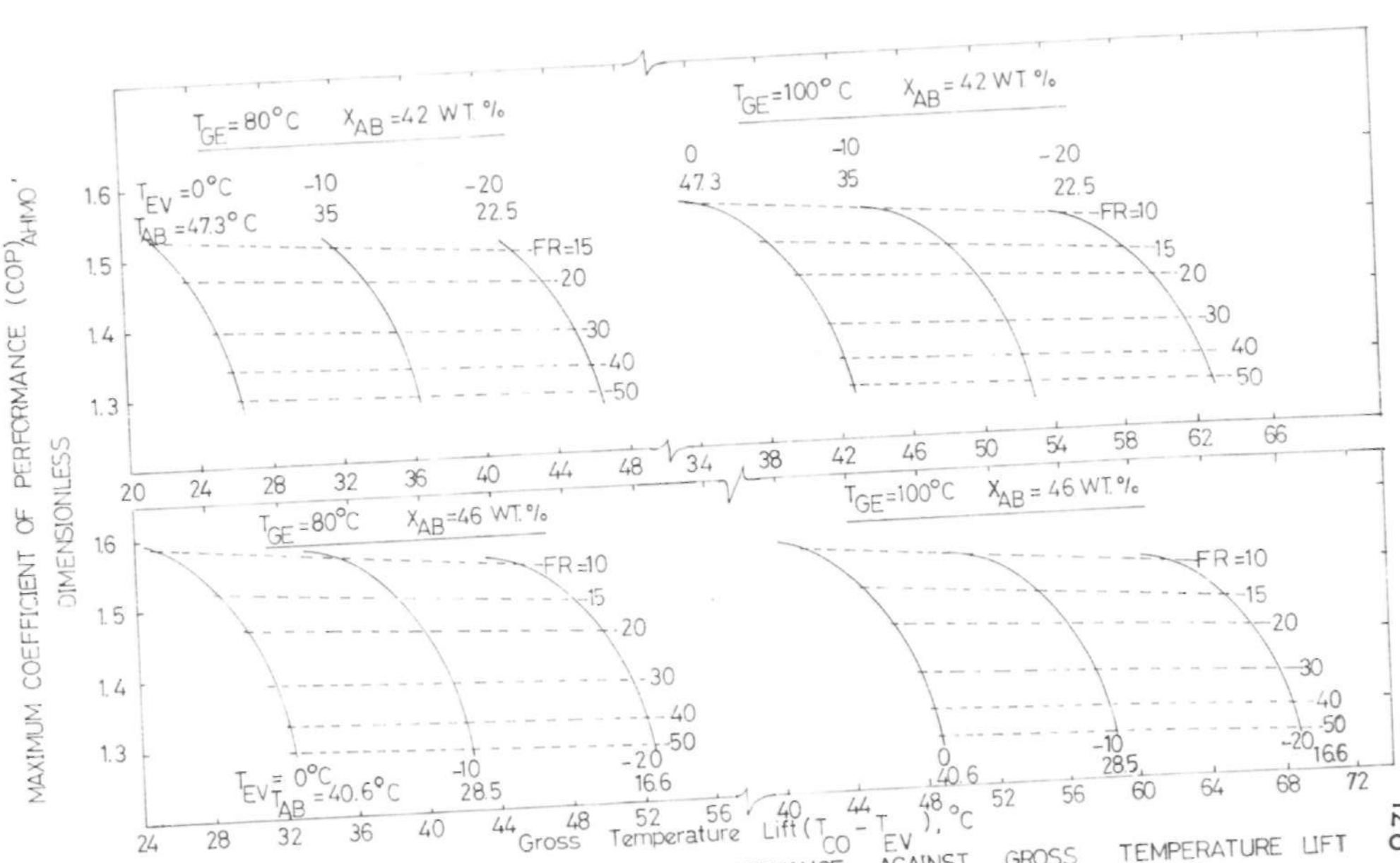


FIGURE 5.10 MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GROSS TEMPERATURE LIFT FOR VARIOUS GENERATOR TEMPERATURES AND ABSORBER CONCENTRATIONS

evaporator is amendable to required changes.

The above discussions assume equal absorber and condenser temperatures. However, Figures 5.9 and 5.10 show the effect of temperature lift for various absorber concentrations and conditions for unequal condenser and absorber temperatures. From these figure it can be seen that for a decrease in temperature lift, the coefficient of performance increases rapidly for given generator and absorber concentrations and tends to a maximum value. This goes to substantiate the fact already established that the absorber and generator are important components whose design must be properly carried out for a given heat load. Higher absorber concentrations yield higher temperature lift. For a given evaporator temperature, the absorber temperature decreases for an increase in absorber concentration.

All the forgone figures vis-a-vis discussions are based on an approach temperature to the absorber of $10^{o}C$. The effect of various approach temperatures are shown in Figure 5.11. The figure shows that a decrease of the approach temperature leads to a significant increase in the coefficient of performance. For example, at a generator temperature of $102^{o}C$ a decrease of the approach temperature from $10^{o}C$ to $5^{o}C$ leads to an increase in the coefficient of performance of about 10 percent. The percentage increase is approximately maintained for other generator temperatures. This emphasises the importance of

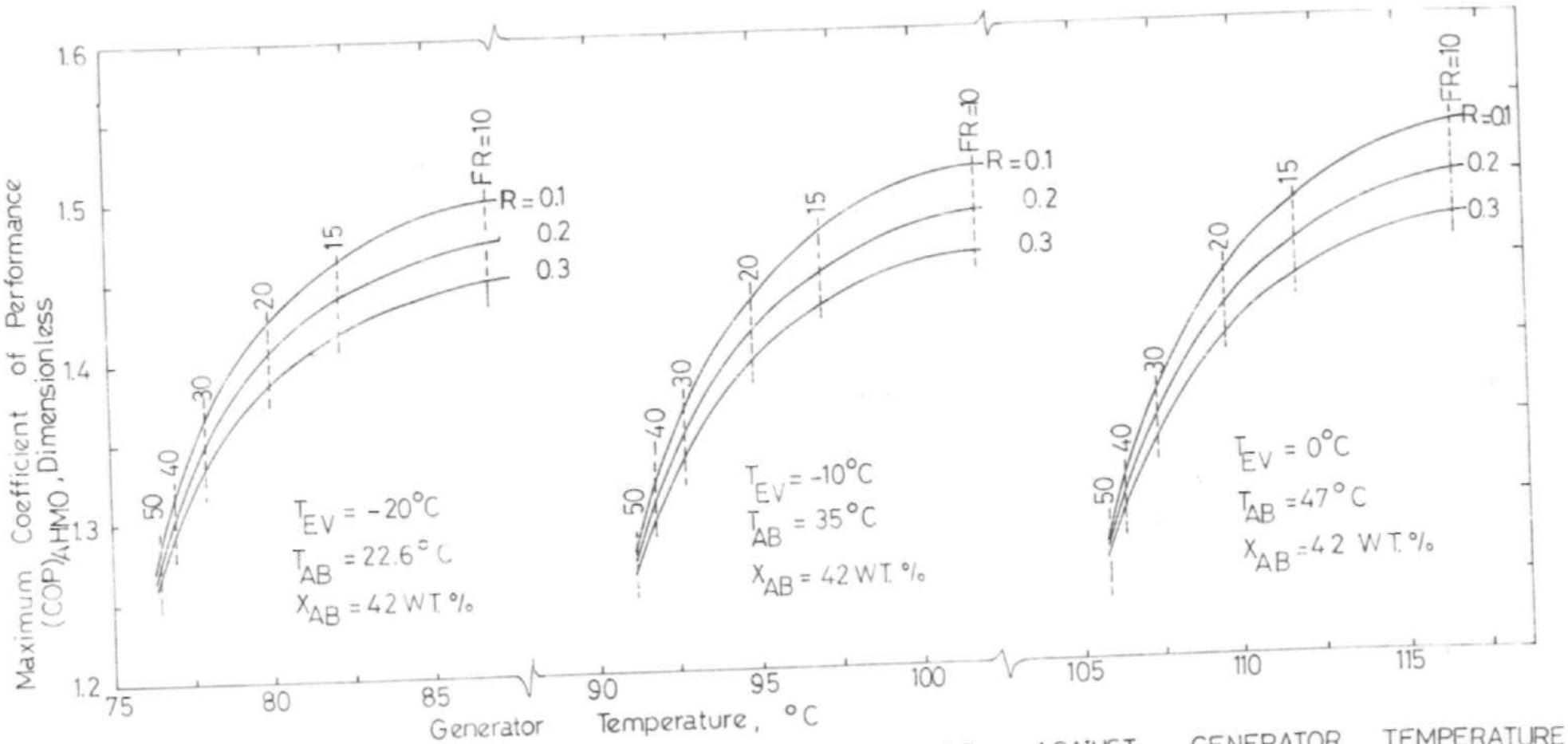


FIGURE 5.12 MAXIMUM COEFFICIENT OF PERFORMANCE AGAINST GENERATOR TEMPERATURE WITH REFLUX AND RECIRCULATION RATIOS AS PARAMETERS

proper lagging of the economiser to reduce heat losses to the environemnt.

5.6.2. Experimental Absorption Heat Pump System

Figure 5.12 shows the plot of coefficient of performance against temperature lift with reflux ratio (R) as parameter. It can be seen from the figure that the coefficient of performance is inversely proportional to reflux ratio. For example, a 5 percent decrease in the coefficient of performance results to an increase of the reflux ratio from 0.1 to 0.3. This quantitative relation is approximately maintained for different flow ratio and absorber concentrations. Although there is a decrease in performance with the inclusion of a reflux stream, it is however needed in the experimental ammonia - water absorption heat pump system if we are to get pure working fluid is the condenser. Water in the condenser leads to the lowering of the performance of the system since condensed water will eventually find its way into the evaporator and will reduce the cooling effect. It might even crystallise at sufficiently low evaporator temperature and block the evaporator tubes. Thus just enough reflux should be allowed to minimise the effect of the presence of water in the condenser/evaporator

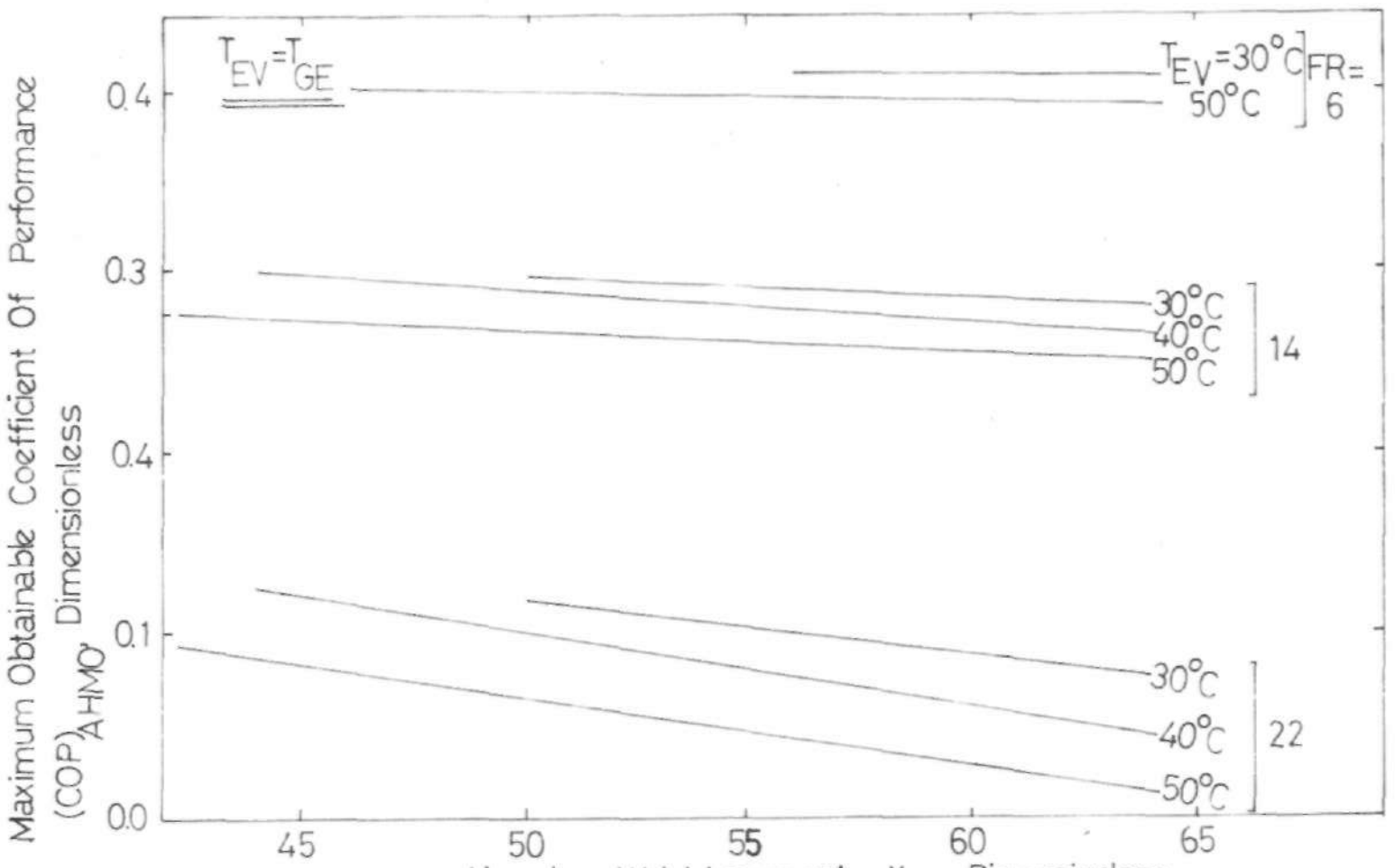


FIGURE 5.13 MAXIMUM OBTAINABLE COEFFICIENT OF PERFORMANCE AGAINST WEIGHT PERCENT OF AMMONIA IN ABSORBER WITH EVAPORATOR TEMPERATURE AND RECIRCULATION RATIO AS PARAMETERS.

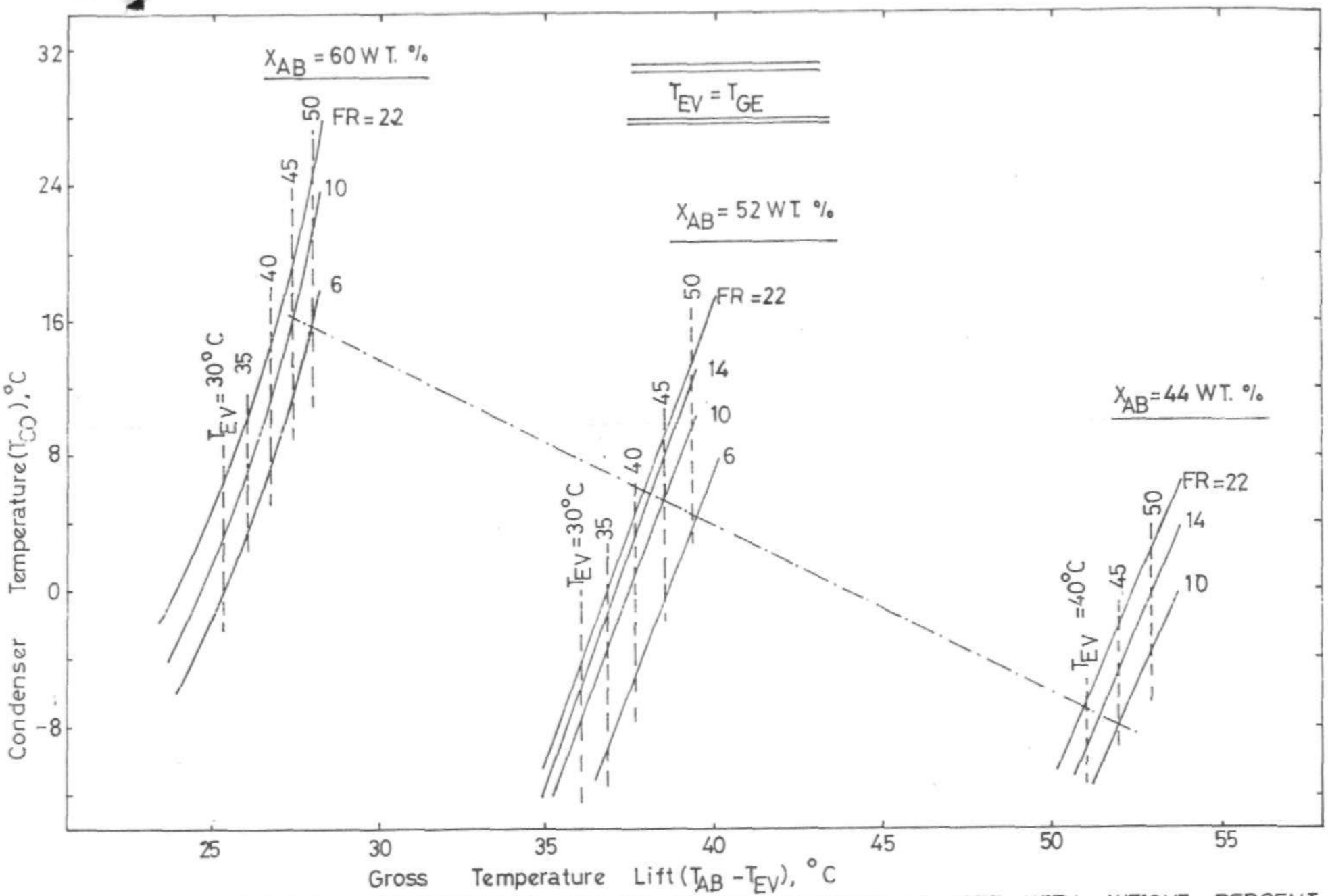


FIGURE 5.14 CONDENSER TEMPERATURE AGAINST GROSS TEMPERATURE LIFT WITH WEIGHT PERCENT OF AMMONIA IN ABSORBER, RECIRCULATION RATIO AND EVAPORATOR TEMPERATURE AS

