POTENTIALLY VIABLE SOLAR POWERED APPLIANCES COOLING AND DISTILLATION

BY

NIMA ASHRAFI

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School of Electronic and Mechanical
Engineering, Institute of Technology,
Sligo
Ballinode, Sligo
Republic of Ireland

The work described in this thesis has not been accepted for any degree and is not being concurrently submitted in the candidature for any degree.

Candidate: ....

Supervisor: ....

Summary

In this thesis, firstly, a comprehensive review of all the existing solar powered refrigeration systems has been presented.

Following this four different approaches in solar powered refrigeration were examined by developing four different prototype systems and assessing their individual performances. A number of major breakthroughs, in the field of solar powered refrigeration were achieved.

Also a high output solar powered desalination unit was developed using the ground-breaking multi-cycle intermittent solar powered cooling technique used for rapid condensation and increased output.

Three papers have been published in connection with the works described in this thesis. The new pioneering approaches used in solar powered refrigeration and water desalination are the subject of two patent applications.

Acknowledgements

It is a pleasure to thank, most sincerely, all the people who contributed to the work reported in this thesis.

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The help and encouragement I received from my supervisor Prof. H. Valizadeh during the entire work is greatly appreciated.

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The author is indebted to his friends Misters John Hession, Sean Shannon and Patrick Cunningham for all their encouragement, help and support.

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Abstract

Stand alone solar powered refrigeration and water desalination, two of the most popular and sought after applications of solar energy systems, have been selected as the topic of research for the works presented in this thesis.

The water desalination system based on evaporation and condensation was found to be the most suitable one to be powered by solar energy. It has been established that high-output fast-response solar heat collectors used to achieve high rates of evaporation and reliable solar powered cooling system for faster rates of condensation are the most important factors in achieving increased outputs in solar powered desalination systems.

Comprehensive reviews of Solar powered cooling/refrigeration and also water desalination techniques have been presented. In view of the fact that the Institute of Technology, Sligo has a well-established long history of research and development in the production of state of the art high-efficiency fast-response evacuated solar heat collectors it was decided to use this know how in the work described in this thesis. For this reason achieving high rates of evaporation was not a problem. It was, therefore, the question of the solar powered refrigeration that was envisaged to be used in the solar powered desalination to facilitate rapid condensation of the evaporated water that had to be addressed first.

The principles of various solar powered refrigeration techniques have also been reviewed. The first step in work on solar powered refrigeration was to successfully modify a

conventional refrigerator working on Platen-Munters design to be powered by high-output fast-response evacuated solar heat collectors. In this work, which was the first ever successful attempt in the field, temperatures as low as –19°C were achieved in the icebox.

A new approach in the use of photovoltaic technology to power a conventional domestic refrigerator was also attempted. This was done by modifying a conventional domestic refrigerator to be powered by photovoltaic panels in the most efficient way. In the system developed and successfully tested in this approach, the power demand has been reduced phenomenally and it is possible to achieve 48 hours of cooling power with exposure to just 7 hours of sunshine.

The successful development of the first ever multi-cycle intermittent solar powered icemaker is without doubt the most exciting breakthrough in the work described in this thesis. Output of 74.3kg of ice per module with total exposure area of 2.88 m^2, or 25.73kg per m^2, per day is a major improvement in comparison to about 5-6kg of ice per m^2 per day reported for all the single cycle intermittent systems. This system has then become the basis for the development of a new solar powered refrigeration system with even higher output, named the "composite" system described in this thesis.

Another major breakthrough associated with the works described in this thesis is the successful development and testing of the high-output water desalination system. This system that uses a combination of the high-output fast-response evacuated solar heat collectors and the multi-cycle icemaker. The system is capable of producing a maximum of 141 litres of distilled water per day per module which has an exposure area of 3.24m^2, or a production rate of 43.5 litres per m^2 per day. Once again when this result is compared to the reported daily output of 5 litres of desalinated water per m^2 per day the significance of this piece of work becomes apparent.

In the presentation of many of the components and systems described in this thesis CAD parametric solid modelling has been used instead of photographs to illustrate them more clearly.

The multi-cycle icemaker and the high-output desalination systems are the subject of two patent applications.

Contents

Chapter 4: Review of Solar Desalination Techniques

Chapter 5: Development of a Continuous Cycle Solar Thermal Refrigeration System

Chapter 6: A New Approach in Using Photovoltaic Power in Refrigeration

Chapter 10: General Discussion and Conclusion

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Appendices

Nomenclature

E_b	Emissive power or radiosty	Wm^{-2}
σ	Stefan-Boltzmann constant	5.67051E-8 $Wm^{-2}K^{-4}$
T	Temperature / Thickness	K or °C / mm
λ	Wave length	Å
R_s	Sun's radius	1.496E11 m
E_s	Radiosty of sun	Wm^{-2}
T_s	Surface temperature of sun	K or °C
$E_\perp$	Solar flux on a horizontal surface	Wm^{-2}
E_{sc}	Solar constant	1353 Wm^{-2}
τ	Transmission coefficient	
R_1, R_2	Radii of curvature	m
n	Index of fraction	
F	Focal length	m
C_{2D}	Concentration ratio in a two dimensional concentrator	
C_{3D}	Concentration ratio in a three dimensional concentrator	
θ_c	Acceptance angle	° or rad
Ra	Rayleigh number	
q	Heat flux per unit time	W
L	Characteristic dimension, the width of the channel	m
ρ	Density	kgm^{-3}

y	Volumetric expansion of a fluid	$m^3m^{-3}K^{-1}$
c_p	Specific heat of a fluid for a constant pressure	$Jkg^{-1}K^{-1}$
μ	Dynamic viscosity	$kgm^{-1}sec^{-1}$
k	Thermal coefficient of conductivity	$Wm^{-1}K^{-1}$
P	Pressure	Nm^{-2}
F	Liquid filled in a heatpipe	
Z	Total thermal resistance	KW^{-1}
COP	Coefficient of performance	
T	Temperature	K or °C
T_A	Absorber temperature	K or °C
T_E	Evaporator temperature	K or °C
M_d	Distillate product	kg sec^{-1}
M_b	Recirculation flow	$kgsec^{-1}$
h	Enthalpy	$kJkg^{-1}$

Chapter 1

Introduction

Chapter 1

Introduction

1.1 Background

The sun is the source of life and energy for all of the earth inhabitants. Most of the energy resources are direct or indirect results of solar energy. Chemical energy is formed, through the photosynthesis process, when the sun's rays falls on plants. This chemical energy helps in the growth and function of plants. Animals in turn derive their energy from plants. Without the sun, plants could not grow, and life would cease to exist. For hundreds of millions of years before man first trod the earth, plants and simple-celled organisms had flourished. After they had died and decayed, they gradually formed deposits of coal, oil, and natural gas [1]. Wind results when uneven solar heating of the atmosphere occurs. Wave, hydroelectric, OTEC wave and biogas are the resultants from solar energy. Solar energy also determines the temperature of the earth and its surrounding atmosphere (various climatic conditions in various spot of the planet).

Utilisation of solar radiation, as a source of energy is not new idea and can be referenced to ancient history with some numerous authentic evidences in the historical literature [2]. Perhaps one of the most impressive examples is the optical system devised by Archimedes. The system consisted of big concentrating mirrors, specifically designed, to repel the invading Roman fleet. In fact it is reported that he had written a book named "On Burning Mirrors" but unfortunately no copy remains today [3].

Mankind dependence on solar appliances has forced various inventors to devise systems, which are directly applicable in particular instances. Again here are numerous examples reported but highlighting them here is beyond the scope of this work.

It is worth pointing out that despite the ancient history of the appliances and systems, using solar and other renewable energy resources, they have not managed to make any significant impact in the world market up to now. Perhaps the most convincing argument to this is the fact that they did not meet the most basic requirements for a product (or system) to be commercially viable. High cost and lack of sufficient features to be used by great number of consumers are some of the most important factors leading to their present status.

The oil embargo in 1973-74 drew the world's attention to our heavy dependence on fossil fuels especially oil. In fact the questions associated with fossil fuels such as their limited reserves, environmental & political problems and the associated hazards with nuclear energy have given more urgency to more meaningful utilisation of solar and other renewable energy resources. The exploitation of solar energy is a genuine contribution in this respect, particularly "low temperature" applications. Its application is clean, environmentally friendly and it is an abundant and ceaseless energy source.

One of the central questions for the long term, as far as sensible solar appliances are concerned, is transition to renewable energy. At present the situation does not seem promising in terms of both quality and competitiveness. According to the very optimistic estimations renewable energies make up between 14% to 20% of the world total energy, comprising mainly hydropower [4].

To this effect research into commercially viable solar appliances has gained more urgency. The works described in this thesis is concerned with design development and evaluation of solar powered systems particularly solar cooling and solar desalination systems.

1.2 Basic Concepts of Solar Energy

Sun is a normal middle age star, formed by gravitational forces between its constituents and is kept from collapse by pressure of its released radiation. As shown in figure 1.1 the sun comprises four regions which are core, radiation zone, convection zone and photosphere.

Solar energy is electro-magnetic radiation produced as a result of thermonuclear fusion in the interior core of the sun, at an estimated temperature of 27 million°F. Nuclear fusion is a nuclear reaction in which nuclei are joined together to constitute heavier nuclei and make a heavier element, with the simultaneous release of energy. A simple description of a nuclear fusion reaction in the sun's core region was initially expressed as [5]:

$$4\,{}^{1}_{1}H \Rightarrow {}^{4}_{2}He + 2e^{+} + 2\nu + \gamma$$

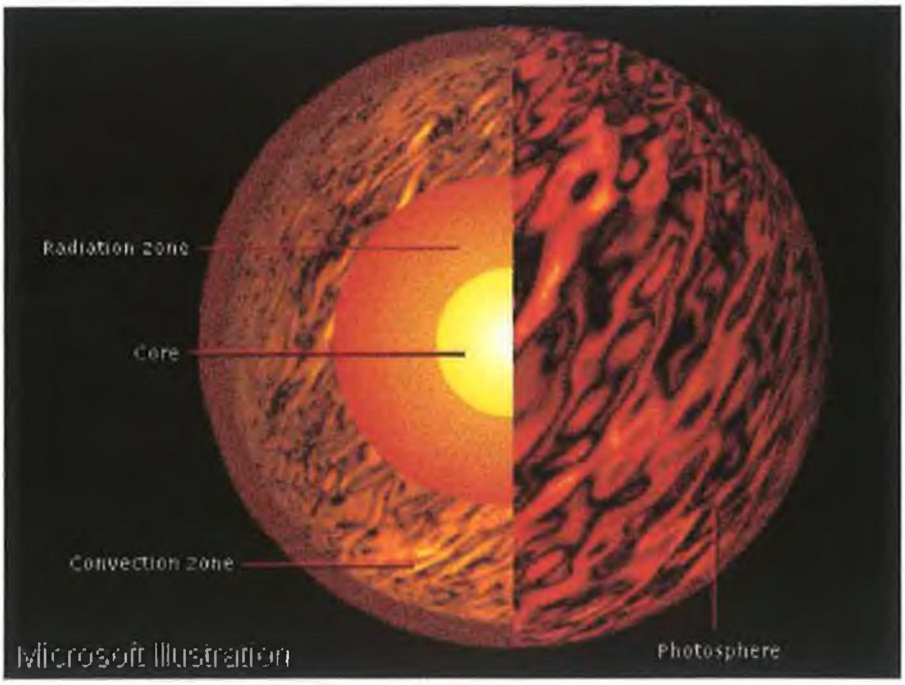
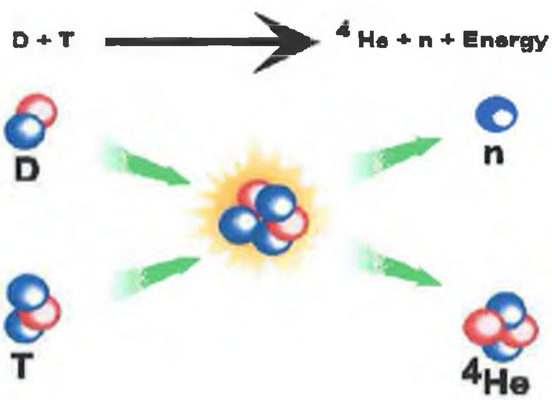


Fig. 1.1 Sun and its different regions: Core, Radiation zone, Convection zone, and Photosphere.

As shown in the above equation four hydrogen atoms combine to form one helium atom, two positrons and two neutrinos. The imbalance in the reaction arises from the release of energy in the form of gamma radiation. The released energy through this process is in the order of 26.72 MeV. Thus the sun is emitting electromagnetic thermal radiation.

However, the main fusion reaction involved occurs between the nuclei of the two hydrogen isotopes, namely, Deuterium and Tritium. The reaction between two hydrogen isotopes Deuterium (D) and Tritium (T) in the sun's core region can be depicted as follows:



1.2.1 Radiation Physics

Matter emits incoherent electromagnetic radiation, usually referred to as thermal or heat radiation. The radiated power depends on the temperature and material properties. The emissive power is given by Stefan-Boltzmann law:

$$E_b = \varepsilon .\ \sigma .\ T^4 \tag{1.1}$$

where E_b is the emissive power known as radiosty, ε is emissivity presenting the object properties, σ is the Stefan-Boltzmann constant equal to $5.67051s10^{-8}Wm^{-2}K^{-4}$ and T is absolute temperature. For a black body the value of ε is equal to unity.

According to Plank's law the monochromatic emissive power from a black body can be expressed as following:

$$E_{b\lambda} = \frac{C_1}{\lambda^5 (e^{C_2/\lambda.T_s} - 1)} \tag{1.2}$$

where C_1 is $3.742\times10^8 W.\mu m^4/m^2$ and C_2 is equal to $1.4387\times10^4 \mu m.K$.

By integration of $E_{b\lambda}$ over wave length, λ, the total radiation can be obtained:

$$E_{b\lambda} = \int_0^\infty \frac{C_1 \lambda^{-5} d\lambda}{(e^{C_2/\lambda.T_s} - 1)} \tag{1.3}$$

or:

$$E_{b\lambda} = Const \times T^4 \tag{1.4}$$

This shows that Plank's law is consistent with Stefan-Boltzmann law.

The energy flux received from the sun out of the earth's atmosphere is constant known as the extraterrestrial solar constant or simply the solar constant. According to the data confirmed by NASA (NASA standard data) the solar constant is equal to $1.3530\ kWm^{-2}$ [6]. Assuming the sun to be a spherical black body emitting homogeneously to space, the radiant flux can be expressed as follows:

$$\Phi_s = 4\pi R_s^2 . E_s \quad \text{or} \quad \Phi_s = 4\pi R_s^2 \sigma . T_s^4 \tag{1.5}$$

where R_s is the sun's radius equal to 6.96×10^8m, E_s is the radiosity of the sun and T_s is the temperature of photosphere.

Figure 1.2 shows the diagram of solar ray's reaching the earth. According to conservation law of energy:

$$4\pi R_S^{\,2}.E_S = 4\pi D_{ES}^{\,2}.E_{SC} \tag{1.6}$$

in which E_{Sc} is the solar constant, D_{ES} is the distance between earth and sun equal to 1.496×10^{11}m. The value of E_S and T_S can be calculated:

$$E_S = 63.2 \text{ MWm}^{-2} \quad \& \quad T_S = 5{,}777\text{K}$$

Therefore, the radiation coming from the sun is equivalent to that from a black body at a temperature of 5800K.

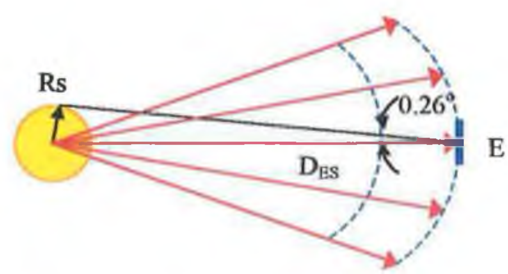


Fig. 1.2 Geometry of Sun-Earth

The spectrum of solar radiation on the earth can be categorised as follows:

1. Ultraviolet (λ< 3500Å)
2. Visible (3500Å<λ< 7000Å)
3. Infrared (λ>7000Å)

The ratios of total solar insolation in the earth surface are 2%, 51% and 47% for ultraviolet, visible and infrared, respectively [7]. Any hot object radiates energy in the form of electromagnetic wave.

Figure 1.3 shows the terrestrial and extraterrestrial solar energy flux at different wavelengths of light. The solar radiation curves peak at the wavelengths of yellow light in the visible section. Hence, the sun appears to be yellow. The area under the extraterrestrial solar radiation flux is equal to solar constant.

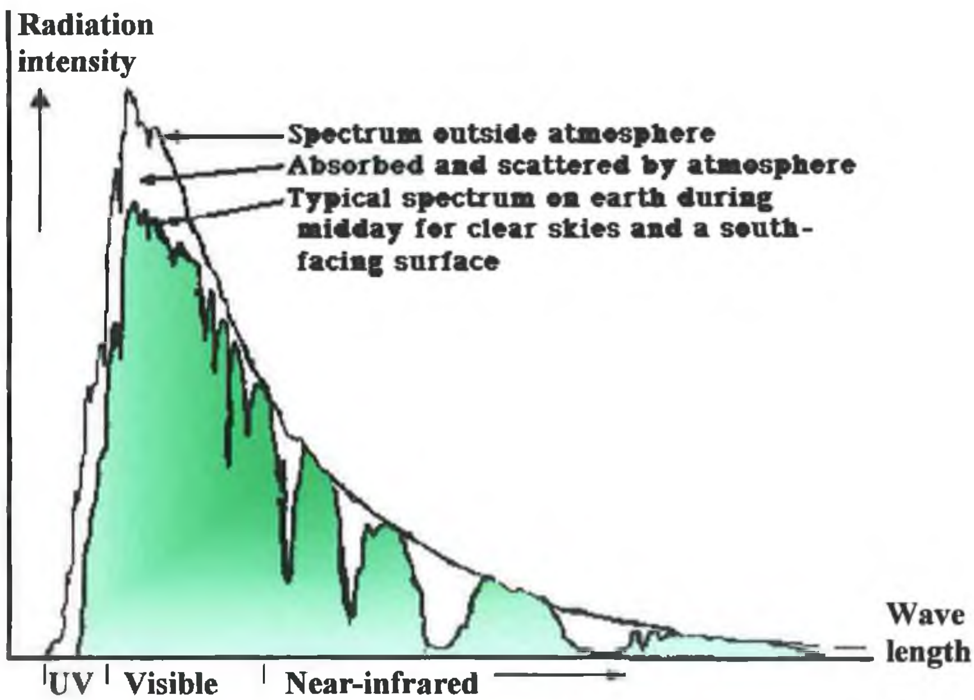


Fig. 1.3 Solar radiation at different wavelength of light

Solar radiation is received at the earth's surface after being subjected to the attenuation, reflection and scattering in earth's atmosphere. Solar radiation through its travel to earth is weakened by absorption. In good weather conditions, highest quantity of the sun's energy received at the surface of the earth is, at best, $1000W/m^2$.

Solar radiation in the atmosphere splits into two categories: direct (or beam) radiation and diffuse (or scattered) radiation. Direct radiation is the part of solar radiation that strikes a surface without change of direction in traversal of the atmosphere. Scattered radiation is the non-directional component of solar radiation induced by Rayleigh scattering, dust, cloud, aerosols and the content of water and CO_2 in the atmosphere. The global or total radiation is referred to the sum of beam and diffuse radiation.

A simple way to distinguish the beam and diffuse radiation is that the sun's image can be formed through concentrating the beam component but with the scattered component of the solar radiation cannot form an image of sun. The ratio of direct radiation to diffuse radiation varies with time and location. On an overcast day, the diffuse radiation exceeds the beam radiation. The amount of direct radiation diminishes as air pollution hikes. For a large city the ration of the direct radiation to diffuse radiation may only be on the order of 2 [8].

The attenuation of solar radiation passing through the atmosphere relies on various independent factors. The solar flux falling on a horizontal surface on earth can be expressed as following [9]:

$$E_{\perp} = \tau_{sc} \, . \, \tau_{Ra} \, . \, \tau_{O3} \, . \, \tau_{Ga} \, . \, \tau_{Wa} \, . \, \tau_{Ae} \, . \, \tau_{Ci} \, . \, E_{sc} \tag{1.7}$$

where $E_{\perp}$ is solar flux on a horizontal surface, E_{sc} is solar constant and τ_i represents the individual transmission coefficients. The τ-indexes are described as following:

- R_a: Rayleigh scattering by molecules of the air.

- O_3: is absorption by ozone.

- G_a: absorption by uniformly mixed gases.

- W_a: absorption by water vapour.

- A_e: extinction by aerosol particles.

- C_i: extinction by cirrus clouds.

Except cirrus scattering, scattering and absorption are strongly dependent on wavelength.

Figure 1.4 describes a breakdown of the absorptive and reflective losses due to the atmospheric attenuation.

1.2.2 Global Distribution of Solar Radiation

The distribution of solar energy incident on the earth relies on the geographical situation. The geographical weather alternations are resulted from the earth's atmosphere. Weather modifies geographical distribution of solar flux extensively. Variation in solar flux caused by clouds has a great influence on the performance of a solar system. If solar thermal collectors are assumed as linear transducers of the sun's energy then the average flux can be predicted from the average system performance.

The global annual isoflux contours of solar radiation is drawn in figure1.5. The regions having the highest solar flux lie in two latitude bands, at approximate 20-30^{o}N and 20-30^{o}S. There are very high insolation area's extending over Africa, The Middle East, India, Central South America, southern Mexico, USA and Australia. This global distribution is a function of both angle of incident and weather condition. Both of these independent variables

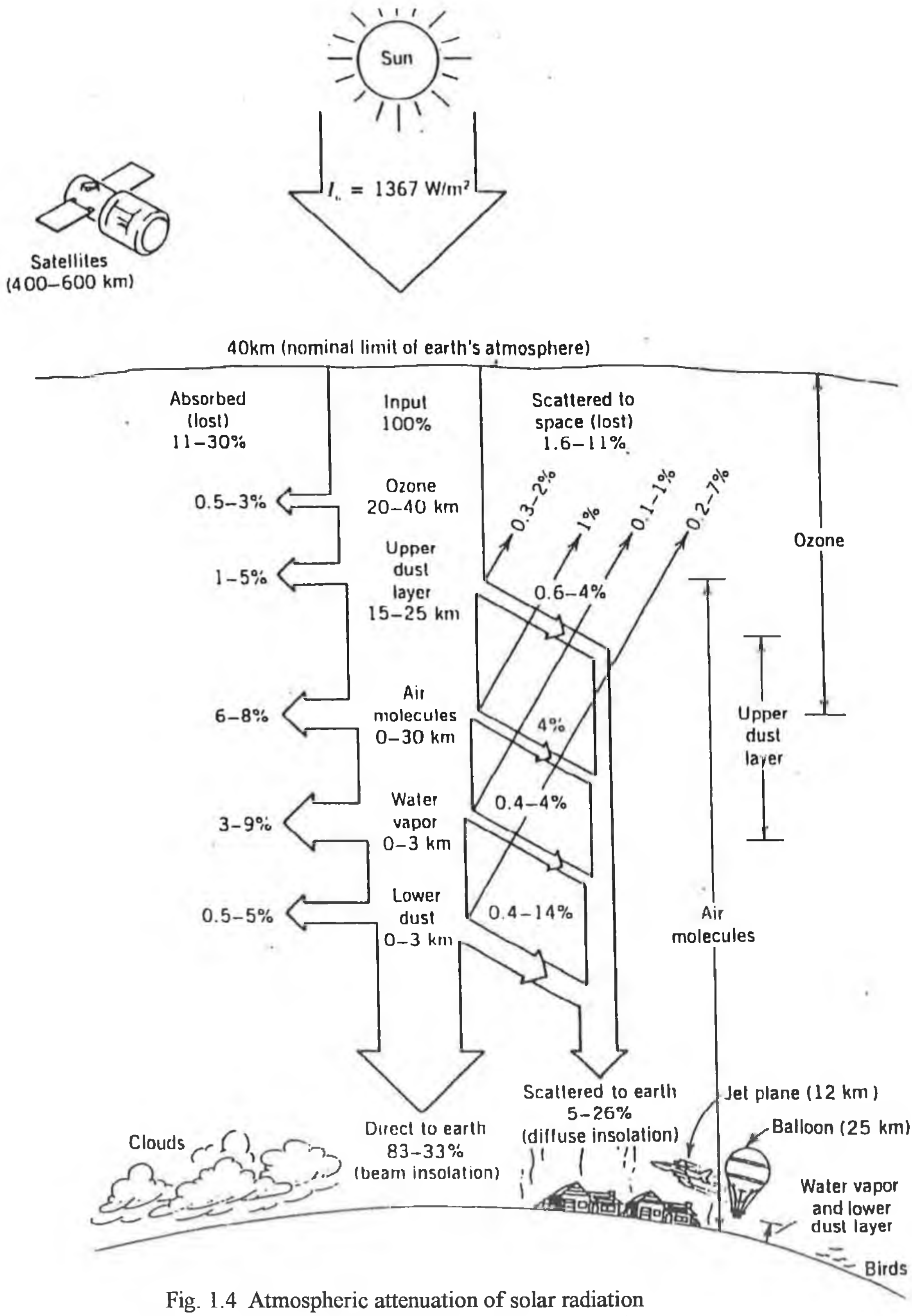


Fig. 1.4 Atmospheric attenuation of solar radiation (Reproduced from W.B. Stine and R. W. Harrington, 1985, Solar Energy fundamentals and design with computer application, New York, Wiley, p.84. Fig 4.9)

fluctuate during the year. Figures1.6.a.b.c.d show the global isoflux counter of four different months [10].

However, this never precludes the application of solar energy in regions with lower insolation. It is in fact in this context, that the design of better solar energy collection and distribution systems has begun to make utilisation of solar energy a feasible option.

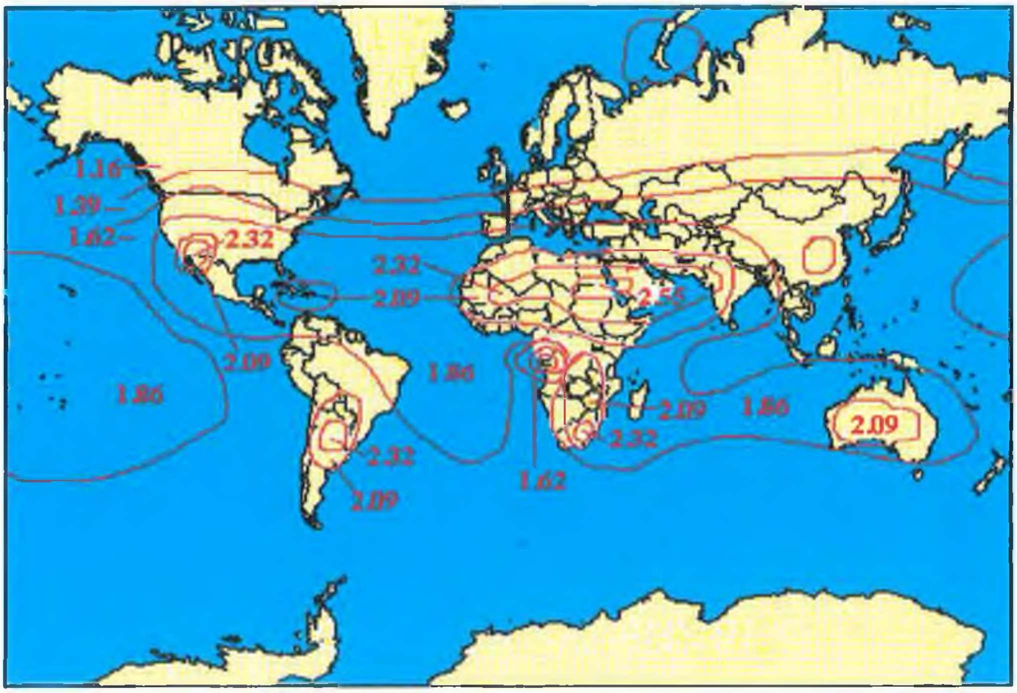


Fig. 1.5 Global distribution of the annular solar radiation, falling on a horizontal surface at the ground level - Measurements are in terawatt hours per km^2 per year. (Source: International Solar Collectors, Inc.)

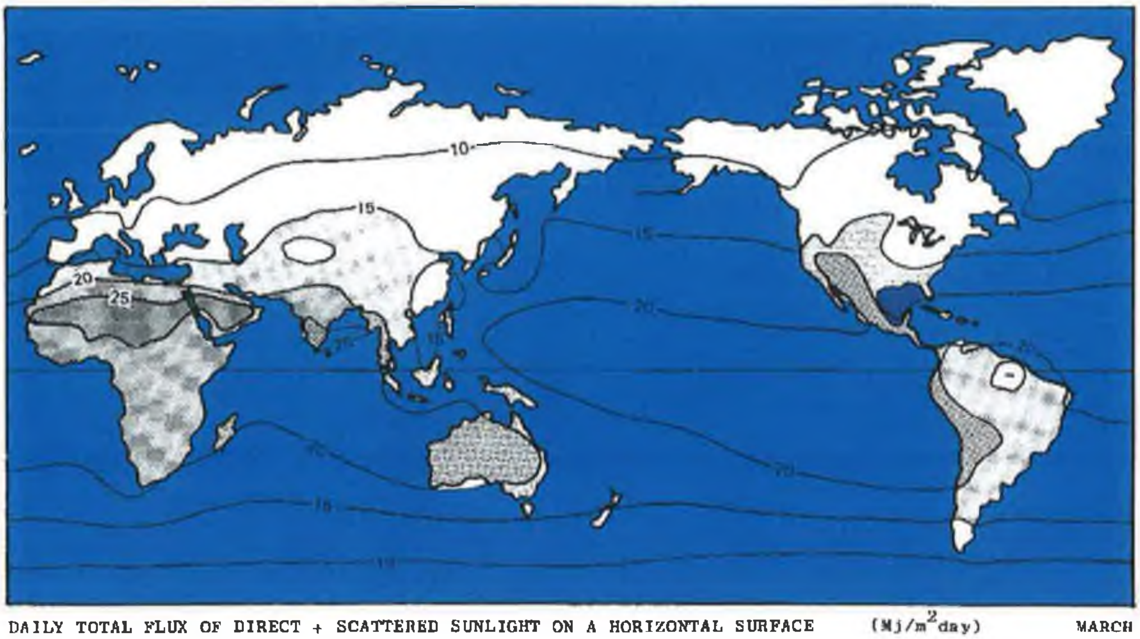


Fig. 1.6.a Global isoflux counter for March.

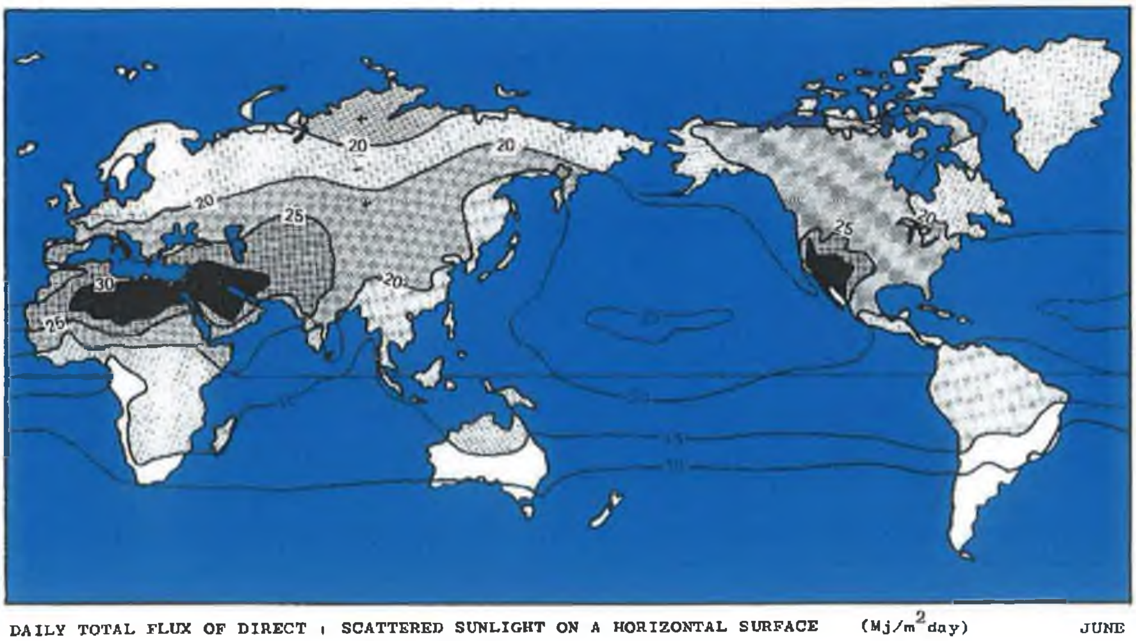
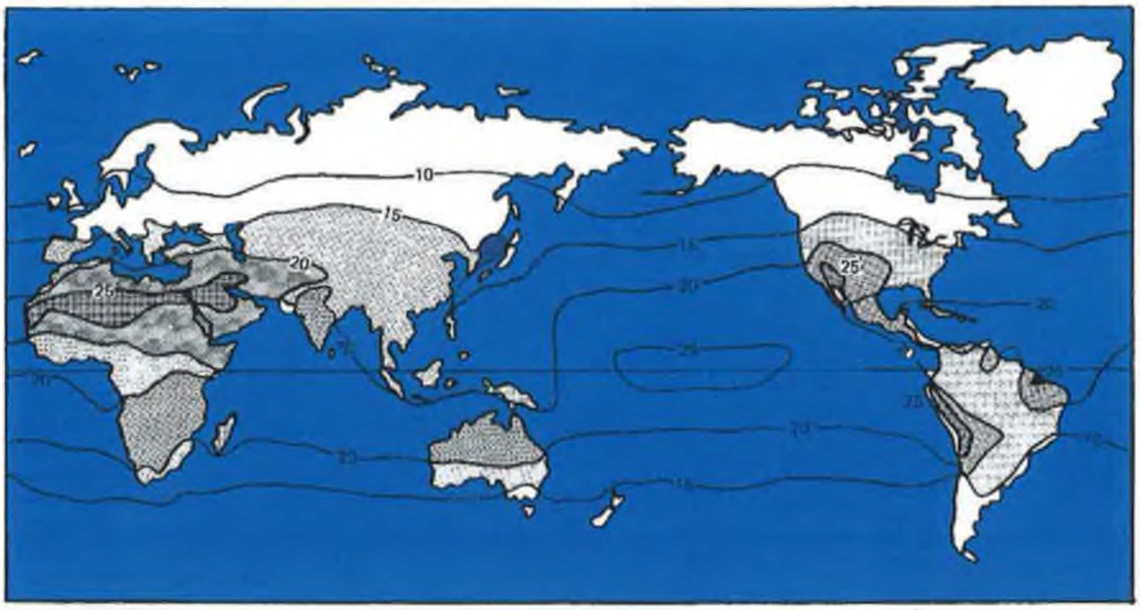
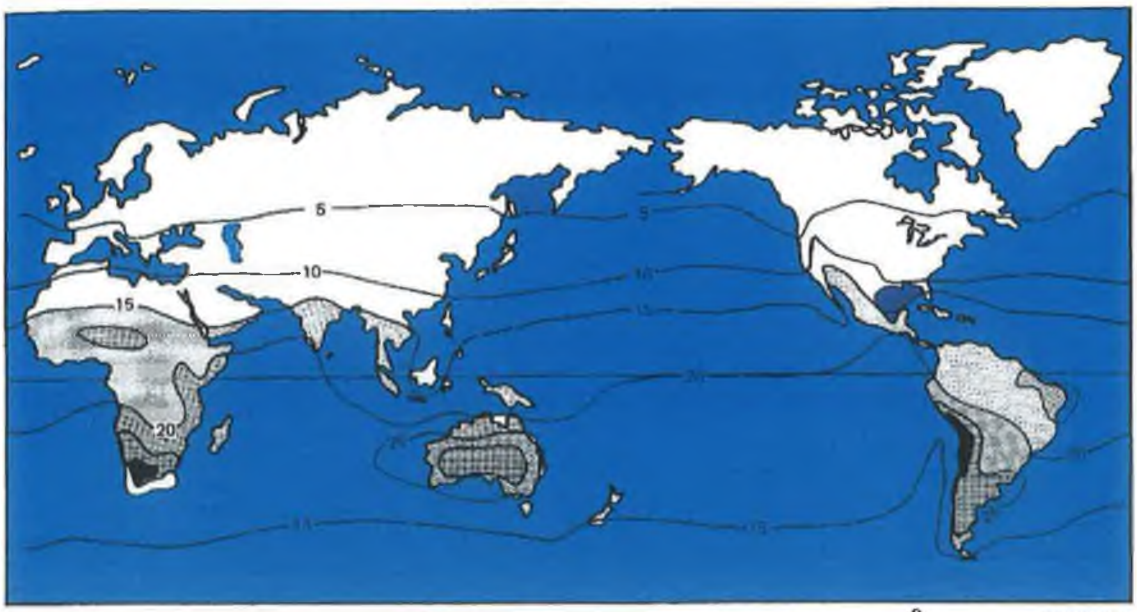


Fig. 1.6.b Global isoflux counter for June.



DAILY TOTAL FLUX OF DIRECT + SCATTERED SUNLIGHT ON A HORIZONTAL SURFACE (Mj/m^{2}day) SEPTEMBER

Fig. 1.6.c Global isoflux counter for September.



DAILY TOTAL FLUX OF DIRECT + SCATTERED SUNLIGHT ON A HORIZONTAL SURFACE (Mj/m^{2}day) DECEMBER

Fig. 1.6.d Global isoflux counter for December.

1.3 Solar Options and Constraints

Solar energy can be considered as an alternative, or even a substitute, for the conventional sources of energy in many cases. However, one must be convinced that there are acceptable reasons for turning from conventional to renewable energy resources such as solar energy. Hence persuasive answers must be found to the following questions:

1. Why change?
2. Why solar energy?
3. What are the advantages and disadvantages of solar energy?

There is some chilling statistic in relation to the increased reliance on fossil fuel for our energy production. The world energy consumption has been increased dramatically in 20th century. Over 50 years, fossil fuel consumption has increased fivefold, from approximately 57 exajoules in 1937 to around 282 exajoules in 1988 [11].

Fossil fuels are linked to severe environmental problems. Fossil fuels account for 90% of SO_2 and NO_x emission into atmosphere. Green house effect, global warming, acid rains and severe climate change especially during the recent decades are directly related to consumption of fossil fuels. Apart from the environmental problems, due to depleted oil reserves the oil era will come to an end very soon [12].

Replacement of fossil fuels for nuclear energy is considered not to be a straight forward and trouble free option. As the result of Chernobyl accident at least 9 million people have been affected in three Republics of former Soviet Union [13]. In total, over 160,0000 km^2 are contaminated in the three republics, Belarus, Ukraine and Russia. In Belarus 30% of country is contaminated with caesium 137 with 46,000km^2 exceeding 1curie/km^2 [14]. When a nuclear power plant is shut down, virtually all of its parts must be encapsulated for at least 22000 years. It is therefore worth exploring the solar and other renewable energy options with a much research as possible.

In contrast, solar energy offers some advantages. These advantages can be summarised as following:

1. It is abundant and ceaseless: It is estimated that the sun will stay in the main sequence more than five billion years radiating energy at about the same rate. The annual energy of the sun received by earth is in the order of 1×10^{18} kWh.
2. Environmentally sensitive and safe: Unlike nuclear energy which is faced with dramatic environmental problems such as nuclear accidents, or fossil fuels environmental problems; solar energy is clean, noiseless and safe.
3. Save resources: By using the renewable energy resources such as solar energy, the reliance on fossil fuels will eliminate.
4. It protects against inflationary fuel costs and possible fuel shortage.
5. Easy installation and virtually maintenance free: Installation of solar energy equipment is relatively easy, and since there are few mechanical or electrical moving parts, the maintenance expenses are relatively low.

However, despite the above-mentioned advantages, the application of solar energy is restricted due to a number of problems.

The initial argument against is that solar energy is expensive. High cost of the existing solar systems is the most important question to be addressed. Although the usage of solar heating panels are cost effective with an acceptable payback period but still there is long way to go on this issue. Applicability of solar assisted systems is restricted because of expenses for collectors and auxiliary systems such as storage, conversion and distribution systems. It is therefore, goes without saying that the only way to make solar option commercially viable is to come up with practical reliable solar powered systems and appliance affordable by a large portion of population.

Another argument against solar energy is the intermittent nature and fluctuations and the dilute nature of solar energy flux. The sun is just usable for few hours during a day. Also as pointed out earlier the solar energy flux on earth's surface varies due to geographical, climate and seasonal changes. Another limiting factor in this respect is low density of solar energy flux. The maximum solar energy intensity assumed for design of solar system is in the order of 1000KW/m^2. This requires the enlargement of the collection area or increasing the energy flux density with assistance of concentrating systems.

In addition to the above-mentioned factors, government legislation and policies have given advantages to fossil fuels. Many countries directly or indirectly subsidies fossil fuels and nuclear power. This includes tax write-off (e.g. UK oil sector's oilrig decommissioning costs are offset against tax), direct subsidies (e.g. German coal support through a levy), preferential R&D support, and pricing systems which encourage the status quo. The sums are not insubstantial, amounting to more than £9 billion for the UK nuclear industry between 1990-1998 and more than $40 billion per annum in the USA [15].

1.4 Conversion Methods of Solar Energy

Almost all the reported applications of solar energy are based on direct conversion of solar energy into:

(a) electricity, known as photovoltaic principle or PV, and

(b) heat, known as photothermal.

These topics are extensively researched and reported in literature [16,17]. They are only being briefly described.

1.4.1 Photovoltaic (or PV) Energy Conversion

In photovoltaic energy conversion the energy of the sun is directly converted into d.c electricity. This is done by using especially designed diodes, known as PV cells. Because this conversion is only possible when the diode is used in its so-called " reverse bias" mode operation they have very low efficiencies. The overall efficiencies of the widely used systems are less than 10-15% and those of most elaborate technology, high efficiency cells made from arsenide alloys, is 29.5% for a one-sun cell and 30.2% for a concentrator cell at 180 suns [18]. Their costs vary from US$4-7 per watt for the popular modules and more than US$500 per watt for the most advanced one [19].

Because of the cost constraints there is still a fair way to go before their application can break new grounds.

1.4.2 Photothermal Energy Conversion

In photothermal systems solar energy is directly converted to heat. The conversion is done using either concentrators or specially designed materials known as absorber or especially enclosure. The efficiency of these conversion systems is much higher than that of PV systems. They range from 25% to over 80%. The cost of photothermal systems is phenomenally lower than that of PV systems.

1.5 Broad Description of Solar Powered Applications

Both PV and photothermal systems have their own limited applications that are reported extensively in literature [20,17]. It must be pointed out that because of higher efficiencies and lower costs photothermal systems enjoy much broader applications compared to PV systems.

1.5.1 Application of PV conversion

Up to about 20 years ago the only noticeable application of PV systems were in space technology. However, since the late 1970s applications of PV systems broke new grounds in vaccine preservation, rural electrification, and telecommunication, etc [21]. It is interesting to note that despite the high cost PV systems are, indeed, a viable option in locations far away from main grid network [22].

1.5.2 Applications of Photothermal Conversion

The application of photothermal systems dates back to ancient times. They were mainly heating buildings and drying. Since mid 1970s the applications of photothermal systems become more wide spread. The first phase of this era was in generation of moderate temperature <80^{o}C water for general domestic and commercial needs. Several new applications reported in drying crops, cooking, cooling, and more exciting than all generation of electricity by generation of superheated steam [23,24,25,26].

1.6 Presentation

The main theme of the work presented in this study is concerned with potentially viable solar appliances from a commercial viewpoint. Solar powered refrigerators and desalination systems are two main applications that form the main part of the present work.

Since generation of thermal energy form the core of the energy supply in appliances described here a comprehensive review of solar heat collection system is presented in chapter 2. In view of the fact that the efficiency of solar heat collection system depends, very strongly, on the speed of removal of the collected heat from the collector to the "application point" heatpipes which are the most relevant mechanism of heat transfer (as far as the speed of the removal of heat from the collectors are concerned) are reviewed and their merits are also assessed in chapter 2.

In chapter 3 the principle of operation of the refrigeration systems and a review on various efforts and studies that has been accomplished to adopt refrigeration technologies with the solar energy are presented.

Chapter 4 is concerned on the solar desalination technologies. In this chapter a review on distillation processes, non-thermal desalination techniques and different solar powered desalination systems is presented.

Chapter 5 is concerned with the development, construction and evaluation of the performance of the new solar power refrigeration.

Chapter 6 is focused on modification of a household refrigerator, which operates based on vapour comparison cycle, to be adopted with solar energy.

Chapter 7 and 8 present the works that have been done on intermittent calcium chloride – ammonia adsorption refrigeration system. The refrigeration systems are block icemakers, which are capable of production of substantial amount of ice for small communities.

Chapter 9 describes the details of the design of the new solar powered desalination system. The constructional details and evaluation of the performance of the system are also presented.

Finally, chapter 10 contains final conclusions and discussions, and also, suggestions for future works.

Three scientific papers have been published as the result of the research works presented in this thesis. The titles of these papers are:

- A Continuous Cycle Solar Thermal Refrigeration System

- A Comprehensive Outlook on Solar powered Cooling Systems

- High-Output Solar Powered Desalination System

Appendix I includes a copy of each paper.

Chapter 2

Methods of Solar Heat Collection

Chapter 2

Methods of Solar Heat Collection

2.1 Introduction

The first step in application of solar energy is the collection and conversion of solar radiation. The two main tasks in the design of solar energy systems are:

1. Collect and convert as much as possible of solar radiation.
2. Transfer and distribute the generated energy (heat) with minimum energy losses.

For implementation of the above-mentioned tasks several factors can be considered to improve the performance of the solar collection system. First, because of the dilute nature of solar energy, it is desirable to increase the density of solar radiation. This can be accomplished with the assistance of the focusing or concentrating systems. The second issue is to minimise the energy losses from the solar collector, and consequently, attain higher temperatures and efficiencies. To achieve this target, a proper insulation is an important factor. Also with the application of vacuum technology in evacuated tube collectors the convection and conduction heat losses between absorber surface and glass cover is eliminated. The third category that helps to improve the efficiency of a solar energy collecting system is concerned to absorber surface. Absorber surfaces with specific optical selectivity have been developed to minimise the radiative losses. Such a design is often called "selective absorber surface". This terminology has been reported in several articles and reference [27] is only one among many. Selectivity in the broadest definition refers to use of separation of the input solar spectrum from the thermal infrared emitted by the collector and by the environment to emphasise the desired effect [28].

2.2 Focusing Lenses

Focusing lenses may be incorporated with solar heat collectors or PV modules to increase the intensity of the sunlight striking the collecting surface. The performance of these systems relies on of the refraction of light at the interface between the air and medium of the lens. Although by application of focusing lenses, the solar energy intensity on the absorber surface can be increased, but they can only utilise the direct component of the total radiation. The principal of concentration in a lens is demonstrated in figure 2.1.

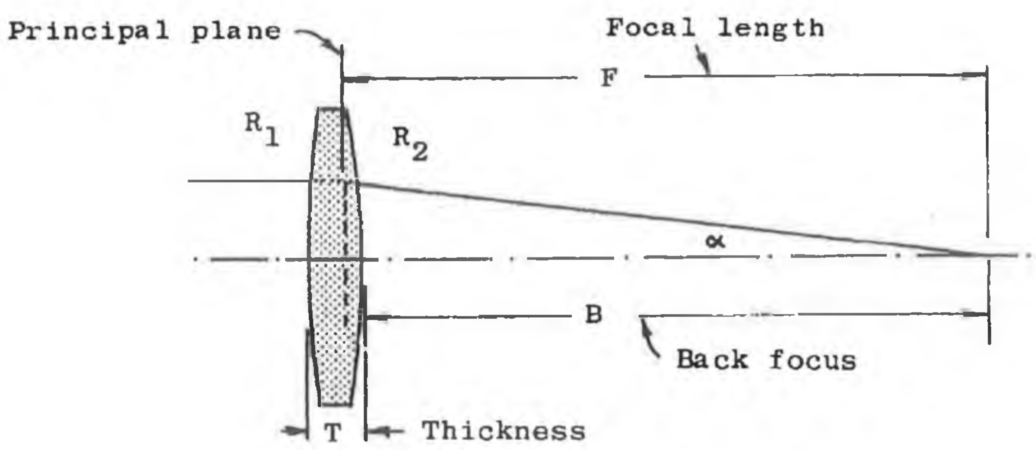


Fig. 2.1 Schematic diagram of a lens

For a thin lens, in which the thickness of the lens is small in comparison to the focal length of the system, the focal length, F, can be calculated as follows:

$$\frac{1}{F} = (n - 1)\left(\frac{1}{R_1} - \frac{1}{R_2}\right) \tag{2.1}$$

where R_1 and R_2 are radios of curvature for the front and rear surfaces and n is index of refraction of material from which the lens is made.

For a thick lens in which the thickness of lens is considerable in comparison to the focal length of the optical system, the equation of focal length can be expressed as follows:

$$\frac{1}{F} = (n - 1)\left(\frac{1}{R_1} - \frac{1}{R_2}\right) + \left(\frac{n-1}{n}\right)\left(\frac{T}{R_1 R_2}\right) \tag{2.2}$$

where T is the thickness of the lens.

It must be noted that the above equations refer to the focal length of the paraxial rays.

Ordinary lenses are rarely used because of their weight and high cost. Thus, special designs of focusing lenses are suggested [29].

One of the interesting designs of focusing lenses, especially for PV systems, is the Frensel lens, which features a miniatures saw-tooth design, as shown figure 2.2 [30]. When the teeth run are straight rows, the lenses act as line focusing concentrators; and when the teeth are arranged in concentric circles, light is focused at central point.

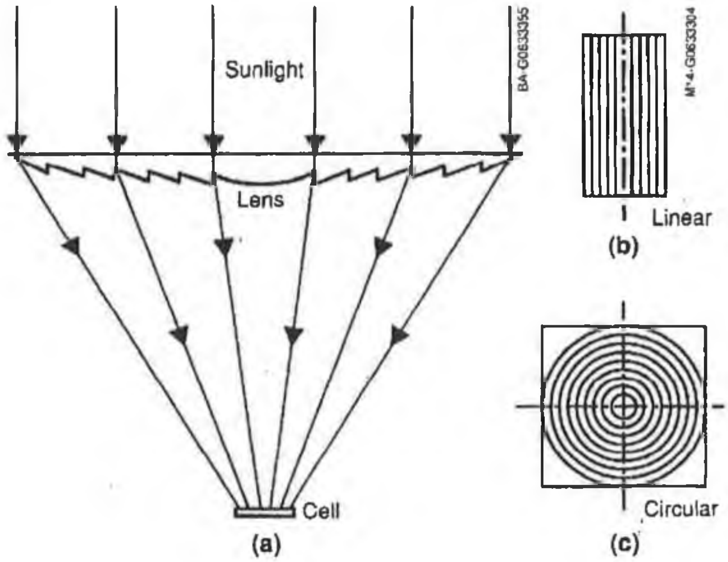


Fig. 2.2 Schematic diagram of a Frensel lens
(Reproduced from Photovoltaic Fundamentals, Cook, G. et. al., 1990, US Department of Energy, p53)

Another design of focusing lenses is the cylindrical lens [31]. These systems are appropriate for solar energy collection systems as their line focus can be matched to an absorber pipe. Figure 2.3 shows the schematic of a cylindrical lens.

A special type of cylindrical lens is the water-filled pipe collector. As shown in figure 2.4 the incoming radiation at an incident angle i_1, is refracted at an angle u_1, and after the second refraction the ray strikes the optical axis.

2.3 Concentrating Mirrors

In addition to focusing lenses, concentrating mirrors can be utilised to increase the intensity of solar radiation. Perhaps one of the simplest designs for increasing the solar energy intensity is the usage of the auxiliary side mirrors, known as booster mirrors. This optical system, booster mirrors, is usually incorporated with the flat plate collectors to

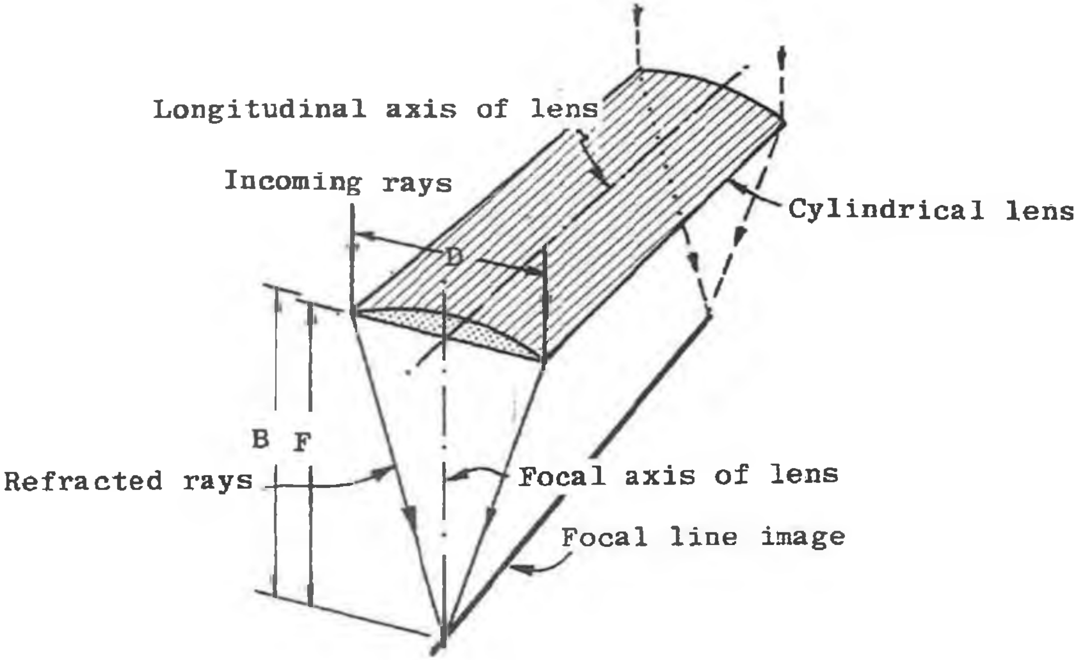


Fig. 2.3 Schematic of a cylindrical lens

The Optics of Solar Collection

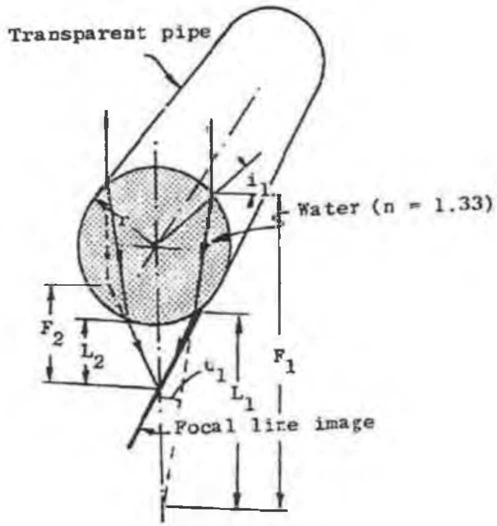


Fig. 2.4 A water-filled type cylindrical lens

achieve higher generating temperatures. Also application of booster mirror in PV panels has been reported [32]. Although booster mirrors just softly increase the solar intensity, but due to their geometry, they can just softly increase the intensity of radiation. To obtain higher intensities of solar radiation over the collection area, or in other word to increase the concentration ratio, different profiles of concentrating mirrors have been developed by inventors. This section is concerned on different types of concentrating mirrors and their practical working.

2.3.1 Parabolic Concentrators

Figure 2.5 shows a line imaging parabolic concentrator, and is so called because an image of sun is formed at its focus. The collector consists of a cylindrical parabolic mirror segment, which focus the radiation onto absorber tube placed on the focal line of the concentrator.

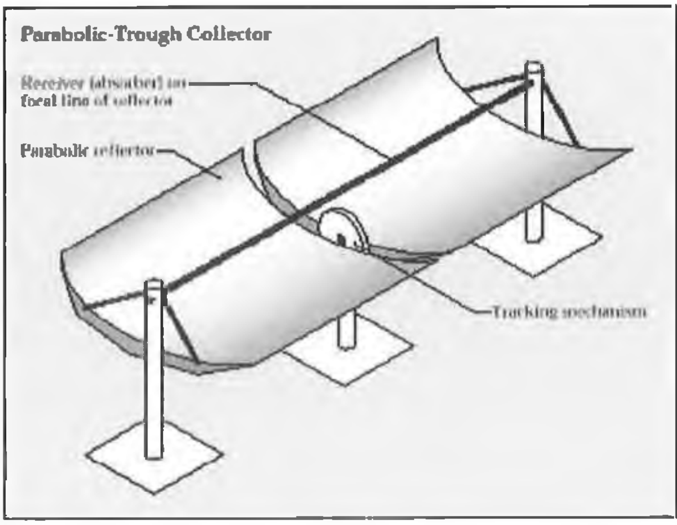


Fig.2.5 A parabolic trough collector

There are various types of absorber tubes, which can be incorporated with the concentrator, including [33]:

❶ black absorber tube with glass sheathing

❷ transparent tube with black absorber liquid

❸ transparent tube with absorber star arrangement

In the transparent tube with absorber star arrangement, through the star arrangement of the absorber the absorption coefficient is improved by multiple reflection. If the absorber has an absorption coefficient (α) of 0.8, an absorption coefficient of 0.96 and 0.99 would be achieved after double and triple reflection, respectively. As shown in figure 2.5 the concentrator can be used with a tracking system to achieve higher efficiencies and temperatures. Tracking systems can be either single axis- trackers or dual-axis trackers. Although a concentrator with a tracking system can achieve higher temperature, but it is more expensive and requires regular maintenance and a parasite power to operate.

Perhaps the most important application of parabolic concentrators is in electrical power generation, known as line focus or distributed collector power generation system. In a typical line focus power plant the reflective surface of a parabolic trough concentrates sunlight onto a receiver tube located along the trough's focal line and warms up a working fluid which is circulated through the absorber pipe. The troughs are normally designed to track the sun. Concentration ratios of parabolic troughs range from 10 to 100, and temperature range up to 400°C. In Southern California, nine operational plants of line focus systems, initially developed by LUZ Ltd, are producing a total capacity of 354MW to the utility grid [34]. These solar electric generating systems use thermal oil as a heat transfer fluid. Oil is pumped through a series of conventional heat exchangers which generate superheated steam of 390°C to derive a turbine.

2.3.2 Compound Parabolic Concentrators

The compound parabolic concentrator is a non-imaging concentrator sometimes known as Winston cusps. The CPC consists of two symmetrical parabolic reflectors, which funnel the solar radiation from the aperture to the absorber. The two parabolas must satisfy two conditions: i- the focus of one half-parabolic profile must be placed on the other, and ii- the profile must be tilted inwards until their tops are parallel to the cusp axis. Figure 2.7.a shows the cross sections of a symmetrical two-dimensional nontruncated CPC.

The maximum concentration ration in a two dimensional CPC is expressed as follows:

$$C_{2D} = \frac{1}{\sin\theta_c} \qquad (2.3.a)$$

where θ_c is the acceptance angle.

Although the concentration ratio of a CPC is relatively high, but in contrast to a simple parabola a CPC is very deep, and therefore it requires a large reflector area. This disadvantage can be offset to some extent. At the upper points of the parabolas in a CPC, the concentrator profile is parallel to the concentrator central axis. Thus, the upper ends of the reflectors contribute little to the radiation reaching the absorber, and can be truncated, as shown in figure 2.7.b.

