

AN ANALYTICAL AND EXPERIMENTAL PERFORMANCE
ANALYSIS OF A FLAT-ABSORBER CYLINDRICAL PARABOLIC
CONCENTRATOR AND A FLAT-PLATE SOLAR COLLECTOR

by

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Thesis submitted to the school of Graduate Studies,
University of Ottawa, in partial fulfilment
of the requirements for the degree of

MASTER OF APPLIED SCIENCE

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September, 1978

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ABSTRACT

A non-tracking, cost effective, prototype flat-absorber cylindrical parabolic concentrator (CPC) has been designed, built, and tested under outdoor conditions along with a conventional type flat-plate collector. The proposed collector design utilizes all the important design features of a traditional flat-plate collector except that an additional cylindrical parabolic surface was placed underneath the absorber plate. The most distinct feature of the flat-absorber CPC is that the absorber plate is illuminated on both the front and the back sides which allows the collector to operate even on cloudy or hazy days. The complete module is composed of four identical parabolic channels which were placed adjacent to each other. Each channel consists of a cylindrical parabolic reflecting surface and a flat-plate absorber blackened on both the front and the back sides was placed parallel to the focal plane and symmetric to the focal axis of the reflecting surface.

The experimental performance results have revealed that the flat-absorber CPC is more efficient and cost-effective as compared with the flat-plate collector when tested about solar noon under specified conditions. The efficiencies were obtained experimentally and the test data was correlated with analytical equations that describe collector performance. The design of the experimental facility and testing procedure was based on the recommendations of the National Bureau of Standards as well as ASHRAE Standard 93-77.

ACKNOWLEDGEMENTS

The author wishes to express his sincere thanks and gratitude to his supervisor, Dr. K.F. Schenk for his invaluable guidance and encouragement which made this work possible.

Thanks are due to workshop staff, Department of Physics and Chemical Engineering for providing the facility to construct the collectors. The assistance received from Mr. Le Henaff is gratefully acknowledged.

The author is grateful to Solar Energy group, Building Research Division of the NRC for supplying a few instruments used in this work, and for their suggestions.

The author would like to extend his thanks to Mrs. Y. Rowe and her assistant, the AES-90 Word Processor for editing and typing of this thesis.

The Financial assistance of the NRC and the Government of India, for providing a National Scholarship, is gratefully appreciated.

Last but not least, the author is indebted to his wife, Pushpalata, for her patience and time made available for the completion of this work.

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CHAPTER 1

INTRODUCTION

The world-wide energy crisis induced by the 1973 oil embargo, the increasing environmental pollution caused by over consumption of fossil fuels, specially in industrialized countries, and the potential hazards of nuclear power have focused increased attention on exploring the economic, technical and environmental feasibility of non-depleting energy resources. These resources include biomass, tidal, geothermal, ocean-thermal, wind and solar energy. Out of these resources, solar energy stands out as the most promising and, both in the medium as well as long term future, it will undoubtedly play a very important role in supplying the needed energy to meet the increasing energy demand.

At the present time solar energy is more attractive than ever before as an alternative energy source to provide low-grade heat requirements at economically competitive costs and without environmental degradation. The decision as to what type of energy source should be utilized must, in each case, be made on the basis of economic, environmental and safety considerations. Because of the desirable environmental and safety aspects of solar energy, it should be utilized in spite of its costs being slightly higher as compared with conventional energy resources.

Solar energy radiated by sun is the net effect of nuclear fusion taking place inside the sun where hydrogen atoms are changed into helium atoms. In this process, four million tons of matter are changed into energy every second. The sun continuously produces 390×10^{18} kW of power. But as this radiation propagates out, it spreads and becomes less concentrated. It amounts to about 1.35 kW/m^2 outside the earth's atmosphere. However, the spatial relationship of the sun to the earth results in a wide variation in solar radiation on the surface of the earth. This poses some difficulties in the efficient extraction of this inexhaustible source of energy.

Harnessing solar energy for other useful purposes is not a new idea. Techniques for many applications are known and are technically feasible. However, following the oil embargo, the use of solar energy for certain applications, i.e., space and water heating, is now economically competitive in many countries including Canada [1] with conventional energy sources. Solar energy can also be converted directly into electricity by solar cells or by concentrating mirrors to produce steam for power stations, but at present these schemes are not economically viable and uncompetitive with conventional power systems.

It is expected that the most immediate large scale application of solar energy would be for space and water heating as well as supplying heat for industrial applications. In Canada, of the total energy produced, about 25% is used for space and water heating. The utilization of solar energy for

space and water heating would net great savings of precious oil and gas reserves, which are more important for other industrial applications. The relationship between demand and availability, however, poses some unique problems for Canada in utilizing this abundant source of energy. This problem could be overcome, by providing an appropriately sized seasonal storage system. This would require a moderate temperature to account for heat losses of the storage system.

The most important component of any solar system is the solar collector which is used to collect the sun's energy. A solar collector, in general, consists of a blackened metallic plate which is housed in a box. The back side of the plate is insulated, and the front side, the one facing the sun, is covered with a sheet of transparent material. Incoming solar radiation is absorbed by the metallic plate and retained as thermal energy. A typical solar collector is shown in Fig. 1.1.

If a heat transfer fluid is allowed to pass through the collector, a portion of the retained heat energy is transferred to the circulating fluid and the rest accounts for the thermal losses of the system.

Several types of solar collector designs have been utilized with varying degrees of success. They are generally classified into two categories: flat-plate solar collectors and concentrating solar collectors. Flat-plate solar collectors are sensitive to both the direct and diffuse components of solar radiation while concentrating collectors utilize only the direct

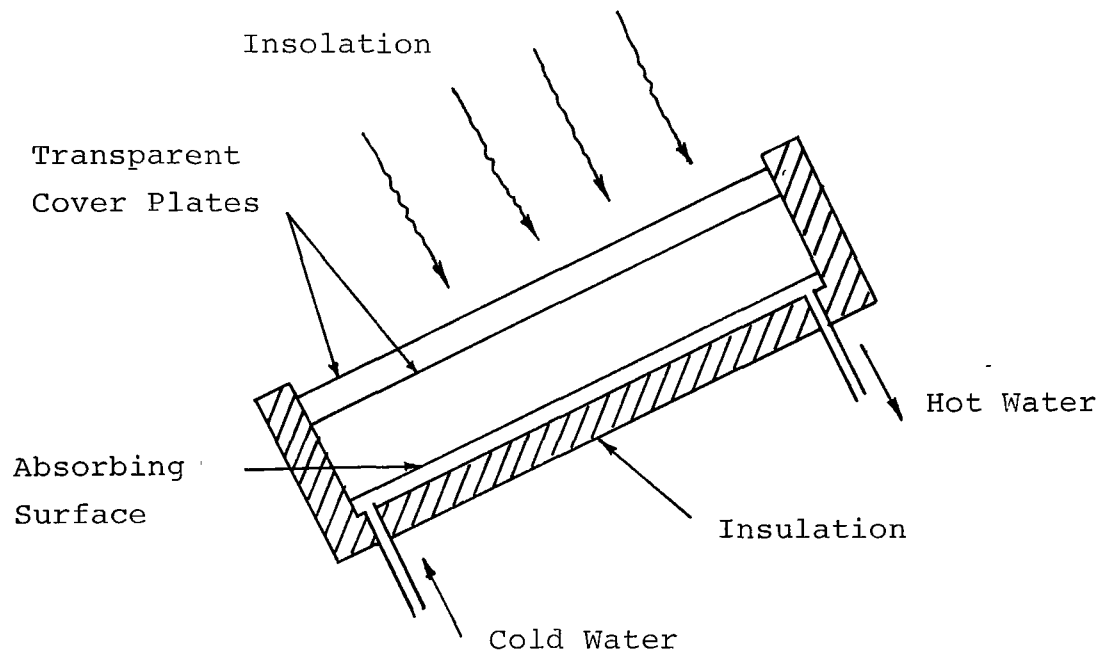


Fig. 1.1 Basic Flat-Plate Solar Collector

component of solar radiation. Concentrating collectors are suitable for high temperature applications while the low intensity of solar radiation makes flat-plate collectors attractive for low-grade heat applications such as space and water heating. However, in Canada where average solar flux density even during the summer season is approximately half of that of tropical regions, flat-plate collectors require a large surface area to provide the required energy for space and water heating. This makes the complete solar system capital intensive. The high capital cost of solar systems can be reduced by developing a less expensive and/or more efficient design of solar collector.

Cylindrical parabolic concentrators with flat absorbers are found suitable for low-to-moderate temperatures since they have a low concentration ratio. In a cylindrical parabolic concentrator with a flat-absorber, the reflecting surface concentrates the direct radiation on its focal plane which is captured by the flat-plate absorber. The front side of the absorbing surface is also exposed to both the direct and diffuse components of radiation. This allows the collector to operate during cloudy or hazy days. For the same transparent cover area, the absorbing surface area in this type of collector is much less than that of the conventional flat-plate collector. This makes this type of collector cost-effective.

This thesis reports on an analytical and experimental investigation of the thermal performance of a flat-absorber

cylindrical parabolic concentrator and a conventional flat-plate collector under identical outdoor conditions. Both the collectors were designed and built at the University of Ottawa and were tested on the roof of the Colonel By Building. The experimental setup and necessary instrumentation were based on the recommendations of the National Bureau of Standards 2 as well as ASHRAE STANDARD 93-77 [3].

In Chapter 2 of this thesis, solar radiation characteristics, its availability, and measurement are described. Chapter 3 outlines the general features of solar collectors. The detailed design of the two collectors tested is also described. The thermal performance analysis of the flat-plate collector as well as the CPC-flat absorber are presented in Chapter 4. Analytical equations to determine thermal performance are developed, specifically for the CPC-flat absorber. The method of testing the solar collectors, the experimental facility and the instrumentation are described in Chapter 5. The experimental results are presented and discussed in Chapter 6. Chapter 7 contains the conclusion and some of the recommendations.

CHAPTER 2

SOLAR RADIATION

2.1 Introduction

Although the energy available on the surface of the sun is immense, as this energy propagates, it spreads out and becomes less concentrated. The mean intensity of solar radiation just outside the earth's atmosphere at the average earth-sun distance of $1.5 \times 10^8 \text{ km}$ is measured as 1.35 kW/m^2 . This energy per square meter per unit time is continuously available outside the earth's atmosphere and is referred to as the solar constant. Most of this radiation lies in the visible light spectrum. As the sun rays traverse the earth's atmosphere, the direct beam is attenuated by absorption by atmospheric gases, by reflection, by dust and water droplets, by scattering, and by air molecules. When it reaches the earth's surface, its density is marginally reduced. The total radiation reaching the earth's surface is the sum of the direct and scattered (diffuse) radiation. The total radiation varies with geographical location, the season, the altitude and the time of day. This is quite obvious from Figures 2.1 and 2.2. On cloudless days, the diffuse (scattered) radiation amounts to 10-20% of the total radiation, while on cloudy days the total radiation is much less and most of the radiation is diffuse. The proportion of diffused over direct radiation depends on the amount of cloud cover.

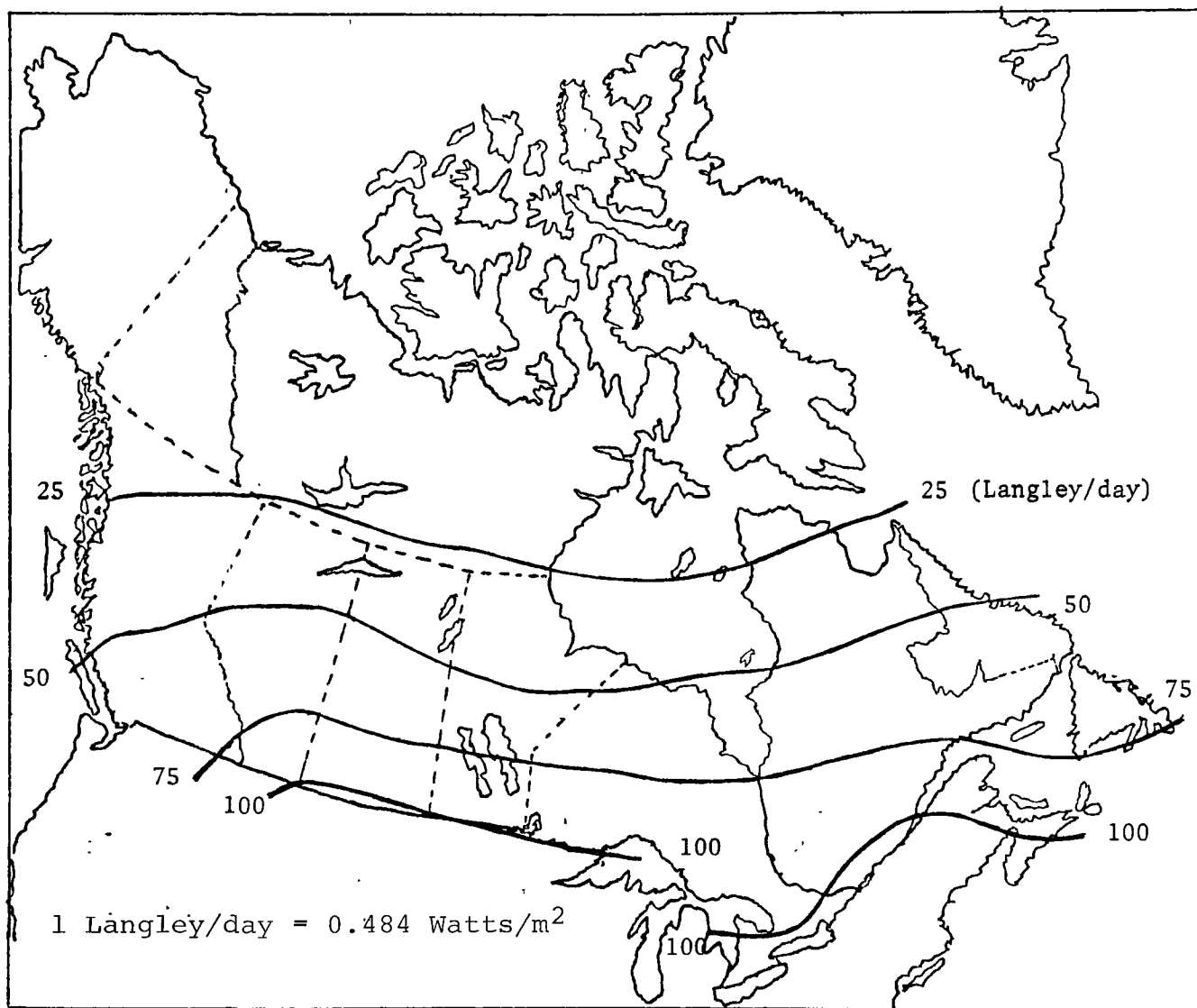


Fig. 2.1 Mean Daily Global Solar Radiation For December

Source: Meteorological Atlas of Canada [23]

2.2 Availability of Solar Energy in Canada

The viability of solar energy in Canada is ridiculed by many, on the false assumption that the resource is simply inadequate. A close look at Fig. 2.3 reveals, however, that most of the Canadian population lives between the 43rd and 49th parallels. The meteorological data, as depicted in Fig. 2.3, indicates that solar radiation received on a horizontal surface at ground level amounts to 150 W/m^2 . Although this value of solar insolation is half of the insolation level in tropical regions of the world; with properly oriented collectors a significant increase in the captured insolation can be achieved. The day-long radiation received by a fixed collector surface is highest in the northern hemisphere when it faces south, and when it is inclined at $\theta + 20^\circ$ for winter season and $\theta - 20^\circ$ for summer conditions, where θ is the angle corresponding to the latitude [4].

Another important factor that increases the solar insolation impinging on a tilted collector surface is the amount of solar energy reflected by snow. Lue and Jordan [5] have suggested the following values of ground reflectance: 0.7 for snow-covered ground and 0.2 without snow cover. Thus, the total radiation impinging on the collector surface consists of the direct, the diffuse, and the reflected radiation from the ground.

On the basis of the available data it is possible to state that there is a great potential for solar heating in Canada. In

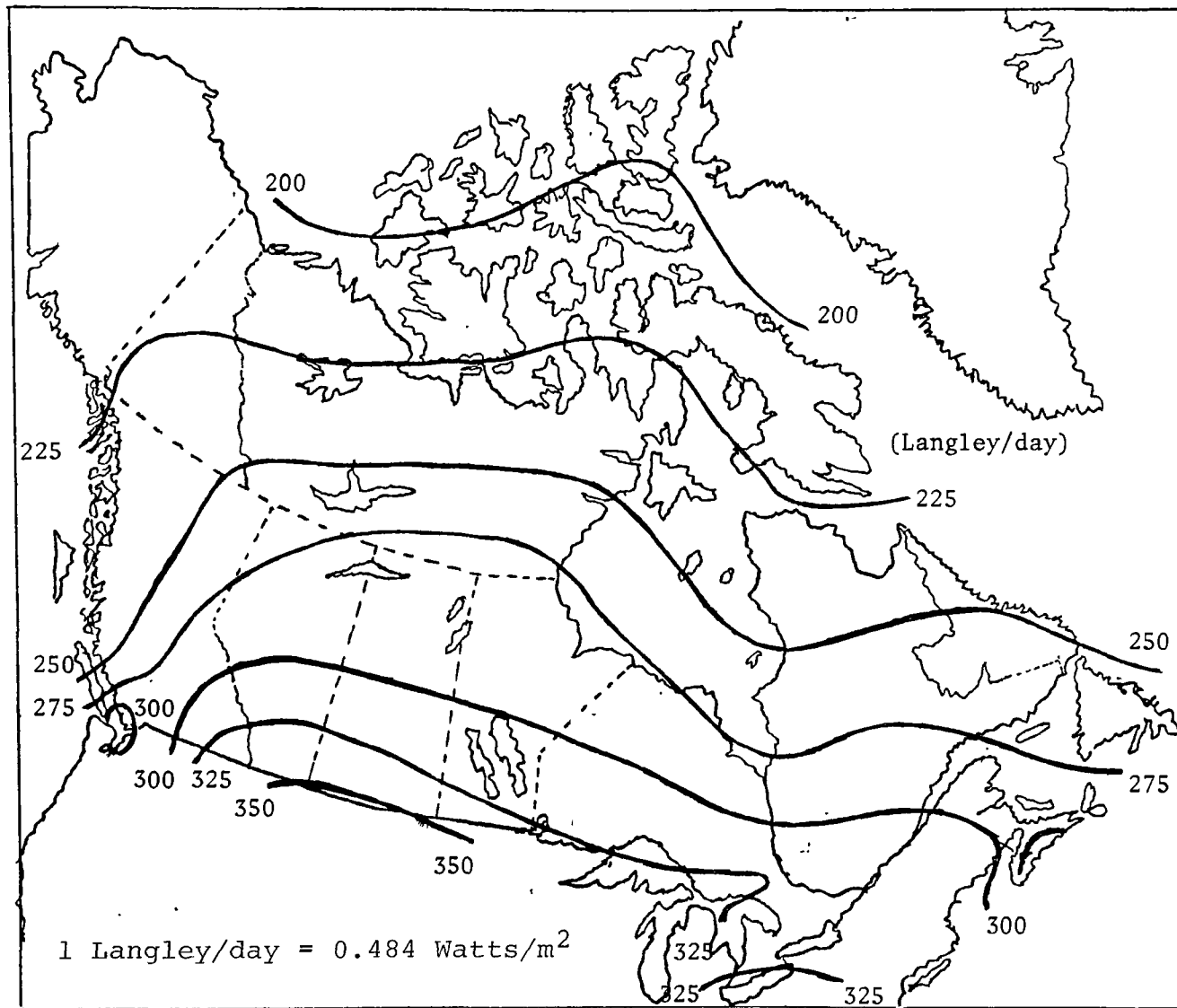


Fig. 2.3 Annual Mean Daily Radiation on a Horizontal Surface

Source: Meteorological Atlas of Canada [23]

line with this, there are many bright sunny days in winter in many Canadian cities. This means that the heat storage unit would be charged to its peak capacity and could meet the required heating load during the sunless periods.

2.3 Solar Radiation Measurement

Solar radiation data has been recorded by about 20 weather bureau stations across Canada. These data are usually reported as energy per unit time per unit area which is striking on a horizontal surface. Pyranometers are used to measure the total radiation, normally known as global radiation, while pyrhelimeter are used to measure the direct beam radiation. Many types of pyranometers have been developed in the past. On the basis of their operating principles, pyranometers can be classified into two main categories: in one type, which is the most commonly used, the temperature difference between two surfaces, painted white and black, is detected by a thermopile. The output of the thermopile is the measure of solar radiation. The Eppley pyranometer which operates on this principle is most widely used by weather bureau stations. A second type of pyranometer, which has been recently developed, is based on the photovoltaic effect. Silicon solar cells are used to measure the solar radiation in terms of the short circuit current. This short circuit current changes linearly with changes in solar insolation. However, the application of these pyranometers is limited because the

calibration is a function of the spectral distribution.

Nevertheless, they are cheaper than the Eppley type.

Solar radiation can also be estimated from the data available on bright sunny hours [6], usually recorded by a Campbell-Stokes recorder.

Global solar radiation available on a horizontal surface has been recorded, at most weather stations, in a form similar to that shown in Figures 2.4 and 2.5, using thermopile pyranometers. Daily data is integrated and reported as hourly or monthly averages in units of langley/hr or langley/day (1 langley = 1 cal/cm²). As such the data is unsuitable and misleading for the purpose of design and evaluation of solar systems because there is no indication of local cloudy conditions. The effect of transient clouds on the useful energy gain is illustrated in Figure 2.6. The figure shows how the useful energy gain is affected by a sudden interruption of solar radiation instantly to zero by intermittent clouds. Because of the transient effect of collector operation under such a varying insolation condition the useful heat energy gain may be insignificant in spite of sufficient solar insolation, as Fig. 2.6 shows.

Clear day insolation tables, prepared by ASHRAE [7] for a variety of latitudes, provide average clear day insolation on horizontal and normal surfaces, and on south-facing surfaces tilted at $\theta+10^\circ$, $\theta+20^\circ$, and 90° (vertical). These tables are an extremely valuable design tool. The limitation of these tables is that the values listed do not include the reflected radiation

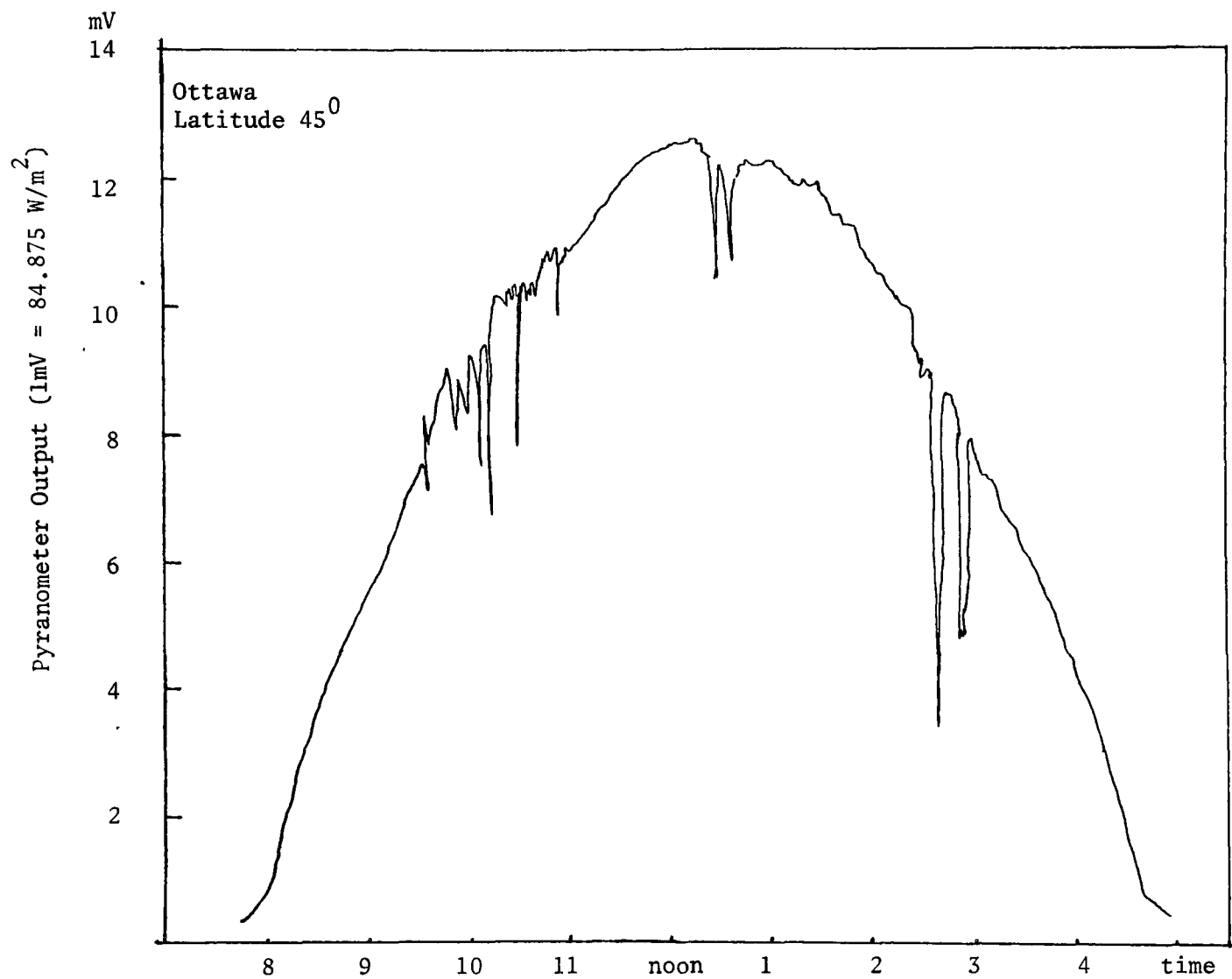


Fig. 2.4 Total Solar Radiation on an Inclined Surface
Versus Time for a Cloudless Day at the University
of Ottawa

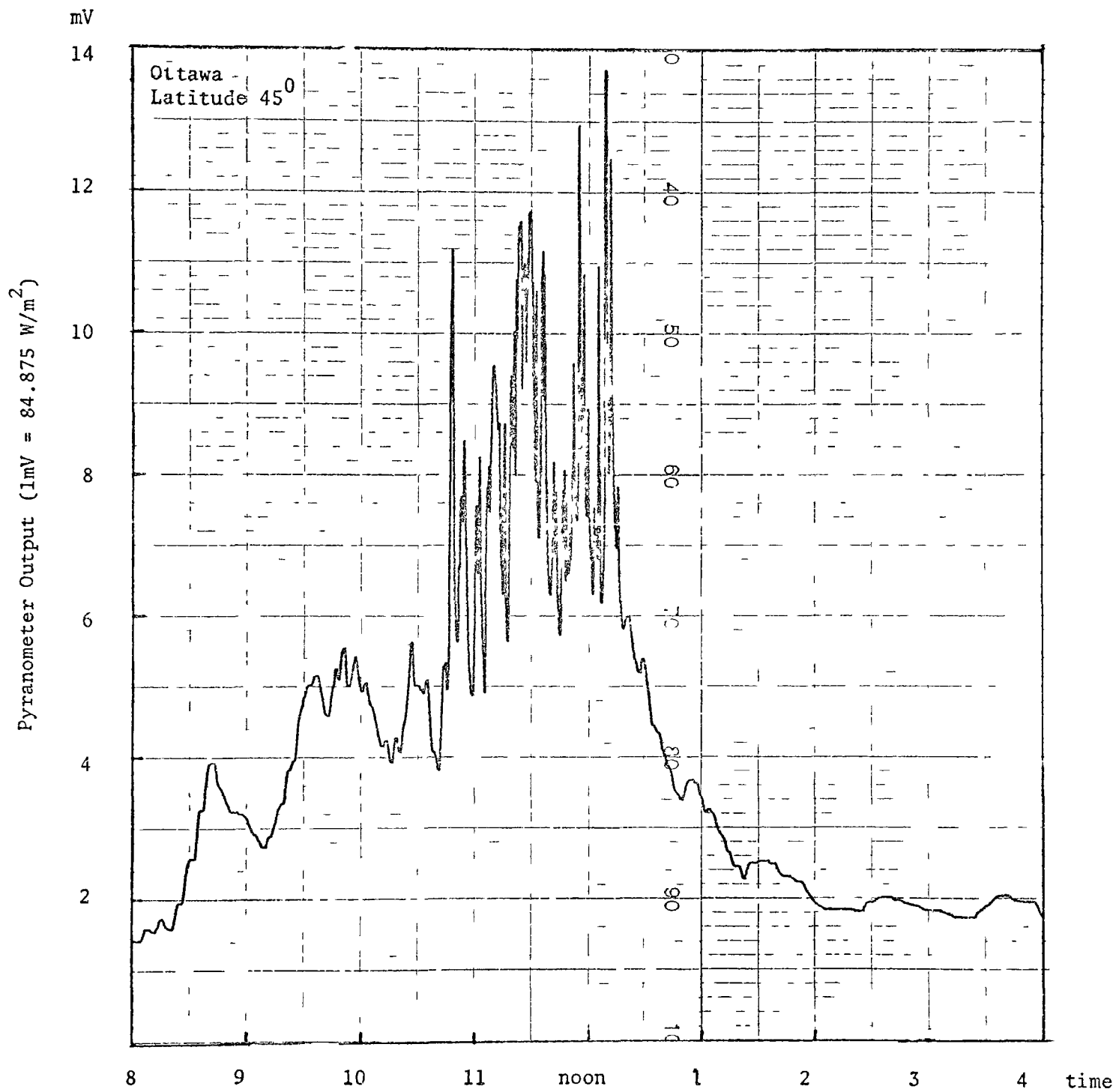
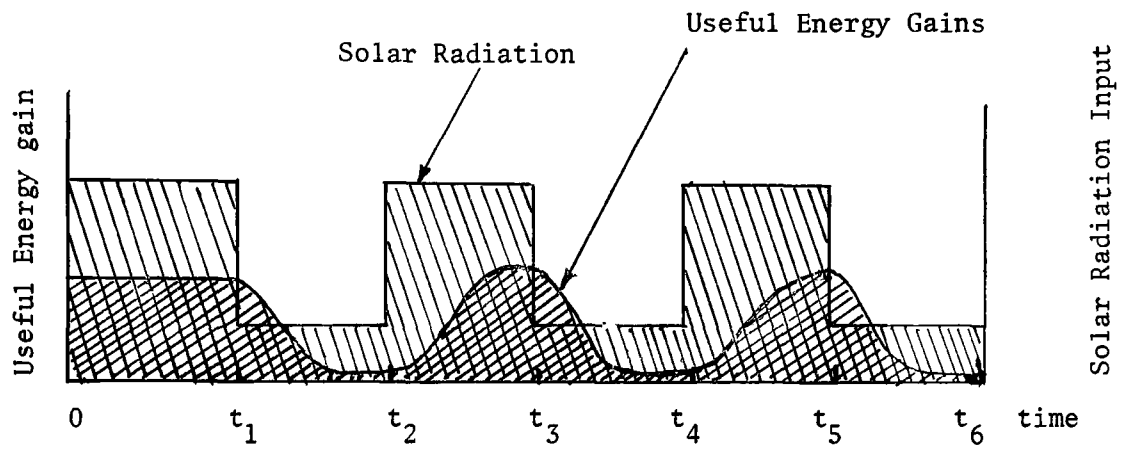