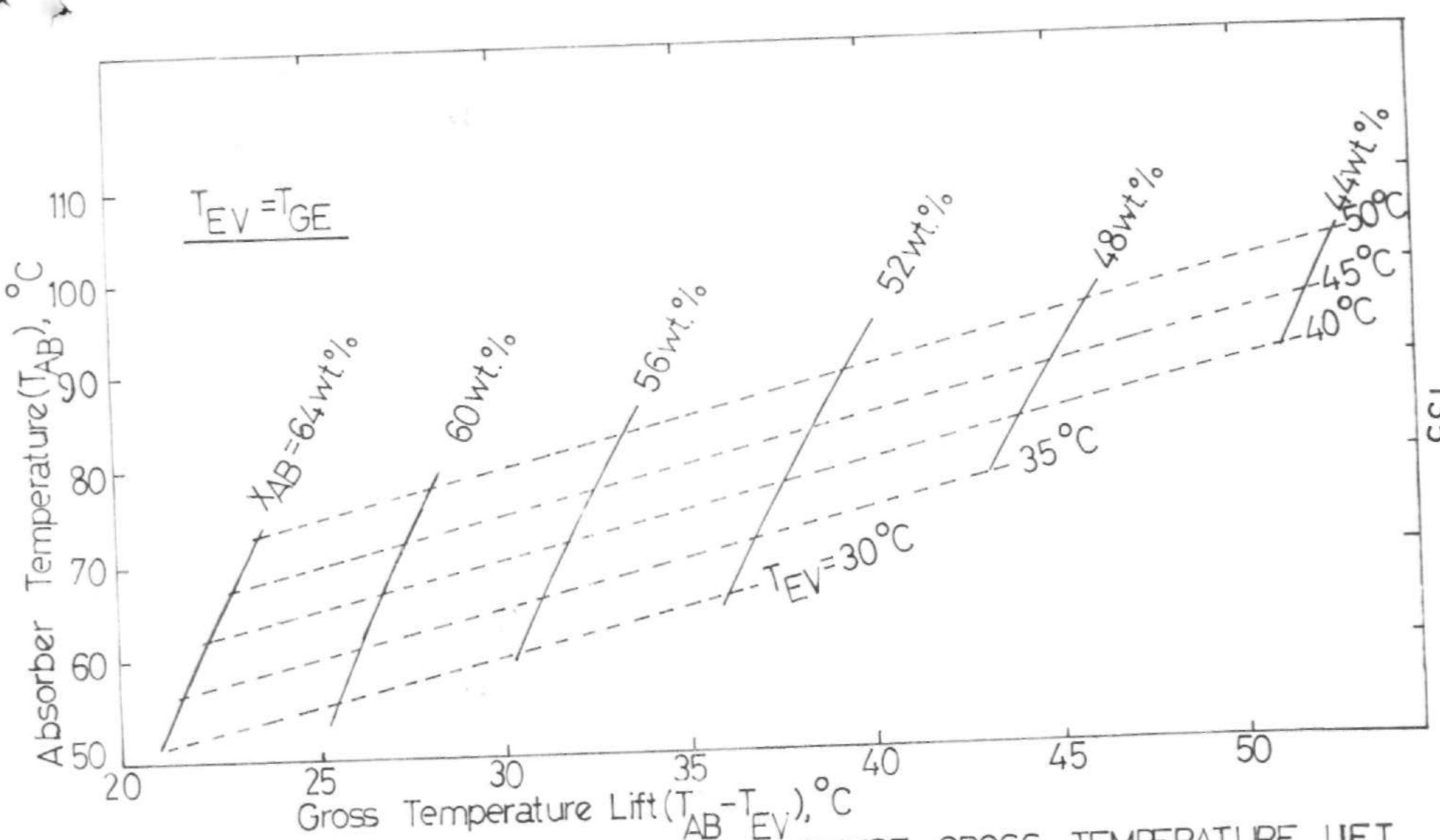


FIGURE 5.15 ABSORBER TEMPERATURE AGAINST GROSS TEMPERATURE LIFT WITH EVAPORATOR TEMPERATURE AND WEIGHT PERCENT OF AMMONIA IN THE ABSORBER AS PARAMETERS.

secondary circuit.

5.6.3 Reversed Absorption Heat Pump System

Figure 5.13 shows the influence of the concentration of ammonia in water for the absorber, X_{AB}, on a plot of absorber temperature against absorber concentration. It can be seen from the illustration that the flow ratio significantly influences the coefficient of performance. For an evaporator temperature of $30^{o}C$, an increase of the flow ratio from 6 to 14 leads a 30 percent decrease of the coefficient of performance. However for a given flow ratio the coefficient of performance does not vary significantly with changes in the evaporator temperature.

Figure 5.14 shows the effect of the values of X_{AB} on a plot of condenser temperature against gross temperature lift. From the plot it can be seen that, operating at lower values of X_{AB} require lower condenser temperatures. For a given value of X_{AB}, the condenser temperature increases weakly with an increase in flow ratio. The values of various condenser temperatures for given absorber concentrations can be obtained by interpolation using Figure 5.14

Figure 5.15 shows the influence of absorber concentrations on a plot of absorber temperature against generator temperature. It can be seen from Figure 5.15

that lower absorber concentration gives higher absorber temperature, but as the gross temperature lift increases, the required generator and evaporator temperatures also increase.

The practical operation of the reversed ammonia - water absorption heat pump system is very limited because of the low condenser temperature it requires to operate effectively. Very high concentrations lead to high pressure in the absorber since the temperature in this component is always very high. In the tropics the ambient and cooling water temperature hardly goes below 20^{O}C. Thus an observation of Figure 5.14 show that, indicated condenser temperatures (approximately equal to or below ambient) are impossible to be obtained except a cooling source for the condenser is cheaply available. In the temperate climate, where temperatures sometimes go below 20^{O}C, the condenser can be sufficiently cooled by placing it outside buried in ice during winter. The heat generated in the absorber can then be used to heat the environment.

The forgone analyses can be used to predict the performance of absorption heat pumps based on modified Rankine cycle, close to the practical cycles. On the basis of the comparision of such theoretical values with practical results, further design parameter may be obtained for specified conditions.

CHAPTER SIX

PERFORMANCE STUDIES FOR SOLAR- POWERED AMMONIA - WATER HEAT PUMP SYSTEM

6.1 INTRODUCTION

Existing storage facilities for agricultural products and other perishables in Nigeria are inadequate to meet the present demand. Almost 30 percent of these products are lost due to lack of suitable storage facilities. Almost all of the food production centres in Nigeria are in the rural areas. Preservation of the agricultural products could be achieved through inter -alia cooling. However most rural areas lack electricity for the running of large refrigeration plants. Therefore it seems that solar - powered heat pump systems for cooling may provide an effective solution for the provision of food preservation facilities in the rural areas. Besides such, the use of such systems enhance energy conservation.

Previous studies indicate that both the vapour compression and vapour absorption systems can be operated with solar energy. However solar - powered vapour compression heat pump systems have relatively low efficiency with corresponding high cost. The vapour absorption heat pump system is more amenable to running with solar energy because even a low energy source can substain its operations.

A lot of work has been carried out with the

intermittent type of absorption systems (Staicovici, 1986), (Farber, 1966). However, not much work has been presented on the solar- powered ammonia - water continuous system, when compared with solar- powered continuous lithium bromide system. This may be because the solution is very corrosive and very toxic. Further more, its operating pressure is usually very high. However, with the use of stainless steel parts and expert welding, these problems can be overcome.

In the foregone chapter, it has been shown that substantial cooling can be achieved by using ammonia - water heat pump systems. Generator temperature is always a limitation for solar- powered heat pump systems. However simiulated performance studies have indicated that cooling can be achieved with a generator temperature of as low as 65^{o}C. Suitable constructed flat plate collector can supply outlet temperatures as high as 120^{o}C, when there is high insolation. With the inclusion of some of the augmentation methods listed in Chapter Two, higher outlet temperatures can also be achieved during periods of low insolation.

Since theoretical predictions and work by other researchers reveal that substantial cooling can be achieved with the use of solar energy, a decison was made to design and develop an experimental solar - powered ammonia heat pump system, using locally available materials and local skills for the construction.

6.2 EQUIPMENT SPECIFICATIONS

The design of the system was carried out by Ajeboye (1986) and Bugage (1987). The material of construction of the heat exchangers is 1/4 inch galvanised iron. The specifications for the other components of the equipment are as follows:

6.2.1 Evaporator (shell and tube heat exchanger)

Capacity = 5.0 K watt

Number of tubes = 40

Tube diameter I.D = 0.0167m, O.D = 0.0213m

Tube length = 0.85m

Shell diameter I.D = 0.23m, O.D = 0.235m

Baffle spacing = 0.05m

Baffle number = 16

Baffle diameter = 0.205m

Working fluid inlet nozzle, I.D = 0.0167m

Working fluid outlet nozzle (2), I.D = 0.052m each

Tube plate diameter = 0.30m

Tube pitch; 0.025m, triangular

Bolts and nuts = 12, each 1 inch x 10mm

6.2.2 Flat Plate Collector

It is made up of two collectors. Each has the following specification:

Total collector area = $4.5m^2$ (3x1.5)

Tilt (towards south) = 11^{o}

Tubing diameter (I.D) = 16.7mm (1/2 inch)

Tubing spacing (centre to centre) = 50mm

Header diameter (I.D) = 52mm (2 inches)

Material of plate and tube = Galv. Steel

Material of insulation = fibre board

Thickness of insulation = 70mm

Cover material = corrugated fiber glass

Cover transmittance = 0.80

Plate absorptance, emittance = 0.95,0.95

Fluid inlet temperature = $65^{O}C$

Fluid outlet temperature = $110^{O}C$

Design efficiency of collector = 45%

6.2.3 Condenser (Shell and Tube Heat Exchanger)

Total condenser cooling load = 4.2k wett

Number of tubes = 10

Tube diameter I.D = 16.7mm, O.D. = 21.3mm

Tube length = 850mm

Tube pitch = 25mm triangular

Shell diameter = 102mm (I.D.)

Baffle diameter = 100mm

Baffle spacing = 50mm

Shell - side nozzle diameter = 16.7mm (I.D)

Tube - side inlet nozzle = 52mm (I.D.)

Tube - plate diameter = 150mm

Bolts and nuts = 6, each 1 inch x 10mm

6.2.4 Absorber (Shell and Tube Heat Exchanger)

Number of tubes = 18

Tube length = 800mm

Tube diameter;I.D = 16.7mm O.D. = 21.3mm

Tube pitch: 26mm triangular

Shell diameter = 152mm (1.D.)

Baffle diameter = 150mm

Baffle spacing = 50mm

Tube - plate diameter = 200mm

Shell side nozzle diameter = 27mm (1.D.)

Tube- side solution inlet nozzle = 27mm (1.D.)

Tube - side solution outlet nozzle = 27mm (1.D.)

Packing inside the tubes: rasching rings 4mm

diameter and length.

Bolts and nuts = 6; each 1 inch x 100mm

6.2.5 Rectifier (Shell and Tube Heat Exchanger)

Number of tubes = 7

Tube length = 500mm

Tube diameter; I.D. = 16.7mm; 0.D. = 21.3mm

Tube pitch = 26mm triangular

Shell diameter = 101mm (1.D.)

Baffle diameter = 99mm

Baffle spacing = 50mm .

Tube - plate diameter upper = 150mm lower = 200mm

Tube - side outlet nozzle = 22mm (1.D.)

Shell - side nozzle = 27mm (1.D.)

Bolts and nuts = 6; each 1 inch x 10mm

6.2.6 Economiser (Double - Pipe Heat Ecxchanger)

Centre pipe diameter 1.D. = 38mm 0.D. = 42mm

Outer pipe diameter 1.D. = 70mm 0.D. = 74mm

Total length of outer pipe = 4000mm

Total length of centre pipe = 4900mm

Number of loops = 4

Outer pipe nozzles (1.D.) = 27mm

Centre pipe nozzles (1.D.) = 27mm.

6.2.7 Auxillary components

- Gasket Material; Asbestos Sheets

The quantity of asbestos sheets was not enough for "gasketing" so the little available sheets were used as gasket material for the rectifier and condenser (high pressure components). Rubber lined with polythene was used for the absorber and evaporator. Lining of the rubber was necessary because ammonia attacks rubber. Complete isolation of the rubber from ammonia could not be guaranteed since cut edges of the polythene sheets could not be completely sealed.

- Thermocouple: Iron - Constantin
- Pressure gauge material; stainless steel.

Stainless steel gauges could not be found locally, so brass guages were used. Brass, because it

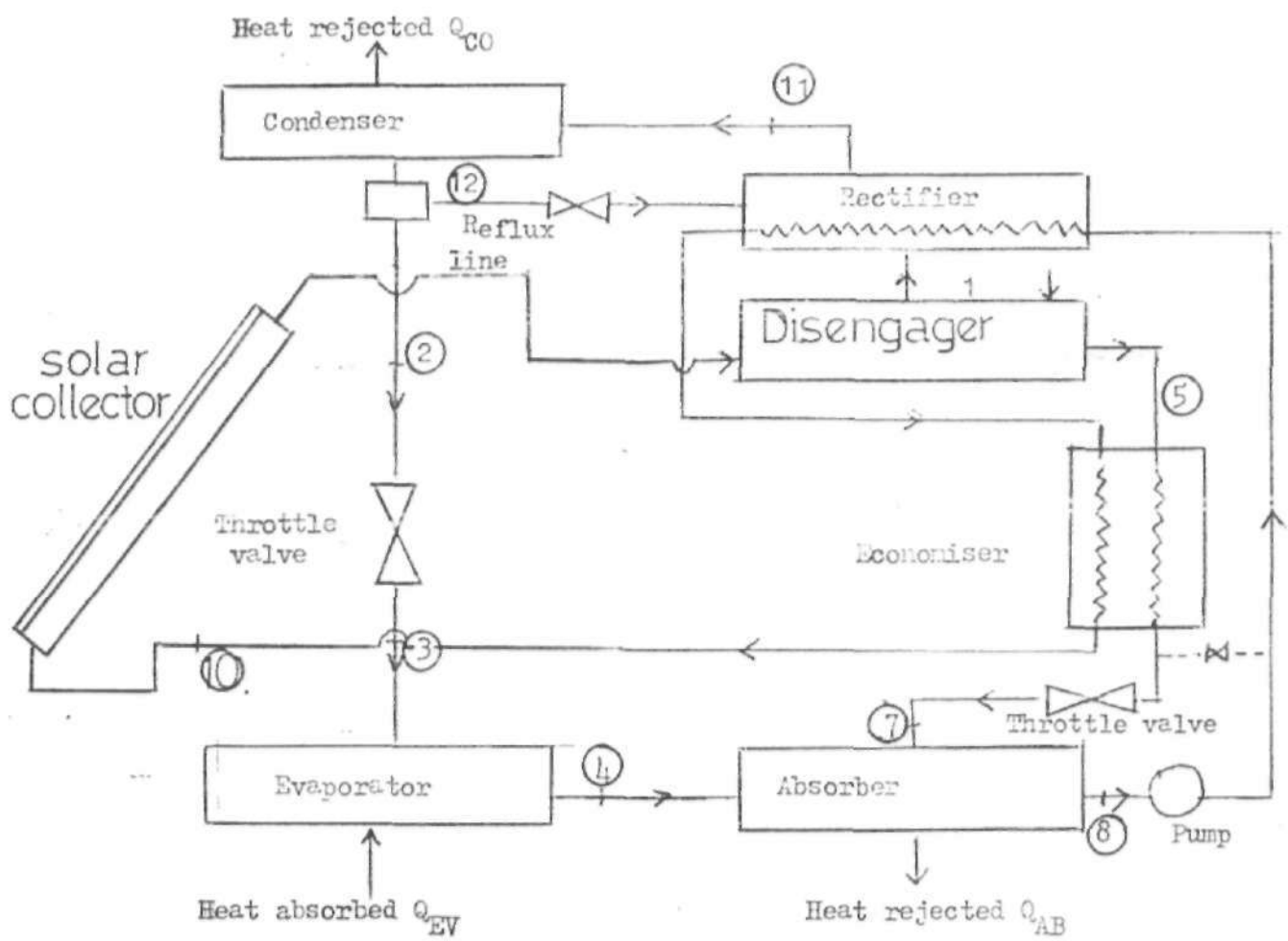


FIGURE 6.1 SCHEMATIC CIRCUIT OF EXPERIMENTAL ABSORPTION HEAT PUMP SYSTEM.

contain Copper reacts with ammonia explosively and so was only used as a temporal alternative for just the testing of the system. Eventual damage of these guages was envisaged.

- Lagging material; polyurethene.

It should be noted that although galvanised iron was used for the fabrication of the heat exchangers, the most compatible material of construction for the ammonia - water systems in literature is stainless steel. Locally available stainless steel tubes and shells were too expensive for the available level of funding for the project. However it is expected that the objective of the project could be achieved within the expected life span of the present alternative. Figure 6.1 shows a schematic diagram for the set up.

6.3 COMMISSIONING THE EQUIPMENT

A most important factor for effective operation of the system is a tight seal. This was monitored and necessary checks were effected as follows:

- The system was evacuated while all the outlet valves were shut.
- Leakages were indicated by unstable pressure guage readings.
- Leakages were isolated using soap solution, marked and subsequently checked by rewelding.

The above process was continued until a perfectly sealed system was obtained.

The following procedure was followed during the charging process:

- The system was evacuated using a vacuum pump
- A cylinder of pure ammonia was connected through an inlet between the condenser and the throttling valve before the evaporator. Since the pressure of ammonia increases substantially with temperature, the equipment was charged in the early hours of the day at low ambient temperature.
- Water was allowed into the evaporator's and absorber's shell side as heating and cooling sources respectively. The pump was switched on at the same time.
- All valves were re-examined and the system was allowed to stabilise with the increasing insolation of the day.
- The collector outlet temperature was monitored to know when a substantial temperature differential existed across the component.
- With a substantial increase in temperature achieved, water was allowed into the condenser shell side.
- An increase in the condenser shell side

water temperature meant that condensation was taking place.

- The system was allowed to stabilised for about 30 minutes.

6.4.1 Results

The results of the single operation are presented in Table 6.1, 6.2 and 6.3. It was not possible to monitor the flowrate of the working fluid and absorbent/working fluid since flow meters were not available. Even the measurement of pressure was limited to the early periods of the start-up because of the incapacitation of the pressure gauges due to corrosion.