The truncation reduces the reflector area and at the same time it also allows the some rays with larger angle of incidence to reach the absorber. This means the acceptance angle of the concentrator is increased, and consequently, the concentration ratio is brought down. Therefore, limited truncation affects the acceptance angle a little, but it does change the concentration ratio and the average number of reflections undergone by radiation before it

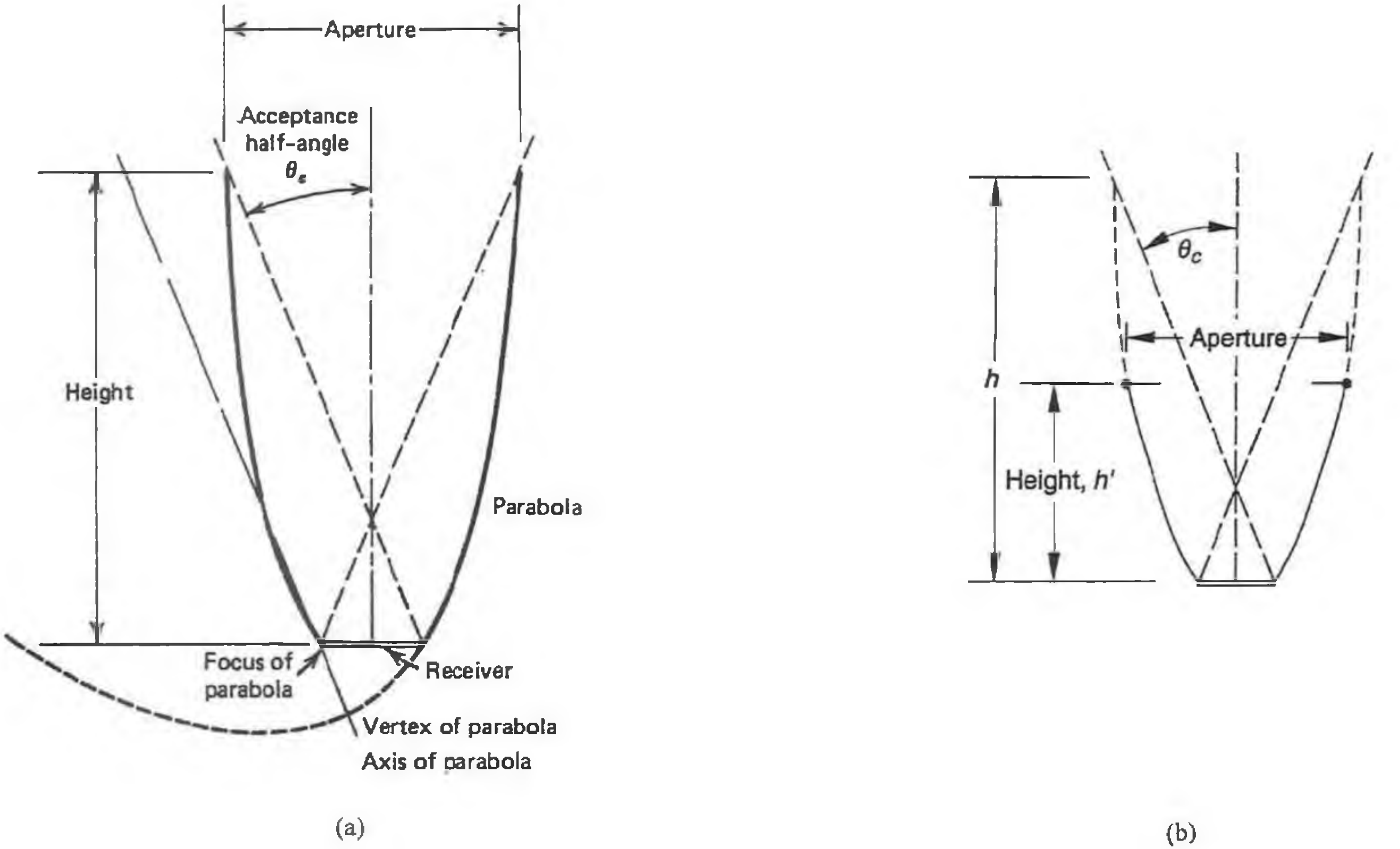


Fig. 2.7 Compound Parabolic Collector : (a) nontruncated CPC, (b) truncated CPC

reaches the absorber surface. The effect of truncation has been extensively studied [35].

To achieve higher concentration ratios, a CPC can be used as a second stage concentrator. In this design, the reflection from the first stage which is a simple parabola functions as the source for the second stage or CPC [36]. Further investigation by Rabel and Winston showed that a elliptical profile, known as compound elliptic concentrator or simply CEC, achieves maximal concentration when used as a second stage concentrator for a flat absorber [37]. Figure 2.8 illustrates the application of a CEC as a second stage concentrator.

Also a three dimensional of the CPC can be designed by rotating the profile about its axis of symmetry. In this case the concentration ratio is:

$$C_{3D} = \frac{1}{Sin^2 \theta_c} \tag{2.3.b}$$

2.3.3 Concentrators with Involute

Trombe (1957) introduced an optical system, which consisted of a cusp mirror with a special profile and an absorber pipe. Later in 1972 Meilnel independently discovered the same properties of such an optical solar collection system. The profile of the curve is the locus of a string unwrapped from about a circular receiver. The resulted profile is depicted in figure 2.9. The instantaneous profile radius of the curvature is the length of the unwrapped portion of the string with the centre of the curvature at its point of tangency with the circle, which is also the local normal of the cusp. The equation of the resulted profile, involute, can be described as follows:

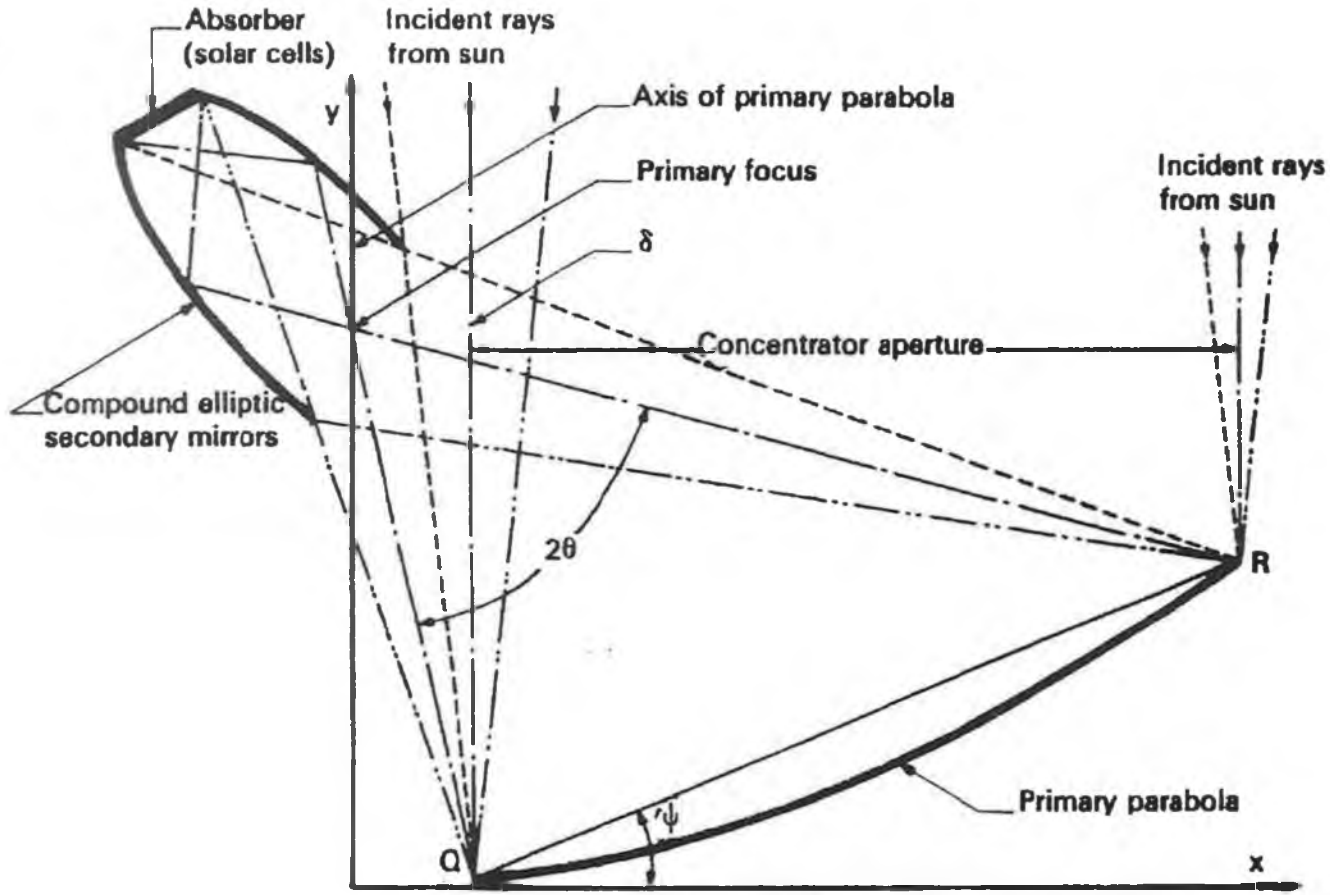


Fig. 2.8 Off-axis parabolic trough compound elliptic concentrator

$$x = a(\cos\phi + \phi\sin\phi) \quad \& \quad y = a(\sin\phi - \phi\cos\phi) \tag{2.4}$$

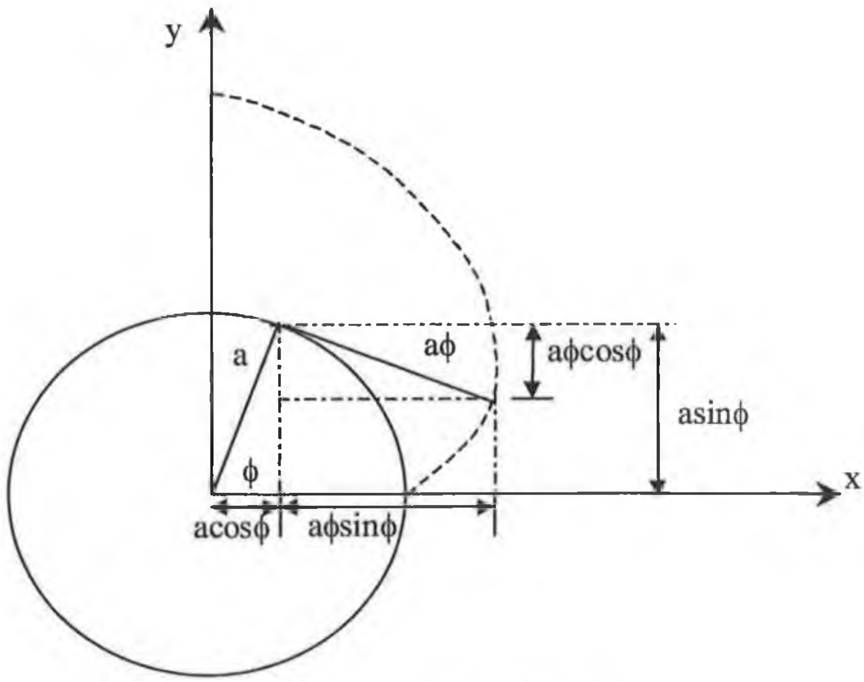


Fig.2.9 Governing equations of the involute

Figure 2.10 demonstrates the schematic of a Trombe-Meinel cusp. The absorber geometry coincides with the circle. The acceptance angle in this case would be equal to π. Thus, the collector functions like a flat plate collector. The concentration ratio, ratio of aperture area to receiver surface area, is unity, and therefore, two dimensional Trombe-Meinel cusp is not really a concentrator. Still it has the advantage of minimising the heat loss surface of the receiver. In this case, πd is the total perimeter of the receiver, whereas, for a flat plate collector of the same aperture, the effective receiving portion would be only the top of the heat absorbing surface, leaving the entire bottom surface open for heat losses.

It is possible to improve the flux concentration. Meinel suggested that the cusp could be expanded with the addition of a circular or parabolic tangent curve [38]. Also the profile can be modified, so that the local normal remains a position just sufficient to reflect the incident ray tangent to the receiver surface. This design with a manual tracking system was used by Menon [39].

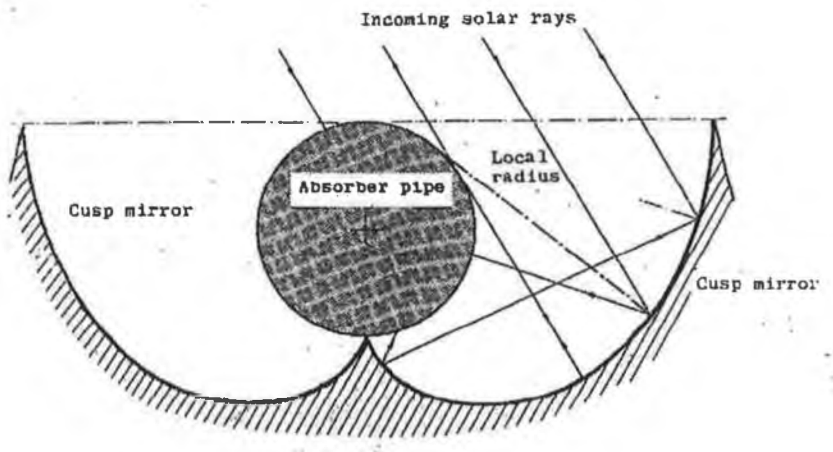


Fig. 2.10 Schematic of an involute cusp

In contrast with CPC concentrators, these collectors have a lower concentration ratio. CPC can offer reasonably a good concentration ratio with flat receivers, and in this case the involute cusp is not a suitable choice. However, Involute cusp is competitive to CPC in the applications involving tubular absorber. Also, involute cusp like CPCs (or CECs) can be used as a second stage concentrator. This design has been studied and used in a solar furnace by Inayatullah and Menon [39,40].

2.4 Flat Plate Collectors

Flat plate collectors are the most extensively used solar heat collectors. These collectors are inherently simple and relatively cheap, and therefore, they are extensively used. These collectors are generally designed for applications with moderate generating temperatures, up to 100°C above ambient temperature [41]. Flat plate collectors utilise both diffuse and direct radiations, therefore they can function in cloudy or hazy conditions, however, they might not be able to generate a desired temperature in these conditions. The applications that flat plate collectors can be used are in provision of hot water, space heating, air conditioning and industrial process heating. The main drawbacks of these collectors are their low efficiencies and low generating temperatures.

2.4.1 Basic Principles of Operation of the Flat Plate Solar Collectors

A basic flat plate collector is just a flat metal sheet appropriately blackened that could lose significant amounts of heat to ambient, particularly in the presence of wind. As shown in figure 2.11, a typical flat plate collector comprises of the following parts:

1- *Transparent cover*: This is a plane glass sheet that provides the thermal protection for the absorber and also keeps rain and dirt away from the blackened surface. If water enters into the space between the collector and the cover glass it evaporates and subsequently condenses on the under side of the glass sheet, and consequently lowers the collector efficiency. The covers must be also air-proof to minimise the convection losses. A collector might not have any cover (bare collector), or it might be

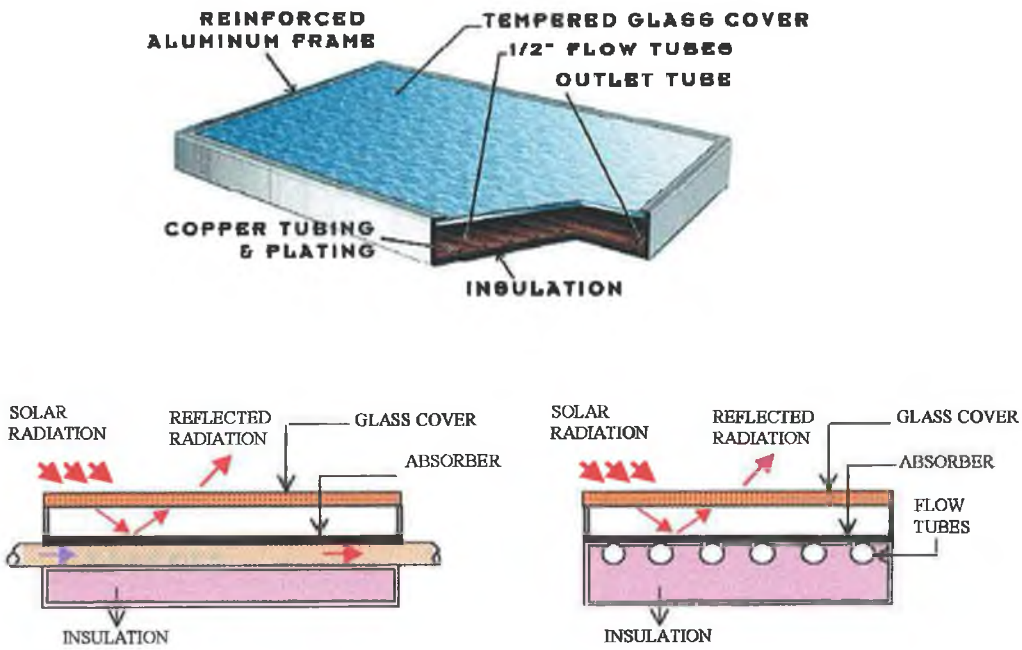


Fig.2.11 Configuration of a typical flat plate collector with its longitudinal and cross sections.

incorporated with one or more transparent covers. The number of transparent covers depends on many factors, but the main one is the temperature deference between the absorber surface and ambient temperature. For low temperature differences, $\Delta T < 80^{o}C$, fewer numbers of glass plate (single- glazing) are suggested. For higher temperature differences, $\Delta T > 80^{o}C$, larger numbers of transparent covers (double-glazing) are suggested. However, the augmented number of covers increases the prices of collector and reduces the quantity of incoming solar radiation to the absorber.

2- *Absorber plate*: A metal blackened surface, which collects the solar energy and subsequently converts it to thermal energy. The absorber plate material can be copper, aluminium or galvanised iron. Special absorber surfaces, known as selective surface, might be used to increase the absorber efficiency and temperature. There are a number of design principles and physical mechanisms in order to create selective solar absorbing surface. Some of these patterns and their properties are described as follows [42]:

❶ *Semiconductor-metal* tandems can provide the desired spectral selectivity in which short wave length radiation is absorbed in a semiconductor with the bandgap around 0.6eV, and because of underlying metal it has low thermal emittance.

❷ *Multilayer absorbers* can be adopted to become efficient selective absorbers. An example of these absorbing surfaces is $Al_2O_3/Mo/Al_2O_3$, which was first used in space technology.

❸ *Metal- dielectric composite coating* is made up of metal particles embedded in a dielectric host. This design takes the advantage of flexibility of solar selectivity, regarding to the constituents, thickness of coating, proportion of constituents, orientation of particles and size of the particles.

The Textured surfaces can offer a high solar absorbance by multi-reflection. The most popular examples of texture surfaces are dendritic tungsten, texture copper, nickel, and stainless steel. The texture surface design is also used in the photovoltaic cells to minimise reflectance losses. Figure 2.12 shows six different coating and surface finishes for selective surfaces.

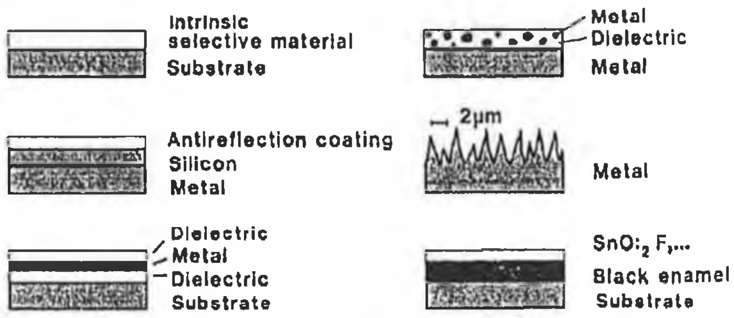


Fig. 2.12 Schematic diagram of six different coating and surface treatment for selective

3- *Flow tubes*: The generated heat in the absorber surface is conducted via the flow tubes to working fluid. The working fluid is a liquid, commonly water that flows throughout the flow tubes. In the case that working fluid is a gas, commonly air, flow tubes are replaced with gas conduits (Figure 2.13).

There are number of variant designs that can be used for flow tubes [43]. Figure 2.14 shows two different designs of flow tubes. In one configuration, flow tubes are routed in parallel, using inlet and outlet headers, and another design is the serpentine pattern. Also heatpipe is utilised to transfer the generated heat from absorber surface [44].

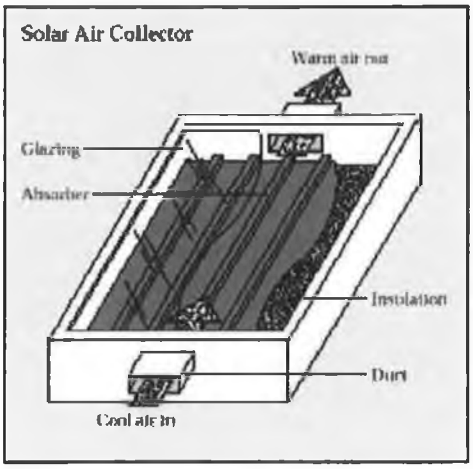


Fig. 2.13 A flat plate air collector

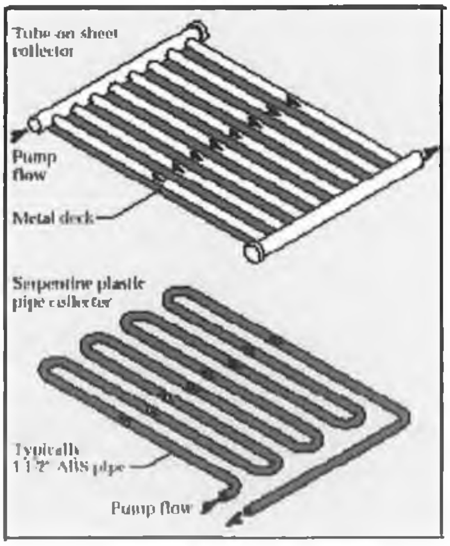


Fig. 2.14 Different configuration of flow tubes

4- *Insulation material*: The insulation reduces the conduction heat loss from the bottom of the absorber plate. The insulation materials could be glass, glass wool, or foams. A minimum 4" conventional insulation material provides enough insulation [45].

5- *Enclosure*: The Enclosure contains the absorber, the flow pipes and the insulation materials.

2.4.2 Flat Plate Collectors with Convection Suppresser

The performance of a collector can be improved by reducing the convective and radiative losses. As pointed out earlier radiative losses may be reduced by application of selective absorber surfaces and selective windows. There are also several methods which can be implemented to suppress the convection losses.

Convection is buoyancy effect that moves the heated air upward. If the heated surface faces down, the air layers are stable up against the heated surface, and therefore, the heat loss is just based on conduction. Based on this theory, the collector could be faced downward and the solar radiation can be beamed on to the collector surface using a mirror. Although this method eliminates the convection heat losses, it is restricted as a result of the reflective losses of the mirror. Therefore, this procedure is suitable when a concentrating mirror is already being used and the reflective losses are relatively small compared to the convective losses.

Also the convection heat loss depends on the shape of the heated surface and the tilt of the collector. The largest convection heat loss is from horizontal position. The convection heat losses in vertical position is 0.8 of that of the horizontal position.

Application of a second of transparent cover might assist to reduce convection losses between the absorber and outer glass cover. Arkon Company, in Denmark, marketed a flat plate collector, which used a double walled transparent cover of polycarbonated [46].

A honeycomb structure, as shown in figure 2.15, can be utilised to suppress the convection and radiative losses [47]. In a small vertical channel the convection does not begin until a certain finite temperature difference between the top and bottom of the channel. The convection occurs when a the Rayleigh number exceeds a minimum value, where the Rayleigh number can be defined as follows:

$$Ra = qL^3\rho^2 g y c_p/\mu k^2 x \tag{2.5}$$

where:

- q is heat flux per unit time (W)

- L is characteristic dimension, the width of the channel (m)

- ρ is fluid density (kgm^{-3})

- y volumetric expansion of the fluid ($m^3m^{-3}K^{-1}$)

- c_p specific heat of the fluid at constant pressure ($Jkg^{-1}K^{-1}$)

- μ dynamic viscosity (kgm^{-1} sec^{-1})

- k thermal coefficient of conductivity ($Wm^{-1}K^{-1}$)

- x length of the channel (m)

The suppression of convection would be effective only for small temperature differences between the bottom and top of the honeycomb. If the absorber temperature rose beyond a certain limit, then the value of the q would cause the Ra value to exceed the limit and convection would begin. The variation of convection heat loss for a honeycomb structure is shown in figure 2.16.

Also honeycomb structure reduces the radiative heat losses.

2.5 Evacuated Tube Collectors

The performance of a solar collector can be further improved if the air in the gap between the absorber and glass cover is evacuated. This would eliminate the convection losses between the absorber and transparent cover. Due to atmospheric pressure and the technical problem related to sealing, construction of an evacuated flat plate collector is not possible. The main problem is the fact that the flat glass sheet and other flat walls of the collector will cave in as a result of the atmospheric pressure outside the collector. This problem can be solved by using the vacuum tubes, since a glass formed as a tube has much higher compression strength than one or two glass plates supported against another one [48]. Although evacuated tube collectors can achieve higher generating temperatures and efficiencies compared to flat plate collectors, but these kind of collectors due to application of vacuum technology are more expensive and hermetically more complex than flat plate collectors.

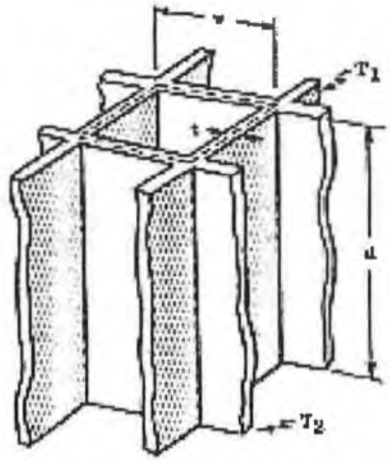


Fig. 2.15 The honeycomb structure

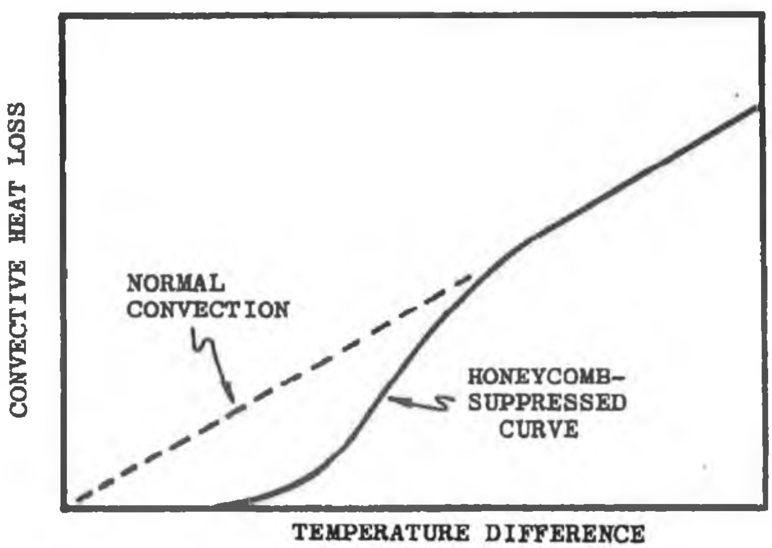


Fig. 2.16 Response of the effect of the convection suppression for honeycomb structure with temperature difference

2.5.1 Thermal Conductivity Through an Evacuated Glass Tube

The heat loss due to molecular conductivity in a collector is independent of the internal pressure over a large range, and is indicated by following equation [49]:

$$Q_C = 4.186.\lambda.(dT/dx) \tag{2.6}$$

where:

- λ is conductivity of the gas

- dT/dx is temperature gradient per unit of length

When the molecular mean free path is greater than collector dimensions, the heat loss decreases linearly with decrease of pressure, as following:

$$Qc = \frac{4}{3} . \frac{\alpha}{(2-\alpha)} . \Omega_{\mu} . P . \left(\frac{273.2}{T_o}\right)^{1/2} . (T_1 - T_o) \tag{2.7}$$

where:

- T_1 is the temperature of the hot surface (absorber) (K)

- T_o is the temperature of the cold surface (glass tube) (K)

- P is the internal pressure (mbar)

- Ω_{μ} is free molecule heat conductivity at 0°C (W.m^{-2}.mbar^{-1}.K^{-1})

- α is the accommodation coefficient of the hot surface

The percentage of heat loss due to molecular conductivity against the internal pressure is shown in figure 2.17. As shown in the graph, for the internal pressure less than 10^{-2} torr[T] (1.33224×10^{-2}mbar) the molecular thermal conductivity, drops dramatically.

[T] International standard term which replaces the old term of millimeters of mercury; (1mmHg = 1torr = 1.33224mbar).

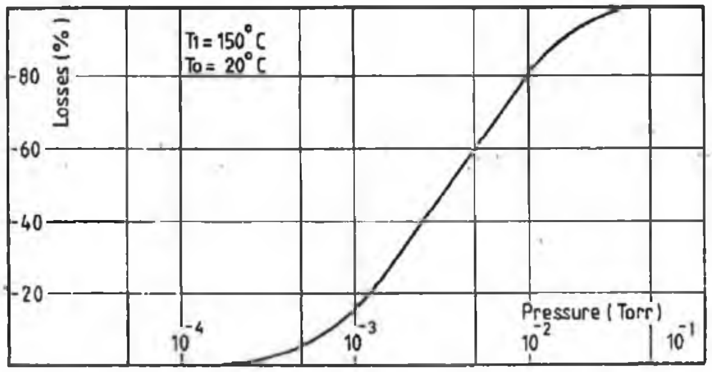


Fig. 2.17 Heat losses due to molecular conductivity versus internal pressure

As a result of the discussion presented above an evacuated heat collectors must be evacuated to levels of less than 10^{-4} torr (1.33224×10^{-4}mbar), if elimination of conduction and convection losses is to remain in force.

It is important that the pressure inside the evacuated tube will rise as a result of permeation of gases, helium in particular, and also as a result of outgassing [50]. The outgasing from the internal components is dominated by the selective surface. A study of gas evolution from selective surfaces at about 280°C revealed that the bulk of outgassed species are H_2O, CO, H_2 and N_2. These gases can be absorbed by a suitable getter to maintain the pressure inside the collector under 10^{-4} torr (1.33224×10^{-4}mbar).

2.5.2 Basic Evacuated Collectors

The first evacuated tube collector was built in 1961 by Speyer, which consisted of a strip of a flat plate absorber containing incoming and outgoing pipes enclosed in an evacuated tube [51]. The original Speyer design used a glass tube that had the lower half aluminised to redirect some of the solar flux that missed the absorbing tube structure. Figure 2.18 demonstrates a typical evacuated tube heat collector based on Speyer design, without the reflector. The most important problem at the time was metal glass connection which was overcome quit well by manufacturers of apparatus.

Since then different designs of evacuated collectors have been introduced. One of these designs is the evacuated collector with two glass tubes. Two different designs of this type of evacuated collector are shown in Figures 2.19 a and b [52]. In the first design the vacuum region is the space between the two glass tubes, avoiding the problem of vacuum-tight glass-metal seal. The inner glass tube is blackened, or covered with selective coating, to act as the absorber. The fluid injection pipe is placed inside of the inner glass tube. In the second design, a copper (or aluminium) tube/fin, incorporated with a U-type fluid tube, is placed inside the second glass tube. Recently, in a collaboration between the University of Sydney of Australia, Peking University of China and Turbosun Company a series of high performance evacuated solar collectors based on this design have been marketed [53]. The outer surface of the inner tube is sputtered with an advanced selective surface, which assists to achieve high temperatures, up to 500°C. Some other designs of evacuated tube collector are introduced in reference [54].

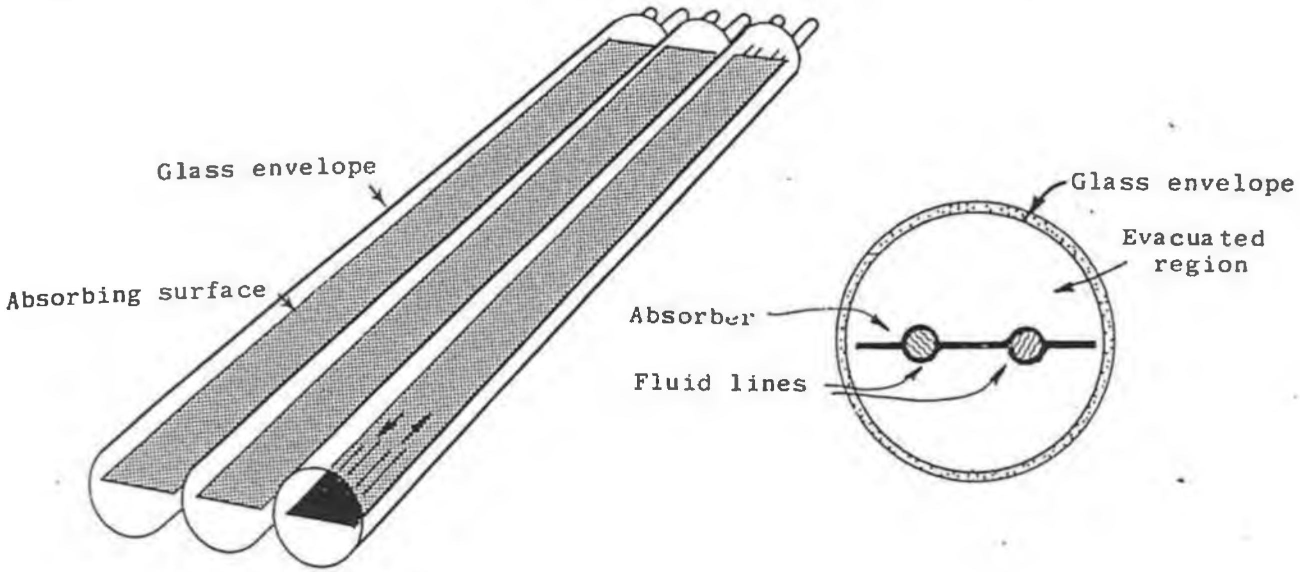
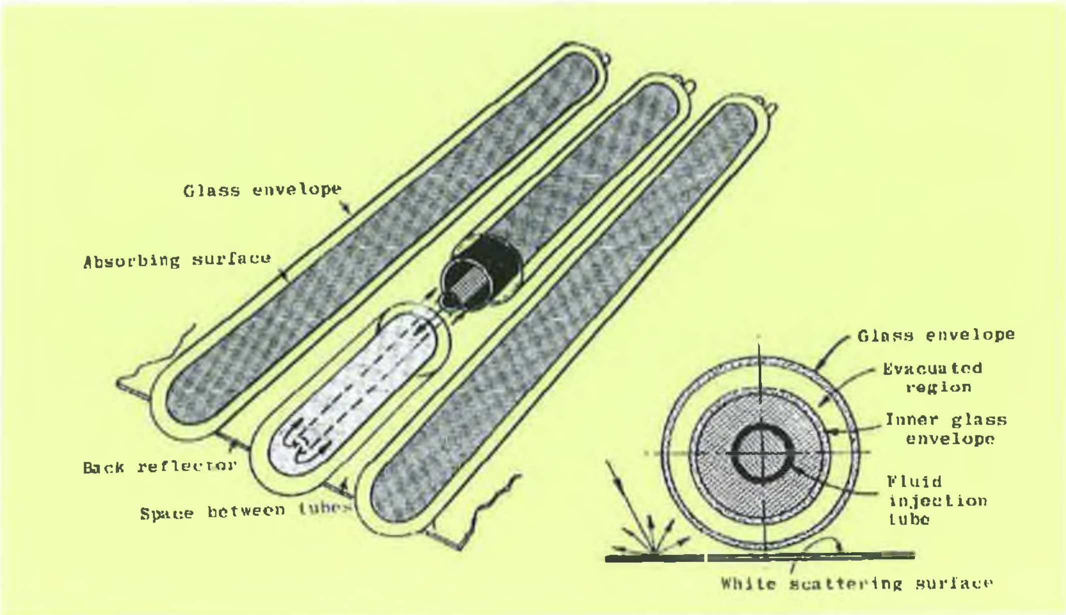
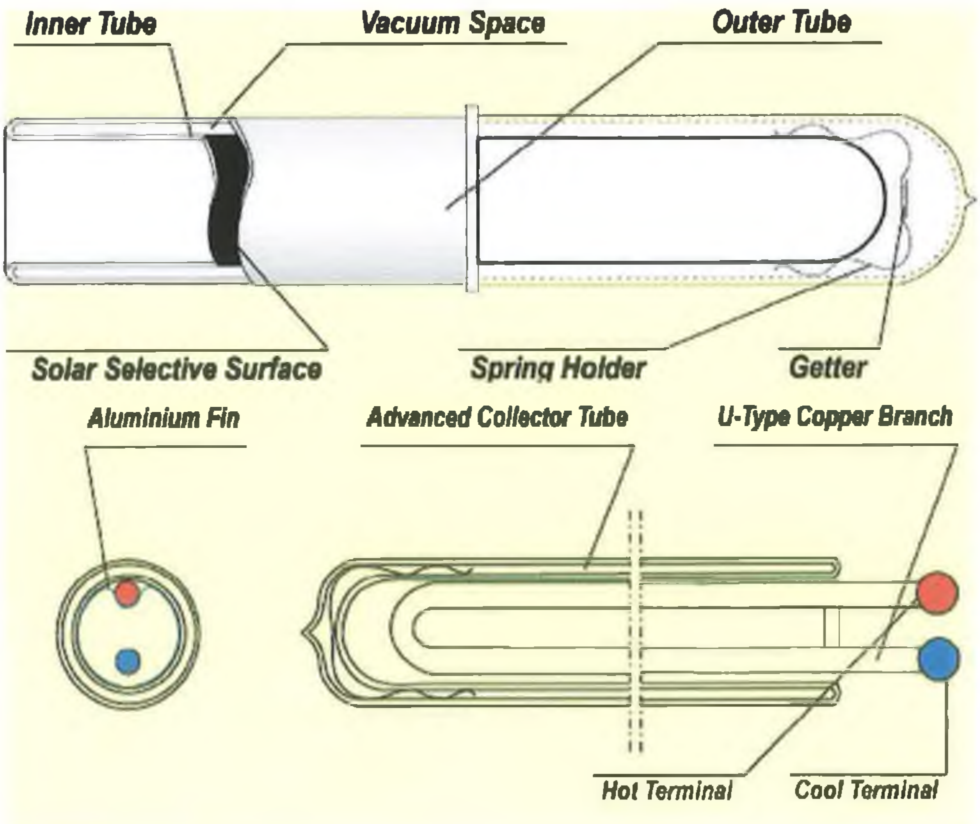


Fig. 2.18 An Evacuated solar collector based on Speyer design



(a)



(b)

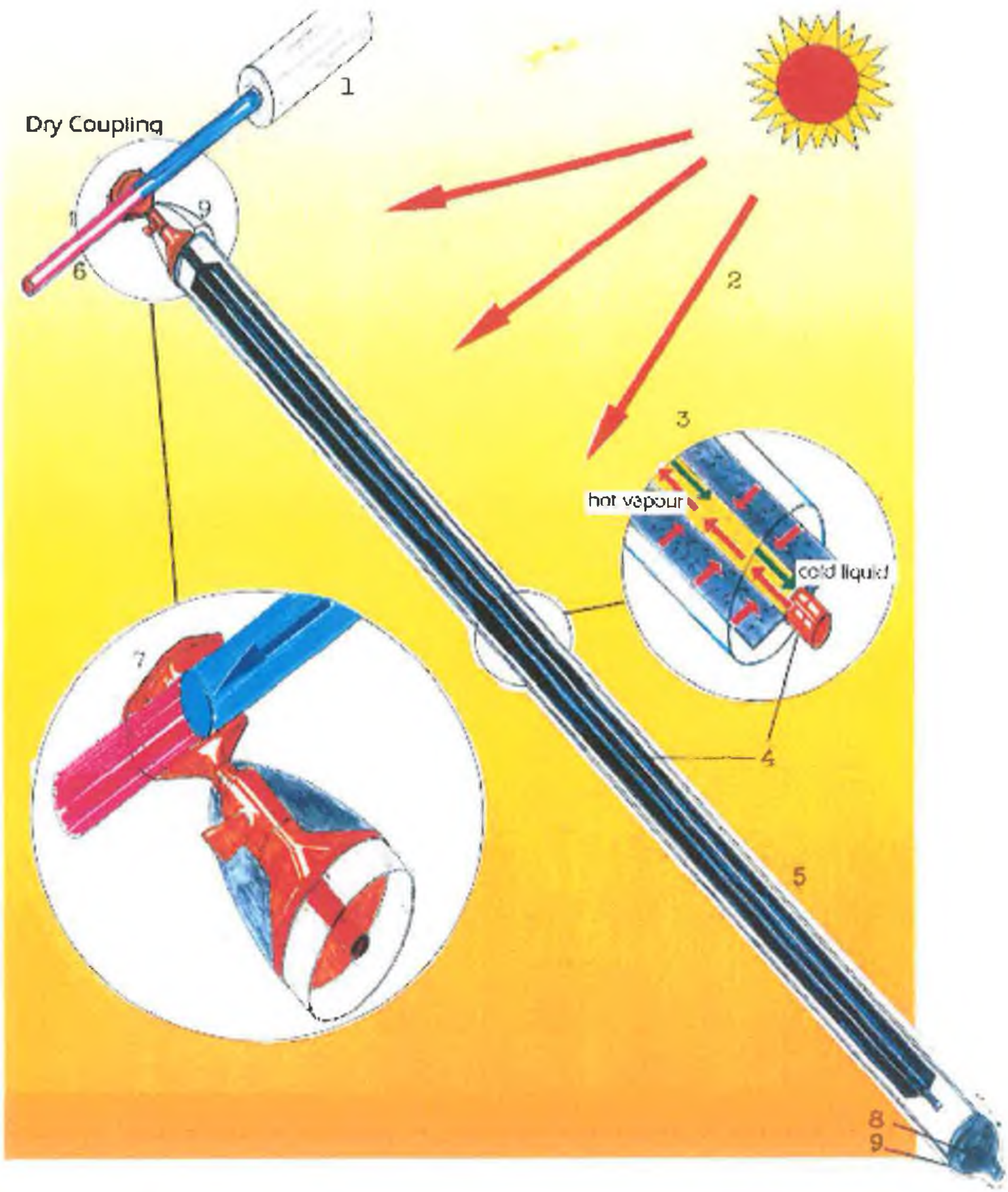
Fig. 2.19 Evacuated collectors with two glass envelopes

2.5.3 High Efficiency Evacuated Heatpipe Collector

Perhaps the most efficient design of evacuated tube collectors is the evacuated heatpipe solar tube which consists of a heatpipe inside a vacuum-sealed tube. The heatpipe is a heat transfer device which is able to transfer large amount of energy with a minimum temperature difference between heat input and heat output. In view of the fact that heatpipes are an extremely important component in high efficiency solar heat collectors a brief review of heatpipe technology is presented in section 2.5.3.1.

As shown in figure 2.20 each tube contains a sealed metal pipe attached to a blackened metal strip. The condenser of the heat pipe protrudes out from the top of the glass tube [55].

There are two alternatives for the absorber fin and heatpipe assembly. The first choice is the system whereby the heatpipe is sandwiched inside an aluminium absorber fin. This ensures a full contact between the absorber strip and the heatpipe. In the second choice heatpipe is stitch welded to the absorber. In this approach the only thermal contact between the heatpipe and the absorber fin is via the thin stitch line of welding between the two. There are also different designs of the condenser of the heat heatpipe.



Evacuated Tube System (EcoTherm)

1. Thermal Insulation
2. Solar Radiation
3. Absorber Plate/Fin
4. Heat Pipe
5. Evacuated Glass Tube
6. System Pipe (Header)
7. Condenser/Heat Exchanger
8. Dome End
9. Rubber Insulation

Fig. 2.20 A typical evacuated solar collector with heatpipe (designed by Ecotherm)

2.5.3.1 Heatpipes

The concept of heatpipe was first introduced by R. S. Gaugler in 1942 [56]. Heatpipes are passive heat transfer devices, which operate by a process of evaporation and condensation of a fluid circulating within a sealed cavity. A typical heatpipe consists of two main sections, namely evaporator and condenser. Many heatpipes also have an adiabatic region, which separates the evaporator and the condenser. Some heatpipes are incorporated a wick lining to improve working fluid circulation. Figure 2.21 shows the cross section of a typical heatpipe.

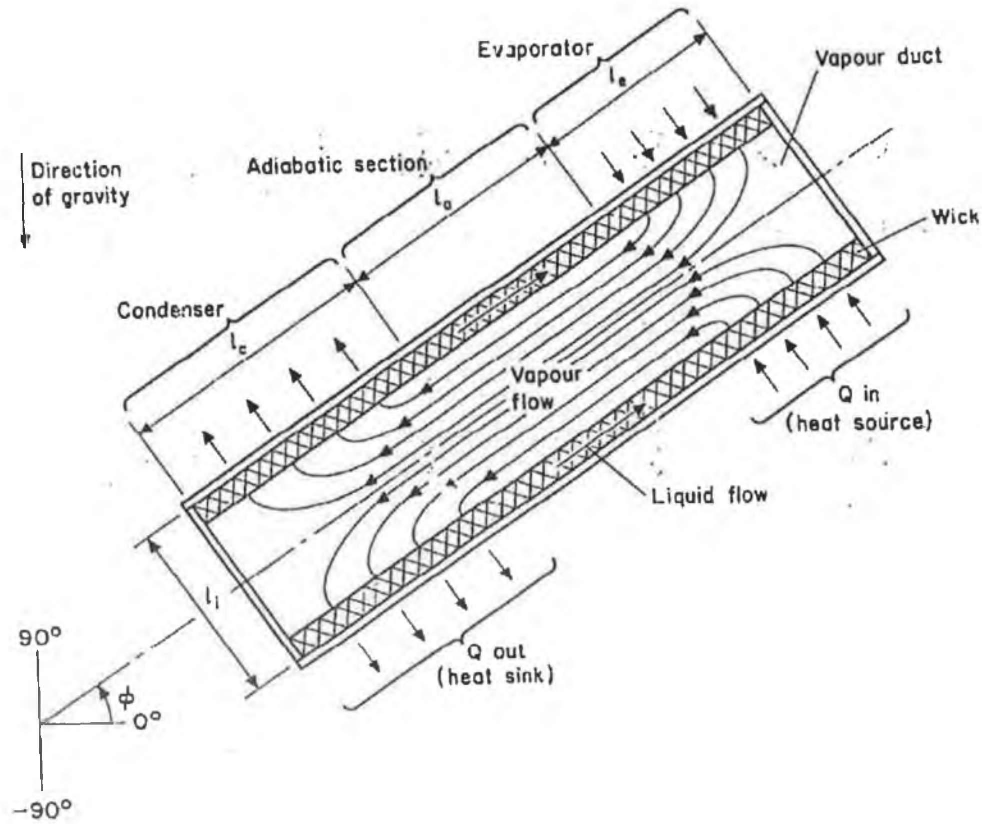


Fig. 2.21 Cutaway section of a heatpipe

The liquid/vapour phase change induced by applying of heat, enables large energy levels to be transmitted in the vapour phase, when a minimal temperature gradient exists. ISOTERIX Ltd. has introduced heatpipes that operates at an overall temperature difference of less than 2°C [57]. The pipe generally has the length to diameter ratio (L/D) in the range of 10 to 100 [58]. In the case that L/D ≤ 10 the heat transfer device is called vapour chamber. Heatpipes have been constructed with diameters varying from 2mm up to almost 1m and with length to diameter ratios from 0.1 up to 450. In theory, the pressure inside the pipe is constant, and therefore, the temperature of boiling fluid is the same as condensing fluid. However, for efficient operation a temperature deference is demanded between the boiling and condensing part. Another temperature difference is required to transfer heat sufficiently from the heat sink and to the evaporator. In fact, what makes the heatpipe unique is that it can transfer a large amount of heat as compared to a solid bar of materials with high thermal conductivity. For example, a cylindrical heatpipe can be designed to have an effective axial thermal conductivity 300 time that of a copper bar having the same dimension [59].

The operating fluid is circulated with assistance of different forces including capillary forces, gravitational forces, rotationally induced forces and sometimes electrostatic or osmotic forces.

The main advantages of the heatpipe are summarised as following:

1. Long operation life.
2. Since there is only a minute amount of liquid present, there is little possibility of corrosion.
3. A Heatpipe is a unit with high thermal conductance.
4. A Heatpipe is an appropriate heat transfer device for applications with low temperature gradients between the heat sink and the heat source.

Since heatpipes are high conductance devices, they have been successfully used in high efficiency solar collectors in both flat plate and evacuated tube types.

2.5.3.1.1 ***Basic Theory of Heatpipe***

The heatpipe must be designed to achieve a maximum effective pumping pressure. In the typical heatpipes used in solar collectors, the effective pumping pressure is made up of three components, namely the capillary (P_c), axial (P_a), and the normal (P_n) pressure. Figure 2.22 shows the free diagram of a heatpipe.

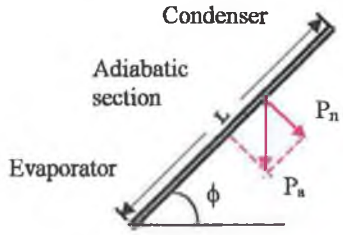


Fig. 2.22 Longitudinal diagram of a heat pipe

As shown in free diagram, the pumping pressure can be calculated as following [60]:

$$P_p = P_c + P_a + P_n \qquad (2.8)$$

The three components in the right side of the above equation are given as follows:

$$P_c = \frac{2\sigma}{r_c} \qquad (2.9)$$

$$P_a = \rho_l . g . L . \sin\phi \qquad (2.10)$$

$$P_n = \rho_l . g . D_v . \cos\phi \qquad (2.11)$$

where:

- σ is the surface tension.

- r_c is the radius of the wick.

- ρ_l is the density of the liquid.

- L is the length of the pipe.

- D_v is diameter of the vapour space.

- ϕ is the angle of pipe makes with horizontal.

In the absence of weak structure, the pumping pressure is just as the resulted from a positive gravitational head. This can be expressed as follows:

$$P_p = P_a + P_n \qquad (2.12)$$

2.5.3.1.2 Classification and Applications

Heatpipes are categorised according to deferent characteristics, such as the temperature range of operation, the degree of thermal resistance variation, geometrical characteristics of the heatpipe, elements of construction.

Heatpipes are classified based on their operating temperature in three categories, as follows [61]:

❶ High temperature heatpipes, 355°C < T ≤ 2725°C. The working fluids for this temperature range are: Mercury, Caesium, Potassium, Sodium, Lithium, Gallium, Silver and Indium.

❷ Moderate temperature range heatpipes, -150°C < T ≤ 355°C, where the working fluids for this temperature rang are: Acetone, Ammonia, Freon, Methanol, Water, Dowtherm.

❸ Low temperatures are cryogenic heatpipes with operating temperatures ranging between -267°C < T ≤ -150 °C. The working fluids for this temperature range are: Hydrogen, Neon, Nitrogen, Oxygen and Methane.

The type of heatpipe applied in the solar collectors belongs to the medium temperature range heatpipes. In solar collector's design the condensation zone is at the higher level than evaporator section, and therefore, the gravitational force helps the circulation of condensed fluid. Hence, commonly solar collators the heat transfer device is without wick structure and it is known as two phase-closed thermosyphon.

Application of heatpipe is not restricted as a heat transfer medium in solar collectors and they are used in extensive range of applications. Heatpipes are designed to satisfy one or more of the following conditions:

❶ Perform as a very high thermal conductance.

❷ Achieve an isothermal surface at low thermal impedance, thermal flattering.

❸ Act as a thermal flux transformer.

❹ The ability to keep a constant source temperature under condition of heat input fluctuations.

❺ Behave as a thermal diode or switch.

2.5.3.1.3 Two phase-closed Thermosyphon

Two phase-closed thermoshyphon is a heat transfer device, which performs same function as heatpipe in which circulation of liquid instead of capillary forces relies on gravity forces. Therefore, the condenser must be situated at a higher level than the evaporator. The thermosyphone must not be confused with gravity assisted heat pipe, since the circulation of the working fluid in gravity assisted heatpipes is resultant from the combination of the capillary and the gravity forces [62]. A typical thermosyphon comprises of a circular tube with uniform cross section and a working fluid. The inclination angle varies between 5° to 90°. Figures 2.23.a and 2.23.b show the cut away sections of a vertical and tilted thermosyphon, respectively.

In a thermosyphon, liquid must be distributed uniformly over part of evaporator not covered by pool. In the case of inclined thermosyphon, used in solar collectors, to improve the liquid contact with evaporator surface, creating spherical grooves or installation of a wick lining along the evaporator section is suggested.

The quantity of working fluid in the tube is an important factor in the performance of the thermosyphon. The insufficient amount of liquid inside the tube leads to dryout and excess must be avoided because any liquid carried up into the condenser reduces the performance by rendering the covered area of the condenser virtually useless. The liquid filled defined as the ratio of the volume of the liquid in an unheated pipe, V_l, to the volume of the vapour:

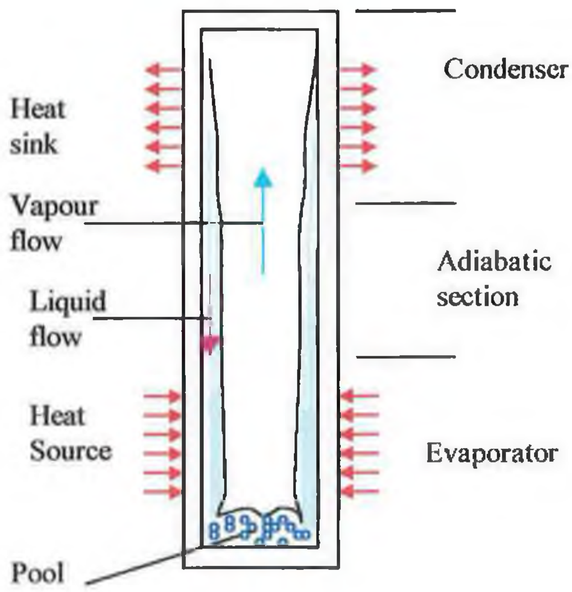


Fig.2.23.a Cross section of a vertical heatpipe

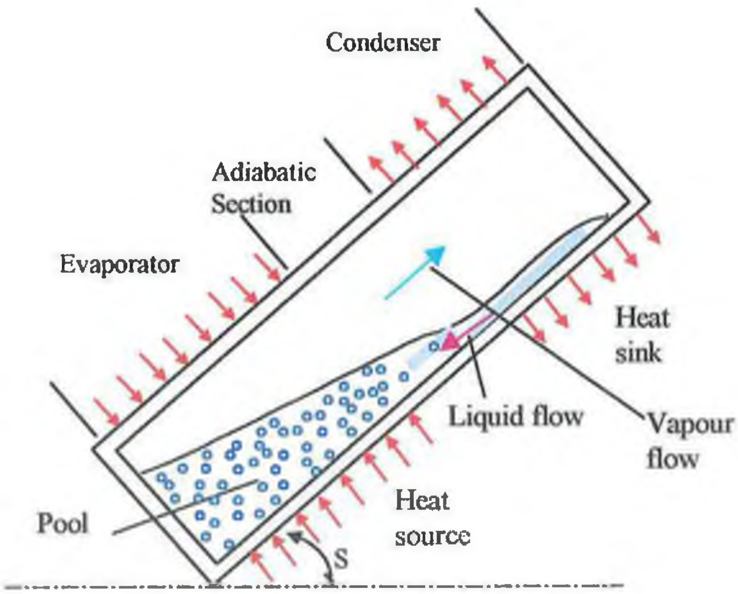


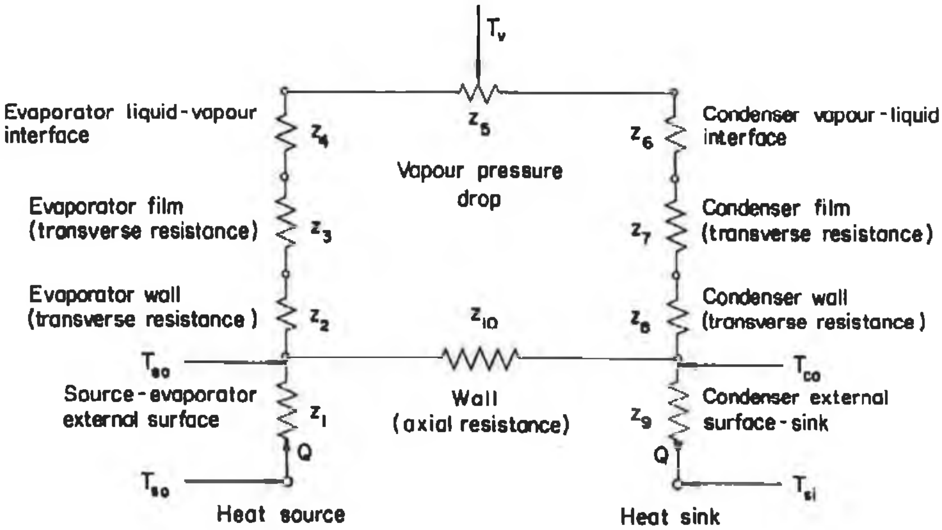
Fig.2.23.b Cross section of a tilted heatpipe

$$F = \frac{V_l}{A.l_e} \qquad (2.13)$$

The heat transfer via heatpipe can be characterised with the overall thermal resistance. The relationship between the heat transfer rate and overall thermal resistance can be expressed as follows:

$$Q = \frac{\Delta T}{Z} \qquad (2.14)$$

The thermal resistance diagram of a thermosyphon depicted in figure 2.24. The values of thermal resistors can be estimated by using equations driven in literature [63].



Z_1 and Z_9	The thermal resistance between the heat source and evaporator and between condenser and heat sink.
Z_2 and Z_8	The thermal resistance across the thickness of the container wall in the evaporator and the condenser.
Z_3 and Z_{12}	The internal thermal resistance of the and the condensing fluid
Z_4 and Z_6	The thermal resistance occurs at vapour–liquid interface in the evaporator and condenser.
Z_5	The effective thermal resistance due to the pressure drop of vapour as it flows from evaporator to condenser.
Z_{10}	The axial thermal resistance of the wall of the container.

Fig. 2.24 The thermal resistance diagram of a thermosyphon

2.5.3.1.4 ***Selection of Working Fluid***

One of the important factors in the function of a heat pipe (or thermosyphon) is related to the working fluid's properties. The selected fluid must have a melting point temperature below and a critical point temperature above the operating temperature. The critical, normal melting and boiling temperatures of a number of working fluids is reported in table 2.1. Recently, environmentally friendly substitutes for some of the CFC gases have been introduced.

Operating Temperature	***Critical temperature (^oC)***	***Melting point (^oC)***	***Boiling Point (^oC)***
Acetone (C_3H_6O)	236	-95	56
Ammonia (NH_3)	132	-78	-33
Freon 11 (CCl_2)	198	-111	24
Methanol (CH_4)	240	-94	65
Water (H_2O)	1165	0	100

Table 2.1 Critical, normal melting and boiling points for working fluids of moderate temperature heatpipe (Heat and Electricity from solar Energy, Solar Enrgie- Technik GmbH)

2.5.3.1.5 ***Operating Limits***

Although heat pipe is a very high thermal conductance heat transfer device, but the rate of heat transfer along a heatpipe (thermosyphon) is subjected to a number of maxima, known as "operating limits" [64]. These limits are originated from the fluid mechanics principles and resulted from a break down or a rate limit in circulation of a working fluid.

The lowest limit at any given operating temperature defines the maximum rate of heat transfer at the operating temperature. The operating limit in two phase closed thermosyphon are expressed as follows:

Entrainment limit

As vapour and liquid flow in the opposite directions, a shear force exists at the liquid-vapour interface. If the vapour velocity is adequately high, a limit can be reached when liquid is torn from the liquid-vapour interface of the wick and entrained in the vapour.