Fig. 2.5 Total Solar Radiation on an Inclined Surface Versus Time for a Cloudy Day at the University of Ottawa



a) Energy Collection During Cloudy Period

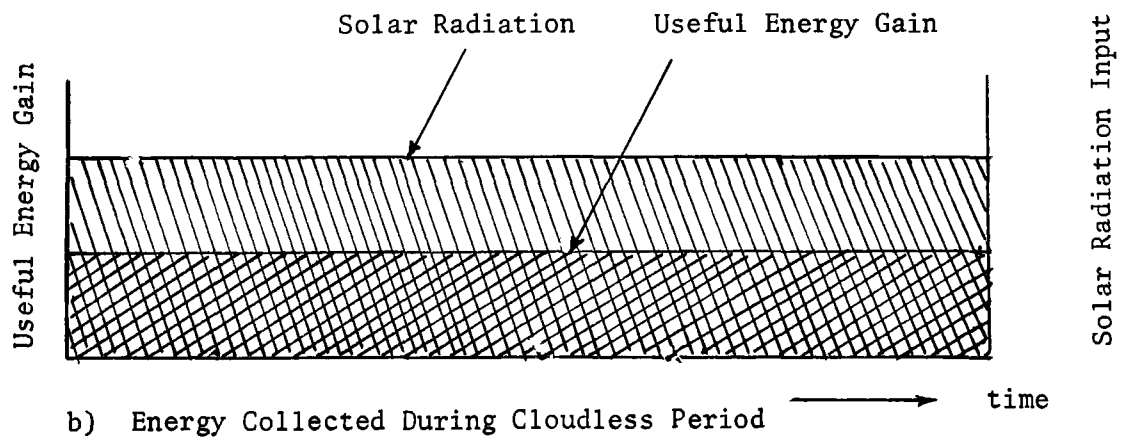


Fig. 2.6 Effect of Intermittent Clouds on Useful Energy Gain

from ground cover. In the winter, this reflected radiation contributes a significant amount onto south-facing walls because the sun is lower in the sky and snow may be covering the ground.

2.4 Solar Radiation on Tilted Surfaces

To ensure the maximum efficiency of the solar system the collector surface is oriented due south and tilted at an optimum angle, latitude $\pm 20^\circ$ to the horizontal surface, depending on the operating season (winter or summer). Unfortunately, most solar radiation data is available in the form of hourly totals on horizontal surfaces. Therefore, for the purpose of solar system design and evaluation, it is necessary to convert these hourly incident totals on a horizontal surface to that incident on a tilted collector surface. This conversion is complicated by the fact that the total incident radiation on the tilted surface is made up of three components; (i) direct or beam radiation; (ii) diffuse or sky radiation; (iii) reflected radiation from the ground or surroundings.

The orientation factor, R_b , for the direct component of radiation, I_b , normal to the collector surface, as depicted in Figure 2.7, can be expressed as [6]

$$R_b = \frac{I_n \cos \theta_T}{I_n \cos \theta_z} = \frac{\cos \theta_T}{\cos \theta_z} \quad (2.1)$$

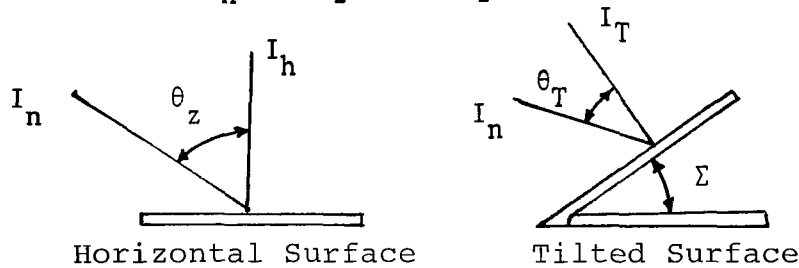


Fig. 2.7 Radiation on Horizontal and Tilted Surfaces

where I_n is the direct normal radiation, and θ_T and θ_z are the angles of incidence of the direct radiation on the tilted and horizontal surfaces, respectively. The expressions for $\cos \theta_T$ and $\cos \theta_z$ derived from the solar geometry are given in Appendix A.

The orientation factor, R_s , for the diffuse component of the radiation, I_d , assuming uniform intensity over the sky, has been derived [6] from the fact that a tilted surface sees a part of sky dome and is given by

$$R_s = (1 + \cos \Sigma)/2 \quad (2.2)$$

where Σ , as shown in Fig. 2.7, is the angle of tilt.

The amount of solar radiation incident on a tilted surface which is reflected by the ground or other surroundings, which have a diffuse reflectance of ρ for solar radiation can be written as

$$I_r = I_H(1 - \cos \Sigma)\rho/2 \quad (2.3)$$

On combining (2.1), (2.2) and (2.3) the total radiation incident on a tilted surface at any time is given by

$$I_t = I_b R_b + I_d R_s + \frac{I_H(1 - \cos \Sigma)\rho}{2} \quad (2.4)$$

and the overall orientation factor R , becomes

$$R = \frac{I_b}{I_H} R_b + \frac{I_d}{I_H} R_s + \frac{(1 - \cos \Sigma)\rho}{2} \quad (2.5)$$

where I_H is the total radiation measured on the horizontal surface.

If the measurements of solar radiation are made along the plane of the tilted surface, the orientation factor R becomes unity.

CHAPTER 3

SOLAR ENERGY COLLECTORS - THERMAL APPLICATION

3.1 Introduction

The most important component of any solar system is the solar collector which captures the heat energy radiated by the sun. There are various types of solar collectors which are subdivided, on the basis of similar geometry and characteristics, into two main categories; flat-plate collectors which absorb both the direct and diffuse components of solar radiation over a wide distributed area, and focusing collectors which concentrate the direct radiation onto a relatively small absorbing surface by means of mirrors or lenses. This energy is transferred to the circulating fluid, water or air, which can be used for other useful purposes. Various geometries of solar collectors of both the focusing and flat-plate type are depicted in Figures 3.1 and 3.2, respectively [6, 7].

The decision as to what kind of collector to use for a specific application is dictated by technical considerations and economics. Since the focusing collector is usually capable of producing higher operating temperatures than a flat-plate collector, it has certain advantages in some applications. However, the complications involved in its manufacture and the continuous tracking mechanism which is required have ruled out the use of this type of collector for low temperature

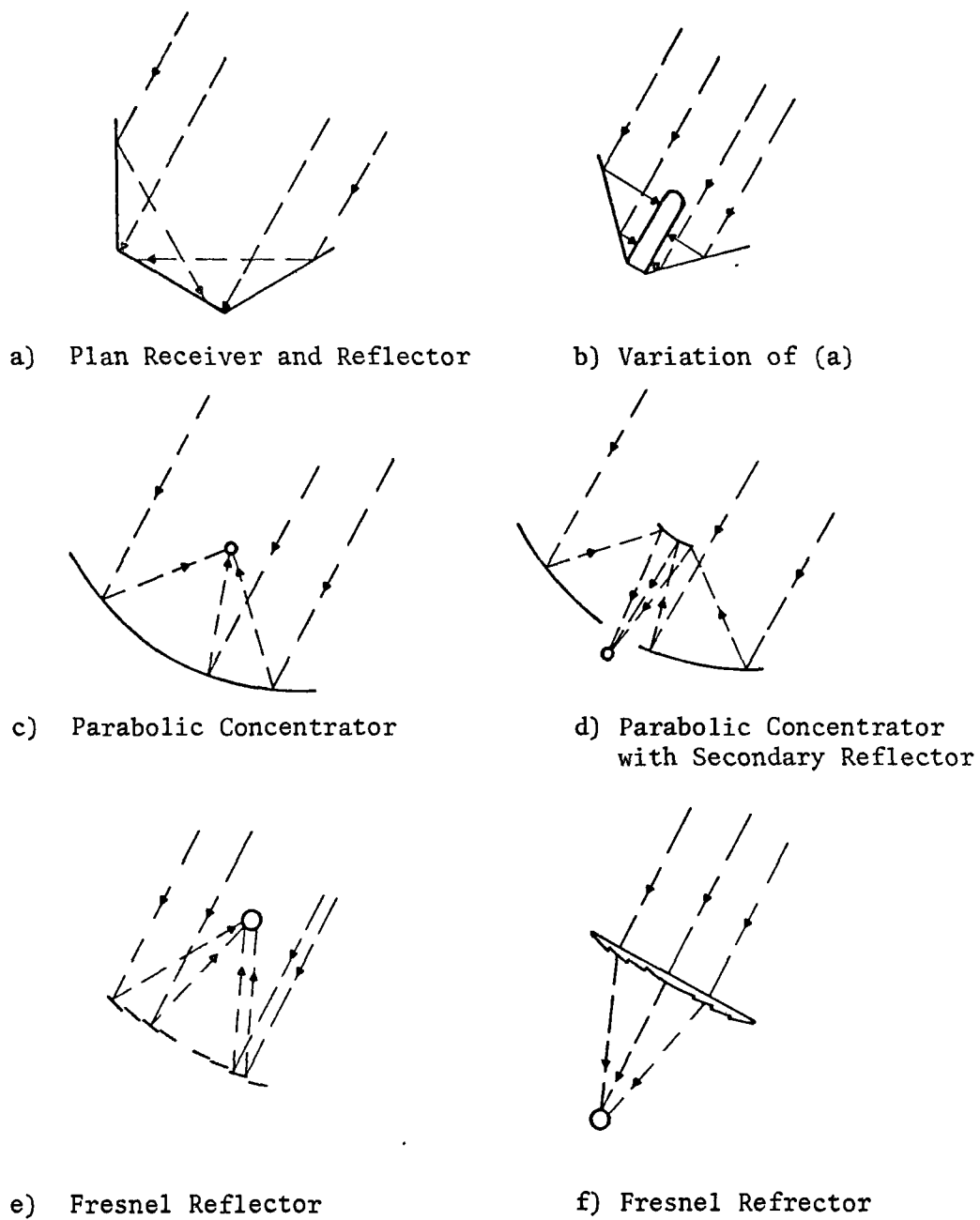


Fig. 3.1 Some Focussing Collector Configurations

Source: Duffie and Beckman, Thermal Processes, 1974 [6]

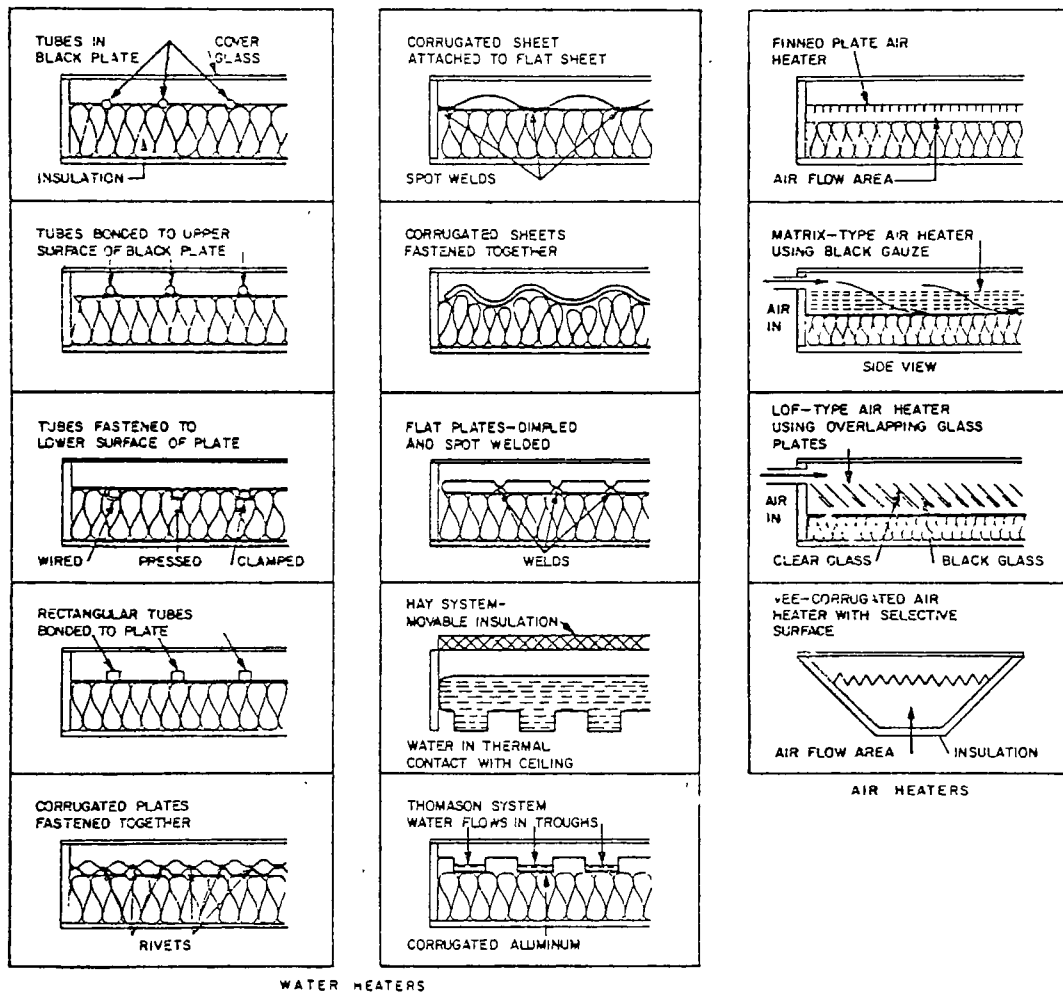


Fig. 3.2 Various Flat-Plate Collector Configurations

Source: ASHRAE, Guide and Data Book Series 1974 [7]

applications such as air and water heating systems. A flat-plate collector, which is usually mounted in a fixed position and can utilize both the direct and diffuse components of solar radiation, is capable of producing temperatures in the range of 150°F (65°C). It is most commonly used for low temperature applications such as space and water heating. However, in regions where the insolation level is low, a huge surface area is needed to provide the required energy for space and water heating. This makes the complete system capital intensive and, in most instances, not yet cost-effective. [1]

A cylindrical parabolic concentrator with a flat-absorber is also suitable for low-to-moderate temperatures since it has a low concentration ratio and utilizes all the basic features of a flat-plate collector. The fundamental principles of this simple but effective device are discussed in some detail in the following sections.

3.2 The Flat-Plate Collector

The basic components of a typical liquid-based flat-plate collector are shown in Figure 3.3.

An absorber plate is covered by one or more transparent cover plates of glass or plastic and the sides and bottom of the box are insulated. The sun's radiation passes with little attenuation through the transparent cover plates and is absorbed by the absorber plate which is heated thereby. This absorbed

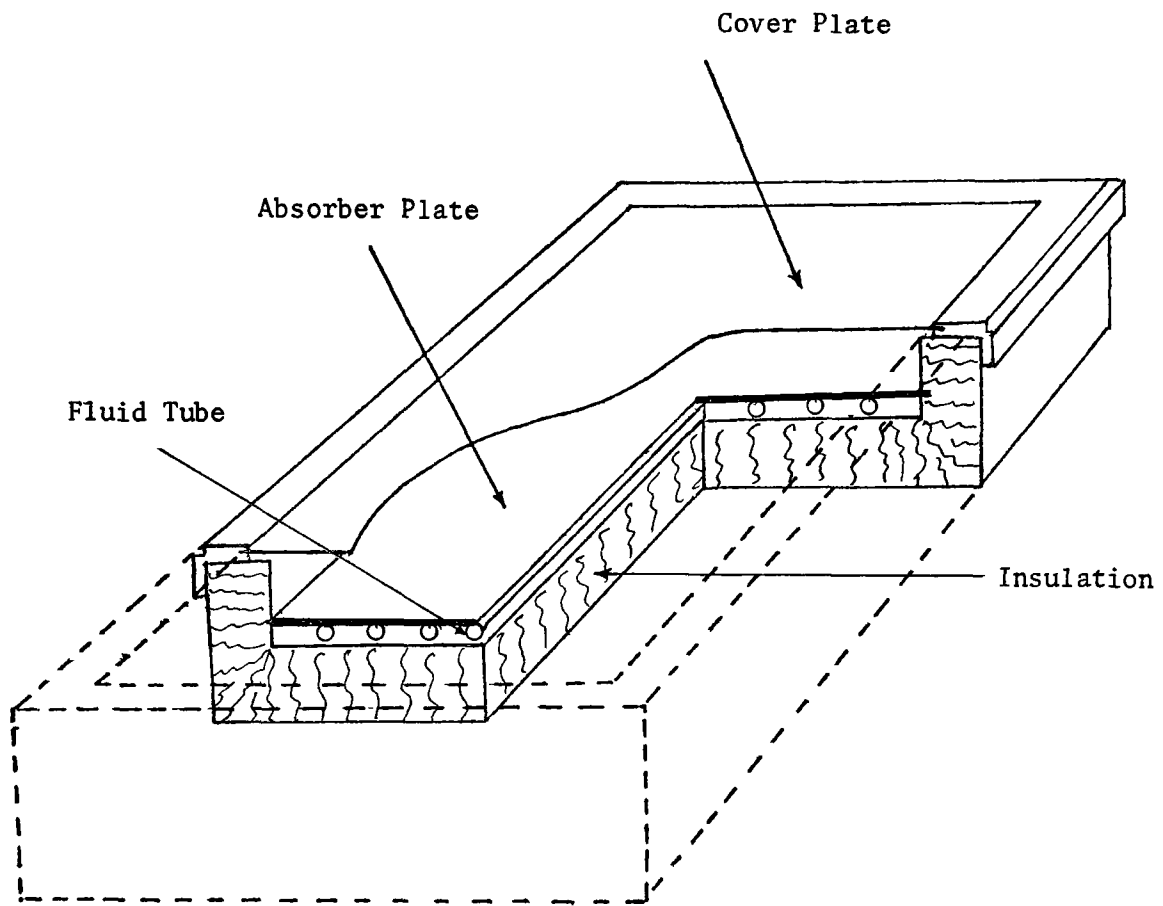


Fig. 3.3 A Typical Liquid Type Flat-Plate Collector

energy is used, in part, to maintain the heat losses from the plate to the surroundings by conduction, convection and radiation; the remainder is available to be transferred to the heat removal fluid (water) flowing through the tubes attached to the absorber plate.

Flat-plate collectors have been used for heating water, air or gases for many years, and the basic technology is well understood. In general practice, the main objective has been to collect maximum useful heat at minimum cost. However, the use of flat-plate collectors on a large scale is restricted by economic reasons rather than technological. Some of the important features of flat-plate collector components are discussed next.

3.2.1 Collector Covers

To increase the useful energy gain, the upward losses are reduced to the lowest attainable value by providing one or more sheets of transparent cover, usually air spaced, on the front side of the absorber plate. The cover must be able to transmit as much solar radiation as possible, and tends to be opaque to the infrared radiation. Glass and plastic glazings are usually used as covers.

Low iron content glass (clear when viewed on end, not greenish) which transmits 85 to 90% of the incoming short wave solar radiation and practically opaque to the long wave infrared radiation emitted by the absorber plate is generally used to glaze the solar collectors. Usually a 3 mm thick glass sheet is

used, but sometimes thicker sheets are employed to provide enough strength against the wind loading.

Although glass is strong, weather-resistant, and will not deteriorate, it is rather hard to work with and is breakable and expensive.

Some of the varieties of plastics now available possess similar optical properties as those of glass. They are often preferred to glass since they are lighter, easier to handle and cheaper. However, they may deteriorate under high temperatures, exposure to ultraviolet light and weathering.

One to three layers of glazing are usually recommended, depending on the climatic conditions and the efficiency required of the collector. Each additional layer of glazing increases the cost of the collector and reduces the solar radiation striking on the absorber plate. It reduces, however, the thermal losses up to a certain degree. Two layers of glazing are recommended for the Canadian climate. However, if the absorber has a selective coating, one glazing layer may be most appropriate.

3.2.2 Absorber Plate

The absorber plate is the most important, and most complex and expensive component of the solar collector. To ensure the maximum useful energy collection, the absorber plate must absorb most of the short wave radiation reaching it through the glazing while not emitting the long wave infrared radiation. The absorptance and emittance of the absorber surface for short wave

and long wave radiation, respectively, depends on the nature and color of the coating and upon the incident angle. For simplicity, the absorber plate can be blackened with a flat black paint which is a good absorber but also a good emitter. Silicon paints and 3M's black velvet paint are the most popular ones being used for flat-plate collectors.

At high temperatures flat-black paints become inefficient, and the absorber plate may be coated with a selective coating which may be produced by electroplating or chemically. Selective surfaces have a high absorption and a low emission, and are very efficient. However, the process is relatively expensive and these surfaces tend to deteriorate rapidly.

The absorber plate material is selected on the basis of the heat collection mode employed. If the heat transfer fluid is in contact with the entire surface area of the absorber plate, the heat transfer properties of the plate are less important and an inexpensive material can be used. The materials which are generally used for collector plates, in decreasing order of cost and thermal conductivity are: copper, aluminium and steel.

The major problem with aluminium or steel is that both are subject to corrosion and deterioration in the absence of a corrosion inhibitor.

Freezing in a water-type system is another problem in winter and so either a self-draining system or an anti-freeze solution must be introduced.

A corrosion inhibitor and an anti-freeze solution, if mixed

with water, must be used in a close loop system where the additives are isolated from the domestic water supply. This necessitates a heat exchanger in the collector loop and the water to be heated.

3.2.3 Fluid Passages

To direct the heat transfer fluid from the inlet header to the outlet header in some conventional type of flat-plate collectors, copper tubes are soldered or fastened to the upper or lower surface of one absorber plate. For better thermal contact with the absorber plate, metal to metal contact is preferred. Sometimes bonded plates with integral tubes are employed, but such plates require mass production facilities.

3.2.4 Insulation

To reduce the heat loss from the absorber plate to the surroundings through the back side and sidewalls of the collector, 5 cm to 10 cm collector of insulation is provided. A low thermal conductivity material which can stand the stagnation temperatures of the collectors may be used.

3.2.5 Casing

The casing which contains the collector plate and insulation and which supports the glazing may be made with treated wood and moisture-resistant hard woodbacking, or it may be constructed from sheet steel, made to resist corrosion. Plastic cases with a

glassfiber lining have been used to a certain extent; they are weather-resistant, do not corrode and well adapted to mass production.

3.3 Flat-Absorber Cylindrical Parabolic Concentrator

In a flat-absorber cylindrical parabolic concentrator with low concentration ratio, a flat-plate absorber blackened on both sides is placed parallel to the focal plane and symmetric to the focal axis of the cylindrical parabolic reflecting surface. The reflecting surface focuses the direct radiation on the back side of the absorber while the front side, facing the sun, receives both the direct and diffuse radiation [see Figure 3.4].

By placing several sections of these cylindrical parabolic channels in series a reasonably large collector surface area can be obtained. To protect the reflecting surface from dust and snow etc., one or more layers of glazing may be used which also reduces the thermal losses from the absorber to the surroundings.

3.4 Design Characteristic of Collectors Tested

The important design features of the conventional flat-plate collector are illustrated in Figure 3.5. A 0.8 mm thick, 0.53 x 1.13 m rectangular copper plate, painted with 3M Velvet paint on the front side was placed in a weather treated wooden box. Circular fluid passages of 1.27 cm OD were soldered 7.62 cm apart on the back side of the plate. To reduce the top losses the plate was covered with two transparent sheets of SUN-LITE

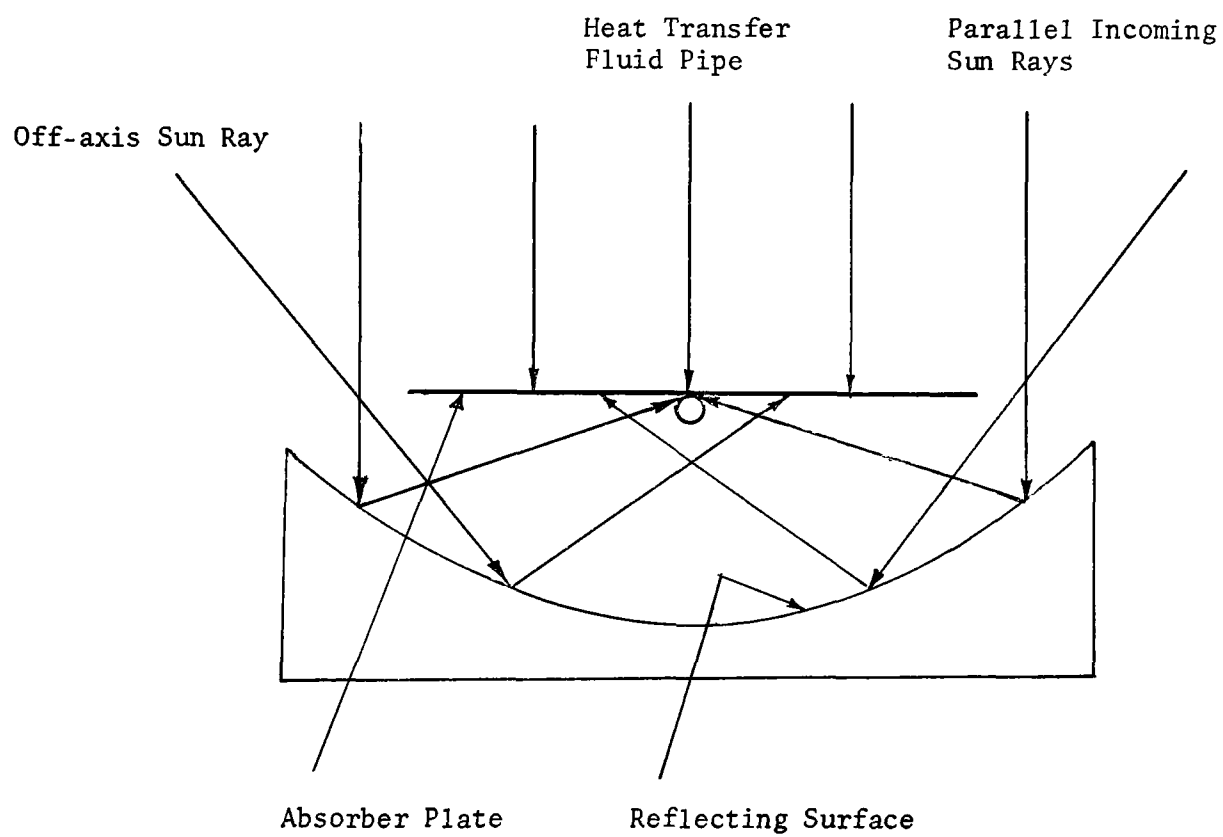


Fig. 3.4 Solar Radiation Absorption in Flat-Absorber CPC

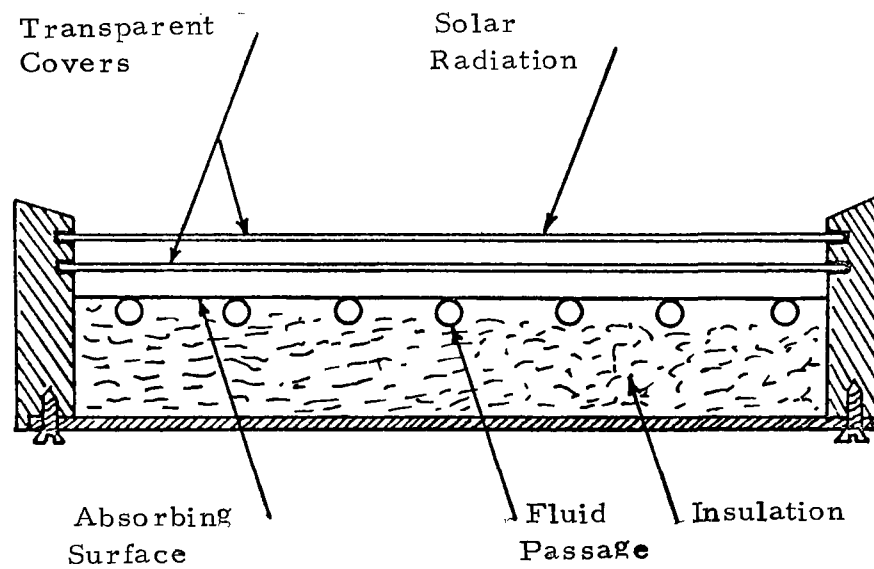


Fig. 3.5 Cross - Section of the Flat-Plate Solar Collector

premium; 0.55 m wide and 1.15 m long. The back side of the wooden box was covered with 6 mm thick plywood, to enclose the 5 cm thick fiberglass insulation.