TABLE 6.1 IRRADIANCE VALUES FOR THE VARIOUS TIMES OF THE DAY

TIME (HOURS)	IRRADIANCE (Wm^{-2})	TIME (HOURS)	IRRADIANCE (Wm^{-2})	TIME HOURS)	IRRADIANCE Wm^{-2}
9.00	315.0	11.05	472.0	13.10	502.6
9.05	336.0	11.10	483.0	13.15	502.6
9.10	340.0	11.15	485.0	13.20	516.6
9.15	344.0	11.20	489.0	13.25	210.0
9.20	347.2	11.25	494.0	13.30	501.2
9.25	350.0	11.30	497.0	13.35	503.0
9.30	368.2	11.35	492.0	13.40	502.0
9.35	378.0	11.40	495.0	13.45	504.0
9.40	382.2	11.45	497.0	13.50	501.0
9.45	386.0	11.50	497.0	13.55	483.0
9.50	392.0	11.55	497.0	14.00	480.0
9.55	399.0	12.00	497.0	14.05	476.0
10.00	410.0	12.05	497.0	14.10	470.0
10.05	420.0	12.10	497.0	14.15	477.0
10.10	427.0	12.15	497.0	14.20	462.0
10.15	434.0	12.20	501.0	14.25	455.0
10.20	448.0	12.25	510.0	14.30	462.0
10.25	445.0	12.30	516.0	14.35	434.0
10.30	448.0	12.35	516.0	14.40	431.2
10.35	455.0	12.40	512.0	14.45	434.0
10.40	458.0	12.45	509.0	14.50	420.0
10.45	458.0	12.50	507.0	14.55	406.0
10.50	460.0	12.55	504.0	15.00	392.0
10.55	462.0	13.00	490.0	15.05	390.0
11.00	466.0	13.05	498.4	15.10	387.0

TABLE 6.2 WORKING FLUID - SOLUTION TEMPERATURES FOR THE DAY

TIME (HOURS)	T_{EV} INLET ^{o}C	T_{EV} OUT-LET ^{o}C	T_{CO} IN-LET ^{o}C	T_{AB} OUT-LET ^{o}C	T_{DIS} ^{o}C	T_{SSFR} ^{o}C	T_{GE} IN-LET ^{o}C	T_{GE} OUT-LET ^{o}C
9.00	3	-3	36	45	30	30	30	32
10.00	2	-2	40	48	35	37	32	37
10.30	2	-6	45	55	38	40	40	50
11.00	3	-5	45	56	42	40	49	68
11.30	-2	-7	45	57	46	42	50	79
12.00	-1	-7	47	60	50	45	51	80
12.30	3	-5	48	54	52	48	56	86
13.00	3	-2	43	55	64	48	57	91
13.30	4	-2	40	50	53	49	60	90
14.00	8	3	36	48	52	47	59	87
14.30	13	7	38	46	50	46	59	83
15.00	15.0	10	34	40	50	42	58	80

TABLE 6.3 COOLING AND HEATING WATER; AMBIENT TEMPERATURES AND FLOWRATES FOR VARIOUS TIMES OF THE DAY

TIME	P_E	P_{CO}	T_{AMB}	WATER IN SHELL SIDES			FLOWRATE OF WATER kg/min		
	bars (gauge)	bar (gauge)	°C	T_{INLETS}	T_{EV} OUT	T_{AB} OUT	T_{CO} OUT	F_{EV}	F_{AB}	F_{CO}
9.00	2.7	-	23.0	22.0	16.0	35.0	23.0	1.0	1.0	0.7
9.30	5.8	7.0	25.0	25.0	15.0	37.0	30.0	1.0	1.0	0.7
10.00	5.8	(faulty)	26.0	25.0	9.0	44.0	36.0	1.0	1.0	0.8
10.30	-	-	28.0	26.0	4.0	50.0	37.0	0.8	1.1	0.7
11.00	-	-	30.0	27.0	4.0	50.0	37.0	0.6	1.1	0.7
11.30	-	-	32.0	27.0	2.0	45.0	36.0	0.4	1.0	0.7
12.00	-	-	33.0	26.0	2.0	43.0	38.0	0.5	0.8	0.7
12.30	-	-	34.0	28.0	3.0	45.0	40.0	1.4	1.2	0.8
13.00	-	-	35.0	28.0	7.0	46.0	35.0	1.0	1.6	0.8
13.30	-	-	35.0	29.0	10.0	44.0	36.0	0.8	1.6	0.7
14.00	-	-	33.0	29.0	18.0	42.0	35.0	0.9	1.6	0.7
14.30	-	-	32.0	28.0	20.0	36.0	33.0	0.8	1.6	0.6
15.00	-	-	30.0	28.0	25.0	36.0	29.0	0.8	1.6	0.7

TABLE 6.4 RELEVANT EQUATION USED FOR DATA DERIVATIONS

GENERATOR $I_H = \int_0^{3600} H_H$ (Simpson Rule)

$$I_o = \frac{(12)(3600)G_{sc}(1+0033\cos\frac{360n}{365})\times}{\Pi}$$

$$(\cos\phi\cos\delta\sin\omega_2 - \sin\omega_1)+(\frac{2\Pi}{360}\times$$

$$(\omega_2-\omega_1)\sin\phi\sin\delta)$$

$$K_T = I_H/H_o$$

$$\frac{I_d}{I_H} = \begin{cases} 1.0-0.249K_T & \text{For } K_T<0.35 \\ 1.557-1.84K_T & \text{for } 0.35<K_T<0.75 \\ 0.177 & \text{for } K_T>0.75 \end{cases}$$

$$I_b = I_H/H_o$$

$$I_T = I_bR_b+I_d\frac{(1+\cos\beta)}{2}+(I_b+I_b)\rho\times\frac{(1-\cos\beta)}{2}$$

$$Q_{Ge} = A_c\eta I_T\alpha\tau$$

EVAPORATOR $Q_{EV} = M_{EVW}\ C_{PW}(T_{EVW-IN}-T_{EVW-OUT})$

ABSORBER $Q_{AB} = M_{EVW}\ C_{PW}(T_{ABW-OUT}-T_{ABW-IN})$

CONDENSER $Q_{CO} = M_{EVW}\ C_{PW}(T_{COW-OUT}-T_{COW-OUT})$

$(COP)_{AC} = Q_{EV}/A_C\ \eta I_T\alpha\tau$

PERCENTAGE ERROR $= \frac{(Q_{GE}+Q_{EV})-(Q_{CO}+Q_{AB})}{Q_{GE}+Q_{EV}}\times 100$

6.4.2 DATA REDUCTION

The Relevant Equations Used to derive required data From Table 6.1,6.2 and 6.3 are Presented in Table 6.4 The derived data are presented in Table 6.5

TABLE 6.5 DERIVED DATA FOR EXPERIMENTAL RIG

TIME (HOURS)	Q_{GE} kJ	Q_{CO} kJ	Q_{EV} kJ	Q_{AE} kJ	$(COP)_{AC}$ -	ERROR %	HEAT LOAD MJ/m^2
10.0	5.123	1.043	2.772	3.694	0.541	+40.0	0.569
11.00	6.2579	2.0034	3.4176	6.1843	0.546	+15.37	0.6953
12.00	6.8623	1.8522	2.961	2.457	0.431	+56	0.762
13.00	7.0619	2.248	5.567	4.642	0.7883	+45.43	0.785
14.00	6.526	1.0584	3.4937	6.048	0.535	+29.1	0.725
15.00	6.2156	0.7656	1.2549	4.032	0.2018	+36.58	0.69

Normally, actual coefficient of performance is obtained from actual values of Q_{GE} and Q_{EV} obtained through energy balance using the component's inlet enthalpy, outlet enthalpy and mass flowrate of the working fluid/absorbent mixture and working fluid respectively.

Thus for the generator we have

$$Q_{GE} = M_{GE} (H_{GE-OUT} - H_{GE-IN})$$

For the evaporator,

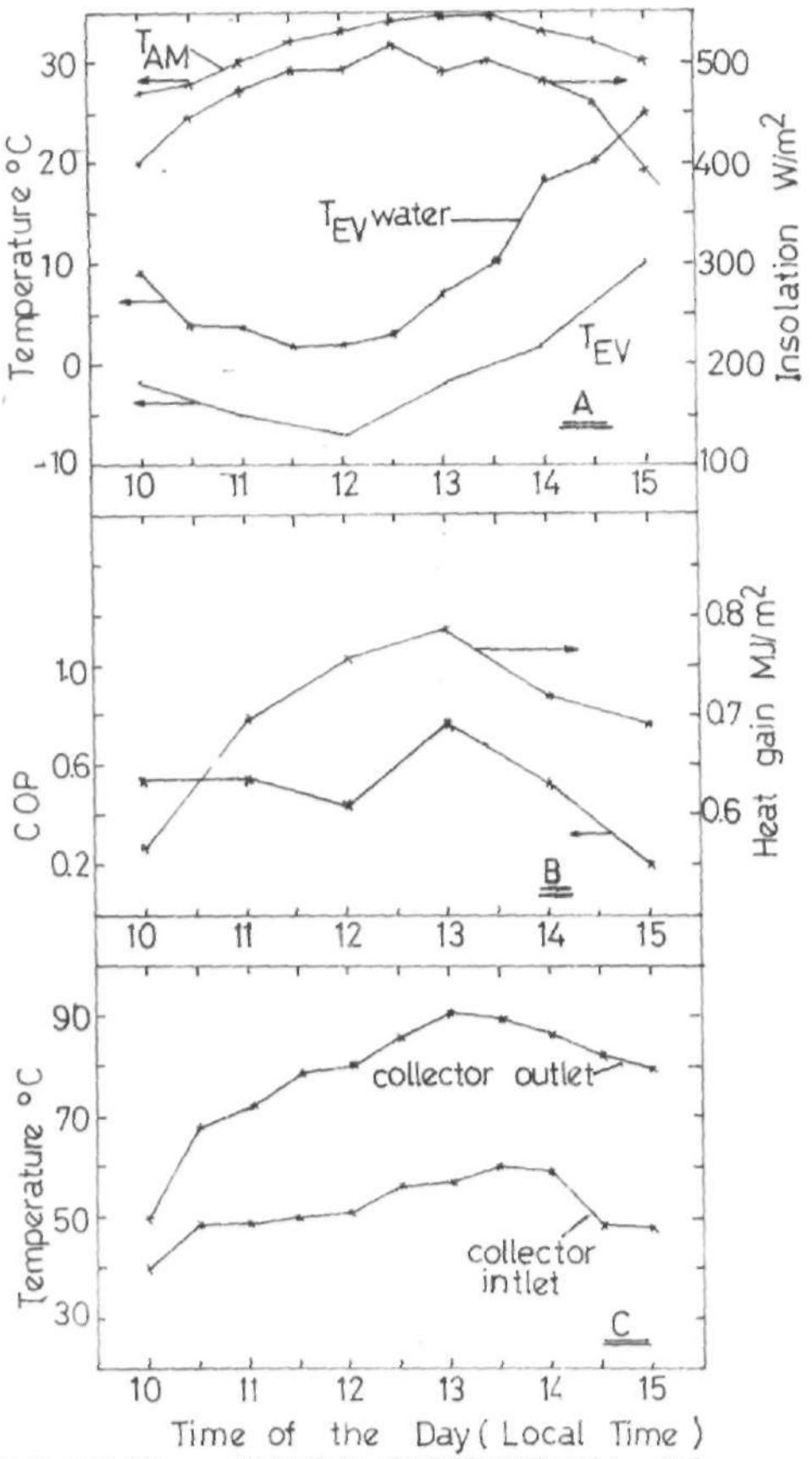


FIGURE 6.2 VARIOUS EXPERIMENTAL RESULTS AGAINST TIME OF THE DAY

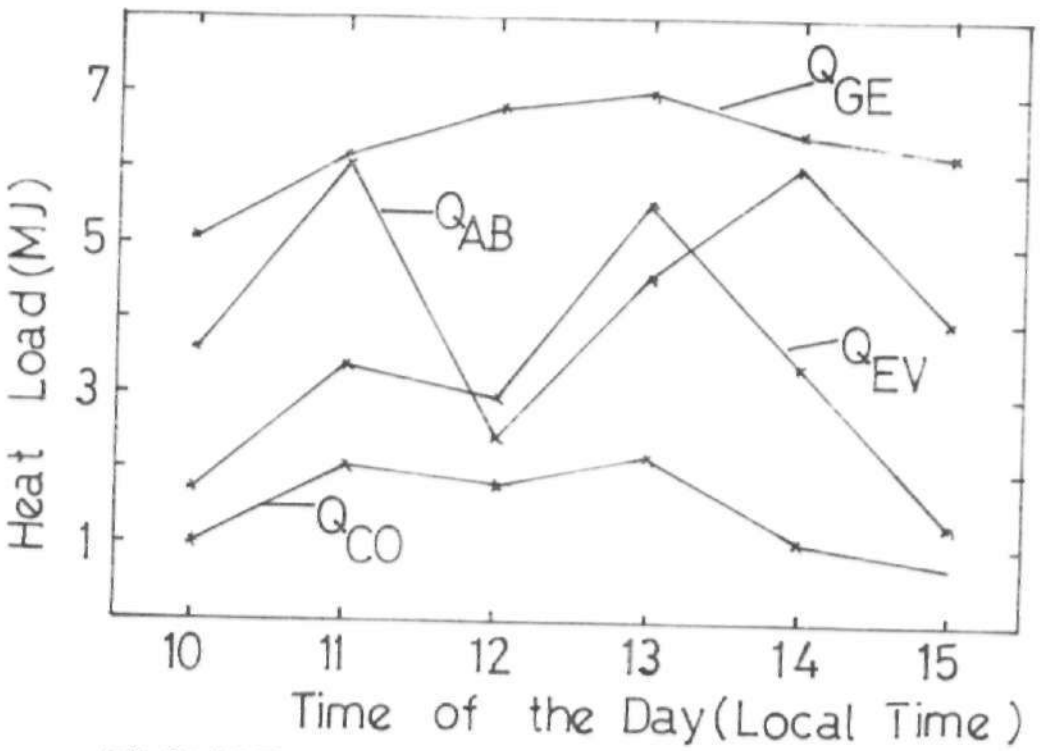


FIGURE 6.3 HEAT LOAD FOR VARIOUS COMPONENTS AGAINST TIME OF THE DAY

$$Q_{EV} = M_{EV} (H_{EV-OUT} - H_{EV-IN})$$

However since the mass flowrates of the ammonia - water solution were not known for both cases, indirect assessment was employed.

Experimental and derived results for the various hours of the day are presented in Figure 6.2 and 6.3

6.5 DISCUSSION OF RESULTS

Figure 6.2A shows the plot of insolation, ambient temperature (T_{AM}), and evaporator temperature (T_{EV}) and evaporator shell side water temprature ($T_{EV-\ water}$) against time of the day. The figure shows that as the insolation increased, the evaporator tube side temperature, and shell side temperature decreased. Evaporator tube side temperature as low as $-7^{o}C$ was achieved. The lowest evaporator outlet shell side water temperature was $2^{o}C$. However, the low evaporator temperatures could not be substained. This could be attributed to the leakage problem that increased during the course of the operation. Make - up ammonia had to be added from time to time to compensate for the losses.

Figure 6.2B shows the plot of the coefficient of performance for cooling and heat gain against time of the day. The heat gain increased from 10 hours local time up to 13 hours local time and then started dropping. The drop was due to the decreasing solar insolation as can be seen from Figure 6.2A. The coefficient of performance

was generally higher than expected. Simulated maximum obtainable results from Chapter Five hardly exceed 0.6. This discrepancy can be explained by the extra refrigerating effect caused when ammonia from the cylinder was charged into the system to make - up for the losses due to leakage problems of the evaporator. It may also be due to errors in the estimation of Q_{GE}. Except for the drop at 12 noon which could be attributed to the severe leakage problems encountered at this time, it can be seen that the coefficient of performance increased to about 13 hours local time and then started dropping. This concided with the peak of the heat gain plot in line with previous indication (Arinze et al, 1986).

Figure 6.2C shows a plot of the inlet and outlet collector temperatures against time of the day. The figure shows a significant heat gain in the collector. The maximum outlet temperature of 91^{O}C was recorded at 13 hours local time. This confirms that the equation for the calculation of ambient temperature introduced in Chapter Two is reasonable. A maximum temperature rise of 34^{O}C was achieved. The low temperature differences was due to the high absorber temperature which could not be adequately controlled during the preliminary testing of the system. However, performance studies for flat plate collectors presented in Chapter Two indicated that the decrease in the collector outlet temperature would not be substantial if the inlet temperature was as low as 25^{O}C.

Thus it could be concluded that the flat plate collector performance was adequate.

Figure 6.3 shows the total load calculated for the hour against time of the day. It is apparent that the condenser did not perform satisfactorily. It implies that either (i) enough ammonia was not passing through this component or (ii) the condenser was undersized. A confirmation could have been possible from either (i) concentration measurements, where they are available, or, (ii)level guages on at least two parts of the system, say the disengager and the absorber. In the event of the former reason, the disengager should be redesigned to ensure flashing and proper vapour - liquid separation. The latter should be established through the re-examination of the design of the condenser and adequate measures taken. Another important factor was that, the throttling valve for the weak solution from the economiser could not be adequately regulated (gate valves were used as neddle valves were not available) to achieve a substantial pressure difference (a substantial pressure difference between the high and low pressure sides implies adequate separation of working fluid from the working fluid / absorbent mixture). Thus the condenser performance could not be adequately regulated for maximum output. It may be concluded that the condenser unsatisfactory performance was due to poor instrumentation and possibly due also to poor design. From the plot, the

collector, absorber and evaporator performances seemed adequate. The heat load for the collector (generator) was quite high compared to that of the absorber and evaporator. The perturbating of the heat load for the evaporator and absorber accounts for the large percentage errors in Table 6.5. Leakage programs accounted for such perturbations.

6.6 CONCLUSION AND RECOMMENDATIONS

The analysis of the data so far collected indicates that substantial cooling was achieved. However more data need to be collected for proper characterisation. The following are recommendations which if implemented, improvement can lead to a better performance and understanding of the system:

(1) All existing pressure gauges should be changed to stainless steel pressure gauges; more should be installed.

(2) All rubber gaskets should be replaced with asbestos gaskets or any other flexible gasket material compatible. In the absence of appropriate gaskets, joints requiring them may then be welded, although this reduces flexibility.