Dryout limit

When the liquid fill is not sufficient to cover all the pipe wall above the pool, most of the liquid evaporates before it reaches the pool and leads to dryout.

Sonic limit

In a heatpipe, the vapour stream accelerates and decelerates, as a result of addition and removal of the vapour, respectively. This causes velocity variations along the pipe. The maximum mass flow rate is attained when the exit velocity of evaporator reaches the local sonic velocity.

Vapour pressure limit

When operating a heat pipe or thermosyphon at a pressure below atmosphere, the pressure drop of the vapour may be significant in comparison with the pressure in the evaporator.

Boiling limit

The boiling limit occurs when a stable film of vapour is formed between liquid and the evaporator surface. The boiling limitation is a limitation of radial heat flux density, while others are limitations of the axial heat flux.

2.6 Composite Evacuated Collectors

Evacuated collectors can be incorporated with line focus concentrators to achieve higher temperatures and efficiencies. This combination is known as composite evacuated collector.

O'Gallaher et al. (1980) have shown that for a tubular absorber, the reflector shape leading to a maximum absorption of radiation by cylindrical absorbers is an involute [39]. The characteristic and geometry of the concentrator with involute profile is discussed earlier in section 2.3.3.

Although integration of evacuated tube collectors and reflectors yields to higher temperature, but this increases the associated expenses of these systems.

It must be pointed out that since reflectors can only perform under direct sunshine conditions these systems are not suitable under diffused solar radiation.

2.7 Conclusion

In summary, the solar heat collection system can be listed as follows:

1 - Flat plate collectors.

2 - Evacuated tubes

3 - Concentrating collectors

4 - Composite Evacuated collectors

Selection of an appropriate solar heat collection system relies on several factors including generating temperature, application, costs and climate condition. Each of the above-mentioned systems has some advantages and disadvantages in contrast to others. Flat plate collectors are suitable for applications with moderate generating temperature (up to 100°C). These systems can utilise both direct and diffuse components of solar radiation, and they are relatively cheap. The main drawbacks of these collectors are their poor efficiency and limited generating temperature.

For application with generating temperature higher than 120°C evacuated tubes are a perfect selection. These collectors can produce temperatures up to 500°C. Due to application of vacuum technology the heat losses from the collector are minimised, and the collector enjoys a high efficiency. Also they can perform in cloudy climates. The main drawbacks for these systems is their high initial cost and their relatively complex manufacturing process.

Concentrating collectors like evacuated tubes, are also used in applications with high generating temperatures. Although these systems enjoy high efficiencies, but they can not perform under cloudy climates, since they can only utilise the direct component of the solar radiation. To achieve higher temperature and efficiencies, concentrators can be coupled with tracking systems . However, this increases the total cost of the system and also the tracking system requires regular maintenance.

The last category of solar heat collection systems is the composite evacuated tubes. Although these systems are very efficient, and they can generate very high temperatures, but they are restricted due to their high costs (higher than evacuated tubes), and due to the limits of concentrators.

Chapter 3

Review of Solar Cooling Systems

Chapter 3

Review of Solar Cooling Systems

3.1 Solar Cooling

Cooling systems have taken a major part in improvement of living standards and development in the world. The necessity of cooling systems applications is extended in many fields such as food and medicine preservation, air conditioning etc.

Refrigeration and air conditioning are two distinct but related fields in cooling. Refrigeration in the engineering sense can be described as “maintaining a system at a temperature less than the temperature of the surrounding” [66]. Thus, refrigeration is the process of removal of heat from a low temperature sink to a high temperature sink. Air conditioning is defined as “provision of a comfortable indoor environment by replenishing, heating, cooling, humidifying, or dehumidifying the atmosphere” [67].

According to Clausius statement of second low of thermodynamic “It is impossible to construct a device that operate in a cycle and produces no effect other than transfer of heat from a cooler body to a hotter body” [68]. This statement is related to refrigerator (or heat pump) and states that it is impossible to construct a refrigerator without an input of work.

Conventional refrigeration systems are driven either by electricity or with the heat generated from burning fuels. Unfortunately the dilute nature of the solar energy that reaches our planet and also its intermittent nature make it impractical and commercially viable to operate the existing conventional cooling and refrigeration systems with solar energy. However, it would be a fare comment to say that solar cooling and refrigeration is nothing other than modification of some of the available conventional technologies to be powered by solar energy. In fact the main theme of the challenge in the field of solar powered cooling is the Research & Development to result a practical and commercially viable system.

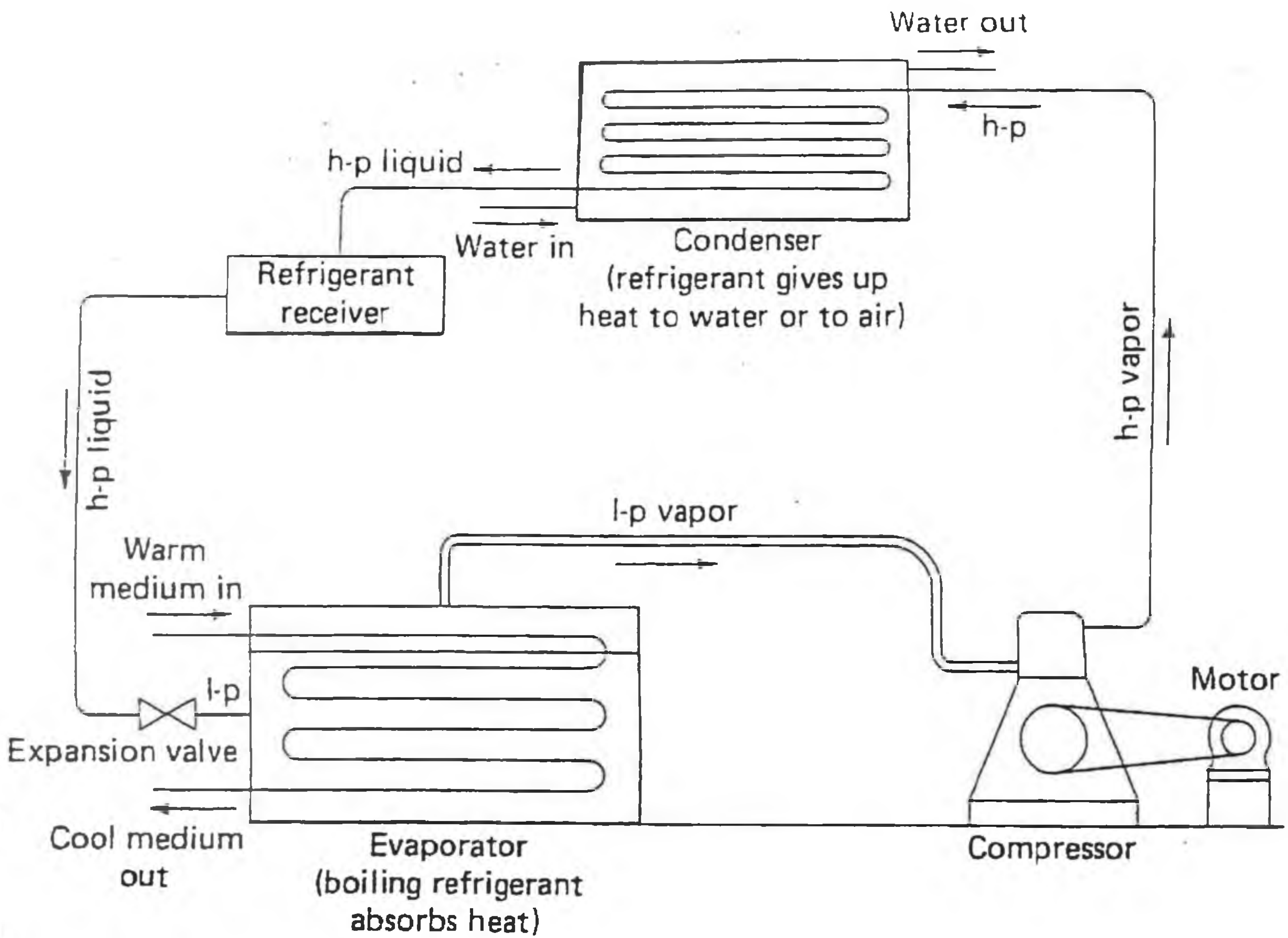
In recent years, great attention is shifted toward solar cooling appliances and considerable R&D programmes have been focused to develop a viable alternative cooling systems. Although solar cooling technologies have not, so far, improved enough to make them commercially viable, but they can be competitive in special applications. These systems are viable and practical in the isolated and remote regions of world that have no access to a utility grid and suffer from inadequacy of fuel transportation. The potential areas that solar cooling can be successfully applied, so far, are storage of vaccine, food preservation, ice production and air conditioning.

There are five main types of design in autonomous solar power refrigerators. The processes by which cooling is obtained include mechanical vapour compression, vapour absorption, humidification-dehumidification (desiccant cooling), adsorption and vapour jet. The systems described in this chapter are those with greatest potential for adoption to solar power.

3.2 Vapour Compression Cooling Systems

Vapour compression cooling cycle is the most known refrigeration cycle, being used in domestic and many other refrigeration systems. The flow diagram of a conventional vapour compression system is brought in figure 3.1. The refrigeration cycle can be divided into two zones, high and low pressure zones. The incoming low-pressure refrigerant vapour from evaporator is pressurised in compressor. This compression (or pressurisation) process is associated with an increase in temperature. An ideal compression is assumed as a reversible and adiabatic process. However, a real compression process is neither reversible nor adiabatic, as a result of changes of refrigerant entropy due to heat transfer between the compressor and environment. After compression, the high pressure refrigerant vapour flows into the condenser and by contacting to the condenser walls gets cold and condenses. Then the liquid refrigerant passes through the expansion valve, and partially evaporates. The complete evaporation occurs in evaporator when the low pressure liquid absorbs the heat from the low temperature sink (or cooling box). The refrigerant vapour returns to compressor and cycle repeats.

Conventional vapour compression systems are electrical powered. They have also been adopted with solar energy. In this case, the system is operated either by PV panels in which solar energy provides the required electrical power for operation of compressor, or the generated heat of photothermal conversion powers a heat engine driving the compressor. Perhaps the main advantage is that the refrigeration system could be built from commercially available components using the standard layout.



h - p = high pressure

l - p = low pressure

Fig. 3.1 A conventional vapour compression refrigeration system

For adoption of a conventional compression cooling system with PV panels several modifications and changes must be accomplished. The a.c. compressor must be replaced with a special d.c. powered compressor. However, by application of an inverter, there is no need to change the a.c. compressor with a d.c. compressor. Also because of the intermittent nature and continuous variation of solar insolation presence of batteries and a regulator is necessary. The schematic of a Photovoltaic refrigeration system is shown in figure 3.2.

A particular novel new approach to the development of a new exciting low cost modification of existing conventional compression cooling system was carried out in IT Sligo by the author. The assessment of the new novel approach together with that of the engineering economics of the project indicates a new and exciting development in the field of solar cooling and refrigeration with reliability and cost-effectiveness being the major governing parameters. The development and evaluation of this new novel system is discussed in details in Chapter 7.

Photovoltaic powered refrigeration systems enjoy number of advantages and suffer from some disadvantages. PV systems are reliable and they require low maintenance costs. However, the price of generated electricity by PV technology is still high. Also the efficiency of the solar cells used in such applications is quite low and glides between 15% to 20% at low light and full sunshine, respectively. It must be noted the price of photovoltaic has been decreased dramatically, over the past years. According to a report from the World Bank in just 15 years the price of photovoltaic models had a decrease by a factor of 10 [69]. PV systems are commercial in cases that extending the power lines is very expensive (more than one-quarter mile away from grid) and for optimum system performance PV modules should at least have a full daily exposure about 5 hours,

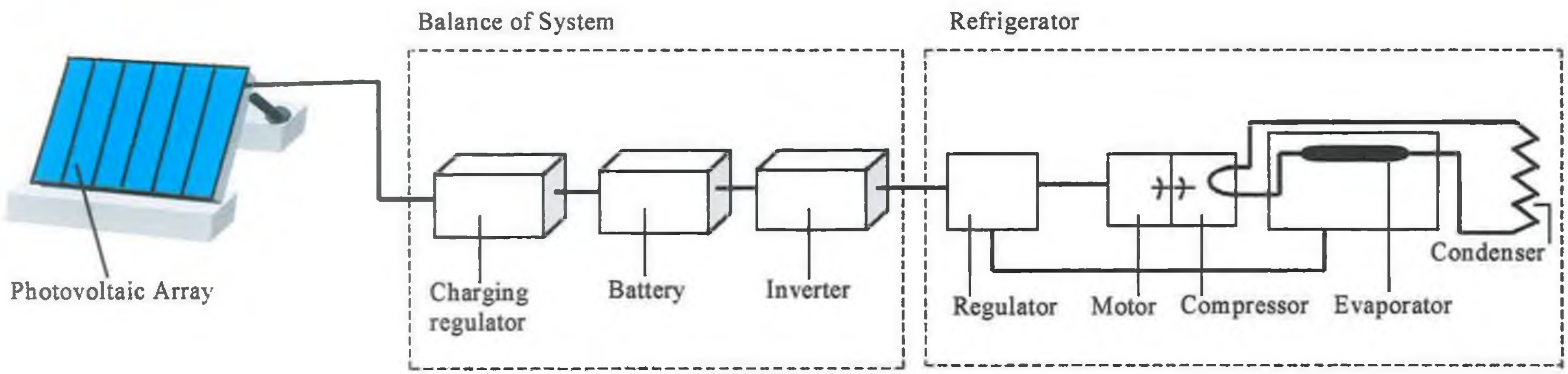


Fig. 3.2 Schematic of a photovoltaic powered refrigeration system

all year. Therefore, it can be argued that PV systems are desirable in especial applications in the remote regions. Perhaps preservation of vaccine can be referred as the most important application of PV powered refrigerators [70,71,72].

The second option to run a vapour compression refrigeration system with the assistance of solar energy is to couple the compressor with a heat engine. A typical solar thermal powered vapour compression system consists of these subsystems: 1- solar collector array. 2 – heat engine. 3 – a vapour compression refrigeration system. There are two types of heat engines, which have been suggested and incorporated with vapour compression systems: Rankine heat engine and Sterling & Bryton heat engine. Sterling-Bryton heat engines perform superior at energy input temperatures over 200°C, whereas Rankine heat engine is more efficient at temperatures below this temperature. Hence, Rankine engines are more interested and most efforts on adoption of a vapour compression cooling systems with solar-thermal power have been focused on this type of heat engine. A typical Rankine engine vapour compression cooling system is depicted in figure 3.3.

Prigmore and Barber introduced a solar powered Rankine cooling system [73]. The system used $58m^2$ of flat plate collector which was coupled to the 3 ton (10.6 kW) Rankine cooling system. The evaporating temperature was 7°C and condensing temperature of 20°C was recorded. The Rankine cycle efficiency was about 11%. A COP of 7.4 for air conditioning and a total solar COP of 0.2 were quoted.

The solar thermal powered vapour compression systems encounter with some drawbacks and disadvantages. The efficiency of a heat engine is limited with unachievable Carnot efficiency ($\eta=1- T_L/T_H$) and for increasing the efficiency higher generating temperature is required which cannot be produced by flat plate collectors. Also the energy losses of photothermal conversion lower the overall solar COP. Aside from the poor performance, heat engines are relatively complicated and require regular maintenance.

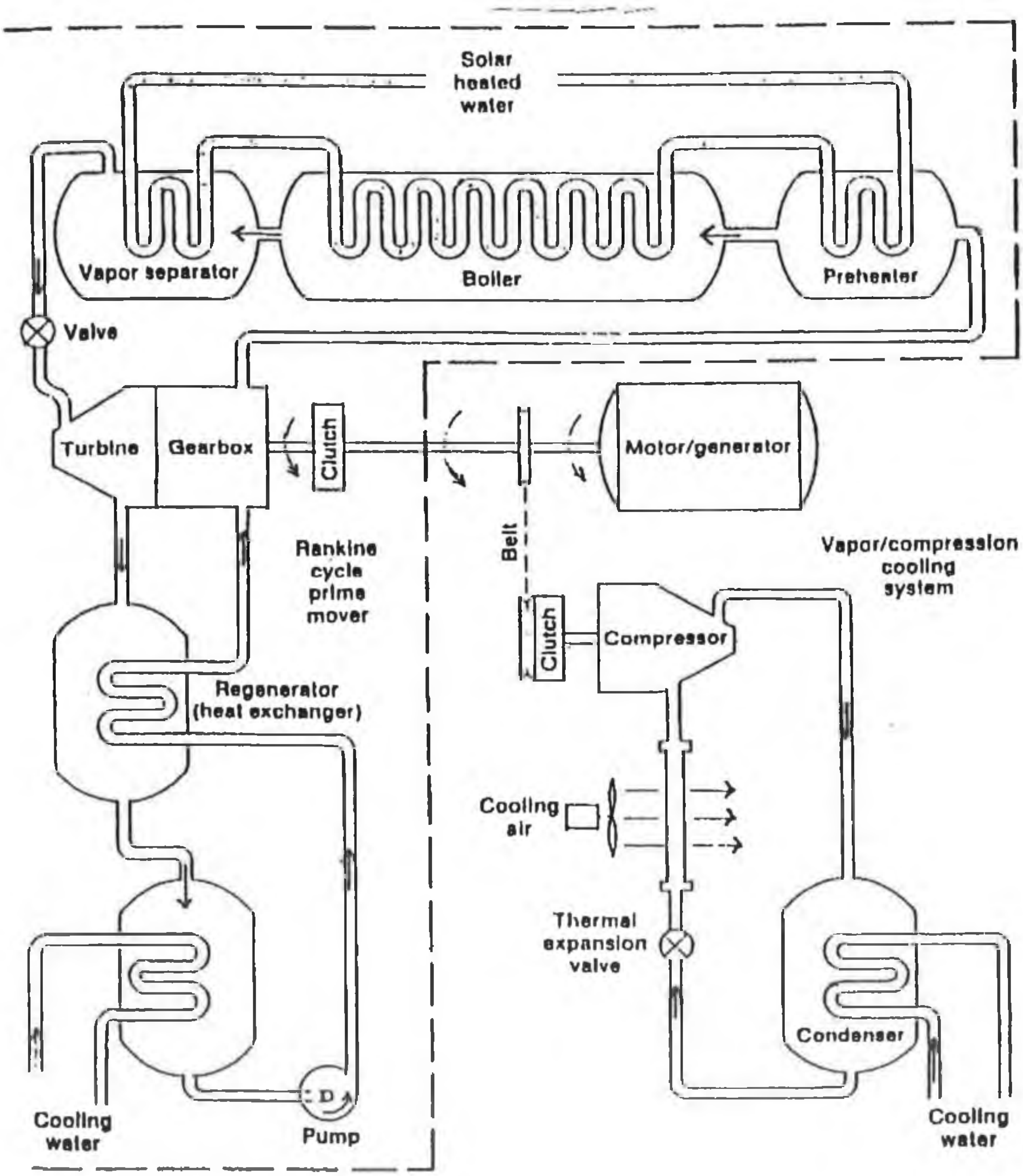


Fig. 3.3 Schematic diagram of a solar driven Rankine engine Vapour compression cooling system (Engineering principles and concepts for active solar systems, Solar Energy Research Institute, Hemisphere p.90)

3.3 Desiccant Cooling Systems

In regions with high humidity, the ambient air contains a large amount of moisture. This moisture is a serious problem for both human comfort and air conditioning systems. Therefore, in places with high humidity direct evaporative cooling systems are not suitable for human comfort and health. For places with this climate condition, desiccant cooling systems are desirable.

In a typical desiccant system, the moisture, latent load, in the process air is removed by a desiccant material in a dehumidifier. A sensible cooler like a heat exchanger or an evaporative cooler reduces the temperature, sensible load, of the dried process air. The desiccant material in the dehumidifier is regenerated. Therefore, the basic component of a desiccant cooling system are as follows: 1 - a dehumidifier, 2 - a sensible cooler, 3 - a regenerator.

Desiccant systems enjoy a number of advantages. Because of separate humidity and temperature control better humidity control is possible. Also application of CFCs is reduced when a desiccant system is incorporated with a vapour compression cooling system or eliminated when it is coupled with an evaporative cooler.

A desiccant material must satisfy a number of properties including noncorrosivity, nontoxicity, and chemical stability. Desiccant materials are either solid or liquid. In solid desiccant systems, air is directed through the bed of desiccant, which is loaded in a rotating disc. The disc rotates between the regeneration air steam and process. In liquid desiccant systems, the air is dehumidified by contact with a strong solution of liquid desiccant on removal of moisture and weakens the solution. The dilute solution is then directed to a regenerator, where heat drives the moisture and strengthens the solution.

The minimum required generating temperature for solid desiccant is about 70°C, and in the case of liquid desiccant is around 50°C to 60°C [74].

Desiccant cooling systems split into open cycle systems and close cycle systems, where desiccant material can be either solid or liquid.

Open cycle systems have been investigated extensively, and different design configurations have been proposed. Among several open-cycle solid desiccant systems, Penington (Ventilation cycle), and Recirculation cycle have been paid more attention. The schematic diagrams of these systems are shown in figures 3.4 and 3.5. In the ventilation mode fresh air is continuously introduced into the conditioned space and exhausted to the ambient air. In recirculation mode the output air from the conditioned space is reintroduced after being reconditioned. Analytical studies indicated a superior performance of ventilation cycle to recirculation cycle [75]. Hence, most of the works on open cycles with a solid desiccant have been carried out on Penington cycle. American Solar King Co. , in 1984, introduced a packaged desiccant cooling/heating system for residential application [76]. The system used a lithium chloride impregnated honeycomb wheel that could be regenerated with solar energy or natural gas. A thermal COP around 1 was achieved with a nominal 3 to 4 ton cooling effect.

These systems have been discussed in more details by Kreith & kreider [77] and Pesaran et. al [78].

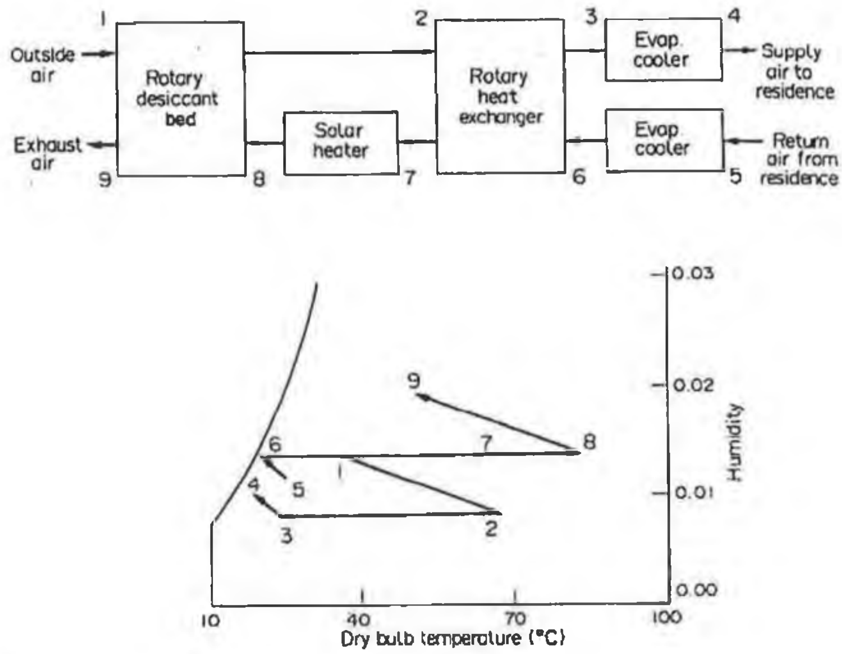


Fig. 3.4 Schematics of the ventilation cycle

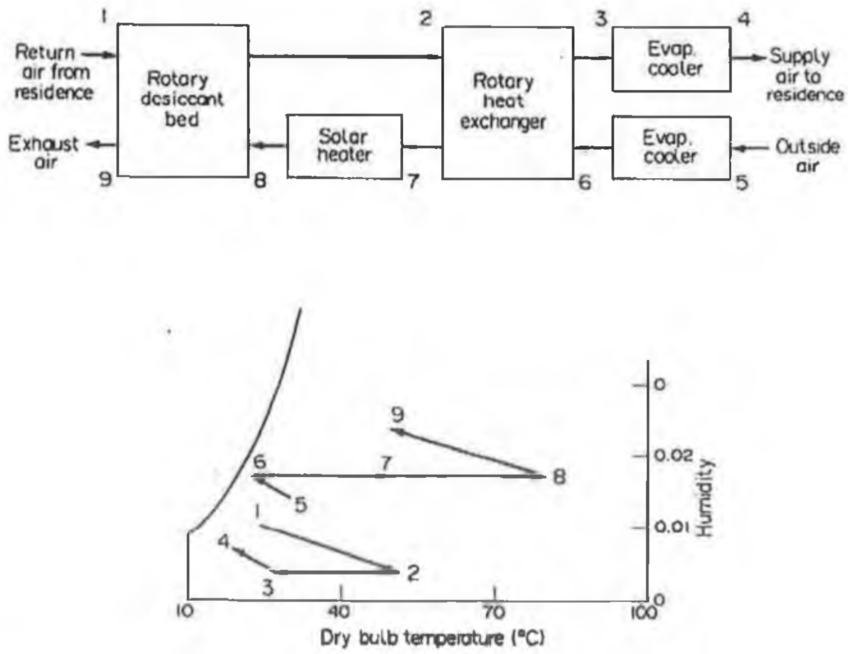


Fig. 3.5 Schematics of the recirculation cycle

As pointed out, in open cycle liquid desiccant systems liquid desiccant is splashed over incoming air and absorbs the moisture. Then the diluted solution is directed to a regenerator where heat is applied and strengthens the solution again. An example of an open cycle liquid desiccant system is given in figure 3.6.

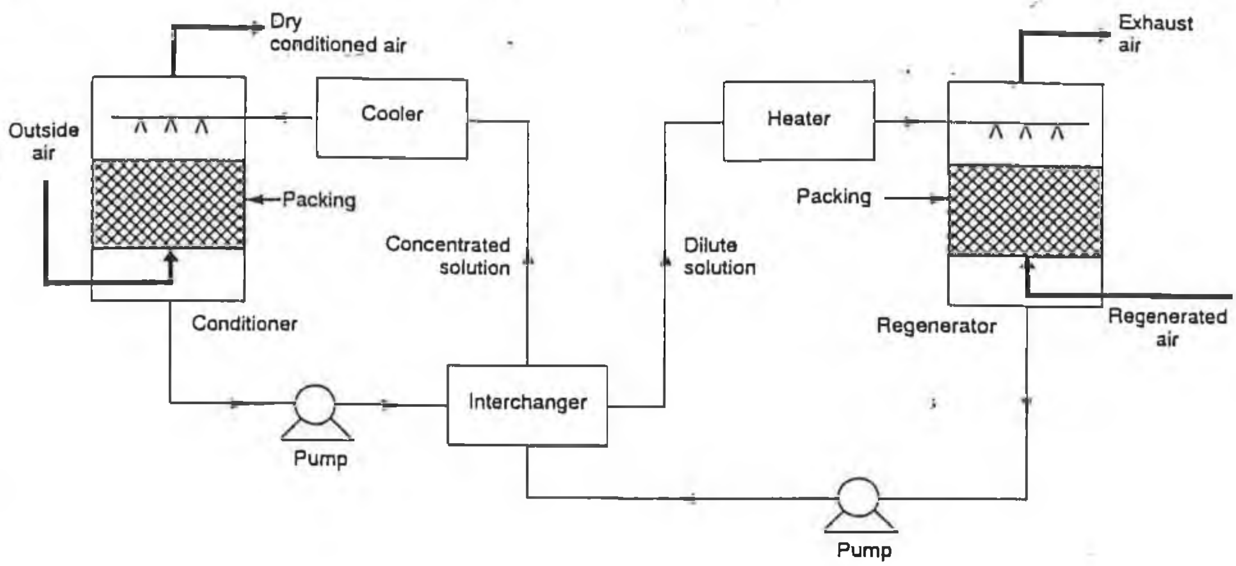


Fig. 3.6 Example of a liquid desiccant cooling system

Liquid desiccant systems have not been extensively investigated in past, because of their predicted modest COP. However, liquid desiccants enjoy some advantages over solid desiccants. As energy stored in the form of chemical energy rather than thermal energy, the reliance on the continuous thermal energy is reduced, and the amount of energy stored is greater. Liquid desiccants do not require complex dehumidifier geometries as the desiccant is usually sprayed over the incoming air and can be regenerated on relatively inexpensive open-flow collectors or tanks. Liquid desiccant systems also offer greater design flexibility than solid systems because the component can be installed in different locations and the liquid can be pumped between them. The main draw back of liquid systems is their large size at low capacities. They can also have carry over problems or corrosion problems if not designed properly.

In fact, there are a few solar powered liquid desiccant cooling systems worthy to mention. A commercially available liquid desiccant cooling system has been reported using 92% tetraethylene glycol solution. The system was coupled with flat-plate collector arrays with generating temperature of about 80°C with a thermal COP of 0.5 [79]. A group of researchers in Colorado State University have investigated and field-tested a solar powered liquid desiccant system which employed commercially available dehumidifying equipment using trithylene glycols as desiccant [80].

The operation of closed cycle solid desiccant cooling systems is similar to that of thermally activated heat pump. The cooling and heating effects are achieved by evaporation and condensation of an absorbate (as refrigerant) on the walls of desiccant container. The major investigation on this system is contributed to the Zeopower [81].

3.4 Adsorption Cooling Systems

Adsorption is defined as a reversible process by which molecules of a fluid are fixed onto a solid matrix, typically a surface or a porous material. It must be noted that there is no chemical combination between the fluid and solid substance. Adsorption cycles for refrigeration were first introduced in early 1900's by Plank and Kuprianoff [82]. An adsorption cooling unit consists of a condenser, an evaporator plus one or several absorbers. The main difference between vapour compression cycle and adsorption cycle is that the mechanical compressor in vapour compression cycle is replaced by a thermal compression system. Figure 3.7 illustrates how an adsorption cooling cycle operates.

Because of adsorbent loading, adsorption cycle operates intermittently.

The cycle operates in two phases. Initially system is in the low pressure, and, the temperature is equal to the ambient temperature. As the adsorber is heated up, pressure increases. Although in theory during the pressurisation, concentration is assumed to be constant, but in fact, pressurisation process gives associated decrease in concentration. As more heat applied to the absorber, the temperature continues to increase and more refrigerant is desorbed. When pressure of the system reaches to the saturation pressure, the refrigerant condenses. As more heat is put into adsorber more refrigerant is drawn off and condenses. In this stage, the pressure of the system is stabilised. This phase stops when the desired adsorbent temperature is attained. At this stage refrigerant concentration in adsorber is minimum.

The second phase starts when adsorber is cooled back or releases heat. This causes the pressure drops from the condensation pressure down to the evaporation pressure. This period is equivalent to the expansion in compression cycle. As the adsorber continues rejecting heat to the ambient, it starts adsorbing the surrounding gas. Since the pressure in the evaporator is less than the saturation pressure of the liquid refrigerant some of the refrigerant evaporates and causes the cooling effect.

Adsorption cooling systems are inherently simple. For operation of these systems no auxiliary electricity or cooling water is required and there is no moving component in the system. Also they avoid certain problems associated with basic intermittent absorption system such as need for a rectifier, and liquid seals and check valves to ensure mixing of solution during absorption. Since they eliminate use of CFCs and HCFCs, these systems are environmentally sensitive. The main disadvantages of these systems are intermittent nature of the cycle and large amount of the adsorbent/refrigerant, even for a small load.

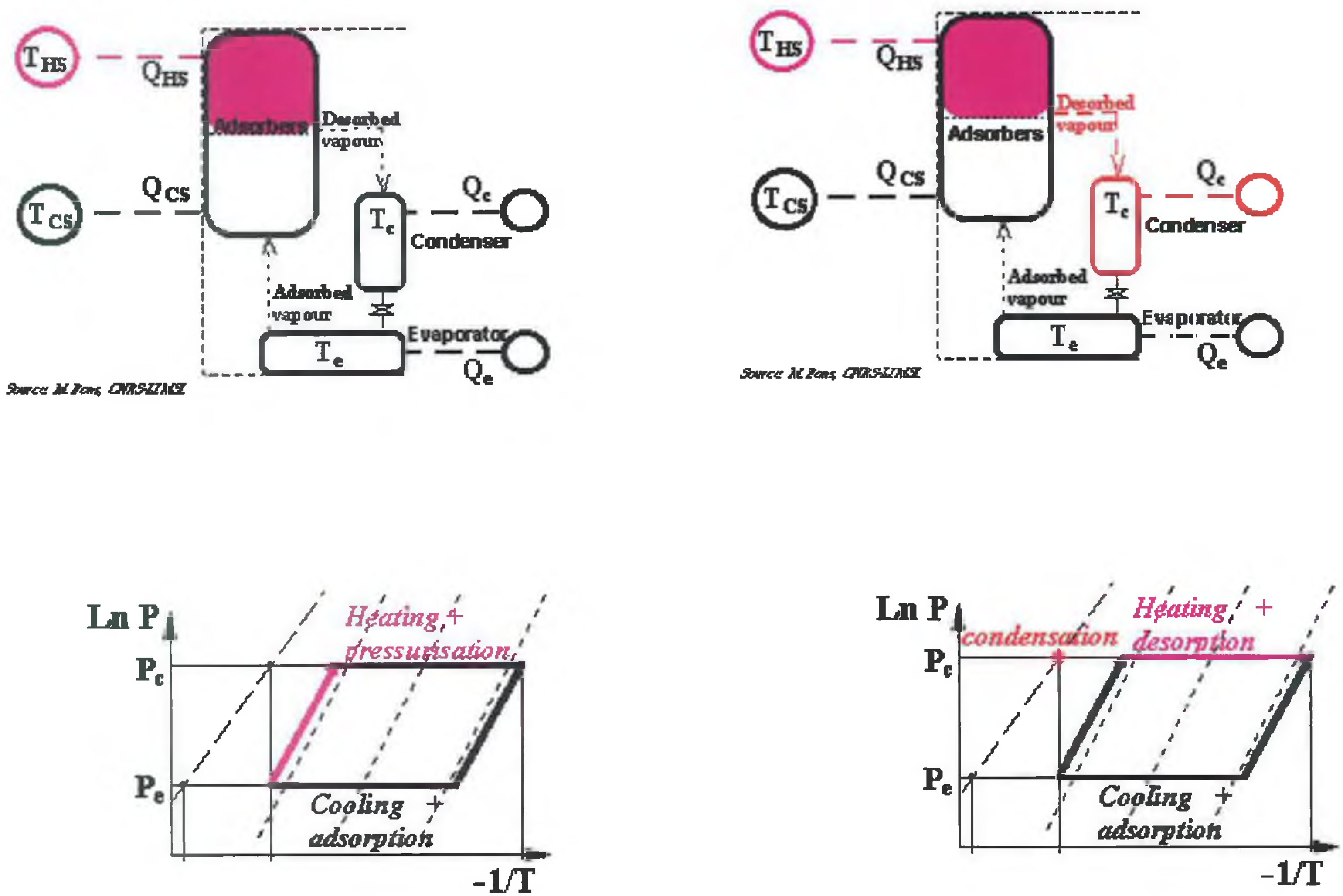


Fig. 3.7.a Pressurisation and desorption processes in the adsorption refrigeration cycle.

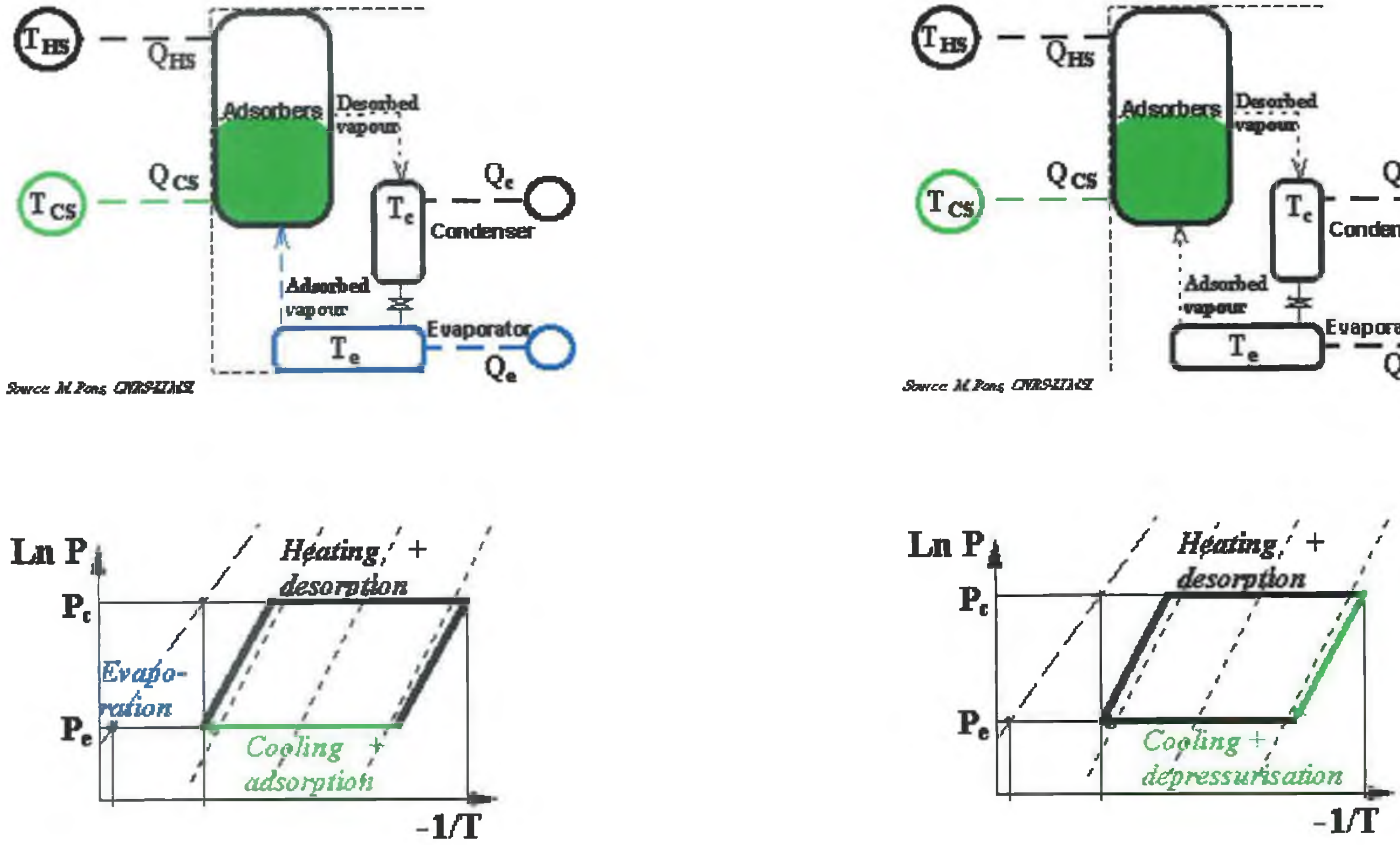


Fig. 3.7.b Depressurisation and adsorption processes in the adsorption refrigeration cycle.

Solar energy has received a great interest as an energy source for the operation of adsorption cooling systems. This is primarily due to the intermittent nature of the adsorption cycle coinciding with the nature of the energy source, sun. Different designs of solar adsorption systems have been introduced by researchers and various refrigerant/adsorbent pairs have been investigated.

An intermittent charcoal-methanol adsorption system was designed and constructed and tested in AIT (Asian institute of technology) [83]. The system was using 15 copper tubes with the length of 1.2 m and diameter of 54mm housed in a galvanised iron sheet with a single glass cover as solar collectors, with effective area of $1m^2$ with efficiency varies between 33% to 44%. The collector tubes contain 17.8kg activated charcoal. The system was using a water-cooled condenser. Evaporator temperature is reported below -7°C with minimum -12°C for some nights, and system was capable to produce 4kg ice per night. Some modifications on the initial design were accomplished. Modifications were: redesigning of the receiver and the evaporator for improvement of ice production and handling the greater amount of methanol, replacement of copper tube with the other type of metal tube to prevent formation of formaldehyde and dimethyl ether.

Also some studies on activated carbon has been carried out in University of Warwick. Critoph reported the design and manufacturing of a prototype ammonia carbon adsorption unit [84]. The adsorber consisted of 15 stainless steel tubes, each 2m length with o.d of 42mm and 1.1mm wall thickness covered with a selective coating. The unit was tested under several laboratory conditions. For the sinusoidal variation over 10 hours with a peak of 942 W/m^2 a maximum collector temperature of 145°C was achieved, after 5 hours insoltaion. For this test the collector was isolated from the cold box for six hours whilst the collector allowed to cool down. Then the isolated valve was opened to achieve the maximum rate of cooling. The evaporator temperature dropped rapidly to -1°C with production of 3Kg ice. A cycle COP of 0.22 and an overall radiation COP of 0.06 have been calculated.

Recently, Critoph and Telto reported a study of a adsorption cooling system with different collectors [85]. The types of collector were single glazing, double glazing and single glazing with transparent insulation material which is a 100mm polycarbonate honeycomb. The collector comprised fifteen 44mm o.d stainless tubes 2m in length with collecting area of 1.43m^2. The surface was coated with selective surface film (α=0.93). The condenser was stainless steel tube with 12.5mm in diameter and length of 2m coiled above the receiver in a water tank. The solar simulator consisted of 48 incandescent 100W lamps with the maximum radiation equal to 969W/m^2. The radiation was increased from zero at 6:00 a.m. to a maximum 960W/m^2 in the midday and falling to zero at 6:00 p.m. the result of the tests are tabulated as follows:

Parameter	Single glazing	Double glazing	Single glazing with TIM
Cycle time (h)	24	24	24
Condensing temperature (°C)	46	32	40
Evaporating temperature (°C)	0.9	0.7	0.9
Ammonia mass collected (g)	968	1109	723
Solar heat input (MJ)	19.8	19.8	13.8
Refrigerating load (MJ)	1.2	1.4	0.9
Solar COP	0.061	0.071	0.065

Z. Chang and L.Z. Fu constructed a solar assisted carbon-methanol adsorption system [86]. The solar collector-adsorber comprises of the absorber in the shape of a steel rectangular box covered with selective paint ($\alpha \geq 0.9$, $\varepsilon \leq 0.65$) and a single glass cover with a V corrugated transparent Mylar film. Also the collector had two windows in the lower and upper sides for cooling the adsorber during the night. The absorber contained 17Kg of activated carbon. The tests revealed that the desorption started at 10:00am and it ceased at 4:00pm. The maximum collector temperature of $105^{o}C$ during the test period was recorded and a maximum Solar COP of 0.111 against a collector temperature of $80^{o}C$ with the ambient temperature varying in the range of 11-25 was estimated. They also carried out a computer simulation analysis. It was defined that a maximum COP of the adsorption cycle (around 0.5) can be reached under the desorbing temperature of $110^{o}C$.

Another refrigerant-adsorbant pair which has been studied by researchers is the zeolite-water pair. An early design of a solar powered zeolite-water adsorption cooling system have been investigated by Zeopower Company and Tchernev [87]. The system was using $0.7m^2$ of flat plate collector holding 36Kg zeolite. The maximum collector temperature of $121^{o}C$ and a solar COP in the range of 0.12-0.15 were reported. A space of volume $0.1m^3$ was cooled with production of 6.8Kg inaccessible ice. Further investigation on zeolite-water pair has been done by Grenier et.al [88]. The unit was a $12m^3$ food preservation using $0.8m^2$ of flat plate collectors with 23 kg zeolite 13X. An overall solar COP of 0.097 for the system was recorded.

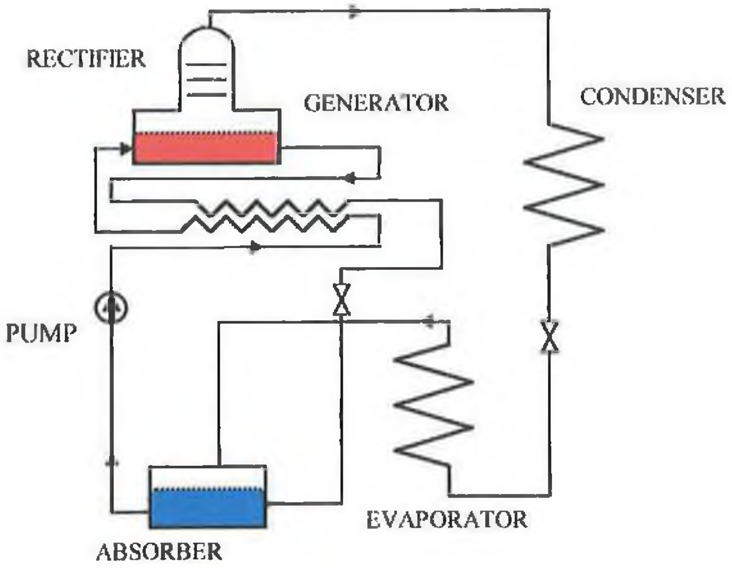
Also adsorption cycle studies on activated carbon- freon and zeolite- freon have been reported [89,90].

3.5 Absorption Cooling Systems

Absorption cooling systems operate based on the same principle as the adsorption cooling systems. Michael Faraday discovered that some vapours like ammonia can be sorbed by solids or liquids such as silver-chloride at low temperature and desorbed in high temperature. Therefore, the compression work would be reduced if refrigerant is dissolved in medium liquid before compression process and drawn off after compression. However, it must be noted that desorption is an endothermic process, and therefore, heat is required to draw off the absorbate, or refrigerant, from absorbent.

3.5.1 Continuous Pumped Absorption Cooling Systems

The continuous pumped cycle is the most commercial and popular design of absorption cooling systems. In this design the presence of a pump for creating the required pressure difference between the low pressure side and the high pressure side of the system is necessary. The schematic diagram of a continuous pumped refrigeration unit is shown in figure 3.8. The strong solution is pumped via the solution heat exchanger to the generator. By introducing heat to the absorbent-refrigerant solution in the generator (boiler) the refrigerant is desorbed. Depending on the refrigerant-absorbent pair, a rectifier column may or may not be required following the generator. The refrigerant vapour flows into the condenser and condenses. The liquid refrigerant passes through the expansion valve and partially evaporates. The complete evaporation occurs in the evaporator and causes the cooling effect. Finally, refrigerant vapour moves toward the absorber and it is absorbed by the incoming weak solution from the generator.



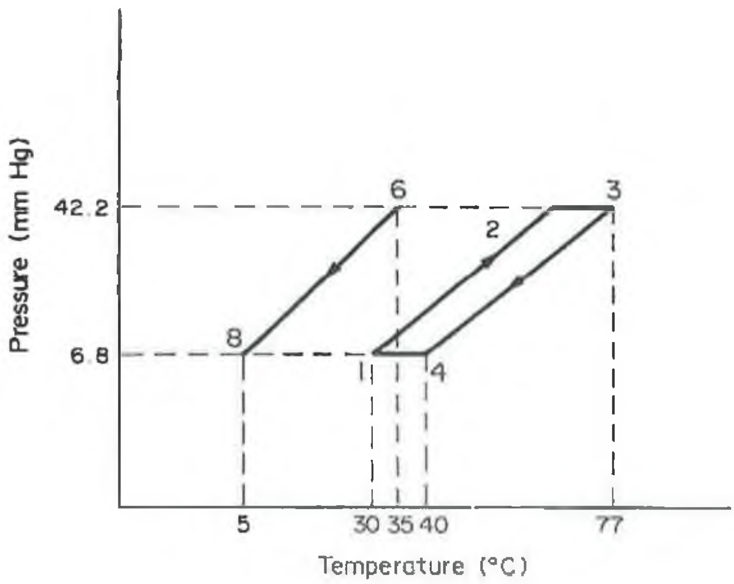


Fig. 3.8 Flow diagram and Pressure-Temperature diagram of the continuos pumped absorption system

The continuous pumped system can be designed to respond with more flexibility to expected variations in solar insolation. So as, it can continue to operate at declining temperature in the afternoon of a clear day. These systems are desirable for larger capacity requirements (50-1000kW). However these systems are associated with feed pump problems. This mechanical part is generally in hydraulic form and noisy with a relatively high maintenance expense and low performance. It also requires a parasite power for operation.

Selection of the absorbent-refrigerant pair depends on factors like application, generating temperature, efficiency, cost. Desirable characteristics for an absorbent-refrigerant pair can be listed as follows:

❶ Low viscosity to reduce pump work.

❷ Good chemical and thermal stability.

❷ Low operation and low pumping power

❹ Inertness, non-corrosiveness, non-toxic, non-flammable.

❺ A refrigerant more volatile than the absorbent in order to be separated from absorbent easily in the generator.

❻ A refrigerant that has a large latent heat so that the circulation rate can be kept at the minimum.

❷ Absence of solar phase absorbent.

Of common refrigerant-absorbent pairs, the pair NH_3-H_2O is being used for low temperature application and the pair H_2O-LiBr is being used for air conditioning. The evaluation and dependence of performance of these two systems on the solar heated generating temperature have been well investigated and studied by researches [91,92]. The H_2O-LiBr systems can be operated with lower generating temperatures (89°C to

93°C) compared to that of the NH_3-H_2O (121°C to 149°C). In the NH_3-H_2O absorption systems a rectifier column is required to prevent water vapour entering to the evaporator where it could freeze, while in the H_2O-LiBr systems there is no need of a rectifier column. In the H_2O-LiBr systems, because of refrigerant properties temperatures below 4°C can not be obtained. Also the H_2O-LiBr pair might suffer from crystallisation problems if not designed properly.

Kaushik conducted a comparative study between NH_3-H_2O and NH_3-$LiNO_3$ pairs [93]. It was resulted that the NH_3-$LiNO_3$ achieves higher COP than that of NH_3-H_2O in the both single and two stage systems, especially at higher generating temperatures.

Several solar assisted continuous pumped absorption systems have been developed by inventors. An early design of solar continuous absorption system was introduced by Chung et. al. (1953) using R21-TEG-dimethyl pair [94]. The overall cycle COP indicated by test results was in the range of 0.28-0.34. Later, Duffie and Sheridan (1963) adopted a commercially available technology H_2O-LiBr unit with solar energy [95]. The overall COP varied between 0.11 and 0.15 with the evaporator temperature in the range of 9°C to 13°C. The cooling effect of 0.056 ton/m^2 of collector was recorded.

More recently, Sloetjes (1988) presented a solar powered absorption unit [96]. The unit was capable to produce 13kW of cooling continuously for 5.5 hours of sunshine, or 3kW mean over 24 hours. The solution pump was driven by conventional power.

There are also a number of modified and special designs of absorption system that can be used to obtain lower temperatures and achieve higher COPs. These designs are referred to multistage and complex absorption cycles. Some of these cycles are listed as follows [93,97]:

1 - Multi stage absorption

In vapour compression cooling systems, to achieve very low temperatures, cascade systems, in which a series of refrigerants with progressively lower boiling points are coupled in a series of single stage units, are suggested [98]. Similar to vapour compression systems, single stage absorption systems can be cascaded. The most common design in this case is to couple two single stage absorption unit together which is known as two stage absorption cooling system. In this case, the first stage (or refrigeration unit) is used to provide the cooling effect for the absorber of the second stage. Hence, the evaporating temperature in the first stage is almost equal to the second stage absorber temperature ($T_{E1}=T_{A2}$). Figure 3.9 explains a typical two stage absorption cooling system. The refrigerant-absorbent pair can be different or same in stages. For applications which require temperatures above 0°C such as air conditioning use of the pair LiBr-H_2O in both stages is commonly recommended and for temperatures below 0°C dual fluid systems are recommended. H_2O- LiBr according to its higher coefficient of performance and modest generating temperature at about 80°C minimum which is attainable by flat-plate collectors is appropriate for first stage. To achieve of temperature below 0°C pairs like NH_3-H_2O and NH_3-$LiNO_3$ can be suggested for the second stage.

2 - Double effect generation absorption

Another design to increase COP of absorption cycle is to incorporate a second generator (effect) which is known as double effect generation. The first prototype unit was built in South West Research Institute in 1956-58. Figure 3.10 shows a schematic diagram of this design. The required heat for the second generator is supplied by condensation of the refrigerant vapour which is generated in the first generator. For condensing the refrigerant in the second generator, pressure must be increased, and subsequently, pressure and temperature of the first stage generator must be increased. Hence, in a double effect generation system, three different pressure zones as following are existed: i- The low pressure zone in the evaporator and the absorber. ii - The medium pressure zone dominating in the condenser and the second effect generator. iii - The high pressure zone in the first generator. To achieve the maximum COP, all the refrigerant vapour produced in the first stage generator must be condensed in the second stage generator, and the generated heat from condensation must match with the required heat for the second stage generator. Although this cycle achieves higher COPs compared to a single effect cycle, but it requires higher operating temperatures (<160°C) which can be only generated by concentrating and evacuated tube collectors.

Most of the efforts on development of this type of the cooler have been conducted by Yazaki Corporation in Japan, which marketed an H_2O-LiBr Chiller based on this cycle. Also a number of studies was conducted on different types of refrigerant-absorbent pairs. Philips argued that the NH_3-H_2O pair is not well suited to air-cooled double-effect operation [99]. Other pairs that have been studied are R21-dimethyl formamide, R22-dimethyl formamide, R22- dimethyl ether and a theoretical COP up to 1.82 has been obtained [100].

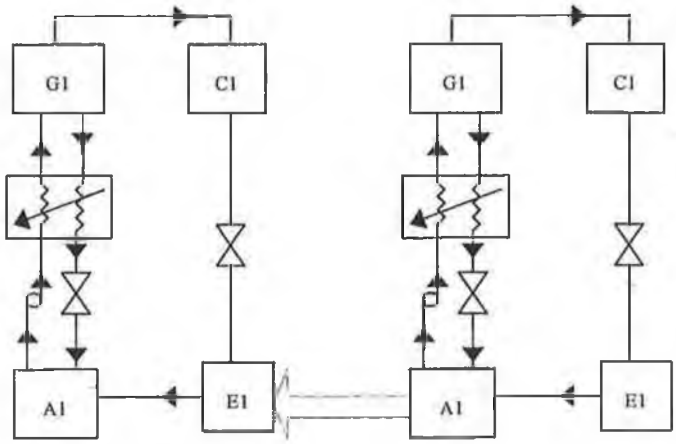


Fig. 3.9 Flow diagram of a two stage absorption cooling system

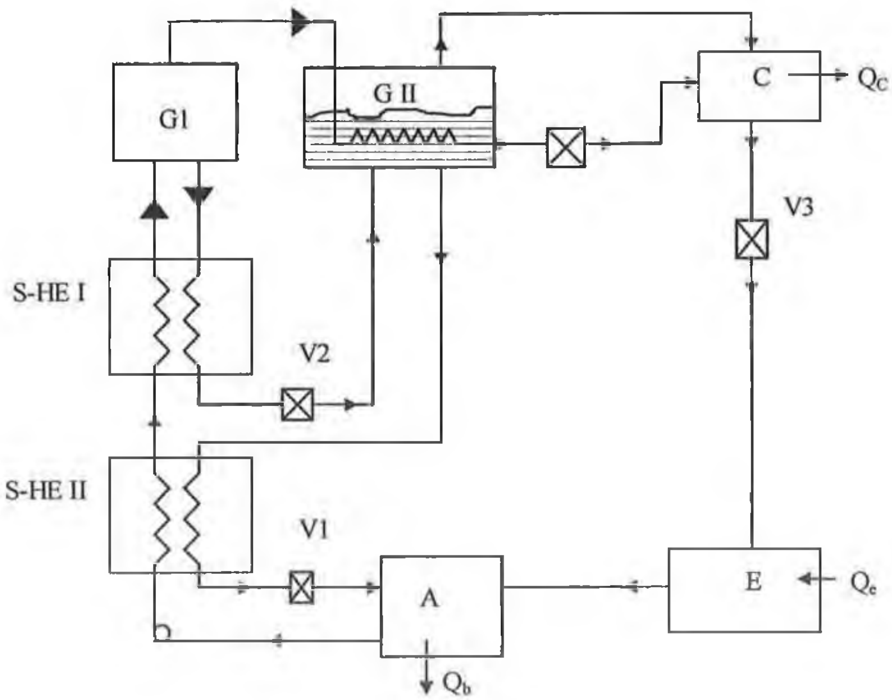


Fig. 3.10 Schematic of a double effect generation absorption cooling cycle

3 - Double effect absorption cooling system

Figure 3.11 illustrates the operation of the double effect absorption cooling system. In this design, two solution circuits known as primary circuit and secondary circuit are existed. The refrigerant vapour is generated in the primary circuit enters to the evaporator by way of the reflux condenser, condenser and expansion valve. The first cooling effect occurs in the evaporator. After leaving the evaporator, the refrigerant enters to the secondary solution circuit and is absorbed in the resorber. The solution boils in the desorber and produces the secondary cooling effect.

Despite the required higher operating temperatures some efforts have been done to adopt these systems with solar energy. Tanaka in 1977 proposed an absorption unit that works in double effect by using fuels at a higher COP and in single effect using solar energy so as to achieve an overall higher COP [101]. Two absorption unit based on Tanka design have been built. Figure 3.12 shows the schematic of this type of absorption chiller. When solar radiation is not high enough, plant can operate under double effect absorption system, with a cooling load of 20kW and COP of 1.1. As the solar energy is high enough, the plant is converted to solar powered single effect absorption, with the cooling load of 30KW and 0.6 in COP.