The cross-sectional view of the flat-absorber CPC is shown in Figure 3.6. This collector retains most of the design features of a flat-plate collector. Similar design considerations as those used for the flat-plate collector regarding glazing, absorber plate and casing, etc., are also followed for this collector. The complete module consists of four identical cylindrical parabolic channels. The detailed design of an individual channel is shown in Figure 3.7. The aluminium reflecting surface (3M SCOTCHCAL film) was glued on preshaped cylindrical parabolic troughs. Four sections of absorber plate painted black on both sides were placed at the focus of each parabolic section. The plate measured 7.62 cm wide by 1.22 m long. The heat transfer fluid circulates through copper pipes of 1.27 cm OD which were soldered at the center of the back side of the absorber plates. The wooden casing supports two air spaced transparent covers: 45.7 cm wide by 1.24 m long, on the front side of the absorber; 5 cm thick fiberglass insulation on the inside of the back cover.

3.5 Collector Orientation and Tilt

The solar radiation impinging upon the collector surface strongly depends on the tilt angle from the horizontal surface and the collector azimuth angle. The collector tilt angle depends on the latitude, but also on the different climate and

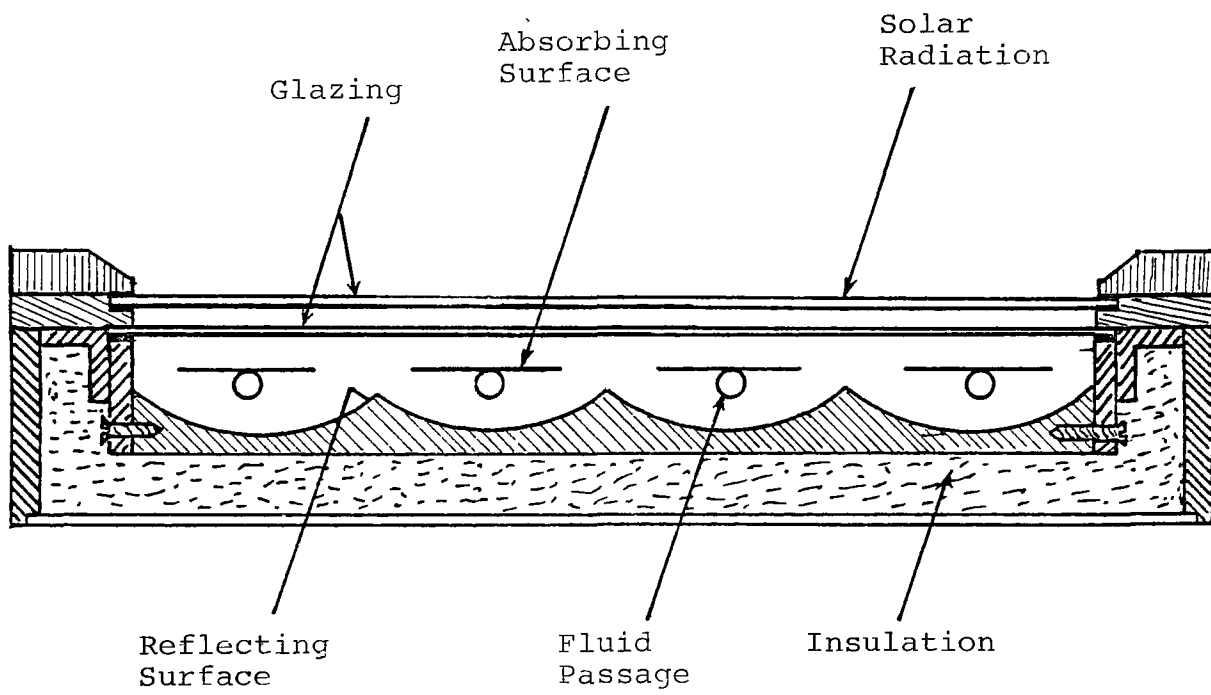


Fig. 3.6 Cross-section of Cylindrical Parabolic Concentrator

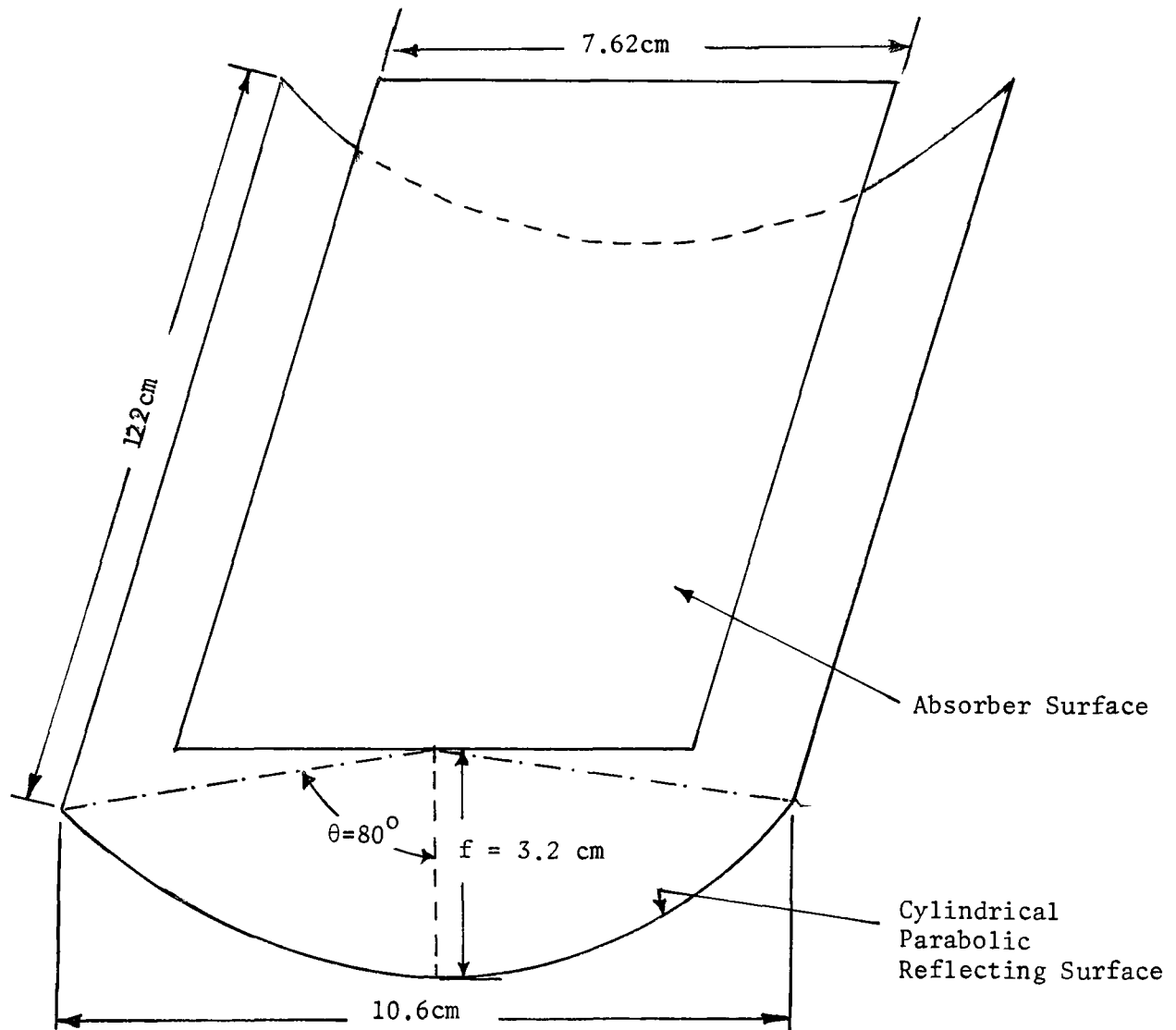


Fig. 3.7 Design Specification of Parabolic Channel

demand structure. Since solar heating systems are designed to maximize the energy collected during the winter season, a collector surface oriented due south and placed at an angle perpendicular to the sun, usually at the angle of latitude $+20^{\circ}$, would result in maximum energy collection (See Figure 3.8)[4].

However, the deviation by $\pm 15^{\circ}$ from the optimum positions would not reduce the system performance significantly. For summer conditions the best tilt angle is the latitude minus 20° , and for year-round collection a tilt angle slightly greater than latitude, say 5° , would be most appropriate.

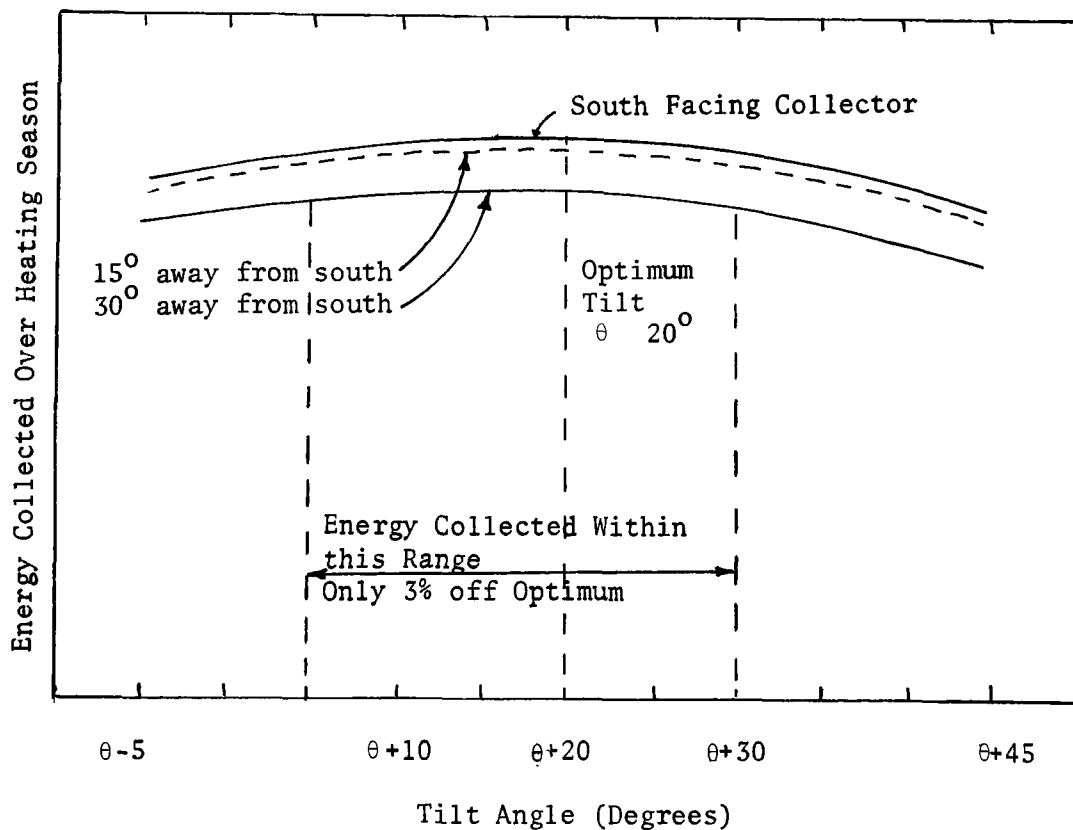


Fig. 3.8 Variation of Seasonal Energy Collected with Tilt Angle and Azimuth Angle

CHAPTER 4

PERFORMANCE OF SOLAR COLLECTORS

4.1 Introduction

In general, the performance of a solar collector is predicted by its collection efficiency which is the ratio of the useful energy gain over a specified time period to the incident solar energy over the same time period. The instantaneous rate of useful energy gain at which absorbed energy is removed from the collector by the heat-transfer fluid is the difference between the amount of incident solar radiation ultimately absorbed by the blackened absorber plate and the heat loss from the collector caused by the temperature gradient existing between the absorber plate and the outside air.

The amount of energy absorbed is affected by (i) solar radiation reaching the tilted collector surface, (ii) number and spacing of cover plates, (iii) material used for glazing (iv) absorptivity of the absorber surface.

The thermal losses from the absorber plate depend on the collector operating conditions such as the average plate-temperature, the local weather conditions and ambient temperatures, the wind velocity and sky temperature. The average plate temperature is primarily determined by the temperature of the inlet fluid, and to a lesser extent by the fluid flow rate and the insolation rate. In addition, other factors which

influence the thermal losses are: emissivity of the absorber surface, number and spacing of cover plates, material used for glazing, and back and edge insulation.

Generally speaking, the thermal performance of a collector varies with geographical location, local weather conditions, and time of year. An analysis of the thermal performance of the conventional flat-plate collector and flat-absorber CPC which is the special case of a flat-plate collector is described in the following sections.

4.2 Performance of Flat-Plate Collector

The first law of thermodynamics applied to the solar collector for steady state operation gives: [9]

$$q_u = q_a - q_\ell \quad (4.2.1)$$

where

q_a = rate of energy absorbed per unit collector area

q_u = rate of useful heat collected per unit area

q_ℓ = total heat loss rate per unit area

The undesirable heat loss, q_ℓ , in (4.2.1) is given by

$$q_\ell = U_L (T_p - T_a) \quad (4.2.2)$$

in which

U_L = overall heat loss coefficient

T_p = average plate temperature

T_a = ambient air temperature

The useful energy gain, q_u , can be expressed as

$$q_u = q_a - U_L(T_p - T_a) \quad (4.2.3)$$

To obtain q_u , the problem is the evaluation of the absorbed energy q_a and the overall loss coefficient U_L . The only difficulty in using (4.2.3) is that the plate temperature is usually non-uniform caused by the location of the fluid flowing passages. In fact, the temperature of the entering fluid is usually known, and can be measured and controlled to a predetermined level. It is, therefore, desirable to replace the average plate temperature T_p , with the inlet fluid temperature, $T_{f,i}$. For this purpose, Hottel and Whillier [8] have defined a heat-removal factor, F_R , which is the ratio of the actual heat gain to the useful heat gain attainable if the collector plate is at the inlet fluid temperature. Then

$$q_u = F_R[q_a - U_L(T_{f,i} - T_a)] \quad (4.2.4)$$

A complete discussion of the heat-removal factor F_R is available in reference [6]. Effectively, the evaluation of q_u involves the calculation of the terms q_a , U_L and F_R in (4.2.4).

4.2.1 Absorbed Energy in the Collector

The fraction of the incident solar radiation absorbed by the blackened absorbing plate is influenced by the transmittance of the cover system and the absorptivity of the absorbing surface. A detailed mathematical analysis of the calculation of the transmittance of n-plate cover system, all of same material, is to be found in Duffie and Beckman [6] and Whillier [9]. The transmittance of an n-plate cover system is given by

$$\tau_{1,2\dots n} = e^{-(k_1 L_1 + k_2 L_2 + \dots k_n L_n)} \times \left[\frac{(1-\rho_1)}{1+2(n-1)\rho_1} + \frac{(1-\rho_2)}{1+2(n-1)\rho_2} \right] / 2 \quad (4.2.5)$$

where

τ = transmittance of cover plate system making allowances for the losses due to both reflection at the air interfaces and absorption in the material

k = extinction coefficient for the material for solar radiation. For glass, the value of k varies from 0.04/cm for "water white" glass to 0.32/cm for poor glass

L = actual path of the radiation through the material (see Figure 4.1)

ρ_1 = surface reflectance in the plane perpendicular to the plane of incidence

ρ_2 = surface reflectance in the plane parallel to the plane

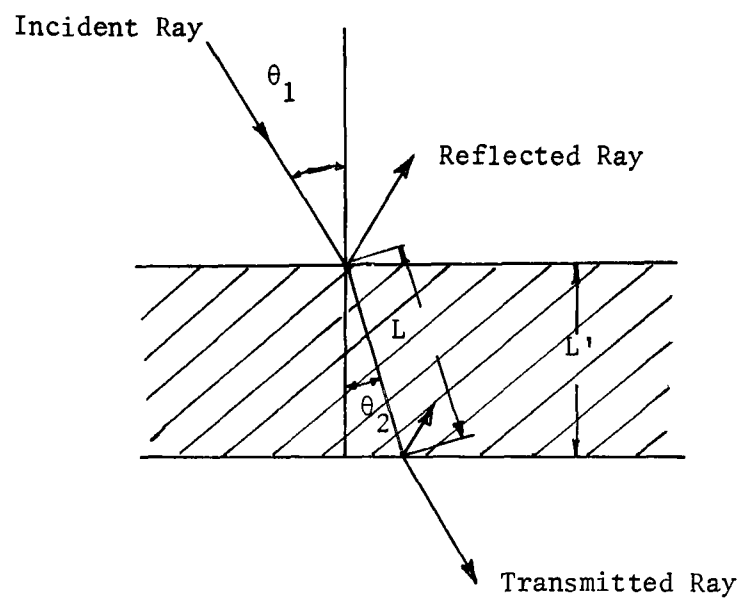


Fig. 4.1 Transmission of Solar Radiation Through Glass

of incidence

If θ_1 and θ_2 are the incident and refracted angles, respectively, then

$$\rho_1 = [\sin(\theta_1 - \theta_2) / \sin(\theta_1 + \theta_2)]^2$$

$$\rho_2 = [\tan(\theta_1 - \theta_2) / \tan(\theta_1 + \theta_2)]^2$$

4.2.2 Transmittance-Absorptance Product ($\tau\alpha$)

It is noticed from Figure 4.2 that a fraction, $(1-\alpha)\tau$, of the total radiation striking the absorbing surface is reflected back to the cover system, and a part of that $(1-\alpha)\tau\rho_d$ is reflected back to the absorbing surface. The factor ρ_d is the reflectance of the cover-plate system for incident diffuse radiation estimated by using the specular reflection at an incidence angle of 60° . The approximate values of ρ_d for one, two, three, and four glass cover system are 0.16, 0.24, 0.29, and 0.32, respectively [6]. The process of multiple reflection continues and the ultimate energy absorbed is given as

$$(\tau\alpha) = \frac{\tau\alpha}{1 - (1-\alpha)\rho_d} \quad (4.2.6)$$

in which α is the absorptivity of the absorbing surface.

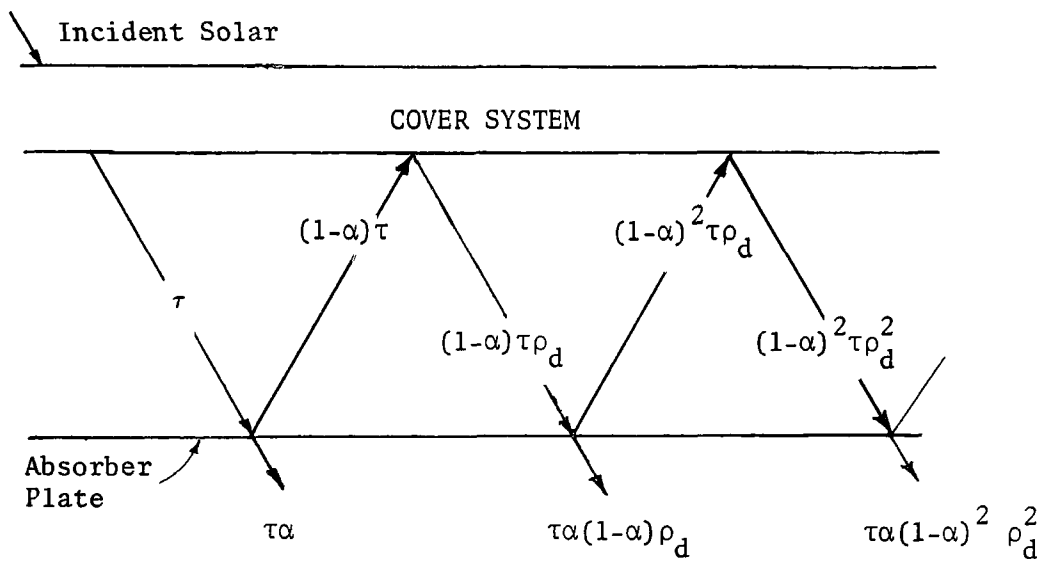


Fig. 4.2 Absorption of Solar Radiation by Absorber Plate

4.2.3 Effective Transmittance-Absorptance Product

As discussed above, the calculation of the $(\tau\alpha)$ product using (4.2.6) takes into account the absorption losses within the transparent cover system. However, all of the radiation retained by cover plates is not lost in the system, since a part of this absorbed energy tends to increase the cover temperature which in turn reduces the outward flow of heat from the absorber plate. To utilize the usefulness of this effect, the effective transmittance-absorptance product concept was introduced and can be calculated using the relation [9]

$$\begin{aligned}
 (\tau\alpha)_e = (\tau\alpha) + a_1(1 - e^{-k_1 L_1}) + a_2 \tau_1(1 - e^{-k_2 L_2}) \\
 + a_3 \tau_1 \tau_2(1 - e^{-k_3 L_3}) + \dots
 \end{aligned}
 \quad (4.2.7)$$

where the constants 'a' are given in Table 4.1, and subscripts 1, 2, 3 etc. refer to the first (outer) second and third layer of the cover system.

Table 4.1 Constants for Use in Equation 4.2.7

<u>Covers</u>	<u>a_i</u>	<u>$\epsilon_p = 0.95$</u>	<u>$\epsilon_p = 0.50$</u>	<u>$\epsilon_p = 0.10$</u>
1	a_1	0.27	0.21	0.13
2	a_1	0.15	0.12	0.09
	a_2	0.62	0.53	0.40
3	a_1	0.14	0.08	0.06
	a_2	0.45	0.40	0.31
	a_3	0.75	0.67	0.53

4.2.4 Effect of Dust and Shading

The long-time experimental results for different locations have shown that the effect of dust on useful heat collection in industrialized regions is as low as 1% [10] while for arid regions it was found close to 8% [11]. However, for design purposes, it is suggested to allow a 2% loss due to dust on the glass, and the dust loss factor (1-D) is taken as 0.98 [9].

Solar radiation reaching the absorber plate is further reduced by shading of the absorber plate by the side walls that support the glass cover, especially when the sun is not at normal incidence to the collector surface. The effect of shading on the useful energy collection varies with incidence angle as shown in Figure 4.3 [9]. However, at normal incidence the recommended value of the factor (1-S) is 0.97 [9]. Thus, the net energy absorbed by the collector becomes

$$q_a = [(1-D)(1-S)(\tau\alpha)_e]I_t \quad (4.2.8)$$

in which I_t is the total radiation incident on the collector surface.

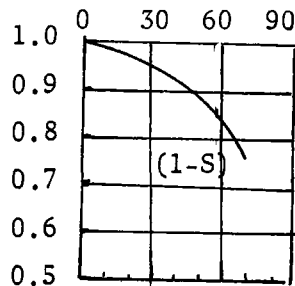


Fig. 4.3 Effect of Angle of Incidence on Shading Factor

The difficulty in using (4.2.8) is that not all of the radiation reaching the collector surface is direct radiation, a portion of total radiation comes from the sky as diffuse radiation. Hence, in determining the transmittance factors, it is necessary to treat both direct and diffuse components individually. Whillier [9] has suggested a modified form of (4.2.8) with the assumption that the average angle of incidence, θ_i , for diffuse radiation can be taken as 50° . Then

$$q_a = I_b [(1-D)(1-s)(\tau\alpha)_e]_{\theta_i} + I_d [(1-D)(1-s)(\tau\alpha)_e]_{\theta_i=50^\circ} \quad (4.2.9)$$

where I_b and I_d are the direct and diffuse components of the solar radiation incident on the tilted collector surface. On further simplification

$$q_a = I_t \left[(1-D)(1-s)(\tau\alpha)_e \right]_{\theta_i} \left[\frac{I_b}{I_t} + \frac{I_d}{I_t} \frac{\{(1-D)(1-s)(\tau\alpha)_e\}_{\theta_i=50^\circ}}{\{(1-D)(1-s)(\tau\alpha)_e\}_{\theta_i}} \right] \quad (4.2.10)$$

The numerical value of the quantity in square brackets in (4.2.10) is close to unity. In fact, this factor is seldom less than 0.98 during the sunny parts of the day. Then

$$q_a = 0.98 I_t [(1-D)(1-s)(\tau\alpha)_e] \theta_i = F_e \dot{I}_t \quad (4.2.11)$$

in which F_e is the fraction of incident solar radiation usually absorbed by the absorber surface after taken into account all the optical losses involved, dirt and shading effects.

4.3 Collector Overall Loss Coefficient

The temperature gradient existing between the absorber plate and the outside air causes the heat loss from a collector to flow outward through the transparent covers, through the bottom insulation, and through the side insulation by the usual heat transfer processes. The calculation of these losses involves a complicated analysis. For convenience the concept of overall loss coefficient was introduced [6] which is the sum of all the three loss coefficients: upward loss coefficient, back loss coefficient, and edge loss coefficient. To evaluate the overall loss coefficient, the thermal network of the collector with two cover system is drawn in Figure 4.4.

The absorbed energy ($F_e I_t$) at the collector plate is distributed to useful energy which is transferred to the circulating fluid and to the losses. The thermal loss through the bottom insulation is represented by the thermal resistance R_B , upward losses are represented by thermal resistances $R_{p,c1}$, $R_{c1,c2}$ and $R_{c2,s}$, and R_{edge} represents the losses through the wall insulation.

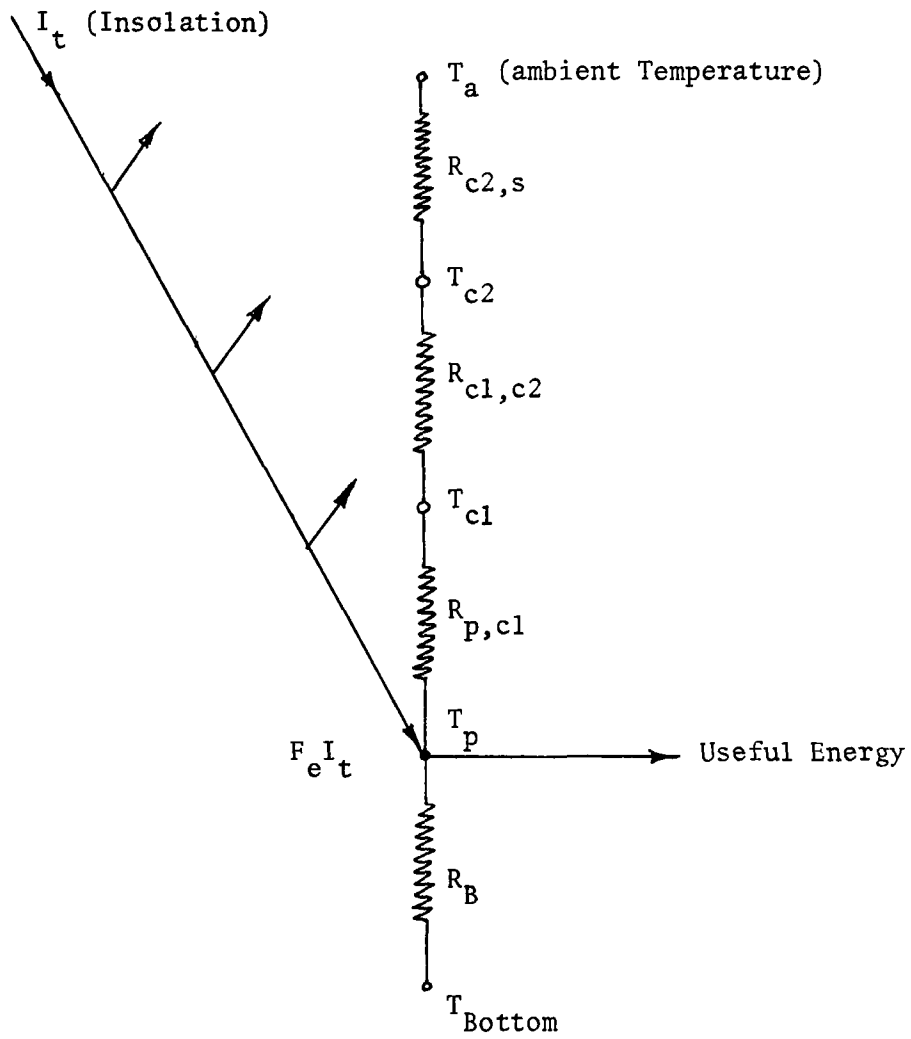


Fig. 4.4 Thermal Network for Flat-Plate Collector

4.3.1 Upward Heat Loss Coefficient

The upward heat loss from the absorber plate through the transparent covers depends primarily on the absorber plate temperature T_p and the ambient temperature T_a , and to a lesser extent on the number of cover plates and their spacing, collector tilt angle from the horizontal, and wind speed over the top plate. The transfer of heat through the cover plates occurs by radiation and convection between the air spaced parallel plates. Duffie and Beckman [6] have expressed the basic equation of heat loss from a heated plate covered by one or more air spaced transparent cover plates as

$$q_l = h_{c_{p,c1}}(T_p - T_{c1}) + h_{r_{p,c1}}(T_p - T_{c1}) \quad (4.3.1)$$

for transfer from collector plate to the cover 1, and

$$q_l = h_{c_{c1,c2}}(T_{c1} - T_{c2}) + h_{r_{c1,c2}}(T_{c1} - T_{c2}) \quad (4.3.2)$$

for transfer from cover 1 to cover 2. Similar expressions may be obtained when additional covers are present.

