

THERMODYNAMIC FEASIBILITY OF
SOLAR-ASSISTED ABSORPTION COOLING

by

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ABSTRACT

Solar cooling is probably among the most promising ways for solar energy utilization in near future. The purpose of the present study is to investigate the thermodynamic feasibility of operating the solar absorption system for space cooling. The thermodynamic analysis is performed for various ranges of the operation parameters selected to conform to typical climatic conditions of Turkey. At the beginning of the study, both ammonia-water and lithium-bromide-water absorption systems have been considered. The lithium-bromide-water system is preferred since it has become more popular for air conditioning applications. The most important advantage of lithium-bromide-water absorption system is that, it operates satisfactorily at generator temperatures of 85 to 90°C, achievable by a flat plate collector.

A theoretical model is developed to investigate the characteristics of a lithium-bromide-water absorption type refrigeration cycle. A computer program is developed to find the

design conditions of this system. Another computer program is developed to calculate the amount of useful heat energy that can be obtained from flat plate solar collectors. Combined system consists of a flat plate solar collector, generator, condensor, evaporator, absorber and economizer. The results indicate that the lithium-bromide-water absorption system combined with flat plate solar collectors may provide energy savings in considerable amounts.

ÖZET

Yakın bir gelecekte,güneş enerjisi kullanımları arasında, güneş enerjisi ile soğutma,en çok kullanılabilirliği olanlardan biri olacaktır.Bu çalışmanın amacı,ısı kaynağı olarak güneş enerjisini kullanan abzorpsiyonlu sistemlerin,bina soğutmasında kullanılmasının olanaklı olup olmadığının incelenmesidir.Türkiye'nin iklim koşullarına uygun parametreler kullanılarak,termodinamik çözümleme gerçekleştirildi.Başlangıçta hem $\text{NH}_3\text{-H}_2\text{O}$ hem de $\text{LiBr-H}_2\text{O}$ abzorpsiyonlu sistemler ele alındı.Netice olarak,iklimlendirme uygulamalarına daha uygun olduğu için $\text{LiBr-H}_2\text{O}$ abzorpsiyonlu sistem tercih edildi.Bu sistemin en önemli üstünlüğü,güneş enerjisi toplayıcılarıyla elde edilebilecek sıcaklıkların $\text{LiBr-H}_2\text{O}$ sisteminin jenaratör sıcaklıklarına ($85\text{-}90^\circ\text{C}$) uygun olmasıdır.

$\text{LiBr-H}_2\text{O}$ abzorpsiyonlu soğutmanın özelliklerini incelemek için kuramsal bir model geliştirildi.Bu sistemin dizayn koşullarını bulmak için,bir bilgisayar programı hazırlandı.Bir başka program da düz güneş toplayıcılarından elde edilebilecek faydalı ısıyı hesaplamak için geliştirildi.En son sistem bir toplayıcı,jenaratör,kondansatör,evaporatör ve ekonomizörden ibarettir.Sonuçlar $\text{LiBr-H}_2\text{O}$ sisteminin önemli miktarlarda enerji kazancı sağlayabileceğini göstermektedir.

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LIST OF SYMBOLS

- α : The angle between the collector and the vertical plane
 α_z : The solar azimuth angle
 A : The area of the medium
 A_c : The area of the collector
 b : Thickness
 b_e : Altitude (sun's altitude angle)
 B : Atmospheric extinction coefficient
 c_p : Specific heat factor
 C : The ratio of diffuse to direct normal irradiation on a horizontal surface
 d : Declination
 e : Emittance
 e_g : Emittance of glass
 e_p : Emittance of plate
 F_R : Heat removal factor
 g_r : Ground reflectance
 G_d : Diffuse instantaneous solar radiation
 G_D : Reflected instantaneous solar radiation
 G_{DN} : Direct instantaneous solar radiation
 G_T : Total instantaneous solar radiation
 h : Enthalpy
 H : Monthly average daily radiation on a horizontal surface
 HD : Monthly average diffuse radiation on a horizontal surface