3.5.2 Intermittent Basic Absorption Cooling Systems

The operation of intermittent basic absorption system is similar to the adsorption cooling systems. Figure 3.13 describes the refrigeration cycle. The refrigeration cycle operates in two phase. Phase I includes pressurisation and desorption processes. As heat is applied

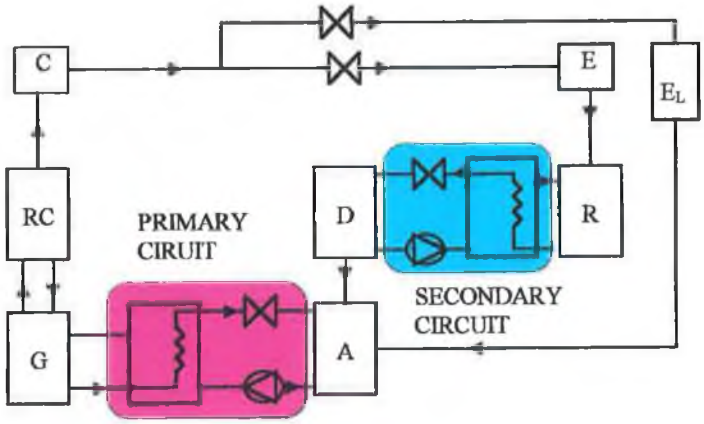


Fig. 3.11 Schematic of the double effect absorption cooling cycle

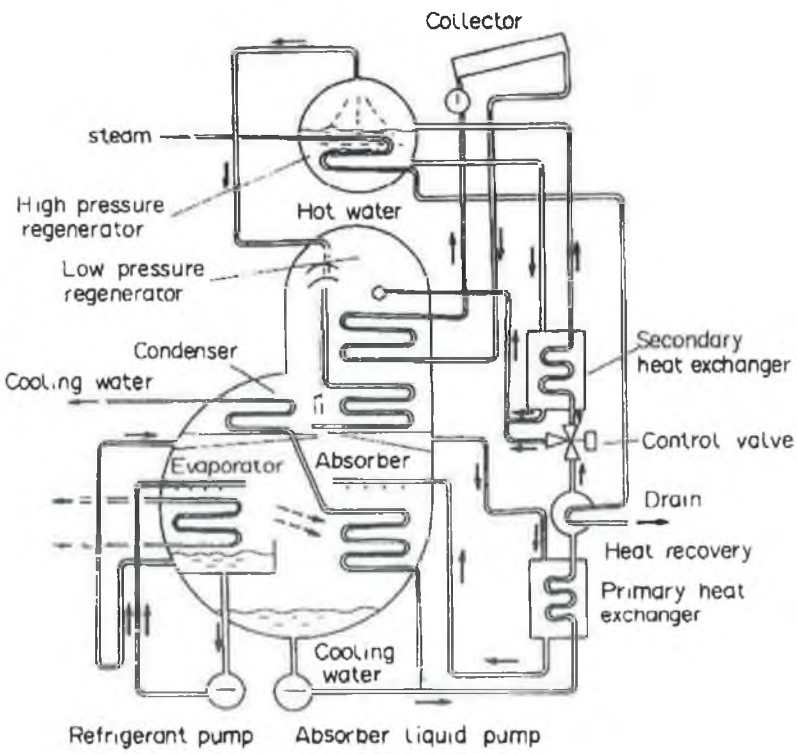


Fig. 3.12 Single-Double effect convertible absorption chiller of H_2O-LiBr type

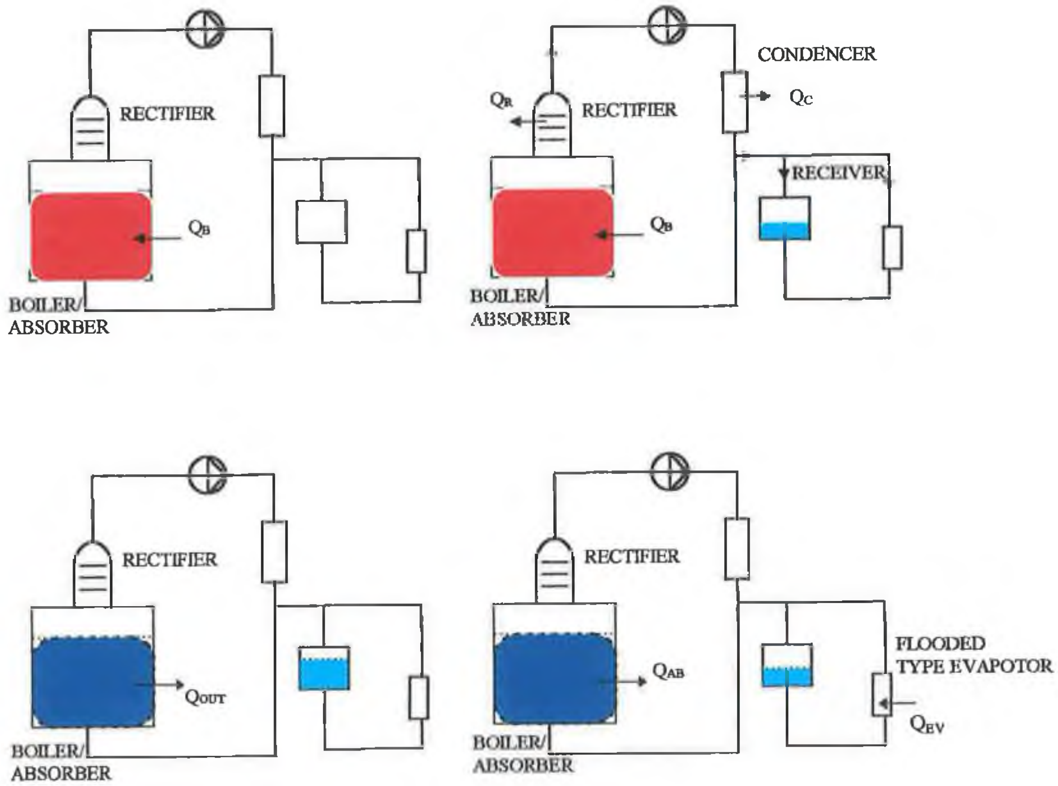


Fig. 3.13 Schematic of the intermittent basic absorption cycle

to the absorber, the pressure and the temperature of the system increase gradually (pressurisation). As more heat is applied some refrigerant is driven out(desorption). Depend on the absorbent-refrigerant pair presence of rectifier column is required or not. As the pressure rises a little from saturation pressure, the refrigerant vapour by contact to the condenser wall (which is in equilibrium with ambient) gets cold and condenses. Finally, the liquid refrigerant flows into the receiver. The check valve is adjusted to allow the vapour to flow from boiler to condenser with negligible pressure drop.

Phase II includes depressurisation and adsorption. This phase starts when solution in absorber rejects heat to ambient. During depressurisation process the solution concentration is assumed constant. However, depressurisation is associated with some sorption. Additional pressure drop induces the absorbent to absorb the surrounding absrobate vapour. As the pressure drops from the saturation pressure, some refrigerant liquid in evaporator vaporises. The heat of absorption heat is rejected to environment.

A convenient illustration of intermittent basic absorption cycle is the graph of temperature versus concentration, shown in figure 3.14.

These systems combine with some advantages including simplicity, absence of a mechanical or moving parts, low maintenance cost and low operating temperature (up to120°C). In addition liquid systems have superior heat transfer to solid adsorbent. However, their cyclic nature might be a disadvantage. All such solar systems have a diurnal cycle with heating during day and cooling during the night.

Number of studies on intermittent absorption systems have been conducted in AIT (Asian Institute of Technology). Exell (1978) introduced an aqua-ammonia solar powered refrigerator. The system was using an integrated collector/generator with the collecting area of 1.44m^2 [102]. An auxiliary plane mirror was applied to increase the intensity of

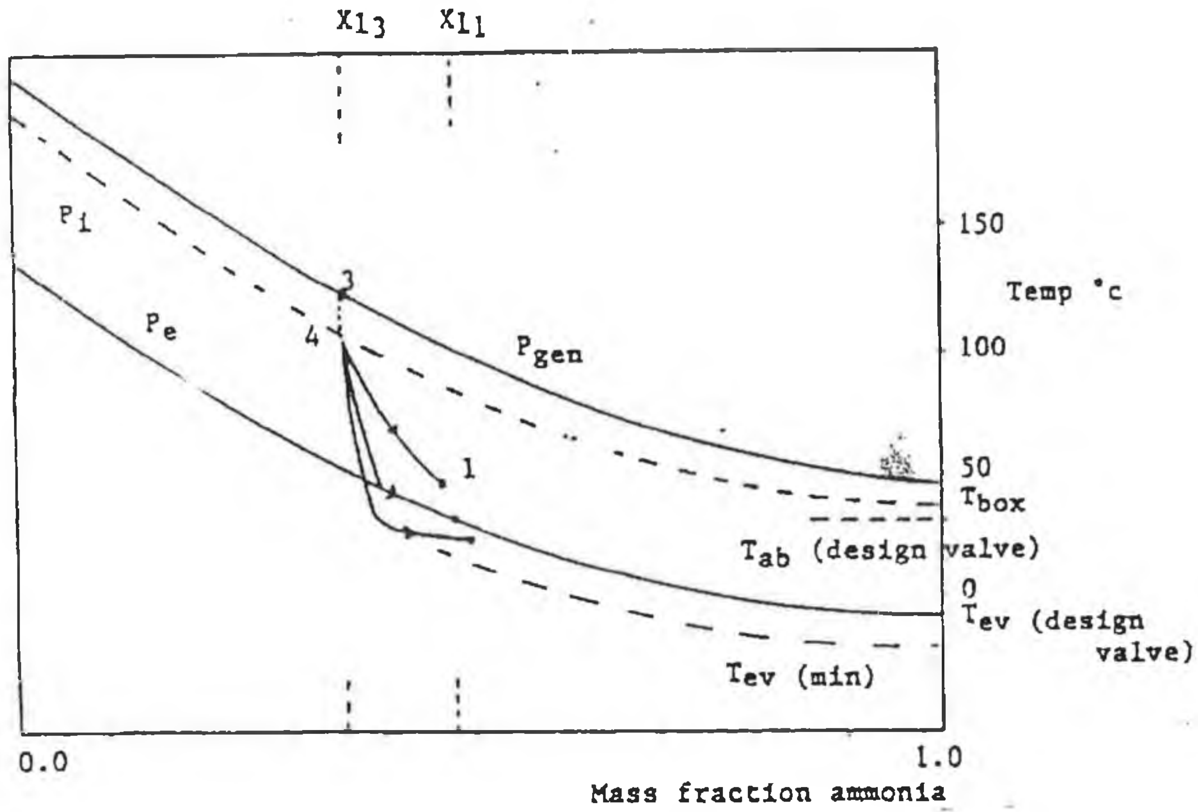


Fig. 3.14 (a) Phase I of the intermittent basic cycle (solution concentration verses temperature).

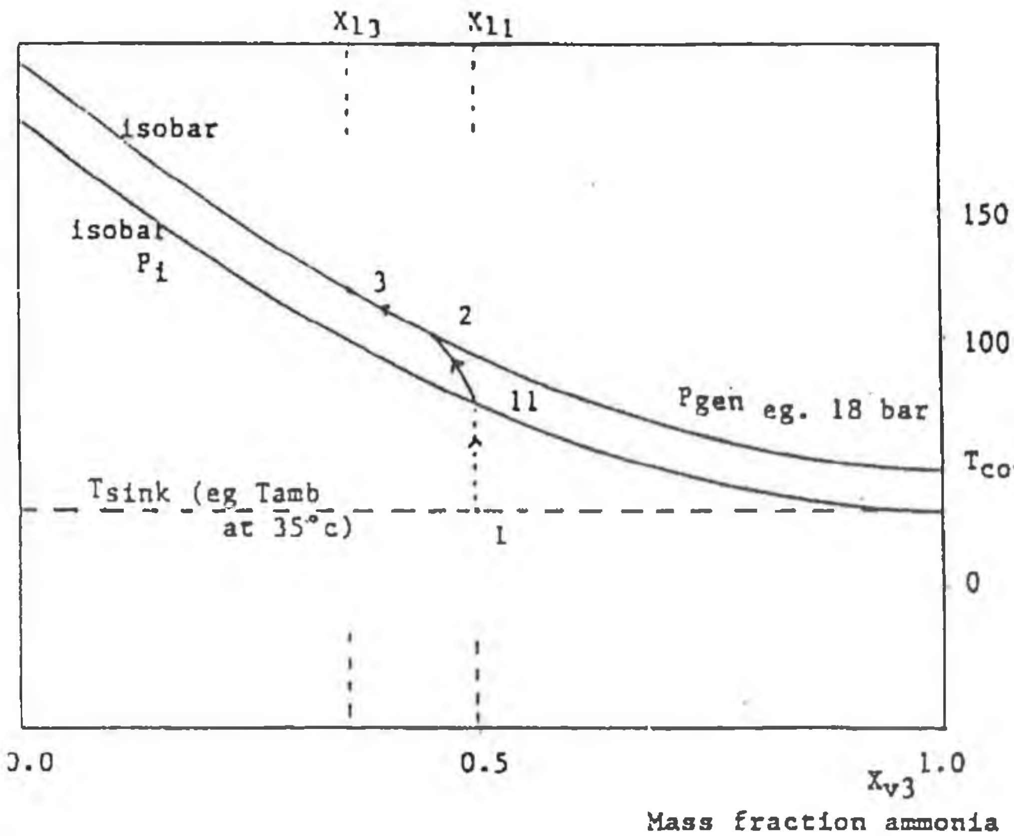


Fig. 3.14 (b) Phase II of intermittent basic cycle (solution concentration verses temperature).

solar radiation on the collector surface. In a typical sun day, the efficiency of 28% for the solar energy collection system was recorded. The refrigerator was charged with 19Kg solution, and generation started at 11am. A cycle COP of 0.26 and a solar COP of 0.07 were reported. Production of 4.2 Kg/m^2 ice as the cooling effect of was reported.

Later (1981), Exell introduced two prototype models, one with a tube-in-sheet collector like the early design, and the other one with non-focusing cylindrical concentrating reflectors mounted below the finned riser tube [103]. Both model had total collecting area of $5m^2$ incorporated with flat mirrors hinged to the east and west edge of collector. To provide sufficient solution for a full day's operation model II was fitted with a reservoir from which fresh solution was to be supplied to the collector during generation. Both system were charged with 65-67Kg of solution with concentration of 0.46. On a sunny day the first prototype model was capable to produce 28Kg ice from 28°C initial temperature. A COP in order of 0.14 was estimated for the $5m^2$ of collector area, or for the total collecting area a COP of 0.07 was reported. Since mirrors were less effective during the high scattered radiation and low direct insolation the performance of system was very sensitive to solar radiation pattern. Consequently, in a day with the same overall insolation, but greater diffuse component COP decreased from 0.14 to 0.09. The generation started at 10 am about 4 hours after sunshine. The performance of the second plant was in order of 1 to 2% lower in comparison with the first model.

Exell (1984) reported the design and construction of a village-size refrigerator [104]. The plant was able to produce 100Kg ice per day. In this design the reservoir tubes containing the solution were placed to either side of the solar window. The thermosyphon action causes the circulation of liquid within the collector. The collector comprised twelve flat plate panels with total $25m^2$ in area. The condenser was constructed of 66m of 0.5" stainless steel cooled by water bath. The evaporator was a gravity circulation type suitable for circulating of 51kg refrigerant during the night. The solar COP 0.11 of was reported.

Agrawal et. al. conducted a study on solar intermittent absorption system using freon 22 ($CHClF_2$)- dimethyl formamide (R22-DMF) pair [105]. The system was able to cool down 60Kg water from a temperature of 30°C to 15°C daily. They argued that the use of pair R22-DMF eliminates the associated problems with NH_3-H_2O (or H_2O-LiBr) such as corrosion, infallibility, high volatility and salting out.

3.5.3 Intermittent Regenerative Absorption Cooling Systems

The first intermittent regenerative absorption cooling cycle was developed by Trombe and Foex (1957). This design enjoys the advantages of a circulating solution in the generator, allowing reclaim of sensible heat, and simplicity of an intermittent system with requiring minimum intricate parts and no active solution pump [106]. A modified system based on the Trombe and Foex design is shown in figure 3.15. The intermittent regenerative absorption system is similar to the continuous pumped absorption system. The main difference is the replacement of the mechanical feed pump with a bubble pump. Similar to the intermittent basic absorption system absorption and desorption in this system are preceded by pressurisation and depressurisation, respectively. The operation of the system can be described as follows. The strong solution boils in Boiler tube/ Bubble pump. The associated reduction in bulk density creates a thermosyphonic pressure head which draws up the reach solution from the upper part of the reservoir. The generated vapour partially condenses in the rectifier. In the absence of a transfer tank separator vessel does the removal of solution mass during desorption phase. Use of a transfer tank assists to reduce the separator size, which in term, increases the performance of the system. Finally, the refrigerant vapour passes through the condenser and drops into the reservoir. The condenser is mounted in the reservoir vessel, as the condensation heat can be removed by the same cooling circuit for removal of absorption heat. During absorption phase refrigerant evaporates along the evaporator, and finally refrigerant vapour is re-absorbed in the lower part of absorber.

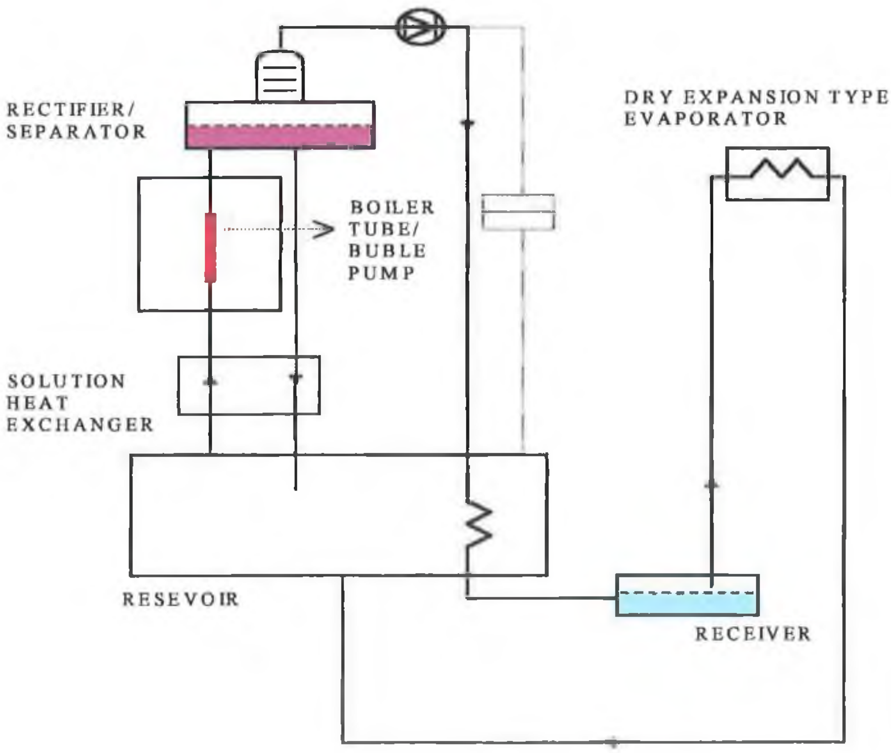


Fig. 3.15 Schematic of intermittent regenerative absorption system

These systems have advantages of a circulating solution in a generator, recovery of sensible heat, and simplicity, absence of mechanical pump, low maintenance. Intermittent nature and large absorbent-refrigerant solution even for low capacity can be mentioned as the most disadvantages of these systems.

Van Passen (1986) in university of Delft, Netherlands designed and constructed an intermittent regenerative system [107]. The system was coupled with flat plate collector/generator with area of 1.63m^2. A cooling capacity of 2.6MJ against the solar COP of 0.1 was recorded.

3.5.4 Diffusion Absorption (Platen-Munters) Cooling Systems

In 1922 Swedish engineering students, Baltzar von Platen and Carl Munters, from the Royal Institute of Technology in Stockholm, presented, an invention, a refrigeration system using a new and brilliant application of the absorption process. The cycle is known as diffusion absorption (or Platen-Munters) cycle. In diffusion absorption systems the pressure difference between the low pressure side and the high pressure side is provided by assistance of an inert gas in the system. Although the total pressure in the system is constant, the partial pressure of the refrigerant and the inert gas varies along the system.

Figure 3.16 shows the schematic diagram of a Platen-Munters cycle. The strong solution boils in generator and rises to the rectifier (or separator), water returns to the absorber, and ammonia moves toward the condenser. In the condenser, ammonia rejects its latent heat to the surrounding and condenses. Then the liquid ammonia flows through a liquid trap to evaporator. In the evaporator ammonia corresponding to its low partial pressure evaporates as the inert gas (hydrogen) flows over the refrigerant, ammonia. As ammonia evaporates, its partial pressure along the evaporator increases gently, and therefore evaporation temperature rises gradually. The mixture of hydrogen and ammonia flows down through a heat exchanger to the absorber, and the ammonia vapour is reabsorbed into the cold water solution. From the absorber hydrogen corresponding to its lower density rises back to the evaporator, while the ammonia-in-water solution flows into the generator to complete the cycle.

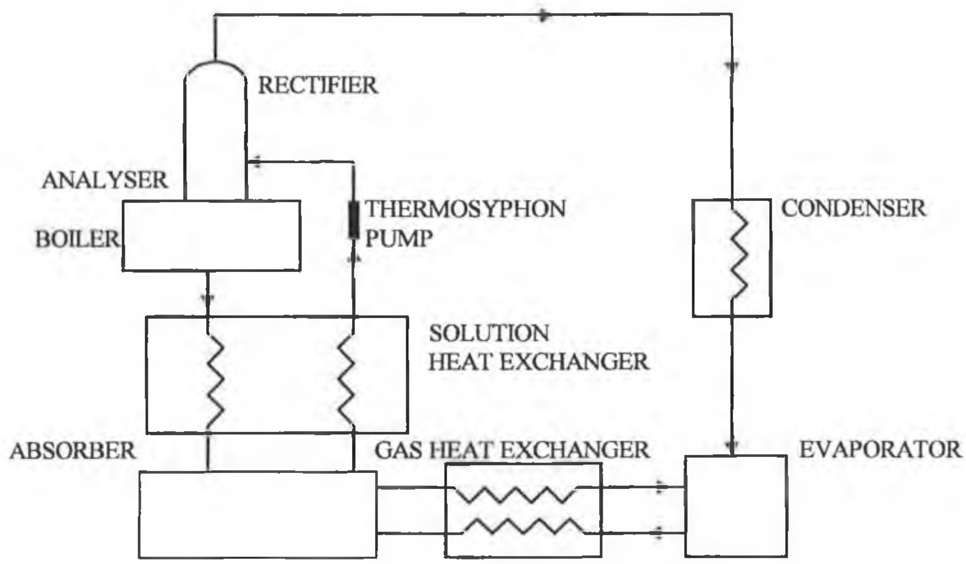


Fig. 3.16 The schematic diagram of a diffusion absorption refrigeration system

According to Harvey diffusion absorption cooling is inherently less efficient due to greater pressure drops experienced in the gas transport process and undesirable heat flow from the absorber to the evaporator due to gas circulation [108]. However, when power consumption is low the efficiency is not the prime consideration.

Cullimore introduced a modified solar powered diffusion absorption cooling system based on Electrolux refrigerator [109]. The normal COP of the cycle was 0.23. For decreasing of generating temperature from 186°C to 130°C, solution concentration was increased from 35% to 55% and total pressure reduced from 26 bar to 19 bar. The maximum (electrical) COP that could be obtained under any conditions was 0.13 if all cooling below ambient is accounted, and 0.05 if only cooling below 0°C was accounted. The system was incorporated with solar collectors, using evacuated tube collectors with area of $3m^2$. An oil circuit thermosyphone was transferring heat from collector to the generator. Cooling period of about five hours was recorded. The solar COP is not recorded but it is expected to be below 0.01.

Hinotani and Sanyo Company (1984) investigated and reported an experimental solar Platen-Munters refrigerator [110]. Type of collector for this purpose comprised of evacuated tube collectors enclosing with parabolic concentrating mirror with concentrating ratio of 1.5. The generator was incorporated with a bubble lift pump. Evaporator temperature less than -15°C during most of the cycle was recorded which can be assumed as the temperature at the end of evaporator and a temperature of 0°C for cooling box was recorded.

Gutierrez (1988) in Mexico reported an effort on conversion of conventional diffusion absorption to solar powered [111]. The selected refrigerator system was a 250 litres capacity, which was using gas burner to produce generating temperature of 160°C. The burner was replaced with 2.5 m^2 of flat plate solar collectors. To reduce the boiling temperature to 105°C the solution concentration increased from 32% to 42%. Since the new circulation ratio (weak solution circulated per unit mass refrigerant generated) was expected to be 5 times larger than original, therefore, the solution heat exchanger was replaced with a bigger version. A number of tests under different controlled laboratory conditions were conducted.

A new solar powered refrigeration system, based on Platen-Munters, has been developed as part of the research programme presented here. This development is described in Chapter 5 and the outstanding results have been presented in the Congress of the World

Renewable Energy Network and published by Pergamon Press.

Chapter 4

Review of Solar Desalination Techniques

Chapter 4

Review of Solar Desalination Techniques

4.1 Introduction

Desalination is the process used to supply fresh water from seawater or brackish water. Drinking water contains dissolved solids in the range of 0.05% to 0.15% dissolved solids. The average concentration of salt in seawater is 3.44% [112]. However, due to climate and seasonal conditions and geographical situation, this concentration has large variations.

In the desalination treatment, energy is required to separate water and salt. Only part of input energy is used for separation of water and salt, and so, the efficiency the process is limited.

In general, desalination technologies fall in two main categories. The first category is referred to desalination processes in which heat is applied to vaporise water. In these processes different distillation methods are used for purification of water. The efficiency of distillation processes is limited by Carnot efficiency ($\eta=1-T_L/T_H$). However, in practice the actual efficiency is considerably lower than Carnot efficiency. Second category refers to non-thermal desalination processes, which do not use distillation methods to provide fresh water. There are several methods to separate impurities from water. These methods are reverse osmosis, electrodialysis and freezing. Non-thermal desalination processes are suitable for production of potable water from brackish water, as cost effectiveness of these systems decrease with increase of water salinity.

Although the technical process for desalting water has been known for a long time, but the process was too expensive. One of the continuing needs for desalted water was on ocean-going ships. Aside from carrying stores of water, some attempts were made to distil the water with the heat of ship's cook stove or engine. This technique was use in 19th century and it was partially successful since the expense involved to produce potable water was relatively high. Another step in development of destination plants were obtained in 1940s during the second world war when various military establishments in arid areas to supply their troops with fresh water. After the war, due to industrialisation and population growth, the demand of fresh water has been growing steadily. Nowadays, water scarcity has become a reality especially in countries in arid and semi-arid regions of the world. The status of fresh water demand and supply has been discussed in literature [113]. Therefore, because of the limited natural resources of fresh water, desalination processes have become more interested and many R&D programmes have been conducted.

Large and medium size scale desalination plants are operated with fossil fuels, electricity and some times nuclear energy [114]. Although large scale desalination technologies are fairly developed and many plants are now operating, but more than half of the erected plants are installed in Middle-East countries, because of their rich oil resources.

Because distillation processes require medium-grade heat, solar energy can be regarded as a potential thermal energy source, particularly in small and medium size desalination plants. Small-scale solar powered desalination systems are commonly suggested for household demands. The application of small solar powered desalination plants is mainly concerned for small communities in the remote regions of the world. For the larger demands of fresh water hybrid systems are suggested [115]. Solar desalination has also been investigated and considered for supply of the fresh water in the developed countries, especially southern European countries [116].

4.2 Distillation Methods

Over 60% of the world desalting water is produced by distilling fresh water from the sea [117]. Distillation is a process in which heat is first applied to vaporise a liquid and then heat is extracted to condense the vapour. Distillation is a method to separated two or more dissolved substances with different evaporation temperature from a solution. A typical distillation process consist of three discrete steps:

1. Vapour formation
2. Removal of the vapour from liquid surface
3. Condensation of vapour

Distillation can be achieved via the different methods. The most extensive used desalination methods are presented as follows [118]:

1- Single effect distillation

In the single effect distillation the seawater is distilled only once to produce fresh water.

2- Multiple effect distillation

A multiple effect distillation plant consists of several evaporators operating at different pressures. Figure 4.1 shows the flow diagram of such a system. The incoming water partially evaporates in the first effect (vessel) and moves toward the second evaporator. The generated vapour releases its latent heat and condenses. This heat is utilised for additional evaporation in the second effect, at a lower pressure and temperature. Hence, further evaporation occurs in successive effect, each at lower temperature and pressure. It must be noted that the seawater feed to undergo multiple boiling without supplying the additional heat after the first effect. Presence of a pump is necessary to deliver the fresh water at atmospheric pressure. Also sort of pumping system is required to exhaust the steam space of evaporator.

This process was used in some of the early designs of desalination plants. It was later replaced by multi- stage flash (MSF) units, discussed below, due to cost factor and their higher efficiencies.

3- Flash distillation

In this process seawater is first preheated and subsequently it is exposed to a pressure lower than its vapour pressure. The vapour flashes off from the warm water. Figure 4.2 shows the flow diagram of a multi-stage flash desalination plant. As shown, seawater is heated in a vessel called brine heater. Then the heated water is directed to a chamber known as the first stage. The sudden introduction of the heated water into the chamber causes it to flash rapidly. The phase change from liquid to vapour incorporates with a temperature drop in brine, which reduces the process efficiency. This problem can be solved with introducing multiple stages. Typically, a multiple effect flash desalination can contain from 4 to about 40 stages.

4- Vapour compression desalination

In this method saline water is first boiled in one side of a heat transfer barrier. The generated vapour is then compressed. The compression can be done either by a mechanical compressor or a steam jet. The compressed steam is returned to the outer side of heat transfer barrier and condenses. The latent heat resulted from condensation is used to vaporise more water in the tube. The distil is removed through a heat exchanger to the storage. Figure 4.3 shows the schematic diagram of a vapour compression desalination unit.

5- Membrane distillation

It is possible to concentrate aqueous solutions of non-volatile dissolved substances by microporous membrane impermeable for water but permeable for water vapour. As the name implies, the membrane distillation process combines both the use of distillation and membranes. In this process, the saline water is warmed and evaporates. The generated vapour is exposed to a membrane that can pass vapour but not water. After vapour passes through the membrane, it cools down and condenses. The water can not pass through the membrane and it is trapped and can be collected as the output of the plant. Figures 4.4 a and b illustrate the operation of the membrane desalination unit. The main advantages of membrane distillation are its simplicity and the need for a small temperature differential for operation.

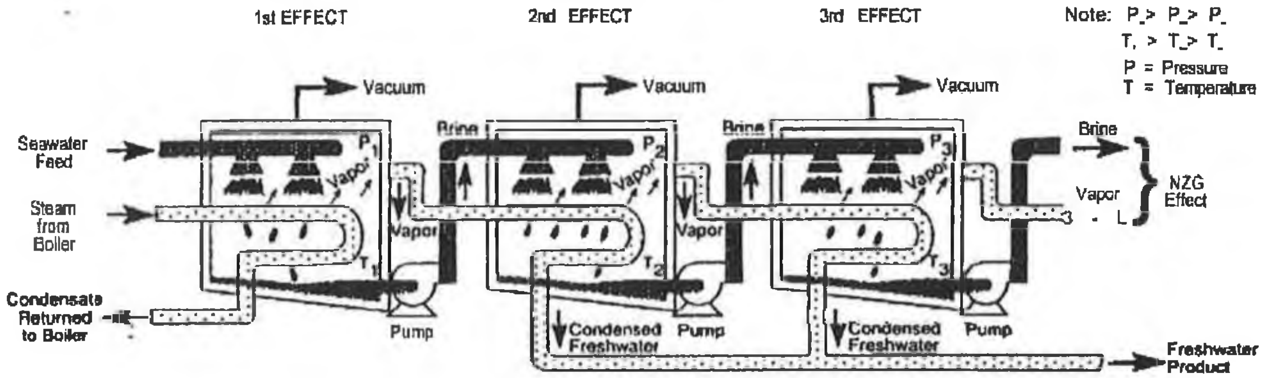


Fig. 4.1 Flow diagram of a multiple effect desalination plant

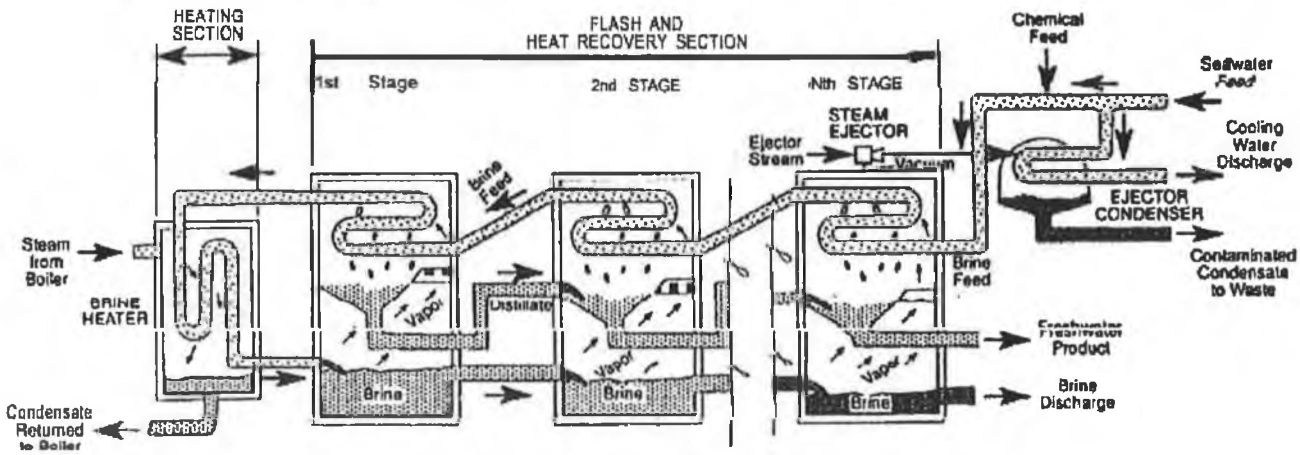


Fig. 4.2 Flow diagram of a multi-stage flash desalination plant

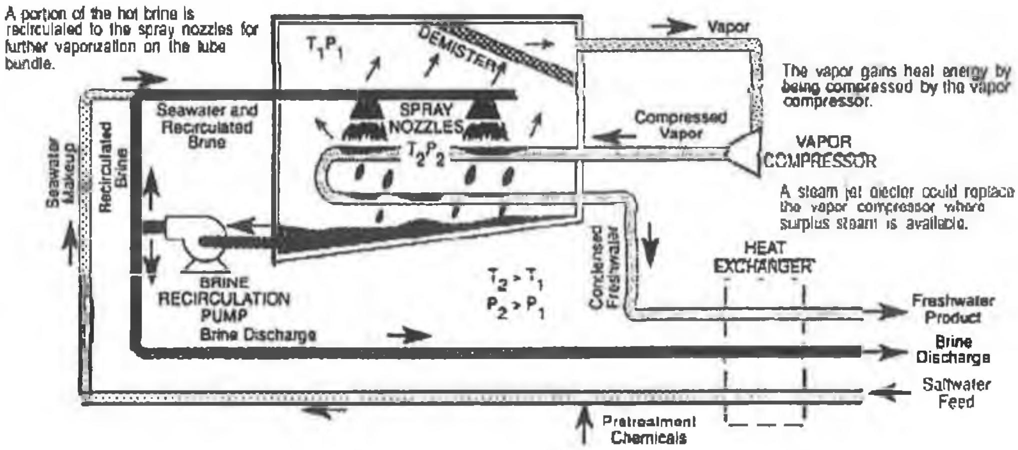


Fig. 4.3 Schematic of a mechanical vapour compression desalination plant

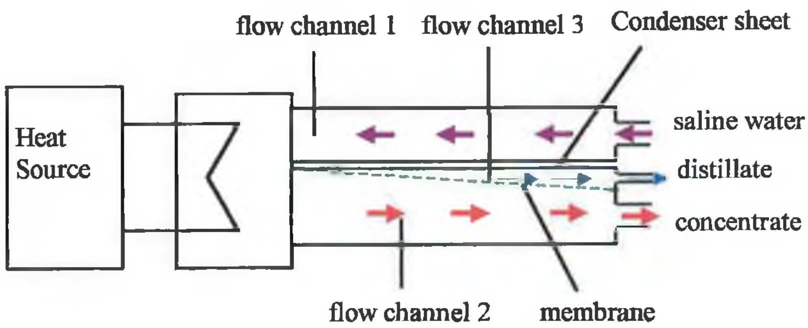


Fig. 4.4.a Schematic of a membrane distillation module

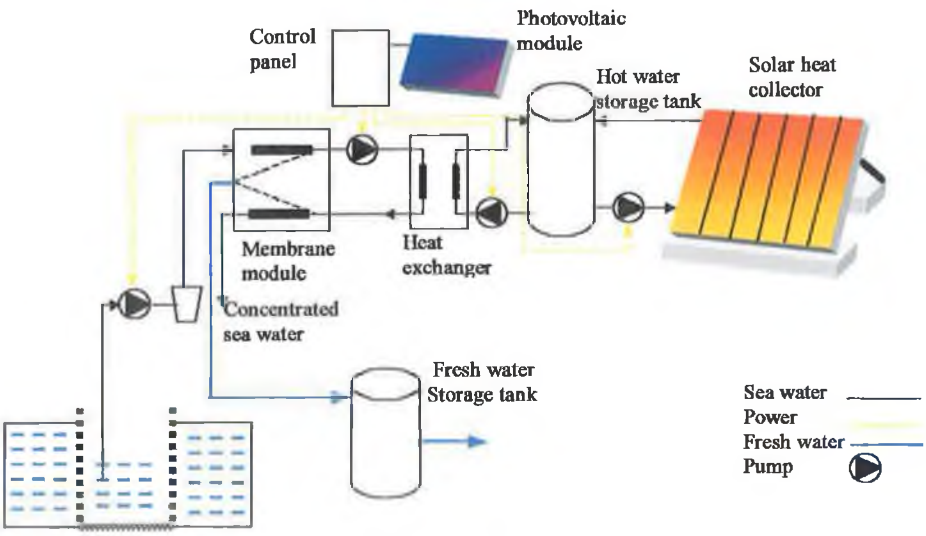


Fig. 4.4.b Schematic diagram of a typical membrane desalination plant

There are also some special designs, which are actually special designs of the above-mentioned distillation systems. Figure 4.5 shows the flow diagram of a desalination system, namely vertical tube falling film, proposed by Aqua-Chem [119].

4.3 Non-Thermal Desalination Techniques

Non-thermal desalination methods are referred to processes which do not utilise thermal energy to vaporise the saline water. The main application of these systems is in some industrial application in which demineralisation is used to produce ultra pure product.

The main non-thermal desalination processes are reverse osmosis, electrodialysis and freezing. As pointed out these processes are commonly suited for desalting of low to medium salt concentration, and therefore, they are not suitable for seawater desalination.

4.3.1 Electrodialysis

The electrodialysis process is primarily used for treatment of water with salinity between 500 to 1000 mg/litre. The electrolysis process is based on the following principles:

- When a salt is dissolved in water, it breaks up into ions, which are positively (cationic) or negatively (anionic) charged. When a potential gradient is applied between two electrodes, ions are forced to migrate between the electrodes.
- Semi-permeable membranes can be deployed alternatively to permit selective passage of either anions or cations.

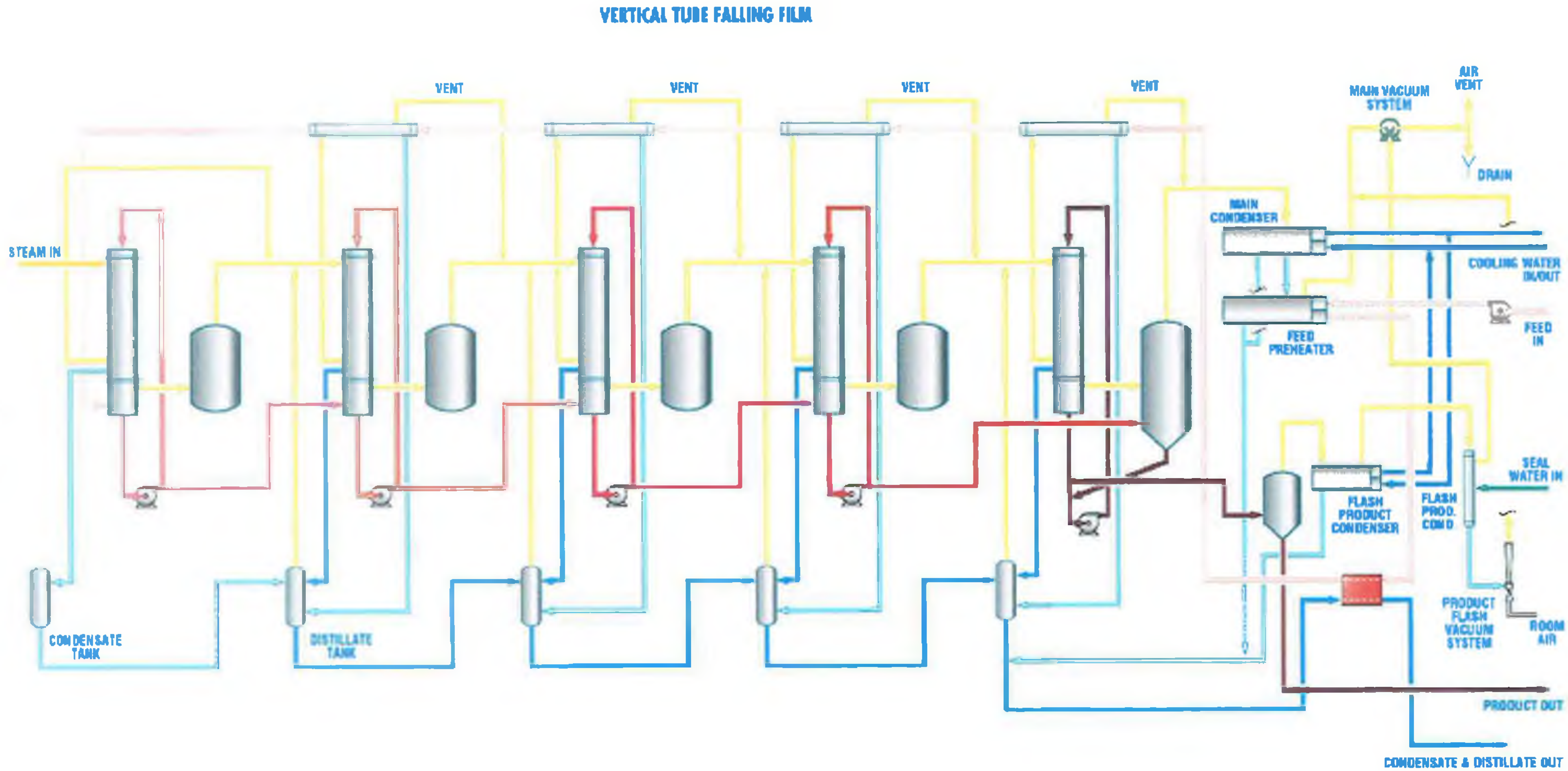


Fig. 4.5 Flow diagram of the vertical tube falling film desalination unit designed by Aqua-Chem Co.

The early electrodialysis units consisted of three-compartment electrolyte cells in which the central compartment was defined by membranes, which were not significantly ion selective [120]. In 1940, Meyer and Strauss suggested a multi-stack electrodialysis using ion-selective membranes [121]. The desalination effect is achieved by the actions of series of cells separated by alternating pairs of cationic and anionic membranes. These stacks can be in turn connected hydraulically in series or parallel [122]. The schematic of a multi-stack electrodialysis desalination unit with ion-selective membranes is illustrated in figure 4.6.

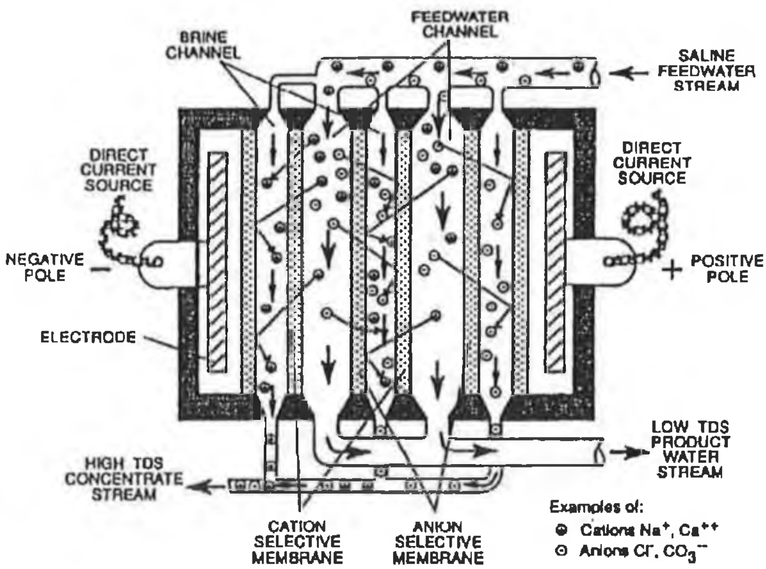


Fig. 4.6 Movement of ions in the electrodialysis process

The early ion-selective membranes were not commercially viable, and also they were not mechanically strong or chemically stable. The other main problem was that membranes with high ion-selectivity had also had high electrical resistance and those which had low resistance also had low selectivity. Ten years after introducing the first ion-selective membranes, Juda and McRac introduced ion-selective membranes, which did not have the associated problems with early ion-selective membranes [123].

Modern cation-selective membranes typically consist of polystyrene having negatively charged sulphonate groups chemically bonded to most of the phenyl groups in the polystyrene. The negative charges of the sulphonate groups are electrically balanced by positive charged counter-ions. Anion selective membranes typically consist of insolubilised polystyrene having positively charged quaternary ammonium groups chemically bonded to the most of the phenyl groups in the polystrene. In this case, the counter-ions are negatively charged and are the principal carriers of the electric power [124].

Figure 4.7 illustrates the basic component of an electrodialysis unit. The raw feed water must be pre-treated to prevent materials that could harm the membranes or clog the narrow channels in the cells from entering the membrane stack. The feed water is circulated through the stack with low pressure pump with enough power to overcome the resistance of the water as it passes through the narrow passage. The post treatment process comprises stabilising water and preparing it for distribution.

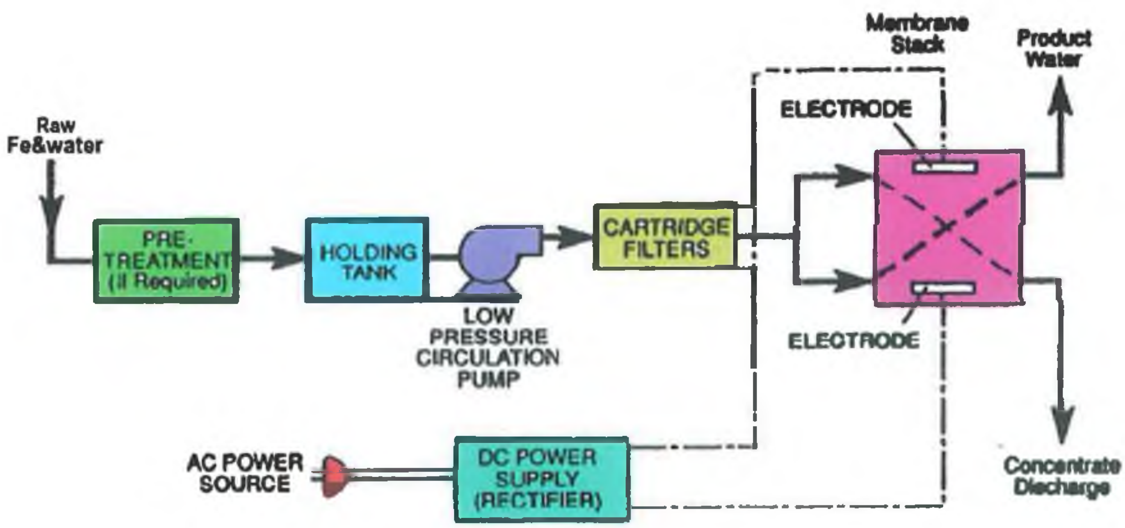


Fig. 4.7 Basic components of an electrodialysis plant

4.3.2 Reverse Osmosis

Reverse osmosis is the process where salts are separated from water by use of hydrostatic pressure forcing the water through a semi-permeable membrane, named permeator, to separate the feed water from salt. As shown in figure 4.8, the basic components of a reverse osmsis process are: 1- Pretreatment, 2- Pressurisation pump, 3- Membrane assembly and 4- Post-treatment.

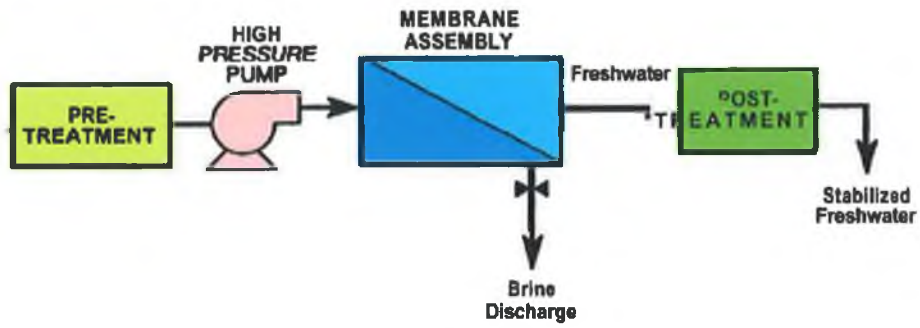


Fig. 4.8 Basic components of a reverse osmosis plant

In comparison to distillation and electrodialysis, reverse osmosis is relatively new, and its technology dates back in 1952 in the University of Florida, when two researchers demonstrated the desalination properties of cellulose acetate membranes [125]. Following a R&D programme in the mid 1960's reverse osmosis devices were commercialised with introduction of spiral configuration and hollow tubes, figure 4.9. Since then capacities of reverse osmosis plants have been rapidly increased, with a continuous decrease in cost of fresh water produced [126].

An intensive study to evaluate different solar desalination processes has been carried out [127]. The result of the study indicated that the RO desalination plant operated by PV, was superior to the others including: Solar still, ME evaporation plant, MF plant and electrodialysis system.

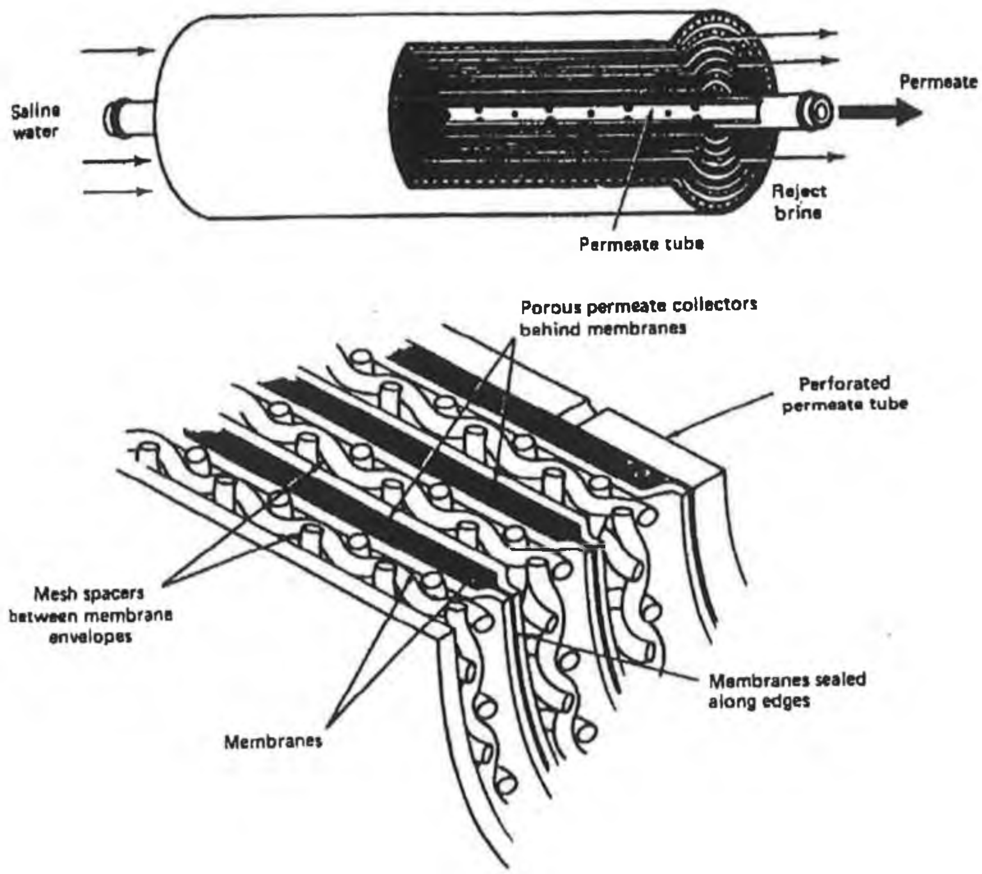


Fig. 4.9 Spiral wound membrane element construction

4.3.3 Freezing

In the freezing desalination process, dissolved salts are naturally exclude during the formation of ice crystals. Seawater can be desalinated by cooling water to form crystals under controlled conditions. Before the entire mass of water is frozen, the mixture is washed and rinsed to remove the salts in the remaining water or adhering to the ice crystal [128].

A small number of plants have been built over the past 40 years, but the process has not been commercially successful. The largest plant using the freezing process for the production of potable water is located in Wrightville Beach in the USA with productivity of 200,000 gallon/day [129]. Another example of a freezing desalting plant was an experimental solar powered unit constructed in Saudi-Arabia [18]. The schematic of a freezing plant is shown in figure 4.10.

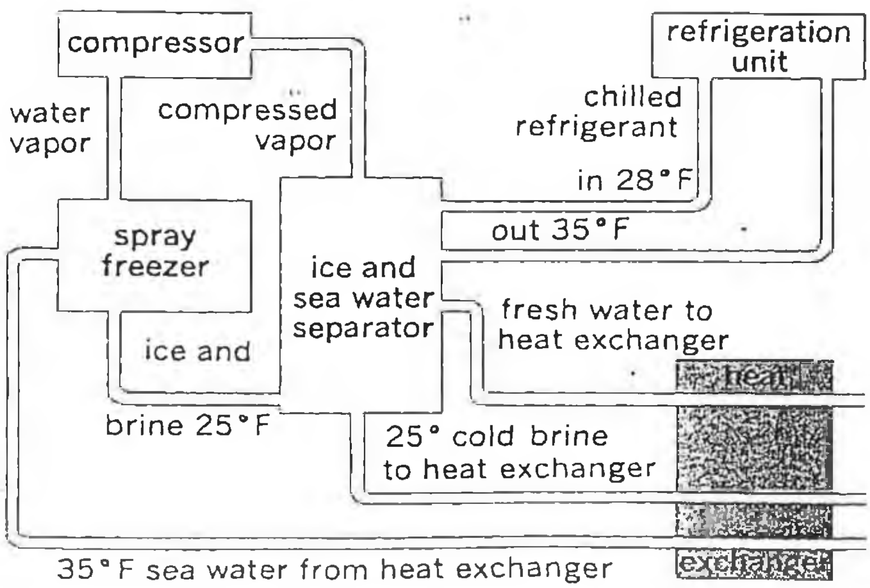


Fig. 4.10 Freezing-evaporation process

4.4 Solar Distillation Technologies

The simplest and cheapest type of solar desalination system is the single effect basin type of solar still. The history of solar stills dates back at least to 1872 when a plant, with 4747m^2 and capacity of 19m^3/day, was installed in Chile [130]. Since then various designs and modifications has been carried by inventors. A typical solar still consists of a glass cover, metal pan, insulation, enclosure and distillation trough. Figures 4.11 a and b present single and double slope solar still, respectively.

(a) (b)

Fig. 4.11 Schematics of basin type solar still: (a) single slope, (b) double slope

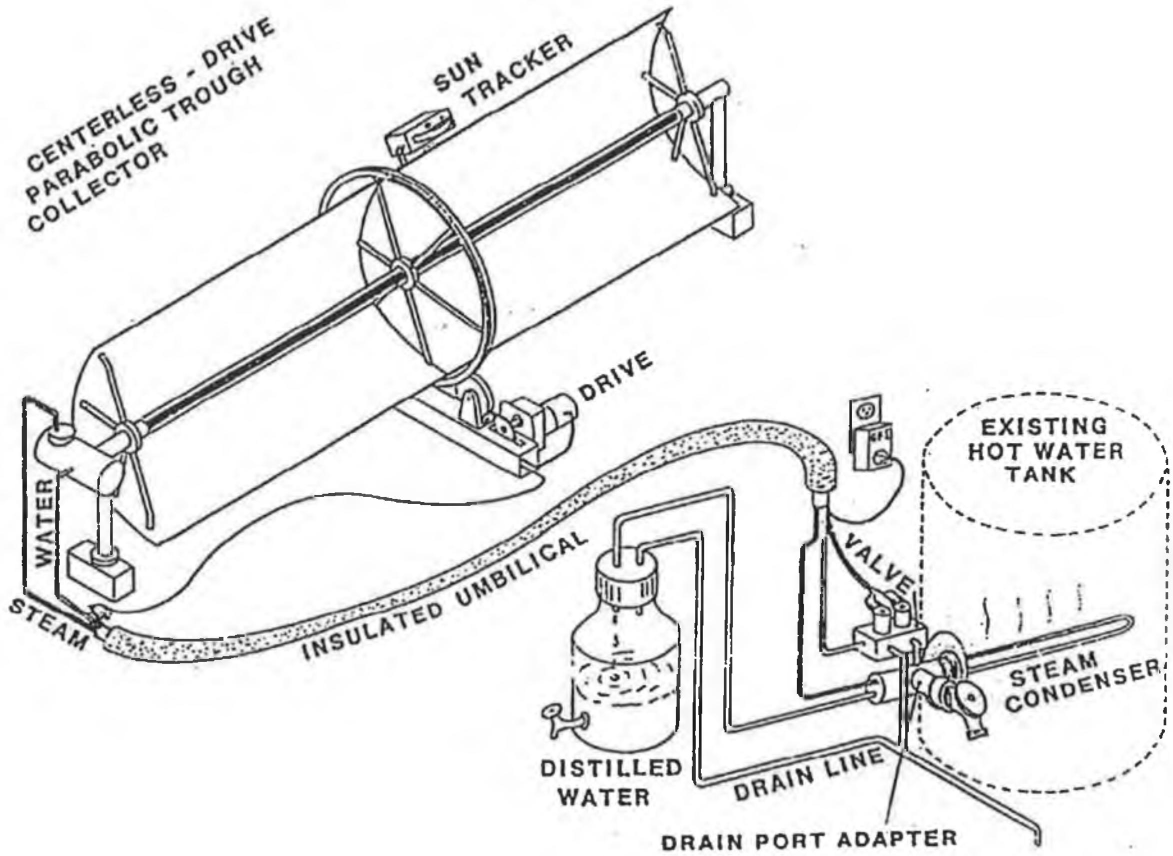
The operation of basin type solar still is based on the "green house" effect. The solar radiation falls on the glass cover. Most proportion of solar radiation is transmitted through the glass cover and rest of it is reflected or absorbed. The transmitted radiation strikes the water surface. Portion of this radiation is reflected and remainder is absorbed via the water mass and metal pan. Most of the absorbed radiation by metal pan is transferred to the water mass and rest of it is lost to the atmosphere through the insulation material. The water warms up and evaporates. The generated vapour by contacting to the glass cover gets cold and condenses in the inner surface of the glass cover and trickles down under its gravity.

The efficiency of the single effect basin type of solar still is relatively low, mainly due to the rejection of heat to the atmosphere. The main operating variables effecting the performance of a solar are expressed as insolation intensity, ambient temperature, brine depth, slope of cover, vapour tightness of the unit, heat losses through the base and the wind speed [131].

To increase the efficiency and productivity of conventional basin type solar steel a number of modifications and different designs of basin type solar steels have been presented by inventors [132,133,134]. Two configurations of solar still, conical and pyramid types has been built and tested in university of Agriculture Fasialabad in Pakistan [135]. The conical solar still was comprised of a blackened circular container enclosed by a transparent conical shaped plastic cover. The water container of the pyramid type solar still was squared shaped covered with pyramid shaped glass cover. The pyramid solar still was capable to produce 161.47 ml/m^2/day and 469.48 ml/m^2/day in winter and summer, respectively. The average productivity of conical type solar still was 200.98 ml/m^2/day for winter and 569.61 ml/m^2/day for summer. More recently, an inverted absorber solar still were introduced in co-operation between the University of Ulster, Northern Ireland, and the L.N. Mithila University, India [136]. The solar energy collection systems comprised of an inclined asymmetric line axis parabolic concentrator, with a concentration ratio of 1.5 and an acceptance angle of 120^o, integrated with a secondary reflector. The distillate collected from this system during a six-hour period was 350militres. Fath proposed and conducted a theoretical study on a double effect basin type solar still, in which the released latent heat from the first stage is used for further evaporation in the second stage [137]. The distillation unit consisted of a second effect still connected to a single sloped solar still incorporated with a shutter type reflector. On the base of numerical analysis it was found that the unit was capable to produce 10.7 kg/m^2 per day of water in the real life cycle during the summer.

Despite all the efforts, the efficiency of the basin type solar stills ranges between 45% and 55% and they need constant attention to operate properly. These operational problems have been discussed in details in literature [138].