In (4.3.1) and (4.3.2) h_c and h_r are the convective and radiative loss coefficients between the parallel plate as specified by the lower subscripts [Appendix B].

The thermal resistance to heat flow from plate to cover 1, $R_{p,c1}$ can be expressed as

$$R_{p,c1} = \frac{1}{h_{c_{p,c1}} + h_{r_{p,c1}}} \quad (4.3.3)$$

and similarly, the thermal resistance $R_{c1,c2}$ is

$$R_{c1,c2} = \frac{1}{h_{c_{c1,c2}} + h_{r_{c1,c2}}} \quad (4.3.4)$$

For the outmost cover plate the heat loss is

$$q_{\ell} = h_w(T_{c2} - T_a) + h_{r_{c2,s}}(T_{c2} - T_a) \quad (4.3.5)$$

assuming the effective radiation temperature of the sky equals T_a , the temperature of the outside air, h_w is the convection heat transfer coefficient for wind blowing on the surface of the outmost cover plate; and can be determined from (B-15) as given in Appendix B.

The thermal resistance to the surroundings is given by

$$R_{c2-s} = \frac{1}{h_w + h_{r_{c2,s}}} \quad (4.3.6)$$

The upward heat transfer coefficient U_t , for two cover system is given by

$$U_t = \frac{1}{R_{p,c1} + R_{c1,c2} + R_{c2,s}} \quad (4.3.7)$$

Equation 4.3.7 gives the exact value of the top loss

coefficient, U_t . However, an approximate empirical relation was developed by Hottel and Wertz [10] as given by

$$U_t = \frac{1}{(N/c) \frac{(T_p - T_a)}{(T_p - T_a)/N + f} + 1/h_w} + \frac{\sigma(T_p + T_a)(T_p^2 + T_a^2)}{\left(\frac{1}{\epsilon_p}\right) + \left(\frac{2N+f-1}{\epsilon_g}\right) - N} \quad (4.3.8)$$

where

N = number of cover plates

f = constant whose values vary with wind velocity as it affects h_w . f has values of 0.76, 0.36, and 0.24 for wind speeds of 0, 16, and 32 km/h

c = coefficient whose value is dependent on collector tilt

For a collector tilt of 45° , Klein [12] suggested a more accurate empirical relation for U_t . This new relation fits the graphs of U_t for plate temperatures ranging from 40°C to 130°C .

$$U_t = \left(\frac{N}{(344/T_p)(T_p - T_a)/N + f} + \frac{1}{h_w} \right)^{-1} + \frac{\sigma(T_p + T_a)(T_p^2 + T_a^2)}{[\epsilon_p + 0.0425N(1 - \epsilon_p)]^{-1} + [(2N+f-1)/\epsilon_g] - n} \quad (4.3.9)$$

where

N = number of glass covers

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

h_w = wind heat transfer coefficient

For tilt angles other than 45° Klein developed another equation as

$$\frac{U_t(\Sigma)}{U_t(45^\circ)} = 1 - (\Sigma - 45^\circ)(2.59 - 1.44\epsilon_\rho) \times 10^{-3} \quad (4.3.10)$$

where Σ is the tilt angle in degrees.

4.3.2 Back Heat Loss Coefficient

The energy loss by conduction through the bottom insulation depends on the thermal conductivity K_b , and the thickness of the material, L_b . Assuming that the back losses are at the same temperature as the top losses, the back loss coefficient U is approximately given by

$$U_b = \frac{K_b}{L_b} = \frac{1}{R_b} \quad (4.3.11)$$

where R_b is the resistance to heat flow through the bottom insulation.

4.3.3 Edge Heat Loss Coefficient

In a well designed large collector, the edge losses may be negligible, but in a small collector where the perimeter to collector area is relatively small, edge losses may be significant. It is recommended that the edge insulation thickness should be the same as the bottom insulation [9].

The edge loss coefficient based on the collector frontal area is given by

$$U_{\text{edge}} = \frac{K_e}{l_e} \left(\frac{A_p}{A_c} \right) \quad (4.3.12)$$

where

A_p = collector perimeter area

A_c = collector transparent frontal area

K_e = thermal conductivity of edge insulation

l_e = thickness of edge insulation

Finally, the overall loss coefficient, U_L is obtained by adding together the three loss coefficients; then

$$U_L = U_t + U_b + U_{\text{edge}} \quad (4.3.13)$$

4.4 Useful Energy Transferred to the Fluid

To evaluate the useful energy gained by the working fluid, consider the classical sheet tube configuration as shown in Figure 4.5. The plate region between the centerline and the tube base is a fin of length $(W-D_o)/2$. If the temperature gradient in the flow direction is assumed to be negligible, an energy balance on an elemental area of width Δx and unit length along the flow direction, as shown in Fig. 4.6, can be written as [6]

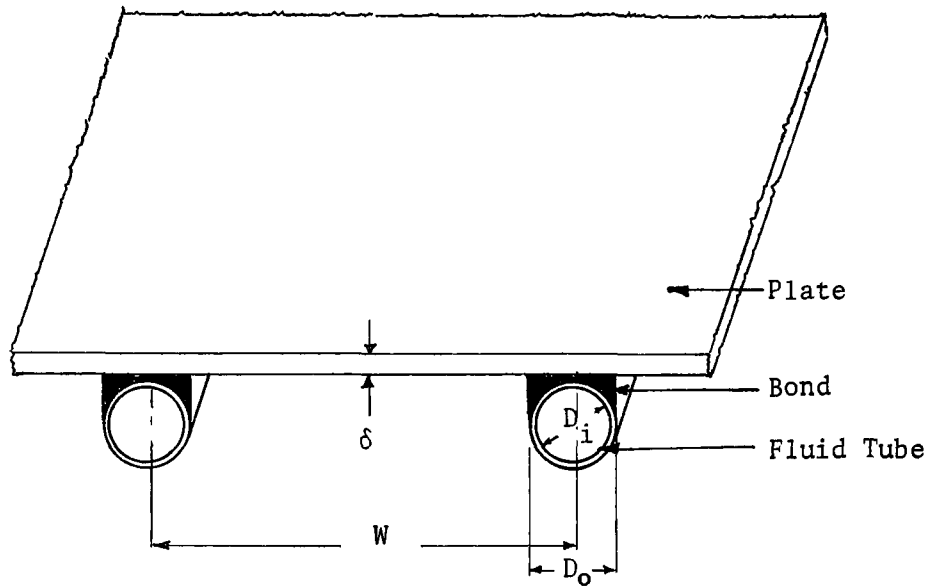


Fig. 4.5 A Typical Tube-in-sheet Collector Plate

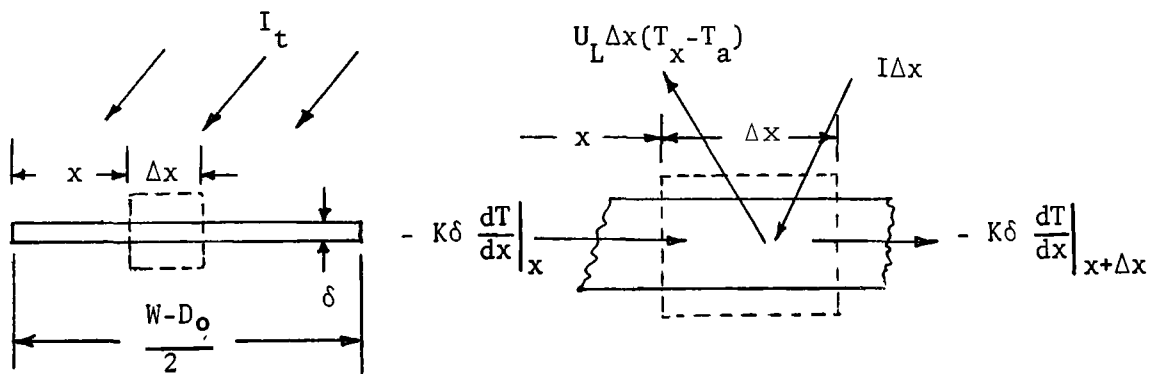


Fig. 4.6 Energy Balance on Fin Element

$$q_a \Delta x + U_L \Delta x (T_a - T) + (K\delta \frac{dT}{dx})|_x - (-K\delta \frac{dT}{dx})|_{x+\Delta x} = 0 \quad (4.4.1)$$

The manipulation of the above equation yields, a second order differential equation which describes the temperature distribution across the collector plate. This second order equation may be written as

$$\frac{d^2 T}{dx^2} + \frac{q_a - U_L (T_b - T_a)}{K\delta} = 0 \quad (4.4.2)$$

The boundary conditions are:

$$\left. \frac{dT}{dx} \right|_{x=0} = 0 \quad T|_{x=(W-D_0)/2} = T_b$$

The solution of (4.4.2) is given by the expression

$$T = c_1 e^{ax} + c_2 e^{-ax} + \left(\frac{q_a}{U_L} \right) + T_a \quad (4.4.3)$$

where $a = \sqrt{(U_L/K\delta)}$ and c_1 and c_2 are constants.

Applying the boundary conditions yields the result

$$T = T_b \frac{\cosh ax}{\cosh a[(W-D_0)/2]} + \left(T_a + \frac{q_a}{U_L} \right) \left[1 - \frac{\cosh ax}{\cosh a[(W-D_0)/2]} \right] \quad (4.4.4)$$

The energy conducted to the region of the tube from both sides of the plate per unit of length in the flow direction has been determined as

$$q_{\text{fin-base}} = - 2K\delta \left[\frac{dT}{dx} \right]_{x=(W-D_o)/2} \quad (4.4.5)$$

Substitution of (4.4.4) results in

$$q_{\text{fin-base}} = (W-D_o) [q_a - U_L(T_b - T_a)] F \quad (4.4.6)$$

where

$$F = \frac{\tanh a(W-D_o)/2}{a(W-D_o)/2} \quad (4.4.7)$$

Thus parameter F is known as the fin efficiency.

The energy gain of the section of plate directly above the tubes due to absorption of solar radiation is given by

$$q_{\text{tube-base}} = D_o [q_a - U_L(T_b - T_a)] \quad (4.4.8)$$

The total energy gain of the collector per unit length in flow direction becomes

$$q_u = [D_o + (W-D_o)F] [q_a - U_L(T_b - T_a)] \quad (4.4.9)$$

This energy is ultimately transferred to the fluid. The resistance to heat flow from plate to the fluid results from three components: (i) resistance due to bond, (ii) resistance due to the wall thickness of the tube, and (iii) fluid to tube wall resistance. Hence

$$q_u = \frac{T_b - T_f}{(1/C_b) + (1/C_w) + (1/\pi D_i h_{fi})} \quad (4.4.10)$$

where

C_b = bond conductance

C_w = tube wall conductance

D_i = inside tube diameter

$h_{f,i}$ = heat transfer coefficient between fluid and tube wall

The bond conductance C_b on a per unit length basis can be estimated using the expression

$$C_b = \frac{K_b l_b}{\delta_b} \quad (4.4.11)$$

Whillier and Saluja [13] have recommended the value of bond conductance, C_b , as $25 \text{ W/m}^2\text{C}$ for metal to metal contact.

The tube wall conductance, C_w , can be calculated from [13] and is given by the expression

$$C_W = \frac{2 K(D_o^2 - D_i^2)}{D_o^2 \ln(D_o/D_i) - [(D_o^2 - D_i^2)/2]} \quad (4.4.12)$$

Some of the empirical relations for $h_{f,i}$ recommended by various investigators are listed in Appendix B. Substituting the value of T_b from (4.4.9) in (4.4.8) gives an expression for q_u in terms of the known parameters as

$$q_u = WF'[q_a - U_L(T_f - T_a)] \quad (4.4.13)$$

where F' , the collector efficiency factor, is given by

$$F' = \frac{1/U_L}{W[1/U_L(D_o + (W - D_o)F + (1/C_b) + (1/C_w) + (1/\pi + D_i h_{fi}))]} \quad (4.4.14)$$

and represents the ratio of actual useful energy gain to the useful energy gain if the collector absorbing surface had been at the average fluid temperature. The denominator of (4.4.14) is recognized as the heat transfer resistance from fluid to the ambient air.

The temperature distribution in the flow direction can be derived by considering the energy balance on a fluid element as in Fig. 4.7 flowing through a pipe of length Δy which is receiving a uniform flux q_u ; so that

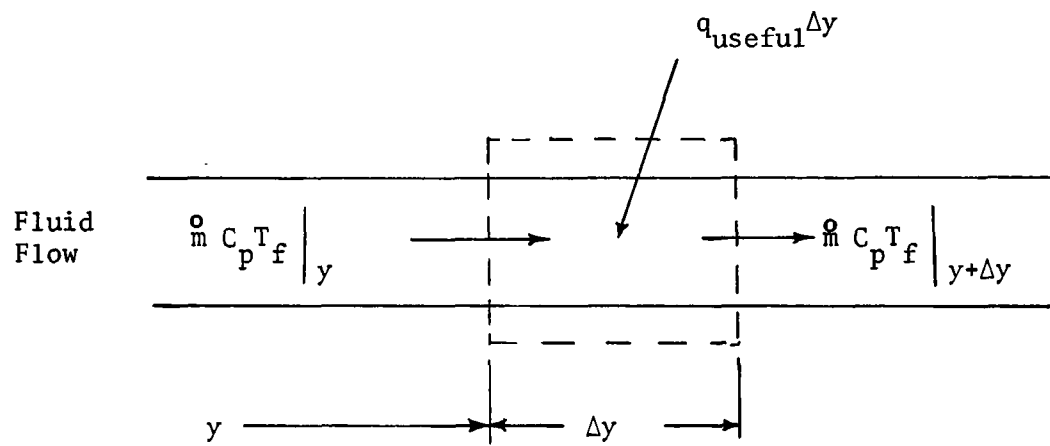


Fig. 4.7 Energy Balance on Fluid Element

$$\dot{m}C_p T_f|_y - \dot{m}C_p T_f|_{y+\Delta y} + q_u \Delta y = 0 \quad (4.4.15)$$

or

$$\dot{m}C_p \frac{dT}{dy} - WF'[q_a - U_L(T - T_a)] = 0 \quad (4.4.16)$$

If F' and U_L are assumed constant and independent of position, the solution of (4.4.16) for any position of y is

$$T = T_a + \left(\frac{q_a}{U_L} \right) - \frac{[(q_a/U_L) - (T_{f,i} - T_a)]}{\exp(-U_L W_c F' y / \dot{m}C_p)} \quad (4.4.17)$$

Substituting $y = L_c$, the outlet temperature $T_{f,e}$ is given as

$$T_{f,e} = T_a + \left(\frac{q_a}{U_L} \right) - \frac{[(q_a/U_L) - (T_{f,i} - T_a)]}{\exp(-U_L F' A_c / \dot{m}C_p)} \quad (4.4.18)$$

The total useful energy gain Q_u is expressed as

$$Q_u = \dot{m}C_p (T_{f,e} - T_{f,i}) \quad (4.4.19)$$

Substituting $T_{f,e}$ from (4.4.17) gives

$$Q_u = A_c F_R [q_a - U_L (T_{f,i} - T_a)] \quad (4.4.20)$$

where, F_R , the heat removal factor is given by

$$F_R = \frac{\dot{m} C_p}{A_c U_L} [1 - \exp(-F' U_L A_c / \dot{m} C_p)] \quad (4.4.21)$$

Referring to (4.4.19) and (4.4.20) the parameter F represents the ratio of actual useful energy gain to the useful energy gain if the collector plate was at fluid inlet temperature.

To facilitate the graphical representation of equation (4.4.21), a new factor F'' , which is equal to F_R/F' was defined. This parameter F'' is a function of single variation $\frac{\dot{m} C_p}{A_c U_L F'}$ and is shown in Figure 4.8.

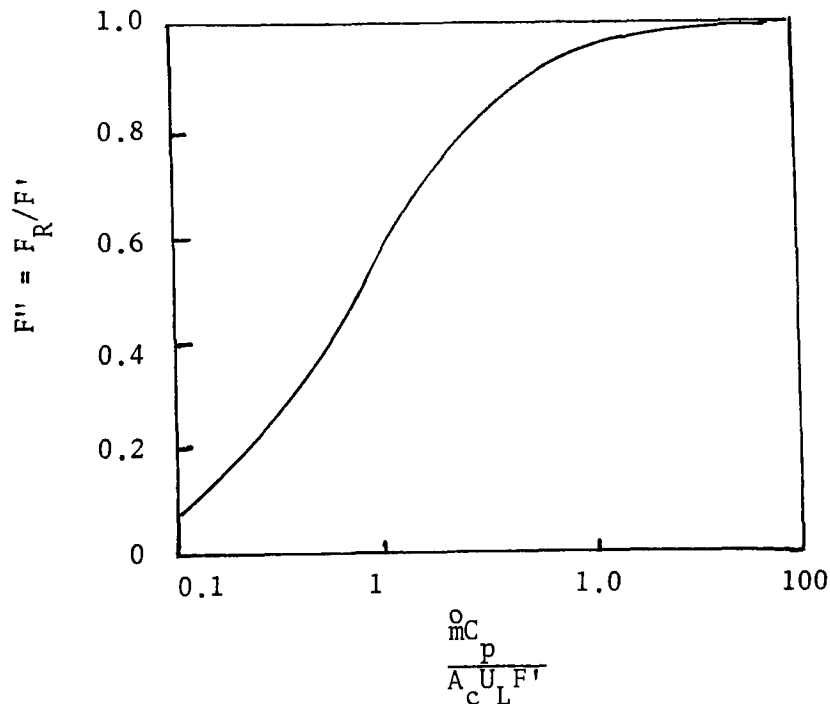


Fig. 4.8 Collector Flow Factor F'' as a Function of $\frac{\dot{m} C_p}{A_c U_L F'}$

4.5 Collection Efficiency

For a collector operating under steady-state conditions, the instantaneous efficiency may be determined from [14] and is given by the expression

$$\eta = \frac{Q_u/A_c}{I_t} = \frac{G C_p (T_{f,e} - T_{f,i})}{I_t} \quad (4.5.1)$$

where

Q_u = useful energy gain (W)

A_c = transparent frontal collector area [m]

I_t = total energy incident on the tilted collector plane
per unit time per unit area [W/m^2]

G = flow rate per unit of collectance [$kg/s-m^2$]

C_p = specific heat of the fluid [$kJ/kg \text{ } ^\circ C$]

$T_{f,e}$ = outlet temperature $^\circ C$

$T_{f,i}$ = inlet temperature $^\circ C$

Based on the useful energy gain calculated using (4.4.19)
the thermal efficiency is given by

$$\eta = F_R [F_e - U_L (T_{f,i} - T_a) / I_t] \quad (4.5.2)$$

The efficiency based on local fluid temperature can be written as

(4.5.3)

$$\eta = F'[F_e - U_L(T_f - T_a)/I_t]$$

and the efficiency based on average plate temperature is

(4.5.4)

$$\eta = [F_e - U_L(T_p - T_a)/I_t]$$

From (4.5.2), (4.5.3) and (4.5.4) it appears that the plot of efficiency against some appropriate $\Delta T/I_t$ will result in a straight line, and the values of design parameters F_R , F_e , F' and U_L may be determined by these curves.

4.6 Thermal Performance of Flat-Absorber CPC

To evaluate the thermal performance of the flat-absorber CPC, the general approach is the same as for flat-plate collectors. Equations (4.5.2), (4.5.3) and (4.5.4) can be used to determine the thermal efficiency of the flat-absorber CPC. However, the parameter F_e needs to be modified because of the additional cylindrical parabolic reflecting surface underneath the absorber plates. Fig. 4.9, illustrates the absorption of solar radiation in the flat-absorber CPC.

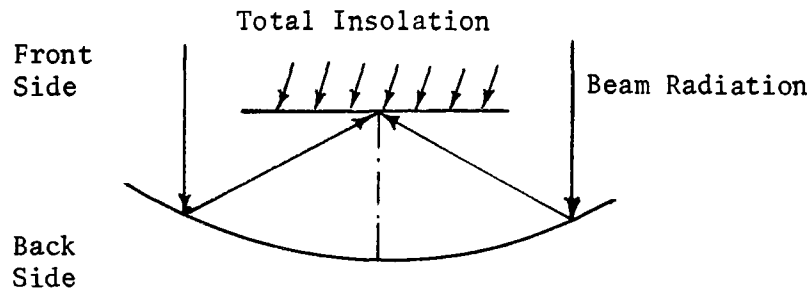


Fig. 4.9 Absorption of Solar Radiation in Flat-Plate CPC at Normal Incidence

Hassan and El-Rafaie [15] have studied the theoretical performance of a flat-absorber cylindrical parabolic concentrator, specifically concerning the energy available at the focal plane and its distribution as well as the energy concentrated on the absorber. The power received on the back side of the absorber, Q_B , can be written as

$$Q_B = N_p \rho_r F_{eb} \gamma f L_c I_b \quad (4.6.1)$$

where

N_p = reduced power, dimensionless

ρ_r = reflectivity of reflecting surface

F_{eb} = fraction of beam radiation available which is transmitted through the collector cover system

γ = intercept factor (unity for this collector)

f = focal length of cylindrical parabolic concentrator

L_c = length of concentrator

I_t = beam component of solar radiation

The dimensionless or reduced power N_p , which takes into account the shading effect on the reflecting surface caused by the flat absorber and may be expressed as

$$N_p = 4(\tan \theta/2 - \tan \psi/2) \quad (4.6.2)$$

provided that $\psi < \theta < \alpha_1$.

In (4.6.2) the angles are defined as (see Fig. 4.10)

θ = rim angle

ψ = shading angle

α_1 = first critical angle

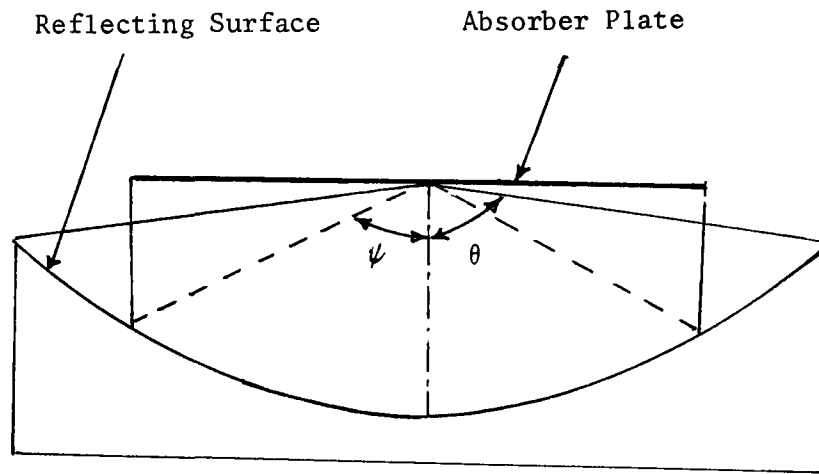


Fig. 4.10 Flat-Absorber CPC Angles

The first critical angle, α_1 , (see page 223, Eq. (11) in [15]) is defined by the expression

$$B = \frac{2}{\cos \alpha (1 + \cos \alpha) (1 - \tan \epsilon \tan \alpha)} \quad (4.6.3)$$

in which B is the width ratio defined as the ratio of absorber width to focal width, that is

$$B = W/2f \tan \epsilon \quad (4.6.4)$$

The angle ϵ is the angular radius of the sun, that is, $\epsilon = 16$ minutes. By equating (4.6.3) and (4.6.4) the first critical angle α_1 may be obtained.

The shading angle ψ from Figure (4.10) can be written as

$$\psi = 2 \tan^{-1} (1/2 B \tan \epsilon) \quad (4.6.5)$$

Since the front side of the absorber is also exposed to direct and diffuse radiation, the power received on the front side of the absorber, Q_F , may be expressed by [16]

$$Q_F = F_e A_r I_t \quad (4.6.6)$$

in which A_r is the collector receiver area.

The total power received by the absorber is therefore [16]

$$Q_A = Q_B + Q_F \quad (4.6.7)$$

From 4.6.1 and 4.6.7, the total power, Q_A , received by the collector can be expressed as

$$Q_A = N_c N_p \rho_r F_{eb} f L_c I_b + N_c F_e A_r I_t \quad (4.6.8)$$

in which N_c is the number of cylindrical parabolic channels.

The total power received per unit collector area, q_a , may be written as

$$q_a = \left[N_c N_p \rho_r F_{eb} \frac{fL_c}{A_c} \frac{I_b}{I_t} + N_c F_e \frac{A_r}{A_c} \right] I_t \quad (4.6.9)$$

or

$$q_a = F_t I_t \quad (4.6.10)$$

where F_t is the quantity in brackets in (4.6.9), and identified as the fraction of incidence energy absorbed by the receiver surface.

The useful energy gain per unit time, Q_u , is therefore given by

$$Q_u = A_c \left\{ I_t F_t - U_L \frac{A_r}{A_a} (T_p - T_a) \right\} \quad (4.6.11)$$

in which A_a is the aperture area of a cylindrical channel, and T_p is the mean plate temperature.

The instantaneous efficiency is given by

$$\eta = \frac{Q_u / A_c}{I_t} \quad (4.6.12)$$

or, in view of (4.6.11)

$$\eta = F_t - U_L \frac{A_r}{A_a} \left(\frac{T_p - T_a}{I_t} \right) \quad (4.6.13)$$

In terms of the inlet fluid temperature $T_{f,i}$, the instantaneous instantaneous efficiency can be expressed as

$$\eta = F_R \left[F_t - U_L \frac{A_r}{A_a} \frac{(T_{f,i} - T_a)}{I_t} \right] \quad (4.6.14)$$

Finally, the efficiency based on local fluid temperature is

$$\eta = F' \left[F_t - U_L \frac{A_r}{A_a} \frac{(T_f - T_a)}{I_t} \right] \quad (4.6.15)$$

4.7 Cost Effectiveness

The rating of solar collectors on the basis of cost effectiveness is important for economic considerations. The cost effectiveness C_{eff} may be defined as [17].

Per unit cost of useful energy gained by
standard collector design (PUC_1)

$$C_{eff} = \frac{\text{Per unit cost of useful energy gained by standard collector design (PUC}_1\text{)}}{\text{Per unit cost of useful energy gained by an alternate collector design (PUC}_2\text{)}} \quad (4.7.1)$$

Let η_1 and η_2 be the thermal efficiencies of the two collectors (tested under identical operating conditions) corresponding to a selected operating point on the graph in Fig. 4.11.