ΔH_x	: Heat of solution
h_c	: Convection heat transfer coefficient
h_w	: Wind heat transfer coefficient
I_b	: Beam insolation for an hour
I_d	: Diffuse insolation for an hour
k	: Thermal conductivity of the material
l	: Latitude
$\dot{m}$	: Mass flow rate
n	: Day of the year
N	: Number of glass covers
σ	: Stephan Boltzman constant
θ	: Incidence angle
θ_z	: The sun's zenith angle
ϕ	: Tilt angle
p	: Pressure
q	: The total rate of heat transfer
q_c	: The rate of heat flow (flux) by convection
q_k	: The rate of heat flow (flux) by conduction
q_r	: The rate of heat flow (flux) by radiation
Q_h	: Supply heat energy
Q_u	: Useful heat gain obtained from a solar collector
R_b	: The ratio of beam radiation on the tilted surface to that on a horizontal surface at any time
R_c	: Resistance to convective heat transfer

- R_d : The ratio of diffuse radiation on the tilted surface to that on a horizontal surface at any time
 R_k : Thermal resistance to the flow of heat by conduction
 S : The solar radiation absorbed by a solar collector
 T : Temperature
 T_a : Ambient temperature
 T_{coll} : Average collector surface temperature
 T_f : Fluid temperature
 $T_{f,i}$: Fluid inlet temperature
 T_i : Fluid inlet temperature
 T_{mean} : Mean ambient temperature
 T_o : Fluid outlet temperature
 T_{out} : Outside temperature
 $T_{p,m}$: Plate mean temperature
 T_s : Surface temperature
 (TA) : Transmittance-absorptance product
 $(TA)_b$: Beam transmittance-absorptance product
 $(TA)_d$: Diffuse transmittance-absorptance product
 $(TA)_g$: Ground transmittance-absorptance product
 $(TA)_n$: Value of (TA) for beam radiation in normal direction to the collector surface
 U_L : Overall heat loss coefficient
 η : Efficiency
 w : Hour angle
 W : Work

W_p : Pump work

x : Distance

X : Concentration (mass/mass)

I. INRODUCTION

It can be easily said that all forms of energy are derived from the sun. However, our most common forms of energy fossil fuels received their solar input eons ago and have changed their characteristics so that they are now in a highly concentrated form. Since it is apparent that these stored, concentrated energy forms are now being used at such a rapid rate that they will be depleted in the not-too distant future, we must begin to supply a large portion of our energy needs not from stored, but from incoming solar energy as soon as possible.

It is known that there are many possible applications of solar energy like heating of buildings; providing hot service water, which has been widely used in South and South-East Regions of Turkey for years; heating of swimming pools; production of hydrogen; cooling etc.

Until ten or fifteen years ago, proposals for using the sun's energy on a significant scale were likely to be received with scepticism. During the last decade, however, the increasing cost of conventional fuels, the political uncertainties related to the supply

of petroleum, and, lastly, the problems associated with electric-energy generation from nuclear sources have modified this skepticism dramatically. The nuclear accident, happened in Chernobyl, in Russia, on April of 1986 is a good example of problems created by nuclear power sources. Additionally, the decrease in petroleum prices in the years of 1985-1986 is only a result of political decisions of OPEC. There is no doubt that petroleum prices will increase drastically. All of these make solar energy more and more important. It must be also added that systems powered by solar energy do not create environmental pollution.

The successful utilization of solar energy on a large scale depends upon the design of systems and a thorough consideration of their economics. In contrast to gas or electric heating systems and mechanical cooling systems, which can be easily standardized, no single system is satisfactory for all tasks. The design and size of a solar system must, therefore, be matched to its task. It should not only meet technical requirements but should also involve and resolve satisfactorily the impact on construction economics, system amortization and building esthetics.

At the beginning, it was said that there are many possible applications of solar energy. Its use for cooling, especially, the cooling of buildings seems the most attractive one. This is used in which the demand for energy closely matches the supply of solar energy, even to variation within the day. Solar energy is an appropriate source for the

heating and cooling of buildings because the temperature are usually moderate and high temperature sources are not required.

In the developed countries the refrigeration of food is normal and air-conditioning of building is fairly common. Cooling is a significant element in the energy economy of these countries, with air-conditioning causing peaks in electrical demand on summer afternoons and evenings. Since Turkey is a developing country, there is an increasing demand for refrigeration of food and air-conditioning of buildings. For this reason, the increase in electrical demand, especially, for air-conditioning will be a serious problem for Turkey in near future.