The parabolic type solar still is an alternative design to basin type solar still. Basically, a simple parabolic solar still consists of a parabolic trough incorporated with an absorber tube and a condensation section. The solar radiation is concentrated over absorber tube and converted to heat. The temperature of saline water by passing through the absorber tubes increases and water partially evaporates. Finally, the vapour condenses in the condensation section, which might be a heat exchanger or a condensation-separation unit. This type of solar still has been marketed by BSAR SOLAR Company [139]. Figure 4.12 illustrates the desalination unit with its operating specifications. More investigation on this type of solar still was carried out in Mu'tah university in Jordan [140]. A maximum productivity of 0.17kg/m^2hr was reported and the plant efficiency was found experimentally to be 12.7% in average. The collector efficiency was calculated in order of 22.3%. The flow diagram and configuration of the plant are depicted in figure 4.13.



SOLAR STILL - HOT WATER SYSTEM

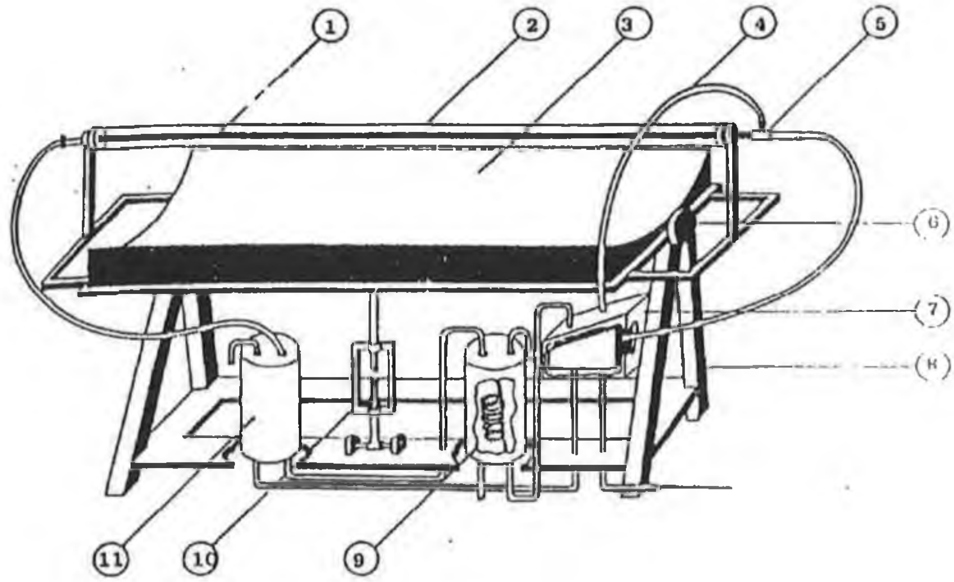
SYSTEM SPECIFICATIONS:

Size	30 in x 16 ft -Collector
Weight	135 pounds -Collector
Area	37 Square Feet
Concentration ...	12 Suns (12:1)
Rim Angle	95 Degrees
Temperature	38-124'C (100-254'F)
Pressure	Up to 32 PSI
Efficiency	40% at 124'C
Reflector	Aluminized Acrylic on Laminated Wood
Receiver	3/4 inch Copper Pipe
Glazing	Pyrex Glass 35 mm OD , 2 mm Wall
Heat Exchanger ..	Single Wall Steam Condenser
Tracking	Patented Single Axis Centerless Drive- 1/80 HP Induction motor, 110 VAC or 12 DC.
Controls	CdS Sun Sensor and Thermistor Temp. Sensors with CMOS IC Digital Controller, 12 VDC
Circulation	Steam Flow
Boiler Blow Down ...	Solenoid controlled Boiler Drain Down Each Night
Boiler Level	Float Switch controlled Feedwater Solenoid
Freeze Protection ..	Thermostat Controlled Feedwater Trace Heating And Boiler Drain Down
System Life	20 Years (Estimated)
Space Heating	Optional Thermostat Controlled Steam Condensing Fan Coil Units
Installation	Do It Yourself (1 Man Day)

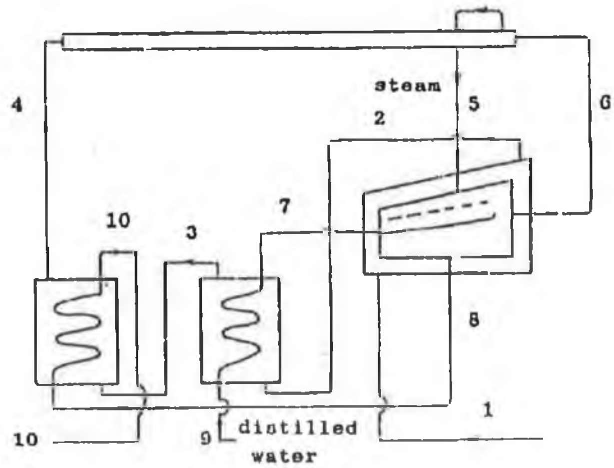
PERFORMANCE: Average Annual (Measured)

Power	3,000 BTU / Hr 878 Watts
Energy	24,000 BTU / 8 Hr Day 7,023 Watt Hrs / Day
Annual Energy	87.6 Therms 2,563 Kilowatt Hrs
Distilled Water	2.9 Gallons / Day 1,050 Gallons / Year

Fig. 4.12 The parabolic solar still marketed by BASAR SOLAR



1:collector tube ; 2:glass cover; 3:parabolic reflector ; 4:flexable tube ; 5:T connection; 6: bearing ; 7:separation and condensation unit;8:frame; 9:first heat exchanger; 10:tilting mechanism ; 11: second heat exchanger.



Flow diagram showing the flow of water and the location of thermocouples.

Fig. 4.13 The parabolic type solar still installed Mu'tah university, Jordan

Also number of studies and works has been done on solar assisted multiple (or single) stage/effect desalination systems. Joyce and et.al. proposed a single effect solar powered desalination system, using CPC solar collector [141]. A prototype electrical powered single effect distillation system was built and tested. The test results indicated a maximum productivity of 2kg/h of distilled water. Hamed and Al-Jebri conducted a study on a multieffect stack desalination system [142]. The plant consisted of three sub-systems, including solar heat collectors, hot water storage and 18 effect. The system was able to produce 120m^3/day of fresh water. The result of mathematical modelling confirmed that the accuracy of the model is reasonably good. The schematic diagram of the plant is brought in figure 4.14. More recently, El-Nashar conducted a theoritical study on multi-effect stack type distillation systems in order to optimise the design of the plant and reduce the cost of fresh water produced [143]. The main variables in the simulation program were the collector area, accumulator capacity and the number of effects of the evaporator. The evaporator capacity was assumed 1200m^3/day. Based on the results of this study it was claimed that drop in collectors cost to 50% of its basic value can result in a drop in cost from \$4.77/m^3 to \$3.03/m^3. It was also estimated a maximum accumulator capacity ratio of 0.2 as the highest limit of accumulator. Sun utility network, Inc. marketed a 30,000gallon/day solar powered multistage flash desalination system. The unit is incorporated with evacuated solar collector or uses a solar pound to gather the solar energy [144]. Also the company introduced a solar assisted mobile desalination unit with productivity of 13,000gallon/day. The heat source for driving the unit is obtained by evacuated solar collector and/or waste heat from the cogeneration system that produces electrical power. ISC Inc. has also proposed a hybrid desalination unit [145]. The desalination plant is a diesel multi-effect unit i.e. is retrofitted with solar trough concentrators. The company claimed that each solar trough is capable to produce

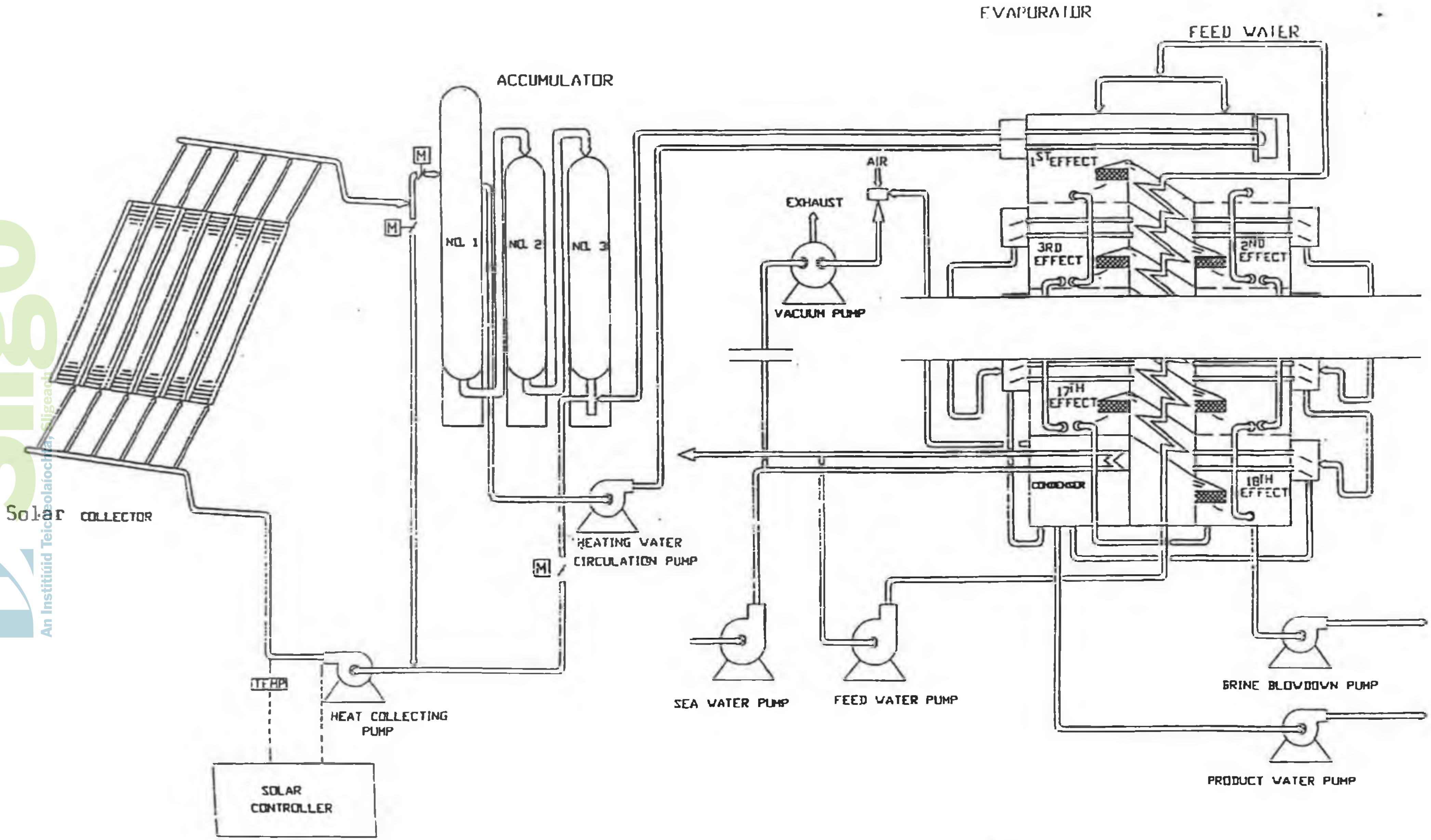


Fig. 4.14 Schematic diagram of the multi-effect desalination plant proposed by Hamed and Al-Jebri.

173MJ/hour i.e. is enough for production of 120galon/day of fresh water.

Bier and Plantikow conducted a study on solar powered desalination by membrane type distillation [146]. A prototype unit was built and tested on the island of Ibiza/Spain. WRPC in a corporation with Tekenaka Co. and Organo Co. has developed a membrane desalination unit [147]. The unit was capable of production of average 40 litres of distilled water per hour.

4.5 Double Chamber Evaporation Condensation System

The newest addition to the family of solar powered desalination based on evaporation and condensation was developed at University of Harare in Zimbabwe. The development of this system was reported in a paper presented at the Congress of World Renewable Energy Network at Reading England in 1994 by Saghafi [148]. Although the reported system was very crude the results reported in connection with this system clearly indicated the promising nature of this design. An improved version of this design with added energy recovery mechanism has been developed as part of the present research programme. This development and the results of its operational performance are presented in chapter 9. The improved version with added energy recovery is rated as a significant development of this project.

Chapter 5

Development of a Continuous Cycle Solar Thermal Refrigeration System

Chapter 5

Development of a Continuous Cycle Solar Thermal Refrigeration System

5.1 Introduction

The comprehensive review of all possible and probable solar cooling systems presented in chapter 3 clearly indicates that the Platen-Munters refrigerator (commonly known as Electrolux refrigerator) is the most suitable system to be powered by solar energy. Perhaps the most convincing reason is the fact that the required pressure is resulted from a partial pressure of the refrigerant in the system, and therefore, there is no need for a compressor or a pump to create the required pressure difference. However, despite this unique attractive features, there has been no report of any attempt to the effect that this approach ever being tried and tested under real life solar power conditions. Furthermore, in a comprehensive paper published by Critoph [149] it has been claimed that Platen-Munters approach is not suitable to be powered by solar energy. The reasons presented were technical complications associated with any large system of this design, low COP and lack of reliable solar power suitable for satisfactory operation. It is very important to highlight the following facts in relation to the above paper:

i- All the findings of the paper were based on purely theoretical estimation and not based on actually developing and testing the performance of a solar powered system of this design, ii- The only source of power considered was flat plate solar heat collectors (which are incidentally associated with the lowest efficiencies in the family of solar heat collectors).

After several private communications between supervisor of this project with the author of the above-mentioned article Ref.[149] with the great depth of expertise available at the Solar Energy Research Centre at the Institute of Technology of Sligo in the fields of the advanced high- efficiency solar heat collectors and also in ancillary solar thermal systems and technologies successful adoption of the Platen-Munters refrigerator to solar energy is indeed a worthwhile piece of research work.

Two prototype Solar Powered Platen-Munters refrigeration systems have been developed and tested as part of the present research programme. The details of development, evaluation and performance results of the prototypes systems are presented in this chapter. The excellent results obtained clearly prove that with deployment of appropriate modification techniques Platen-Munters system is, indeed, a viable option in solar cooling. The results of this development were presented as a scientific paper in the Congress of the World Renewable Energy Network and has been published by Pergamon Press. A copy of this paper is presented in the Appendix.

5.2 Constructional Details

5.2.1 The Refrigerator

The refrigerator was a small commercially available Electrolux design unit, known as the so-called "bedroom refrigerator". This unit is routinely being produced by a company in Poland. The source of power of this refrigerator is an 80W electrical heater. The main point in the successful operation of this refrigerator is to run it with a continuous and adequate (not than 80W) of power. Furthermore, according to the manufacturer's instruction the unit must be kept in a totally horizontal.

The electrical heating element was disconnected and removed from the refrigerator system. special heat transfers devices were then manufactured to provide the collected solar heat to the refrigerator, for each of the two prototypes made.

5.2.2 Solar Heat Collection System

As pointed out earlier in this chapter two prototypes were produced, in this approach. The only difference between the two was the type of solar heat collectors used. The first prototype was powered by the thermal energy collected from the sun using a non-tracking concave reflector, successfully developed and tested by Menon [150]. In fact, there is a great deal of knowledge about this type of collector at the Solar Energy Research Unit at the Institute of Technology of Sligo. This Knowledge was fully perfected by Valizadeh

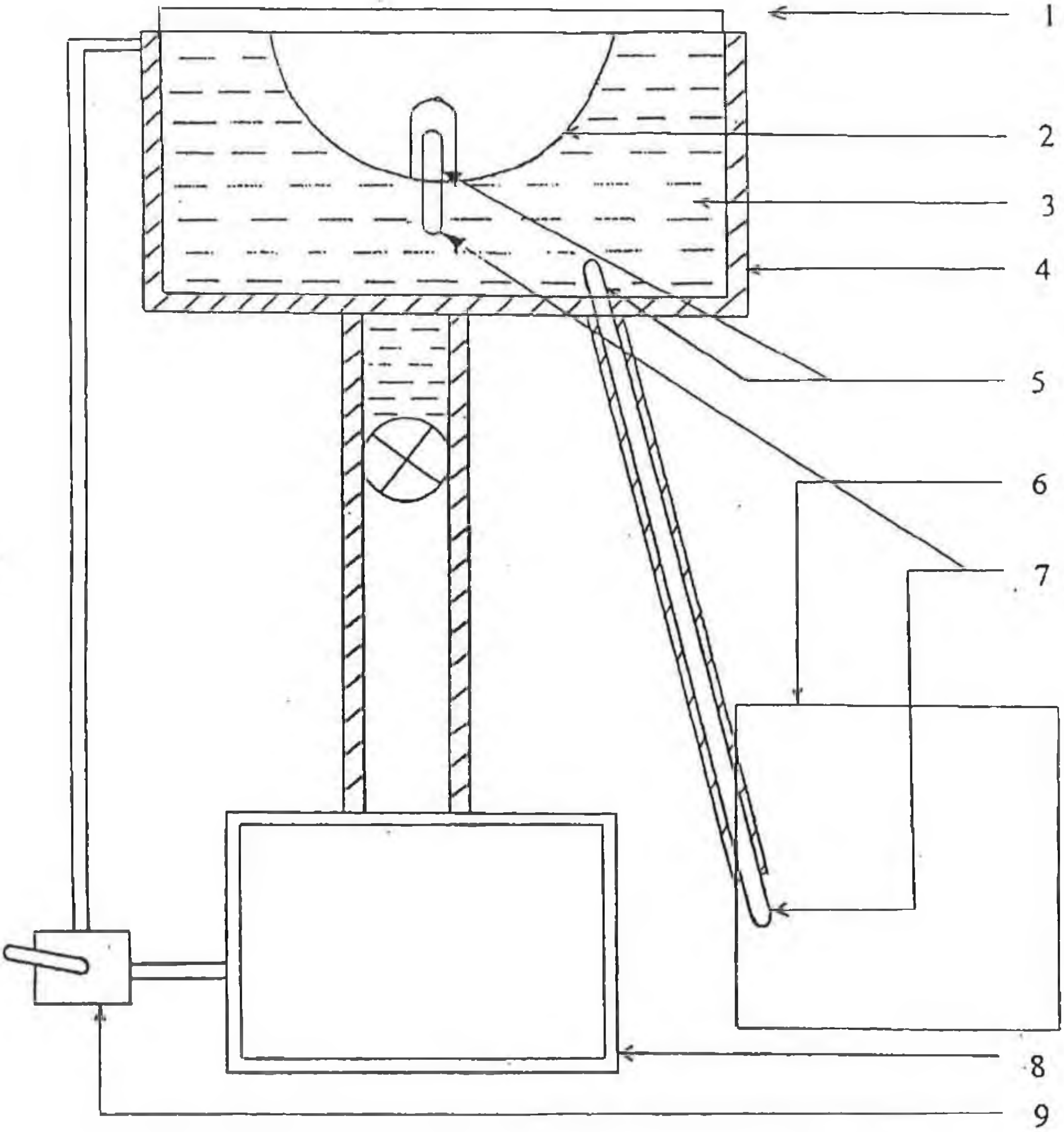
[151] who used this type of collector in a storage type solar powered cooker. A schematic diagram of this system is shown in figure 5.1.

After the successful development and testing of the first prototype where the suitability of Platen-Munters design to solar energy was established in practice in the solar heat collection system of the prototype was replaced by advanced evacuated, high-efficiency collector incorporating heat pipe. The particular know-how and technology in relation to the production of this type of collectors were developed by Valizadeh [152]. All the test results obtained and presented in this thesis in connection with suitability of the operation of Platen-Munters refrigerators to solar energy are obtained using this particular type of collectors, for the size of refrigerator used 5 collector tubes equivalent to about $1m^2$ of collector area was used.

5.2.3 Transfer of Heat from the Collector to the Refrigerator

Transfer of heat from the non-tracking reflector system (used in the first prototype) to the refrigerator was secured using peanut oil tried and tested for the first time as discussed in Ref. [151] at the Solar Energy Research Centre at Sligo.

In the second prototype a special heat exchanger was designed and manufactured to transfer the collected heat from the evacuated collector to the refrigerator. This heat exchanger was machined out of two slabs of 12.5mm thick aluminum. Each slab contained grooves where the condenser of the collectors could be sandwiched. A photograph of this heat exchanger is presented in figure 5.2.



Captions: 1 - pane of glass, 2 - compound parabolic concentrator, 3 - special oil for thermal battery, 4 - the battery tank, 5 - evaporator of the heat pipe, 6 - the refrigerator, 7 - condenser of the heat pipe, 8 - lower level storage tank, 9 - hand pump.

Fig. 5.1 Schematic diagram of the solar heat collection-storage system coupled with the refrigerator

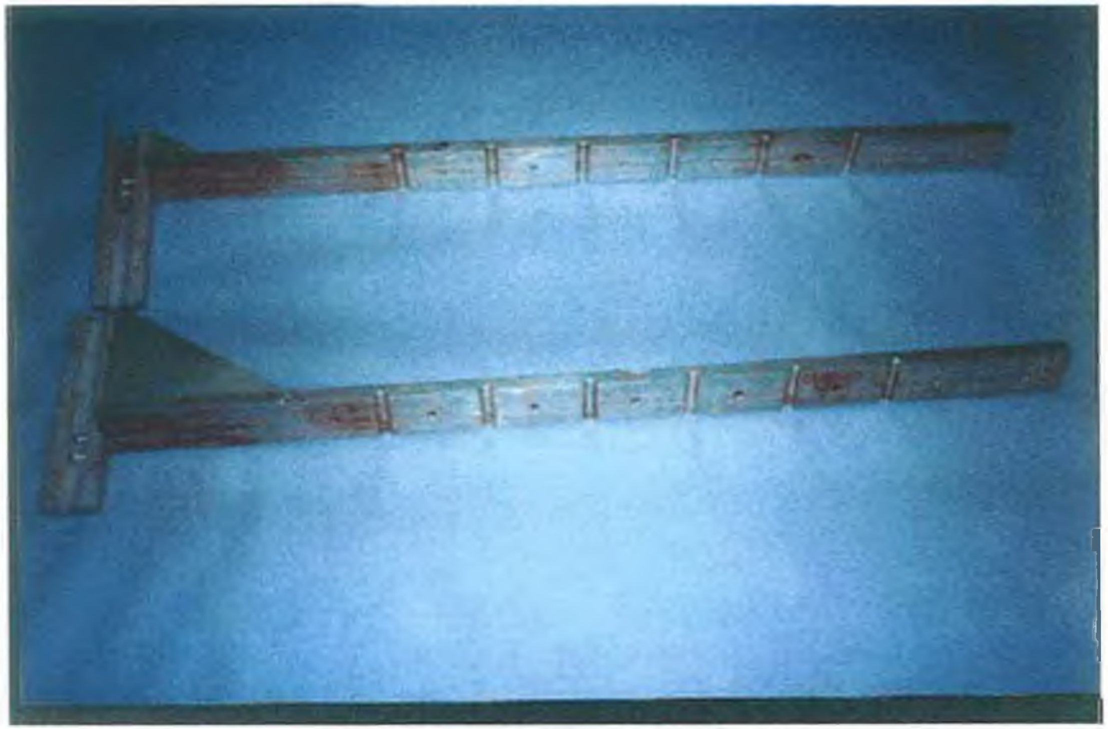


Fig. 5.2 The aluminum slabs of the heat exchanger

It must be pointed out that the prior to the deployment of the above-mentioned heat exchanger several attempts were made to transfer heat from the condensers of the collectors to the refrigerator using a manifold pipe (containing peanut oil) where the condensers were clamped to. Since it was difficult and costly to use a reasonably priced pump to transfer the heated oil (whose temperature was above 150°C) it was hoped to use the thermosyphon principle. However, after several trials this approach proved unsuccessful as the heat loss associated with this approach was to high for any satisfactory operation of the system.

5.2.4 Thermal Insulation

Initially, during the development and testing of the first prototype, no attempt was made to assess the effect of extra thermal insulation on the performance of the refrigerator. However, during the evaluation of the performance of the second prototype it was noticed that lower temperatures could be achieved for lower temperatures of the environment surrounding the refrigerator.

This observation then opened the way to add more layers of thermal insulation materials to the existing 10-15mm thick layers. For this reason layers of glass fibre materials of various thickness were wrapped around the refrigerator and the lowest temperatures obtained, inside the refrigerator, for each case was recorded. It was found that an extra layer of about 150mm thick produced the lowest temperature. However, due to constructional constraints thickness layer of 100mm was the most practical parameters.

It must be pointed out that the insulation materials to the refrigerator could only be added to the top, front and the two side faces of the refrigerator. It was impossible to insert any to the back (where the pipe works are mounted and also to the bottom. The reason that nothing was done to the bottom face was due to logistic constraints for the test set up.

5.2.5 Solar Powered Air Cooling Tower System for Enhancing Heat Removal

In order to expect the best realistic result it must be remembered that every effort must be made to take on board all the adverse points as far as the real life operation of such a refrigerator is concerned. In this connection one of the most important factors to be considered is that the environments under which a refrigerator of this kind may operate have, usually, an ambient temperatures of 40-45°C and perhaps with high humidity. Under these conditions it was thought that it would be an advantage if an artificial air circulation system is added on to the condenser part of the refrigerator to speed up the removal of heat to the atmosphere. For this purpose a simple chimney was designed and mounted to the back of the refrigerator. This chimney was designed to be in the form of an inverted funnel. It was to be mounted to the back of the refrigerator in a vertical position with the broad end of this funnel very close to the ground so that the air is sucked in from the broad end and travel upward toward the narrow end. The funnel was fixed to the back of the refrigerator in such a way that heat dumping condenser is perfectly in the direction of the upward movement of the airflow.

The upward movement of the air was secured by connecting a spiral pipe to the inner wall of the inverted funnel shape chimney. This spiral pipe covered up to 1/8th of the very top part of the chimney. Solar heated water was circulated in this spiral pipe using the standard thermosyphon principle. As the air in the top part of the chimney was heated up, due to the hot spiral pipe then it rose to higher levels and this way a simple but very effective air draft system was created.

Since the solar heated spiral pipe was mounted at the very top of this chimney, and way above the condenser of the refrigerator, there was no danger of the heat reaching it. A narrow pipe providing a small quantity of cooling water (0.25 litre/hour) on the condenser was also used an additional auxiliary heat removal system.

This way a simple, but very effective, cooling tower was used to speed up the removal of heat from the refrigerator. A diagram of this cooling tower is shown in figure 5.3.

5.3 Evaluation

It was very important to establish a general relationship between the temperatures of so-called "hot side" and "cold side" of the system. The hot side is the manifold (or the condenser of the evacuated solar heat collector) and the temperature at which the collected heat is actually applied to the heat intake part of the refrigerator. In total 6 thermocouples were connected to the system. Thermocouples T_1 and T_2 were connected to the hot side whereas thermocouples T_3 to T_6 were connected to the cold side. Three thermocouples T_3 to T_5 were connected to the pipe containing refrigerant inside the icebox (at its entrance,

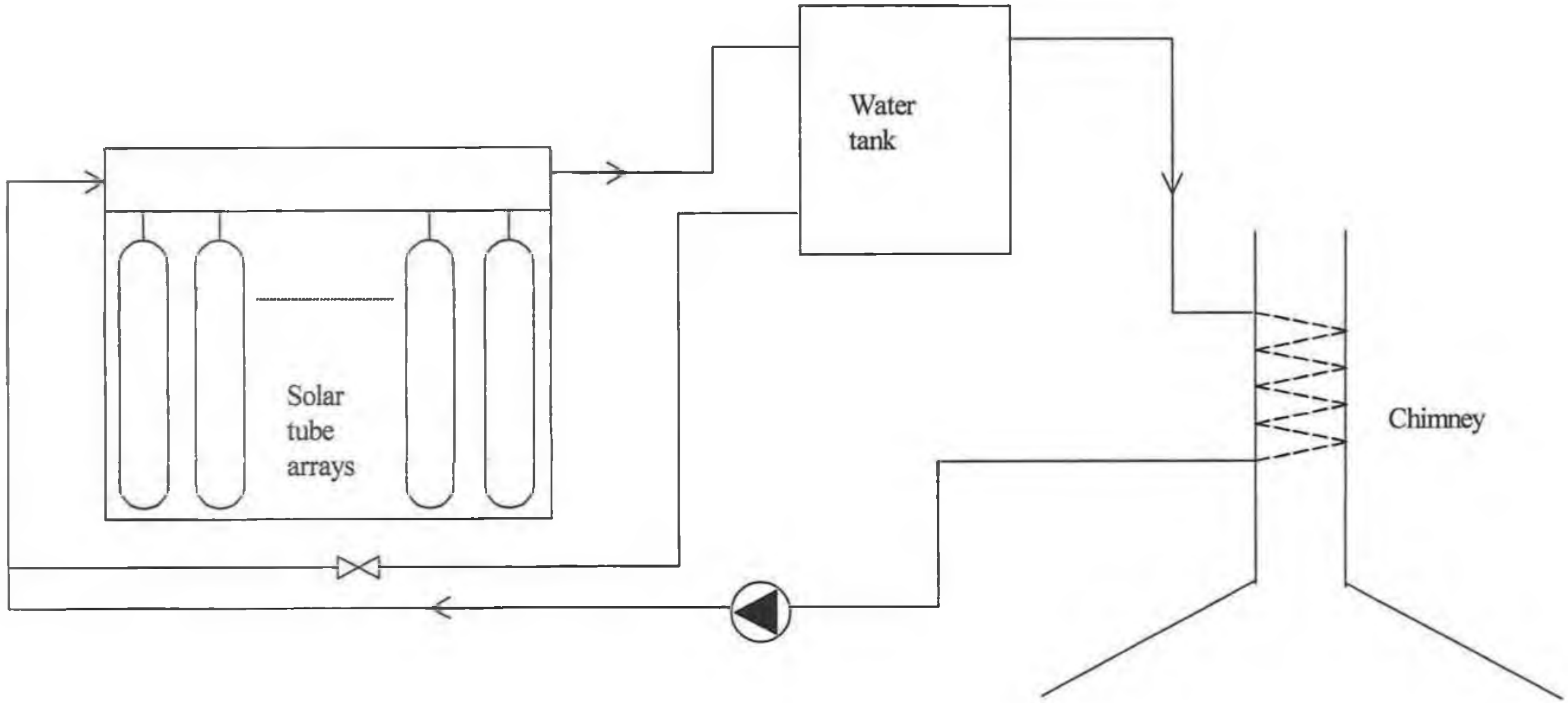


Fig. 5.3 Schematic of the Solar Powered Air Cooling Tower System

middle and exit points) and T_6 was detecting the temperature inside the refrigerator, but outside the icebox.

For both Prototypes tested it was observed that the actual sign of cooling, which was taken as the start of the drop of temperature in the ice box, appeared as soon as the temperature of the heat applied to the generator of the refrigerator reached a range of 110°C-115°C. For the second prototype, this point was reached after 90 minutes of exposure of sunshine whereas in the first prototype this point was reached after longer duration of exposure. However, because of the storage capability of the collector system used the cooling continued after the sunset. As pointed out earlier, the simplicity and ease of operation of the evacuated heat collectors used made this operation more attractive, both from practical and commercial view points. A view of the pipe works at the back of the modified refrigerator is presented in figure 5.4. The place where the aluminium heat exchanger is to be connected to is marked.

5.4 Tests Results and Discussions

The first set of tests were carried out without usage of any additional thermal insulation material on the walls of the refrigerator. The cooling effect started after 90 minutes of exposure. After about 6 hours of exposure temperatures of 7°C and -9°C were obtained in the refrigerator and icebox respectively. The results of test are presented in figure 5.5.

It is important to note that this result is a very exciting development because the range of the temperatures obtained are well within (and even better than) those of commercially available refrigerators.

The place where
the heat exchanger
to be connected

Fig. 5.4 A view of the pipe work at the back of the refrigerator

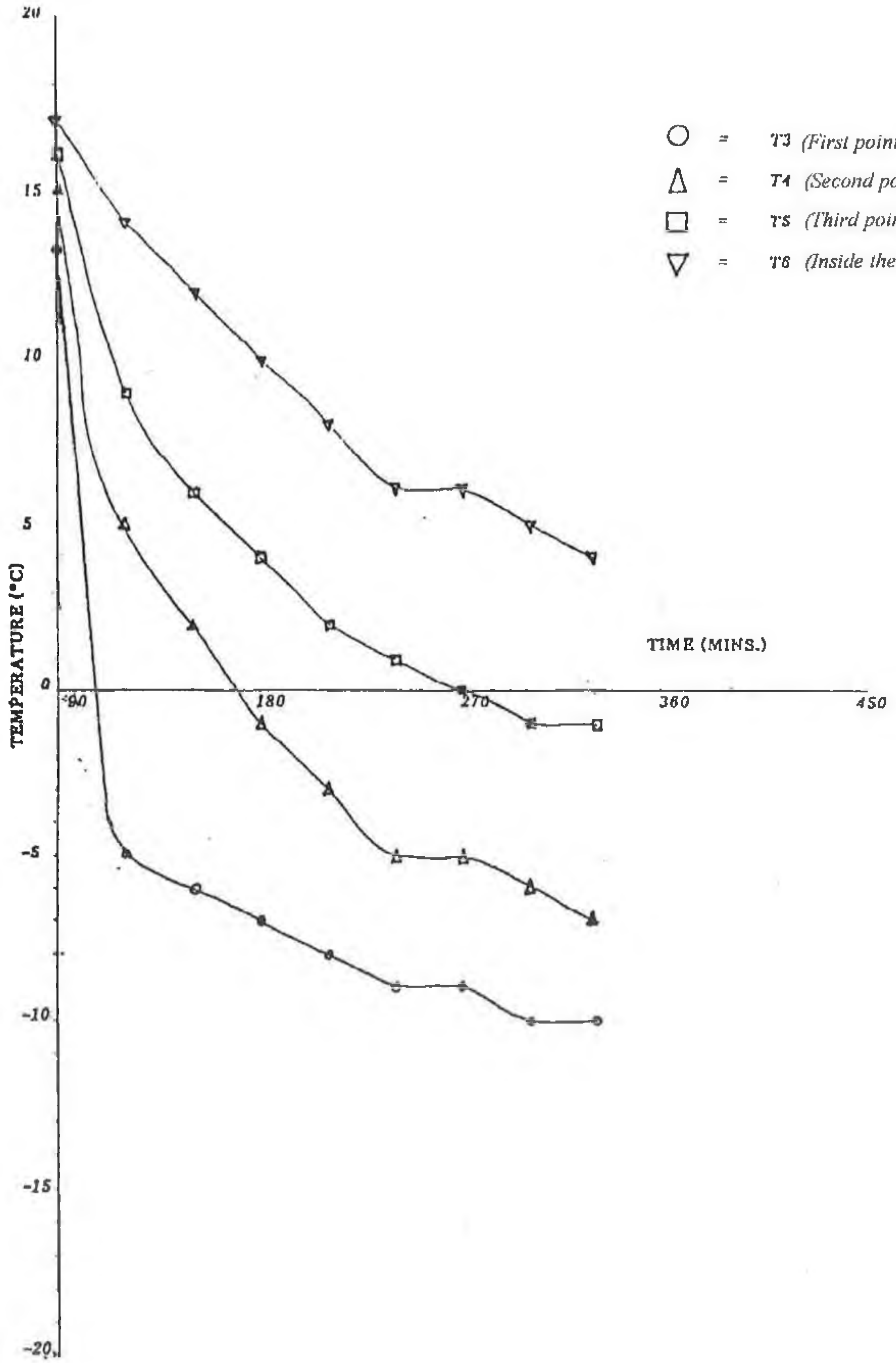


Fig. 5.5 Test Results of the system without additional insulation material

In the next set of the experiment insulation materials with thickness of about 100mm were placed on the front, two sides and top faces of the refrigerator as discussed in section 5.2.4. It was interested to note that under this condition the actual cooling and drops in temperatures on the cold side taken place at much faster rate. The result of this set of tests is presented in figure 5.6. It can be seen that after 4 hours of exposure the temperature of the interior of the refrigerator dropped to 0°C and that of the icebox dropped to -13°C.

This result which shows a drop of 7°C in the refrigerator (from 7°C down to 0°C) and a drop of 4°C in the ice box (from -9°C down to -13°C) is another exciting proof of suitability of Platen-Munters design to be powered by solar energy if appropriate and correct components and modifications are used. This proves that sufficient cooling power does exist under the right conditions. The results indeed show that cooling power to achieve temperatures in the range of lower than -1.5°C in the interior of the refrigerator and lower than -15°C in the ice box is routinely possible.

In another set of tests where the effect of spraying water on the condenser of the refrigerator was investigated about 20-40cm^3 of water was sprayed on the condenser in 10 minutes intervals. This approach has improved both the speed of cooling and the final temperatures achieved. The temperature of the interior of the refrigerator reached 0°C by about 30 minutes, and that of the icebox reached -13°C, by about 60 minutes, compared to the previous case respectively. Furthermore, the ultimate temperatures achieved after about 5 hours of exposure were less than -3°C and -17°C in the interior and the icebox of the refrigerator, respectively. The results of this series of tests are presented in figure 5.7.

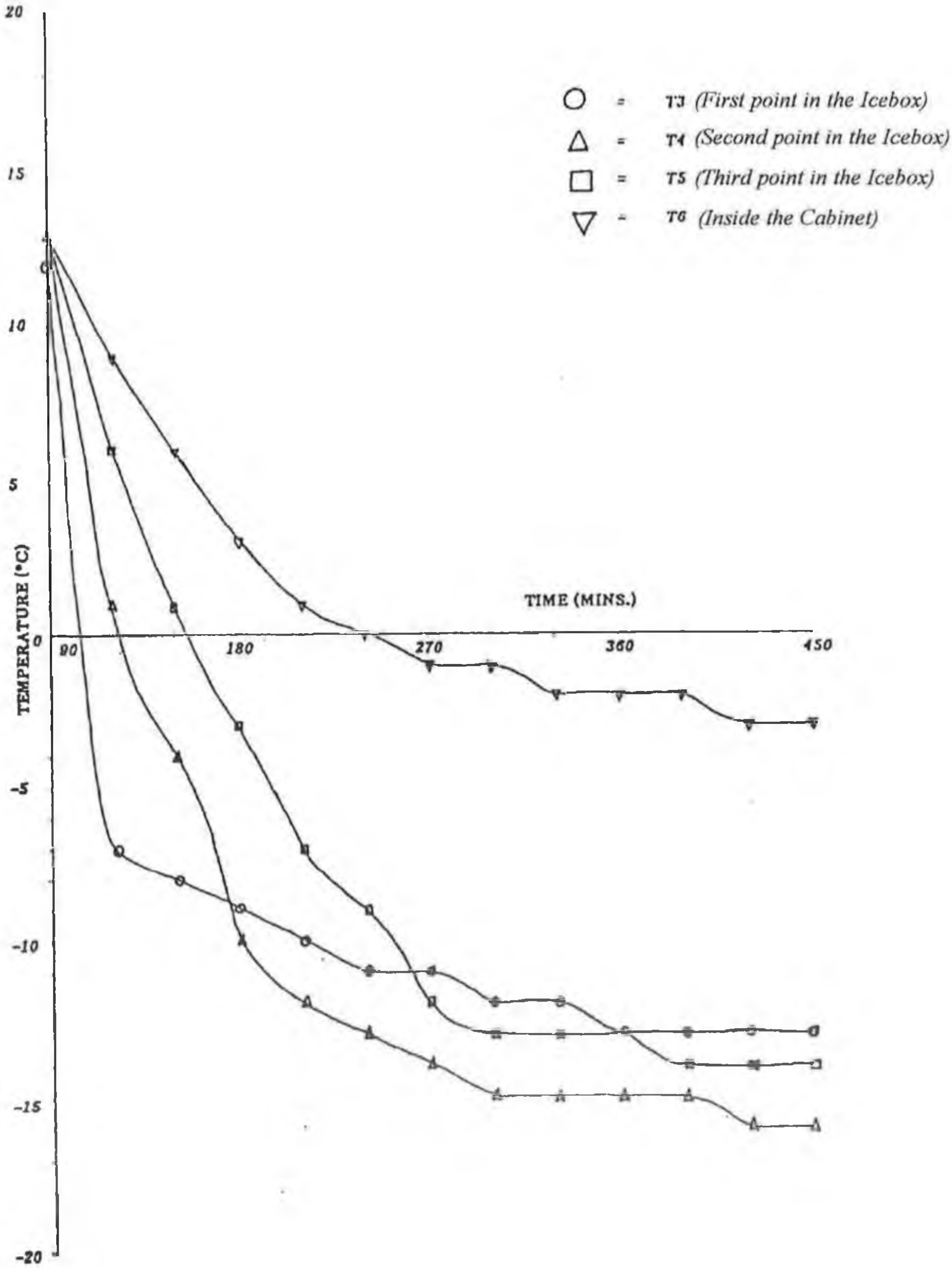


Fig. 5.6 Test results for the refrigerator with 100mm thick of insulation material

The effect of the combined water spraying and air blow using a cooling chimney was investigated in another sets of tests. The results of this approach are given in figure 5.8. It can be seen that the effect of the simultaneously water spraying and ventilation has been faster removal of heat from the condenser, hence shorter cooling time and lower temperatures were achieved.

5.5 Conclusions

The results of the evaluation clearly indicated that Platen-Munters design is indeed very suitable to be powered by solar energy. This is for the first time that the merits of this have been put to test in practice. There is now no doubt that there is high levels of cooling power in solar powered Platen-Munterns refrigeration system. This finding clearly indicate that the system lends itself to generate enough cooling power sufficient for more than the times it is exposed to sunshine. In order words it seems possible to use refrigerators of this type for long hours if they are to be used in conjunction with appropriate storage system where the cooling power can be stored.

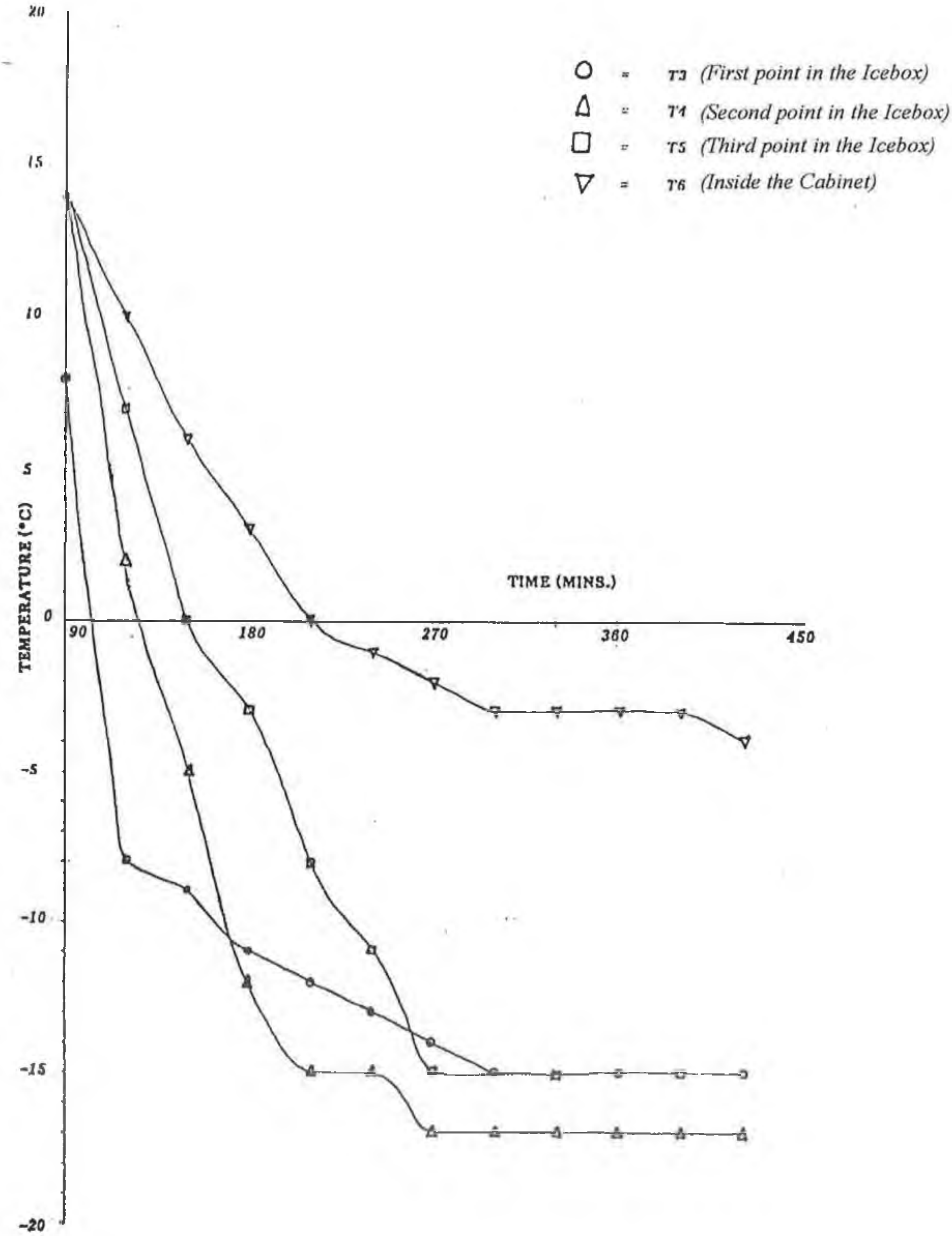


Fig. 5.7 Test results for the water spraying with 10 minutes intervals

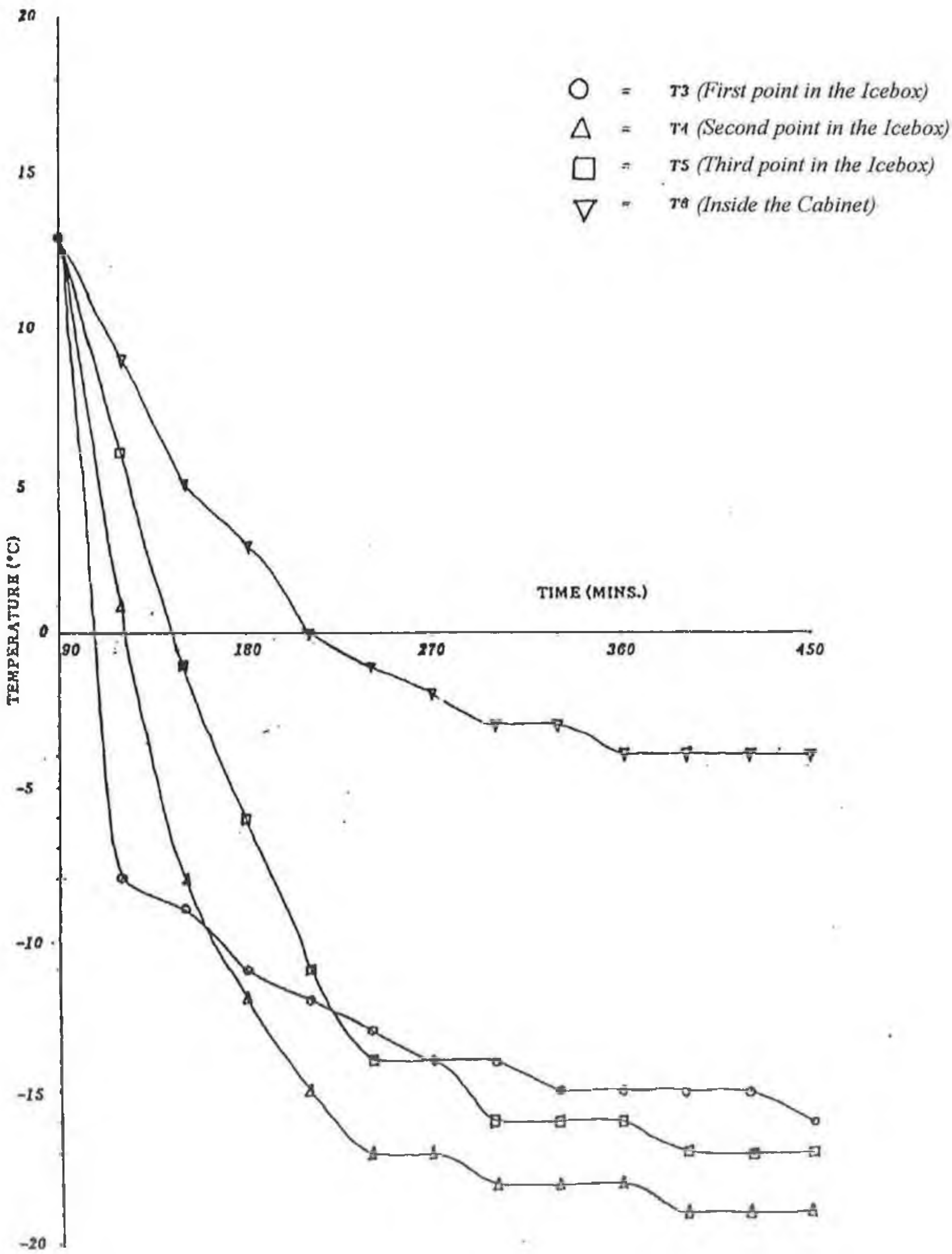


Fig. 5.8 The results of the simultaneous water spring along pipe works at the back of the refrigerator and air ventilation

Chapter 6

A New Approach in Using Photovoltaic Power in Refrigeration

Chapter 6

A New Approach in Using Photovoltaic Power in Refrigeration

6.1 Introduction

The idea of application of photovoltaic panels to power commercially available refrigerators is nothing new. In fact, a lot of commercial developments have been carried out for various programmes financed by the bodies affiliated to the United Nations under various aid programmes. The operation of almost all such system is based on connecting an existing commercial refrigerator (without any meaningful modification) to a photovoltaic panel via an inventor to convert the d.c. power of the photovoltaic panels to a.c, power. A battery system is also used to store power when the solar energy is not available.

The size of the photovoltaic panels for such systems is inevitably very large, and consequently, the overall cost of systems is too high to be assessed in a commercial sense. Even the small vaccine refrigerators, developed for various United Nations programmes, are far too expensive to be remotely considered as a competitor to the existing household refrigerator. The power requirement for the refrigerator and the limited efficiencies of the existing photovoltaic panels clearly indicates that a significantly larger size of photovoltaic arrays is needed for such systems. In fact, the larger required size of the photovoltaic panels is the most significant factor that contributes to increase the cost of systems of this kind. It, therefore, becomes clear that for attempts so far made for there is no commercially viable future in powering household refrigerators by photovoltaic panels. However, a programme of research investigating various design modifications to the existing household refrigerators aimed at substantial reduction in input power can be taken as the first significant step forward. The work described in this chapter is concerned with an investigation and assessment of a number of novel approaches aimed at paving the way forward for the use of significantly smaller sized photovoltaic panels to power household refrigerators, in remote regions where there is no mains electricity supply.

6.2 Description of the Plan of Action

The programme of work in which the possibility of using photovoltaic panels to power a refrigerator as the first step towards commercial viability is planned as follows:

❶ Investigation of all factors that contribute significantly to the input energy demand, in order to aim at reducing at the size of the required photovoltaic panels.

❷ Development of an appropriate mechanism to obviate the need for the battery that is used for the storage, and subsequent usage of the electrical power, generated by photovoltaic panels.

❸ Design and construction of a working prototype based on the findings of step 1 and 2.

❹ Evaluation of the performance of the prototype.

These factors are described in detail in the rest of this chapter.

6.3 Investigation on the Factors Affecting the Input Power Demand

The main factors that contribute to the input power demand can be listed as follows:

1. Efficiency of the refrigeration system
2. Heat gain of the refrigeration cabinet
3. Energy losses from the system

6.3.1 Efficiency of the Refrigeration System

The factor that directly influences the efficiency of the refrigeration system is the coefficient of performance, abbreviated as COP. The COP of the system is dependent on the outside ambient temperature, temperature of the refrigerator cabinet and efficiencies of all the components used.

6.3.1.1 Temperature Difference between the Cabinet and Outside Atmosphere

So far, as the temperatures of the refrigerator cabinet and the outside ambient are concerned, it is a well-known fact that the COP is inversely proportional to the difference between these two temperatures. In other words higher COPs can be achieved as the temperature difference between the inside of the cabinet and the outside ambient, through which heat is to be pumped, reduces. In order to achieve the smallest possible temperature difference between the refrigerator cabinet and outside ambient the following two approaches are thought to be most appropriate:

1. increase the size of the evaporator, and
2. increase the size of condenser

of the refrigerator system.

This is because an increase in the size of the evaporator helps with improved collection of heat from the inside of the refrigerator cabinet, and an increase in the size of the condenser will speed up the dumping of the heat collected from the inside of the cabinet. It would, therefore, be easier for the system to pump heat from the inside cabinet to the outside cabinet.

6.3.1.2 Utilising the Evaporator and Condenser of the Refrigerator for Longer Periods at Higher COPs

The COP of the refrigerators depends on the maximum utilisation of the evaporator and the condenser over longer periods. This means that the speed of the pumping of the heat from the inside of the cabinet to the outside ambient must be totally in harmony with the cooling load demand. The cooling load demand depends on the quantity of the food in the refrigerator cabinet, the ambient temperature and the frequency of the opening and closing of the cabinet door.

It has been decided that using a compressor with variable speed, whereby it could accelerate during high cooling demand and decelerate during low demand, will be the most appropriate step to achieve the desired goal. This is because as a result of the acceleration and deceleration during high and low load demands the evaporator and condenser will be used for longer periods and will perform at higher COPs.

6.3.1.3 Saving Energy Wasted by Sub-Cooling

It has been found that the starting and stopping temperature settings of the compressors and the temperature differentials are also important parameters in the COP of refrigerators. A more efficient and precise control of these temperatures could bring about a saving of waste energy by sub-cooling.

It has been thought that using an electronic thermostat will facilitate the desired control thus leading to an improved COP.

6.3.1.4 Improving the Liquid to Gas Ratio in the Capillary and Recovering Unusable Energy

The capillary in a refrigerator passes liquid more rapidly than gas. In this respect an improvement in the liquid to gas ratio can, therefore, increase the mass flow rate of the refrigerant. This in turn improves the efficiency of the refrigerator.

The introduction of a heat exchanger to recover available cooling from the suction line can bring about an improvement in the liquid to gas ratio of the mix in the capillary tube. This point can be substantiated by referring to the graph in figure 6.1, where the temperature and pressure distribution in the capillary tube is shown. The temperature is constant until it reaches the critical point 2 where the first bubble appears. This critical point can be brought further down, by additional cooling, which will result in a better refrigerant flow rate through a reduction in resistance.

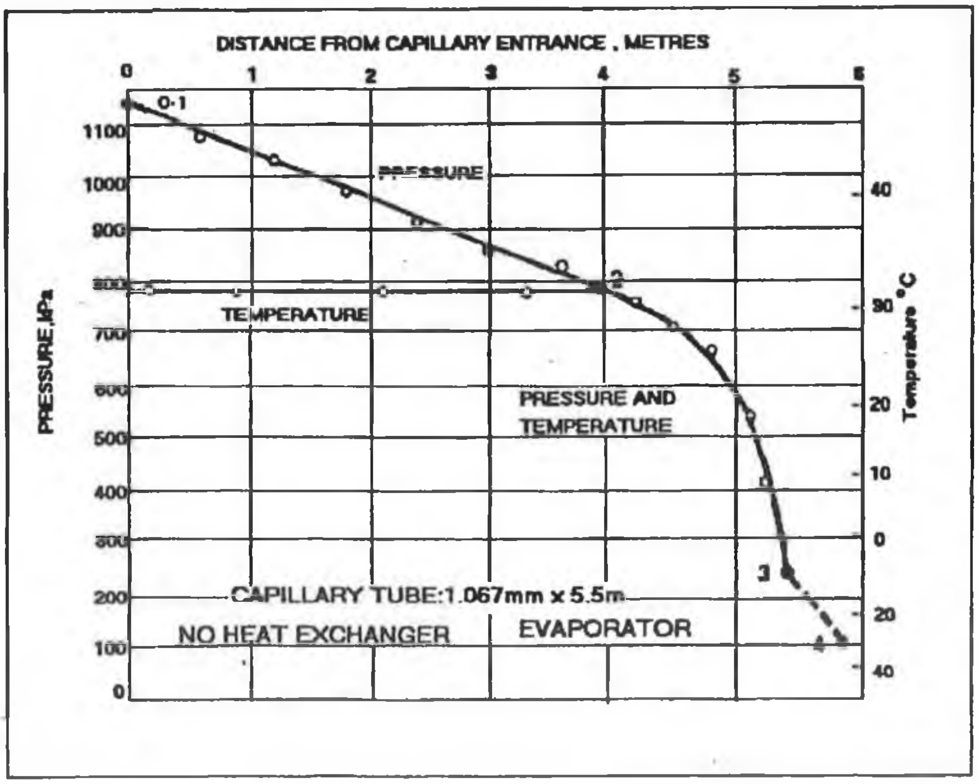


Fig. 6.1 Pressure and Temperature distribution along a typical capillary tube

6.3.2 Heat Gain of the Refrigerator Cabinet

The heat gain of the refrigerator cabinet depends very strongly on the following factors:

1. The outside cabinet temperature,
2. The temperature difference between the outside ambient temperature and the refrigerator cabinet, and
3. The surface area of the cabinet walls exposed to the outside world, and the thermal conductivity of the walls of the insulating material.

It is easily recognisable that the only parameters worth investigating is the degree of thermal insulation between the inside and outside of the refrigerator cabinet, as discussed in point 3 above. .

In all the household refrigerators investigated the wall thickness of the thermal insulation material used is, at best, about 30mm. Figure 6.2 shows a typical characteristic of cellular polyurethane, which is the most commonly used thermal insulation because of its higher insulation property and low cost. This graph presents the effect of the thickness of the material over the annual energy consumption for a 200 litre refrigerator when the compressor runs for 2.8 hours. It can be seen that an increase of 45mm, from 30mm, in thickness of the thermal insulation material energy consumption drops from about 720kWh down to about 340kWh, or a reduction of over 52%. This graph clearly illustrates that better thermal insulation can significantly reduce heat gain hence reduce the energy demand in refrigerators. It is important to highlight the significance of this result in the case of refrigerators being powered by photovoltaic panels. This is because the extra cost of the thickness of the thermal insulation material is significantly lower compared to the cost of the extra panel size and storage facility for the collected electrical energy from the photovoltaic panels.

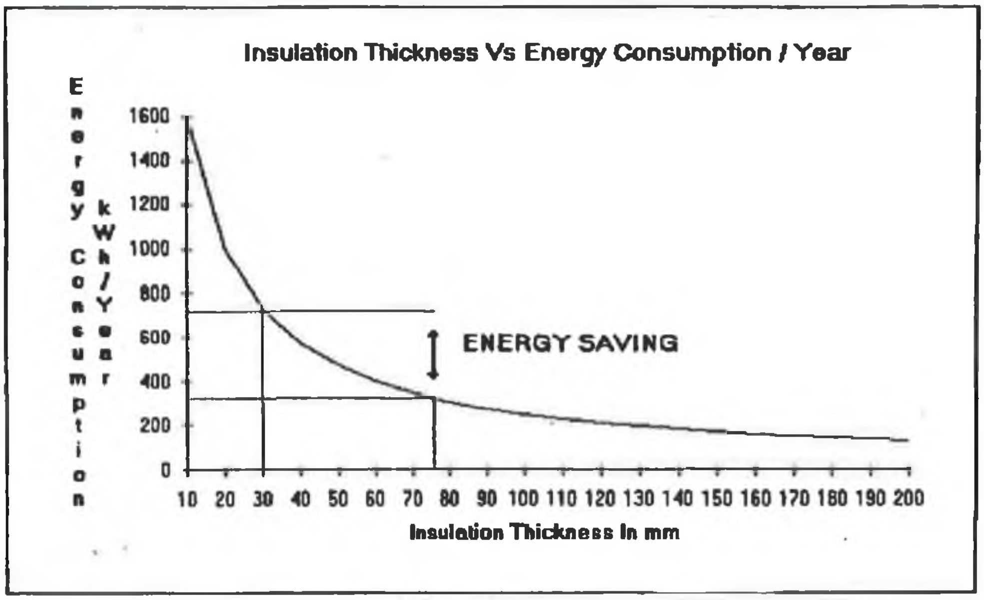


Fig. 6.2 Insulation material thickness versus Energy consumption

6.3.3 Elimination of the need for the Electrical Battery and Inverter

There are three main reasons that give credibility to the idea of eliminating the need for the electrical battery and inverter. They are:

1. loss of power as a result of using the battery
2. loss of power as a result of using the inverter, and
3. the costs associated with the battery and the inverter and also their maintenance.