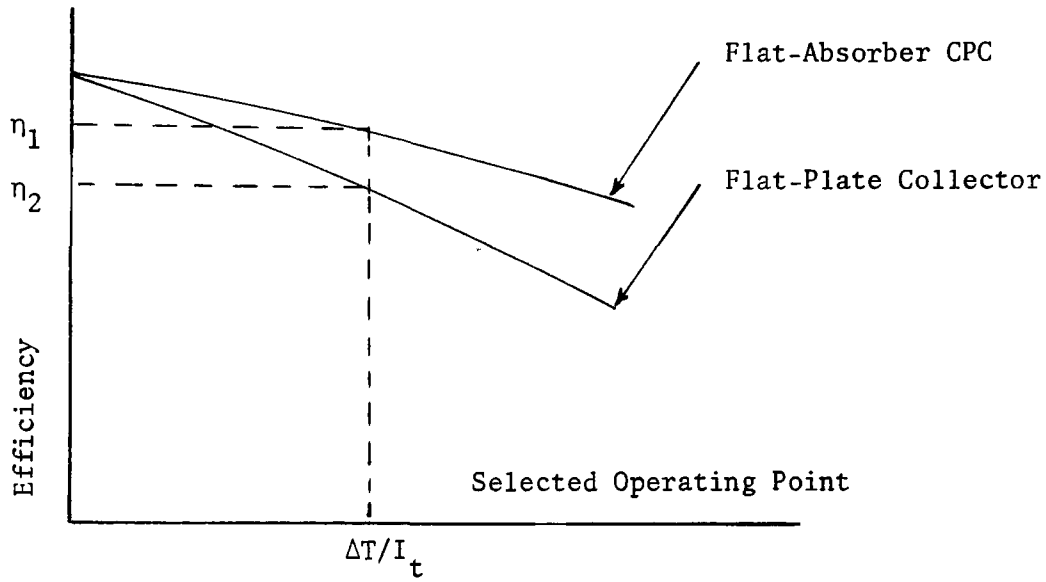


Fig. 4.11 Thermal Efficiency, η , as a function of $\Delta T/I_t$

The useful energy gains, Q_1 and Q_2 , per unit time per unit area for the two collectors are given by

$$Q_1 = \eta_1 I_t \quad (4.7.2)$$

$$Q_2 = \eta_2 I_t \quad (4.7.3)$$

If c_1 and c_2 are the costs per unit area for the two collectors, respectively, then the per unit cost of thermal energy gained by the two collectors may be expressed as

$$PUC_1 = C_1/Q_1 \quad (4.7.4)$$

$$PUC_2 = C_2/Q_2 \quad (4.7.5)$$

In view of (4.7.1), the cost effectiveness becomes

$$C_{eff} = \frac{PUC_1}{PUC_2} \quad (4.7.6)$$

or

$$C_{eff} = \frac{(C_1/Q_1)}{(C_2/Q_2)} \quad (4.7.7)$$

Substitution of (4.74) in d(4.75) results in

$$C_{eff} = \frac{(C_1/\eta_1)}{(C_2/\eta_2)} \quad \begin{array}{l} \text{standard design} \\ \text{Alternate design} \end{array} \quad (4.7.8)$$

Since cost effectiveness C_{eff} is the function of thermal efficiency, it depends, therefore, on the slope of the efficiency curve. Thus, it may vary slightly over the operating range of the two collectors under consideration.

CHAPTER 5

SOLAR COLLECTOR TESTING

5.1 Introduction

Several methods of testing solar collectors have been used to determine the thermal performance characteristics for rating purposes, which can be classified into

1. Instantaneous Procedure
 - a. closed-loop system
 - b. open loop system
2. Calorimetric Procedure
 - a. simultaneous testing
 - b. closed loop system

In the instantaneous method the useful heat collection rate is determined by measuring the mass flow rate of the heat-removal fluid and the temperature difference of the heated fluid across the collector continuously. All collectors are oriented in the same direction, are located under identical weather conditions and insolation, and are tested for the same inlet-fluid temperature and flow rate.

In the closed loop system the heated fluid in the collector is recirculated after rejecting the heat to the ambient by means of a heat exchanger. In the open cycle system the flow is "once-through", introducing new fluid and storing it continuously. The open cycle system does not require a heat

exchanger but it wastes the fluid and requires continuous fluid pretreatment, such as filtering. The system governing equation is given by

$$q_u = G C_p (T_{f,e} - T_{f,i}) \quad (5.1)$$

where G is the fluid mass flow rates Kg/s-m^2 .

The calorimetric procedure employs a closed-loop system in which heated fluid from the collector is mixed with a large volume of the heat-removal fluid stored in an insulated tank. The time rate of change of the temperature in the tank is measured to determine the heat collection rate. The system governing equation is

$$q_u = m C'_p (dT/d\tau) \quad (5.2)$$

where

m = mass of the fluid in the calorimeter per unit
collector area kg/m^2

C'_p = specific heat of the fluid in the calorimeter $\text{kJ/kg-}^\circ\text{C}$

T = average temperature of the fluid in the calorimeter $^\circ\text{C}$

τ = time s.

Figures 5.1 to 5.3 illustrate the conceptual schematic of the two methods.

5.2 Recommended Method

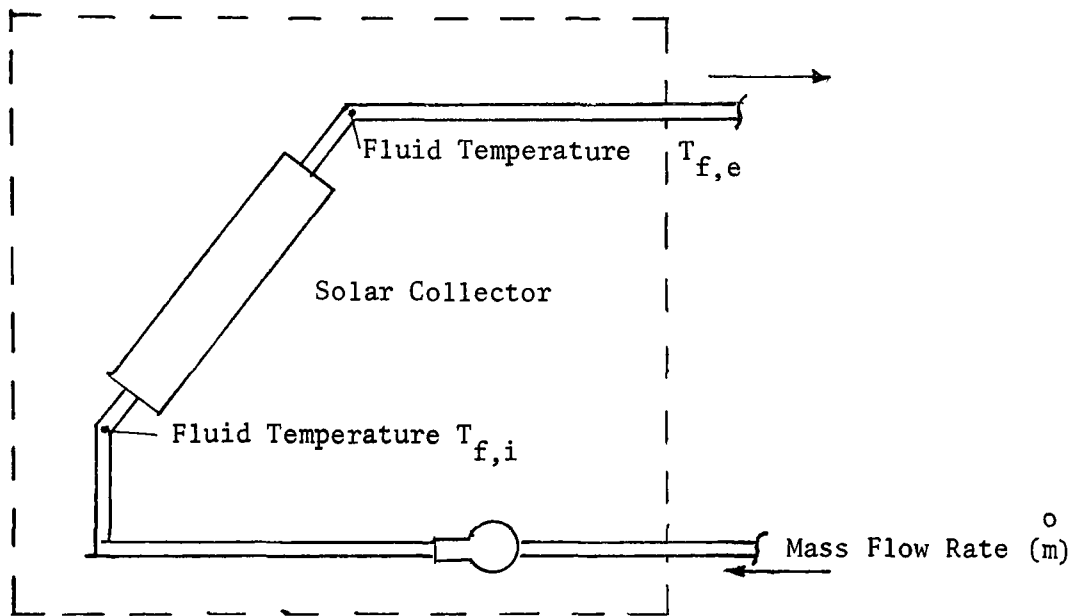


Fig. 5.1 Standard Test Method - Open Loop System

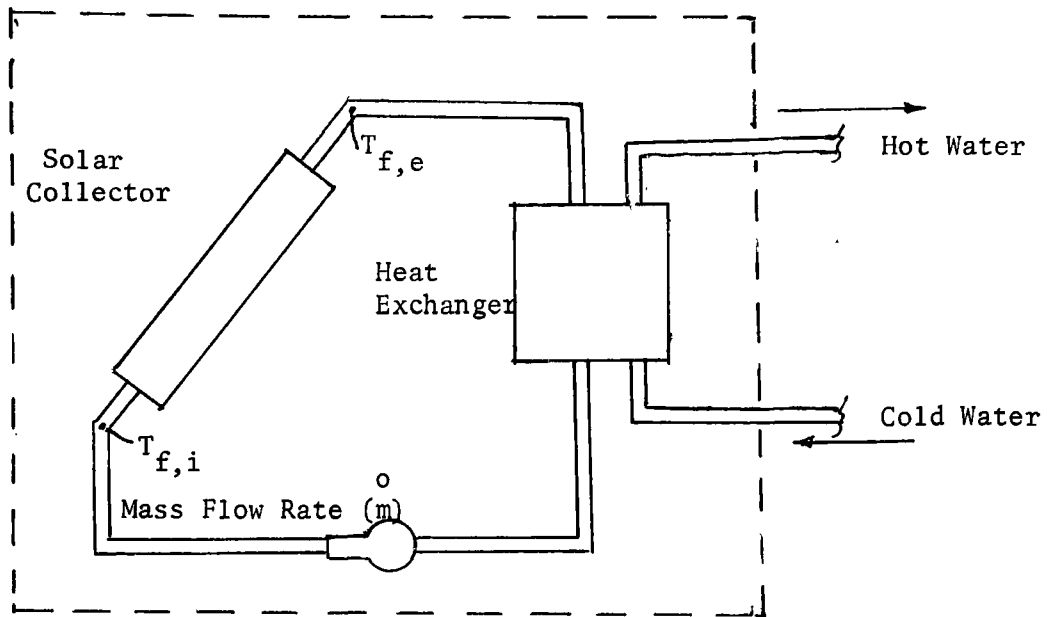


Fig. 5.2 Standard Test Method - Closed Loop System

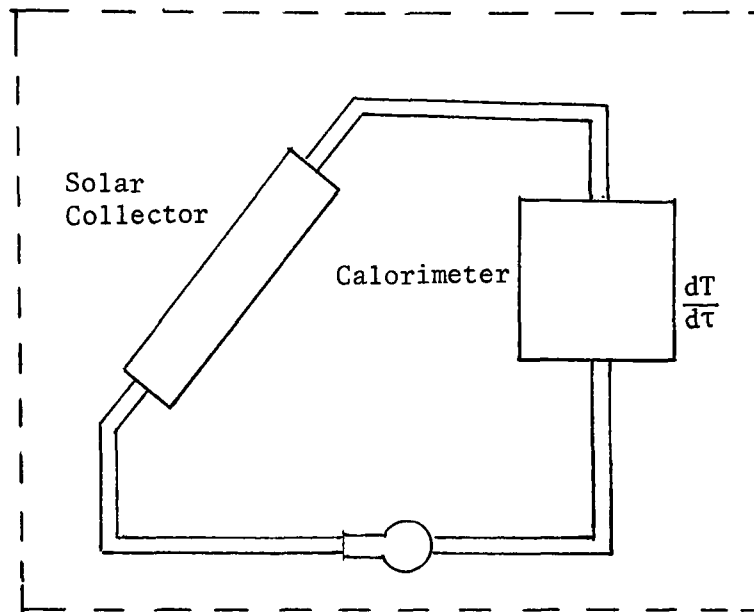


Fig. 5.3 Calorimeter Method - Close Loop System

Any one of the two methods described above can be employed to determine the thermal performance of collectors. However, there are certain advantages and disadvantages to each method. The instantaneous procedure requires costly instrumentation and creates some difficulties in maintaining the steady state conditions. The accuracy requirements in the measurements of ΔT , $\dot{m}$, and I_t are more stringent. The performance test generally takes days of effort to produce enough data for a complete evaluation of the collector.

In the calorimetric method only one variable need to be measured, namely, the storage tank temperature. The method is simple and takes a few hours to evaluate the average performance of the collector. The calorimetric method is limited to liquid-based collectors because the heat capacity of air is small and the collected heat cannot be readily stored while the instantaneous method is suitable for both types of collectors (liquid and air). Another disadvantage with a calorimetric method is that if several collectors are to be tested simultaneously, each collector unit needs an individual circulation loop, calorimeter, and pump etc. Nevertheless, the calorimetric procedure is more appropriate for determining the average efficiency while the instantaneous method is more accurate to determine the "instantaneous efficiency." Moreover, in the instantaneous procedure all the measurements are made around the collector which provides the detailed information on the collector performance, while in the calorimetric method all the

efforts are diverted towards a very careful analysis of the calorimeter itself.

In 1974, Hill and Kasuda [2] reviewed the various methods of testing and recommended the instantaneous procedure as the standard method of testing the solar collectors to determine the thermal performance.

5.3 Standard Test Method

The National Bureau of Standards [2] and AHSRAE [3] have published a testing procedure for solar collectors which outlines the required outdoor test conditions, instrumentation and measurement techniques for flow, temperature difference, and insolation. The test procedure involves determining the solar collector performance by obtaining values for collector instantaneous efficiency for a large combination of values for incident insolation, ambient temperature and inlet fluid temperature.

The outdoor conditions under which tests must be conducted are specified as

- . "The orientation of the collector shall be such that the incident angle (measured from the normal to the collector surface) is less than 30° during the period in which solar data is being taken.
- . The range of ambient temperatures for all reported test points comprising this efficiency curve shall be less than 30°C . (54°F).

- . The test shall be conducted on days having weather conditions such that the 15 minute integrated average insolation measured in the plane of collector surface shall not be less than 630 W/m^2 ($199.8 \text{ Btu/hr-ft.}^2$)"

5.3.1 Instrumentation

The parameters to be measured in the instantaneous procedure (standard method) usually consists of

1. Temperatures:
 - a. fluid inlet and outlet
 - b. ambient
 - c. transparent covers (to evaluate the losses)
 - d. plate temperature
2. Temperature Differences:
 - a. fluid outlet - fluid inlet
3. Insolation:
 - a. total radiation
 - b. direct or diffuse radiation
4. Fluid Flow Rate

To obtain the required accuracy, it is desirable to measure the temperature difference directly, by thermopile or resistance thermometer in the bridge circuit. For insolation, a pyranometer must be oriented in the plane of collector's surface and two precautions should be taken: (i) it does not receive the reflected radiation from the collector covers; (ii) it should not project any shadow on the collector surface. Also, the wind

velocity must be measured in the vicinity of the test stand.

5.4 Experimental Facility

An outdoor testing facility to determine the experimental performance of liquid-based solar collectors was constructed, and instrumented to meet the specifications of National Bureau of Standards and ASHRAE Standard 93-77. The facility is capable of testing two collectors simultaneously, and the major components of the system are shown, schematically, in Figure 5.4. The system consists of the two collectors mounted on a south facing stand tilted at 65° (for winter operation) to the horizontal surface, two fluid flowing circuits served by a single circulating pump and an insulated fluid storage tank. A heat exchanger used to reduce the flowing-fluid temperature by employing city water as coolant, and an electric heater activated by a temperature controller were employed to maintain the inlet fluid temperature constant within the specified limit of $\pm 1^{\circ}\text{C}$.

A 65/35 (by weight) mixture of water and propylene glycol used as heat transfer fluid was stored in the insulated tank which was maintained at atmospheric pressure to suppress any vapour pressure formation.

5.5 Instrumentation and Data-Acquisition System

To obtain the thermal performance of a collector, the following parameters were measured: fluid flow rate, fluid inlet and outlet temperatures, glazing and absorber plate

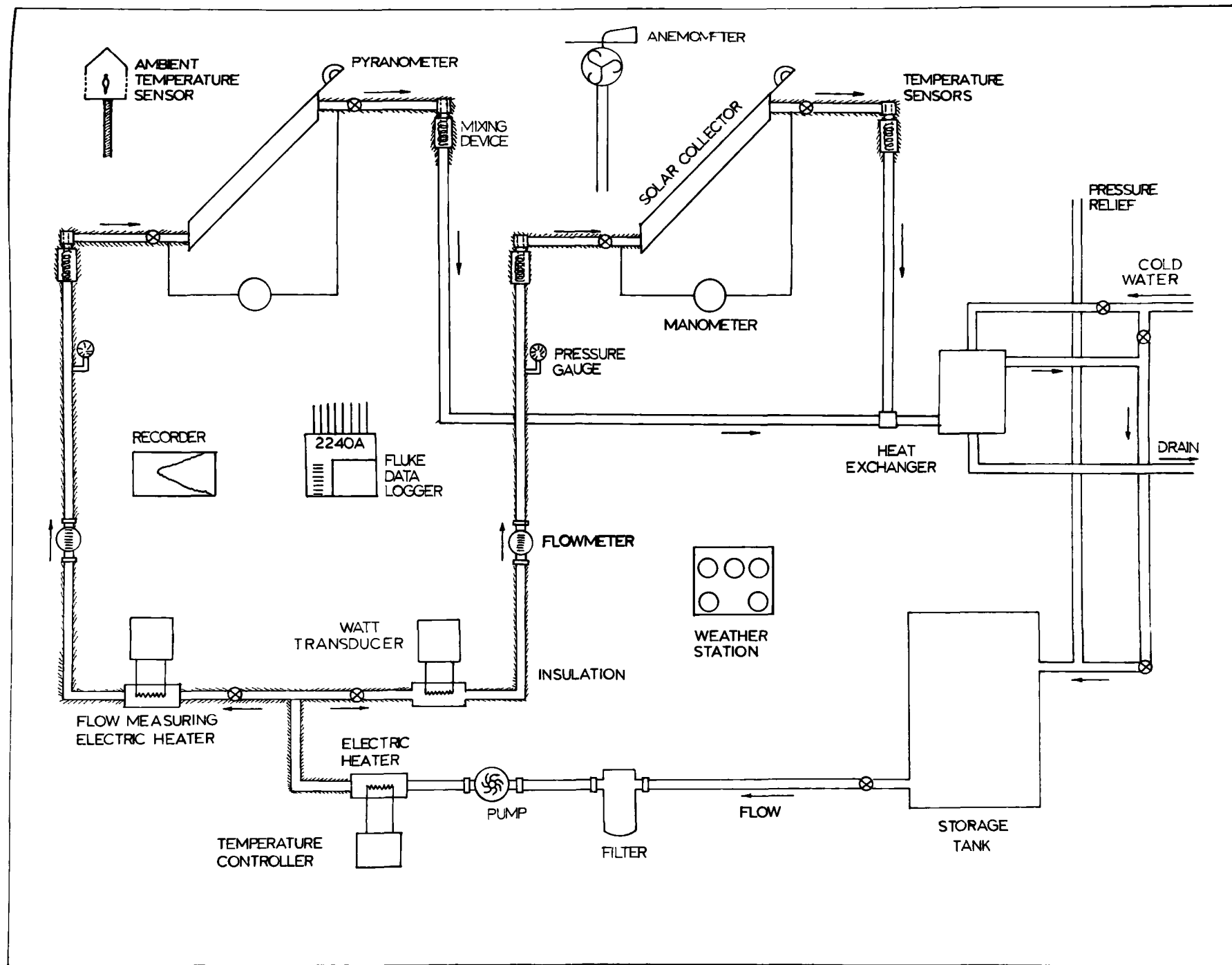


Fig. 5.4 CLOSED-LOOP TESTING CONFIGURATION FOR LIQUID-BASED SOLAR COLLECTOR

temperatures, solar flux, wind speed, and ambient temperature. In addition to the above mentioned parameters, the reflecting surface temperature was measured for the flat-absorber CPC.

Copper-Constantan (type T) 24 gauge teflon fiber glass covered thermocouples were used for all temperature measurements. The error in the absolute temperature measurement is $\pm 0.5^{\circ}\text{C}$. Nine thermocouples in all were used for the flat-absorber CPC. Four thermocouples were attached to measure the plate temperatures. Two were placed at the center of the transparent covers and two thermocouples were employed to measure the inlet and outlet temperatures. One thermocouple measured the reflector surface temperature.

Eight thermocouples were employed to measure the absorber plate, transparent covers, and inlet-outlet temperatures across the conventional flat-plate collector. One thermocouple was mounted in a radiation shielded chamber to measure ambient air temperature. The temperature difference between inlet and outlet was also measured with two thermopiles, one for each, consisting of eight thermocouple junctions in each. The differential error of the thermopile is less than 0.1°C .

The flow rate for each collector was obtained by measuring the input power and temperature rise across a 500 watt heating unit installed in the liquid loop and placed at the inlet to the collector. In case of pump failure, a fail safe device, Surface Aquastat Limit Control, Model LA409, was used to disconnect the 500 watt heater elements. This device was attached to the

surface of the pipe in which the heating unit was immersed. The schematic of flow meter unit is shown in Figure 5.5.

To determine the fluid flow rate, the input power to individual heaters was measured by a calibrated watt-transducer and the temperature difference across the heaters was measured by two thermopiles consisting of four thermocouple junctions in each. Moreover, for visual inspection and rapid adjustment of the flow rate, a glass tube flowmeter, Brooks Model 1307, was also installed in the liquid loop.

An Eppley pyranometer, Eppley Model 50, oriented along the collector's surface measured the total solar radiation incident to the collector surface. The output of the pyranometers was recorded continuously by a strip chart recorder, Linear Instruments Model 355, and printed on a digital printer. Two Rho-Sigma pyranometers, Rho-Sigma Model 1008, were employed to determine the direct and diffuse components of solar radiation. One of these was shaded permanently by a semi-circular black painted strip of 1m radius 10cm wide while the other was placed along the plane of the collectors.

A weather station records speed and direction of the wind as well as humidity and barometric pressure.

A data logger, Fluke Model 2240A, was employed to record the temperature at selected points, the output of the thermopiles, pyranometers, and that of the watt-transducers.

5.6 Test Procedure

As recommended by NBS and ASHRAE Standard 93-77, all the

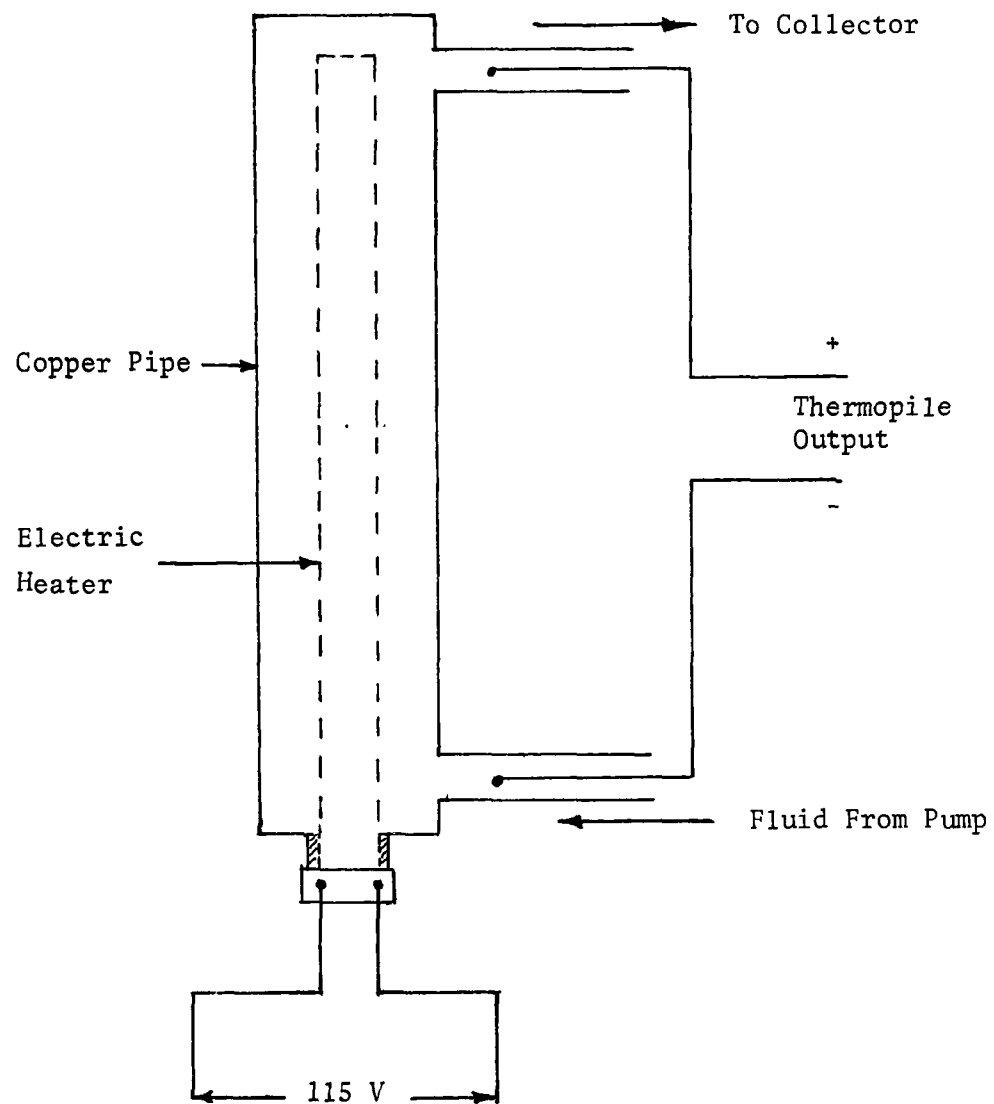


Fig. 5.5 Schematic of Flow-meter

tests were conducted about solar noon on bright sunny days when solar radiation was almost steady. The pyranometer output recorded on a bright sunny day is shown in Figure 5.6. Fluid temperature at the inlet to the collectors was controlled by adjusting the inflow of cold water to the heat exchanger of Figure 5.4, and by an electric heater activated by a precision temperature controller. To reach thermal equilibrium, the system was operated for one-half hour or more before any data was recorded. Two tests were performed before noon and two in the afternoon for a selected inlet temperature. Four days data completed the test results for one particular flow rate.

The collectors were tested for three flow rates corresponding to 10.0, 14.7, and 20.0 lb/hr-sq ft² (or 0.0135, 0.02, and 0.271 kg/s-m²) of collector transparental area. Fluid mass flow rate was kept constant for a 15 minute test period. All test data, except the wind speed, were printed at one minute intervals by the digital printer.

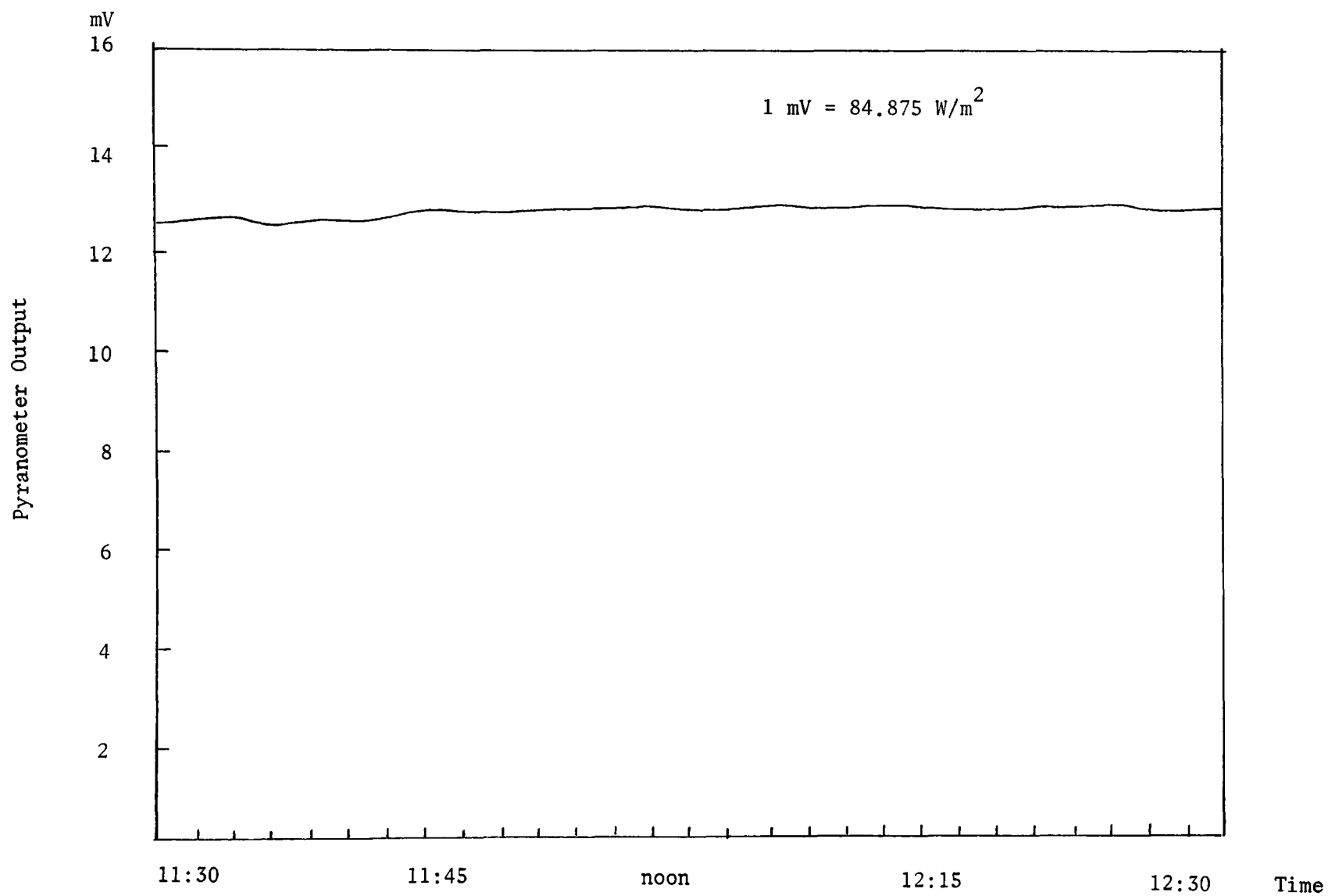


Fig. 5.6 Pyranometer Output on a Cloudless Day
About Solar Noon

CHAPTER 6

EXPERIMENTAL RESULTS AND DISCUSSION

The typical experimental performance data for the two collectors obtained under identical outdoor conditions are presented in Tables 6.1(A,B), 6.2(A,B) and 6.3(A,B) corresponding to three flow rates: 0.02, 0.0271 and 0.0135 kg/s-m², respectively. The thermal performance of the two collectors obtained by using equations (4.5.2)-(4.5.4) and (4.6.13)-(4.6.15) is depicted in Tables 6.4(A,B), 6.5(A,B) and 6.6(A,B) for the three flow rates. These tables include the values of key parameters F_R , F' , U_L and $F_t(F_e)$ computed by using the analytical performance equations. However, as explained in Chapter 4, Section 4.5, of this thesis the experimental values of these collector parameters can be obtained by plotting instantaneous efficiency, η , against temperature difference divided by total insolation; $(T-T_a)/I_t$.