(3) The rectifier should be cooled with water or air instead of strong solution from the absorber. This is because the absorber temperature might rise to an extent that insufficient cooling can be achieved in the rectifier. Thus water might find its way into the condenser, a condition which must be

avoided.

(4) The height of the condenser- liquid collector should be below the evaporator so that no accumulated water if any should drain into the evaporator during shut-down periods.

(5) All thermocouples should be changed to copper-constantin increase sensitivity (iron - constantin thermocouples were used before).

(6) A water pump and an insulated tank of capacity about 1000 litres should be provided for the running and storage of cooling water. If possible, a cooling tower for the cooling of absorber and condenser water should be added to the setup to minimise the wastage of water.

(7) All leakage parts of the system should be rewelded.

(8) Stainless steel throttling valves should be provided.

(9) Flowmeters, concentration measurement instruments, compatible with ammonia should be

provided for the characterisation of the system.

(10) The collector sizes should be reduced to standand pipe length sizes (say 12 feet) and about eight constructed. Facilities should be provided for bypassing any of the collectors if leakage problems occurs. They should be constructed in such a way that they can be easily opened and cleaned if need be. Facilities for the addition of side reflectors to increase absorbed radiation should be provided.

(11) Level gauges and controllers should be provided for the disengager for properly monitoring of the level of weak solution. Absence of a level controllers may results in weak solution of ammonia in the disengager moving into the rectifier and thus increasing the possibility of water condensing in the condenser. In the absence of level controllers, the disengager level can be controlled by connecting an additional collector to the disengager at the level which is considered optimum.
With such a connection, excess accumulated

fluid in the disengager will flow into this collector which will be subsequently be drained into the absorber. Two valves should be positioned, one before this collector inlet and the other after the collector outlet. The valves are present so that pressure equalisation between the high and low pressure sides is checked. The extra collector should be checked continuously so that fluid should not accumulate there for a long time.

(12) More piping and joining materials should be provided to make the above alterations possible.

It is believed, if the above recommendations are implemented, the equipment performance will substantially improve.

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APPENDIX A

WORKING FLUID AND ABSORBENT MIXTURE

The fluid that circulates in the vapour compression heat pump system is homogeneous. The system consists basically of two heat exchange components, the condenser and the evaporator. The pressure and temperature changes result from the compression effect and throttling effect of the compressor and throttling valve. The absorption heat pump system also has these two heat exchange components and the throttling valve but the compressor is replaced by two extra heat exchange components - the absorber and generator, and a pump. The fluid in the condenser-evaporator loop is homogeneous but that in the absorber-generator loop is not.

It is conventional to refer to fluids in the vapour compression heat pump as "working fluids". With reference to the absorption heat pump system, only the fluid in the condenser-evaporator loop will be referred to as "working fluid". The fluid in the absorber-generator loop will be referred to as "absorbent mixtures".

APPENDIX B

THEORETICAL EVALUATION OF VARIOUS TRACKING MODES

```
C THIS PROGRAM IS WRITTEN TO CALCULATE THE OPTIMUM ANGLE
C FOR VARIOUS TRACKING MODES;FIXED POSITION,FIXED SPEED
C SEMI-TRACKING,VARIABLE SPEED SEMI-TRACKING AND FULLY
C TRACKING. FOR EACH CASE VF IS ADJUSTED ACCORDINGLY AS
C EXPLAINED IN CHAPTER TWO. THE SIMULATION IS CARRIED OUT
C FOR VARIOUS MONTHS OF THE YEAR AND FOR DIFFERENT LOCATIONS
      PROGRAM SOLAR(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
      REAL ITH(13),IOH(13),HI(13)
        REAL LAT,LT,LATT,KD
C SYMBOLS USED; SC SOLAR CONSTANT, LAT,LATT=LATITUDE OF LOCATION
C HH=TOTAL DAILY SOLAR RADIATION, DN=DAY NUMBER,DE=DECLINATION,
C AS=SUNSET ANGLE,HA=HOUR ANGLE,IOH=EXTRATERRESTRIAL HOR.DAILY
C TOTAL SOLAR RADIATION,PG=GROUND REFLECTANCE
C THIS EXAMPLE IS FOR FIXED POSITION
      LAT=0.1957677
      PI=3.141592
        SC=4.8708E+06
         READ(5,99)TITLE
          WRITE(6,99)TITLE
99         FORMAT(1X,6A4)
         READ(5,10)DN
              READ(5,*)HH
10          FORMAT(F8.2)
        N=6
        DT=1.0
       DO 40 L=1,45
       FJ=FLOAT(L)
         STP=0.0
        SF=(FJ-10.0)*0.087266462/2.0
        SFF=SF*180.0/PI
        DE=0.40928*SIN((284.+DN)*360.*PI/(365.*180.))
        AS=ACOS(-(SIN(LAT)*SIN(DE))/(COS(LAT)*COS(DE)))
         DO 30 I=1,12
       FI=FLOAT(I)
         HI(1)=AS
          IF(I.LT.N.AND.I.GT.1)HI(I)=HI(1)-DT*15.*PI*(FI-1.0)/180.
         IF(I.EQ.N)HI(I)=0.
         IF(I.GT.N)HI(I)=-DT*15.*PI*(FI-N)/180.
         HA=HI(I)
          RO=1.0+0.033*COS(2.0*PI*DN*PI/365.0*180.0)
  IOH(I)=(SC*24.*RO/PI)*(COS(LAT)*COS(DE)*SIN(AS)+AS*
1SIN(LAT)*SIN(DE))
  HKT=HH/IOH(I)
  IF(HKT.GT..17.AND.HKT.LE..8)DKD=1.188-2.272*HKT+9.473
1*(HKT**2)-21.856*(HKT**3)+14.648*(HKT**4)
  IF(HKT.LE..17)DKD=.99
  KD=DKD*HKT
   AA=0.409+0.5016*SIN(AS-1.047)
    BB=0.6609-0.4767*SIN(AS-1.047)
     RH=0.1368997*(AA+BB*COS(HA))*(COS(HA)-COS(AS))/
```

```
     1(SIN(AS)-AS*COS(AS))
       RD=PI*(COS(HA)-COS(AS))/(24.*(SIN(AS)-AS*COS(AS)))
         SID=KD*IOH(I)*RD
          STH=HKT*IOH(I)*RH
         VF=0.0
         AI1=ACOS(SIN(DE)*SIN(FAT)*COS(SF)-SIN(DE)*COS(FAT)*SIN(SF)
     1*COS(VF)+COS(DE)*COS(FAT)*COS(SF)*COS(HA)+COS(DE)*SIN(FAT)*
     1SIN(SF)*COS(VF)*COS(HA)+COS(DE)*SIN(SF)*SIN(VF)*SIN(HA))
       AI2=ACOS(SIN(DE)*SIN(FAT)+COS(DE)*COS(FAT)*COS(HA))
       RB=COS(AI1)/COS(AI2)
           SIBA=STH-SID
       PG=0.2
       S=SIBA*RB+SID*((1.0+COS(SF))/2.0)+PG*(SIBA+SID)*
     1((1.0-COS(SF))/2.0)
         STP=STP+S
          IF(I.EQ.12)WRITE(6,22)SFF,STP
 22       FORMAT(1X,2E14.4)
30       CONTINUE
40       CONTINUE
          STOP
          END
-END OF FILE-
```

APPENDIX C

PERFORMANCE PREDICTIONS

```
C THIS PROGRAM IS WRITTEN TO FIND THE EFFECT OF FLOWRATE,
C INLET TEMPERATURE TO CHANGES OF RADIATION FOR A FLAT PLATE
C COLLECTOR. SYMBOLS USED IN THE PROGRAM IN APPENDIX A ALSO
C APPLIES HERE.OTHERS INCLUDE TAV=MEAN TEMPERATURE OF THE DAY,
C TMAX=MAXIMUM TEMPERATURE OF THE DAY,TMIN=MINIMUM TEMPERATURE
C OF THE DAY,TDD=TIME OF THE DAY,FR=FLOWRATE,TEMI=INLET TEMPER-
C ATURE,TPMM=MEAN PLATE TEMPERATURE,TOUT=COLLECTOR OUTLET TEMPE-
C RATURE.RELEVANT EQUATIONS FROM DUFFIE AND BECKMAN(1980)
      PROGRAM SOLAR1(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
     REAL ITH(13),IOH(13),HI(13)
        REAL LAT,LT,LATT,KD
    PI=3.141592
      SC=4.8708E+06
       LAT=0.1957677
       WRITE(6,*)LAT
         DO 20 M=1,2
      WRITE(6,*)M
         QUT=0.0

       WRITE(6,*)M
       HH=20670000.0
        DN=198.0
        TAV=24.14
        TMAX=28.5
        TMIN=19.7
        SFF=11.13
            SF=SFF*PI/180.0
      N=6
      DT=1.0
      DE=0.40928*SIN((284.+DN)*360.*PI/(365.*180.))
      AS=ACOS(-(SIN(LAT)*SIN(DE))/(COS(LAT)*COS(DE)))
       DO 30 I=1,12
      FL=FLOAT(I)
        HI(1)=AS
         IF(I.LT.N.AND.I.GT.1)HI(I)=HI(1)-DT*15.*PI*(FL-1.0)/180.
        IF(I.EQ.N)HI(I)=0.
        IF(I.GT.N)HI(I)=-DT*15.*PI*(FL-N)/180.
        HA=HI(I)
         RO=1.0+0.033*COS(2.0*PI*DN*PI/365.0*180.0)
      IOH(I)=(SC*24.*RO/PI)*(COS(LAT)*COS(DE)*SIN(AS)+AS*
     1SIN(LAT)*SIN(DE))
      HKT=HH/IOH(I)
      IF(HKT.GT..17.AND.HKT.LE..8)DKD=1.188-2.272*HKT+9.473
     1*(HKT**2)-21.856*(HKT**3)+14.648*(HKT**4)
      IF(HKT.LE..17)DKD=.99
      KD=DKD*HKT
       AA=0.409+0.5016*SIN(AS-1.047)
        BB=0.6609-0.4767*SIN(AS-1.047)
         RH=0.1368997*(AA+BB*COS(HA))*(COS(HA)-COS(AS))/
     1(SIN(AS)-AS*COS(AS))
```

```
      RD=PI*(COS(HA)-COS(AS))/(24.*(SIN(AS)-AS*COS(AS)))
        SID=KD*IOH(I)*RD
         STH=HKT*IOH(I)*RH
          VF=0.0

        AI1=ACOS(SIN(DE)*SIN(LAT)*COS(SF)-SIN(DE)*COS(LAT)*SIN(SF)
     1*COS(VF)+COS(DE)*COS(LAT)*COS(SF)*COS(HA)+COS(DE)*SIN(LAT)*
     1SIN(SF)*COS(VF)*COS(HA)+COS(DE)*SIN(SF)*SIN(VF)*SIN(HA))
        AI2=ACOS(SIN(DE)*SIN(LAT)+COS(DE)*COS(LAT)*COS(HA))
        RB=COS(AI1)/COS(AI2)
           SIBA=STH-SID
        PG=0.2
        S=SIBA*RB+SID*((1.0+COS(SF))/2.0)+PG*(SIBA+SID)*
     1((1.0-COS(SF))/2.0)
            AT=0.85
             STT=S*AT
          STT=STT*0.90
            CA=17.0
            PTC=430.0
            PT=0.001
            TS=0.05
            TDI=0.0157988
            TDO=0.021336
            HTC=300.0
            IF(I.LE.8)TEA=TMIN+(TMAX-TMIN)*SIN((PI/2.0)*(FT-6.0)/8.0)
            IF(I.GT.8)TEA=TAV+(TMAX-TAV)*SIN((PI/2.0)*(FT-6.0)/12.0)
            EP=0.95
            EG=0.88
            TNO=1.0
            HW=10.0
            TPM=TEA
101         TPM=TPM+0.5
          IF(TPM.GT.130.)GO TO 20
            FU=(1.0+0.089*HW-0.1166*HW*EP)*(1.0+0.07866*TNO)
           IF(SFF.LT.0.0)SFT=0.0
            IF(SFF.GT.0.0)SFT=SFF
            CU=520.0*(1.0-0.000051*SFT**2)
            EU=0.43*(1.0-100.0/(TPM+273.15))
            SB=5.6697E-8
       TTP=TPM+273.15
       TEE=TEA+273.15
         UT=1.0/(TNO/(((CU/TTP)*((TTP-TEE)/(TNO+FU))**EU)+1.0/HW))+
     1(SB*(TTP+TEE)*(TTP**2+TEE**2))/((1.0/(EP+0.00591*TNO*HW))+
     1((2.0*TNO+FU-1.0+0.133*EP)/EG)-TNO)
            TK=0.045
            TI=0.05
            UB=TK/TI
            UE=(0.045/0.025)*26.0*0.075/17.0
            UT=UT+UB+UE
```

```
      OM=SQRT(UI/(PTC*PT))
      FF=(TANH(OM*((TS-TDO)/2.0)))/(OM*((TS-TDO)/2.0))
       TDD=FI+5.5
    F=(1.0/UI)/(TS*((1.0/(UI*(TDO+(TS-TDO)*FF)))+(1.0/(PI*TDI*HTC)
   1)))
       TEMI=55.0
            TEMM=TPM+273.15
      A=0.6741
      B=2.825E-03
      C=-8.371E-06
      D=8.609E-09
      FR=0.03

     CP=4186.8*(A+B*TEMM+C*TEMM**2+D*TEMM**3)
       CAA=1.0/EXP((CA*UI*F)/(FR*CP))
       CAC=1.0-CAA
       FRC=((FR*CP)/(CA*UI))*CAC
       QU=CA*FRC*(STT-UI*(TEMI-TEA)*3600.0)
       TFM=TEMI+(((QU/CA)/(UI*3600.0*FRC)))*(1.0-FRC/F)
       TEMM=TFM+273.15
       CP=4186.8*(A+B*TEMM+C*TEMM**2+D*TEMM**3)
     CAA=1.0/EXP((CA*UI*F)/(FR*CP))
     CAB=TEMI-TEA-STT/(UI*3600.0)
     CAC=1.0-CAA
      FRC=((FR*CP)/(CA*UI))*CAC
       QU=CA*FRC*(STT-UI*(TEMI-TEA)*3600.0)
       QUP=QU/CA
          IF(QU.LE.0.0)GO TO 30
     TOUT=(CAA*CAB)+TEA+STT/(UI*3600.0)
     TPMM=TEMI+(((QU/CA)/(UI*3600.0*FRC)))*(1.0-FRC)
                  IF(ABS(TPMM-TPM).GT.1.0)GO TO 101
      IF(QU.LE.0.0)QUP=0.0
                QUT=QUT+QUP
                WRITE(6,23)TDD,FR,TEMI,QUP,TPMM,TOUT
                IF(I.EQ.12)WRITE(6,24)QUT
   23       FORMAT(1X,6E14.4)
   24        FORMAT(3X,E14.4)
   30        CONTINUE
   20          CONTINUE
                  END
-END OF FILE-
```

APPENDIX D

EVALUATION OF DATA EQUATIONS FOR AMMONIA-WATER MIXTURES

```
C THIS PROGRAM IS USED TO FIND THE RELATIONSHIP
C AMONG ANY THREE OF THE FOLLOWING VARIABLES FOR
C AMMONIA-WATER MIXTURE;TEMPERATURE, PRESSURE,
C CONCENTRATION AND ENTHALPY.THE SUBROUTINE
C E02CAF FORMS AN APPROXIMATION TO THE WEIGHTED
C LEAST-SQUARES CHEBYSHEV SERIES SURFACE FIT
C TO DATA ARBITRARILY DISTRIBUTED ON LINES
C PARALLEL TO ONE INDEPENDENT CO-ORDINATE AXIS.
C IN THIS EXAMPLE PRESSURE(Y) AND CONCENTRATION
C ARE INDEPENT VARIABLES AND ENTHALPY, THE DEPENDENT
C VARIABLE. DATA ARE REPORED IN ENGINEERING UNITS,
C SUITABLE EQATIONS HAVE BEEN INCLUDED TO CONVERT
C THEM TO SI UNITS.
      PROGRAM QW1(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
        REAL PRESID,RESID,YMAX,YR
       INTEGER I,IFAIL,J,K,KP1,L,MI,MJ,N,NA,NIN,NOUT,NW,NX,R,T
        REAL A(100),F(400),FF(400),TITLE(7),W(400),WORK(1500),X(400)
     1,XMAX(20),XMIN(20),Y(20)
        INTEGER M(20)
C SUBROUTINE REFERENCES.....E02CAF,E02CBF
         DATA NIN,NOUT/5,6/
       NA=300
        NX=1000
         NW=3000
C INPUT THE NUMBER OF LINES Y=Y(I) ON WHICH
C DATA IS GIVEN AND THE REQUIRED DEGREE OF FIT
C IN THE X AND Y DIRECTIONS
        READ(NIN,99)TITLE
         WRITE(NOUT,95)(TITLE(I),I=1,6)

  20      READ(NIN,98)N,K,L
      WRITE(6,98)N,K,L
        IF(N.EQ.0)STOP
        KP1=K+1
       MJ=0
C INPUT Y(I) THE NUMBER OF DATA POINTS ON Y(I)
C AND THE RANGE OF X VALUES ON THIS LINE,FOR I=1,2..N
         DO 40 I=1,N
       READ(NIN,97)Y(I),MI,XMIN(I),XMAX(I)
C CONVERTING TO BARS
        Y(I)=Y(I)/14.696
         M(I)=MI
         MJ=MJ+MI
 40      CONTINUE
C TERMINATE PROGRAM IF THE ARRAYS HAVE NOT BEEN
C DECLARED LARGE ENOUGH TO CONTAIN THE DATA
C TERMINATE PROGRAM IF THE ARRAYS HAVE NOT BEEN DECLARED LARGE ENOUGH TO
C CONTIAN THE DATA.
       IF(NX.GE.MJ)GO TO 60
        WRITE(NOUT,94)MJ
```

```
      STOP
C INPUT THE VALUES AND FUNTION VALUES F TOGETHER
C WITH THEIR WEIGHTS,W
C INPUT THE X VALUES AND FUNTION VALUES F TOGETHER WITH THEIR WEIGHTS,W.
  60      READ(NIN,96)(X(I),F(I),W(I),I=1,MJ)
C CONVERTING ENTHALPY UNITS TO SI UNITS
        DO 21 I=1,MJ
         F(I)=2.3245662*F(I)
  21      CONTINUE
          IFAIL=1
C EVALUATUATE THE COEFFICIENTS A OF THE FIT
C TO THIS SET DATA
C EVALUATE THE COEFFICIENTS A OF THE FIT TO THIS SET OF DATA.
       CALL E02CAF(M,N,K,I,X,Y,F,W,NX,A,NA,XMIN
     1,XMAX,Y,1,Y,1,WORK,NW,IFAIL)
          IF(IFAIL.EQ.0)GO TO 80
          WRITE(NOUT,93)IFAIL
          GO TO 20
  80      MI=0
          DO 140 R=1,N
          IFAIL=1
      T=MI+1
       MI=MI+M(R)
       YR=Y(R)
          YMAX=Y(N)
          IF(N.EQ.1)YMAX=YMAX+1.0
C EVALUATE THE FITTED POLYNOMIAL AT EACH OF THE
C DATA POINTS ON THE LINES Y=Y(I)
C EVALUATE THE FITTED POLYNOMIAL AT EACH OF THE DATA POINTS ON THE LINES
C Y=Y(R)
       CALL E02CBF(T,MI,K,I,X,XMIN(R),XMAX(R),YR,Y(1),YMAX
     1,FF,A,NA,WORK,NW,IFAIL)

          IF(IFAIL.EQ.0)GO TO 100
          WRITE(NOUT,92)IFAIL
          GO TO 20
 100      DO 120 I=T,MI
          RESID=FF(I)-F(I)
         PRESID=(RESID/FF(I))*100.0
 120      CONTINUE
          WRITE(NOUT,87)
 140      CONTINUE
C OUTPUT THE CHEBYSHEV COEFFICIENT OF THE FIT
C OUTPUT THE CHEBYSHEV COEFFICIENTS OF THE FIT
          WRITE(NOUT,89)
          T=0
          DO 160 J=1,KP1
          R=T+1
          T=R+I
      WRITE(NOUT,88)(A(I),I=R,T)
```

```
 160      CONTINUE
          GO TO 20
 99       FORMAT(6A4,1A3)
 98       FORMAT(3I5)
 97       FORMAT(F8.3,I5,2F8.3)
96    FORMAT(3F8.3)

 95       FORMAT(4(1X/),1H,5A4,1A3,7HRESULTS/1X)
 94       FORMAT(40H NX IS TOO SMALL. IT SHOULD BE AT LEAST , I5)
 93       FORMAT(22H E02CAF FAILURE NUMBER,I3)
 92       FORMAT(22H E02CBF FAILURE NUMBER,I3)
 89       FORMAT(34H CHEBYSHEV COEFFICIENTS OF THE FIT)
 88       FORMAT(1H0, 1P5E14.4)
 87       FORMAT(1X)
          END
-END OF FILE-
```