Solar cooling of buildings represents a potentially significant application of solar energy for building air-conditioning in most sunny regions of Turkey. Solar-cooling technology is presently not as advanced as solar-heating technology, but research work is expected to close the gap between two in 1980's.

As a result, it can be said that the practical implementation of solar cooling will depend on social acceptability, economical competitiveness, political decisions, and technical suitability for local conditions. Local factors include the availability of electric power and cooling water. It is difficult for solar cooling to compete with compression refrigeration where electric power is plentiful. Political decisions that will effect the competitive position of solar energy include deregulation of natural gas prices-when natural gas pipeline between Russia and Turkey is completed, it will be important-, cost of petroleum, and tax incentives.

The importance of the thesis subject has been discussed in this chapter. The Nature and historical perspective of solar energy have been introduced in Chapter 2.

The next chapter is a review of heat transfer mechanism. Methods of solar energy collection and use has been discussed in Chapter 4. The importance of solar cooling has been emphasized in Chapter 5. In addition, in this chapter vapor-compression systems and heat pumps have been introduced.

The topic of Chapter 6 is Absorption Refrigeration. As its name refers to, this chapter covers the introduction of absorption refrigeration and fundamental characteristics of binary mixtures. The comparison of $\text{LiBr-H}_2\text{O}$ and $\text{NH}_3\text{-H}_2\text{O}$ absorption systems and analytical evaluation of the $\text{LiBr-H}_2\text{O}$ cycle have been done. Finally, optimization of this system by means of a computer program has been achieved. The results of this study are listed in Appendix A1. At this stage of the study, the source of heat is not so important since the aim is only to investigate the characteristics of a $\text{LiBr-H}_2\text{O}$ absorption type refrigeration cycle.

In Chapter 7 solar angles, solar radiation on the surface of the Earth and calculation of useful heat energy obtained from flat plate collectors have been discussed. A computer program has been developed to calculate the amount of useful heat energy that can be obtained from flat plate collectors. The results are listed in appendix A2.

In the last chapter, namely, Chapter 8 the results of above mentioned computer programs have been combined. Consequently, it was found that solar assisted $\text{LiBr-H}_2\text{O}$ absorption system might provide energy savings in considerable amounts.

II. THE NATURE AND HISTORICAL PERSPECTIVE OF SOLAR ENERGY

2.1 ENERGY-CONTEMPORARY SOURCES

Energy is defined in classical thermodynamics as the capacity to do work. From a practical point it is the basic ingredient for all industrialized societies. In industrialized countries energy is currently derived from five primary sources: petroleum, natural gas, coal, wood and electricity generated from hydraulic, thermal and nuclear power plants. The supplies of these common energies, except wood and electricity generated from hydraulic and nuclear power plants, are finite. Additionally, there are some problems associated with electricity generation from nuclear power plants. As current energy sources become exhausted energy gap will develop depending on the population growth and increased dependence on energy.

After nonrenewable energy sources are consumed in what some authors call this "Fossil Fuel Age" mankind must turn to longer-term permanent

energy sources. The two most significant of these are nuclear and solar energy. Nuclear energy requires highly technical and costly means for its safe and reliable utilization and may have undesirable side effects. Solar energy, on the other hand, shows promise of becoming a dependable energy source with new requirements of a highly technical and specialized nature for its widespread utilization. In addition, there appear to be significant polluting effects from its use.

Turkey has not any natural energy sources in considerable amounts. It has to import petroleum and natural gas from another countries. Coal, which is not good in quality and sufficient in quantity, and hydraulic sources are not sufficient neither to meet all energy requirements of Turkey. All these makes solar energy an important energy source, especially, for developing and sunny countries like Turkey.

2.2 THE NATURE OF SOLAR ENERGY

Solar energy is the world's most abundant permanent source of energy. The amount of solar energy intercepted by the planet earth is 170 trillion kW, an amount 5.000 times greater than the sum of all other inputs (terrestrial, nuclear, geothermal, and gravitational energies and lunar gravitational energy).