An inspection of equations (4.5.2)-(4.5.4) reveals that

- . $F_R U_L$ and $F_R F_e$ are the negative of the slope and the y axis intercept of the instantaneous efficiency curve when plotted η vs $(T_{f,i} - T_a)/I_t$
- . $F' U_L$ and $F' F_e$ are the negative of the slope and the y axis intercept of the instantaneous efficiency curve when plotted η vs $(T_f - T_a)/I_t$.
- . U_L and F_e are the negative of the slope and the y axis intercept of the instantaneous efficiency curve when plotted η vs $(T_p - T_a)/I_t$.

The instantaneous efficiency for recommended flow rate of 0.02 kg/s-m^2 (Table 6.1) is plotted against $(T_p - T_a)/I_t$, $(T_f - T_a)/I_t$ and $(T_{f,i} - T_a)/I_t$ in Figs. 6.1, 6.2, and 6.3, respectively. Since the test results include the operating range of $\Delta T/I_t$ from 0.017 to $0.051^\circ\text{C m}^2 \text{ W}^{-1}$, a least square fit was used to plot these curves to obtain the respectively intercepts, as shown in Figs. 6.1-6.3. The correlation, r , is calculated for each curve. The significant scattering in the results happened due to the varying nature of outdoor conditions; wind and angle of incidence variation. The experimental values of key parameters which determine the collectors performance are obtained from these curves and are presented in Table 6.7 along with the calculated values. The experimental results are found satisfactorily with the calculated values. The plot of efficiency based on the inlet fluid temperature is presented in Fig. 6.4 for the three flow rates, 0.02, 0.0271 and 0.0135 Kg/s-m^2 , illustrates the fact that the efficiency increases with increase in mass flow rate. However, the increase in efficiency is not directly proportional to the increase in mass flow rate. In fact, as the flow rate is further increased, the efficiency appears to reach an upper bound. This is to be expected. As it appears from these curves, the efficiency of the flat-absorber CPC was higher than that of the conventional type flat-plate collector, under identical outdoor conditions and for the tested temperatures. The efficiency of both the collectors reduces with increase in $\Delta T/I_t$. However, the decrease in efficiency for the flat-plate collector is higher mainly because of the larger absorber area. In addition to this the absorbed energy in the flat-absorber CPC is slightly higher than that of the flat-plate collector.

From Table 6.8 [18] the estimated cost per unit area for each collector is about the same i.e. $\$70/\text{m}^2$. However, the cost effectiveness determined from (4.7.8) is higher for flat-absorber CPC than that for the flat-plate collector under selected operating conditions. As shown in Table 6.9 the cost effectiveness of a collector varies slightly with operating conditions.

TABLE 6.1 A EXPERIMENTAL DATA FOR FLAT-ABSORBER CPC
FOR FLOW RATE = 0.0200 kg/s-m²

SET	IP	IRH1	IRH2	T1	TO	TP1	TG1	TG2	TA1	WV1	TR1	DELTW	MVF	MVC	VL ^m
1.0	12.606	516.0	87.64	13.5	19.7	25.5	22.5	5.4	-6.1	6.0	21.6	8.8	1.357	1.961	3.0114
1.0	12.752	515.2	94.97	14.0	20.2	25.8	22.5	5.5	-6.4	5.0	22.1	9.2	1.555	1.995	3.0491
1.0	12.834	516.6	85.03	14.0	20.5	25.5	22.9	4.3	-4.8	2.0	22.1	7.3	1.450	2.123	3.0834
1.0	12.708	519.8	83.70	14.0	20.4	25.8	21.9	2.9	-5.0	3.0	22.0	9.0	1.457	1.886	3.0983
2.0	11.512	498.6	118.65	19.6	25.8	31.1	27.1	10.4	-1.2	0.0	27.4	10.5	1.630	1.933	3.1360
2.0	12.890	510.2	124.16	19.6	26.0	32.1	28.1	11.1	-1.9	0.0	28.1	9.2	1.479	2.021	3.1237
2.0	12.573	508.0	123.97	19.6	25.9	32.0	28.1	11.5	-1.3	0.0	25.9	9.1	1.488	2.086	3.1207
2.0	11.889	501.9	119.60	19.6	25.6	31.3	27.5	11.0	-1.1	0.0	27.5	9.7	1.566	1.919	3.1215
3.0	12.753	526.0	108.89	21.6	28.6	32.2	24.6	2.6	-9.5	4.0	22.3	8.1	1.356	2.005	2.7856
3.0	13.016	527.5	109.40	21.3	28.8	32.1	23.3	0.4	-9.0	5.0	28.9	11.6	1.526	2.193	3.2077
3.0	13.270	526.8	102.05	20.9	29.4	32.1	24.0	1.4	-8.6	4.0	29.0	9.9	1.696	2.415	3.1838
3.0	13.053	524.9	101.16	21.9	29.7	33.1	24.4	1.4	-9.0	2.0	28.9	10.3	1.746	2.303	3.1732
4.0	12.425	511.7	128.22	37.4	42.6	45.2	33.4	10.0	-7.5	3.0	41.3	10.0	1.455	1.763	3.1377
4.0	12.737	521.1	122.11	37.9	43.9	46.3	33.1	5.0	-8.8	5.0	41.8	9.0	1.460	1.530	3.1430
4.0	12.745	519.5	134.32	37.1	44.2	46.3	33.2	7.2	-6.2	4.5	41.5	9.7	1.545	1.545	3.1271
4.0	12.583	521.8	132.06	38.0	44.3	46.6	33.9	9.8	-5.8	3.0	41.2	8.7	1.444	1.564	3.1095

TABLE 6.1 B EXPERIMENTAL DATA FOR FLAT-PLATE COLLECTOR
FOR FLOW RATE = 0.0200 kg / s-m²

SET	IP	IRH1	IRH2	T1	TO	TP1	TG1	TG2	TA1	WV1	DELTW	MVP	MVC	VLT
1.0	12.606	516.0	87.64	15.9	21.8	25.0	20.6	3.4	-6.1	6.0	9.0	1.401	1.751	3.4743
1.0	12.752	515.2	94.97	16.1	22.4	25.5	20.2	2.3	-6.4	5.0	9.2	1.594	1.744	3.5162
1.0	12.834	516.6	85.03	14.8	22.3	25.8	20.7	2.0	-4.8	2.0	9.6	1.481	1.933	3.5764
1.0	12.708	519.8	83.70	16.4	23.3	26.6	18.0	0.0	-5.0	3.0	10.4	1.620	1.875	3.5335
2.0	11.512	498.6	118.65	21.1	25.2	29.1	25.6	9.0	-1.2	0.0	9.0	1.375	1.991	3.5794
2.0	12.890	510.2	124.16	21.4	25.1	28.9	25.4	7.6	-1.9	0.0	8.9	1.496	1.844	3.5526
2.0	12.573	508.0	123.97	21.4	26.1	29.5	27.5	9.0	-1.3	0.0	9.0	1.513	1.840	3.5358
2.0	11.889	501.9	119.60	21.2	27.0	29.3	26.0	9.0	-1.0	0.0	10.1	1.604	1.887	3.5446
3.0	12.753	526.0	108.89	20.9	25.5	29.6	19.7	-0.1	-9.5	4.0	8.1	1.458	1.514	3.5521
3.0	13.016	527.5	109.40	21.3	26.0	30.1	20.0	-1.1	-9.0	5.0	10.1	1.359	1.592	3.5730
3.0	13.270	526.8	102.05	21.0	26.7	29.6	20.1	-0.1	-8.6	4.0	10.8	1.775	1.847	3.5542
3.0	13.053	524.9	101.16	22.0	27.1	31.1	22.1	0.7	-9.0	2.0	9.6	1.608	1.647	3.5245
4.0	12.425	511.7	128.22	37.3	40.9	43.4	29.5	7.0	-7.5	3.0	8.5	1.492	0.544	3.5747
4.0	12.737	521.1	122.11	38.1	41.5	44.1	27.8	2.6	-8.8	5.0	8.5	1.424	0.626	3.5775
4.0	12.745	519.5	134.32	37.5	41.0	45.0	27.9	5.0	-6.2	4.5	8.5	1.484	1.173	3.5805
4.0	12.583	521.8	132.06	38.1	41.8	45.0	29.7	-7.1	-5.8	3.0	9.7	1.380	0.919	3.5783

TABLE 6.2 A EXPERIMENTAL DATA FOR FLAT-ABSORBER CPC
FOR FLOW RATE = 0.0271 kg/s-m^2

SET	IP	IRH1	IRH2	T1	TO	TP1	TG1	TG2	TA1	WV1	TR1	DELTW	MVP	MVC	VL"
1.0	13.482	529.0	103.00	16.0	21.9	26.8	22.2	12.3	-9.1	0.0	26.5	7.2	1.056	1.772	2.9040
1.0	13.476	527.9	104.55	16.0	22.1	26.5	23.2	2.8	-9.9	7.5	26.3	7.5	1.102	1.832	2.9540
1.0	12.892	527.5	130.96	16.1	22.1	26.5	22.5	0.6	-9.1	12.0	25.5	6.9	0.991	1.792	2.8680
1.0	12.604	527.4	112.67	16.4	22.3	26.6	22.9	1.7	-8.5	7.0	25.7	6.8	0.958	1.807	2.8720
2.0	13.024	514.3	95.48	25.7	32.2	35.5	30.6	11.0	-2.8	0.0	33.0	5.9	1.115	1.925	2.8720
2.0	13.141	513.4	98.47	25.2	30.0	34.3	29.9	11.1	-3.3	0.0	32.2	6.1	1.158	1.292	2.8830
2.0	13.188	520.8	97.80	25.5	30.4	33.9	28.8	7.6	-3.9	0.0	31.5	6.3	1.808	1.400	2.8840
2.0	12.829	510.2	95.07	25.7	32.1	35.7	31.0	13.2	-1.2	0.0	33.1	6.4	1.190	1.860	2.8750
3.0	13.123	517.1	95.42	29.4	35.9	39.3	31.7	9.9	-4.4	10.0	36.6	6.4	0.962	1.813	2.9130
3.0	13.396	518.6	94.27	29.6	35.9	39.2	31.4	7.9	-5.0	10.0	37.6	6.8	1.022	1.749	2.9510
3.0	13.374	517.0	91.09	29.9	34.6	38.6	30.9	8.6	-5.6	5.0	38.1	6.8	1.102	1.214	2.9150
3.0	13.372	517.2	91.12	29.9	35.3	39.2	31.4	7.9	-5.0	10.0	37.6	5.9	1.070	1.436	2.9050
4.0	12.690	524.0	88.00	36.2	41.2	44.0	32.4	4.7	-6.6	10.0	37.7	5.4	1.149	1.208	2.7950
4.0	12.747	524.2	75.14	36.2	41.1	44.2	32.0	5.5	-6.7	7.5	38.8	6.2	1.176	1.263	2.8550
4.0	12.896	524.2	93.01	35.9	41.2	44.0	32.8	5.8	-6.4	6.0	8.4	5.7	1.167	1.331	2.8090
4.0	12.684	524.7	74.86	36.2	41.5	44.3	32.5	4.7	-6.3	10.0	38.7	6.3	1.158	1.344	2.8330

TABLE 6.2 B EXPERIMENTAL DATA FOR FLAT-PLATE COLLECTOR
FOR FLOW RATE = 0.0271 kg/s-m^2

SET	IP	IRH1	IRH2	T1	TC	TP1	TG1	TG2	TA1	WV1	DELTH	MVP	MVC	VL1
1.0	13.482	529.0	103.00	15.2	19.4	29.7	21.4	-1.6	-9.1	0.0	7.0	1.094	1.410	3.6150
1.0	13.476	527.9	104.55	15.4	19.9	30.3	23.1	-2.2	-9.9	7.5	9.3	1.427	1.512	3.6880
1.0	12.892	527.5	130.96	15.7	20.1	30.6	19.4	-2.2	-8.3	7.0	9.4	1.528	1.466	3.5750
1.0	12.604	527.4	112.67	15.6	20.0	30.3	21.3	-2.4	-9.1	12.0	6.8	0.881	1.486	3.5680
2.0	13.188	520.8	97.80	25.5	28.8	33.1	30.7	3.7	-3.9	0.0	6.5	1.031	1.163	3.5240
2.0	13.024	514.3	95.48	26.3	31.6	35.0	7.1	-2.8	0.0	0.0	7.1	1.182	1.856	3.5200
2.0	13.141	513.4	98.47	25.6	29.2	33.6	30.6	7.7	-3.0	0.0	6.6	1.062	1.250	3.5410
2.0	12.829	510.2	95.07	26.6	31.3	35.1	34.4	9.1	2.2	0.0	7.7	1.226	1.651	3.5260
3.0	13.374	517.0	91.09	29.9	32.4	40.6	29.2	4.7	-5.6	5.0	6.9	1.110	0.863	3.4820
3.0	13.372	517.2	91.12	30.0	32.9	41.7	29.0	4.7	-5.2	6.0	6.2	1.021	1.034	3.4510
3.0	13.396	518.6	94.27	29.9	33.5	41.2	28.9	3.5	-5.0	10.0	7.2	1.151	1.242	3.5110
3.0	13.123	517.1	95.42	29.4	33.2	42.3	29.7	5.2	-4.4	10.0	6.8	1.093	1.341	3.4720
4.0	12.684	524.7	74.86	37.0	39.3	44.6	29.3	0.2	-6.3	10.0	6.7	1.082	0.638	3.4520
4.0	12.690	524.0	88.00	37.0	38.4	44.4	31.3	1.2	-6.6	5.0	7.0	1.088	0.926	3.5140
4.0	12.896	524.2	93.01	36.6	39.1	44.5	29.8	1.0	-6.4	6.0	6.6	1.078	0.932	3.4990
4.0	12.747	524.2	75.14	37.1	38.8	44.5	29.5	2.2	-6.7	7.5	6.7	1.116	1.263	3.5460

TABLE 6.3A EXPERIMENTAL DATA FOR FLAT-ABSORBER CPC
FOR FLOW RATE = 0.0135 kg/s-m^2

C	SET	IP	IRH1	IRH2	T1	TO	TP1	TG1	TG2	TA1	WV1	TP1	DELTW	MVF	MVC	VLT
	1.0	12.020	501.8	72.77	17.5	31.7	36.0	27.5	7.3	-2.0	8.0	34.6	4.4	0.701	4.330	0.7420
	1.0	12.457	509.4	74.45	17.2	33.5	36.0	27.8	6.3	-3.7	5.0	34.7	4.3	0.746	5.006	0.7340
	1.0	13.053	516.3	79.42	17.1	32.4	36.5	27.5	7.3	-4.1	5.0	34.3	4.4	0.629	4.676	0.7300
	1.0	12.754	512.6	76.84	17.5	33.5	36.2	27.5	5.8	-3.8	7.0	34.3	4.2	0.711	4.926	0.7500
	2.0	12.288	517.5	84.17	22.6	37.7	37.9	30.9	7.9	-3.9	7.0	36.5	4.3	0.781	4.179	0.7410
	2.0	12.958	518.6	83.27	22.2	38.2	38.3	31.0	10.4	-3.4	1.0	36.6	4.3	0.775	4.528	0.7520
	2.0	13.150	517.3	81.18	22.6	38.9	39.2	31.5	9.1	-2.9	6.0	36.7	4.3	0.789	4.701	0.7520
	2.0	12.833	517.3	85.72	22.3	38.9	39.3	31.0	10.4	-3.4	1.0	36.6	3.9	0.651	4.804	0.7450
	3.0	12.819	512.9	72.91	30.2	46.8	47.5	36.3	11.4	-2.5	10.0	40.1	4.8	0.684	4.589	0.7550
	3.0	13.048	515.6	74.45	30.1	44.2	45.5	34.5	11.0	-1.9	5.0	38.7	3.8	0.624	3.961	0.7450
	3.0	12.634	514.3	71.46	30.4	47.5	48.8	36.2	10.6	-1.7	5.0	41.2	4.1	0.630	4.776	0.7580
	3.0	12.282	508.9	6.93	30.1	44.2	44.2	34.9	13.4	-1.2	6.3	41.2	4.7	0.731	3.770	0.7480
	4.0	12.292	519.9	90.73	37.1	47.4	48.4	35.6	8.5	-6.3	10.0	37.0	15.6	2.778	2.998	2.8800
	4.0	12.452	519.0	91.32	37.9	48.6	49.4	36.5	9.8	-6.0	7.0	37.6	13.8	2.275	3.061	2.8700
	4.0	12.635	514.9	88.12	37.0	48.8	49.1	39.0	12.2	-4.4	0.0	38.8	14.0	2.765	3.481	2.9130
	4.0	12.520	519.8	89.87	36.6	48.7	49.1	36.3	7.8	-6.3	10.0	38.0	13.6	2.164	3.629	2.8930

TABLE 6.3B EXPERIMENTAL DATA FOR FLAT-PALTE COLLECTOR
FOR FLOW RATE = 0.0135 kg/s-m^2

SET	IP	IRH1	IRH2	T1	TO	TP1	TG1	TG2	TA1	WV1	DEITW	MVF	MVC	VLT
1.0	12.457	509.4	74.45	16.4	27.2	32.6	23.8	5.8	-3.7	5.0	3.4	0.568	3.386	0.8090
1.0	12.020	501.8	72.77	17.0	25.5	32.4	23.5	7.7	-2.0	8.0	3.3	0.559	2.872	0.8890
1.0	13.053	516.3	79.42	16.9	26.9	34.2	23.5	5.3	-4.1	5.0	2.8	0.512	3.168	0.8880
1.0	12.754	512.6	76.84	16.8	26.9	34.1	23.4	4.7	-3.8	7.0	3.4	0.562	3.584	0.8920
2.0	12.958	518.6	83.27	23.0	32.0	38.1	25.6	6.3	-3.4	7.0	3.2	0.521	3.025	0.8950
2.0	12.288	517.5	84.17	23.6	30.9	36.7	27.9	6.7	-3.3	7.0	3.1	0.533	2.461	0.8800
2.0	12.833	517.3	85.72	23.0	32.4	37.8	27.0	8.0	-3.4	1.0	3.9	0.666	3.122	0.8910
2.0	13.150	517.3	81.18	23.4	32.5	37.9	25.7	7.8	-2.9	6.0	3.6	0.581	3.077	0.8960
3.0	12.819	512.9	72.91	31.5	39.6	44.2	31.0	9.3	-2.5	10.0	3.2	0.521	2.805	0.9030
3.0	13.048	515.6	74.45	30.7	37.1	42.9	30.5	8.9	-1.9	5.0	3.1	0.576	2.268	0.8990
3.0	12.634	514.3	71.46	31.9	40.5	40.5	31.6	8.8	-1.7	5.0	3.6	0.593	3.327	0.8990
3.0	12.282	508.9	6.93	31.1	36.9	45.2	31.5	11.5	-1.2	6.0	3.3	0.544	2.021	0.8960
4.0	12.292	519.9	90.73	35.2	37.7	44.9	28.6	6.0	-6.3	10.0	13.5	2.247	0.984	3.4460
4.0	12.452	519.0	91.32	36.2	38.6	46.1	28.0	6.5	-6.0	7.0	14.3	2.324	0.859	3.4530
4.0	12.635	514.9	88.12	35.6	40.1	46.8	29.7	8.5	-4.4	0.0	13.0	2.140	1.581	3.5080
4.0	12.520	519.8	89.87	35.6	39.7	46.1	28.2	5.1	-6.3	10.0	11.9	1.950	1.459	3.4790

TABLE 6.4A THERMAL PERFORMANCE OF FLAT-ABSORBER CPC
FOR FLOW RATE = 0.0200 $\text{kg} / \text{s-m}^2$

SET	IT	FLOW	FR	FP	FT	UL	IRTI	EFTI	IFTF	EFTF	IPTP	EFPP	EFMEFS
1.0	1069.93	0.0117	0.9202	0.9417	0.6911	4.2	0.018	0.5786	0.021	0.5828	0.030	0.5906	0.4777*
1.0	1082.33	0.0108	0.9182	0.9411	0.6885	4.2	0.019	0.5738	0.022	0.5790	0.030	0.5881	0.4410
1.0	1089.29	0.0126	0.9271	0.9459	0.6921	3.9	0.017	0.5904	0.020	0.5933	0.028	0.6030	0.5423
1.0	1078.59	0.0115	0.9217	0.9430	0.6927	4.1	0.018	0.5848	0.021	0.5890	0.029	0.5983	0.4648
2.0	977.08	0.0101	0.9220	0.9447	0.6787	3.8	0.021	0.5646	0.024	0.5692	0.033	0.5757	0.4520
2.0	1094.04	0.0113	0.9253	0.9458	0.6777	3.9	0.020	0.5702	0.023	0.5742	0.031	0.5805	0.4686
2.0	1067.13	0.0113	0.9254	0.9459	0.6776	3.8	0.020	0.5704	0.023	0.5743	0.031	0.5801	0.4848
2.0	1009.08	0.0107	0.9239	0.9454	0.6787	3.8	0.021	0.5680	0.023	0.5724	0.032	0.5787	0.4533
3.0	1082.41	0.0112	0.9166	0.9394	0.6844	4.3	0.029	0.5347	0.032	0.5374	0.039	0.5490	0.4913
3.0	1104.73	0.0101	0.9114	0.9370	0.6843	4.4	0.027	0.5343	0.031	0.5380	0.037	0.5513	0.4685
3.0	1126.29	0.0104	0.9137	0.9383	0.6868	4.3	0.026	0.5433	0.030	0.5455	0.036	0.5596	0.5281
3.0	1107.87	0.0100	0.9152	0.9399	0.6870	4.2	0.028	0.5421	0.031	0.5455	0.038	0.5581	0.4829
4.0	1054.57	0.0109	0.9129	0.9370	0.6764	4.5	0.043	0.4759	0.045	0.4801	0.050	0.4944	0.3963
4.0	1081.05	0.0115	0.9114	0.9352	0.6794	4.7	0.043	0.4694	0.046	0.4717	0.051	0.4855	0.4130
4.0	1081.73	0.0107	0.9096	0.9347	0.6750	4.6	0.040	0.4767	0.043	0.4783	0.049	0.4920	0.4263
4.0	1067.98	0.0116	0.9148	0.9375	0.6760	4.5	0.041	0.4809	0.044	0.4827	0.049	0.4962	0.4385

* For computation refer to Appendix C

TABLE 6.4B THERMAL PERFORMANCE OF FLATE -PLATE COLLECTOR
FOR FLOW RATE = 0.0200 kg/s-m²

SET	IT	FLOW	FR	FP	FT	UL	IRTI	EFTI	IFTP	EFTF	JFTP	EFTP	EFMPAS
1.0	1069.93	0.0131	0.8817	0.9114	0.6592	6.1	0.021	0.4707	0.023	0.4712	0.029	0.4820	0.4321
1.0	1082.33	0.0123	0.8792	0.9105	0.6592	6.1	0.021	0.4686	0.024	0.4692	0.029	0.4802	0.4155
1.0	1089.29	0.0127	0.8837	0.9134	0.6592	5.9	0.018	0.4888	0.021	0.4867	0.028	0.4935	0.4909
1.0	1078.59	0.0115	0.8776	0.9105	0.6592	6.0	0.020	0.4743	0.023	0.4747	0.029	0.4839	0.4237
2.0	977.08	0.0136	0.8897	0.9169	0.6592	5.7	0.023	0.4701	0.025	0.4734	0.031	0.4814	0.4433
2.0	1094.04	0.0130	0.8879	0.9162	0.6592	5.7	0.021	0.4769	0.023	0.4832	0.028	0.4978	0.3482
2.0	1067.13	0.0128	0.8875	0.9161	0.6592	5.7	0.021	0.4772	0.023	0.4811	0.029	0.4944	0.3944
2.0	1009.08	0.0117	0.8831	0.9142	0.6592	5.7	0.022	0.4707	0.025	0.4723	0.030	0.4870	0.4232
3.0	1082.41	0.0138	0.8843	0.9125	0.6592	6.1	0.028	0.4317	0.030	0.4336	0.036	0.4391	0.3699
3.0	1104.73	0.0128	0.8797	0.9102	0.6592	6.2	0.027	0.4313	0.030	0.4343	0.035	0.4412	0.3443
3.0	1126.29	0.0108	0.8725	0.9078	0.6592	6.1	0.026	0.4354	0.029	0.4390	0.034	0.4525	0.3440
3.0	1107.87	0.0120	0.8797	0.9113	0.6592	6.0	0.028	0.4327	0.030	0.4357	0.036	0.4427	0.3447
4.0	1054.57	0.0133	0.8806	0.9100	0.6592	6.2	0.042	0.3471	0.044	0.3491	0.048	0.3581	0.2118
4.0	1081.05	0.0136	0.8795	0.9087	0.6592	6.4	0.043	0.3367	0.045	0.3388	0.049	0.3475	0.2133
4.0	1081.73	0.0133	0.8788	0.9086	0.6592	6.3	0.040	0.3540	0.042	0.3567	0.047	0.3588	0.2768
4.0	1067.98	0.0129	0.8787	0.9091	0.6592	6.3	0.041	0.3530	0.043	0.3554	0.048	0.3613	0.2500

TABLE 6.5 A THERMAL PERFORMANCE OF FLAT-ABSORBER CPC
FOR FLOW RATE = 0.0271 kg/s-m^2

SET	IT	FLOW	FR	FP	FT	UL	IRTI	FPTI	IFTF	PFTF	IPTP	PFTF	EFFVAC
1.0	1144.28	0.0141	0.9344	0.9504	0.6866	3.7	0.022	0.5803	0.025	0.5828	0.031	0.5928	0.5004
1.0	1143.78	0.0137	0.9228	0.9417	0.6860	4.3	0.023	0.5593	0.025	0.5619	0.032	0.5738	0.5050
1.0	1094.21	0.0146	0.9220	0.9404	0.6769	4.5	0.023	0.5464	0.026	0.5478	0.033	0.5577	0.5524
1.0	1069.76	0.0150	0.9254	0.9428	0.6832	4.3	0.023	0.5563	0.026	0.5575	0.033	0.5675	0.5763
2.0	1105.41	0.0149	0.9318	0.9478	0.6883	3.9	0.026	0.5645	0.029	0.5652	0.035	0.5774	0.6035
2.0	1115.34	0.0145	0.9315	0.9478	0.6871	3.9	0.026	0.5645	0.028	0.5679	0.034	0.5801	0.4097
2.0	1119.33	0.0110	0.9233	0.9446	0.6879	3.9	0.026	0.5579	0.028	0.5642	0.034	0.5803	0.3264
2.0	1088.86	0.0139	0.9300	0.9471	0.6881	3.9	0.025	0.5666	0.028	0.5681	0.034	0.5799	0.5566
3.0	1113.81	0.0156	0.9200	0.9380	0.6884	4.8	0.030	0.5252	0.033	0.5249	0.039	0.5365	0.6107
3.0	1136.99	0.0148	0.9183	0.9372	0.6890	4.8	0.030	0.5240	0.033	0.5247	0.039	0.5378	0.5518
3.0	1135.12	0.0141	0.9207	0.9396	0.6900	4.5	0.031	0.5289	0.033	0.5327	0.039	0.5462	0.3805
3.0	1134.95	0.0153	0.9193	0.9376	0.6900	4.8	0.031	0.5243	0.033	0.5261	0.039	0.5385	0.4807
4.0	1077.06	0.0148	0.9164	0.9358	0.6915	4.9	0.040	0.4886	0.042	0.4903	0.047	0.5043	0.4344
4.0	1081.90	0.0140	0.9164	0.9364	0.6960	4.8	0.040	0.4964	0.042	0.4990	0.047	0.5129	0.4124
4.0	1094.55	0.0144	0.9185	0.9377	0.6898	4.7	0.039	0.4976	0.041	0.4993	0.046	0.5134	0.4478
4.0	1076.55	0.0139	0.9143	0.9349	0.6961	4.9	0.039	0.4925	0.042	0.4944	0.047	0.5086	0.4419