APPENDIX E

THERMODYNAMIC ANALYSIS OF AMMONIA-WATER HEAT PUMP SYSTEMS

```
C THIS PROGRAM WAS DESIGNED TO CALCULATE THE MAXIMUM OBTAINABLE COP
C FOR A RANGE OF CONCENTRATION OF AQUEOUS SOLUTION AND A RANGE OF
C TEMPERATURE APPROACH BETWEEN THE HOT SOLUTION LEAVING THE ECONOMISER
C FOR THE ABSORBER VIA THE EXPANSION VALVE, AND THE WARM COLUTION ENTERING
C THE ECONOMISER FROM THE ABSORBER.
C THE NUMBERING OF THE FLOW OF STREAM OF FLUIDS IN THE ABSORPTION HEAT
C PUMP SYSTEM AS FOLLOWS
C     (1)     WATER VAPOUR FROM THE GENERATOR INTO THE CONDENSER
C     (2)     CONDENSED AMMONIA LEAVING THE CONDENSER INTO THE EVAPORATOR
C             VIA THE EXPANSION VALVE
C     (3)     STREAM (2) AFTER THE EXPANSION VALVE
C     (4)     AMMONIA VAPOUR FROM THE EVAPORATOR IN THE ABSORBER
C     (5)     SOLUTION LEAVING THE GENERATOR FOR THE ECONOMISER
C     (6)     SOLUTION LEAVING THE ECONOMISER FOR THE ABSORBER VIA THE
C             EXPANSION VALVE
C     (7)     STREAM (6) AFTER THE EXPANSION VALVE
C     (8)     SOLUTION LEAVING THE ABSORBER INTO THE ECONOMISER VIA THE
C             SOLUTION PUMP
C     (9)     STREAM (8) AFTER THE SOLUTION PUMP
C    (10)     SOLUTION FROM THE ECONOMISER INTO THE GENERATOR
C             NOMENCLATURE
C COPH,COPC,COPCM,CPHC,CPC COEFFICIENT OF PERFORMANCE
C             (DIMENSIONLESS)
C FX,FFX      CIRCULATION RATIO (RATE OF FLOW  OF AMMONIA-RICH AQUEOUS
C             SOLUTION TO THE RATE OF FLOW OF PURE AMMONIA IN THE CONDENSER
C             AND EVAPORATOR (DIMENSIONLESS)
C HX,HHX      ENTHALPY PER UNIT WEIGHT OF FLUID AT FLOW STREAM POSITION
C             (X) (KJ PER KG)
C PX,PPX      PRESSURE AT FLOW POSITION (X)(BARS)
C TA          TEMPERATURE APPROACH AT ABSORBER END OF THE ECONIMSER (DEG.C)
```

```
C   TBMTV,TBMT    ABSORBER TEMPERATURE LIFT (DEG.C)
C   TCMTV,TCMT    CONDENSER TEMPERATURE LIFT (DEG.C)
C   TEV           EVAPORATING TEMPERATURE OF AMMONIA IN THE EVAPORATOR
C                 (DEG.C)
C   TGE           TEMPERATURE OF SOLUTION IN THE GENERATOR (DEG.C)
C   TOST          LOWEST CIRCULATION RATIO (DIMENSIONLESS)
C   XB,XXAB       PERCENT WEIGHT OF AMMONIA IN WATER IN THE ABSORBER
C                 (PERCENT)
C   XWGE,XXGE     PERCENT WEIGHT OF AMMONIA IN WATER IN THE GENERATOR
C                 (PERCENT)
C   EBIT          LOWEST PERCENT WEIGHT OF AMMONIA IN WATER (PERCENT )
C                 THE PROGRAM
C   SET COP=1,IF CONVENTIONAL SYSTEM, SET COP = 2 IF REVERSED
C                 INITIALISATION
C
C                 PROGRAM COP (INPUT.OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
                  DIMENSION (P4(20),XB(20),T7(20),TBMV(20,20)H4(20,20)
                 *H3(20),T5(20),H7(20),H5(20),H10(20),P1(20),FX(20),
                 *XWGE(20),H8(20,20),TGE(20,20,20),H1(20),H2(20
                 *TCTV (20), COPH(20),T2(20),T4(20),COPC(20),CPCM(20),
                 *T6(20),H6(20),TT4(20),PP4(20),XXAB(20,20),TT8(20,20),
                 *TBMT(20,20),HH3(20,20),TT7(20),HH4(20,20),HH8(20,20),
                 *TT2(20,20),PP2(20,20),HH2(20,20),FFX(20),XXGE(20),TT5(20),
                 *TT1(20),HH1(20),HH7(20),HH10(20),CPH(20),TCMT(20),XEB(20),
                 *TV(20),ZT(20),CPHC(20),HH5(20),CPC(20)
                  COMMON/BLK3/P(20)
                  COMMON/BLK1/F(20)
                  COMMON/BLK2/T(20)
                  REAL XT4,TA,TGEI,TOST,XBIT
                  IF(COP.EQ.1) GO TO 37
                  IF(COP.EQ.2) GO TO 38
                  TA=10.0
                  TGEI=333.15
                  TOST=20.0
                  XBIT=30.0
```

```
C FOR CASES FOR WHICH ABSORBER TEMPERATURE IS EQUAL TO CONDENSER
C TEMPERATURE, DIRECT TO STATEMENT NUMBER 99
      GO TO 17
      GO TO 99
   17 WRITE (6,501)
      WRITE (6,502)
      WRITE (6,503)
      WRITE (6,508)
      1AL=7
C BECAUSE OF TIME INVOLVE IN THE    ITERATIONS IAL CAN BE SET TO 2
C AND DATA GOT FOR TWO EVAPORATOR TEMPERATURE; THAT IS IF THE
C    ITERATION PROCEDURE IS FOLLOWED. SEE AHEAD
      DO 1 I=1,IAL
      F1=FLOAT(I)
      T4(I)=XT4+(F1-1.0)*5.0
      P4(I)=EXP(11.29261 2600.0341/T4(I)-25395.259/T4(I)**2)
      PT=P4(I)
      TEV=T4(I)
      IF (PT.GT.20.413) GO TO 1
C THE ABOVE STATEMENT CAN BE CHANGE ACCORDING TO THE USERS NEED
      JAL=18
      DO 2 J=1,JAL
      FJ=FLOAT(J)
      XB(J)=XBIT+(FJ-1.0)*2.0
      P40=PT
      XAB=XB(J)
      TV(J)=TEV
      JP=1
C USERS WHO HAVE ACCESS TO THE NAG LIBRARY NEED NOT MAKE USE OF THE
C SUBROUTINE BIOL AND ENTH AND  PRESS, INSTEAD THE SHOULD CALL BIOLA,
C ENTHA AND PRESSA- SET NAG=1 IF USING BIOLA, ENTHA, PRESSA.
      NAG=1
      IF(NAG.EQ.1) GO TO 600
      CALL BIOL(P40,XAB,TEQ)
      T8(J)=TEQ
      GO TO 800
```

```
600   CALL BIOLA (JP,XAB,P40)
      T8(J)=F(JP)
800   TAB=T8(J)
      IF (TAB.GT.323.15) GO TO 2
C     WE DON'T INTEND TO WORK ABOVE ABSORBER TEMPERATURE OF 323.15K
      T8(J)=TAB
      TEMTV(I,J)=T8(J)-TEV
      H4(I,J)=819.27972-7.149116E-05*P40*(TEV+45652.165)
     *+11.6804322*TEV
      T4(J)=TV(J)
      T7(J)=T8(J)+TA
      ZX=XAB
      T77=T7(J)
      ZT(J)=T8(J)
      JQ=1
      IF(NAG.EQ.1) GO TO 601
      CALL ENTH(ZT(J),ZX,HQ)
      H8(I,J)=HQ
      GO TO 801
601   CALL ENTHA(JQ,ZX,ZT(J))
      H8(I,J)=T(JQ)
801   KAL=7
      DO 3 KO=1,KAL
      FKO=FLOAT(KO)
      TGE(I,J,KO)=TGEI+(FKO-1.0)*10.0
      MAL=9
      DO 4 MN=1,MAL
      FMN=FLOAT(MN)
      FX(MN)=TOST+(FMN-1.0)*5.0
      T7(MN)=T77
      XB(MN)=XAB
      XWGE(MN)=(100.0-FX(MN)*XB(MN))/(1.0-FX(MN))
      T5(MN)=TGE(I,J,KO)
      G7=EWGE(MN)
      TO7=T7(MN)
      MNQ=1
```

```
      IF(NAG.EQ.1) GO TO 602
      CALL ENTH (TO7,G7,HQQ)
      H7(MM)=HQQ
      GO TO 802
  602 CALL ENTHA (MNQ,G7,T07)
      H7(MN)=T(MNQ)
  802 G5=XWGE(MN)
      T05=T5(MN)
      MNR=1
      IF (NAG.EQ.1) GO TO 303
      CALL ENTH (T05,G5,HQR)
      H5 (MN)=HQR
      GO TO 803
  303 CALL ENTHA (MNR,G5,T05)
      H5(MN)=T(MNR)
  803 H10(MN)=H8(I,J)+(H5(MN)-H7(MN)*(1.0-(1.0/FX(MN))
C         IN THE CALCULATION OF THE GENERATOR PRESSURE, WE CAN USE
C         THE SUBROUTINE PRESS OR PRESSA, OR WE CAN USE BIOLA BIOL
C         BY ITERATION WHICH, EVER OPTION THE USERS MAY CHOSE
C         BY ITERATION, SET IT = 1
      IT=1
      IF (IT.EQ.1) GO TO 604
      XWG1=XWGE (MN)
      L=1
      IF (NAG.EQ.1) GO TO 710
      CALL PRESS (T55,XWG1,PR)
      P1(MN)=PR
      GO TO 804
  710 CALL PRESSA (L,XWG1,T55)
      P1(MN)=PX(L)
      GO TO 804
  604 CONTINUE
      CO=0.0
    5 CO=CO+1.0
      XWG1=XWGE(MN)
      L=1
      PA=1.5414+(CO+1.0)*0.01
```

```
      IF (PA.GT.20.413) GO TO 4
      G10=XWG1
      PAA=PA
      IF(NAG.EQ.1) GO TO 605
      CALL BIOL (PAA,G10,TK)
      TA5=TK
      GO TO 805
605   CALL BIOLA (L,G10,PAA)
      TA5=F(L)
805   IF (TA5.GT.394.14) GO TO 4
      WE DON'T INTEND TO GO ABOVE SUCH GENERATOR TEMPERATURE
      CONSIDERED
      IF (ABS(TA5-T5(MN).GT.0.2) GO TO 5
      P1(MN)=PA
804   H1(MN)=819.2797+7.149116E-05*P1(MN)*(T5(MN)+45652.165)
     *+11.6804322*T5(MN)
      IF(IT.EQ.1) GO TO  5

      DO=0.0
 6    DO=DO+1.0
      TA2=273.15+(D=-1.0)*0.1
      IF (TA2.GT.394.15) GO TO 4
      P2=EXP(11.29261-2600.0341/TA2-25395/Ta2**2)
      IF (ABS(P2-P1(MN).GT.0.2) GO TO 6
      T2(MN)=TA2
      IF(NAG.EQ.1) GO TO 606
      CALL BIOL (P1(MN),100.0,TT7)
      T2(MN)=TT5
      GO TO 900
606   LA=1
      CALL BIOLA (LA,100.0,P1(MN))
      T2(MN)=F(LA)
900   IF (T2(MN).LE.293.0) GO TO 4
        H2(MN)= -1344.1126+4.196453*T2(MN)
      H3(MN)=H2(MN)
```

```
      TCMTV(MN)=T2(MN)-TEV
      COPH(MN)=(H4(I,J)-H7(MN)+FX(MN)*(H7(MN)-H8(I,J)+H1(MN)
     *-H2(MN))/(FX(MN)*(H5(MN)H10(MN)+H1(MN)-H5(MN))
      COPC(MN)=(1.0-TEV/T5(MN))*(T2(MN)/(T2(MN)-TEV))
      CPCM(MN)=(H4(I,J)-H3(MN))/(FX(MN)*(H5(MN)-H10(MN))+H1(MN)-H5(MN))
      WRITE (6,504)TCE(I,J,KO),T2(MN),TEV,T8(J),T7(MN),T5(MN)
      WRITE (6,505)H1(MN),H2(MN),H4(I,J),H8(I,J),H7(MN),H5(MN),H10(MN)
      WRITE (6,506)P40,P1(MN),P1(MN),FX(MN),COPC(MN),COPH(MN),CPCM(MN)
      WRITE (6,507)TCMTC(MN),TBMTV(I,J),XWGE(MN)
    4 CONTINUE
    3 CONTINUE
    2 CONTINUE
    1 CONTINUE
   99 GO TO 51
   99 CONTINUE
  501 FORMAT(1X,"T1   T2   T4   T8   T7   T5")
  502 FORMAT(1X,"H1   H2   H4   H8   H7   H5   H10")
  503 FORMAT(1X,"P4   P1   FR   COPH(C)   COPH(MO)   COPC(MO)")
  508 FORMAT(1X,"T(CO)-T(EV)   T(AB)-T(EV)   X(AB)   X(GE)")
  504 FORMAT (1X,6E11.4/)
  505 FORMAT (1X,7E10.3/)
  506 FORMAT (1X,6E10.3/)
  507 FORMAT (1X,E14.4,3X,E14.4,2E14.4//)
C     CASES FOR ABSORBER TEMPERATURE EQUALS AND DENSERT
      WRITE (6,501)
      WRITE (6,502)
      WRITE (6,503)
      WRITE (6,508)
  601 FORMAT(1X,"   CASES FOR ABSORBER'S TEMPERATURE EQUALS
      CONDENSER'S")
      1A=7
      DO 701 1=1,IA
      F10=FLOAT(I)
```

```
      TT4(I)=XT4+(F10-1.0)*5.0
      PP4(I)=EXP(11.29261-2600.0341/TT4(I)-25395.295/TT4(I)**2)
      IF(PP4(I).GT.20.414) GO TO 701
      JA=18
      DO 702 J=1,JA
      FJO=FLOAT(J)
      XXAB(I,J)=EBIT+(FJO-1.0)*2.0
      XXB=XXAB(I,J)
      PP44=PP4(I)
      TET=TT4(I)
      XX=XXB
      PP=PP44
      JZ=1
      IF(NAG.EQ.1)  GO TO 606
      CALL BIOL (PP,XX,TZZ)
      TT8(I,J )=TZZ
      GO TO 806
606   CALL BIOLA (JE,XX,PP)
      TT8(I,J)=F(JZ)
806   IF(TT8(I,J).GT.323.15) GO TO 702
      XEB(J)=XXB
      PP4(J)=PP44
      TT4(J)=TET
      TBMT(I,J)=TT8(I,J)-TT4(J)
      HH4(I,J)=819.27971-7.149116E-05*PP4(J)*(TT4(J)+45652.165)
     *+11.6804322*TT4(J)
      XEB=XEB(J)
      TF8=TT8(I,J)
      TET=TT8(J)
      JY=1
      IF(NAG.EQ.1) GO TO 607
      CALL ENTH (TF8,XEB,HPZ)
      HH8(I,J)=HPZ
      GO TO 807
607   CALL ENTHA (JY,XEB,TF8)
      HH8(I,J)=T(JY)
807   TT2(I,J)=TT8(I,J)
```

```
      TCMT(J)=TT2(I,J)-TET
      PP2(I,J)=EXP(11.29261 2600.0431/TT2(I,J)-25395.295/
     *TT2(I,J)**2)
      IF(PP2(I,J).GT.20.414) GO TO 702
      HH3(I,J) =-1344.1126+4.9162453*TT2(I,J)
      HH3(I,J)=HH2(I,J)
      PPF=PP2(I,J)
      XXAC=XEB(J)
      MALA=9
      DO 703 M=1,MALA
      FKM=FLOAT(M)
      FFX(M)=TOST+(FMK-1.0)*5.0
      XXGE(M)=(100.0-FFX(M)*XXAC)/(1.0-FFX(M))
      MA=1
      IF (NAG.EQ.1) GO TO 608
      CALL BIOL (PPF,XXGE(M),TPD)
      TT5(M)=TTD
      GO TO 808
608   CALL BIOLA(MA,XXGE(M),PPF)
      TT5(M)=F(AM)
808   IF(TT5(M).GT.394.14) GO TO 703
      IF((TT5 (M)-TF8).LT.15.0) GO TO 703
      MV=1
      IF (NAG.EQ.1) GO TO 609
      CALL ENTH (TT5(M) ,XXGE(M),HDS)
      HH5(M)=HDS
      GO TO 809
609   CALL ENTHA (MV,XXGE(M),TT5(M))
      HH(M)=T(MV)
809   TT1(M)=TT5(M)
      HH1(M)=819.27972-7.149116E-05*PPF*(TT1(M)+45652.165)
     *+11.6804322*TT1(M)
      TT7(M)=TF8+T8
      MB=1
      IF (NAG.EQ.1) GO TO 610
      CALL ENTH (TT7(M),XXGE(M),HQD)
      HH7(M)=HQD
```

```
610   ENTHA (MB,XXGE(M),TT7(M)
      HH7(M)=T(MB)
810   HH10(M)=HH8(I,J)+(HH5(M)-HH7(M))*