However, the implementation of this approach means that the compressor of the refrigerator must be replaced by a d.c. one. Apart from the technical argument it must be pointed out, that as d.c. compressors are not mass produced to the scale which a.c. are, d.c. compressors are more expensive. The commercial merit of the usage of the more expensive d.c. compressor in the overall context of cost benefit discussion will be assessed later in this chapter. At this stage it is worth investigating this approach with particular focus on technical aspects of the argument. It would be then that the overall merit approach could be realistically assessed.

6.3.3.1 Technical Aspects

a) Eliminating of the Battery

As pointed out earlier not a great deal of innovation has gone into the refrigeration systems powered by photovoltaic panels so far. The block diagram of such a system has already been shown in figure 3.2. In this system a conventional refrigerator that must operate with mains a.c. electricity is to be powered by a photovoltaic panel. Since the output of the photovoltaic panel is always a d.c. voltage an inverter is needed to convert the d.c. voltage to mains a.c. to power the refrigerator. However, although it may look straightforward enough, the system suffers from a number of drawbacks. The main disadvantages of such an approach are the considerable power losses associated with using the battery and the inverter as well as with the maintenance of such devices. There is also a problem associate with usage of batteries with particular reference to the required short charging period. Although there are several attempts by various companies to address this problem and a number of batteries have been marketed as "solar batteries" it has been seen in practice that this problem does not seem to be completely resolved. Despite these problems it must be pointed out that the main technical advantage of the system shown in figure 3.2 is the fact that a commercially available refrigerator can be used without the need for any modification.

Although the elimination of the battery (and the inverter for that matter) has the advantage of cost reduction and removes the need for maintenance, it introduces a number of new problems. A major part of the research programme reported in this chapter is how to address these problems. Among all the questions perhaps the most important ones to be addressed are:

❶ how can a d.c. power of the photovoltaic be used to power a conventional household refrigerator?, and

❷ how can the refrigerator operate satisfactorily during the long hours that there is no solar irradiation i.e. night?

Once d.c. power is to be used to operate any conventional refrigerator it immediately necessitates the replacement of the a.c. compressor with an equivalent d.c. one. Although this approach solves one set of problems, as outlined above, it raises a number of new points to be taken in account.

The most fundamental questions to be considered refer to the variable nature of the output power of the photovoltaic panel. Due to the fact the incident solar radiation is not uniform throughout the sunshine hours, how does a d.c. compressor perform under variable input power?

Also the question of the need for cooling power during the times when there is no or insufficient solar radiation, such as nighttime, becomes a problem if an alternative approach is not adopted. A thermal (cold) storage mechanism is thought to be a satisfactory approach. This is possible with the use of a suitable phase change material with high energy density, which will occupy minimal space inside the refrigerator cabinet. This approach has the advantage of higher conversion efficiency, lower cost and greater life expectancy.

6.4 The Question of Cost

The approaches discussed above that result in significant reduction in the size of the photovoltaic panel, elimination of electrical battery and d.c. to a.c. inverter brings about very significant cost reduction in the overall system. However, a number of new problems that necessitate design modification bring about their own cost. All these questions are addressed in section 6.6 of this chapter.

6.5 Practical Evaluations of the Suggested Design Improvements

A full description of all the major points that can contribute to significant improvement, in terms of reduction of the input power demand, in conventional household refrigerators has been presented in Section 6.3. The various recommended points have been arrived at after a thorough theoretical assessment of the design features of household refrigerators. In this section the various suggested design improvement are evaluated. The maximum cabinet temperature for these experiments was 42°C. Three sets of refrigerators namely 100litres, 156litres and 250 litres refrigerators were used as test models.

6.5.1 Thermal Insulation

As pointed out earlier in Section 6.3 the thickness of the thermal insulation materials (that are less than 30mm) is not sufficient to provide any satisfactory resistance to heat flow from outside the cabinet to the interior of the refrigerator cabinet. The effect of the thickness of the thermal insulation materials on reduction of heat gain of the refrigerator cabinet was assessed for following cases. In all the cases cellular polyurethane has been used as the thermal insulation material. It must be highlighted that particular attention was paid to find the optimum thickness for each case and not indulging in "over engineering" by not going for excessive thickness. This way the cost effectiveness of this approach was always ensured.

In the first set of experiments a 100 litre refrigerator was used. The wall thickness of the thermal insulation material was varied in stages of 5mm, from 30mm. The performance of each test was compared against a conventional 100 litre with 30mm insulation material. It was found that 75mm of wall thickness produced the best result and any further increase did not produce any significant improvement. It was found that increasing the wall thickness of the thermal insulation material from 30mm to 75mm the input power demand of the refrigerator showed a drop of 33%.

In the second set of tests the same procedure was repeated for a 156litre refrigerator. In this case it was found that the optimum results was obtained when the wall thickness of the thermal insulation material was 125mm. The reduction in the input power demand was about 36%.

In the third set of experiments a 250litre refrigerator was used as the test model. After repeating the entire procedure, a thickness of 125mm resulted in the optimum performance, and the reduction in the input power demand was about 30%.

6.5.2 Increased Evaporator Size

Increasing the size of the evaporator proved not to be as straightforward as increasing the wall thickness of the thermal insulation material. For this reason it was decided to increase the evaporator to twice that of the original size for each of the 3 test cases.

It was found that for the 100litre size the saving in input energy was about 7.6%. For the 156 and 250 litre sizes the saving was in the order of 7.3% and 6.9%, respectively.

6.5.3 Waste Energy Recovery

As pointed out in section 6.3 recovery of available cooling from the suction line improves the liquid to gas ratio of the mix in the capillary tube. In order to achieve this goal, transfer of heat from the capillary tube to the vapour coming from the evaporator is an obvious step. To this end a heat exchanger was introduced in the suction line with capillary tube coiled inside. This approach has resulted in 2.3% reduction in energy demand.

6.5.4 Temperature Control

Mechanical thermostats were found unsuitable for satisfactory control of temperature and efficient operation of the refrigerator. This is because they have wide differentials or hysteresis. Although they favour the off period required to equalise the high and low pressures, they are unsuitable as far as prevention of waste energy is concerned. The wide differentials (that cause wastage of energy) cause over-cooling and over-heating. Electronic thermostats, because of their more accurate control characteristics and temperature sensing capabilities, have been found more capable of preventing this wastage of energy.

It is worth pointing out that in a system with thermal energy storage the latent heat is stored at a constant temperature by changing phase. Therefore to make maximum use of thermal energy storage the compressor should start and stop immediately after and before the phase change. With an electric thermostat the set point and differential can be adjusted to suit the evaporator and refrigerator temperatures. It was not easy to arrange the set point and differential in order to adjust to suit the evaporator and refrigerator temperatures to achieve the most desired operation. This is because the freezing and thawing processes occur at separate interface locations. They are the evaporator storage medium interface when thawing and the storage medium cabinet air interface when freezing. It therefore required sensing at these two interfaces.

In practice it has been found that with storage the compressor cycling is far less frequent. The losses due to overcooling and overheating of thermostatic hysteresis are minimal.

6.5.5 Thermal (Cold) Storage

Bridging the time gap between the availability and use of energy is an essential part of a refrigerator if it is to be powered by solar energy. The idea of using an electrical battery was to address this issue. However, since usage of photovoltaic panels as a power source, minimisation of the size of the photovoltaic panels, elimination of electrical batteries and inverters with particular focus on cost reduction are the main themes of the work described in this Chapter an alternative mechanism of storage of energy must be adopted.

Thermal storage is the model substitute approach. This is because both the output and input of such a storage system are thermal energy and there will be no losses due to intermediate electromechanical energy conversion. In this connection phase change material is the answer because space is at a premium in refrigerators. For this reason larger size refrigerators are more practical for the implementation of this technique. The criteria in selection of the most suitable phase change material are melting point, specific heat, latent heat, heat of fusion, chemical and physical stability, toxicity, availability and compatibility with the container material and cost. Water was selected as the most obvious choice. The most significant deciding factors are its higher latent heat energy, non-toxic characteristics and cost.

The container for the phase change material (water) was designed with particular attention to the capacity and configuration. The importance of these factors will now be briefly discussed.

The capacity of the thermal (cold) storage is dictated by the availability, the cooling load and the capacity of the refrigeration system. In determining the cooling load the most important point is to generate enough cold storage to provide sufficient cooling, during the sunshine hours, so that cooling is maintained in the refrigerator cabinet for 36 hours. Obviously, the capacity increases with the thermal load but a good design takes care of this point. The design of the container is based on its ability to act as an effective heat exchanger. In this respect it must release all its stored energy effectively to maintain the desired temperature inside the refrigerator cabinet for a period of 24 hours for a sunshine duration of 7 hours.

6.5.6 Design and Evaluation of Performance of a Prototype

On the basis of the findings from all the points described earlier in this chapter a prototype was designed and its performance was evaluated under "real life conditions". This matter is described in this sub-section.

6.5.6.1 Choice of Capacity

It was initially intended to construct a 250 litre capacity prototype. However, the fact that it is intended to use the output d.c. power of a photovoltaic panel directly to power the refrigerator the question of a suitable compressor, with particular reference to availability and cost, had to be addressed very carefully. First of all, d.c. compressors are the only type that can be used. At the time of construction of the prototype it was not easy to find a d.c. compressor with a power rating about 280 Watts within our budgetary constraints. In fact, the question of cost, indeed, had to be considered very carefully as the potential commercial viability of the end product is a major theme of the present work. After some careful consideration modification of a 156 litre, based on the findings of the points described earlier, was thought to be the most suitable approach in the construction of the prototype.

The accurate size of the container for the thermal (cold) storage was calculated as follows:

The daily energy requirement=487Whr/day

Assuming the total sunshine hours to be 7 hours.

Then the required thermal energy storage is:

$$\frac{487\ (24-7)}{24} \approx 345\text{W.hr}$$

With the enthalpy of fusion of water being:

$$\frac{333.8\text{kJ/kg}}{3.6} = 92.7\text{W.hr/kg}$$

then the quantity of the water required:

$$\frac{345\text{W.hr}}{92.7\text{W.hr/kg}} = 3.72\text{kg}$$

It, therefore, becomes possible to maintain the refrigerator cabinet temperature around 4°C over a 24 hours period with only 3.72kg of thermal (cold) storage.

At this stage it is important to point out that as the ice thaws its specific heat changes. In other words the specific heat of ice is 2.1kJ/kg°C whereas that of water is 4.23kJ/kg°C. This, obviously, affects the rate of heat transfer. For this reason allowance must be made by increasing the thermal (cold) storage by more than 100% to provide more than twice as large a heat transfer surface area. The volume of the phase change material (water) used was, therefore, taken as 10 litres. Since this had to be accommodated inside the cabinet then the power rating of the compressor must be estimated for a 156 litre refrigerator. The question of the power requirement for charging phase change material will now be addressed.

In order to achieve a temperature of 4°C inside the cabinet under an ambient temperature of 40°C for a refrigerator with 125mm thick wall of polyurethane the energy requirement for a 156 litre refrigerator is about 487Whr/day. Assuming about 7 hours of solar radiation the cooling load is:

$$\frac{487\text{Whr/day}}{7\text{hr}} = 69.57\text{W}$$

Since 60W d.c. compressors were readily available at reasonable costs it was decided to use two 60W d.c. compressors in the prototype and investigate the performance. It has been envisaged that these compressors are adequate to cool the cabinet and charge the phase change storage.

The evaporator and condenser were chosen to be 200% and 300% the size for the conventional similar capacity refrigerator, respectively. The electronic thermostat switch used was a conventional Radionics switch. The recovery of available cooling from the suction line was secured by coiling the capillary tube inside the line. There was no alternation done to the refrigeration line and the conventional refrigerant was left intact.

6.5.6.2 Performance

The refrigerator was filled with perishable food items such as milk, butter, cheese, fruit, meat and fresh vegetables. A thermocouple was also placed inside the cabinet. The photovoltaic panels purchased were made of amorphous silicon. The panels were capable of producing 70W at 12volts d.c. for each compressor.

In the evaluation of the performance of the prototype the temperature of the ambient and the cabinet were recorded at regular intervals of 30 minutes. The refrigerator door was opened for 6 times during each run with each opening not lasting more than 1.5 minutes. The test was conducted for 15 runs. Each run lasted for 48 hours and at the end of every 7 hours of exposure of the photovoltaic panels to the sunshine the panels were disconnected from the refrigerator and remained so for 41 hours. At the end of the each run the cabinet door was left open until all the ice in the thermal storage container was fully thawed and its temperature rose to above 25°C. The temperature of all the foodstuff (apart from the milk and meat) was also allowed to rise above 25°C. The surface of the photovoltaic panels was cleaned at the start of each run. Figure 6.3 shows the variation of the temperature inside the cabinet during a 48 hour operation. It was observed that after 4 hours and 50 minutes of operation of the compressors the phase change material reached the predetermined status and switched off automatically. In other words the compressors need to operate for only 4 hours and 50 minutes to maintain the refrigerator temperature for a period of about 24 hours. The spikes in figure 6.3 represent the time when the refrigerator door is opened for about 1.5 minutes and then closed.

The graph clearly indicates that it is, indeed, possible to maintain the refrigerator temperature below 4°C with smaller size for the container of the phase change material for 24 hours duration. It is also important to point out that for lower ambient temperatures the case becomes more attractive.

Another important point to highlight is the potential for further improvement in the performance of such a system. This can be explained as follows:

The cooling load demand varies with ambient temperature. The output power of the photovoltaic panels varies in such a way that it is at its maximum during the mid-day hours. It is in fact during this period that the cooling load and the ambient temperature are at their maximum. For this reason a variable speed compressor will be a perfect match to the system. This is because the compressor speed could accelerate during high cooling demand periods. This way both the evaporator and condenser will be utilised for longer periods that will inevitably improve the coefficient of the performance. However, it proved impossible to source a d.c. compressor with variable speed when the research programme was in progress. An alternative approach to cater for variable output of photovoltaic panel is implemented in the research work described in Chapter 8.

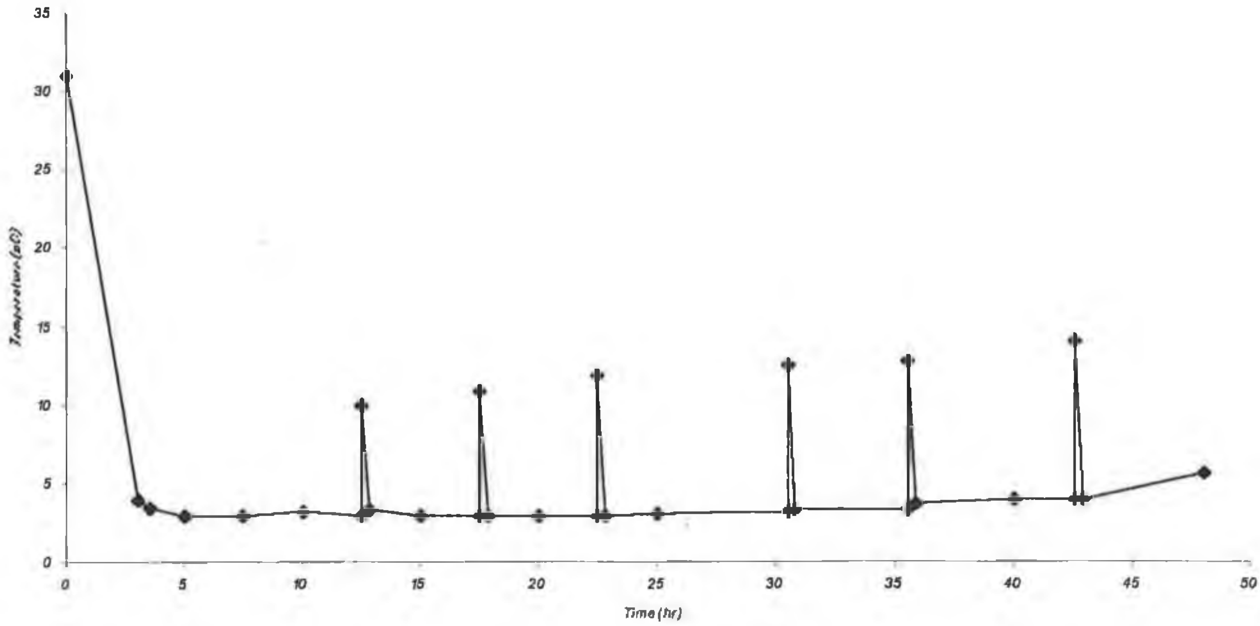


Fig. 6.3 Variation of the temperature inside the cabinet during a 24 hour operation

6.6 Discussion

As a result of the research work presented in this chapter it has now been firmly established that the approach adopted so far whereby conventional refrigerators are powered by a photovoltaic panels via battery and inverter is not the most cost effective and energy efficient approach in solar refrigeration. The presented discussion is, therefore, about how advantageous the solar refrigeration system based on the new design idea compared with the current approach described in this chapter. This is because the superiority of the new design over the systems using photovoltaic panels, battery and inverter is now a foregone conclusion. This point has already been addressed earlier in this Chapter both from the technological and costs point of view. The new design is undoubtedly a major step forward in improving the commercial potential of solar power refrigeration systems powered by photovoltaic panels for rural regions where the supply of mains electricity is either non-existent or very intermittent and unreliable.

Another important point to be discussed is how the new design refrigerator compares with conventional refrigerators using mains a.c. as their source of power. The discussion on this issue must, therefore, be centred on increased cost of the new design and savings that the reduced input energy brings about.

6.6.1 The Question of Increased Cost

Perhaps it would be most appropriate to address the question of the increased cost in terms of following points:

❶ Photovoltaic panels

❷ New design in relation to improved efficiency, and

❸ d.c. compressors

For the prototype designed and tested the photovoltaic panels were chosen to provide 140 Watts of power at 12 volts d.c. Although the efficiency of amorphous silicon panels are lower than that of the other types of photovoltaic panels the fact that the cost of the amorphous panels is about US$3.00/Watt compared to US$7.00/Watt for the nearest competitor made amorphous silicon panels very attractive. The costs for the amorphous photovoltaic panels to provide 140W of output power, and those of installation frames and brackets came to about €461 (or US$452 based on the exchange rate of €1 = US$0.98).

To implement the ideas of the new design necessitates usage of extra materials. However since the manufacturing procedure for the mass production of the new design will be very close to that of the conventional refrigerators there will not be a significant increase in production cost. It would, therefore, be reasonable to assume a cost increase of about 6%.

As far as d.c. compressors are concerned they are more expensive that their a.c. counter-parts. This is because they are not produced in as large of a volume as the a.c. compressors are. A full survey of costs of compressors showed that d.c. compressors, on average are about 20%-25% more expensive. Here again, in order to discuss the question of cost in a critical manner the upper figure of 25% is being taken into account.

6.6.2 Saving Due to Reduction in Energy Demand

It has been shown in this chapter that each stage of design improvement has resulted in a certain level of reduction in the energy demand. The most significant points are in relation to using thicker walls for thermal insulation, thermal (cold) storage, larger evaporator & condenser, and the heat recovery from the suction line. The reductions related to thermal insulation, larger evaporator & condenser and heat recovery are found to be 36%, 7.3% and 2.3%, respectively. The energy reduction in relation to thermal (cold) storage turns out to be a by-product of the design. For this reason it was not deemed necessary to present a figure. In other words, the major breakthrough here is the smart design that has facilitated the required cooling power for the refrigerator to function satisfactorily over a 24 hour period.

6.6.3 Final Cost Calculation

The overall cost discussion can be based on the argument that once the photovoltaic panels are paid for and installed there will be no running cost associated with them. It would, therefore, be necessary to see whether the saving associated with the elimination of the running cost of the refrigerator has any significant impact on the commercial viability of the new design.

A conventional 150 litre refrigerator requires a maximum of 1400 W.hr/day in area where the ambient temperature varies between 20°C to 40°C. If the cost of the electricity required to run a 150 litre refrigerator in Ireland is taken into account as the basis for calculation then for the case in hand annual running cost and the calculation leading to potential viability can be presented as follows:

The annual power requirement = 1400 W.hr/day×365 = 511 kW/year

The cost of each kW.hr (kilowatt.hour) for domestic users of energy = €0.12 after taking into account the rate, service demand charge and service capacity charge as specified by the Electricity Supply Board. On this basis the saving will be €64.9 per year.

The 6% increase in the cost for implementation of the new design ideas must not be worked out against retail price of a conventional refrigerator. Instead it must be worked out against the cost of materials used in mass production. For the retail price of €305 the cost of the materials for each refrigerator cannot be over €102. The 6% extra cost is, therefore, around €6.10.

For the d.c. compressor the extra 25%, which is the current figure for the difference between d.c. and a.c. compressor, is about €31.75.

The extra cost involved, therefore, is the sum of the following figures:

€432 (for the panels) + €6.1 (for the new design) + €31.75(for the d.c. compressors) = €469.85

The saving per year =€64.9

The payback time = 7.2 years.

It must be noted that the above figure has arrived at without taking a discount-cash-flow analysis.

6.7 Conclusions

It has been established that there is, indeed, a more practical and reasonable lower cost approach to the way photovoltaic panels are being used to power conventional refrigerators.

As far as the comparison between this new solar powered refrigeration system and the conventional a.c. mains powered refrigerators is concerned the cost of the new system is far too expensive for the new design to be considered as a competitor. This is because the cost of the new system is about 154% more expensive compared to the conventional refrigerators. The fact that there is no running cost associated with the new design makes the design attractive up to some point but the payback time of 7.2 years is still too long to make it a viable commercial proposition. However, it does not mean that the new design can be disregarded completely. This is because the useful lifetime of the photovoltaic panels is over 20 years and the useful lifetime of conventional refrigerators is normally about 15 years. In this context the fact that there will still be at least over 8.3 years of life left in the system after the payback time gives the new design strong merits as far a commercial viability is concerned.

There are also a number of important points that need careful attention. The new advances in the manufacturing of photovoltaic panels promises a cost per watt in the region of around US$1.00, due to the new Cadmium Telluride cells. This will undoubtedly bring the overall cost down by about €280 and the payback time goes down to 3.84 years. This figure speaks for itself in terms of commercial viability. Another point that is worth pointing out is that if a greater demand develops for d.c. compressors then their costs will inevitably come down. It would, therefore, be reasonable to claim that pay back times of about 3.3 years are not too far away. Finally, in rural areas where there is no reliable supply of mains electricity the new design, indeed, offers the most attractive option in refrigeration.

Chapter 7

Design and Development of a Stand-Alone Multi-Cycle Intermittent Solar Powered Block Icemaker Plant

Chapter 7

Design and Development of a Stand-Alone Multi-Cycle Intermittent Solar Powered Block Icemaker Plant

7.1 Introduction

Nowadays the idea of solar powered cooling and refrigeration is one of the most attractive ideas in the field of application of solar energy not only in the developing countries but also in the developed countries. It is therefore, easy to see why the idea of solar powered cooling and refrigeration has attracted so much interest among the researchers and technologists.

In view of the fact that solar energy is available on an intermittent basis a great majority of all the works carried out so far have been concerned with intermittent systems. In these systems various conventional cooling and refrigeration systems were modified to operate in conjunction with the intermittently available solar radiation. The simplest of all the conventional cooling and refrigeration systems to be adopted were those using liquid absorbents such as water-ammonia, or lithium bromide-water and physical absorption systems such as methanol in activated charcoal. Once again, it has to be emphasised that the only reason for the usage of these systems were their simplicity and certainly not their coefficient of performance (COP). The highest COPs reported for all the above systems have been reported [153-158] to be in the range of 0.02 to 0.12.

Systems using physical absorbents such as calcium chloride in conjunction with ammonia, on the other hand, did not receive the same initial interest among the researchers despite their higher inherent COPs. This was primarily because of a number of practical problems associated with such refrigeration systems using solid absorbent such as calcium chloride in conjunction with ammonia. However, in more recent works these problems have been reported to be solved [159].

It has now been realised that intelligently designed systems with better COPs are the only way forward if the question of solar powered cooling and refrigeration is to be taken seriously. The higher COP will offset any marginal initial extra cost.

In this chapter design and performance of a new novel solar powered multi-cycle intermittent ice making system is described. The novel system uses an advanced evacuated solar heat collector in conjunction with calcium chloride-ammonia mixture and ancillary photovoltaic peripheral technology. Increased yield and relatively low cost are the most advantages features of the new novel design.

7.2 Background to Refrigeration Systems Using Calcium Chloride – Ammonia Mixture

The history of the application of calcium chloride-ammonia mixture for refrigeration dates back to the late 1920s. A number of practical problems had to be tackled at the time in Research and Development stage. The problems as outlined by Chinappa [159] are:

1. Poor heat transfer to and from the solid adsorbent
2. The solid absorbent swells up to 3-4 of times its original size when it absorbs ammonia
3. The absorbent tends to lose its crystalline structure and break up into powder, which makes subsequent absorption difficult.
4. The solid absorbent is packed into regions less accessible to ammonia, which causes a decrease in absorbing capacity with cycling.

The system proved very reliable once all the above problems were sorted out by the early 1930s. Siemens-Protos successfully manufactured and marketed household refrigerators using intermittent absorption systems using calcium chloride – ammonia mixtures in the 1930s and 1940s. The system was abandoned in the late 1940s when the so-called Electrolux patents expired.

7.3 A Brief Theoretical Description of the Calcium Chloride – Ammonia Cooling and Refrigeration Systems

Calcium chloride absorbs a total of eight moles of ammonia in four steps. The steps are:

i- $CaCl_2 + NH_3 \Leftrightarrow CaCl_2(NH_3)$

ii- $CaCl_2(NH_3) + NH_3 \Leftrightarrow CaCl_2(2NH_3)$

iii- $CaCl_2(2NH_3) + 2NH_3 \Leftrightarrow CaCl_2(4NH_3)$

iv- $CaCl_2(4NH_3) + 4NH_3 \Leftrightarrow CaCl_2(8NH_3)$

In the cases related to all solar powered cooling and refrigeration systems reported so far the source of heat has been flat plate collectors. In view of the limited energy conversion efficiencies associated with flat plate collectors and the moderate ultimate obtainable working temperatures only six moles can be cycled, working with flat plate collectors as their source of energy. In this respect only reactions iii and iv described above are applicable to solar powered cooling and refrigeration using flat plate collectors. This phenomenon is shown in figure 7.1.

The heat of reaction is approximately twice that of the latent heat of vaporisation which sets an upper limit for the obtainable coefficient of performance (COP) of the process of about 0.5. The actual COP is, however, lower because of the losses caused by the heating of the absorber from the absorption temperature to the desorption temperature.

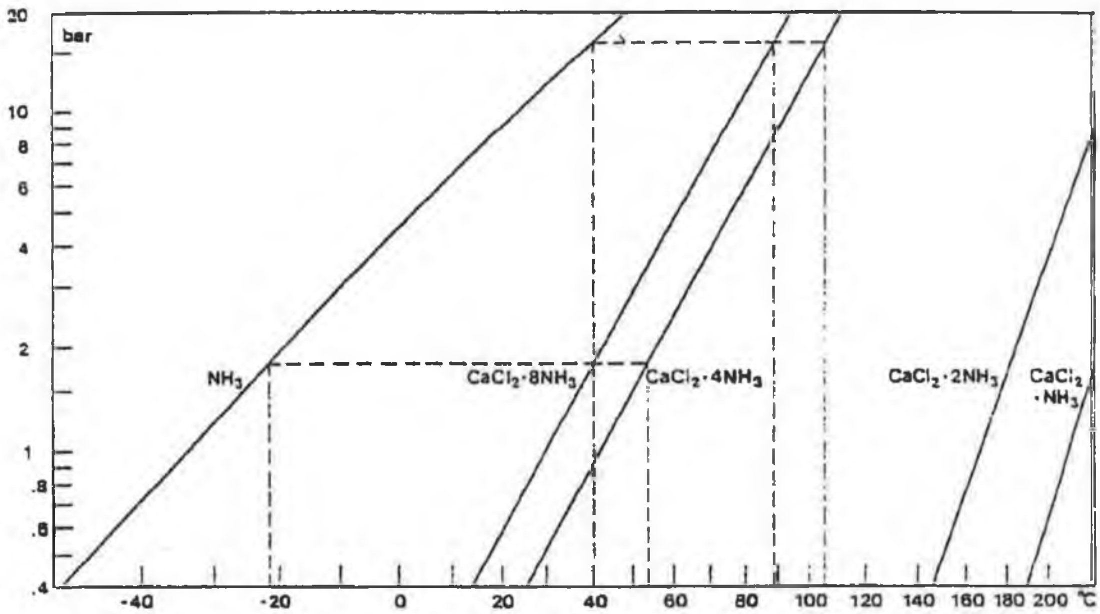


Fig. 7.1 Calcium chloride – ammonia equilibrium

As shown in figure 7.1 the reaction temperatures are determined by the pressure in the system. The desorption temperatures are somewhat higher than those needed in a water-ammonia system, and one does not have the freedom, offered by latter system, of choosing the concentration according to the specific evaporation and condensation temperatures. In view of the fact that the thermal capacity of the calcium chloride is smaller than that of the strong solution of water and ammonia mixture a higher COP can be obtained with the calcium chloride – ammonia system. Unlike water - ammonia, or other systems with liquid absorbent where the COP is very sensitive to changes in the evaporation and condensation temperature, the COP of the calcium chloride – ammonia system is only slightly effected by such changes.

7.4 Systems Using High-Efficiency Evacuated Solar Heat Collector Tubes

In order to obtain sufficiently high rates of reaction in calcium chloride-ammonia mixture a certain superheating of the absorbent during desorption is necessary. This point is also shown in the equilibrium diagram of figure 7.1. In order to detach and cycle the two strongly bound moles to the solid, temperatures excess of 200°C are required. It is, indeed, for this particular reason that flat plate collectors are not capable of cycling the last two moles.

The high efficiency fast response evacuated solar heat collectors developed at the Institute of Technology, Sligo is capable of supplying operating temperatures of over 250°C, routinely. The solar selective absorber material used in this type of collector is the highly stable cobalt oxide on aluminium, developed at the Indian Institute of Technology, Delhi [160]. It produces stagnation temperature over 300°C. This type of collector is ideal as the source of heat for calcium chloride – ammonia mixtures. The application of this type of collector has been the main step in the design of the novel multi-cycle icemaker. The excellent results obtained very clearly prove a major breakthrough brought about by using this type of collector.

7.5 An Overview of the Novel Intermittent Stand-Alone Multi-Cycle Solar Powered Block Ice Plant

The design idea leading the development of the novel multi-cycle stand alone intermittent solar powered block ice plant stems from the successful trial of a single cycle intermittent system developed by Sunice company [161] and tested in Gambia in 1989. In this system 12 modules of generator using double glazed flat plate solar heat collector were used. These collectors were capable of delivering heat at 90°C to the calcium chloride – ammonia mixture. Each module contained 4.2kg of ammonia and the system was capable of production of 100kg of ice per day. In view of the fact that 4.2kg of ammonia is sufficient to freeze about 10kg of ice the output of the system proved very satisfactory. In this analysis the output was over 83% of the expected 120kg of ice per day.

Due to technological advancements in the fields of stable selective absorber material and in the design and development of highly efficient evacuated solar heat collectors at Institute of Technology, Sligo it was decided to explore the possibility of developing the novel multi-cycle system.

7.5.1 Brief Description of the Overall System

A schematic diagram of the overall system is presented in Figure 7.2. The system comprises the following parts:

i- Generator modules with each module comprising 10 evacuated solar heat collector tubes. Each collector tube contains the housing for the calcium chloride – ammonia mixture and the evaporator of an internal cooling system using R11 refrigerant.

ii- Condensers for ammonia circuits with air coolant pipes welded to its outer surface.

iii- Condenser of R11 circuit for the internal cooling with air coolant pipes welded to its outer surface

iv- A bi-stable solenoid valve to be used for the starting and stopping of the internal cooling for the commencement of absorption and desorption, respectively.

v- Evaporator for the Ammonia Circuit (block icemaker part).

vi- A flow meter to record the flow rate of liquid ammonia into the reservoir before it is allowed to flow into the evaporator of the block icemaker plant.

vii- The housing of the plant

viii- Pond of water for storage of cold water to help to cool the condensers of ammonia and the internal cooling circuits.

ix- Air circulating pumps to cool air from the inside of the pond to the condensers described in (ii) and (iii) above.

x- Electrical system comprising of photovoltaic panels, battery and timer circuit.

xi- An electronic integrator to signal the end of the desorption cycle and the start of the refrigeration.

xii- Shading system to prevent the generator from receiving solar irradiation during absorption of vaporous ammonia, and

xiii- Bi-stable solenoid stoppers.

The function of the parts listed above are described in Section 7.6

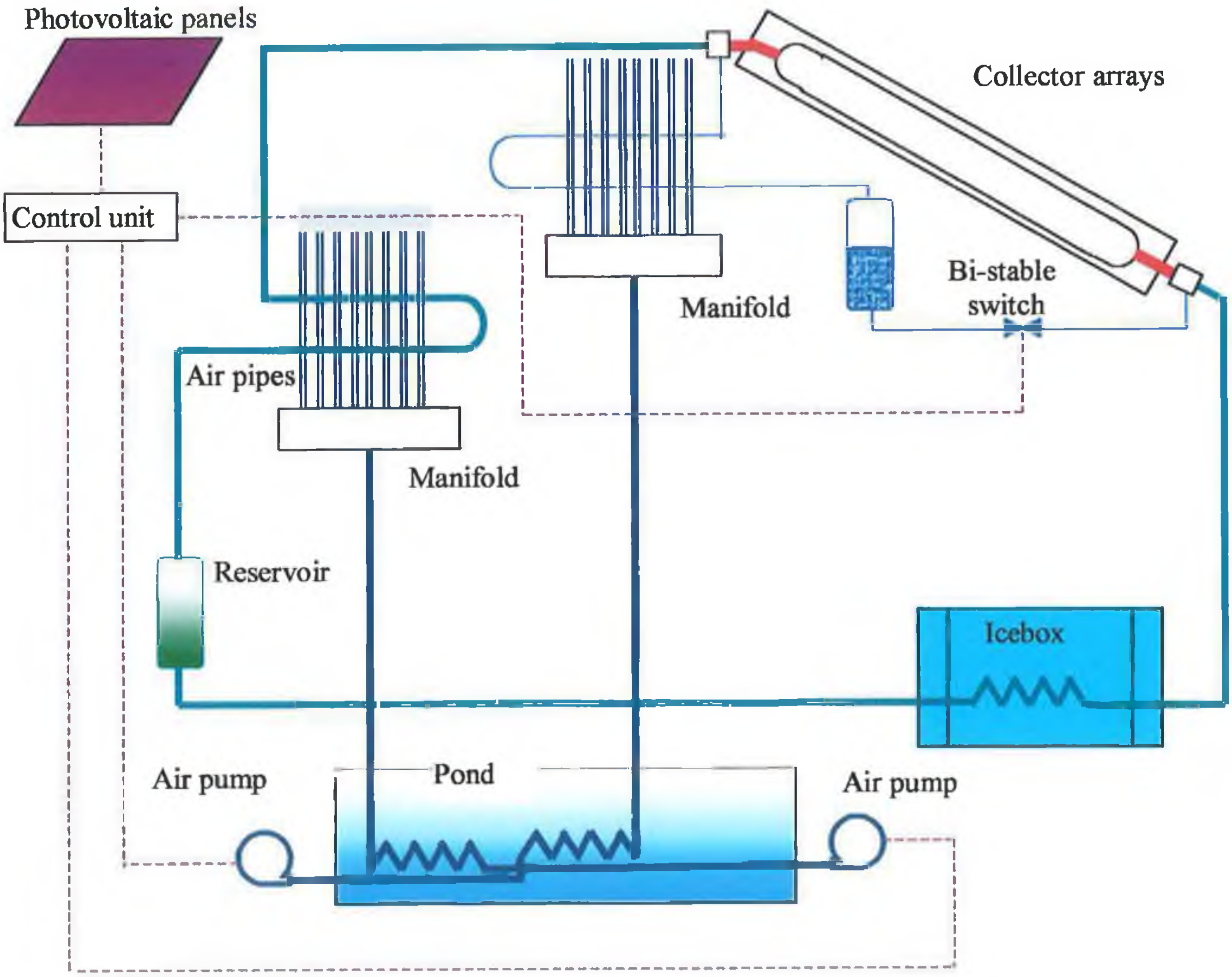


Fig. 7.2 The schematic diagram of the system

7.6 Description of the Various Parts of the Plant

7.6.1 The Generator Tube

The term generator in this context is referred to as the part of the plant where ammonia is boiled off the solid – ammonia mixture. Since it is intended to develop a multi-cycle system the generator must be capable of boiling off all the ammonia from calcium chloride – ammonia mixture in the shortest possible time. For this reason the mixture of solid – ammonia has to receive the maximum possible thermal energy from the solar irradiation. In view of the fact that there is considerable expertise present at the Institute of Technology, Sligo in high-output evacuated solar heat collectors an evacuated tube generator was the most obvious option. The housing for the solid - ammonia mixture is an extruded aluminium profile of the shape shown in figure 7.3. The length of the profile is 1.9m. The inner diameter of the inner tube is 15mm and its wall thickness is 2.5mm. It is connected to the outer tube by 6 fins. The thickness of each fin is 2.5mm. The outer tube has an O.D. of 80mm and its I.D. is 75mm. The space between the inner and outer tube is the housing for the solid – ammonia mixture. The inner tube is for cooling purpose when the solid is to be cooled down to appropriate temperatures in order to facilitate absorption of the ammonia vapour at the end of each cycle. The outer surface of the extruded aluminium is coated by cobalt oxide selective coating. There are three reasons for the use of aluminium as the profile. They are:

Fig. 7.3 The housing for the solid - ammonia mixture

❶ It can easily be produced by extrusion

❷ Its low cost, and

❸ Cobalt oxide selective coating can easily be deposited on aluminium.

The above-mentioned dimensions for the housing of the solid – ammonia were chosen in order to allow for the expansion of the calcium chloride when it absorbs the ammonia. As pointed out earlier its volume can expand to 3-4 times of that its original volume (when it is dry) as it absorbs ammonia. In order to be on absolute safe side and facilitate repeated cycles of usage of the solid over many years the housing is designed to tolerate expansion of as high as 10 times. This also increases the collector area which in turn accelerates the desorption of ammonia from the solid during the desorption phase. The choice of dimension for the inner tube and thickness of the fins have been based on providing the necessary cooling power for the dry solid to facilitate efficient absorption.

The problems associated with the interaction of calcium chloride and ammonia that can adversely affect the solid as outlined earlier in this Chapter has been addressed using the approach suggested by Iloeje [162]. This is achieved by making a mixture of calcium chloride and Portland cement (Ca_2SO_4). The mixture was 80% calcium chloride and 20% Portland cement. The aluminium profile containing the solid was placed inside a 2.1m long borosilicate tube of 100mm O.D. The wall thickness of the glass tube was 2.6mm. Two concentric stainless steel bellows of 15mm and 80mm I.D. where connected (by ultrasonic welding) to either end of profile. A nickel chromium iron alloy flange with an 80mm hole to accommodate the 80mm O.D. bellows were used at either end of the glass tubes. The flange was TIG welded to the 80mm O.D. bellows. The flange was then fused to the neck of the glass tube at either end of the glass tube using thermal compression technique for a hermetically sealed glass to metal fusion. The volume between the extruded aluminium part and the inner wall of the glass tube was evacuated down to 2×10^{-5} mbar and then sealed off.

Ten of these evacuated tubes with a distance of 120mm between them were mounted in an aluminium box covered by a pane of glass to form one module. Each generator tube was integrated with a highly polished involute aluminium reflector to reflect the solar irradiation to the to the back of the aluminium profile of generator tube. The arrangement of a tube and a reflector, known as one Generator Unit, is shown in figure 7.4. The reason behind using the glass pane was to facilitate cleaning and removing dust on regular basis without damaging the highly polished surface of the reflector and also the generator tubes. It must be pointed out that if the modules were to be used without any protection against dust and dirt then it would be necessary to clean the reflectors and the generator tubes regularly and frequently. This would, especially, result in scratching and adversely effecting its reflecting characteristics of the reflectors. A cutaway view of generator tubes in a module and a fully completed module are shown in figures 7.5.a and 7.5.b.

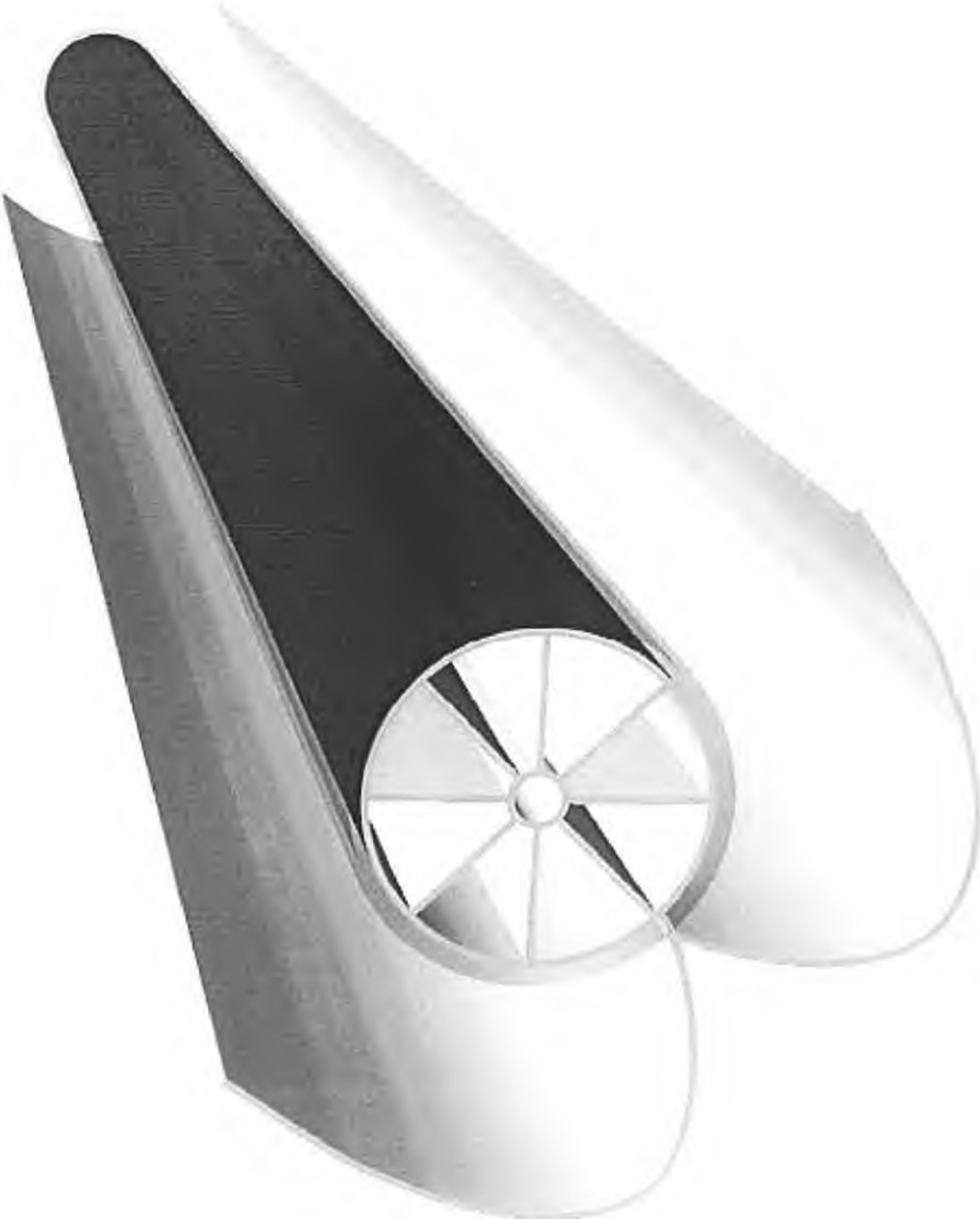


Fig. 7.4 The generator Tubes and the reflectors

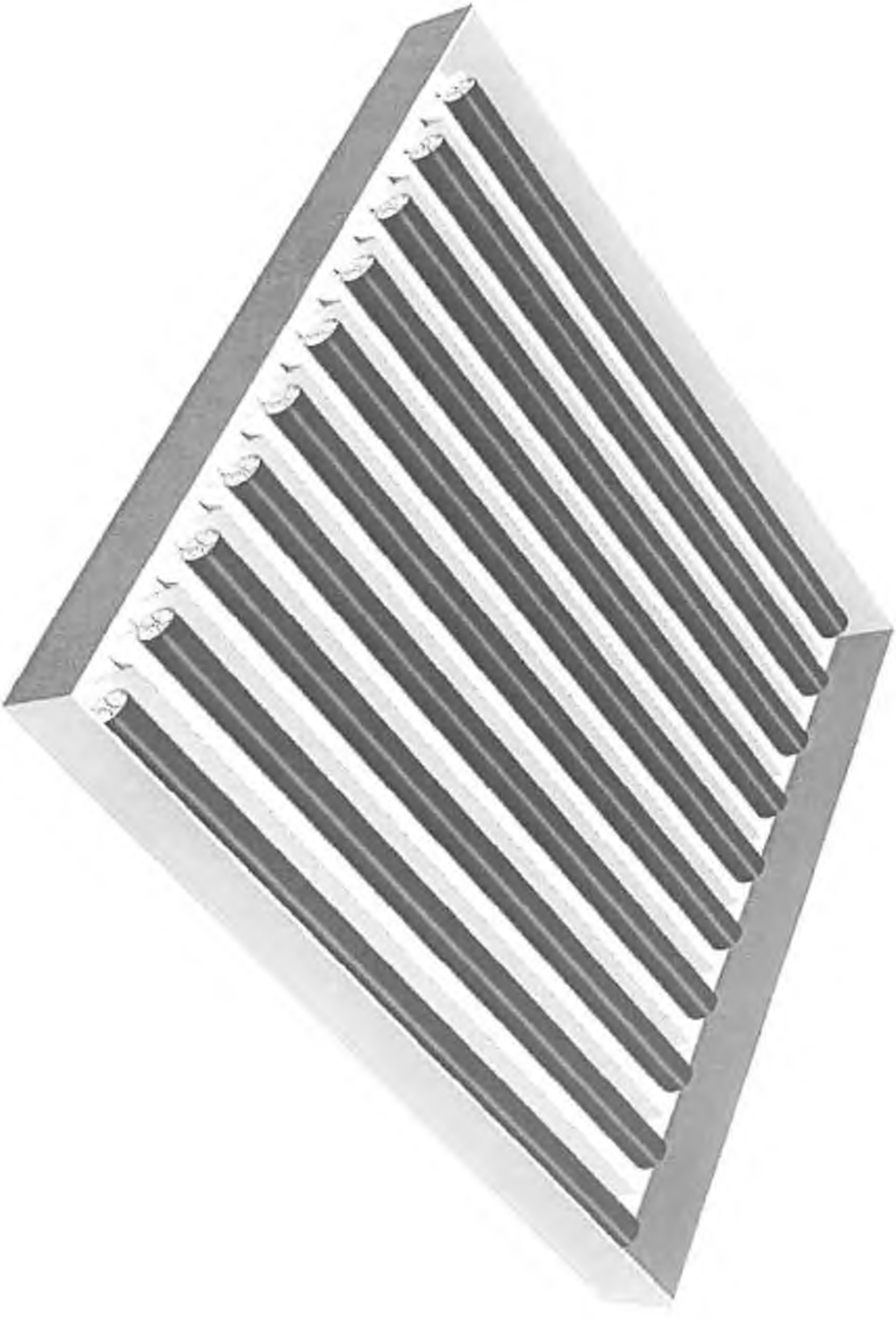


Figure 7.5.a A cutaway view of generator tubes in a module

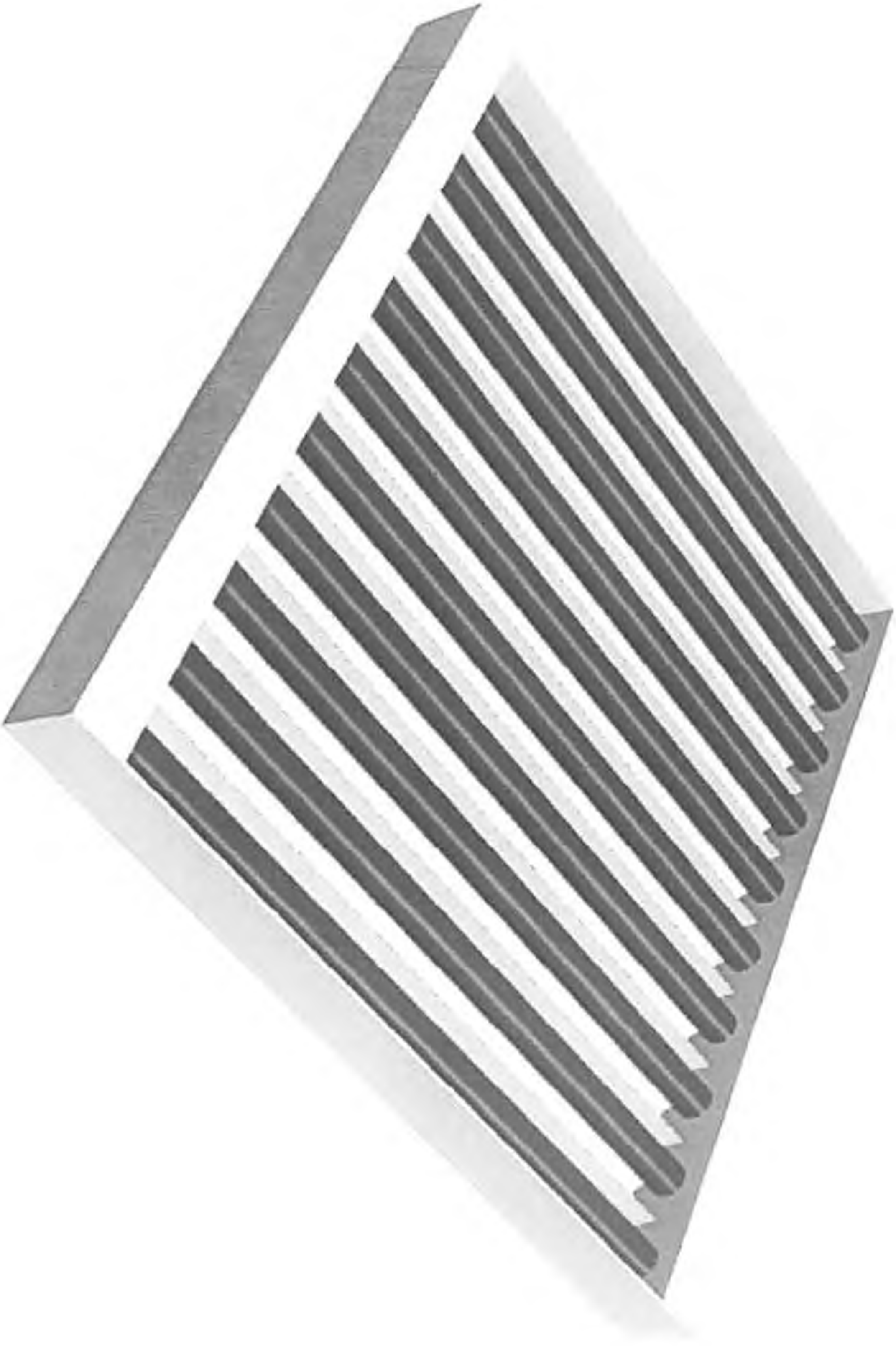


Figure 7.5.b   A fully developed generator module

7.6.2 The Refrigeration Circuit

A simple illustration of the block icemaker is schematically depicted in figure 7.6. A cellar type room of about 5 metres under outside ground level is an integral part of the icemaker. The "generator module" and its shading screen together with the photovoltaic panels are mounted on the top of the cellar housing (A full description of the shading system is presented later in this subsection). The reason behind the usage of cellar room is the fact that the temperature inside the cellar is always in the range of 16-20°C while the ambient temperature outside the cellar is about 32°C in the shade at ground level. The cellar, therefore, provided a unique cooling facility to the refrigeration circuit were placed inside cellar:

❶ the condenser of the internal cooling circuit

❷ the reservoir of the refrigerant of the internal cooling circuit,

❸ the condenser of the ammonia circuit,

❹ the reservoir of the ammonia circuit

❺ the evaporator of the block ice maker (the ice box), and

❻ a pond containing 5 cubic metres of water and two submerged air heat exchanger metallic coils at the bottom of the cellar.

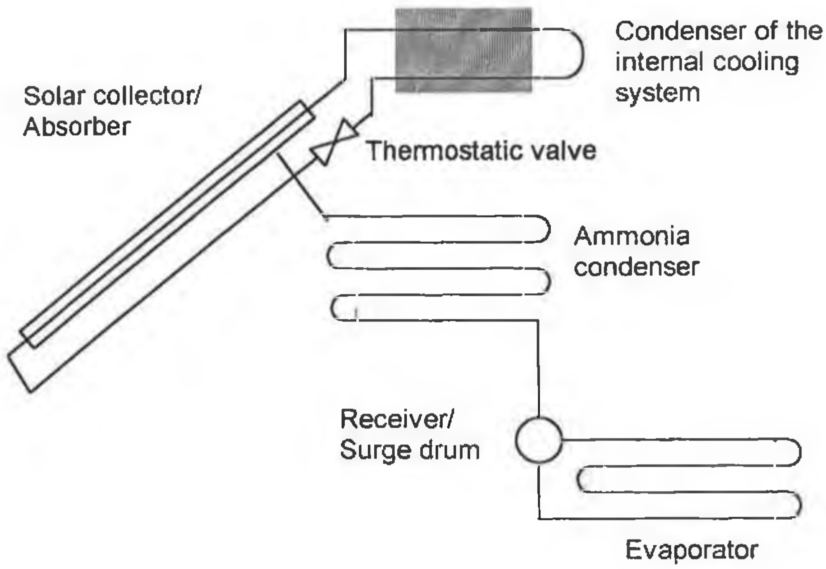


Fig 7.6 The principle of the operation of the absorption system

Metallic pipes at a distance of about 25mm apart were welded to the outside wall of both condensers. The bottom ends of these pipes were left open. Each manifold was supplied with cool air from the submerged heat exchanger coils from the pond at the bottom of cellar. This arrangement facilitated as a very effective cooling mechanism for the condensers. Two d.c. air circulating pumps powered by photovoltaic panels provided the coolant air to the condensers.

The ammonia vapour generated during the desorption phase in the evacuated generator tubes is liquefied in the condenser and finds its way, by gravity, to the ammonia reservoir before it is allowed to flow into the evaporator of the block ice maker. The refrigerant vapour emerging from the internal cooling pipe of the generator tubes is liquefied in the condenser and then finds its way, by gravity to its reservoir.

The liquefied ammonia goes to the receiver and evaporator that are placed at the bottom of the cell. A flow meter is placed at the lowest point of the condenser coil. The output of the flow meter is connected to an integrator to measure the quantity of the liquefied ammonia that flows to the evaporator.

In order to facilitate absorption of the ammonia vapour, emerging from the evaporator of the icemaker circuit, by the calcium chloride inside the evacuated generator tube it is necessary to cool the calcium chloride to temperatures in the range of 25°C or less. This is achieved by incorporating a separate cooling system. The evaporator of this cooling system, that has been called the “internal cooling”, is the 15mm I.D. finned tubing inside the extruded aluminium housing of calcium chloride. As pointed out earlier, the condenser of this cooling system is also in the cellar housing. This condenser is also air-cooled the in the same manner as the condenser of the ammonia circuit. A 4mm I.D. pipe connects the reservoir of the internal cooling circuit to the bottom of the internal cooling pipe in the extruded aluminium profile via a bi-stable valve. The cooling is switched on by the opening of the bi-stable solenoid valve connected between the condenser and the evaporator. As the valve opens the liquid refrigerant is fed to the bottom of the internal cooling tube. As the liquid travels to the 15mm I.D. tube from the 4mm I.D. pipe it rises up and is turned into vapour. This action generates the required internal cooling.

7.6.3 Shading System

By 10 am on a good day all the ammonia is boiled off from the generator modules and at this time it is appropriate to activate the internal cooling refrigeration cycle. This can be achieved primarily by preventing the evacuated generator tubes from receiving any solar radiation. In order to do this the modules must be shaded completely so that no solar incident reaches the evacuated tubes.

The shading is facilitated by a screen made of 2.2m long 20mm O.D. hollow fibreglass tubes of 1mm wall thickness. There are 2 grooves at either ends of these pipes and they are at a distance of 10mm from the nearest end. The depth and the width of the grooves are 0.25mm and 2mm, respectively.

These grooves are used to interconnect the fibreglass tubes as closely as possible to each other by a nylon cord. This arrangement provided the most ideal flexibility to the screen as it moves on or off the module.

Wooden pulleys were mounted along both sides of the module, at distances of about 74cm from each other. Both ends of the fibreglass tubes rest and slide over these pulleys and the middle part of the tubes on an aluminium rod with inverted U-shaped supports to prevent sagging of the tubes. This arrangement provided the ideal structure for the positioning and movement of the screen over the modules. Figure 7.7 shows the arrangement of the above parts for ideal positioning and movement of the screen on/or off the module. This structure also prevents the movement of the fibreglass shading screen to the right or left. In other words it takes care of careful positioning of the screen on the module frame. Seeing as there are some small gaps in between the fibreglass pipes of the screen there will be a small amount of solar irradiation that could reach the module tubes, from the gaps between adjacent fibreglass while the screen is on. In order to prevent this event a second screen of similar type and support structure was placed right above the first screen. With this arrangement no direct sunshine will reach the generator module.