TABLE 6.5B THERMAL PERFORMANCE OF FLAT-PLATE COLLECTOR
FOR FLOW RATE = 0.0271 kg/s-m^2

SET	IT	FLOW	FR	FP	FT	UL	IRTI	EPFI	IRTP	FPFP	IRTP	FPFP	PFMEFS
1.0	1144.28	0.0175	0.8995	0.9210	0.6592	5.7	0.021	0.4832	0.023	0.4851	0.034	0.4644	0.4076
1.0	1143.78	0.0135	0.8802	0.9096	0.6592	6.3	0.022	0.4579	0.024	0.4620	0.035	0.4384	0.3386
1.0	1094.21	0.0126	0.8774	0.9086	0.6592	6.3	0.022	0.4581	0.024	0.4630	0.036	0.4370	0.3214
1.0	1069.76	0.0193	0.8933	0.9146	0.6592	6.4	0.023	0.4569	0.025	0.4558	0.037	0.4236	0.5070
2.0	1119.33	0.0182	0.9006	0.9213	0.6592	5.8	0.026	0.4571	0.028	0.4598	0.033	0.4683	0.3497
2.0	1105.41	0.0162	0.8983	0.9210	0.6592	5.7	0.024	0.4714	0.026	0.4708	0.032	0.4802	0.5063
2.0	1115.34	0.0178	0.8996	0.9208	0.6592	5.8	0.026	0.4593	0.027	0.4615	0.033	0.4690	0.3714
2.0	1088.86	0.0153	0.8952	0.9193	0.6592	5.7	0.022	0.4756	0.025	0.4772	0.030	0.4868	0.4309
3.0	1135.12	0.0167	0.8887	0.9128	0.6592	6.3	0.031	0.4096	0.032	0.4143	0.041	0.4011	0.2390
3.0	1134.95	0.0182	0.8912	0.9136	0.6592	6.4	0.031	0.4104	0.032	0.4133	0.041	0.3945	0.3073
3.0	1136.99	0.0162	0.8842	0.9097	0.6592	6.5	0.031	0.4052	0.032	0.4074	0.041	0.3931	0.3397
3.0	1113.81	0.0169	0.8857	0.9103	0.6592	6.6	0.030	0.4074	0.032	0.4085	0.042	0.3839	0.3789
4.0	1076.55	0.0170	0.8855	0.9100	0.6592	6.6	0.040	0.3488	0.041	0.3520	0.047	0.3473	0.2128
4.0	1077.06	0.0168	0.8882	0.9123	0.6592	6.4	0.040	0.3553	0.041	0.3611	0.047	0.3560	0.2086
4.0	1094.55	0.0174	0.8888	0.9123	0.6592	6.4	0.039	0.3608	0.040	0.3636	0.047	0.3594	0.2189
4.0	1081.90	0.0171	0.8872	0.9113	0.6592	6.5	0.040	0.3511	0.041	0.3559	0.047	0.3512	0.2772

TABLE 6.6A THERMAL PERFORMANCE OF FLAT-ABSORBER CPC
FOR FLOW RATE = 0.0135 kg/s-m^2

SET	IT	FLOW	FR	FP	FT	UL	IRTI	FPTI	IFTF	EFTF	IRTP	ERTP	EFFACS
1.0	1020.20	0.0056	0.8790	0.9252	0.6956	4.6	0.019	0.5485	0.026	0.5531	0.037	0.5554	0.5470
1.0	1057.29	0.0055	0.8809	0.9269	0.6954	4.5	0.020	0.5492	0.027	0.5519	0.038	0.5588	0.5440
1.0	1107.87	0.0058	0.8845	0.9280	0.6940	4.5	0.019	0.5524	0.026	0.5563	0.037	0.5610	0.5625
1.0	1082.50	0.0058	0.8813	0.9261	0.6947	4.6	0.020	0.5477	0.027	0.5500	0.037	0.5570	0.5497
2.0	1042.94	0.0054	0.8760	0.9242	0.6924	4.6	0.025	0.5227	0.033	0.5263	0.040	0.5415	0.5207
2.0	1099.81	0.0055	0.8886	0.9319	0.6928	4.2	0.023	0.5459	0.031	0.5447	0.038	0.5650	0.5385
2.0	1116.11	0.0054	0.8773	0.9249	0.6935	4.6	0.023	0.5335	0.030	0.5372	0.038	0.5525	0.5402
2.0	1089.20	0.0062	0.8951	0.9336	0.6919	4.2	0.024	0.5479	0.031	0.5475	0.039	0.5594	0.6472
3.0	1088.01	0.0056	0.8699	0.9197	0.6962	5.0	0.030	0.5001	0.038	0.5004	0.046	0.5107	0.5633
3.0	1107.45	0.0065	0.8857	0.9267	0.6958	4.7	0.029	0.5193	0.035	0.5210	0.043	0.5336	0.5514
3.0	1072.31	0.0063	0.8828	0.9253	0.6968	4.7	0.030	0.5135	0.038	0.5098	0.047	0.5157	0.6713
3.0	1042.43	0.0054	0.8740	0.9230	0.7196	4.7	0.030	0.5285	0.037	0.5342	0.044	0.5524	0.4774
4.0	1043.28	0.0058	0.8727	0.9203	0.6903	5.0	0.042	0.4553	0.047	0.4617	0.052	0.4774	0.3432
4.0	1056.86	0.0068	0.8844	0.9247	0.6900	4.9	0.042	0.4649	0.047	0.4676	0.052	0.4826	0.4684
4.0	1072.40	0.0062	0.8947	0.9334	0.6909	4.2	0.039	0.5009	0.044	0.5051	0.050	0.5216	0.4707
4.0	1062.64	0.0071	0.8837	0.9234	0.6905	5.0	0.040	0.4650	0.046	0.4645	0.052	0.4783	0.5604

TABLE 6.6B THERMAL PERFORMANCE OF FLAT - PLATE COLLECTOR
FOR FLOW RATE = 0.0135 kg/s-m^2

SET	IT	FLOW	FR	PP	PT	UL	IRTI	EPTI	IFTP	PERTP	IRTP	EFTP	EFFTPS
1.0	1057.29	0.0086	0.8576	0.9017	0.6592	6.2	0.019	0.4643	0.024	0.4596	0.034	0.4464	0.5437
1.0	1020.20	0.0087	0.8563	0.9004	0.6592	6.3	0.019	0.4640	0.023	0.4643	0.034	0.4467	0.4655
1.0	1107.87	0.0099	0.8653	0.9043	0.6592	6.2	0.019	0.4684	0.023	0.4641	0.035	0.4442	0.5532
1.0	1082.50	0.0086	0.8553	0.9001	0.6592	6.3	0.019	0.4612	0.024	0.4588	0.035	0.4384	0.5267
2.0	1099.81	0.0093	0.8585	0.9007	0.6592	6.4	0.024	0.4346	0.028	0.4325	0.038	0.4188	0.4828
2.0	1042.94	0.0092	0.8577	0.9005	0.6592	6.4	0.026	0.4245	0.029	0.4256	0.038	0.4149	0.4085
2.0	1089.20	0.0074	0.8519	0.9014	0.6592	6.0	0.024	0.4381	0.029	0.4403	0.038	0.4330	0.4028
2.0	1116.11	0.0083	0.8524	0.8990	0.6592	6.3	0.024	0.4349	0.028	0.4355	0.037	0.4281	0.4312
3.0	1088.01	0.0093	0.8548	0.8978	0.6592	6.6	0.031	0.3874	0.035	0.3849	0.043	0.3763	0.4498
3.0	1107.45	0.0089	0.8565	0.9000	0.6592	6.4	0.029	0.4039	0.032	0.4078	0.040	0.4014	0.3346
3.0	1072.31	0.0081	0.8514	0.8984	0.6592	6.4	0.031	0.3918	0.035	0.3905	0.039	0.4092	0.4516
3.0	1042.43	0.0089	0.8547	0.8988	0.6592	6.5	0.031	0.3925	0.034	0.3966	0.045	0.3719	0.3225
4.0	1043.28	0.0083	0.8479	0.8955	0.6592	6.6	0.040	0.3369	0.041	0.3488	0.049	0.3362	0.1385
4.0	1056.86	0.0080	0.8470	0.8959	0.6592	6.5	0.040	0.3392	0.041	0.3522	0.049	0.3397	0.1195
4.0	1072.40	0.0088	0.8643	0.9059	0.6592	5.9	0.037	0.3781	0.039	0.3850	0.048	0.3754	0.2429
4.0	1062.64	0.0096	0.8568	0.8985	0.6592	6.6	0.039	0.3420	0.041	0.3473	0.049	0.3341	0.2441

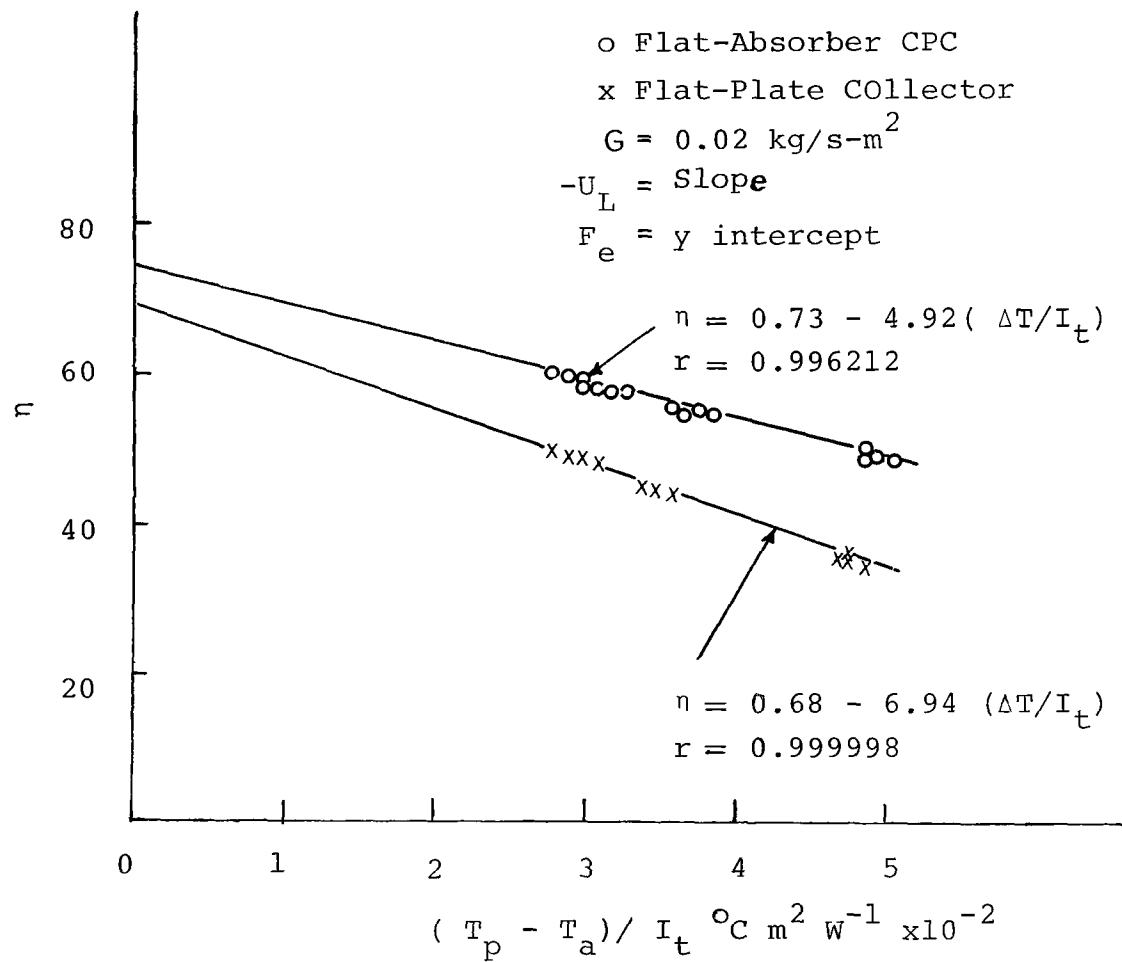


Fig. 6.1 Thermal Performance Based on Average Plate Temperature

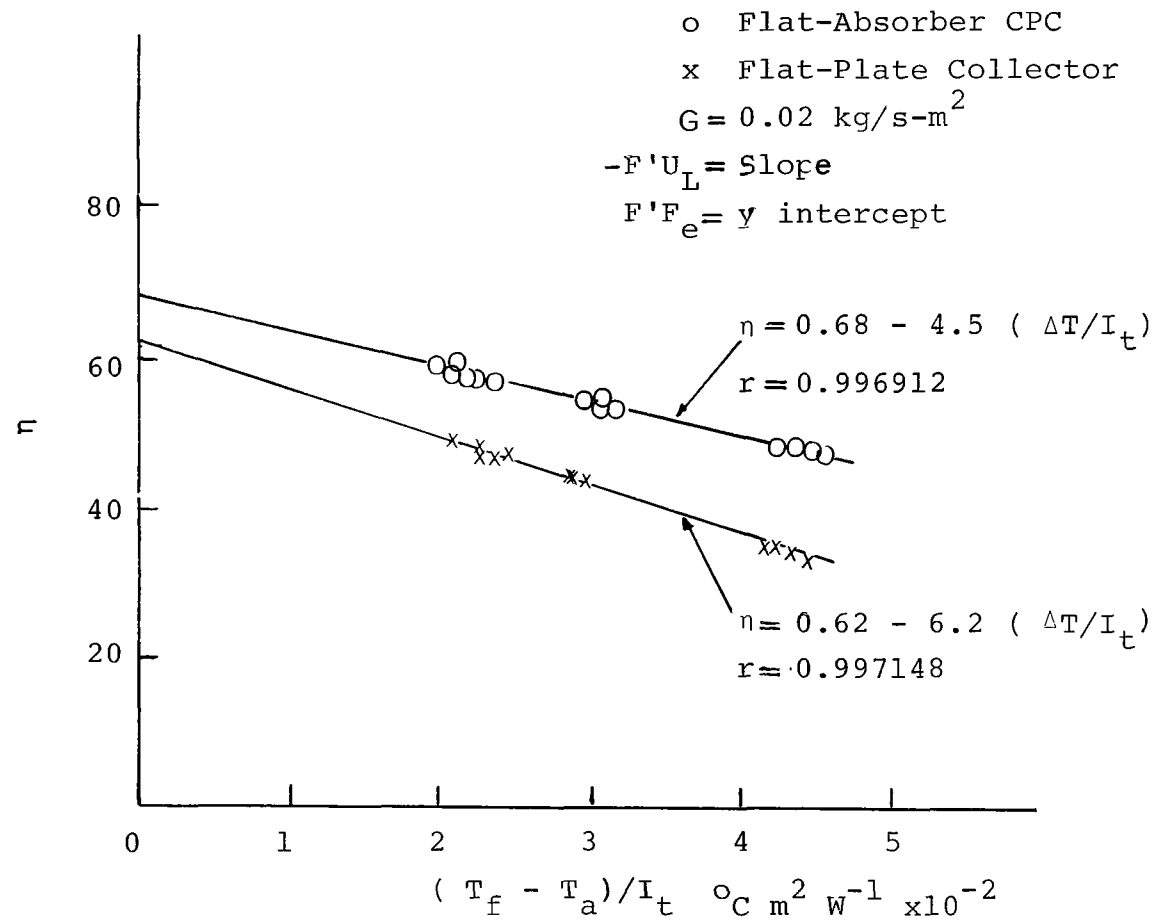


Fig. 6.2 Thermal performance Based on Average Fluid Temperature

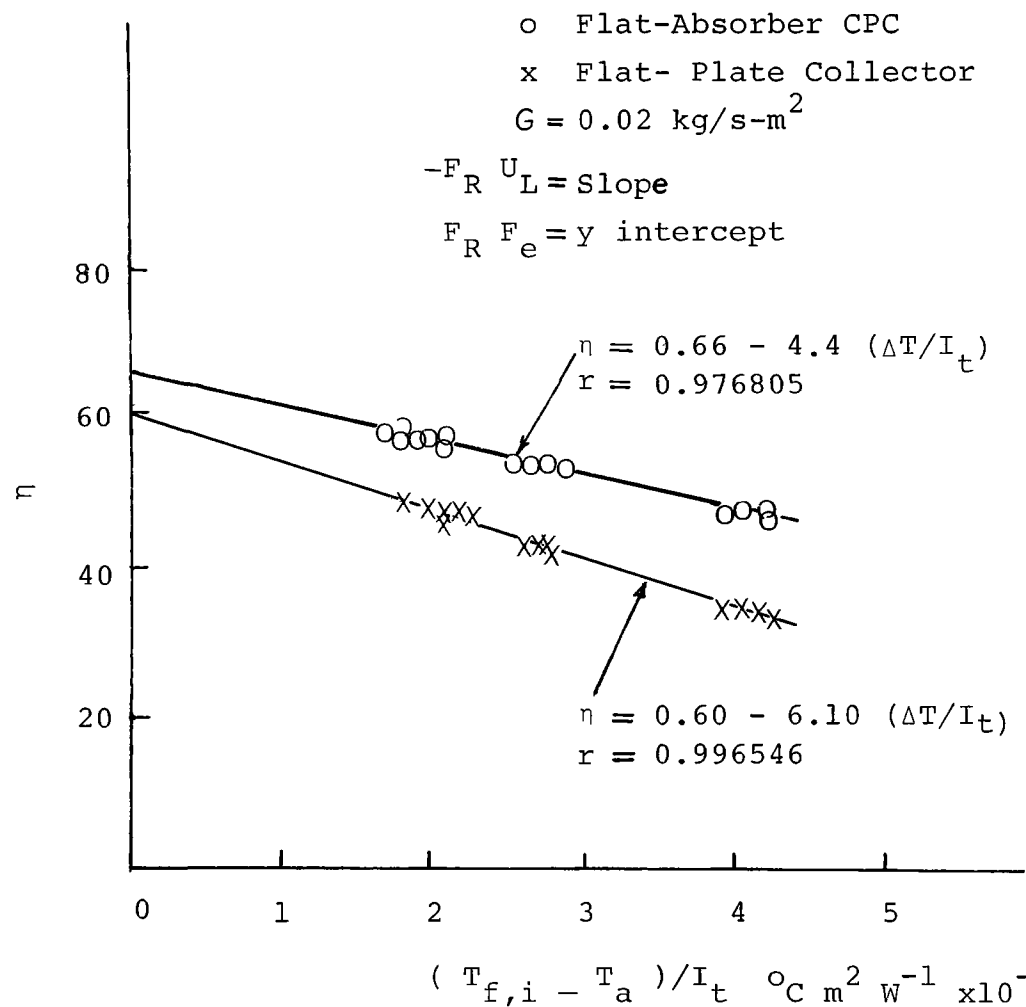


Fig. 6.3 Thermal performance Based on Inlet Fluid Temperature

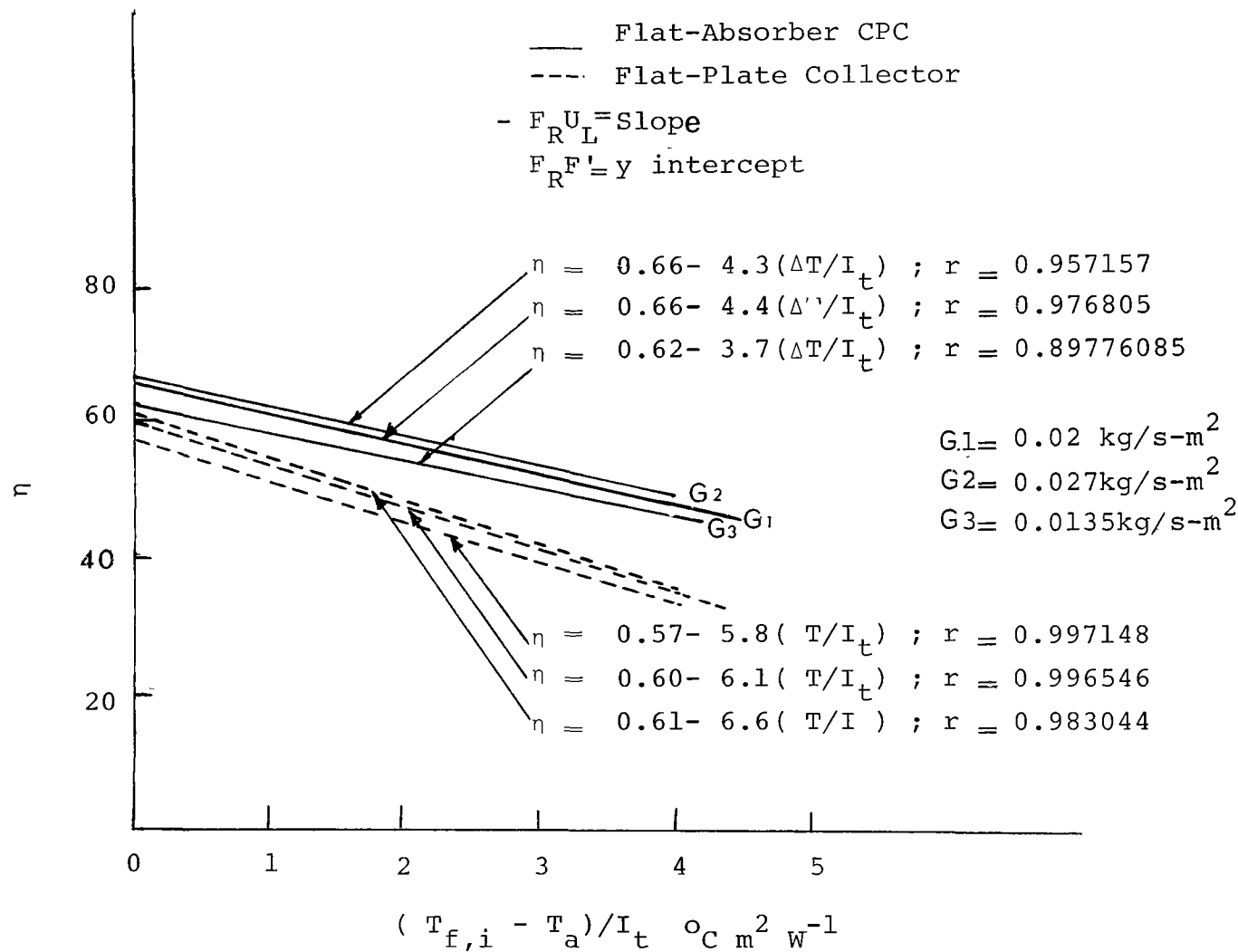


Fig. 6.4 Thermal performance of Solar Collectors for Varying Flow Rates

TABLE 6.7
DESIGN PARAMETERS OF THE TWO COLLECTORS

	Flat-Absorber CPC		Flat-Plate collector	
	Exp.	Cal.	Exp.	Cal.
Heat Removal Factor F_R	0.89	0.91	0.88	0.87
Collector Efficiency Factor F'	0.91	0.93	0.90	0.91
Fraction of Energy Absorbed F_t, F_e	0.73	0.68	0.68	0.66
Overall Loss Coefficient U_L $W^{\circ}C\ m^{-2}$	4.92	4.22	6.94	6.03

TABLE 6.8
ESTIMATED COST OF THE TWO COLLECTORS

	Flat-Plate Collector	Flat-Absorber CPC
	\$	\$
Materials		
Absorber plate	15.00	10.00
Fluid tube	9.00	6.00
2 Sun-Lite covers	10.80	9.40
Insulation	1.00	1.00
Housing	3.00	4.00
Reflecting surface	-	7.00
	<u>38.80</u>	<u>37.40</u>
Overhead and labour	10.00	20.00
total	<u>48.80</u>	<u>57.40</u>

TABLE 6.9
COST EFFECTIVENESS OF THE FLAT-PLATE CPC

$\Delta T/I_t$	Efficiency η_1 (%)	Efficiency η_2 (%)	C_{eff}
0.01	57	53	1.07
0.02	53	47	1.12
0.03	48	41	1.17
0.04	44	35	1.25

CHAPTER 7

CONCLUSION AND RECOMMENDATIONS

Experimental results have revealed that the flat-absorber CPC is more cost-effective than the conventional flat-plate collector, if operated at solar noon under the specified conditions. Although the flat-absorber CPC utilizes much less absorber plate area, the total cost per unit area for both the collectors balances out because of the additional cost of the reflecting surface. However, it is expected that the cost of labour in building the parabolic surface would be reduced marginally by mass production. Another important consideration is that since the absorber width is large enough to receive even the off-axis incident rays, no tracking mechanism is needed and it also allows the collector to operate on cloudy or hazy days.

Although the flat-absorber CPC is tested with water as the heat-removal fluid, the parabolic surface underneath the absorber plate provides a large volume which makes the collector design attractive for air heating systems too. This would eliminate the water tubes. The absorber plate can then be made from an inexpensive material such as aluminium, plastics, etc. This may further reduce the overall cost of the collector. It is recommended that thermal performance of the flat-absorber CPC may be investigated using air as the heat transfer fluid. However, air systems are less efficient than water systems necessitating a large collector for the same performance as a liquid-based system.

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APPENDIX A

SOLAR ANGLES

The position of the sun in the sky can be defined by its altitude, β , above the horizontal and the solar azimuth, A_s , measured from the south. To find these angles the following equations may be used:

$$\sin \beta = \cos \phi \cos \delta \cos \omega + \sin \phi \sin \delta \quad (A1)$$

$$\sin A_s = \cos \delta \sin \omega \cos \beta \quad (A2)$$

where ϕ = Local latitude (north positive)

δ = Solar declination; (i.e. angular position of the sun at solar noon with respect to the plane of the equator)

ω = Hour angle; $0.25 \times$ (number of minutes from local solar noon).

The solar declination δ , can be obtained from the approximate equation [6].

$$\delta = 23.45 \sin\left[360 + \frac{284 + n}{365}\right] \quad (A3)$$

where n is the day of the year. Table A1 gives the values of solar declination for each day of the particular year.

The solar position angles and incident angles for horizontal and tilted surfaces are shown in Figure A1. The angle of incidence for any surface is defined as the angle between the incoming solar direct beam and a line normal to that surface. For a surface with a tilt angle, Σ , and surface azimuth, γ_t , the incident angle θ for the direct solar beam is

$$\cos \theta = A + B \cos \omega + C \sin \omega \quad (A4)$$

where $A = (\cos \Sigma \sin \phi - \sin \Sigma \cos \phi \cos \gamma_t) \sin \delta$

$B = (\cos \Sigma \cos \phi + \sin \Sigma \sin \phi \cos \gamma) \cos \delta$

$C = \sin \Sigma \sin \gamma \cos \delta$

For horizontal surfaces where $\Sigma = 0^\circ$, equation (A4) is simplified as

$$\cos \theta_z = \sin \phi \sin \delta + \cos \Sigma \cos \phi \cos \delta \quad (A5)$$

In this case angle of incidence, θ_z , is known as zenith angle.

The relationship for the angle of incidence, θ_T , on the tilted surfaces with slop Σ to the north or south ($\gamma_t = 0$) can be derived from equation (A4) as

$$\cos \theta_T = \sin (\phi - \Sigma) \sin \delta + \cos (\phi - \Sigma) \cos \delta \cos \omega \quad (A6)$$

Solar Time

Solar time generally differs from local standard time mainly because of the two factors. First, the difference in longitude between the location and the meridian on which local standard time is based. The second factor is the equation of time which is the measure of the extent by which solar time runs faster or slower than mean time. Table A2 gives the values of equation of time for each day of every month of a typical year. Thus the solar time can be calculated from standard time by this equation.

$$\text{Solar time} = \text{Standard time} + E + 4(L_{SM} - L_{LM})$$

where E = Equation of time, in minutes

L_{SM} = Standard time meridian

L_{LM} = Local meridian (place in consideration).