      (1.0-(1.0/FFX(M)))
      CPH(M)=(HH4(I,J)-HH7(M)+FFX(M)*(HH7(M)-HH8(I,J)))+(HH1(M)-HH2
     *(I,J)))/(FFX(M)*(HH5(M)-HH10(M))+HH1(M)-HH5(M))
      CPHC(M)=(1.0-TET)/TT5(M))*TT2(I,J)/TT2(I,J)-TET))
      CPCC(M)=(HH4(I,J)-HH3(I,J))/(FFX(M)*
      (HH5(M)-HH10(M))+HH1(M)-HH5(M))
      CPHC(M)=TT2(I,J)/TT5(M)*((TT5(M)-TET)/(TT2(I,J)-TET))
      CPC=(HH4(I,J)-HH3(I,J))/(FFX(M)*
      (HH5(M)-HH10(M)+HH1(M)-HH5(M))
      WRITE (6,504)TT1(M),TT2(I,J),TET(TT8(I,J),TT7(M),TT5(M)
      WRITE (6,505)HH1(M),HH2(I,J),HH4(I,J),HH8(I,J),HH7(M),HH5(M),
     *HH10(M)
      WRITE (6,506)PP4(I),PP2(I,J), FFX(M),CPHC(M),CPH(M),CPC(M)
      WRITE (6,507)TCMT(J),TBMT(I,J),XXAC,XXGE(M)
703   CONTINUE
702   CONTINUE
701   CONTINUE
 51   STOP
      END
```

```
 38   XT4=303.15
      TA=10.0
       TOST=6.0
         XBIT=30.0
        WRITE(6,500)
500       FORMAT(" REVERSED ABSORPTION HEAT PUMP SYSTEM")
      WRITE(6,501)
      WRITE(6,502)
      WRITE(6,503)
      WRITE(6,508)
      IAI=7
      DO 1 I=1,IAI
      FI=FIOAT(I)
      T4(I)=XT4+(FI-1.0)*5.0
      P4(I)=EXP(11.29261-2600.0341/T4(I)-25395.295/T4(I)**2)
        IF(P4(I).IE.0.81655)GO TO 1
       IF(P4(I).GT.20.414)GO TO 1

          PT=P4(I)
           TEV=T4(1)
         H4(I)=819.27971-7.149116E-05*PT*(TEV+45652.165)+
    11.6804322*TEV
      JAI=18
      DO 2 J=1,JAI
      FJ=FIOAT(J)
      XB(J)=XBIT+(FJ-1.0)*2.0
         P40=PT
         XAB=XB(J)
       XAR=XAB
      TV(J)=TEV
         JP=1
         CALL BIOIA(JP,XAB,P40)
         T8(J)=F(JP)
         TAB=T8(J)
        JQ=1
         CALL ENTHA(JQ,XAR,TAB)
         H8(I,J)=T(JQ)
         IF(TAB.GT.394.14)GO TO 2
      TBMTV(I,J)=TAB-TEV
        T4(J)=TV(J)
       T7(J)=TAB-TA
          T77=T7(J)
       MAI=6
       DO 4 MN=1,MAI
       FMN=FIOAT(MN)
        FX(MN)=TOST+(FMN-1.0)*4.0
         T7(MN)=T77
       XB(MN)=XAB
      XWGE(MN)=(100.0-FX(MN)*XB(MN))/(1.0-FX(MN))
      T5(MN)=TEV
?
```

```
      G7=XWGE(MN)
       T07=T7(MN)
        MNQ=1
       CALL  ENTHA(MNQ,G7,T07)
     H7(MN)=T(MNQ)
      G5=XWGE(MN)
       T05=T5(MN)
        MNR=1
      CALL  ENTHA(MNR,G5,T05)
     H5(MN)=T(MNR)
    H10(MN)=H8(I,J)+(H5(MN)-H7(MN))*(1.0-(1.0/FX(MN)))
      TAD=T5(MN)
       XWG1=XWGE(MN)
       G10=XWG1
       I=1
       CALL  PRESSA(I,G10,TAD)
       P1(MN)=P(I)
      IF(P1(MN).LE.0.81655)GO TO 4
     H1(MN)=819.27971-7.149116E-5*P1(MN)*(T5(MN)+45652.165)+
11.6804322*T5(MN)
   PP1=P1(MN)
    IA=1
    CALL  BIOLA(IA,100.0,PP1)
    T2(MN)=F(IA)
       IF(T2(MN).LE.263.15)GO TO 4
      IF(T2(MN).GT.TEV)GO TO 4
    H2(MN)=-1344.1126+4.9162453*T2(MN)
    TCMTV(MN)=TEV-T2(MN)
     COPH(MN)=((FX(MN)-1.0)*H7(MN)+H4(I)-FX(MN)*H8(I,J))/((FX(
1MN)-1.0)*H5(MN)-FX(MN)*H10(MN)+H1(MN)+H4(I)-H2(MN))
       IF(COPH(MN).LE.0.0)GO TO 4
      COPC(MN)=(1.0-T2(MN)/T5(MN))*(TAB/(TAB-T2(MN)))

    WRITE(6,504)T5(MN),T2(MN),TEV,TAB,T7(MN),T5(MN)
    WRITE(6,505)H1(MN),H2(MN),H4(I),H8(I,J),H7(MN),H5(MN),H10(MN
   1)
    WRITE(6,506)P40,P1(MN),FX(MN),COPC(MN),COPH(MN)
    WRITE(6,507)TCMTV(MN),TBMTV(I,J),XAR,XWGE(MN)
4       CONTINUE
2       CONTINUE
1       CONTINUE
         STOP
          END
```

```
C     THE NECESSARY SUBROUTINE
C   **SUBROUTINE BIOLA**
C     SUBROUTINE BIOLA CALCULATES THE SATURATION TEMPERATURE GIVEN THE
C     SATURATION PRESSURE AND CONCENTRATION OF AMMONIA IN THE SOLUTION BY
C     CALLING SUBROUTINE EØ2CBF FROM THE NAG LIBRARY.
C     NOTE-THOSE WITHOUT THIS FACILITY SHOULD USE AS STATED BEFORE BIOL
C     AND FOR THE OTHER SUBROUTINES, ENTH AND PRESS
      SUBROUTINE BIOLA (MDUM,XDUM,PDUM)
      COMMON/BLK1/F(20)
      REAL XDUM,PDUM,WK(12),AO(12),BO(30),CO(30),DO(30)
      DATA AO/1.1808E+03,2.2756E+01,-2.1393E+00,-1.3296E-02,-2.0565E+00,
     *2.0270E-01,3.7784E01,1.8645E-01,1.5058E-02,-1.7086E+00,1.1242E-01,
     *2.4552E-02/
      DATA BO/1.3432E+03,5.0530E+01,-7.4615E+00,1.5989E+00,-4.4155E-01,-
     *1.4677E+02,-4.4559E+00,7.0187E-01,-1.0395E-01,2.7736E-02,3.8944E+01,
     *2.9585E-01,-2.0679E-02,8.4342E-03,-2.4076E-02,7.7446E-01,2.0930E-01,-
     *3.0539E-02,-1.9785E-02,2.6029E-02,-1.0967E+00,-4.3997E-01,-2.7643E-02,
     *3.8438E-02,9.3406E-03,-1.2267E+00,3.5413E-01,-3.2161E-02,-2.1582E-02,-
     *2.1347E-02/
      DATA CO/1.4940E+03,2.8658E+01,-2.1132E+00,2.1205E-01,-8.1942E-03,-
     *1.5924E+02,-1.9905E+00,3.6542E-01,-5.9377E-02,-2.6529E-02,3.9428E+01,-
     *4.8887E-02,-2.0674E-02,1.2702E-02,3.5557E-03,-1.4843E-01,-4.0882E02,-
     *8.9779E-02,2.0667E-02,1.8525E-02,-2.2580E+00,-2.7157E-02,1.0244E-01,-
     *2.9635E-02,-8.9233E-03,2.9753E-01,2.1228E-02,-1.3925E-01,2.643E-02,
     *1.2928E-02/
      DATA DO/1.5745E+03,1.3224E+01,-2.3889E-01,7.6243E-02,-2.0605E-02,-
     *1.6455E+02,-9.0433E-01,5.6693E-02,1.4544E-02,-3.9139E-03,3.8984E+01,-
     *1.2025E-01,6.9276E-02,4.0042E-03,-6.5407E-03,-4.3264E-01,-1.5508E-02,
     *6.7470E-02,8.40691E-03,-2.0490E-02,-2.0433E+00,9.8179E-02,-5.9511E-03,-
     *2.0067E-02,3.4546E-04,-7.4846E-01,-5.0549E-02,4.4410E-02,-1.5876E02,
     *9.1042E-03/
```

```
      IF(PDUM.LT.2.04137) GO TO 10
      IF(PDUM.LT.10.0207) GO TO 11
      IF(PDUM.LT.20.414)  GO TO 12
      IF(PDUM.LT.27.218)  GO TO 13
   10 K=3
      L=2
      NWK=12
      PMIN=0.81655
      PMAX=2.0414
      IFAIL=1
      XMIN=0.0
      XMAX=100.0
      NA=12
C     CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
      CALL EØ2CBF(MDUM,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,F,AO,
     *NA,WK,NWK,IFAIL)
      GO TO 100
   11 K=5
      L=4
      NA=30
      XMIN=0.0
      XMAX=100.0
      PMIN=2.0414
      PMAX=10.0207
      IFAIL=1
      NWK=30
C     CALL  THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
      CALL EØ2CBF(MDUM,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,
      F,BO,NA,WK,NWK,IFAIL)
      GO TO 100
   12 K=5
      L=4
      NA=30
      XMIN=0.0
      XMAX=100.0
```

```
      PMIN=10.0207
      PMAX=20.414
      IFAIL=1
      NWK=30
C     CALL THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
      CALL EØ2CBF (MDU,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,F,
     *CO,NA,WK,NWK,IFAIL)
      GO TO 100
   13 K=5
      L=4
      NA=30
      XMIN=0.0
      XMAX=100.0
      PMIN=20.414
      PMAX=27.218
      IFAIL=1
      NWK=30
C     CALL THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
      CALL EØ2CBF (MDUM,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,F,
     *DO,NA,WK,NWK,IFAIL)
  100 RETURN
      END
```

```
      SUBROUTINE ENTHA (MDU,XDU,TDU)
C     THIS SUBROUTINE CALCULATES THE ENTHALPY OF THE SOLUTION
C     GIVEN THE CONCENTRATION AND THE TEMPERATURE USING THE NAG
C     LIBRARY
      COMMON/BLK2/T(20)
      REAL XDU,TDU,WK(100),FO(30),GO(42),HO(30),PO(12)
      DATA FO/-6.6586E+02,1.2289E-02,1.6225E+ 01,-5.5491E+00,
     *1.7900E+00,-3.4341E+01,3.9441E+01,-2.5658E+01,1.2577E+01,
     *-4.8478E+00,2.8283E+02,-2.7887E+01,1.9848E+01,-9.5408E+00,
     *3.6936E+00,-2.8241E+01,1.8012E+01,-1.2881E+01,5.8303E+00,
     *-2.3703E+00,-8.9504E+00,-9.2031E+00,6.5372E+00,-2.5547E+00,
     *1.1633E+00,2.9213E+00,3.1212E+00,-2.3136E+00,5.2531E-01,
     *-3.2884E-01/
      DATA GO/3.2255E+02,3.6289E+02,3.3806E+00,-7.2415E-01,1.4402E-02,
     *6.6832E-02,3.4532E+01,1.4225E+01,1.6401E+00,8.9253E-02,-7.0264E-02,
     *4.8604E-02,2.4860E+02,-2.9560E+00,-4.1666E-03,1.6996E-01,
     *-3.2549E-01,-3.8759E-01,-8.7244E+00,1.4846E+00,1.9850E-02,
     *-5.0182E-01,-3.3891E-01,-3.7317E-01,-1.9291E+01,-1.4657E+00,
     *-6.5413E-01,-5.4658E-01,-5.9881E-01,-5.1416E-01,4.6776E+00,
     *-4.9093E-01,-7.9984E-01,-5.4673E-01,-2.6851E-01,-3.1059E-01,
     *1.4003E+00,4.0202E-01,-4.5985E-01,-4.6205E-01,-6.0398E-01
     *9.9959E-02/
     DATA HO/1.2758E+03,2.4429E+02,5.2142E+01,2.9296E+01,8.2448E+00,
     *-8.7811E+01,6.5408E+01,4.8115E+01,2.7352E+01,7.8816E+00,1.6829E+02,
     *4.3925E+01,3.7903E+01,2.2064E+01,6.4675E+00,4.7818E+01,2.8686E+01,
     *2.4066E+01,1.4303E+01,4.1339E+00,5.2834E+00,1.3162E+01,1.2173E+01,
     *7.4215E+00,2.0025E+00,8.5591E-01,3.4482E+00,3.7938E+00,1.7389E+00,
     *-1.3170E-01/
      DATA PO/1.909E+03,1.7852E+02,4.1156E+00,-2.0197E+02,5.6062E+00,
     *3.6370E+00,3.2848E+01,-1.9626E+00,2.5930E+00,9.5341E+00,
     *-6.7737E-01,9.7004E-01/
```

```
      IF(TDU.LT.277.59) GO TO 22
      IF(TDU.LT.355.37) GO TO 23
      IF(TDU.LT.394.16) GO TO 24
      IF(TDU.LT.433.15) GO TO 25
   22 NA=30
      K=5
      L=4
      XMIN=0.0
      XMAX=100.0
      TMIN=244.26
      TMAX=277.59
      IFAIL=1
      NWK=30
C     CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
      CALL EØ2CBF (MDU,MDU,K,L,XDU,XMIN,XMAX,TDU,TMIN,TMAX,T,
     *FO,NA,WK,NWK,IFAIL)
   23 K=6
      L=5
      IFAIL=1
      XMIN=0.0
      XMAX=100.0
      TMIN=277.59
      TMAX=355.37
      NWK=42
C     CALLING THE SUBROUTINE EØ2CBF THE NAG LIBRARY
      CALL EØ2CBF (MDU,MDU,K,L,XDU,XMIN,XMAX,TDU,TMIN,TMAX,
     *T,GO,NA,WK,NWK,IFAIL)
   24 K=5
      L=4
      NA=30
      XMIN=0.0
      XMAX=70.0
      TMIN=355.37
      TMAX=394.16
      IFAIL=1
      NWK=0.30
```

```
C       CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
        CALL EØ2CBF (MDU,MDU,K,L,XDU,XMIN,XMAX,TDU,TMIN,TMAX,T,
       *HO,NA,WK,NWK,IFAIL)
        GO TO 101
   25   NA=12
        K=3
        L=2
        XMIN=0.0
        XMAX=50.0
        TMIN=394.16
        TMAX=433.15
        IFAIL=1
        NWK=120
C       CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
        CALL EØ2CBF (MDU,K,L,XDU,XMIN,XMAX,TDU,TMIN,TMAX,T,PO,
       *NA,WK,NWK,IFAIL)
  101   RETURN
        END
```

```
C       ***SUBROUTINE PRESSA***
C
C          THIS SUBROUTINE CALCULATE THE SATURATED PRESSURE OF THE
C          SOLUTION GIVEN THE CONCENTRATION AND THE SATURATED TEMPERATURE
C          USING THE NAG LIBRARY
           SUBROUTINE PRESSA (MDU,XDU,TDU)
           COMMON/BLK3/P(20)
           REAL XDU,TDU,WK(100),FO(30),GO(42),PO(12)
           DATA FO/3.2448E+00,1.5855E+00,3.9354E-01,4.9709E-02,6.1097E-04,
          *1.9682E+00,9.9792E-01,1.8535E-01,3.5790E-02,6.7818E-04,2.6682-01,
          *9.3827E-02,4.2136E-02,-9.5208E-03,-3.7834E-04,-4.3904E-01,
          *-9.5458E-02,-3.5255E-02,5.3924E-03,1.8212E-03,-3.3559E-02,
          *-1.3301E-02,1.0906E-02,-2.4895E-03,-2.0879E-03,5.6255E-02,
          *2.5078E-02,-1.9766E-04,-1.2476E-04,2.4761E-03/
           DATA GO/4.1695E+01,2.2234E+01,5.3339E+00,7.9489E-01,1.4056E-01,
          *1.0985E-01,2.5041E+01,1.2902E+01,2.8558E+00,3.6478E-01,6.0991E-02,
          *6.6399E-02,3.6000E+00,18077E+00,4.5798E-01,2.2088E-01,1.0704E-01,
          *8.8219E-02,-1.6133E+00,-4.3590E-01,2.5530E-01,3.2605E-01,1.7448E-01,
          *1.0163E-01,3.9136E-01,6.3426E-01,5.3495E 01,4.1737E-01,2.1193E-01,
          *1.2323E-01,1.0303E+00,7.3074E-01,4.3854E-01,3.0260E-01,1.6554E-01,