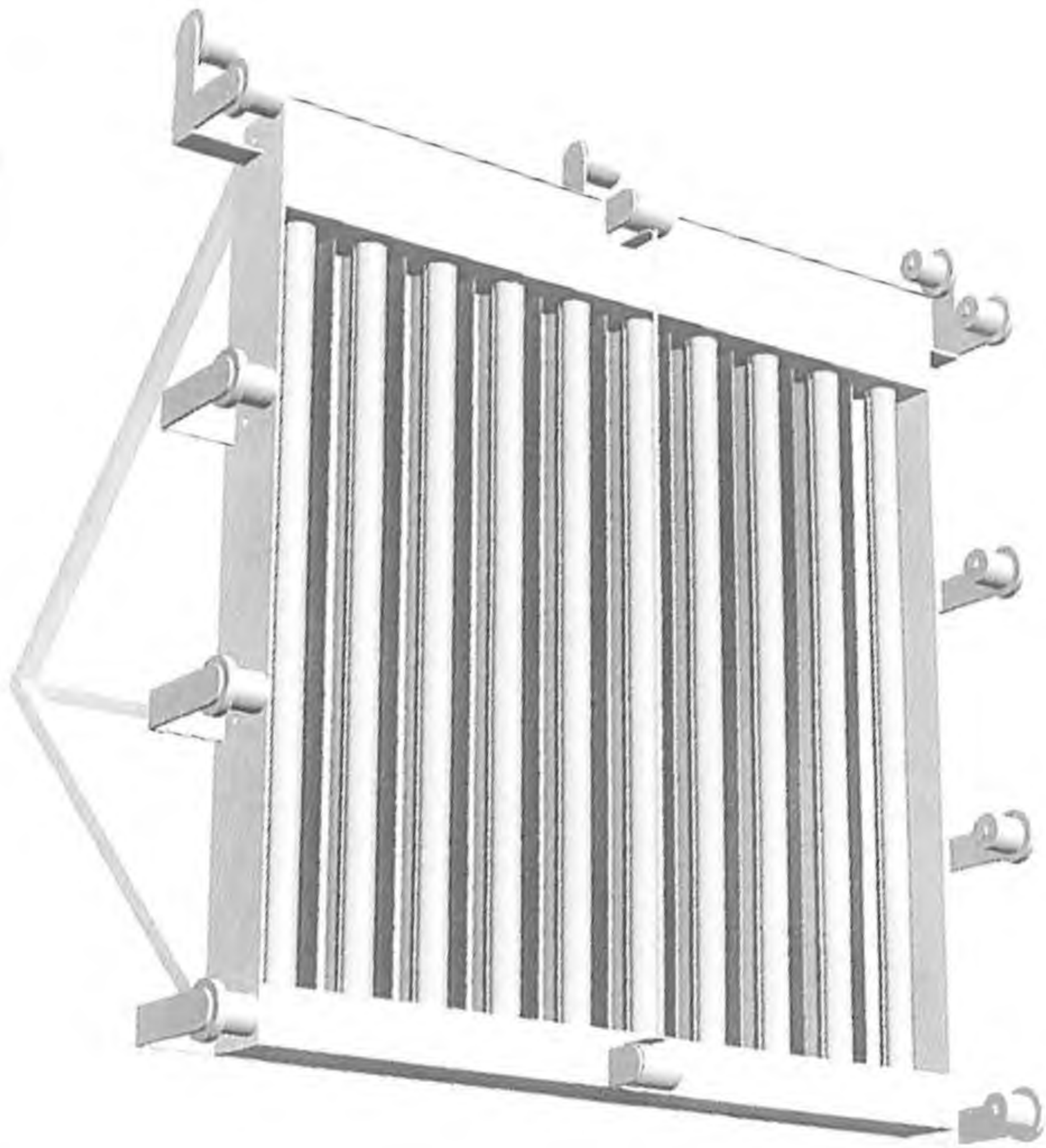


Fig. 7.7 The shading screen support structure with the arrangement of the pulleys

The movement of this screen over and away from the module is secured by dropping a spherical weight of 5kg to one of the baskets connected to each end of the shading system. Each screen, therefore, is connected to a basket at each end to accommodate the weight in question

Perhaps it would be best to describe the operation of the mechanism used to drop the weight into the appropriate baskets and makes the screen move on or away from the modules by referring to the diagrams shown in figures 7.8.a and 7.8.b.

Once the need for the shading arises an electrical signal will be received at the "holding station". The holding station consists of a holding pipe, a standby box, a standby shell and the outlet door. The holding pipe is at a slight angle with respect to horizontal, and it contains 3 of the spherical weights. A small stop is mounted at the end of this holding station. The wall at the outlet of this box is hinged and forms a door that pivots around the hinge. It can be closed or opened by moving a small stopper rod that is activated by a bi-stable solenoid.

The spherical ball in the box sits inside the "standby" shell. This standby shell is a segment of spherical shell whose inner diameter is almost equal to the outside diameter of the spherical weight. The depth of the shell is roughly equal to half of its radius. This shell is manufactured from casted aluminium with a rod protruded from the centre of the outside wall of the shell, in one piece. The shell can therefore pivot around the central hole. The standby box contains the standby shell and the outlet door. There are two 10mm diameter holes along the central line of the bottom of the box. The holes are situated in the direction of movement of the spherical weights from the holding pipe to the outlet door of the standby box. One of the holes is positioned at the exit side of the station box and the other one is at its inlet side. In the stand by mode the stopper at the exit side is up and the one at inlet hole is down. When the need to move the screen arises the stopper at inlet moves up and, simultaneously, the one in exit move down. The stoppers are 7mm rods and are connected to a mechanism that pivots in such a way that the action of one is 180° out of phase with that of the other. The stopper mechanism is activated by the bi-stable solenoid. Therefore, in the stand by mode, when there is no need to move the screen, the positions of the stoppers at the bottom of the standby shell are as follows:

❶ the one nearest to the inlet of the standby box is down, and

❷ the one nearest to the hanging door of the basket is up.

In this arrangement the standby shell is in such a position that prevents the ball from moving toward the hanging door of the standby situation. When the time for moving the screen arrives, the position of the stopper changes from the standby mode and the ball moves to the basket. The distance by which the up and down movements of the rods are adjusted is carefully selected so that the weight can easily drop into the basket of the moving screen which is of the same size as the standby box of the standby station mentioned above.

The basket has an open inlet wall and the bottom of this basket has a square piece of aluminium of 15mm thickness that can pivot around an axis fixed to the sidewalls of the basket that is an integral part of the basket.

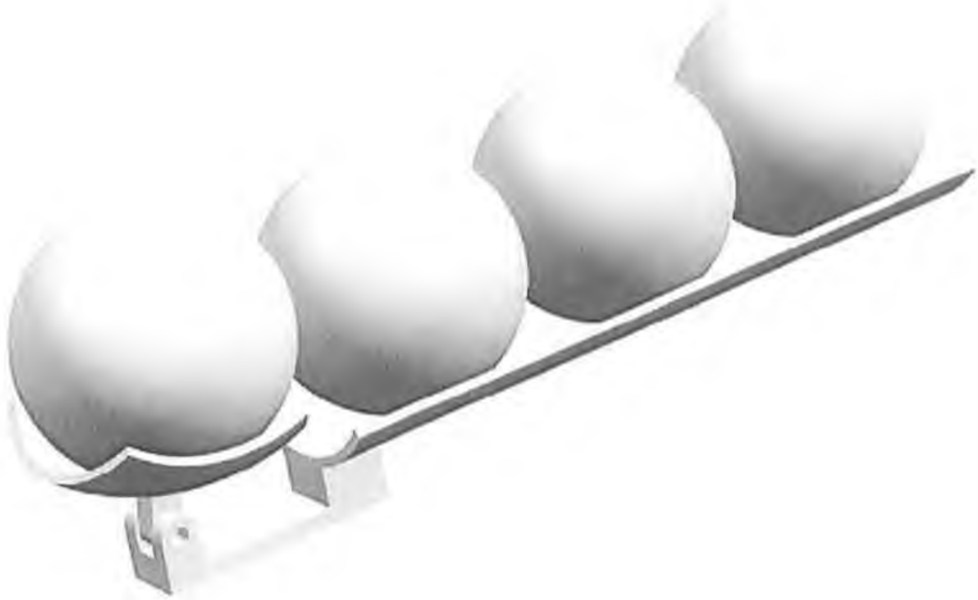


Fig. 7.8.a The position of holding station prior the need for moving of the shading screen arises

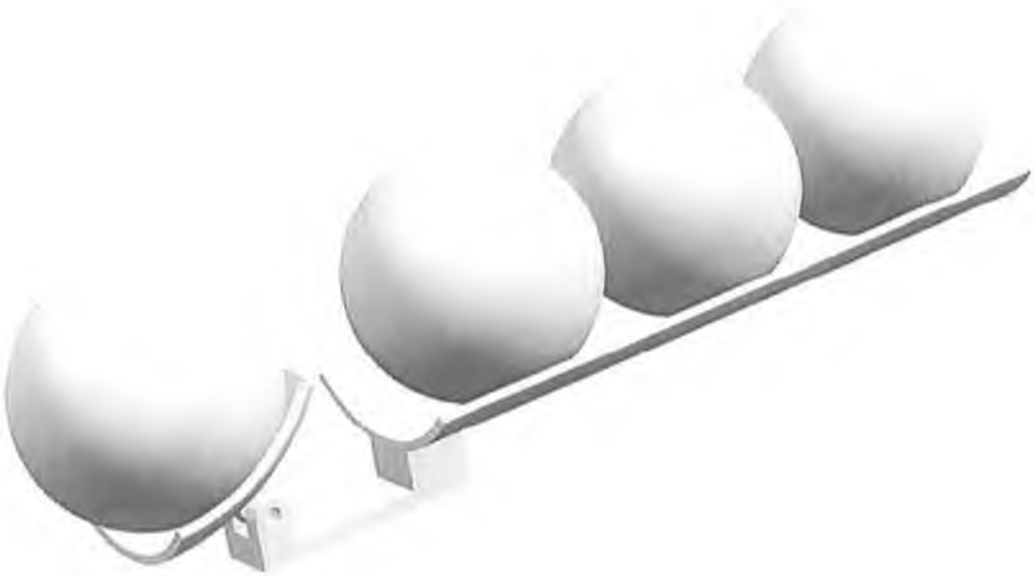


Fig. 7.8.b The position of holding station when the need for moving of the shading screen arises

This basket is placed into a long vertical wooden box and as the weight is dropped in it, it moves down the long vertical box and the distance that it should move is worked out to be sufficient enough to allow the screen to cover the entire generator module. Figure 7.9 shows the position of the ball in the basket as it falls into the basket and travels along the vertical box. There is an opening on one face of this long box where the basket is to arrive at the end of its downward journey. A similar size opening is at the bottom of the opposite face of the long box to allow the weight to leave the basket and long box. At the bottom of the long box there is a stopper rod that helps the bottom floor of the basket to tilt towards the outlet opening of the long box. This means as the basket reaches the end of the distance it has to travel the spherical weight easily finds its way out of the basket and the long box. Figure 7.10 shows the position of the ball and the aluminium sheet as the basket reaches the end of the long box. The screen remains over the generator module up to the end of the refrigeration cycle which is worked out to be about 80 minutes. A similar arrangement is used at the opposite end of the screen to move it away from the generator and starts the generation cycle all over again. With this arrangement it is then possible to activate more than one full refrigeration cycle during the day. Figures 7.11.a and 7.11.b show a generator module incorporated with the shading system before and after the shading cover moves over it the module.

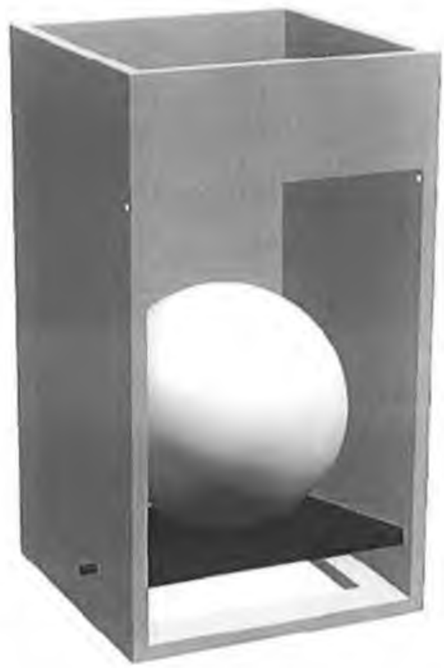


Fig. 7.9 The position of the ball and the revolving sheet as the ball falls into the basket

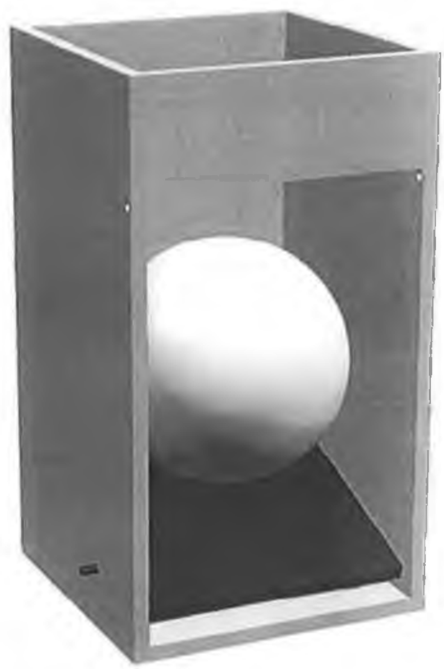


Fig. 7.10 The position of the ball and the revolving sheet as the sheet hits the stand bar at the end of long box

Fig. 7.11.a The position of the shading cover at the end of the absorption phase and start of the desorption phase

Fig. 7.11.b The position of the shading cover at the end of the desorption phase and start of the absorption phase

7.6.4 The Electrical Part

There are a number of components that are actuated by d.c. electrical signal in the block icemaker. In view of the fact that the system is intended to be used in remote areas where the supply of mains electricity is highly unreliable at best and non-existent under most circumstances, it was decided to use photovoltaic panels in conjunction with a battery storage system. As photovoltaic systems are very expensive the electronic systems were designed specially to consume the least amount of power and for the shortest possible time. This has been achieved by using the bi-stable solenoid relays and interlinking them to each other as much as possible to reduce power consumption. Furthermore, the photovoltaic panels used are amorphous silicone panels with proven track record of stability. The battery used is a long-life tractor battery which can be used in conjunction with panels without any adverse reaction.

7.7 Technical Data

7.7.1 Quantity of Solid in Each Generator Tube

As pointed out earlier in this Chapter special care is needed to make sure that the solid calcium chloride does not undergo degradation as a result of the generation of ammonia and refrigeration. This question has already been addressed by Ileoje [162]. For this reason calcium chloride was mixed with Portland Cement to form a solid mixture with 80% calcium chloride and 20% Portland Cement. Since the volume of calcium chloride can increase 3-4 times that of its dry state as a result of absorbing ammonia special attention was paid to the quantity of the solid mixture placed in the housing of the solid in the evacuated generator tube. For this reason the volume of the mixture of dry calcium chloride and Portland cement for each generator was calculated to be 702cm^3. With six separate parts in the housing for the solid ammonia mixture in each generator tube about 117cm^3 of dry solid mixture was placed in each subsection of the housing of each generator tube.

7.7.2 Quantity of Ammonia in Each Generator Tube

The quantity of liquid ammonia for each generator tube was worked out to be about 1kg which is sufficient for freezing 2.38kg of ice. Each module is, therefore, capable of freezing about 23kg of ice per cycle on a good day. In view of the fact that it is possible to get as high as 4 cycles per day each module is potentially capable of producing more 100kg of ice. All the above calculations are based on the assumption that only 6 moles of ammonia will be desorbed. In view of the fact that with high temperatures achievable from the evacuated tubes the possibility of desorption of more than 6 moles is always there. In this respect higher output of ice respected.

7.7.3 Vacuum and Pressure Data

Both the internal cooling system (in which R11 was used as refrigerant) and the ammonia circuit was first evacuated to 2×10^{-5} mbar and leak tested first. The ammonia circuit was pressure leak tested to 30×10^{5} Pa (30bar) and the internal cooling circuit is tested under a pressure of 3×10^{5} Pa (3bar).

The reason of using R11 refrigerant was the availability of the refrigerant at the test site at the time.

7.7.4 Level of Vacuum in the Generator Tube

The Glass tube of the generator tube is evacuated down to 2×10^{-5}mbar and sealed. A 200g barium getter maintains this level of vacuum after the tube is sealed off from the evacuation system.

7.8 Description of the Operation of the System

The combination of the generator tubes and the reflector allows maximum collection of thermal energy from the solar radiation. In view of the fact that the glass envelope is evacuated down to 2×10^{-5}mbar there will be no loss of heat due to the conduction or convection. The question of heat loss by radiation is addressed by using cobalt oxide as the selective coating. Since this selective coating is highly stable at high temperatures (up to 800°C), the absorber surface temperature can rise as high as 250°C without any adverse effect. This type of arrangement of the generator tubes in a 10 tube module receives enough energy to boil off all the ammonia from the solid by 10:00-10:30 am on a good sunny day during so-called "Low Season". At this instance the output signal from the electrical integrator circuit connected to the flow meter, mounted between the condenser and evaporator of the ammonia circuit, triggers the end of the boiling off phase of ammonia from the solid and the start of the absorption phase. It is, therefore, possible to run two cycles in the low season. In the so-called "High Season" the desorption phase will be completed by about 9:30 am and it will be possible to run as high as four cycles. It is, therefore, possible to achieve 3 cycles between the low and high seasons.

As soon as the end of the desorption phase is signalled to the electronic control system, trigger signals will go out from the control system which activates the following:

❶ pulling the shading screens over the generator tubes module.

❷ activating the internal cooling by opening the bi-stable valve in the system, and

❸ pumping the cold air to the condenser of the internal cooling system.

In the internal cooling system R11 vapour is liquefied at the condenser and the liquid is fed to the bottom of the 15mm I.D. tube in the extruded aluminium profile which is inside the generator tube. The liquid is converted into vapour in this tube and generates a cooling effect which cools the calcium chloride. This action, in turn, brings about a reduction in pressure in the ammonia circuit, which in turn causes the ammonia in the circuit to boil off in the evaporator of the icemaker (the icebox) and bring about temperatures in the range of -15°C to -25°C which causes the formation of block ice around the evaporator. This action continues for about 80 minutes until all the ammonia is boiled off in the evaporator and absorbed by calcium chloride. At this stage the electronic control system sends appropriate signals which switch off all the internal cooling system described above, pulls the shading screen away from the generator module and the desorption cycle starts again. The above procedure is repeated for each cycle.

7.9 Trial Unit

In order to assess the performance of the block icemaker under real life conditions two different sites in India were chosen. The first site was chosen at Jallindar in Punjab which enjoys four seasons through the year. The second was in Trivandrum in Kelara which has one hot humid season through the year. In this system a simple housing with wall heights of 2.5m in front and 3.5m in the rear was used. The estimation of these heights was based on allowing easy and trouble-free of the movement of the shading screen over and away from the modules. The height of the housing is 7.5m in total with about 4m of the height under ground level. Temperature in hot summer days can go to as high as 45°C. However, the temperature at night in the high season, summer, could go as low as 14°C. For this reason there was actually no need for the internal cooling in the final cycle of the day. However, it was decided to run the system without relying on the use of the low night time temperature. This was done deliberately in order to assess the performance of the system under more unfavourable conditions. In the months of November, December, January and February the maximum day time temperature, in Jallindar, can be as low as 10°C and the night time temperature can go to about freezing. However, both sides enjoy more than 320 days of sunshine in a year.

The roof of the housing was chosen so as to be large enough to accommodate 3 modules of generators side by side. The size of photovoltaic panel used was 6 panels with a power rating of 24W each. As pointed out earlier a maintenance free tractor battery was used for storage of the d.c. power output from the photovoltaic panels.

7.10 Tests Results

The performance of the system has been assessed from November 2000 to September 2001. In the months of November up to January a maximum of two cycles per day achievable. The quantity of ice produced was about 43kg of ice per day. This figure was well below the predicted one. In other words the output ice was approximately 32% of the expected value for two cycles. The reason behind the poor performance was investigated thoroughly and eventually the cause was narrowed down to the source of the water that was to be frozen. The water was being supplied from a shallow (14 metre deep) well and after analysing its composition the water contained salts and other minerals in it that were approximately 8-14 times higher than the figures recommended for the drinking water. It was then decided to use the treated drinking water from the town.

The trial was repeated with the town water during the months of January and February. Again it was only possible to achieve two cycles per day. The output of the plant went up to about 115-120kg of ice per day for the two cycles. It was then decided to use town water from then on for the rest of the test period.

In the months of March, April and May it was possible to achieve 3 cycles per day and the productivity went up to about 192-197kg of ice per day. From mid May to approximately mid August it was possible to run four cycles daily, with a maximum output of 217-223kg of ice per day. From mid August the number of cycles per day went down to three.

In terms of percentage value the output for two cycles for 2 cycles per day reached between 83% and 87% of the expected figure. For three cycles per day the output reached between 80% and 82% of the expected figure. Finally, for the 4 cycles per day operation the output has been in the range of 71% to 73%.

7.11 <u>Conclusion</u>

The main theme behind the work presented in this chapter was to design and develop a multi-cycle intermittent stand-alone solar powered block icemaker. It is worth pointing out that no successful development and trial of such a system has been reported so far. Among all the available single cycle intermittent systems calcium chloride – ammonia system was chosen as the basic system for the operation. Higher coefficient of performance and the fact that this system is not affected very much due to changes in the evaporation and condensation temperatures are the most important reasons for choosing calcium chloride – ammonia systems. A number of practical problems associated with calcium chloride – ammonia mixture which have caused researchers to look at alternative cooling systems has been successfully addressed in the design of the system reported in this chapter.

A novel, and at the same time very simple and practical, multi-cycle system has been proposed and developed. This system has undergone real-life trial tests and its performance has been assessed. The system is capable of producing up to four full cycles per day.

In the summer months 4 full cycles of operation have resulted in the production of 217kg to 223kg of ice per day and in the months where only two cycles are possible 115kg to 120kg of ice has been produced. In terms of percentage values the ice production is between 71% and 73% of the expected value in the summer months and between 83% and 87% in the winter months. The reason for the slightly lower output in the four cycles per day operation is due to the fact that in the fourth cycle all the ammonia was not boiled off the solid.

It is interesting to note that in all the solar power intermittent refrigeration systems (with all being single cycle) the maximum quantity of ice produced per square metre of solar collector area has been 6kg per day maximum. The fact that the production rate as high as 223kg highlights the significant breakthrough brought about by this novel system.

It is worth pointing out that the increased production is not solely due to more cycles per day. In fact it is also due to the deployment of stable solar selective coating (cobalt oxide) and high vacuum technology which have resulted in higher temperatures to facilitate desorption of significantly higher quantities of ammonia from the adsorbent, calcium chloride, in relatively short times.

This novel system opens a new chapter in practical solar powered cooling and air conditioning systems.

Chapter 8

Design, Construction and Evaluation of a Novel Composite Solar Powered Icemaker

Chapter 8

Design, Construction and Evaluation of a Novel Composite Solar Powered Icemaker

8.1 Introduction

The exciting results obtained from the performance of the multi-cycle intermittent solar thermal powered block icemaker system described in previous Chapter, which is the only functional system of its kind, is undoubtedly the most significant breakthrough in the field of solar powered cooling and refrigeration. After careful assessment of the system it is believed that to explore the possibility of using photovoltaic technology in the system with view to increasing the output of the novel multi-cycle intermittent solar thermal system would be worthwhile attempt. The name "composite" is used for the system because it is powered both by solar thermal and photovoltaic technologies.

A small system has been designed developed and successfully tested. In this chapter the details of design and construction and the results of the performance evaluation tests are presented.

8.2 Description of the System

The system is a combination of two different types of solar powered cooling and refrigeration systems. One being the solar thermal powered section and the other one being powered by photovoltaic panels. The two sections are combined together in a novel way so that the efficiencies of both sections are at their maximum and they complement each other to yield maximum output in the system.

8.2.1 The Solar Thermal Section

The solar thermal section of the system is made of a number of evacuated solar heat collector units called the "generator" as in the previous Chapter. The only different between the "generator" used in this chapter and the one used in Chapter 7 is in the inner extruded aluminium pipe in the evacuated tube envelope. The inner aluminium pipe of the "generators" used in here has an extra feature. A spiral baffle fin is placed inside the inner aluminium tube to secure maximum heat transfer between the calcium chloride and the air. This extra feature is used in a dual-purpose fashion. It is used to accelerate the cooling of the calcium chloride and Portland cement mixture in order to speed up absorption of ammonia during and at the end of each ice making cycle. This reduces the absorption period by, at least, 30 minutes. This pipe and baffle fin combination is also used to speed up the heating and consequently boiling off of the ammonia from the calcium chloride and Portland cement mixture. This is achieved by pumping hot air into the pipe and baffle combination at the start of the desorption phase. The exact operation of the extra cooling and heating is described in Section 8.3.

8.2.2 The Photovoltaic Section

The photovoltaic section of the system is a conventional domestic type refrigerator modified to be powered by photovoltaic panels. In view of the fact that the output of the photovoltaic panel is a d.c. signal it is natural to use a d.c. compressor instead of an a.c. one. With the deployment of d.c. compressors, where the need for an inverter is eliminated, the power loss to operate the compressor will be brought down to a negligible level. Although this is a technically valid argument it was nevertheless decided to use a conventional refrigerator using an a.c. compressor. This necessitated using an inverter to convert the d.c. output of the photovoltaic panels to an equivalent 220V mains signal. This was a somewhat difficult choice as deploying an inverter not only meant additional cost but also meant the loss of about 10% of the power. The cost could be justified because d.c. compressors are much more expensive than the a.c. ones. What tilted the argument in favour of a.c. compressors was that a conventional commercially available refrigerator could be used with very little modifications. Furthermore, the fact that no significant work was needed to be done in this approach made it more attractive in the design of the so called the "Extremum System" described in Section 8.2.2.1, where more than one compressor was used.

At this stage it must be emphasised that the idea of using a.c. compressors does not in any way undermines the novelty of the approach used in the design of the composite system. There is no reason why in the long term d.c. compressors could not be used. In fact d.c. compressors were used in initial stage of this work and a paper has been published on the findings of the research work.

In order to obtain the maximum efficiency from the compressor it must receive the full input power it was designed to operate under. A very important point to bear in mind when photovoltaic technology is being used to power the compressor is that the output power of photovoltaic panels is not constant during the day. There are, therefore, certain periods in a day where it would be delivering less than its rated output power. In this respect it would not be reasonable to expect a compressor with a certain power rating to be operating at its optimum efficiency when powered directly by photovoltaic panels without taking necessary steps to ensure that the compressor receives adequate input power. One can, therefore, conclude that the so-called "ON-OFF" system, where the compressor is ON when there is some input power and OFF when there is no input power, the compressor would not be operating at the optimum efficiency. This problem can be addressed to some extent by using a so-called "Extremum" system, where better efficiency, compared to the ON-OFF system, can be obtained. A description of this approach being is presented now.

8.2.2.1 The Extremum System

In the Extremum system a number of compressors are used in such a way that the sum of their individual ratings is slightly less than the output power rating of the photovoltaic panels. This slight difference is to allow for the slight power loss from the photovoltaic panels to the compressors' unit. As pointed out the word "compressor unit" is referred to as the combination of the individual compressors.

The photovoltaic panels chosen were of the low-cost amorphous silicon type and sufficient panels were used to provide 1500W of output power. The output of the panels was connected to a commercially available inverter. The output of the inverter was connected to the compressor unit via an electronic control unit. This control unit continuously monitors the output power level of the photovoltaic panel. At the early hours of the sunshine when the output power is low the control unit provides power to the first compressor only. This situation remains unchanged until the output of the photovoltaic panels becomes higher than that the power rating of the first compressor. As soon as the panels produce more power needed for the first compressor the control unit switches the extra power to the second compressor and turns it on. In this process the first compressor naturally remains on. This process goes on as higher power is produced at the panels. As soon as the output power of the panel exceeds the level demanded by the first and second compressors, the control unit switches the extra power to the third compressor. As the output power of the photovoltaic panel starts to drop from its peak value the control unit starts to switch off the compressors one by one until all eventually all compressors are off.

Three compressors, with a power rating of ½ Horsepower each, were cascaded in this work.

8.2.3 Energy Storage Section

The energy storage section contains two storage tanks. The first storage tank is for storing hot water and the second one for storage of cold water. The size of the hot water storage tank is about 30 litres and that of the cold water storage tank is 100 litres. They are both heavily insulated by 100mm thick polystyrene foam frozen on their entire outer surface. Both the hot water and cold water storage tanks contain heat exchanger coils to allow air to go through them for pre-heating and also cooling of the calcium chloride and Portland cement mixture in the evacuated generator tubes. The source of heating and cooling for both these storage tanks is the compressors. Each compressor is designed as a heat pump where it takes energy from cool tank and after compressing it in delivers the thermal energy.

8.3 Description of the Operation of the Composite System

The schematic diagram of the composite system is given in figure 8.1. As pointed out above the composite system is a modified version of the novel multi-cycle system described in Chapter 7 working in parallel wit P.V. powered extremum system. The composite system is, therefore, partially powered by photovoltaic panels.

In this respect all the stages related to the operation of the multi-cycle system remain equally valid for the Composite System. The icebox contains two separate exchanger coils one belonging to the solar thermal part and the other to the photovoltaic part of the system. Each compressor has its own separate circuit.

The heat pump action starts to remove energy from the icebox and it also cools the water storage tank. The warm end of the heat pump heats up the water in the hot water storage tank.

The temperature of the cold-water storage can go down to about 8°C and the temperature of the hot water storage can go to about 90°C.

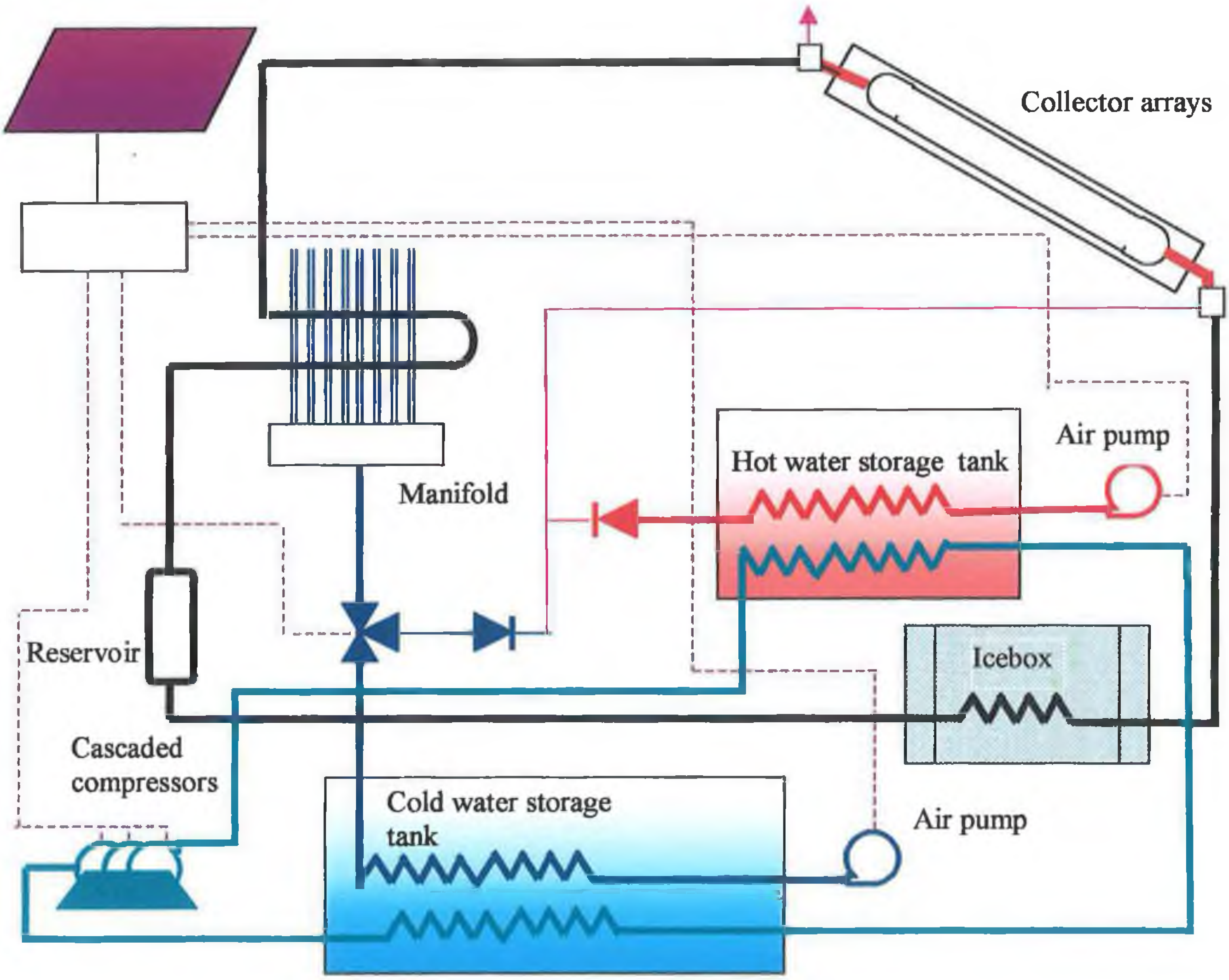


Fig. 8.2 Schematic diagram of the Composite System

These storage tanks are used for rapid heating and rapid cooling of the calcium chloride and Portland cement mixture of the multi-cycle system. The coolant air enters the cold-water storage tank after it leaves the heat exchangers which are placed in the ponds. This arrangement allows cold are of about 8°C-10°C to reach the condensers and housing of the solid mixture in the evacuated generator tubes. Hence faster condensation of ammonia in the condensers and faster absorption of ammonia by the solid mixture become possible. This arrangement allowed the 4th cycle to be completed by 4:00pm.

8.4 Testing and Evaluation

The performance of the system was tested under "real life" conditions for a period of 10 days. The Extremum system, as pointed out, was designed to cool down the condenser of ammonia circuit to accelerate the condensation of ammonia and also to cool down the condenser of the internal cooling system in order to speed up the absorption of ammonia. The total amount of ice produced was in the range of 330-335kg ice per day for four cycles.

8.5 Conclusion

The significance of the works described in this Chapter may be considered in terms of successful design and operation of the extremum system in the first instance. There is no significant degree of innovation in relation to the combination of the multi-cycle and extremum system. It is only natural to expect higher output when the two systems are combined together. However, what is of innovative significance is the fact that the absorption and desorption phases of the intermittent multi-cycle system are shortened significantly, in the composite system. This, in turn, has made it possible to get, at least, 4 cycles in the multi-cycle system before 4:00pm in the day.

What is important at this stage is to look at the question of cost benefit in relation to this new approach, the composite system, compared to the multi-cycle system described in Chapter 7. The total cost of a multi-cycle refrigeration system is in the range of €25000. This includes the costs for generator modules, housing, shading system photovoltaic panels & control unit and refrigeration system. The extra cost of the composite refrigeration system compare to the intermittent multi-cycle refrigeration system can be addressed to the deployment of the extremum system. For the extremum system there is a need for 1500W of output power from the photovoltaic panels. Furthermore, the cost of the required compressor system and the electronic control unit must also be included in the analysis. The total extra cost is:

€9000 (photovoltaic panels for 1500W) + €760 (compressor system) + €360(control unit and the refrigeration system) =€10070.

It can be seen that the extra cost of combining the extremum system to the multi-cycle system is about €10070. As pointed out earlier the maximum output of the multi-cycle system is about 223kg of ice per day and that of the composite system is 335kg of ice per day. In order words the composite system is able to produce about 112kg of ice per day.

Now the important question to address is to find out whether production of 112kg of ice at an extra cost of €10070 is commercially justified. This question can be addressed by referring to the unit cost of ice. The cost of 1kg of ice is about €0.11. Hence 112kg of ice will cost €12.32. The commercial benefit of ice is therefore, €12.32 per day and allowing 320 days per year the annual benefit is about €3942.4. In order to work out the payback time one has to include the cost of borrowing in the calculation. If the rate of borrowing is taken about 8% then the cost of borrowing €10070 over 2.5 years will be €2014. Now the question is how long will the payback time for €10070 + €2014 = €12084. A simple calculation reveals a period of just above 3 years.

At the end, it is important to emphasise that the composite system is a breakthrough in technological achievement by itself as far as high output stand-alone solar powered ice making systems are concerned.

Chapter 9

Design, Development and Evaluation of the Performance of a Novel Reliable High-Output Solar powered Water Desalination System

Chapter 9

Design, Development and Evaluation of the Performance of a Novel Reliable High-Output Solar powered Water Desalination System

9.1 Introduction

The ever-dwindling supply of the fresh water supply and the inevitable hardship that this trend might bring about makes the question of solar powered desalination more urgent than ever. The fact that good quality drinking water is being retailed at prices way above those of refined petroleum products, such as petrol and diesel, for equal volumes proves that drinking water should not be treated as a "cheap" commodity.

There have been several attempts by solar scientists and technologists to develop solar powered water desalination plants or systems. However, none of the reported systems have been reliable and practical enough to be used in any meaningful and significant way. This is because water has been treated as a commodity that enjoyed no significant commercial respect for a long time and this perception seems to be evident even in the last decade when it came to the design of solar powered desalination systems. In other words the most important factor that appears to have been the main influence on researchers in the design of nearly all solar powered desalination systems that have been reported so far has been cost. In this respect as far as solar powered desalination is concerned the end users have always been assumed to be the most deprived inhabitants of the most impoverished regions of the world. The end user is, therefore, assumed to have no financial ability to pay for his/her drinking water. It goes without saying that once this idea was allowed to be the most important deciding criterion in the design then cost became the only deciding factor. All the solar powered desalination systems had to be inevitably of extremely low cost. In this respect all the reported systems, which were designed to be very cheap, turned out to have a maximum output of about 5 litres per square metre of collector per day. All the reported systems of this type have been intermittent and with a single cycle for every 24 hour period. In these systems the evaporation takes place during the day time sunshine hours and the condensation takes place at the night time.

In the novel system reported in this chapter the idea of cost has been taken on board together with increased reliability and output yield at the design stage. This system is, without doubt, a major pioneering breakthrough in the field of solar powered desalination. The cost of the system is, indeed, justifiable with reference to the quantity of fresh water it can provide per day and the cost of similar quality drinking water marketed by various suppliers worldwide.

9.2 <u>Brief Review of the Available Techniques</u>

There have been numerous articles, papers and reports on water desalination techniques. Perhaps the most elegant review on various up-to-date techniques is the one presented by El-Bahi [163]. He has outlined the most popular water desalination systems as; Multi-

Stage Flash {MSF) process, Multi-Effect Boiling (MEB), Vapour Compression (VC), Reverse Osmosis (RO) and Electrolysis Desalination (ED). All these systems are powered by conventional (non-solar) energy. It is important to highlight that the operation of all the large plants is based on high input energy. In this respect their running cost is also a major factor (in addition to the substantial initial capital outlay) in assessing their overall commercial viability. However, despite the huge cost implication the rate of installation of various conventional energy based systems has been consistently increasing over the last 15 years.

As far as solar powered water desalination techniques are concerned nearly all reported cases of passive design of one kind or other [164-165]. The reason for this approach has been presented above in Section 9.1. They are solar stills of different design and the maximum reported yield associated with them is about 5 litres of desalinated water per square metre of the system exposed to the sunshine. In fact the main factor for this low yield is the lack of appropriate condensation facility. This is because although the evaporation phase can take place during the sunshine hours the condensation phase can only take place at night time when the temperature drops to a level for it to happen. In view of the fact that there will always be a limited space to collect the evaporated water the rate of evaporation slows down significantly in the absence of condensation. Another factor contributing to the low yield is the limited input solar energy that can be supplied to the seawater in the shallow pond at the bottom of the still where the evaporation must take place. This is due to the fact that the system is a passive one. In some reported cases black plastic beads coated with a mat black paint are placed in the shallow pond to absorb more energy and hence increase the evaporation rate. However, despite the fact that this approach has brought about some initial increase in the rate of evaporation there could not be any significant increase in the overall output yield of the system. Again, this can only be attributed to the fact there was no cooling during the evaporation to help the condensation of the evaporated water.

9.3 Background to the Design of the High-Output Novel Active System

The main inspiration for design of the novel active system described in this chapter stems from the work reported by Saghafi [166]. A schematic diagram of the system is shown in figure 9.1. In this system an array of flat plate solar heat collectors heats up the seawater. The heated water is then showered in the evaporation chamber of the system. The evaporation chamber contains a number of jute cloth trays made out of stainless steel frames. The jute cloth trays were then placed inside the evaporation chamber at various levels about 15cm apart. As the heated seawater was showered onto the first jute cloth tray on the top it went trough the jute cloth to the trays in the lower levels. In doing so a portion of the heated seawater turned into vapour and salt accumulated on the jute cloth trays.

The vapour of the evaporation chamber is then directed to the condensation chamber where it cools down. The seawater flowed through a coil placed inside the condensation chamber on its way to the flat plate collectors. This coil formed a heat exchanger to cool the condensation chamber to accelerate the condensation of the vapour emerging from the evaporation chamber. This simple, but very innovative, system became the basis for the design of the active high output system, described in this chapter.

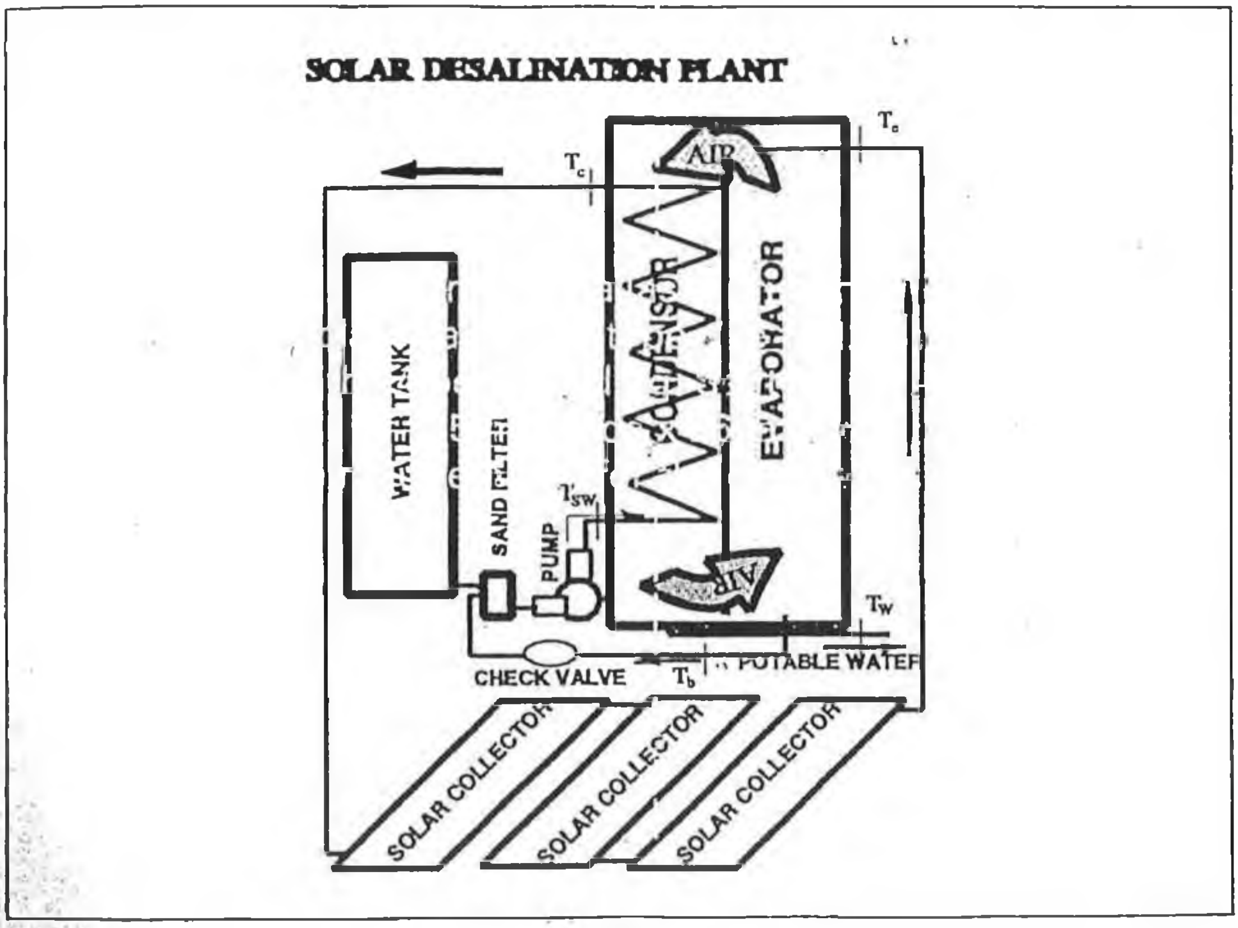


Fig. 9.1 The Proposed Solar Powered desalination plant by Saghafi

9.4 Description of the Novel Active High-Output System

As pointed out above the system described in this chapter is a modified version of the basic system described in section 9.3. There are several new features used in its design that help to increase the output yield significantly. These features are described in details in this section. The flow diagram of the system is given in figure 9.2.

9.4.1 The Water Heating

The seawater was heated by a bank of high-efficiency evacuated solar heat collectors developed at the Institute of Technology, Sligo. Unlike the flat plate collectors used by Saghafi these evacuated collectors are able to heat the seawater to about 90°C very rapidly. In order to cut cost and also avoid heat losses it was found to be most appropriate to pump this heated water to the evaporation chamber directly, without storing the heated water prior to entering the evaporation chamber.

The array of solar collectors used contained 10 evacuated collector tubes each with a length of about 2m and an outside diameter of about 100mm. The absorber fin inside the glass tube was a thin (0.26mm thick) copper sheet coated with the selective coating material, black chrome. The fin was shaped in the form of semi-cylinder with diameter of about 90mm and length of 1.8m. This arrangement provided an angle of incidence of 90° between the rays of the sun and the absorber fin for the entire sunshine hours. The perpendicular "effective" surface of the absorber fin, nevertheless, remained about

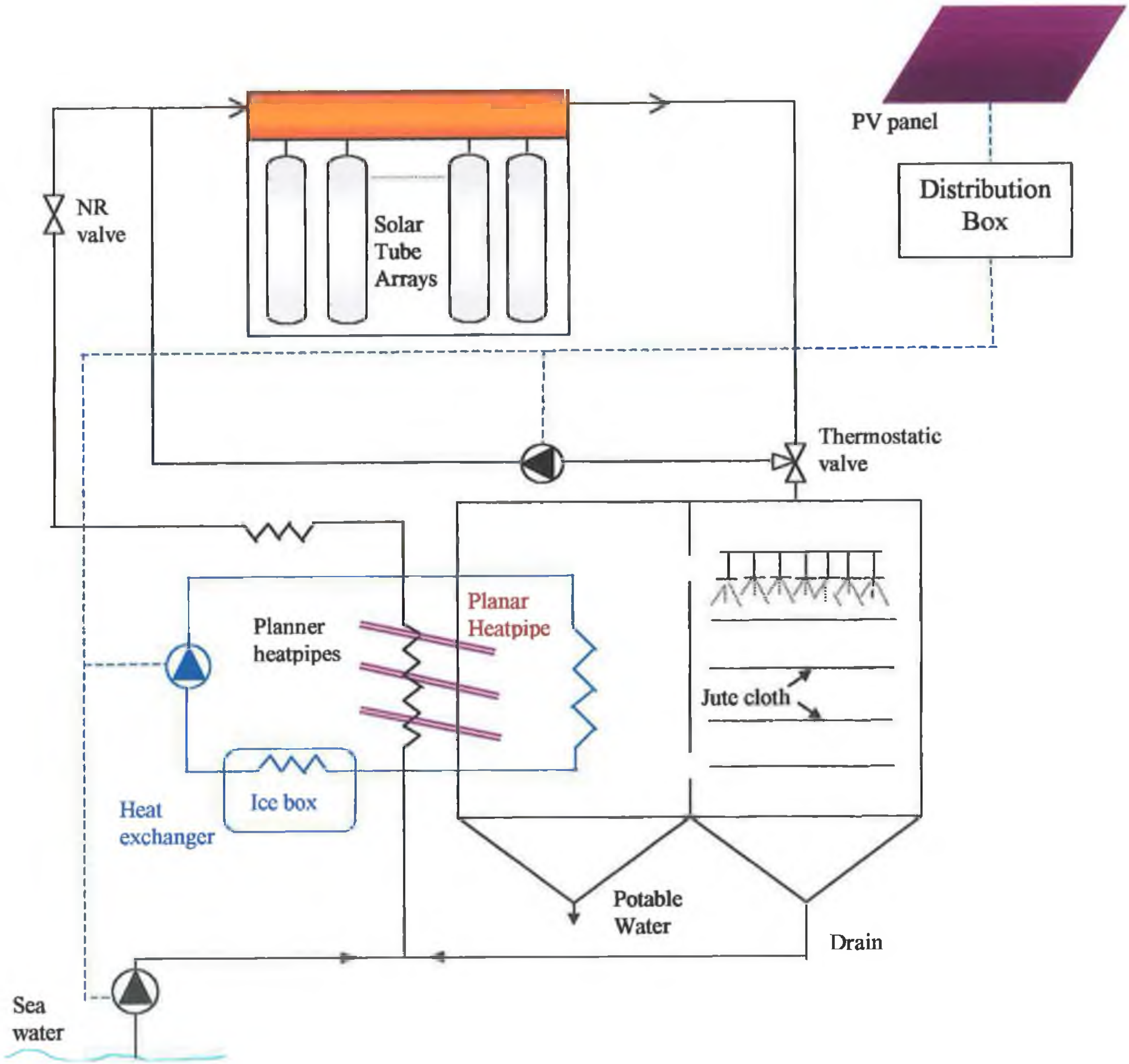


Fig. 9.2 Flow diagram of the Composite desalination plant

0.162m^2 per tube. The curved shaped fin has also been found to be useful in the collection of some of the diffused solar rays. The fin is welded to a copper heatpipe. The heatpipe is 12mm O.D., 10mm I.D. and its length including its condenser is about 2.25m. In order to increase the heat collection capability of each tube a second fin and heatpipe assembly was also placed inside the glass tube to cover the empty space behind the first fin. With this arrangement the collector tube can absorb and convert the energy of the solar irradiation that passes through the gap between the evacuated tubes and shine on the reflectors placed behind the array of collector tubes. Figures 9.3.a and 9.3.b demonstrate a cutaway view and a completed collector tube with an involute reflector, respectively. The ten tubes were integrated with ten involute reflectors and mounted into a box with a glass pane with a 200mm distance between the tubes in the box. The reflectors were made out of anodised aluminium. As the reflectors were highly polished the reflection factor was greater than 90%. The open end of the involute reflectors was 220mm and they were each 1.8m long. This arrangement of the collectors and reflectors is known as the solar collector module. The solar collector module is shown in figure 9.4. The first fin and the heatpipe assembly were exposed to the direct sunshine to collect the direct and diffuse incidence rays of the sun and the second one was to collect the rays emerging from the reflector. The glass pane covering the collector module was to keep the collector tubes and especially the reflectors clean and protected from dust and other undesirable coatings. There are therefore two condensers of the heatpipe that are coming out off the metal flange of the glass to metal seal in each evacuated collector tube.

Another element of the water heating section is the waste heat energy recovery system. This system is described in Sections 9.4.3 and 9.4.4.

Fig. 9.3.a The cutaway view of the solar heat collector and the involute reflector

Fig. 9.3.b The fully developed solar heat collection unit comprising the collector tube and the involute reflector

Fig 9.4 The solar collector module

9.4.2 Evaporation Chamber

The evaporation chamber is similar to the one reported by Saghafi. However there is one additional feature that has been used in the new system and that is the thermal insulation. In order to keep the output yield of the desalinated water one of the most important steps is to reduce the energy loss from the evaporation chamber to the ambient as much as possible. To this end the entire evaporation chamber was covered with 100mm thick of thermal insulation material.

9.4.3 Condensation Chamber

The condensation chamber is another box covered with 100mm thick of thermal insulation all around its walls. The thermal insulation is to prevent the ambient heat getting into the condensation chamber.

In view of the fact that the condensation of the vapour in the chamber brings about release of the latent heat from the vapour it is very important to remove this released energy from the chamber as soon as possible. This is done with the use of a number of heatpipes that are the building block of the energy recovery system described in section 9.4.4.

As mentioned earlier one of the most important novel features of the entire system is the rapid condensation that takes place in the condensation chamber. The source of cooling is a multi-cycle icemaker system similar to that described in Chapter 7. A heat exchanger coil containing circulating cold water at about 5-7°C is placed inside the condensation chamber to secure rapid condensation.

9.4.4 Waste Heat Recovery System

As pointed out above it is very important to make sure that the latent heat, released as a result of condensation of the water vapour in the condensation chamber, is rapidly removed from the chamber. The rapid transportation of the released energy from the condensation chamber is secured by using a number of copper heatpipes. As one might expect each heatpipe has an evaporation and condensation section. The evaporator of the heatpipe (where the released energy is absorbed by the heatpipe) is placed inside the condensation chamber and the other side of the heatpipe, the condenser, is outside the condensation chamber, so that the latent heat is removed from the condensation chamber. The condenser of the heatpipe is placed inside the pipe transporting the seawater to the solar heat collectors. The seawater, is therefore, preheated by the energy removed from the condensation chamber. Without this arrangement the released heat would have gone to waste.

Another section of the waste heat recovery system is the seawater not evaporated in the evaporation chamber which collects at the bottom of the chamber. This was due to the fact that the collected water at the bottom of the evaporation chamber is much warmer than the seawater emerging from the sea. This warm salty water is pumped to the solar heat collector circuit to be heated again to 90°C and showered into the evaporation chamber again. The combination of these heat recoveries has a great positive impact on the effectiveness of the solar heat collectors, and so, helps to heat up larger volumes of the seawater, and therefore, enhances the output of the system.

9.4.5 Electrical System

The source of electrical energy needed to operate the pumps have been photovoltaic panels made of amorphous silicon. The output 12V d.c. of the panels is fed to an ordinary tractor battery. The reason for the choice of the tractor battery was its ability to withstand the fast charging rate of the battery. A car battery usually gets too hot during charging by the photovoltaic panels and might explode. The pumps used are, therefore, d.c. pumps.

9.5 Theoretical Calculation Heading to Estimation of the Output of the System

The theoretical calculations are presented under two headings:

1. Total thermal power collected by the solar heat collectors modules
2. Quantity of the desalinated water

The details of these calculations are now being presented. It is worth pointing out that all the presented calculations are based on the most conservative assumptions.

9.5.1 Power Collected by the Module of the Solar Heat Collectors Tubes

Assuming an overall efficiency of 60% for the absorber fin, over the range of operating temperatures, an effectiveness of 70% due to shadowing, and considering 90% of global incident can pass trough the glass pane, the heat collected by the front fin of the collector tubes per tube, P_f ,will be:

$$P_f = 0.6 \times 0.162m^2 \times 0.9kW/m^2 \times 0.9 \times 0.7 = 0.0551kW$$

In this calculation the effective surface area of the curved fin is $0.162m^2$ and the effective available solar power is taken as $0.9kW/m^2$.

In calculation of the power collected by the rear fin per tube, P_r, a reflection efficiency of the reflector 90% is considered. Again, an overall efficiency of 60% for the absorber fin, an effectiveness of 70% due to shadowing, and 10% losses of incident rays undergo by going through the glass pane is assumed. With an effective surface area of the curved fin $0.162m^2$ and the effective available solar power of $0.9kW/m^2$, the power collected by the rear fin per tube can be calculated as follows:

$$P_r = 0.9 \times 0.9 \times 0.9kW/m^2 \times 0.6 \times 0.162m^2 \times 0.7 = 0.0496kW$$

The total power collected by each collector tube, P_c, is then:

$$P_c = P_f + P_r = 0.0551 + 0.0496 = 0.1047kW$$

The total power, P_t, for the 10 tube-reflector unit is then:

$$P_t = P_c \times 10 = 1.0471kW$$

9.5.2 Calculation of Quantity of the Desalinated Water

The quantity of the desalinated water can be calculated by working out the quantity of the seawater that can be heated to about 90°C before reaching the evaporation chamber this can be done easily by using the well known formula:

$$Q = m \times C\,(\theta_2 - \theta_1)$$

where:

Q is the input energy

C is the specific heat capacity of the seawater

θ_2 is the final temperature

θ_1 is the initial temperature

m is the quantity of seawater

In view of the fact that it is not possible to have the exact value of C for various types of seawater the well documented value of the specific heat capacity for 20% salt brine (C = 3.559 kJ/kg.K) is taken as the acceptable value.

The input power is P_t = 1.0471 kW.

In this respect energy collected per hour is 1.0471Wh.

Since 1000kJ = 0.277778 kWh, then the input energy, E_t, is worked out by:

$$E_t = \frac{1000 \times 1.0471\text{kWh}}{0.277778} = 3769.56 \text{ kJ}$$

The initial temperature, θ_1, is taken as 30°C and the final temperature, θ_2, is taken as 90°C. The initial quantity (volume) of the water, m_1, to reach 65°C in an hour is worked out from:

$$3769.56 = m_1 \times 3.559(90 - 30)$$

$$m_1 = \frac{3769.56}{213.54} = 17.65 \text{ litres /hour}$$

In these calculations one litre of the brine is taken to be almost equal to 1kg.

As the time goes on the recovered energy, that would otherwise have gone to waste, will increase the initial temperature. This is estimated to reach a value of about 45°C, in about one hour. From then on the quantity of the water at 90°C, m_2, is then worked out by:

$$3769.56 = m_2 \times 3.559(90 - 45)$$

or:

$$m_2 = \frac{3769.56}{3.559 \times 45} = 23.54 \text{ litres /hour}$$

It is important to note that after the first hour the initial temperature, θ_1, must be taken at, at least, 45°C. This is due to the contribution from the heat recovery system. In this respect the total quantity of processed seawater, V_t, must be taken as:

$$V_t = m_1 + 6m_2$$

In this expression the second term in the right side of the equation, $6m_2$, indicates the 6 hours of operation after the first hour. V_t in other words a total of 7 hours of operation is taken into account per day. Is then worked out as:

$$V_t = 17.65 + 6 \times 23.54 = 159 \text{ lit/day}$$

Allowing about 4 litres of loss per hour then the expected output, Vo, is:

$$V_o = 159 - 7 \times 4 = 131 \text{ lit/day}$$

9.6 Evaluation of the Performance of the System

The performance of the system was evaluated under two conditions. The first set of evaluation tests were carried out in the laboratory at the Institute of Technology, Sligo, and final evaluation under real life condition.

9.6.1 Evaluation Conducted in the Laboratory

The first set of tests were conducted on the first trial system were developed to investigate whether the design is a workable one or not.

In this trial system the seawater was used as the only source of cooling to help condensation of the water vapour. In this arrangement seawater from the sea was pumped through a heat exchanger placed in the condensation chamber. The outlet of the heat exchanger was connected to the inlet of the solar heat collector module. In the initial set of tests an output volume of about 64 litres per day was achieved. Various attempts were made to increase the output. Eventually an additional cooling by simply spraying the exterior walls of the condensation chamber with the seawater was tried. The temperature of the seawater for this job was about 22°C. It was found that this additional cooling improved the output to about 108 to 111 litres per day. This exciting result strengthened the author's belief that a reliable and efficient cooling is the ultimate key in improving the output of the system. This idea lead to the design of the "Ultimate System".

9.6.2 Test Results of the Ultimate System

The ultimate system using the composite system in conjunction with the extremum system, described in Chapter 8, was put to the test in the coast line of Trivandrum in the south of India for a period of 15 days.

The seawater temperature in this coastline was between 26-30°C for the 15 days the system was under test. It was found that after about 35-40 minutes from the start of the operation the temperature of the preheated water reached a steady state value of about 39-41°C. In this respect the top temperature of 90°C was reached very easily after passing through the solar heat collector module. The quantity of the desalinated water obtained per day turned out to be about 139-141 litres. This is turned out to be higher than the predicted value.

9.6.3 Analysis of the Output Water

The distilled water was analysed at the Science laboratories of the Institute of Technology, Sligo and the results are now being presented. The tests in relation to these results were conducted several times to spot any variation from the normal observations. Table 9.1 shows the general overview of the results. In these analysis minerals have been removed to trace levels from the "Solar" sample.

	Calcium	Magnesium	Sodium	Potassium	Chloride	Sulphate	Nitrate	Conductivity (μS)
Sea Water	500	1000	12000	900	21000	3000	>100	>5000
Ballygowan	114	16	15	3	28	15	9	-
Tipperary	37	23	25	17	-	10	0.5	409
IT Distilled	<1	<0.3	<1	<1	<1	<1	<1	4
Distillate	<1	<0.3	<2	<2	<1	<1	<1	14.5

Table 9.1 Laboratory results (Plasma Tech - IT Sligo): Comparison of mineral content in solar distillate, mineral water (typical concentrations

The test looked at the concentration of Calcium, Magnesium, Sodium, Potassium, Chloride, Sulphate, Nitrate and also the conductivity. As can be seen from the result the outcome of this research programme has been very successful in both the quality of the distilled water and also the quantity. This is a very exciting result because all the existing solar stills using more solar input power can only produce a maximum of 5 litres per day under most favourable conditions.

It is worth pointing out that there has been no attempt made on pre-treatment (such as filtration). There is every reason to foresee that the pre-treatment can speed up the evaporation hence improve the output even further. It is also possible to expect higher yield if the evaporation takes place in a chamber with a pressure lower than atmospheric pressure.