TABLE 1

Mean value of the solar declination (for 1962, noon GMT)

DAY	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.
1	-23°01'	-17°09'	-7°40'	+4°28'	+15°01'	+22°02'	+23°08'	+18°04'	+8°22'	-3°06'	-14°22'	-21°46'
2	-22°56'	-16°52'	-7°17'	+4°51'	+15°19'	+22°10'	+23°03'	+17°49'	+8°00'	-3°30'	-14°41'	-21°56'
3	-22°50'	-16°35'	-6°54'	+5°14'	+15°37'	+22°17'	+22°59'	+17°34'	+7°38'	-3°53'	-15°00'	-22°04'
4	-22°44'	-16°17'	-6°31'	+5°37'	+15°54'	+22°25'	+22°54'	+17°18'	+7°16'	-4°16'	-15°19'	-22°13'
5	-22°38'	-15°59'	-6°08'	+6°00'	+16°12'	+22°31'	+22°49'	+17°02'	+6°54'	-4°39'	-15°37'	-22°21'
6	-22°31'	-15°40'	-5°44'	+6°22'	+16°29'	+22°38'	+22°43'	+16°46'	+6°31'	-5°02'	-15°55'	-22°28'
7	-22°24'	-15°22'	-5°21'	+6°45'	+16°45'	+22°44'	+22°37'	+16°29'	+6°09'	-5°25'	-16°13'	-22°35'
8	-22°16'	-15°03'	-4°58'	+7°08'	+17°02'	+22°50'	+22°30'	+16°12'	+5°46'	-5°48'	-16°31'	-22°42'
9	-22°08'	-14°44'	-4°34'	+7°30'	+17°18'	+22°55'	+22°23'	+15°55'	+5°24'	-6°11'	-16°48'	-22°48'
10	-21°59'	-14°25'	-4°11'	+7°52'	+17°34'	+23°00'	+22°16'	+15°38'	+5°01'	-6°34'	-17°05'	-22°54'
11	-21°50'	-14°05'	-3°47'	+8°14'	+17°50'	+23°04'	+22°08'	+15°20'	+4°38'	-6°57'	-17°22'	-22°59'
12	-21°40'	-13°45'	-3°24'	+8°36'	+18°05'	+23°08'	+22°00'	+15°02'	+4°16'	-7°20'	-17°39'	-23°04'
13	-21°31'	-13°25'	-3°00'	+8°51'	+18°20'	+23°12'	+21°52'	+14°44'	+3°53'	-7°42'	-17°55'	-23°08'
14	-21°20'	-13°05'	-2°37'	+9°20'	+18°35'	+23°15'	+21°43'	+14°26'	+3°30'	-8°04'	-18°11'	-23°12'
15	-21°09'	-12°44'	-2°13'	+9°42'	+18°49'	+23°18'	+21°34'	+14°07'	+3°07'	-8°26'	-18°26'	-23°16'
16	-20°58'	-12°24'	-1°49'	+10°03'	+19°03'	+23°21'	+21°24'	+13°49'	+2°43'	-8°49'	-18°41'	-23°19'
17	-20°47'	-12°03'	-1°25'	+10°24'	+19°17'	+23°23'	+21°14'	+13°29'	+2°20'	-9°11'	-18°56'	-23°21'
18	-20°35'	-11°42'	-1°02'	+10°45'	+19°30'	+23°24'	+21°04'	+13°10'	+1°57'	-9°33'	-19°11'	-23°23'
19	-20°22'	-11°21'	-0°38'	+11°06'	+19°43'	+23°25'	+20°53'	+12°51'	+1°34'	-9°54'	-19°25'	-23°25'
20	-20°09'	-10°59'	-0°14'	+11°27'	+19°56'	+23°26'	+20°42'	+12°31'	+1°11'	-10°16'	-19°39'	-23°26'
21	-19°57'	-10°38'	+0°09'	+11°47'	+20°09'	+23°27'	+20°31'	+12°11'	+0°47'	-10°37'	-19°52'	-23°26'
22	-19°43'	-10°16'	+0°33'	+12°08'	+20°21'	+23°26'	+20°19'	+11°51'	+0°24'	-10°59'	-20°05'	-23°27'
23	-19°29'	-9°54'	+0°57'	+12°28'	+20°32'	+23°26'	+20°07'	+11°31'	+0°01'	-11°20'	-20°18'	-23°26'
24	-19°15'	-9°32'	+1°20'	+12°48'	+20°44'	+23°25'	+19°55'	+11°11'	-0°23'	-11°41'	-20°31'	-23°25'
25	-19°00'	-9°10'	+1°44'	+13°07'	+20°55'	+23°24'	+19°42'	+10°50'	-0°46'	-11°02'	-20°43'	-23°24'
26	-18°46'	-8°47'	+2°08'	+13°27'	+21°05'	+23°22'	+19°29'	+10°29'	-1°10'	-12°22'	-20°54'	-23°22'
27	-18°30'	-8°25'	+2°31'	+13°46'	+21°16'	+23°20'	+19°16'	+10°08'	-1°33'	-12°43'	-21°05'	-23°20'
28	-18°15'	-8°02'	+2°55'	+14°05'	+21°26'	+23°18'	+19°02'	+9°47'	-1°56'	-13°03'	-21°16'	-23°18'
29	-17°59'	-7°38'	+3°18'	+14°24'	+21°35'	+23°15'	+18°48'	+9°26'	-2°20'	-13°23'	-21°27'	-23°15'
30	-17°43'	-7°14'	+3°41'	+14°42'	+21°44'	+23°11'	+18°34'	+9°05'	-2°43'	-13°43'	-21°37'	-23°11'
31	-17°26'	-6°49'	+4°05'	+15°01'	+21°53'	+23°07'	+18°19'	+8°43'	-3°07'	-14°03'	-21°47'	-23°07'

TABLE A.2

Mean value of the equation of time, in minutes (at true noon)

DAY	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.
1	-3.4	-13.6	-12.5	-4.1	+2.8	+2.3	-3.6	-6.3	-0.2	+10.1	+16.3	+11.2
2	3.9	13.7	12.3	3.8	3.0	2.2	3.8	6.2	+0.1	10.4	16.4	10.8
3	4.3	13.8	12.1	3.5	3.1	2.0	4.0	6.2	0.5	10.8	16.4	10.4
4	4.8	13.9	11.9	3.2	3.2	1.9	4.2	6.1	0.7	11.1	16.4	10.0
5	5.2	14.0	11.7	2.9	3.3	1.7	4.4	6.0	1.1	11.4	16.4	9.6
6	5.7	14.1	11.5	2.6	3.4	1.5	4.6	5.9	1.5	11.7	16.3	9.2
7	-6.1	-14.2	-11.2	-2.3	+3.4	+1.3	-4.7	-5.8	+1.8	+12.0	+16.3	+8.8
8	6.5	14.2	11.0	2.1	3.5	1.2	4.9	5.7	2.1	12.3	16.3	8.3
9	6.9	14.3	10.7	1.8	3.6	1.0	5.0	5.5	2.5	12.6	16.2	7.9
10	7.3	14.3	10.5	1.5	3.6	0.8	5.2	5.4	2.8	12.8	16.1	7.5
11	7.8	14.3	10.2	1.2	3.7	0.6	5.3	5.2	3.2	13.1	16.0	7.0
12	8.2	14.3	10.0	0.9	3.7	0.4	5.4	5.1	3.5	13.4	15.9	6.5
13	-8.5	-14.3	-9.7	-0.7	+3.7	+0.2	-5.6	-4.9	+3.9	+13.6	+15.8	+6.1
14	8.9	14.3	9.4	0.4	3.7	0.0	5.7	4.7	4.2	13.8	15.6	5.6
15	9.3	14.2	9.1	-0.2	3.7	-0.2	5.8	4.5	4.6	14.1	15.5	5.1
16	9.6	14.2	8.9	+0.1	3.7	0.4	5.9	4.3	5.0	14.3	15.3	4.6
17	9.9	14.1	8.6	0.2	3.7	0.7	6.0	4.1	5.3	14.5	15.1	4.1
18	10.3	14.0	8.3	0.5	3.7	0.9	6.1	3.9	5.5	14.7	14.9	3.6
19	-10.6	+13.9	-8.0	+0.7	+3.6	-1.1	-6.2	-3.7	+6.0	+14.9	+14.7	+3.2
20	10.9	13.8	7.7	0.9	3.6	1.3	6.2	3.5	6.4	15.1	14.5	2.7
21	11.2	13.7	7.4	1.2	3.5	1.5	6.3	3.2	6.7	15.2	14.3	2.2
22	11.5	13.6	7.1	1.4	3.5	1.7	6.3	3.0	7.1	15.4	14.0	1.7
23	11.8	13.5	6.8	1.6	3.4	2.0	6.4	2.8	7.4	15.6	13.7	1.2
24	12.0	13.4	6.5	1.8	3.3	2.2	6.4	2.5	7.8	15.7	13.4	0.7
25	-12.3	-13.2	-6.2	+1.9	+3.2	-2.4	-6.4	-2.2	+8.1	+15.8	+13.1	+0.2
26	12.5	13.1	5.9	2.1	3.1	2.6	6.4	1.9	8.4	15.9	12.9	-0.3
27	12.7	12.9	5.6	2.3	3.0	2.8	6.4	1.7	8.8	16.0	12.5	0.8
28	12.9	12.7	5.3	2.4	2.9	3.0	6.4	1.4	9.1	16.1	12.2	1.3
29	13.1		5.0	2.6	2.8	3.2	6.4	1.1	9.5	16.2	11.9	1.8
30	13.3		4.7	2.7	2.6	3.4	6.4	0.8	9.8	16.3	11.5	2.3
31	+13.4		4.4		+2.5		-6.3	-0.5		+16.3		2.8

After the *Ephémérides Nautiques* for the year 1963.

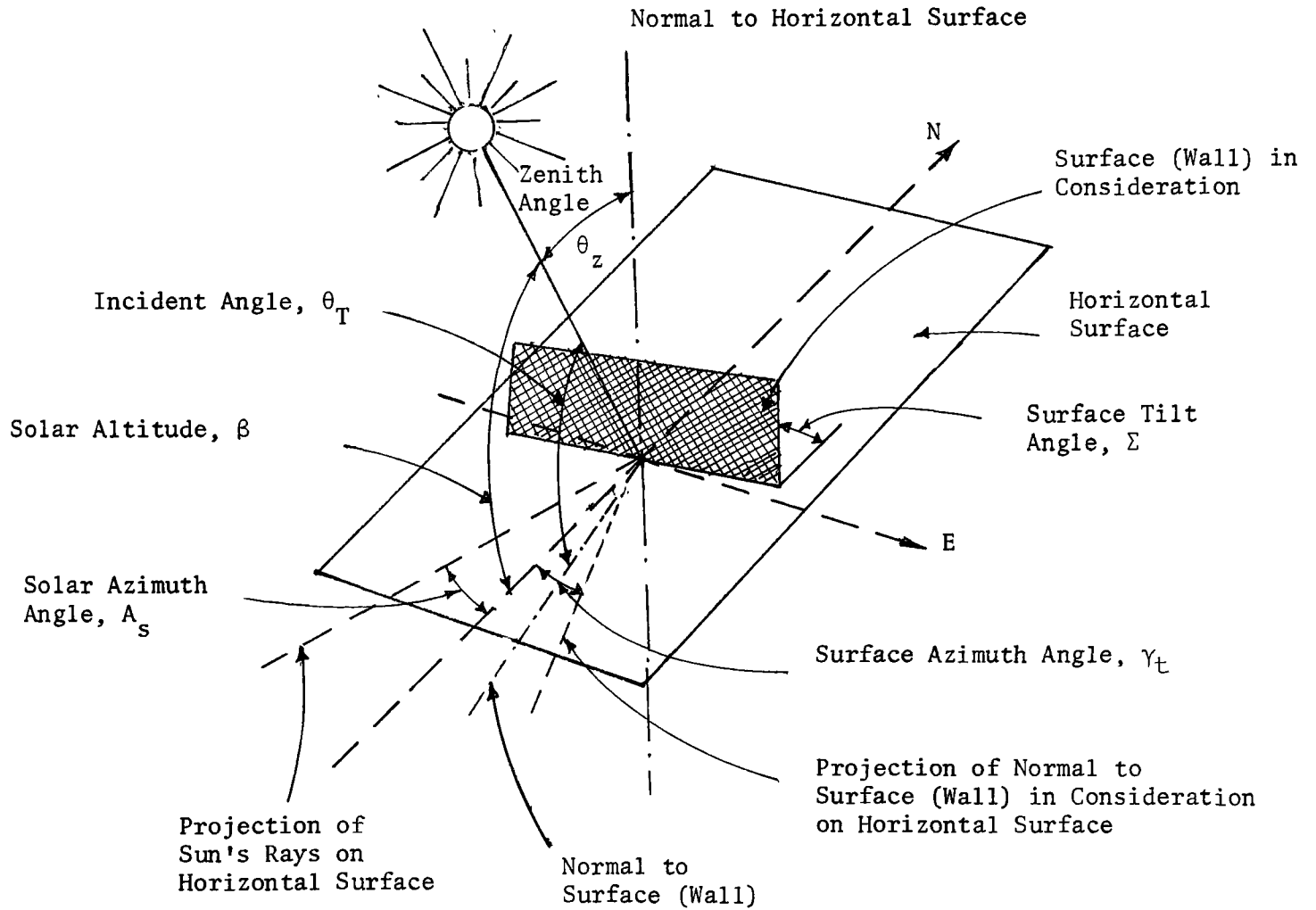


Fig. A1 Solar Angles for Tilted and Horizontal Surfaces

APPENDIX B

MISCELLANEOUS HEAT TRANSFER RELATIONS

The thermal performance prediction of a solar collector usually involves the use of numerous empirical correlations for all of the heat transfer mechanisms related to energy collection system. Some of the important heat transfer coefficients generally used in design and/or analysis of solar collectors are described here.

General

Prandtl Number, P_r , is defined as

$$P_r = \frac{\mu C_p}{K_f} \quad (B1)$$

Rayleigh Number, R_a , can be expressed as

$$R_a = \frac{\beta g \Delta T x^3}{D_a \nu} \quad (B2)$$

Grashof Number G_r , is expressed as

$$G_r = \frac{\beta g \Delta T x^3}{\nu^2} \quad (B3)$$

Nusselt Number, N_u , is written as

$$N_u = \frac{h_f a}{K_f} \quad (B4)$$

Convection Heat Transfer Between the Collector Plate and the Glass Cover

Tabor [19] examined the results of a number of investigations, and presented them in the nondimensional forms as,

For horizontal planes, and lower plate at high temperature,

$$N_u = 0.168 R_a^{0.281}, \quad 10^4 < G_r < 10^7 \quad (B5)$$

For planes inclined at 45° , with heat flow upwards,

$$N_u = 0.102 R_a^{0.310}, \quad 10^4 < G_r < 10^7 \quad (B6)$$

For vertical planes,

$$N_u = \begin{matrix} 0.0685 R_a^{0.327} & 1.5 \times 10^5 < G_r < 10^7 \\ 0.0369 R_a^{0.381} & 1.5 \times 10^4 < G_r < 1.5 \times 10^5 \end{matrix} \quad (B7)$$

Hollands [20] studied the natural convection of air between the inclined parallel plates with the lower plate at higher temperature than the upper plate. Hollands recommends the relation

$$N_u = 1 + 1.44 \left[1 - \frac{1708}{R_a \cos \Sigma} \right] \left[1 - \frac{1708 (\sin 1.8 \Sigma)^{1.6}}{R_a \cos \Sigma} \right] + \left[\left(\frac{R_a \cos \Sigma}{5830} \right)^{1/3} - 1 \right] \quad (B8)$$

Note: dotted brackets go to zero when negative.

Duffie and Beckman proposed the use of the relation presented by Dropkin and Somerscales [21]. Using liquid in between parallel planes for tilt angles $0^\circ - 90^\circ$ and G_r greater than 2×10^5 as

$$N_u = [0.069 - 0.020 \left(\frac{\Sigma}{90} \right)] (G_r P_r)^{1/3} (P_r)^{0.074} \quad (B9)$$

For air with Prandtl number 0.7 this relation simplified to

$$N_u = [0.060 - 0.01 \left(\frac{\Sigma}{90} \right)] G_r^{1/3} \quad (B10)$$

Forced Convection in an Inclined Tube

Duffie and Beckman proposed the use of correlation quoted in Krieth [22, p. 391].

$$N_u = \frac{G_z}{4} \ln \left[\frac{1}{1 - (2.654/P_r)^{0.167} G_z^{0.5}} \right] \quad (B11)$$

Whillier [9] quoted the following relation without reference.

$$\bar{N}_u = 4.36 + \frac{0.067(D/L)R_e P_r}{1+0.04[(D/L)R_e P_r]^{2/3}} \quad (B12)$$

Seider and Tate (Ref. Krieth [4, p. 391]) suggested an empirical relation which is widely used to correlate experimental results for liquids.

$$\bar{N}_u = 1.86(G_z)^{0.33} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} \quad (B13)$$

The correction factor $\left(\frac{\mu_b}{\mu_s} \right)^{0.14}$ takes into account the effect of temperature variation on the physical properties of the liquid used.

Heat Transfer from a Collector Cover Exposed to the Atmosphere

Duffie and Beckman have quoted the linear relation given by McAdams which relates the heat transfer coefficient, $h_w(W/m^2{}^{\circ}C)$ to the wind speed, $V(m/s)$

$$h_w = 5.7 + 3.8 V \quad (B14)$$

Radiation Heat Transfer Coefficient

The heat transfer by radiation between the two arbitrary surfaces is given by

$$Q = A_1 h_r (T_2 - T_1)$$

where h_r is the heat transfer coefficient which is expressed as

$$h_r = \frac{\sigma (T_2^2 + T_1^2) (T_2 + T_1)}{[(1/E_1) + (1/E_2) - 1]} \quad (B15)$$

APPENDIX C

COMPUTER PROGRAM

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C      THERMAL PERFORMANCE ANALYSIS OF A FLAT-ABSORBER CYLINDRICAL
C      PARABOLIC CONCENTRATOR
C
C      TRANSMITTANCE-ABSORPTANCE PRODUCT
1      IMPLICIT REAL*4 (A-Z)
2      DIMENSION TEST (4,6) , DATA (16,16) , PFSULT (16,14)
3      INTEGER J,O,K,I
4      LTD=45.45
5      TILT=65.0
6      LSTM=75.0
7      OTTM=75.617
8      K=1
9      DO 100 O=1,4
10     READ 10, ET,DICL,CPF,MU,RHO
11     10  FORMAT (F6.2,2X,F6.2,2X,F6.4,2X,F6.4,2X,F6.1)
12     DO 100 J=1,4
13     READ 20, LSTD,IP,IRH1,IRH2
14     20  FORMAT (F5.2,2X,F6.3,2X,F5.1,2X,F6.2)
15     DATA (K,1)=O
16     DATA (K,2)=IP
17     DATA (K,3)=IRH1
18     DATA (K,4)=IRH2
19     LAT=LSTD+ ( (ET+4. *(LSTM-OTTM)) /60.)
20     OMEGA=(12.-LAT)*15.
21     PI=3.1416
22     RAD=PI/180.
23     W=OMEGA*RAD
24     D=DICL*RAD
25     X=(LTD-TILT)*RAD
26     THET1 =ARCOS ( (COS (X)*COS (D)*COS (W)) + (SIN (X)*SIN (D)) )
27     N1=1.
28     N2=1.526
29     THET2 =ARSIN ( (N1/N2)*SIN (THET1) )
30     YP=(SIN (THET2-THET1)) **2/(SIN (THET2+THET1)) **2
31     ZP=(TAN (THET2-THET1)) **2/(TAN (THET2+THET1)) **2
32     TOR=(((1-YP)/(1+3*YP)) + ((1-ZP)/(1+3*ZP)))/2
33     TX=0.1016
34     LT=TX/COS (THET2)
35     EK=0.161
36     TOA=EXP (-EK*LT)
37     TAU=TOR*TOA
38     ALF=0.95
39     TOL=(TOR*ALF)/(1-(1-ALF)*.24)
40     A1=0.15
41     A2=0.62
42     FEI=TOL+(A1*(1.-TOA)) + (A2*TAU*(1.-TOA))
43     SF=0.03
44     DF=0.02
45     FI=FEI*(1-SF)*(1-DF)*0.95
46     FE=0.98*0.95*(1-DF)*(1-SF)*FI
47     LB=1.22
48     AB=0.0929
49     AA=0.13
50     AZ=AB/AA
51     AC=0.520
52     FL=3.175E-2
53     WF=7.62E-2
54     REF=0.88

```

```

55      NT=4.
56      IT=IP*84.875
57      IRHT=IRH1*2.0063
58      IRHD=IRH2*1.9565
59      RTD=IRHD/IRHT
60      IB=IT*(1-RTD)
61      RBT=IB/IT
62      PHY=80.*RAD
63      ETA=0.266*RAD
64      IW=2* FL *TAN(ETA)
65      B=WF/IW
66      ZZ=0.5*B*TAN(ETA)
67      CIE=2*ATAN(ZZ)
68      NP=4*((TAN(PHY/2)-TAN(CIE/2)))
69      FB=(NT*REF*NP*FI*FL*LB*RBT)/AC
70      FF=FE*AZ
71      FT=FB+FF

      C
      C      CALCULATION OF LOSSES
      C
      C      BACK AND EDGE LOSS COEFFICIENTS, UB, UE
72      KINS=0.036
73      LINS=0.5E-1
74      KBW=0.1267
75      LBW=0.625E-2
76      KPW=0.1584
77      LPW=1.27E-3
78      RB1=LINS/KINS
79      RB2=LBW/KBW
80      RB3=LPW/KPW
81      R11=RB1+RB2+RB3
82      AG=0.854
83      KEW=0.1584
84      LEW=0.203E-1
85      DEP=0.1374
86      RINS=LINS/KINS
87      REW=LEW/KEW
88      PER=4.023
89      UE=(1.0/(REW+RINS+REW))*(DEP*PER/AG)

      C
      C      TOP LOSS COEFFICIENT
      C
90      READ 50,T1,TO,TP1,TG1,TG2,TA1,WV1,TR1
91      50  FORMAT(8(F4.1,2X))
92      DATA(K,5)=T1
93      DATA(K,6)=TO
94      DATA(K,7)=TP1
95      DATA(K,8)=TG1
96      DATA(K,9)=TG2
97      DATA(K,10)=TA1
98      DATA(K,11)=WV1
99      DATA(K,12)=TR1
100     SIGMA=5.668E-8
101     EP=0.95
102     EC=0.88
103     EM=0.30
104     FPC=1
105     FPR=FPC
106     EA=1/((1/FPC)+((1/EP)-1)+(AZ*((1/EC)-1)))
107     EZ=1/((1/FPR)+((1/EP)-1)+(AZ*((1/EM)-1)))

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```

108      TA=TA1+273
109      TS=0.0552*(TA**1.5)
110      TC1=TC1+273.
111      TC2=TC2+273.
112      TP=TP1+273.
113      TR=TR1+273.
114      HR1=SIGMA*(TC1+TP)*(TP**2+TC1**2)/(EA)
115      HR2=SIGMA*(TC2+TC1)*(TC2**2+TC1**2)/((2./FC)-1.)
116      HFS=SIGMA*(TP+TS)*((TP**2)+(TP**2))/(FZ)
117      HRS=SIGMA*(TC2+TS)*(TC2**2+TS**2)*FC
118      KA=0.025
119      TPC=TP-TC1
120      TCC=TC1-TC2
121      TPR=TP-TR
122      LPC=2.2E-2
123      LCC=1.5E-2
124      LPR=PL*0.5
125      MODP=0.1284E9
126      MODC=0.1558E9
127      GR1=MODP*TPC*(LPC**3)
128      GR2=MODC*TCC*(LCC**3)
129      GR3=MODP*TPR*(LPR**3)
130      PFA=0.71
131      PA1=GR1/PRA
132      RA2=GR2/PFA
133      RA3=GR3/PRA
134      PEI=TILT*RAD
135      SEI=PEI*1.8
136      NU1=1+1.44*(1-(1708.0/(PA1*COS(PEI))))*(1-(((SIN(SEI))**1.6)*
      & (1708.0/(PA1*COS(PEI)))))+(((PA1*COS(PEI)/5830.0)**0.333)-1.0)
137      NU2=1+1.44*(1-(1708.0/(RA2*COS(PEI))))*(1-(((SIN(SEI))**1.6)*
      & (1708.0/(RA2*COS(PEI)))))+(((RA2*COS(PEI)/5830.0)**0.333)-1.0)
138      NU3=1+1.44*(1-(1708.0/(RA3*COS(PEI))))*(1-(((SIN(SEI))**1.6)*
      & (1708.0/(RA3*COS(PEI)))))+(((RA3*COS(PEI)/5830.0)**0.333)-1.0)
139      HPCT=NU1*KA/LPC
140      HCCT=NU2*KA/LCC
141      WV=WV1*0.444
142      HW=5.7+3.8*WV
143      HPRT=NU3*KA/LPF
144      R1=1/(HR1+HPCT)
145      R2=1/(HR2+HCCT)
146      R3=1/(HFS+HW)
147      UT=1./(R1+R2+R3)
148      R4=1/(HRF+HPIT)
149      UB=1/(R11+P4)
      C
      C      OVERALL LOSS COEFFICIENT
      C
150      UL=UB+UT+UF
      C
      C      COLLECTOR EFFICIENCY FACTOR
      C
      C      FLUID HEAT TRANSFER COEFFICIENT, HFC
151      READ 40 , DELTW,MVF,MVC,VLT
152      40  FORMAT(F4.1,2X,F5.3,2X,F5.3,2X,F6.4)
153      DATA(K,13)=DELTW
154      DATA(K,14)=MVF
155      DATA(K,15)=MVC
156      DATA(K,16)=VLT
157      KW=0.440

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158      DI=0.9525F-2
159      DO=.127F-1
160      AI=(PI/4)*(DI**2)
161      CP=CPF*4.184E3
162      PP=MU*CP/KW
163      QF=VLT*(500./4.04)
164      DELTF=MVF/0.1696
165      DELTFW=(DELTF+DFITW)/2.
166      MFL=QF/(CP*DELTFW)
167      VFL=MFL/FHO
168      VEL=VFL/(AI*NT)
169      PE=(VFL*DI)*(FHO/MU)
170      GZ=RE*PP*(DI/LB)
171      HFI=1.86*(KW/DI)*(GZ**0.333)

      C
      C
      C      FIN EFFICIENCY FACTOR, F

172      KP=385
173      DELTP=0.79E-3
174      G1=MFL/AC
175      M=SQRT(UL/(KP*DELTP))
176      Y=0.5*M*(WF-DO)
177      F=TANH(Y)/Y

      C
      C
      C      COLLECTOR EFFICIENCY FACTOR, FP

178      CW=0.80E2
179      RB=0.2595E2
180      RF=PI*DI*HFI
181      RP=UL*(DO+(WF-DO)*F)
182      UO=1./(WF*((1./RP)+(1./RB)+(1./RF)+(1./CW)))
183      FP=UO/UL

      C
      C
      C      FLOW FACTOR, FR

184      NTU=UL/(G1*CP)
185      FP=(1.-EXP(-FP*NTU))/NTU

      C
      C
      C      INSTANTANEOUS COLLECTOR EFFICIENCY, CLEFF

186      IT=IP*84.875
187      TI=T1+273.
188      TE=TO+273.
189      TF=(T1+TO)/2
190      IRTI=(TI-TE)/IT
191      IRTF=(TF-TA1)/IT
192      IRTP=(TP-TA)/IT
193      EFTI=FP*(FT-(UL*IRTI*AZ))
194      EFTF=FP*(FT-(UL*IRTF*AZ))
195      EFTP=FT-(UL*IRTP*AZ)

      C
      C
      C      INSTANTANEOUS EXPERIMENTAL EFFICIENCY, INEFF

196      TEI=TF-TI
197      TFO=MVC/(8*42.4E-3)
198      TTC=(TEI+TFO)/2.
199      EFMEAS=(G1*CP*TTC)/IT
200      RESULT(K,1)=C
201      RESULT(K,2)=IT
202      RESULT(K,3)=MFL

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203      RESULT(K,4)=FR
204      RESULT(K,5)=FP
205      RESULT(K,6)=FT
206      RESULT(K,7)=UL
207      RESULT(K,8)=IFTI
208      RESULT(K,9)=EFTI
209      RESULT(K,10)=IFHF
210      RESULT(K,11)=DITF
211      RESULT(K,12)=IFTP
212      RESULT(K,13)=EFTP
213      RESULT(K,14)=EFMEAS
214      K=K+1
215 100 CONTINUE
216      PRINT 300
217 300 FORMAT('1',5(/),25X,'EXPERIMENTAL DATA FOR FLAT-ABSORBER CPC FOR F
    @LOW RATE = 0.0200 KG/S-M',////)
218      PRINT 310
219 310 FORMAT(' ','SET',5X,'IP',6X,'IEH1',4X,'IEH2',6X,'T1',5X,
    @'TO',4X,'TP1',4X,'TG1',4X,'TG2',4X,'TA1',5X,'WV1',3X,'T11',5X,
    @'DELTW',4X,'MVF',4X,'MVC',5X,'VL',//)
220      O=0
221      DO 330 I=1,4
222      DO 320 K=1,4
223      O=O+1
224 320 PRINT 340,(DATA(O,J),J=1,16)
225 340 FORMAT(' ',F3.1,4X,F6.3,3X,F5.1,2X,F6.2,4X,F4.1,3X,F4.1,3X,
    @F4.1,3X,F4.1,3X,F4.1,3X,F4.1,3X,F4.1,3X,F4.1,4X,F4.1,4X,F5.3,
    @2X,F5.3,2X,F6.4,/)
226 330 PRINT 335
227 335 FORMAT(' ')
228      PRINT 350
229 350 FORMAT('1',5(/),25X,'EXPERIMENTAL DATA FOR FLAT-ABSORBER CPC FOR F
    @LOW RATE = 0.0200 KG/S-M',////)
230      PRINT 360
231 360 FORMAT(' ','SET',5X,'IP',6X,'FLOW',7X,'FF',7X,'TP',7X,'FT',5X,'UL'
    @,5X,'IFTI',4X,'EFTI',5X,'IFTF',4X,'EFTF',5X,'IPTP',4X,'EFTP',
    @5X,'EFMEAS',//)
232      O=0
233      DO 390 I=1,4
234      DO 370 K=1,4
235      O=O+1
236 370 PRINT 380,(RESULT(O,J),J=1,14)
237 380 FORMAT(' ',F3.1,3X,F7.2,2X,F6.4,4X,F6.4,3X,F6.4,3X,F6.4,3X,F3.1,
    @4X,F5.3,2X,F6.4,4X,F5.3,2X,F6.4,4X,F5.3,2X,F6.4,4X,F6.4,/)
238 390 PRINT 395
239 395 FORMAT(' ')
240      PRINT 400
241 400 FORMAT('1')
242      STOP
243      END

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\$ENTRY

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