          *7.9300E-02,3.3085E-01,2.8291E-01,2.2625E-01,1.7384E-01,9.3184E-02,
          *4.0563E-02/
           DATA PO/1.3104E+02,3.9958E+01,3.9193E+00,7.5969E+01,2.1914E+01,
          *1.7746E+00,1.7146E+01,4.9447E+00,4.6471E-02,7.6560E-01,5.7828E-01,
          *-2.7343E-01/
C          IF(TDUM.LT.273.04) GO TO 22
           IF(TDUM.LT.366.48) GO TO 23
           IF(TDUM.LT.433.15) GO TO 24
      22   NA=30
           K=5
           L=4
           XMIN=10.0
           XMAX=100.0
           TMIN=222.04
           TMAX=272.04
           IFAIL=1
           NWK=30
```

```
C       CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
        CALL EØ2CBF (MDU,MDU,K,L,XDU,XMIN,XMAX,TDU,TMIN,TMAX,P,FO,
       *NA,WK,NWK,IFAIL)
        GO TO 101
   23   K=6
        L=5
        XMIN=0.0
        XMAX=100.0
        TMIN=272.04
        TMAX=366.48
        IFAIL=1
        NWK=42
        NA=42
C       CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
        CALL EØ2CBF (MDU,MDU,K,L,XDU,XMIN,XMAX,TDU,TMIN,TMAX,P,GO,
       *NA,WK,NWK,
        GO TO 101
   24   NA=12
        K=3
        L=2
        XMIN=0.0
        XMAX=70.0
        TMIN=366.48
        TMAX=433.15
        IFAIL=1
        NWK=12
C       CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
        CALL EØ2CBF (MDU,MDU,K,L,XDU,XMIN,XMAX,TDU,TMIN,TMAX,P,
       *PO,NA,WK,NWK,IFAIL)
  101   RETURN
        END
```

```
C   ***SUBROUTINE CONCA***
C      THIS SUBROUTINE CALCULATES THE AMOUNT OF WATER IN THE SATURATED
C      VAPOUR GIVEN THE SATURATED PRESSURE AND SATURATED LIQUID COMPOSI-
C      TION USING THE NAG LIBRARY.
       SUBROUTINE CONC (MDUM,XDUM,PDUM)
       COMMON/BLK4/P(20)
       REAL XDUM,PDUM,WK(20),AO(20),BO(20),CO(20),DO(20),EO(20)
       DATA AO/4.4322E+01,6.4803E+00,-8.3306-01,3.0797E-01,-3.1957E+01,
      *-3.6908E+00,6.2418E-01,-2.2386E-01,1.3859E+01,9.1089E-1,-1.8953E-01,
      *9.9931E-02,-4.1426E+00,-6.8178E-02,1.5439E-02,-1.0640E-02,8.1411E-01,
      *-7.4702E-02,-8.1719E-03,-2.1686E-02/
       DATA BO/6.7780E+01,5.4233E+00,-4.3107E-01,8.7446E-04,-4.3722E+01,
      *-2.3697E+00,2.6328E-02,6.9108E-03,-3.9853E+00,1.3173E-01,-2.1862E-02,
      *-7.5818E-04,1.8745E-01,-1.9827E-01,3.5271E-02,4.5584E-04/
       DATA CO/1.7326E+00,4.3523E-01,-1.5428E-04,-4.8561E-04,-1.0972E+00,
      *-2.6241E-01,-1.0997E-03,3.9816E-04,4,4516E-01,1.0318E-01,3.9169E-03,
      *3.0312E-03,-1.7375E-01,-3.7716E-02,-4.4397E-03,-1.0980E-03,6.0762E-02,
      *1.0568E-02,2.2412E-03,-2.1486E-03/
       DATA DO/8.4643E+01,3.3098E+00,-8.9220E-02,1.3087E-02,-5.0412E+0-1,
      *-1.1951E+00,2.0089E-02,2.5629E-03,1.7012E+01,4.4519E-02,6.6964E-03,
      *-1.0070E-02,-3.5300E+00,1.2305E-01,1.2014E-02,4.5085E-03,-2.2372E-01,
      *-1.4930E-02,-3.3482E-03,-3.3217E-03/
       DATA EO/3.4699E+00,4.4585E-01,1.2187E-02,-1.2318E-03,-2.1525E+00,
      *-2.6951E-01,-5.2767E-03,5.0181E-04,8.5657E-01,9.1178E-02,-
      *-8.7337E-03,-2.1337E-03,-3.2608E-01,-2.5345E-02,9.7675E-03,
      *-5.3357E-04,9.7875E-02,3.4599E-03,-2.0249E-03,2.9406E-03/
       NA=20
       K=4
       L=3
       IFAIL=1
       NWK=20
       IF(PDUM.LT.6.8046.AND.XDUM.LE.50.0) GO TO 22
       IF(PDUM.LT.17.011.AND.XDUM.LE.50.0) GO TO 23
       IF(PDUM.LT.17.011.AND.XDUM.LE.90.0) GO TO 24
       IF(PDUM.LT.28.218.AND.XDUM.LE.50,0) GO TO 25
       IF(PDUM.LT.28.218.AND.XDUM.LE.90.0) GO TO 26
```

```
C       **  NOTE FOR PRESSURE RANGE 0.81655 BAR TO 6.8046 BARS NO
C           POLYNOMIAL EQUATION WAS DEVELOPED BECAUSE OF LARGE % PERCEN-
C           TAGE DEVIATION FROM THE STATE POINT FOR OTHERS THE GANGE OF
C           CONCENTRATION SHOULD NOT BE LESS THAN 10.0WT% OR GREATER THAN
C           90.0WT%
   22       PMIN=0.81655
            PMAX=6.8046
            XMIN=10.0
            XMAX=50.0
C           CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
            CALL EØ2CBF(MDUM,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,P,AO,
           *NA,WK,NWK,IFAIL)
   23       PMIN=6.8046
            PMAX=17.011
            XMIN=10.0
            XMAX=50.0
C           CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
            CALL EØ2CBF(MDUM,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,P,
           *BO,NA,WK,NWK,IFAIL)
            GO TO 101
   24       PMIN=6.8046
            PMAX=17.011
            XMIN=50.0
            XMAX=90.0
C           CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
            CALL EØ2CBF(NDUM,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,P,
           *CO,NA,WK,NWK,IFAIL)
            GO TO 101
   25       PMIN=17.011
            PMAX=28.218
            XMIN=10.0
            XMAX=50.0
```

```
C           CALLING THE SUBROUTINE EØ2CBF FROM THE NAG LIBRARY
            CALL EØ2CBF(MDUM,MDUM,XMUM,XMIN,XMAX,PDUM,PMIN,PMAX,P,
           *DO,NA,WK,NWK,IFAIL)
     46     PMIN=17.011
            PMAX=28.218
            XMIN=50.0
            XMAX=90.0
C           CALLING THE SUBROUTINE EØ2CBF FROM THE NAGE LIBRARY
            CALL EØ2CBF(MDUM,MDUM,K,L,XDUM,XMIN,XMAX,PDUM,PMIN,PMAX,P,
           *EO,NA,WK,NWK,IFAIL)
    101     RETURN
            END
```

```
C   **   SUBROUTINE BIOL   **
C        THIS ROUTINE CALCULATE DIRECTLY TEMPERATURE OF AMMONIA-WATER
C        MIXTURE GIVEN THE PRESSURE AND CONCENTRATION
         SUBROUTINE BIOL (PGEN,XGEN,TGEN)

         DATA AO/1.1808E+03,2.2756E+01,-2.1393E+00,-1.3296E+02,
        *-2.0567E+00,2.0270E-01,3.7784E+01,1.8645E-011.5058E-01,
        *-1.7086E+00,1.1242E-01,-2.4552E-02/
         DATA BO/1.3432E+03,5.05773E+01,-7.4615E+00,1.5989E+00,
        *-4.4155E-01,-1.4677E+02,-4.4559E+00,7.0187E-01-1.0395E-01,
        *2.7736E-02,3.8944E+01,2.9585E-01,-2.0679E-02,8.4342E-03,
        *-2.4076E-02,-7.7446E-01,2.8930E-01,-3.0539E-02,-1.9785E-02,
        *2.6029E-02,-1.0967E+00,-4.3997E-01,-2.7643E-02,3.8438E-02,
        *9.3406E-03,-1.2267E+00,3.5413E-01,-3.2161E-02,-2.1582E-02,
        *-2.1347E-02/
         DATA CO/1.4940E+03,2.8658E+01,-2.1132E+00,2.1205E-01,
        *-8.1942E-03,-1.5924E+02,-1.9905E+00,3.6542E-01,-5.9377E-02,
        *-2.6529E-02,3.9428E-01,-4.8887E-02,-2.0674E-02,12702E-02,
        *3.5557E-03,-1.4843E-01,-4.0882E-02,-8.9779E-02,2.0667E-02,
        *1.8525E-02,-2.2580E+00,-2.7252E-02,1.0244E-01,-2.9635E-02,
        *-8.9233E-03,-2.9753E-01,2.1228E-02,-1.3925E-01,2.6453E-02,
        *1.2928E-02/
         DATA DO/1.5745E+03,1.3224E+01,-2.3889E-01,7.6243E-02,
        *-2.0605E-02,-1.6455E+02,-9.0433E-01,5.6693E-02,1.4544E-02,
        *-3.9139E-03,3.8984E+01,-1.2025E-01,6.9276E-02,4.0042E-03,
        *-6.5407E-03,-4.3264E-01,-1.5508E-02,6.7470E-02,8.4691E-03,
        *-2.0490E-02,-2.0433E+00,9.8179E-02,-5.9511E-02,2.0067E-02,
        *3.4546E-04,-7.4846E-01,-5.0549E-02,4.4410E-02,-1.5876E-02,
        *9.1042E-03/
        FI(PGEN.LT.2.0414) GO TO 10
        IF(PGEN.LT.10.0207) GO TO 11
        IF(PGEN.LT.20.414) GO TO 12
        IF(PGEN.LT.27.218) GO TO 13
```

```
10  U=(2.0*PGEN-(2.0414+0.81655))/(2.0414-0.81655)
    V=(2.0*XGEN-100.0)/100.0
    GO TO 50
11  U=(2.0*PGEN-(10.0207+2.0414))/(10.0207-2.0414)
    V=(2.0*XGEN-(100.0))/(100.0)
    GO TO  50
12  U=(2.0*PGEN-(20.414+10.0207))/(20.414-10.0207)
    V=(2.0*XGEN-100.0)/100.0
    GO TO 50
13  U=(2.0*PGEN-(27.218+20.414))/(27.218-20.414)
    V=(2.0*XGEN-100.0)/100.0
50  X1=COS(ACOS(U))
    X2=COS(2.0*ACOS(U))
    X3=COS(3.0*ACOS(U))
    X4=COS(4.0*ACOS(U))
    Y1=COS(ACOS(V))
    Y2=COS(2.0*ACOS(V))
    Y3=COS(3.0*ACOS(V))
    Y4=COS(4.0*ACOS(V))
    Y5=COS(5.0*ACOS(V))
    IF(PGEN.LT.2.0414) GO TO 14
    IF(PGEN.LT.10.0207) GO TO 15
    IF(PGEN.LT.20.414) GO TO 16
    IF(PGEN.LT.27.218) GO TO 17
14  A=0.25*AO(1)+0.5*AO(2)*X1+0.5*AO(3)*X2
    B=Y1*(0.5*AO(4)+AO(5)*X1+AO(6)*X2)
    C=Y2*(0.5*AO(7)+AO(8)*X1+AO(9)*X2)
    D=Y3*(0.5*AO(10)+AO(11)*X1+AO(12)*X2)

    TGEN=A+B+C+D
    GO TO 100
15  A=0.25*BO(1)+0.5*BO(2)*X1+0.5*BO(3)*X2+0.5*BO(4)* X3+0.5*BO(5)*X4
    B=Y1*(0.5*BO(6)+BO(7)*X1+BO(8)*X2+BO(9)*X3+BO(10)*X4)
    C=Y2*(0.5*BO(11)+BO(12)*X1+BO(13)*X2+BO(14)*X3+BO(15)*X4)
    D=Y3*(0.5*BO(16)+BO(17)*X1+BO(18)*X2+BO(19)*X3+BO(20)*X4)
```

```
      E=Y4*(0.5*BO(21)+BO(22)*X1+BO(23)*X2+BO(24)*X3+BO(25)*X4)
      F=Y5*(0.5*BO(26)+BO(27)*X1+BO(28)*X2+BO(29)*X3+BO(30)*X4)
      TGEN=A+B+C+D+E+F
      GO TO 100
   16 A=0.25*CO(1)+0.5*CO(2)*X1+0.5*CO(3)*X2+0.5*CO(4)*X3+0.5*CO(5)*X4
      B=Y1*(0.5*CO(6)+CO(7)*X1+CO(8)*X2+CO(9)*X3+CO(10)*X4)
      C=Y2*(0.5*CO(11)+CO(12)*X1+CO(13)*X2+CO(14)*X3+CO(15)*X4)
      D=Y3*(0.5*CO(16)+CO(17)*X1+CO(18)*X2+CO(19)*X3+CO(20)*X4)
      E=Y4*(0.5*CO(21)+CO(22)*X1+CO(23)*X2+CO(24)*X3+CO(25)*X4)
      F=Y5*(0.5*CO(26)+CO(27)*X1+CO(28)*X2+CO(29)*X3+CO(30)*X4)
      TGEN=A+B+C+D+E+F
      GO TO 100
   17 A=0.25*DO(1)+0.5*DO(2)*X1+0.5*DO(3)*X2+0.5*X3*DO(4)+0.5*DO(5)*X4)
      B=Y1*(0.5*DO(6)+DO(7)*X1+DO(8)*X2+DO(9)*X3+DO(10)*X4)
      C=Y2*(0.5*DO(11)+DO(12)*X1+DO(13)*X2+DO(14)*X3+DO(15)*X4)
      D=Y3*(0.5*DO(16)+DO(17)*X1+DO(18)*X2+DO(19)*X3+DO(20)*X4)
      E=Y4*(0.5*DO(21)+DO(22)*X1+DO(23)*X2+DO(24)*X3+DO(25)*X4)
      F=Y5*(0.5*DO(26)+DO(27)*X1+DO(28)*X2+DO(29)*X3+DO(30)*X4)
      TGEN=A+B+C+D+E+F
  100 RETURN
      END
```

```
***  SUBROUTINE ENTH  **
     THIS SUBROUTINE CALCULATES DIRECTLY THE SOLUTION ENTHALPY
     GIVEN THE TEMPERATURE AND LIQUID CONCENTRATION
     SUBROUTINE ENTH (TGEN,XGEN,HA5)
     DATA AO/-6.6585E+02,1.2289E+02,1.6225+01,-5.5491E+00,1.7900E+00,
    *-3.4341E+01,3.9441E+01,-2.5658E+01,1.2577E+01,-4.8478E+00,
    *2.8283E+02,-2.7887E+01,1.9848E+01,-9.5408E+00,3.6936E+00,
    *-2.8241E+01,1.8012E+01,-1.2881E+01,5.8303E+00,-2.3703E+00,
    *-89504E+00,-9.2031E+00,6.5372E+00,-2.5547E+00,1.1633E+00,
    *2.9213E+00,3.1212E+00,-2.3136E+00,5.2531E-01,-3.2884E-01/
     DATA BO/3.2255E+02,3.2689E+02,3.3806E+00,-7.2415E-01,
    *1.4402E-01,6.6832E-02,3.4532E+01,1.4225E+01,1.6401E+00,
    *8.9253E-02,-7.0244E-02,4.8604E-02,2.4860E+02,-2.9560E+00,
    *-4.1666E-03,1.6996E-01,-3.2549E-01,-3.8759E-01,-8.7244E+00,
    *1.4846E+00,1.9850E-02,-5.0182E01,-3.3891E-01,-3.7371E-01,
    *-1.9291E+01,-1.4657E+00,-6.5413E-01,-5.4658E-01,-5.9881E-01,
    *-5.1416E-01,4.6776E+00,-4.9093E-01,-7.9984E-01,-5.4673E-01,
    *-2.6951E-01,-3.1059E-01,1.4003E+00,4.0202E-01,-4.5985E-01,
    *-4.6205E-01,-6.0398E-01,9.9959E-02/
     DATA CO/1.2758E+03,2.4429E+02,5.2142E+01,,2.9296E+01,8.2448E+00,
    *-8.7811E+01,6.5408E+01,4.8115E+01,2.7352E+01,7.8816E+00,1.6829E+02,
    *4.3925E+01,3.7903E+01,2.2064E+01,6.4675E+00,4.7818E+01,2.8686E+01,
    *2.4060E+01,1.4303E+01,4.1339E+00,5.2834E+00,1,3162E+01,1.2173E+01,
    *7.4215E+00,2.0025E+00,8.5591E-01,3.4482E+00,3.7938E+00,1.7389E+00,
    *-1.3170E-01/
     DATA DO/1.9090E+03,3.7852E+02,4.1156E+00,-2.0197E+02,5.6062E+00,