9.6.4 Salt Collection

As a result of the downwards flow of the heated sea water where it goes from one tray of the jute cloth to the next one it under goes a filtration process. As a result of the filtration salt is collected on these trays. The trays are mounted on a rack, which can be slotted in and out of the evaporator chamber. The rack can be replaced every 24 hours, and the collected salt can be removed mechanically i.e. by shaking.

9.7 Conclusion

The exciting result of 139-141 litres per module per day, or 43.5 litres per m^2 compared to 5 litres per m^2 per day for all solar stills reported to date is impressive.

This phenomenal increase in the output is due to the performance of the features used in the system. They are high-output solar heat collectors, waste heat recovery and the rapid condensation system. It is important to highlight the fact that while the deployment of the waste recovery and high-output evacuated heat collectors played a major role in the increased output the contribution of the rapid condensation system is, undoubtedly the most significant one. This is because it is the rapid condensation that eliminated the stagnation and saturation situation of the evaporation and, thereby, opened the way for more evaporation (and subsequent condensation) to take place.

It is interesting to note that the output turned out to be more than the calculated value. Perhaps the most important factors contributing to the discrepancy are:

1. The system was operating for more than 7 hours,
2. The quantity of the wasted heated water was less than the predicted quantity, and
3. The preheated water reached a temperature of 45°C in much shorter length of time than expected.

It is important not to compare the output of this novel system with that of the reported solar stills in terms of output per square metre of collector. This will not necessarily be a fair assessment because the system described here is far more elaborate and sophisticated than solar stills.

The analysis of the output water shows that the quality of the output water compares very well with the best available processed and spring water currently available. In this respect the cost of the system (although higher than solar stills) is justifiable in view of the quality and quantity of the output of the system.

Chapter 10

General Discussion and Conclusion

Chapter 10

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10.1 Discussion

Solar powered refrigeration/cooling and solar powered water desalination are, without doubt, among the most attractive applications of solar energy systems. While it is acknowledged that solar powered cooking is another very desirable application the above-mentioned two applications have always been the focus of attention for researchers in the last two decades. Perhaps the most important point to be highlighted in this thesis is that it was deemed absolutely essential to move away from the scenario that solar energy applications must be centred on absolute low cost. This is because the ever-dwindling source of conventional energy and also the alarmingly diminishingly supply of fresh water with all the economic and socio-political problems it brings about necessitate a fresh approach in solar powered desalination and refrigeration. It has been highlighted in the work described in this thesis that it is only by combination of solar heating and solar cooling that high output water desalination systems can operate satisfactorily. The approach adopted in the works described in this thesis has been not to ignore the fact that a suitably designed solar powered desalination and refrigeration can be commercially very attractive propositions so far as they can pay for themselves in a reasonable length of time. The work initially started with the design and development of a continuous cycle solar powered refrigerator based on modifying the Munters-Von Platten system to be powered by high-efficiency evacuated solar heat collectors. This groundbreaking work, which is the first ever approach of its kind, provided temperatures as low as -19°C in the icebox of the refrigerator. Among all the new features used in this work is a novel solar powered air circulation system to assist cooling at the condenser of the refrigerator. The novel feature of this air circulation system is that it uses no pump of any kind. The only source of energy used in it is the heat collected by the solar heat collectors. No such system has ever been reported to be used anywhere. The application of this system has been proven to improve the performance of the refrigerator by reducing the temperature in the icebox from about -10°C down to -19°C.

In another approach an existing household refrigerator has been modified to operate with d.c. power from photovoltaic panels without using a storage battery. The fact that existing household appliances have been used and modified for the job stems from the idea of using tried and tested technologies to bring the cost down as much as possible without compromising performance.

It is now envisaged that the cost of the design modification for both the above refrigeration systems will be very reasonable if they are to be mass-produced. It is important to note that the sources of power for both the above systems (evacuated heat collectors and photovoltaic panels) are the most expensive components used in them. However, the cost of these items is going down rapidly. As a result of the surveys conducted it is now quite possible to expect photovoltaic panels be produced at a cost of about US$1.5 per watt in the next 5 years. The cost of evacuated solar collectors has come down by about 50% in the last 12 years. This reduction in cost is expected to continue. Bearing in mind the fact that the useful effective lifetime of the evacuated collectors and photovoltaic panels is greater than 20 years it becomes clear that the future commercial prospects for these systems are very promising.

The development of the extremum cooling system and the way it has been used to work in conjunction with the ground-breaking pioneering works on development of the first-ever multi-cycle intermittent solar powered block icemaker is perhaps the most outstanding work among all the works reported in this thesis. The application of this composite block icemaker in the novel solar powered desalination system opened a new chapter in the solar powered systems. It is envisaged that it will attract significant new interest in high cost applications such as air conditioning. The fact that the composite block icemaker produces more ice as the sun gets hotter is perhaps the most significant feature of the system that can be effectively used in air conditioning.

The production of about 140 litres of desalinated water per day with a system using one module with the effective heat collection area of $3.24m^2$ to evaporate and $1m^2$ of collector area to generate the cooling power to condense the evaporated water vapour is a major pioneering work. This is a major departure from all the existing techniques that can only produce about 5 litres of water per day. The design of the system is such that it can be used in large scale operation by simply interconnecting the cooling systems of a number of such plants to be operated as one cooling system for increased output and economising on the cost of the needed cooling system.

10.2 <u>Overall Conclusion</u>

In the work described in this thesis the main focus has been potential commercial viability. In this respect the systems chosen as the relevant topics are solar powered refrigeration/cooling and solar powered desalination. This, by no means must be interpreted as the only application because in addition to the above-mentioned projects there are other applications that are very popular and attracted the attention of the researchers for a fairly long time and solar powered cooking is one such application among many.

Perhaps the most interesting point to be highlighted here is that all the applications described in this thesis have promising commercial attraction but, at the same time, the multi-cycle composite block icemaker turned out to be the key system. This is because its application turned out to have a major impact on increasing the output of the novel water desalination system described as part of the work in this thesis.

As far as the commercial viability of the novel multi-cycle composite system is concerned the matter can be addressed in two ways:

(i) *<u>In the field of air conditioning</u>*: It is an established fact that during the peak demand hours (mid day) the main utility supply system is under severe pressure. This is not only a feature for under-developed countries but also highly developed countries. San Francisco is a typical example. It goes without saying that the unit cost of the mains power will be at its maximum premium to the end user. It is, therefore, possible to appreciate that the multi-cycle block icemaker, whose output will be at its maximum during the peak demand hours, is ideally suited to this particular application.

To meet the ever-increasing demand for the peak hours the Utility must upgrade the power generation capacity. The commercial viability of the multi-cycle icemaker, to supplement the existing air conditioning systems, can be appreciated if the cost of upgrading the power generation capacity of the Utility is compared with the overall cost of individual multi-cycle icemakers. At this stage it is not possible to provide a figure for multi-cycle block icemakers manufactured on a mass production basis. However, it will not be an exaggeration to say that the overall cost of the multi-cycle icemaker will be a very small fraction of the cost of upgrading the Utility power generation capacity. This analysis on a report published in 1994 by APS (Advance Photovoltaic Systems) of New Jersey USA, where it was argued that it would be more cost effective to boost up the grid powered by photovoltaic panels to meet the peak demand for air conditioning in San Francisco rather than upgrading the power generation capacity of the Utility.

(ii) *The economics of producing ice*: The multi-cycle composite system is capable of producing about 190kg of ice per day. The cost of ice, in the market where there is adequate mains power is about US$0.1 per kg. However, in remote regions where the supply of mains power is not adequate or non-existent then the price will be much higher. If one adopts a very conservative approach and considers 300 days of operation per year with a basis production capacity of about 190kg ice per day, then annual quantity of ice produced will be $190 \times 300 = 57000$kg.

Once more, if a price of US$0.065 per kg of ice is to be considered, in order to present highly conservative analysis, the saving such a system brings about will be US$3.705 per year. In this respect after 5 years of operation the financial contribution of the system will be 5 × US$3705 = US$18525. The cost of construction of the system, under the worst scenario in a remote region of a developing country will not be over US$25,000 under the worst conditions. In this analysis the payback time will be about 6 years and 9 months.

It must be emphasised that this analysis is based on a scenario where reliable and sufficient mains electricity is readily available. It can, therefore, be seen that for regions where the mains supply is either non-existent or unreliable and intermittent the price of ice per kg will be higher than the figure given above. The payback time will then be much shorter than 6 years and 9 months.

10.2.1 Payback Time for the Composite Desalinisation System

The average volume of desalinated water that the composite system is capable of producing is about 140 litres per day. Again, in order to present the most conservative analysis, 300 days of operation per year is assumed and the ex-works price of drinking water is taken as US$0.2 per litre. The annual volume of desalinated water is therefore $140 \times 300 = 42000$ litres per year. With the cost of water at US$0.2 per litre the plant is capable of producing 42000× US$0.2 = US$8400 worth of water. In 5 years of operation the system will be producing 5 × US$8400 = US$42000 worth of water. The cost of constructing the desired composite solar powered desalination plant will not be over US$50000 under the most unfavourable conditions. In this analysis the payback time will be about 5 years and 11 months.

It must be emphasised in the analysis presented above the figures used are those related to the worst case and, at the same time, for conditions under which reliable mains supply is available. In reality the price of processed drinking water is higher than US$0.2 per litre ex-works. Taking this point into account and also a more realistic figure for the unit cost of the energy in regions where supply of mains electricity is either non-existent or intermittent, at best, the payback period for the composite solar powered desalination system will be much better than 5 years and 11 months.

Once again it would be appropriate to highlight the fact that the useful lifetime of all the components used in the novel multi-cycle block icemaker and the novel composite water desalination system are over 20 years. It is, therefore, possible to appreciate that the commercial prospect of these systems is even more promising than the analysis presented in this Chapter.

10.3 Suggestion for Future Works

The systems described in this thesis have all been the first successful attempt of its kind. It is, therefore, logical to expect that there is a good deal of work that can be attempted to improve the performance of the systems and, at the same time, reduce their unit costs. The most obvious steps are:

(i) To investigate the possibility of eliminating the evacuated solar heat collectors and the reflector assembly that is used in conjunction with these collectors, by using planar heatpipes heated by non tracking compound parabolic reflectors. The planar heat pipes can be placed inside the evaporation chamber of the water desalination plant. It is an established fact that compound parabolic reflectors can produce heat in excess of 400°C, as reported by Mennon [167]. It therefore seems possible to pour seawater directly on the planar heatpipe placed inside the evaporation chamber and produce vapour by flash evaporation. This research necessitates development of a planar heatpipes that can transfer heat in the range of more than 300°C.

(ii) If it proves impossible to develop planar heatpipes to operate at temperatures of 300°C+ then it would be worth looking at the performance of the system by using two or three planar heat pipes, capable of operating at about 260°C to 280°C. In this approach the heatpipes can be placed at various heights in the evaporation chamber.

(iii) In order to increase the rate of evaporation it is worth looking at the usage of vacuum pumps to reduce the pressure at the evaporation chamber. Perhaps the most obvious attempt would be using a vacuum pump known as a "water pump". It is worth pointing out that this "water pump" does not refer to a pump which pumps water in a circuit. It is a simple pump that uses water to produce fairly soft vacuum, and finally;

(iv) It is also worth looking at filtration of the seawater, hence removing some the floating impurities, prior to pumping it to the evaporation chamber. This might lower the evaporation temperature.

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Appendices

Appendix I

Appendix I

There have been three publications in relation to the work described in this thesis. They are:

1- A Continuous Cycle Solar Thermal Refrigeration System

   Presented in the 4th Renewable Energy Network in Denver, Colorado, USA, June 1996.

2- A Comprehensive Outlook on Solar Powered Cooling Systems

   Presented at the International Conference of the World Renewable Energy Network, Sharjah, UAE, Feb 2001.

3- High-Output Solar Powered Desalination System

   Presented at the International Conference of the World Renewable Energy Network, Sharjah, UAE, Feb 2001.

These papers are presented in this Appendix.

A CONTINUOUS CYCLE SOLAR THERMAL REFRIGERATION SYSTEM

H. VALIZADEH and N. ASHRAFI

DEPARTMENT OF ELECTRONIC AND MECHANICAL ENGINEERING
SLIGO REGIONAL TECHNICAL COLLEGE, SLIGO.
REPUBLIC OF IRELAND

ABSTRACT

A Platen-Munters refrigerator has been modified to operate with heat collected from solar energy conversion in a solar thermal continuous cycle refrigeration system. The results show that temperatures as low as -19°C are routinely achievable. The incorporation of a solar thermal battery can increase the refrigeration period to as long as 10 hours to provide cooling power for as long as 36 hours.

KEYWORDS

Platen-Munters, Compound Parabolic Concentrator, Heat Pipe, Condenser, and Solar Thermal Battery.

1. Introduction

Solar powered refrigeration and cooling is one of the most important applications of solar energy technology. The potential positive impact of solar cooling and refrigeration on the quality of life of a great portion of the world population, and its snow ball effect on the world economy has been investigated by many workers [1-4]. Unfortunately, the engineering and technological developments in the field of solar cooling and refrigeration has not moved forward fast enough to make this application commercially viable. However, the encouraging research and technological developments that has taken place in the last 10 years which have resulted in several working solar power cooling and refrigerator models [5-8] are the most positive indicators that research and development in this field is, indeed, worth pursuing.

It is now beyond any doubt that the question of storing and distribution of perishable food stuffs can be addressed in a much meaningful way if reliable and cost-effective cooling systems can be, more easily, deployed. This, together with the increase in the cost of conventional fuel and its distribution, and the fact that a great majority of the world population, whose living standards are being undermined by these increases, live in areas with very high annual sunshine hours add to the argument that a suitable and cost-effective solar power refrigeration system can prove commercially viable. Basically, there are five different types of design in autonomous solar power refrigerators and these are thoroughly investigated in several publications [9-14]. A critical review of all the reported works by the authors revealed that the Platen-Munters is potentially the most promising system for solar powered refrigeration system if the sully of power and temperature can be made to remain constant in the range required by the design. In fact the reason why the Platen-Munters system has not been successfully adopted for energy has been outlined in a recent publication [15] and its difficulty of maintaining the correct liquid and gas flow rates within the system when driven by a source of variable power and temperature. The work reported in this paper is concerned with a solar power refrigeration system using a Platen-Munters design in conjunction with a solar energy collection and storage system coupled to a fast response heat pipe for the transfer of the stored energy to the refrigerator. The system is capable of 10 hours of refrigeration with the potential to provide cooling for over 36 hours.

2. Description of the system

The system comprises a solar heat collection, solar thermal battery, heat transfer, Platen-Munters refrigerator cycle, and heat rejection parts.

2.1 Solar heat collection

The solar heat collector can be either advanced evacuated solar heat collectors or non-tracking compound parabolic reflector. The reason for the development of these collectors is their reliability and ability to produce high temperature collected heat.

2.2 Solar thermal battery

This section is a well insulated storage tank containing a special type of oil which is non-degradable in the operating temperature of about 200°C.

2.3 Heat transfer

The transfer of the collected and stored solar energy to the refrigerator is done by a simple heat pipe. The evaporator of the heat pipe is inside the solar thermal battery and its condenser is thermally coupled to the evaporator of the refrigerator. The temperature layering investigations revealed that the required temperature is obtainable at a certain depth range inside the thermal battery tank. A thermostatically controlled valve at the bottom of the thermal storage tank allows the oil whose temperature is no longer useful for the operation to drain into another tank. This way the temperature at the evaporator of the heat pipe remains at about 200°C as long as there is sufficient heated oil in the tank of the thermal battery.

The combination of the above mentioned 3 parts is now a well tried and tested system which was developed for he first time at Sligo Regional Technical College and reported in a paper in 1994 [16].

2.4 The refrigerator

The refrigerator used was a commercially available Platen-Munters design modified to take the required energy from the solar heat collector or thermal battery. In order to maximise the output of the refrigerator it soon became apparent that the question of its thermal insulation need special attention. This is because the quantity of heat gain of the refrigerator from its surrounding environment is proportional to the surface area of its wall, the difference between its inside and outside temperatures and the thermal conductivity of thermal insulation material used. It soon became clear that the thermal insulation of the available refrigeration is not adequate by any stretch of the imagination. For this reason it was found that in order to proceed properly with this project it was absolutely essential to find the optimum thickness for maximum cost-effective performance.

This was done by covering the refrigerator with thermal insulation material of varying thickness and assessing its performance. (The assessment procedure is described in Section 3 of this paper). It was found that for thickness of around 75mm the energy reduction is about 35%. For thickness around 100mm the energy reduction was around 42% and no significant gain was observed for higher thickness. As a result of this study the refrigerator was wrapped with 100mm thick thermal insulation material during the entire programme.

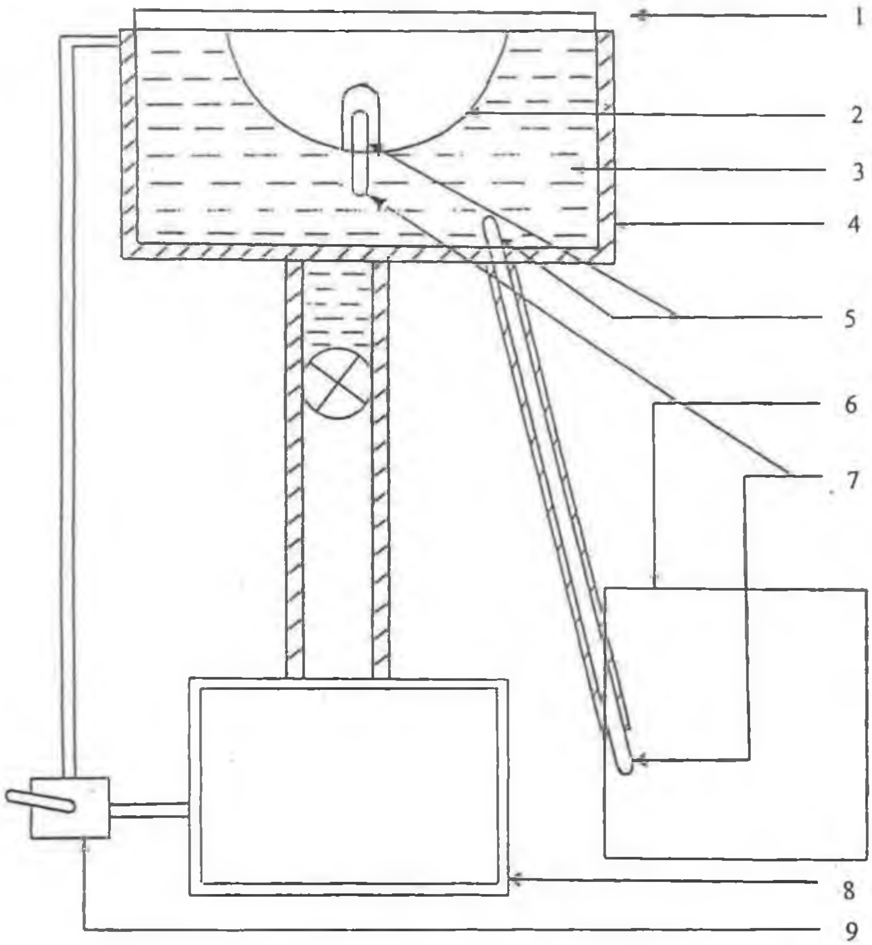


Figure 1

Captions: 1 - pane of glass, 2 - compound parabolic concentrator, 3 - special oil for thermal battery, 4 - the battery tank, 5 - evaporator of the heat pipe, 6 - the refrigerator, 7 - condenser of the heat pipe, 8 - lower level storage tank, 9 - hand pump.

ABSORPTION COOLING UNIT

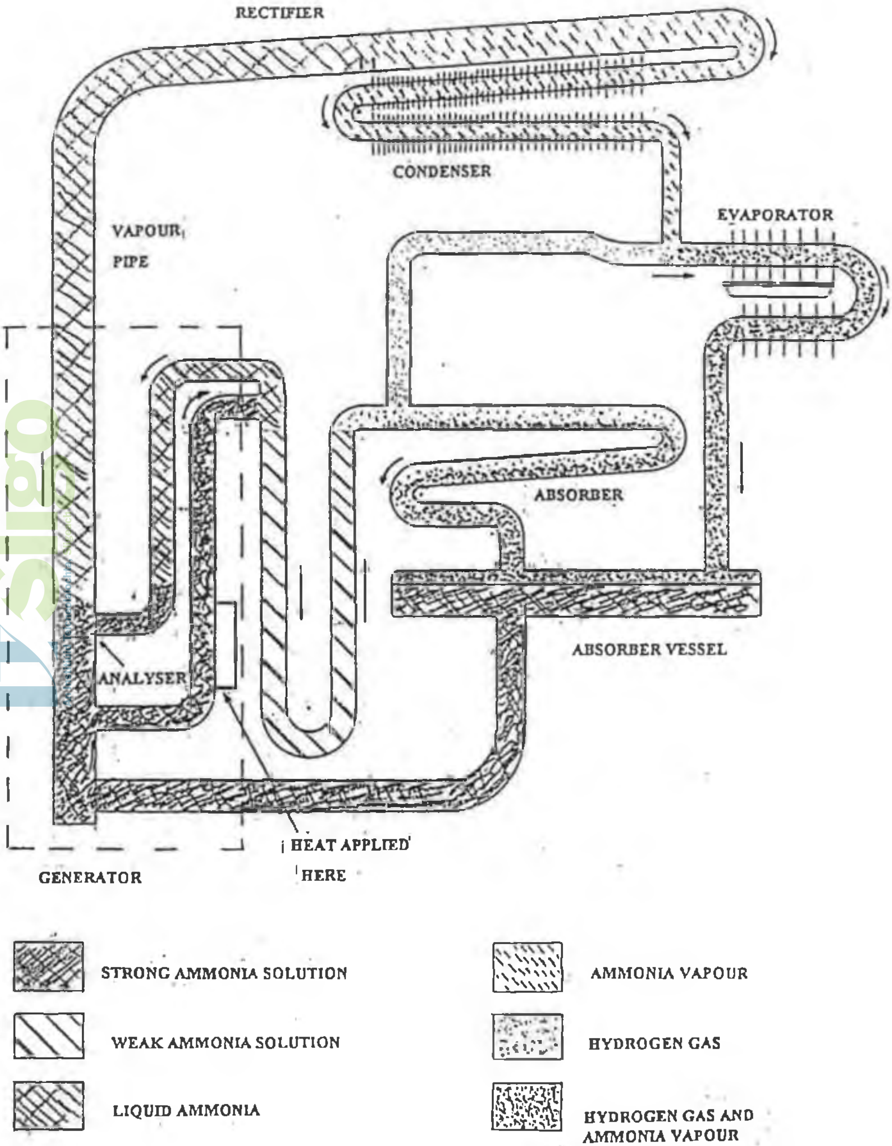


Figure 2

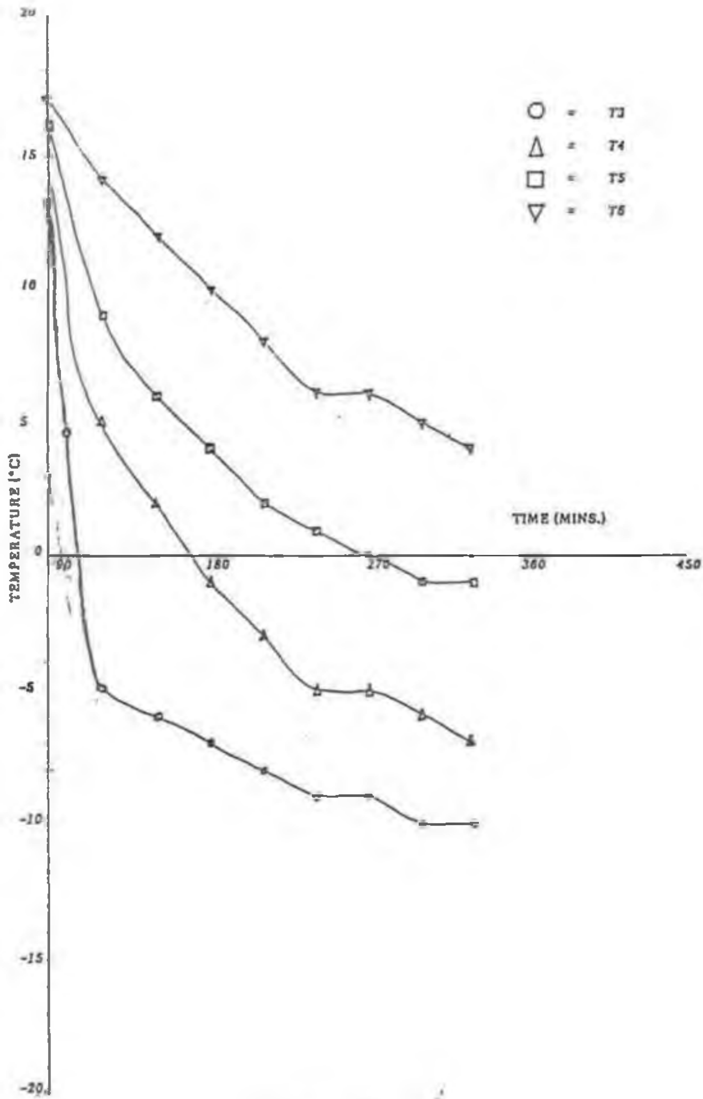
Another modification adapted to the refrigerator was to fit an inverted funnel shape "chimney" to condenser and the rest of the pipe works at its back. The broad end of this contraption contains the condenser and the rest of the pipe works at the back of the refrigerator while the narrow end is at about, at least, 1.2 metres above the condenser. This is to encourage the air movement and speed up the rejection of heat from the condenser. A diagram of the system is shown in Figure 1 and Figure 2 shows the diagram of the refrigerator circuit.

3. Results and Evaluation

In order to collect the necessary data to assess the performance of the system a total of 6 thermocouples were connected to the system. Thermocouples 1-2 were connected to the high temperature end of the systems and thermocouples 3-5 were connected inside the ice box and the last one was fixed outside the ice box but inside the refrigerator. A total of over 50 test runs were conducted to check the repeatability of the results. The results reported here are concerned with the direct coupling of the collected heat to the evaporator of the refrigerator circuit. This is because the most important point was to see whether a system like this could produce any useful cooling cycle. Furthermore the solar thermal battery is now a well proven technique and it can only improve the performance reported here.

The first set of results were obtained with no additional thermal insulator material on the walls of the refrigerator. The result is presented in Figure 3. It is clear that the cooling action started after the system was exposed to 90 minutes of solar radiation and temperature of as low as -10°C was being achieved in the ice box. It was interesting to note that once the refrigerator was covered by 100 mm thick of thermal insulation cover the performance of the system improved dramatically with temperatures of as low as -16°C in the ice box and -3°C in the refrigerator was routinely achievable. The results of this series of tests are presented in Figure 4. No further improvement was observed with thermal insulation covers of thicker than 100mm.

The next question to address was to see which method of rejection of heat from the refrigerator circuit (blowing air, water spraying, or both) can prove most effective as far as the overall performance of the system is concerned. For this reason each one of these approaches



O = T3
Δ = T4
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TEMPERATURE (°C)
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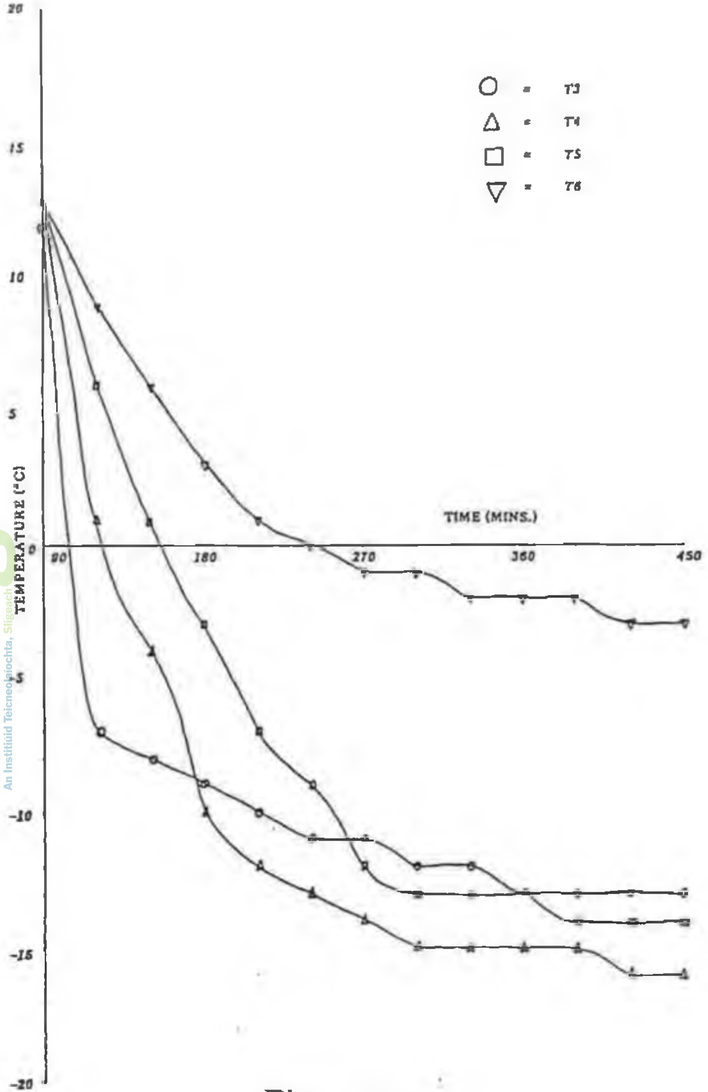
Figure 3



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TEMPERATURE (°C)
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Figure 4

were tried and the results are presented in Figures 5 and 6. The water spraying was repeated every 10 minutes on the condenser of the refrigerator circuit only. The results show that the spraying on its own is sufficient enough to produce the optimum cooling. (The graph shown in Figure 5 shows the results of the water spraying approach and the graph of figure 6 shows the results of the combined water spraying and air ventilation approach). It is interesting to note that temperatures as low as -19° in the ice box and temperatures as low as -4°C in the refrigerator is achievable. These results clearly show that air ventilation does not produce any noticeable advantage. Furthermore, the sparing of the entire pipe works at the touch of the refrigerator has some slight advantage by speeding up the cooling action. The water spraying action was repeated for every 20 minutes and, at a later test, for every 30 minutes and the results are shown in Figure 7 (for every 10 minutes), Figure 8 (for every 20 minutes), and Figure 9 (for every 30 minutes). These results show that the only effect was some slight improvement on the speed of cooling but the final temperatures remained the same.

The results clearly show that the modification and adaptation of the Platen-Munters design of refrigeration system to solar energy is now possible with the high output collectors. The performance of the system can be further improved by the incorporation of the solar thermal battery by storing the excess energy during the peak sunshine hours and using it for longer refrigeration cycles. This system is currently under investigation and the preliminary results obtained so far indicate that it is feasible to expect about 10 hours of cooling cycle with the system.

4. Acknowledgements

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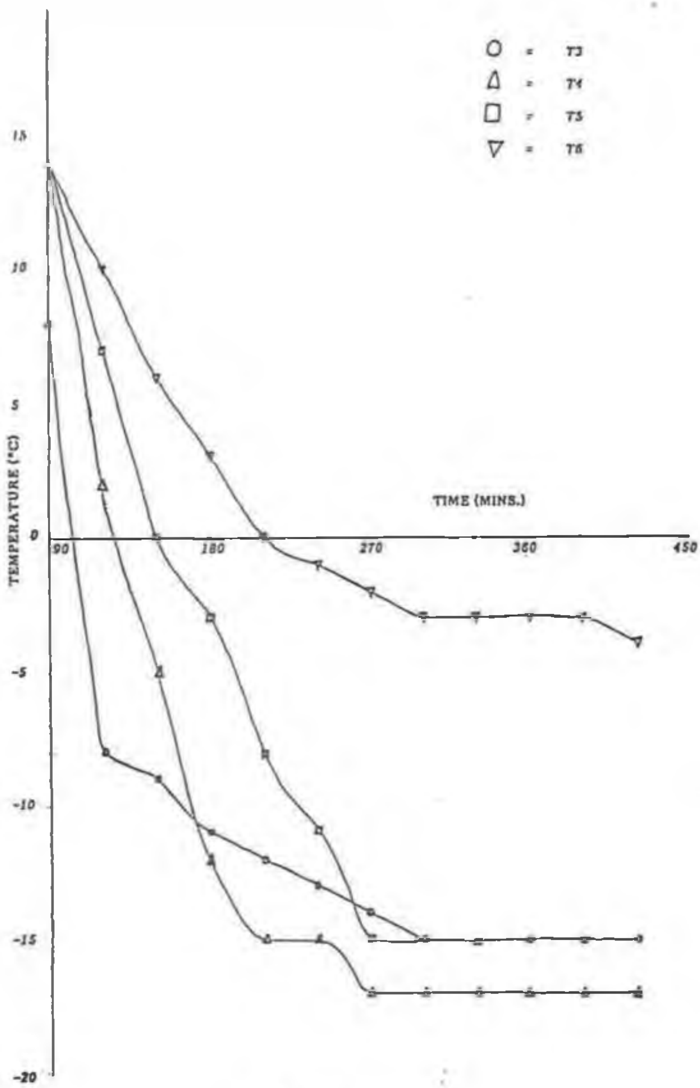

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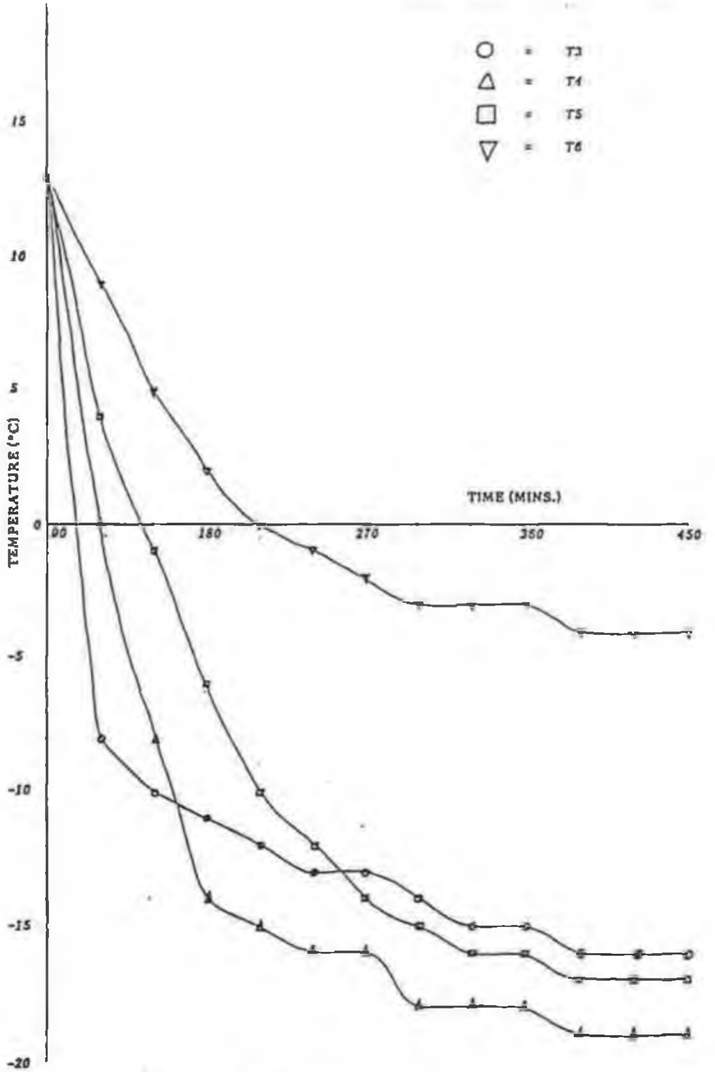
Figure 5



○ = T3
△ = T4
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TEMPERATURE (°C)
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Figure 6

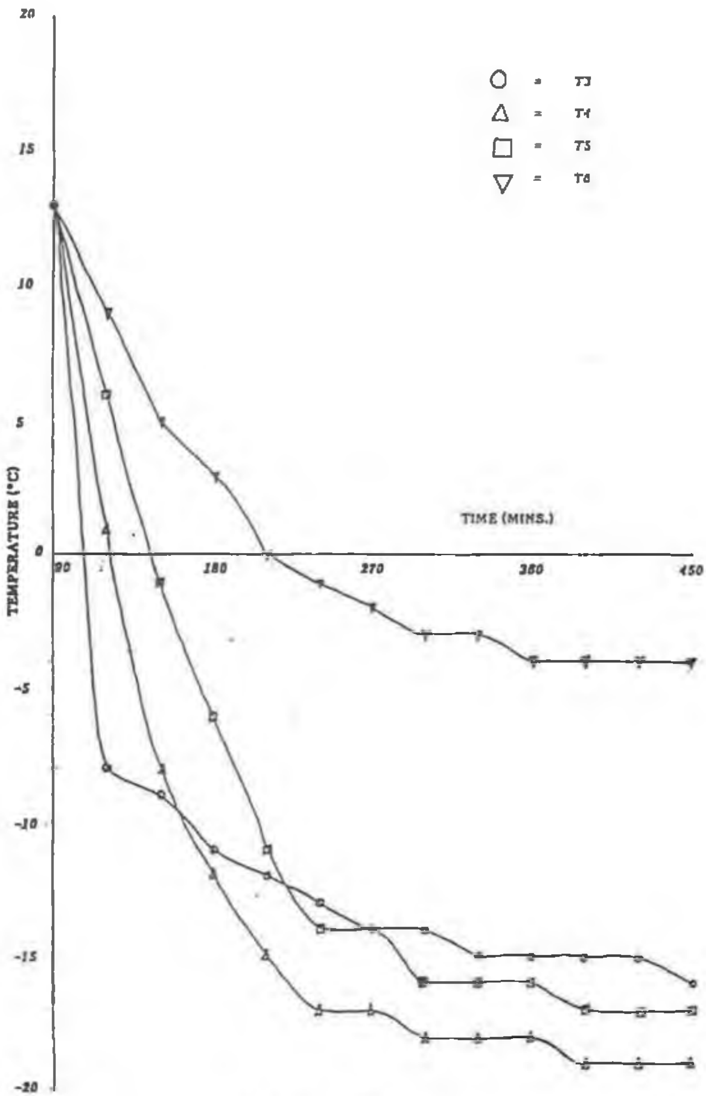

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Figure 7

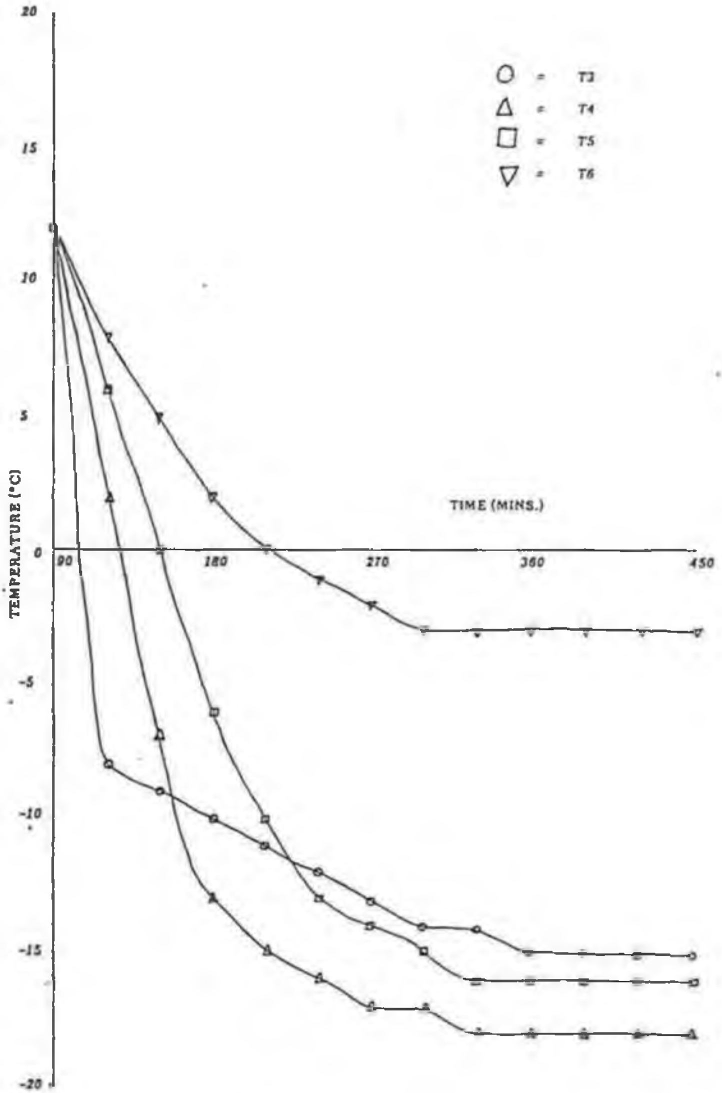
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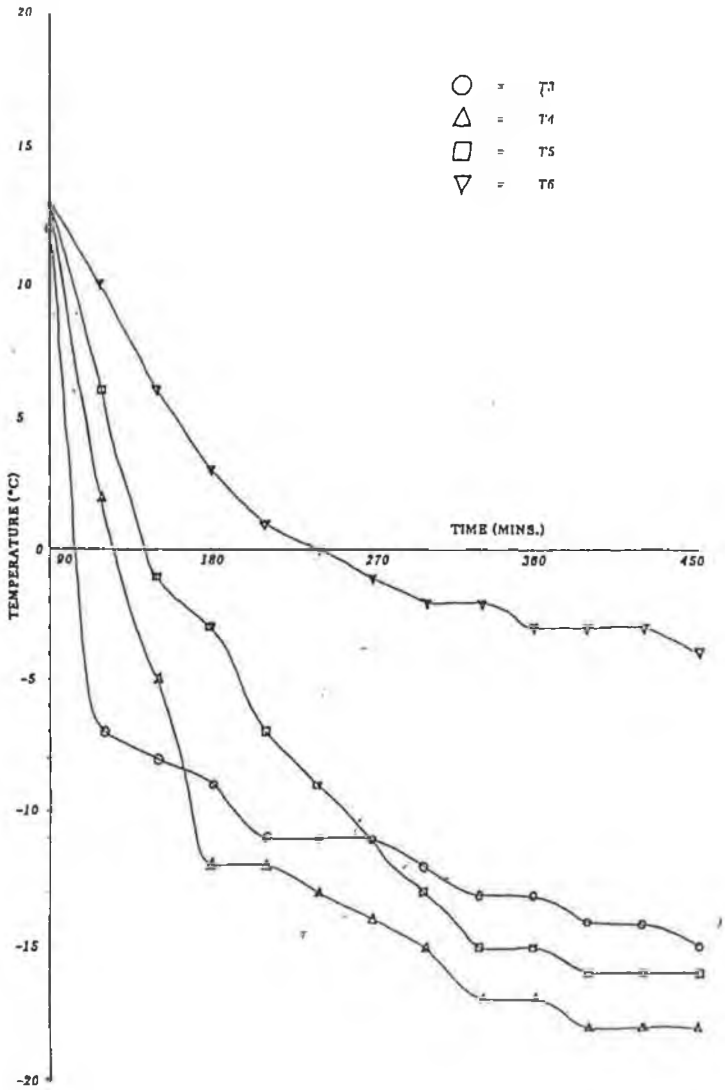
Figure 8



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TIME (MINS.)
TEMPERATURE (°C)
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Figure 9

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A COMPREHENSIVE OUTLOOK ON SOLAR POWERED COOLING SYSTEMS

By

N. Ashrafi and H. Valizadeh
Department of Mechanical Engineering
Institute of Technology, Sligo
Ballinode, Sligo, Republic of Ireland

Abstract

In this paper a general review and assessment of all the solar powered systems using solar thermal and photovoltaic as their source of power, reported so far, has been presented first. A thorough investigation of the potential of calcium chloride and ammonia in a solar powered intermittent refrigeration system has been conducted and the merits of this approach are highlighted. It has been found that deployment of high efficiency evacuated heat collectors, as the source of power, can facilitate more than two cycles for each 24 hours period. Regarding the potential of photovoltaic panels in solar cooling systems it has been found that the maximum efficiency with lowest possible cost is possible by eliminating the need for the storage of solar d.c. power. A system based on this idea has been designed and tested, and we call it an "Extremum System". In this system all the components used are available off-the-shelf and the cooling can be achieved by using a novel control system where maximum output can be achieved for any input power. Finally, it has been found that a sustainable solar powered 24 hours cooling system is possible by combining the high-efficiency intermittent solar thermal system with the extremum photovoltaic system.

Key words

Refrigeration, Absorption, Vapour Compression, Photovoltaic. Compressors, Extremum, Heat pump, Refrigerant, Absorbent and Composite.

1- Introduction

Solar powered cooling is, without doubt, one of the most desirable applications of solar energy in several regions enjoying high rates of solar irradiation. Considerable energies and resources are constantly being put into research and development in this field to make the idea a commercial reality. Although there has been some considerable technological breakthroughs (1-4) there are several difficulties that has prevented the researchers to achieve the desired goal. Technological complexities and the extremely high cost have been the major cause for lack of total success so far.

In this paper after brief review of all the reported systems in Section 2 a trial low cost intermittent system has been designed and tested. For the first time ever it has been practically been proven that an intermittent system can have more than one cooling cycle for every 24 hours. This major breakthrough is described in Section 3.

As far as the application of photovoltaic in solar cooling is concerned a new and novel technique has been used in the design of a system that uses standard components currently in use in refrigeration systems. A novel control system facilitates maximum possible efficiency out of the system for any given input power. This new system which also eliminates the need for storage of solar electricity is, without doubt, the lowest cost system of its kind described in Section 4. A combination of the systems described in Sections 3 and 4 is discussed in Section 5. This combination is potentially capable of providing solar cooling for 24 hours for 6-7 hours of solar irradiation.

Finally, acknowledgement is presented in Section 6.

2- Brief Literature Review

The literature survey of all the reported systems reveals that solar cooling can be divided into two main categories; namely solar thermal and photovoltaic. Since the reporting of all of the systems is beyond the scope of the work presented in this paper the most up-to-date papers/thesis(es) concerned with solar cooling are included in this brief review. One of the most relevant works in this field is presented in literature review of a M.Sc. thesis completed by A. F. Kothediwala (5). Perhaps one of the most significant set of works in the field of solar cooling were completed by Critoph who has developed and tested various solar thermal powered cooling systems and published a valuable reviews in the field (6-10). A comprehensive review of solar thermal cooling systems is reported in a Ph.D. thesis completed by Harvey (11). Sayigh and McVeigh (13) have published the most valuable book containing several papers in the field of solar cooling. Perhaps the most up-to-date review has been completed by Best and Ortega (14) where different solar cooling systems have been outlined.

It is interesting to note that in all the reported cases there are no reports of a tried and tested of a practical and composite design incorporating the merits of more than one technique. The present paper is concerned with a step-by-step approach leading to the design of a practical composite solar powered cooling system.

3- High Output Intermittent System

The brief review presented in Section 2 reveals that intermittent-absorption system with liquid absorbent, particularly water/ammonia (H_2O/NH_3) systems have overall COP of from 0.05 to 0.07. In terms of ice making the performance corresponds to about 4kg of ice per 24 hours, and 1m^2 of collector area. Approximately the same performance has been reported for an intermittent cycle utilising methanol in activated charcoal.

Using a solid absorbent (calcium chloride) and ammonia as the refrigerant, COP of 0.10 is routinely achievable (15).

Calcium chloride absorbs a total of eight moles of ammonia in four steps. It is important, however, to note that the first two moles are too strongly bound to calcium chloride to be driven off at temperatures of 75°C-80°C. In this respect only six moles can be cycled if flat plate heat collectors is to be used as the source of energy. It was found that when evacuated solar heat collectors were used as the source of energy a temperature of about 120°C was routinely achievable. This extra gain of the collector can facilitate a higher rate of reaction.

If the system is to be used as a sole single cycle one (one cycle per 24 hours duration) a COP of better than 0.1 is possible. However, since the high efficiency of the evacuated collector causes total desorption of all the ammonia moles after about 3.5 hours of solar irradiation it was found possible and feasible to consider the idea of getting more than one full cycle per 24 hours duration, if an adequate system of condensation is designed for the refrigerant.

3.1 Description of the system

The schematic diagram of the system is given in Fig.1. The calcium chloride/ammonia mixture was placed in the long pipes, mounted inside the evacuated tubes and welded to absorber plate of the collector. This diagram is, of course, a conventional intermittent system with one cycle per each 24 hours duration. However, there are two features that makes the present design a novel one. They are:

i - deployment of a high efficiency evacuated collector as the source of energy, and

ii - a simple, but very effective, system cooling for condensing the evaporated refrigerant at the condenser.

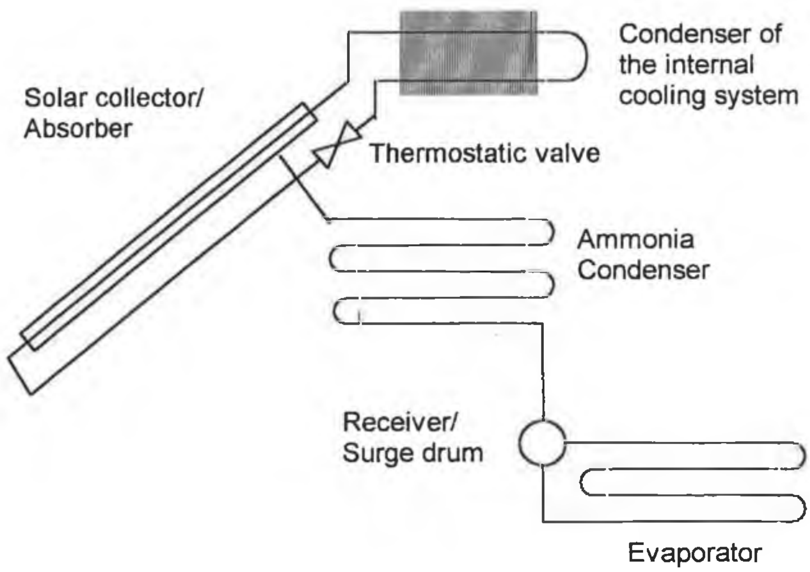

Fig.1

The simple cooling system was achieved by using water at temperatures of about 14°C-18°C poured on condenser. The water is pumped using a pump with photovoltaic panel. The cool water pours over the condenser and as this action starts a fan coil (also powered by photovoltaic panel) becomes active to speed up the removal of heat from the condenser. A simple switch activated by a thermostatic valve activates the water pump and the fan coil. It is possible to get up to 3 full cycles on a good day, of cooling during the daytime hours. However, if the solar conditions are not ideal to get 3 cycles per day then the third cycle will take place at night hours. with this system a target of 18kg of ice per 24 hours for 1m^2 of solar collectors has been routinely achieved.

4- Photovoltaic cooling system

The application of photovoltaic panels to power conventional refrigeration systems is not new. In these systems solar electricity is fed to a conventional refrigerator via an inventor or fed directly to a modified refrigerator using d.c. compressor. The ultimate parameters in deciding to choose one of these approaches are cost and simplicity & reliability of the system. The idea of using a conventional refrigerator and an inventor seemed more attractive. However, after careful consideration of costs and all other technical points the "d.c. compressor" option became more attractive. It is important to point out that the costs of d.c. compressors are coming down as the demand is now increasing. It was, therefore, decided to use d.c. compressor in an otherwise conventional heat pump.

After investigating all aspects related to efficiency of a heat pump it was decided to design a system to have the maximum possible efficiency at various times of the day. This is because the efficiency of the compressor will be at its maximum when it receives its maximum rated power. In view of the fact that the output power of a photovoltaic panel is not constant at various hours of the day called the "Extremum" cooling system was designed.

4.1- Description of the Extremum System

The difference between the so called Extremum system and a conventional system is that in the latter there is one compressor and it comes on as the power is switched on and goes off when there is no power. This system is, obviously, designed when the input power to the compressor remains reasonably constant. Since the output power of photovoltaic panel is not constant during the sunshine hours a conventional system will to be operating with maximum efficiency.

In Extremum System the compressor is made out of a number of smaller compressors connected in parallel. In the early morning hours, as photovoltaic panel starts to produce electricity and its output power is low only one of compressors is activated. As the output power of the panel is increased and there is more power beyond the demand of the first compressor then the second compressor kicks in. This procedure repeats itself for higher powers engaging compressor No.3, 4 etc. As the output power of the panel decreases the reverse action takes place. In this system all the compressors will be operating near their maximum rated efficiencies.

5- The Ultimate Composite System

The Ultimate Composite System is a combination of both the systems described in Section 3 and 4. The basic system is one described in section 3 with the only difference being the cooling system of the condenser. The cooling power for the condensation of the evaporated refrigerant is achieved using the cooling power of Extremum Cooling System described in Section 4. Since the heat pump system produces heat at its "warm side" then this heat is used as part of thermal energy to drive off the ammonia moles from the calcium chloride. The combination of this heat and output heat of the evacuated collectors brings about a more favourable condition to drive off the refrigerant from the calcium chloride at faster rate. This combination of two systems bring about much faster evaporation and condensation. It has been possible to produce more than 30kg of ice per 24 hours duration for a system containing 1m^2 of solar panel.

5.1 Evaluation and Performance

The performance of the system has been evaluated in a number of field trial tests. The trials were conducted in two different regions with totally variable levels of humidity.

In the high humidity region the level of humidity was over 95% and in low humidity region the figure was at about 40%-50%.

The following output yields were routinely obtained for the composite system in the two regions:

- High-humidity region: 20-23.5 kg of ice per 24 duration for 1 every square metre of collector area.

- Low-humidity region: 23-26 kg of ice per 24 duration for 1 every square metre of collector area.

No tests were conducted in the regions with humidity levels less than 40%. However, the yield is expected to be better than 26kg of ice per 24 hours cycle.

6- Acknowledgement

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HIGH-OUTPUT SOLAR POWERED WATER DESALINATION SYSTEM

By

N. Ashrafi and H. Valizadeh
Department of Mechanical Engineering
Institute of Technology, Sligo
Ballinode, Sligo, Republic of Ireland

Abstract

The main theme of the idea behind the design of the solar powered desalination system reported in this paper is based on the findings of a thorough investigation concerning the designs and performances of all the solar powered desalination systems reported by 1997. The conclusions of the investigation highlighted the most important parameters that contribute to the restrictions of the limited output yields of the systems. In fact, the principles of operations of all the reported systems are based on evaporation ad condensation of the sea water.

It has been found that the only way the output yield can be increased is by adopting appropriate technologies in order to increase evaporation temperature, reduce the condensation temperature and recover a high percentage of the energy that, otherwise, go to waste. This has become possible by incorporating high output planar heat pipes, concentrators and d-c heat pumps. This revolutionary new approach has resulted in phenomenal increase in the output yield and, therefore, is a major proof that the present system is, indeed, a commercial reality.

Key words

Desalination, yield, output, saline, distilled, heat pipe, heat pump, evaporation, condensation and heat exchanger.

1- Introduction

There have been several attempts by solar scientists and technologists on using of the sun's power to convert the seawater to drinking water. It would be beyond the scope of the works described in this paper to go into the history of solar powered desalination systems. It has been decided to concentrate on all the works reported in the last 25 years in order to form an accurate and up-to-date picture concerning the principles of operation, performance and all the constrains of all the solar powered desalination plants of the modern times. This is presented in Section 2 of this paper. The findings of this investigation paved the way to the design of the solar powered desalination systems reported in this paper. This is presented in Section 3 of this paper. In fact, the final system, which is a new approach in the general approach to the design of solar powered desalination, is an evolution of the initial system designed and evaluated by authors of this paper in 1997. Section 4 of this paper is concerned with the evaluation of the performance of the most recent design. The exciting high yield of system that strengthens the argument that present system does indeed make solar powered desalination system a commercial reality is presented in section 4. Finally, acknowledgement is presented in Section 5.

2- Brief Review of the available Techniques

The fact that water has always been the most vital commodity in sustaining life, water desalination has gain immense importance in recent years, as the supply of the fresh water is going down. It is, therefore, not surprising that water desalination has become a topic that has received the attention of great number of researchers, in the past 25 years in particular. Perhaps the most elegant concise review on various on various up-to-date techniques has been presented by El-Buhi (1). The most important of all the available techniques can be out lined as: Multi-Stage Flash (MSF) process, Multi-Effect Boiling (MEB), Vapour Compression (VC), Reverse Osmosis (RO) and Electrolysis Desalination (ED).

As far as the solar water desalination techniques are concerned nearly all the reported works in the field have been passive design of one type or the other (2-6). The reason behind this approach seems to be low cost and "simple" technology as the prime users were though to be people remote regions of the third world countries. However, in view of the authors of this paper, the fact that good quality drinking water is, in great majority of countries, more expensive than some popular refined petroleum products (at list by a factor of 2) and also the dwindling supply of good quality drinking water world-wide dictate that the question of solar powered desalination needs a new approach. For this reason an active system incorporating some advanced technologies has been the main theme of the research work presented in this paper.

3- Description of the System

The main inspiration for the design of the system described in this paper stems from excellent work presented by M. Saghafi in Third Congress of the World Renewable Energy Congress in Reading, England in 1994 (7).

In brief, the system, described by Saghafi, comprises a flat plate collector, evaporation chamber and condensation chamber. The sea water is heated by the collector and is then showered down in the evaporation chamber containing a number of jute clothe webs spaced at various levels in this chamber. As the hot seawater is poured down in this chamber a good percentage of it is evaporated here. The vapour is then directed to the condensation chamber. The cooling power in the condensation chamber is secured by the incoming seawater, prior to reaching the solar collector. The potable water is collected from the condensation chamber.

In developing the system described in this paper it soon became apparent to the authors of this paper that a careful design of the system using the technologies developed at our laboratories in Sligo and adopting them to the system reported by Saghafi can open a new horizon in solar powered desalination approach. In our design every effort has gone into the idea that the output of the system must be as high as possible. The factors that determine increased output are:

i - increasing the quantity of water vapour in the evaporation chamber by increasing the temperature of the hot seawater to be showered down in the evaporation chamber. This was achieved by using high-efficiency evacuated collectors instead of flat plate.

ii - compensating for the drop in hot sea water temperature as it passes through the jute cloth webs to lower levels in the evaporation chamber. Incorporation of a number of heat pipes, with their condensers placed in the lower levels of evaporation chamber. The evaporator part of the heat pipes were placed outside the desalination unit and heated by the excess heat produced by the evacuated collectors and the heat pump .

iii - Increasing the speed of condensation. This was achieved by using a so-called "Extremum" heat pump developed by the authors of this paper and also presented to this conference.

It is now easy to see that the heat recovery arrangement described in step (iv) above also facilitates further assistance to the idea of increasing the quantity of water vapour described in step (i).

The heat pump and circulation pumps were powered by low cost amorphous silicon photovoltaic panels.

4- Evaluation of the Performance of the System

It is worth pointing out that the novel design of the system facilitates excellent harmony between the production of vapour and condensation. This means as the dosage of the incoming solar radiation increases, which brigs about higher quantity of vapour, the heat pump will operate at higher power levels and produces more cooling power.

A system using $1m^2$ of evacuated collector and 1kW of photovoltaic panel has been found to produce an average quantity of 28-34 litres of distilled water per hour. It is important to note that another factor that contributes to this high output is the ability of heat pump to absorb considerable amount of thermal energy from the surrounding ambient.

It is envisaged that the performance of the system can be further improved by optimising the sizes of the evacuated collectors' area and the compressor power of the heat pump. Initial investigations point out that the existing 1kW extremum heat pump is capable of coping with, at least, 15% increase in the quantity of the generated water vapour. Based upon the work completed so far output yields of up to 50 litres of distilled water will not be impossible.

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Appendix II

Appendix II

The development of the Solar Powered Multi-Cycle Intermittent Block Icemaker and the Development of the Composite Solar Powered Desalination System are the subject of two separate patent applications.