    *3.6370E+00,3.2849E+01,-1.9626E+00,2.5930E+00,9.5341E+00,
    *-6.7737E-01,9.7004E-01/
     IF(TGEN.LT.277.59) GO TO 22
     IF(TGEN.LE.355.37) GO TO 23
     IF(TGEN.LE.394.16) GO TO 24
     IF(TGEN.LE.433.15) GO TO 25
```

```
22    U=(2.0*TGEN-(277.59+244.26))/(277.59-244.26)
      V=(2.0*XGEN-(100.0))/(100.0)
      GO TO 30
23    U=(2.0*TGEN-(355.37+277.59))/(355.37-277.59)
      V=(2.0*XGEN-(100.0))/(100.0)
      GO TO 30
24    U=(2.0*TGEN-(394.16+355.37))/(394.16-355.73)
      V=(2.0*XGEN-(70.0))/(70.0)
      GO TO 30
25    U=(2.0*TGEN-(433.15+394.26))/(433.15-394.26)
      V=(2.0*XGEN-(50.0))/(50.0)
30    X1=COS(ACOS(U))
      X2=COS(2.0*ACOS(U))
      X3=COS(3.0*ACOS(U))
      X4=COS(4.0*ACOS(U))
      X5=COS(5.0*ACOS(U))
      Y1=COS(ACOS(V))
      Y2=COS(2.0*ACOS(V))
      Y3=COS(3.0*ACOS(V))
      Y4=COS(4.0*ACOS(V))
      Y5=COS(5.0*ACOS(V))
      Y6=COS(6.0*ACOS(V))
      IF(TGEN.LE.277.59) GO TO 26
      IF(TGEN.LE.355.37) GO TO 27
      IF(TGEN.LE.394.16) GO TO 28
      IF(TGEN.LE.433.15) GO TO 29
26    A=0.25*AO(1)+0.5*AO(2)*X1+0.5*AO(3)*X2+0.5*AO(4)*X3+0.5*AO(5)*X4
      B=Y1*(0.5*AO(6)+AO(7)*X1+AO(8)*X2+AO(9)*X3+AO(10)*X4)
      C=Y2*(0.5*AO(11)+AO(12)*X1+AO(13)*X2+AO(14)*X3+AO(15)*X4)
      D=Y3*(0.5*AO(16)+AO(17)*X1+AO(18)*X2+AO(19)*X3+AO(20)*X4)
      E=Y4*(0.5*AO(21)+AO(22)*X1+AO(23)*X2+AO(24)*X3+AO(25)*X4)
      F=Y5*(0.5*AO(26)+AO(27)*X1+AO(28)*X2+AO(29)*X3+AO(30)*X4)
      HA5=A+B+C+D+E+F
```

```
      GO TO 101
27    A=0.25*BO(1)+0.5*BO(2)*X1+0.5*BO(3)*X2+0.5*BO(4)*X3+0.5*BO(5)*X4+
     *0.5*BO(6)*X5
      B=Y1*(0.5*BO(7)+BO(8)*X1+BO(9)*X2+BO(10)*X3+BO(11)*X4+BO(12)*X5)
      C=Y2*(0.5*BO(13)+BO(14)*X1+BO(15)*X2+BO(16)*X3+BO(17)*X4+BO(18)*X5)
      D=Y3*(0.5*BO(19)+BO(20)*X1+BO(21)*X2+BO(22)*X3+BO(23)*X4+BO(24)*X5)
      E=Y4*(0.5*BO(25)+BO(26)*X1+BO(27)*X2+BO(28)*X3+BO(29)*X4+BO(30)*X5)
      F=Y5*(0.5*BO(31)+BO(32)*X1+BO(33)*X2+BO(34)*X3+BO( 35)*X4+BO(36)*X5)
      G=Y6*(0.5*BO(37)+BO(38)*X1+BO(39)*X2+BO(40)*X3+BO(41)*X4+BO(42)*X5)
      HA5=A+B+C+D+E+F+G
      GO TO 101
28    A=.25*CO(1)+0.5*CO(2)*X1+0.5*CO(3)*X2+0.5*CO(4)*X3+0.5*CO(5)*X4
      B=Y1*(0.5*CO(6)+CO(7)*X1+CO(8)*X2+CO(9)*X3+CO(10)*X4)
      C=Y2*(0.5*CO(11)+CO(12)*X1+CO(13)*X2+CO(14)*X3+CO(15)*X4)
      D=Y3*(0.5*CO(16)+CO(17)*X1+CO(18)*X2+CO(19)*X3+CO(20)*X4)
      E=Y4*(0.5*CO(21)+CO(22)*X1+CO(23)*X2+CO(24)*X3+CO(25)*X4)
      F=Y5*(0.5*CO(26)+CO(27)*X1+CO(28)*X2+CO(29)*X3+CO(30)*X4)
      HA5=A+B+C+D+E+F
29    A=0.25*DO(1)+0.5*DO(2)*X1+0.5*DO(3)*X3
      B=Y1*(.5*DO(4)+DO(5)*X1+DO(6)*X2)
      C=Y2*(0.5*DO(7)+DO(8)*X1+DO(9)*X2)
      D=Y3*(0.5*DO(10)+DO(11)*X1+DO(12)*X2)
      HA5=A+B+C+D
101   RETURN
      END
```

```
C       ***SUBROUTINE PRESS***
C          THE SUBROUTINE CALCULATES DIRECTLY THE SATURATED PRESSURE
C          GIVEN THE SATURATED TEMPERATURE AND THE CONCENTRATION OF
C          THE SOLUTION.
           SUBROUTINE PRESS (TGEN,XGEN,TGEN)
           DATA AO/3.2448E+00,1.5855E+00,3.9354E-01,2.9705E-02,6.1097E-04,
          *1.9682E+00,9.9792E-01,1.8545E-01,3.5790E-02,6.7818E-04,2.6682E-01,
          *9.3827E-02,4.2163E-02,-9.5208E-03,-3.7834E-04,-2.3900E-01,
          *-9.5458E-02,-3.5255E-02,5.3927E-03,1.8212E-03,-3.3559E-02,
          *-1.3301E-02,1.0906E-02,-2.4895E-03,-2.0879E-03,5.6255E-02,
          *2.5078E-02,-1.9766E-04,-1.2476E-04,2.4761E-03/
           DATA BO/4.1695E+01,2.2234E+01,5.3339E+00,7.9489E-01,
          *1.4056E-01,1.0985E-01,2.5041E+01,1.2902E+01,2.8558E+00,
          *3.6478E-01,6.6991E-02,6.6399E-02,3.6006E+00,1.8077E+00,
          *4.5798E-01,2.2088E-01,1.0704E-01,8.8219E-02,-1.6133E+00,
          *-4.3590E-01,2.5530E-01,3.2606E-01,1.7448E-01,1.0163E-01,
          *3.9136E-01,6.3426E-01,5.3495E-01,4.1737E-01,2.1193E-01,
          *1.2323E-01,1.0303E+00,7.3074E-01,4.3854E-01,3.0260E-01,
          *1.6554E-01,7.9300E-02,3.3085E-01,2.8291E-01,2.2625E-01,
          *1.7384E-01,93184E-02,4.0563E-02/
           DATA DO/1.3104E+02,3.9958E+01,3.9193E+00,7.5909E+01,
          *2.1914E+01,1.7745E+00,1.7146E+01,4.9447E+00,4.6471E-02,
          *7.6566E-01,5.7828E-01,-2.7343E-01/
           IF(TGEN.LT.272.04) GO TO 21
           IF(TGEN.LT.366.48) GO TO 22
           IF(TGEN.LT.433.15) GO TO 23
21         U=(2.0*TGEN-(272.04+222.04))/(272.04-222.04)
           V=(2.0*XGEN-(100.0+10.0))/(100.0-10.0)
           GO TO 31
22         U=(2.0*TGEN-(366.48+272.04))/(366.48-272.04)
           V=(2.0*TGEN-100.0)/(100.0)
           GO TO 31
23         U=(2.0*TGEN-(433.15+366.48))/433.15-366.48)
           V=(2.0*XGEN-(70.0))/(70.0)
```

```
 31     X1=COS(ACOS(U))
        X2=COS(2.0*ACOS(U))
        X3=COS(3.0*ACOS(U))
        X4=COS(4.0*ACOS(U))
        X5=COS(5.0*ACOS(U))
        Y1=COS(ACOS(V))
        Y2=COS(2.0*ACOS(V))
        Y3=COS(3.0*ACOS(V))
        Y4=COS(4.0*ACOS(V))
        Y5=COS(5.0*ACOS(V))
        Y6=COS(6.0*ACOS(V))
        IF(TGEN.LT.272.04) TO GO 41
        IF(TGEN.LT.366.48) GO TO 42
        IF(TGEN.LT.433.15) GO TO 43
 41     A=0.25*AO(1)+0.5*AO(2)*X1+0.5*AO(3)*X2+0.5*AO(4)*X3+0.5*AO(5)*X4
        B=Y1*(0.5*AO(6)+AO(7)*X1+AO(8)*X2+AO(9)*X3+AO(10)*X4)
        C=Y2*(0.5*AO(11)+AO(12)* X1+AO(13)*X2+AO(14)*X3+AO(15)*X4)
        D=Y3*(0.5*AO(16)+AO(17)*X1+AO(18)*X2+AO(19)*X3+AO(20)*X4)
        E=Y4*(0.5*AO(21)+AO(22)*X1+AO(23)*X2+AO(24)*X3+AO(25)*X4)
        F=Y5*(0.5*AO(26)+AO(27)*X1+AO(28)*X2+AO(29)*X3+AO(30)*X4)
        HA5=A+B+C+D+E+F
        GO TO 101
 42     A=0.25*BO(1)+0.5*BO(2)*X1+0.5*BO(3)*X2+0.5*BO(4)*X3+0.5*BO(5)
       **X4+0.5*+BO(6)*X5)
        B=Y1*(0.5*BO(7)+BO(8)*X1+BO(9)*X2+BO(10)*X3+BO(11)*X4+BO(12)*X5)
        C-Y2*(0.5*BO(13)+BO(14)*X1+BO(15)*X2+BO(16)*X3+BO(17)
       **X4+BO(18)*X5)
        D=Y3*(0.5*BO(19)+BO(20)*X1+BO(21)*X2+BO(22)*X3+BO(23)*X4+BO(24)*X5)
        E=Y4*(0.5*BO(25)+BO(26)*X1+BO(27)*X2+BO(28)*X3+BO(29)*X4+BO(30)*X5)
        F=Y5*(0.5*BO(31)+BO(32)*X1+BO(33)*X2+BO(34)*X3+BO(35)*X4+BO(36)*X5)
        G=Y6*(0.5*BO(37)+BO(38)*X1+BO(39)*X2+BO(40)*X3+BO(41)*X4+BO(42)*X5)
        HA5=A+B+C+D+E+F+G
        GO TO 101
 43     A=0.25*DO(1)+0.5*DO(2)*X1+0.5*DO(3)*X2)
        B=Y1*(0.5*DO(4)+DO(5)*X1+DO(6)*X2)
        C=Y2*(0.5*DO(7)+DO(8)*X1+DO(9)*X2)
        D=Y3*(0.5*DO(10)+DO(11)*X1+DO(12)*X2)
        HA5=A+B+C+D
101     RETURN
        END
```

```
C   ***   SUBROUTINE CONC*
C         THIS SUBROUTINE CALCULATES DIRECTLY THE AMOUNT OF WATER IN
C         SATURATED VAPOUR GIVEN THE SATURATED PRESSURE AND SATURATED
C         LIQUID COMPOSITION.
          SUBROUTINE CONC(PGEN,XGEN,CON)
          DATA AO/4.4322E+01,6.4803+00,-8.3306E-01,3.0797E-01,-3.1957E+01,
         *-3.6908E+00,6.2418E-01,-2.2386E-01,1.3859E+01,9.1089E-01,-1.8953E-01,
         *9.9931E-02,-4.1426E+00,-6.8178E-02,1.5439E-02,-1.0640E-02,8.1411E-02,
         *-7.4762E-02,-8.1719E-03,-2.1086E-02/
          DATA BO/6.7780E+01,5.4233E+00,-4.3107E-01,8.7446E-04,-4.3722E+01,
         *-2.3697E+00,2.6328E-01,-5.7061E-03,1.6273E+01,3.5613E-01,-4.9534E-02,
         *6.9108E-03,-3.9853E+00,1.3173E-01,-2.1862E-02,-7.5818E-04,1.8745E-01,
         *-1.9827E-01,3.5271E-02,4.5584E-04/
          DATA CO/1.7326E+00,4.3523E-01,-1.542E-04,-4.8561E-04,-1,0972E+00,-
         *2.6241E-01,-1.0897E-03,3.9816E-04,4.4516E-01,1.0318E-01,3.9169E-03,
         *3.0312E-03,-1.7375E-01,-3.7716E-02,-4.4397E-03,-1.0980E-03,6.0762E-02,
         *1.0568E-02,2.2412E-03,-2.1486E-03/
          DATA DO/8.4643E+01,3.3098E+00,-8.9220E02,1.3087E-02,-5.0412E+01,
         *-1,1951E+00,2.0089E-02,2.5629E-03,1.7012E+01,4.4519E-02,6.6964E-03,
         *-1.0070E-02,-3.5300E+00,1.2305E-01,1.2014E-02,4.5085E03,-2.2372E-01,
         *-1.4930E-02,-3.3482E-03,-3.3217E-03/
          DATA EO/3.4699E+00,4.4585E-01,1.2187E-02,-1.2518E-03,-2.1525E+00,
         *-2.6951E-01,-5.2767E-03,5.0131E-04,8.5657E-01,9.1178E-02,-8.7337E-03,
         *-2.1337E-03,-3.2608E-01,-2.5345E-02,9.7675E-03,-5.3357E-04,
         *9.7875E-02,3.4599E-03,-2.0249E-03,2.9406E-03/
          IF(PGEN.LT.6.8046.AND.XGEN.LE.50.0) GO TO 22
          IF(PGEN.LT.17.011.AND.XGEN.LE.50.0) GO TO 23
          IF(PGEN.LT.17.011.AND.XGEN.LE.90.0) GO TO 24
          IF(PGEN.LT.28.218.AND.XGEN.LE.50.0) GO TO 25
          IF(PGEN.LT.28.218.AND.XGEN.LE.90.0) GO TO 26
   22     U=(2.0*PGEN-(6.8046+0.81655))/(6.8046-0.81655)
          V=(2.0*XGEN-(50.0+10.0))/(50.0-10.0)
          GO TO 50
```

```
23    U=(2.0*PGEN-(17.011+6.8046))/(17.011-6.8046)
      V=(2.0*XGEN-(50.0+10.0))/(50.0-10.0)
      GO TO 50
24    U=(2.0*PGEN-(17.011+6.8046))/(17.011-6.8046)
      V=(2.0*XGEN-(90.0+50.0))/(90.0-50.0)
      GO TO 50
25    U=(2.0*PGEN-(27.218+17.011))/(27.218-17.011)
      V=(2.0 GEN-(50.0+10.0))/(50.0-10.0)
      GO TO 50
26    U=(2.0*PGEN-(27.218+17.011))/(27.218-17.011)
      V=(2.0*DGEN-(90.0+50.0))/(90.0-50.0)
50    X1=COS(ACOS(U))
      X2=COS(2.0*ACOS(U))
      X3=COS(3.0*ACOS(U))
      Y1=COS(ACOS(V))
      Y2=COS(2.0*ACOS(V))
      Y3=COS(3.0*ACOS(V))
      Y4=COS(4.0*ACOS(V))
      IF(PGEN.LT.6.8046.AND.XGEN.LE.50.0) GO TO 30
      IF(PGEN.LT.17.011.AND.XGEN.LE.50.0) GO TO 31
      IF(PGEN.LT.17.011.AND.XGEN.LE.90.0) GO TO 32
      IF(PGEN.LT.28.218.AND.XGEN.LE.90.0) GO TO 33
      IF(PGEN.LT.28.218.AND.XGEN.LE.90.0) GO TO 34
30    A=0.25*AO(1)+0.5*AO(2)*X1+0.5*AO(3)*X2+AO(4)*0.5*X3
      B=Y1*(0.5*AO(5)+AO(6)*X1+AO(7)*X2+AO(8)*X3)
      C=Y *(0.5*AO(9)+AO(10)*X1+AO(11)*X2+AO(12)*X3)
      D=Y3*(0.5*AO(13)+AO(14)*X1+AO(15)*X2+AO(16)*X3)
      E=Y4*(0.5*AO(17)+AO(18)*X1+AO(19)*X2+AO(20)*X3)
      CON=A+B+C+D+E
      GO TO 101
31    A=0.25*BO(1)+0.5*BO(2)*X1+0.5*BO(3)*X2+0.5*BO(4)*X3)
      B=Y1*(0.5*BO(5)+BO(6)*X1+(7)*X2+BO(8)*X3)
      C=Y2*(0.5*BO(9)+BO(10)*X1+BO(11)*X2+BO(12)*X3)
      D=Y3*(0.5*BO(13)+BO(14)*X1+BO(15)*X2+BO(16)*X3)
      E=Y4*(0.5*BO(17)+BO(18)*X1+BO(19)*X2+BO(20)*X3)
      CON=A+B+C+D+E
```

```
32    A=0.25*CO(1)+0.5*CO(2)*X1+0.5*CO(3)*X2+0.5*CO(4)*X3
      B=Y1*(0.5*CO(5)+CO(6)*X1+CO(7)*X2+CO(8)*X3)
      C=Y2*(0.5*CO(9)+CO(10)*X1+CO(11)*X2+CO(12)*X3)
      D=Y3*(0.5*CO(13)+CO(14)*X1+CO(15)*X2+CO(16)*X3)
      E=Y4*(0.5*CO(17)+CO(18)*X1+CO(19)*X2+CO(20)*X3)
      CON=A+B+C+D+E
      GO TO 101
33    A=0.25*DO(1)+0.5*DO(2)*X1+0.5*DO(3)*X2+0.5*DO(4)*X3
      B=Y1*(0.5*DO(5)+DO(6)*X1+DO(7)*X2+DO(8)*X3)
      C=Y2*(0.5*DO(9)+DO(10)*X1+DO(11)*X2+DO(12)*X3)
      D=Y3*(0.5*DO(13)+DO(14)*X1+DO(15)*X2+Do(16)*X3)
      E=Y4*(0.5*DO(17)+DO(18)*X1+DO(19)*X2+DO(20)*X3)
      CON=A+B+C+D+E
      GO TO 101
      A=0.25*EO(1)+0.5*EO(2)*X1+0.5*EO(3)*X2+0.5*EO(4)*X3
      B=Y1*(0.5*EO(5)+EO(6)*X1+EO(7)*X2+EO(8)*X3)
      C=Y2*(0.5*EO(9)+EO(10)*X1+EO(11)*X2+EO(12)*X3)
      D=Y3*(0.5*EO(13)+EO(14)*X1+EO(15)*X2+EO(16)*X3)
      E=Y4*(0.5*EO(17)+EO(18)*X1+EO(19)*X2+EO(20)*X3)
      CON=A+B+C+D+E
      RETURN
      END
```