About 30 percent of this amount is reflected to space 47 percent is converted to low temperature heat and reradiated to space, and 23 percent powers the evaporation/precipitation cycle of the biosphere;

less than 1/2 percent is represented in the kinetic energy of the wind and waves and in photosynthetic storage in plants. The amount of the sun's energy intercepted by earth is only a tiny fraction--one thousandth of one millionth--of the total released by the conversion of 4 million tons of hydrogen per second to helium in the sun.

Although it is abundant, solar energy having an effect on the earth's atmosphere is relatively less (approximately 430 Btu/hrft² °C). Traversing the earth's atmosphere dilutes it further by attenuation, local weather phenomena and air pollution. Moreover, solar energy is received only intermittently at any point on earth. The solar energy which arrives on the surface of the earth is of two forms: direct radiation and diffuse radiation. Direct radiation is collimated and capable of casting a shadow: diffuse radiation is dispersed or reflected, by the atmosphere, and not collimated. In considering how solar energy can be best be used, the ratio of direct to diffuse radiation becomes important. The ratio of direct to diffuse radiation varies with time and location. The amount of direct radiation diminishes as air pollution increases.

On the average the radiation striking 1 ft² of the earth's surface is on the order of 100 to 200 Btu/hrft² or 315.4 W/m² to 630.8 W/m². Owing to the dilute nature of solar radiation, large collection areas are required, and the initial cost of installing solar heating and cooling equipment is larger than for other contemporary fuels.

However, if it is properly designed and constructed, solar heating and cooling of homes and buildings may be, at current power costs, less expensive than electrical climate control.

2.3 HISTORICAL PERSPECTIVE (1)

The first person to use the sun's energy on a large scale was Archimedes, who reputedly set fire to an attacking Roman fleet at Syracuse, in 212 B.C. He accomplished this "by means of a burning glass composed of small square mirrors moving every way upon hinges... so as to reduce it (the Roman fleet) to ashes at the distance of a bowshot. Serious studies of the sun and its potential began in the seventeenth century-when Galileo and Lavoisier utilized the sun in their researches. By 1700 diamonds had been melted and by the early 1800's heat engines were operating with energy supplied by the sun. In the early twentieth century solar energy was used to power water distillation plants in Chile and irrigation pumps in Egypt. By 1930 Robert Goddard had applied for five patents on various solar devices to be used on this project of sending a rocket to moon. Most of these project, however, were considered curiosities because they were ahead of their time.

By the 1920s and 1930s practical use was being made of the sun's energy in California-solar service-hot-water heaters. Devices similar to solar water heaters were also used to heat buildings in the United States. The first building to be practically heated with converted solar service-hot-water heaters was constructed at the Massachusetts Institute

of Technology in 1938. Some twenty other experimental building heating projects-an average of about one per year-were completed between 1938 and 1960. Performance data, recorded for a number of these projects, are used in solar-heated building design today.

The space age gave solar energy its first post-war boost. The success of solar cells in powering NASA service modules in terrestrial orbit and in lunar excursions led some engineers to propose other uses for solar energy in the space program. Solar-thermal power plants were designed and prototypes built; other exotic devices were proposed-some even proposed use of the small pressure exerted by sunlight to propel interplanetary vehicles. Although none of these projects was directed at providing large scale, terrestrial solar conversion economically, some of this NASA technology has found use in practical, applied project fund economical solar heating and cooling in 1970s.

III. FUNDAMENTALS OF HEAT TRANSFER

To estimate the size, the efficiency, and the cost of equipment necessary to transfer a specified amount of heat in a given time, a heat-transfer analysis must be made. The dimensions of a solar collector, a heat exchanger, or a refrigerator depend not so much on the amount of heat to be transferred under given external conditions. From an engineering viewpoint the determination of the rate of heat transfer at a specified temperature difference is the key problem in sizing a solar collector to provide a given temperature in a home or building.

Heat is transmitted by three distinct modes:

conduction, or heat transfer by molecular scale vibration and rotation; radiation, or heat transfer by photons; convection, or heat transfer by large-scale fluid motion. In nature, heat usually flows not by one but several mechanisms acting simultaneously; however, the three modes of heat transfer are initially analyzed separately in the discussion that follows.

3.1 CONDUCTION

Conduction is a process by which heat flows from a region of higher temperature to a region of lower temperature within a solid, liquid, or gaseous medium or between different media in direct physical contact. Of the three processes, conduction is the only mechanism by which heat can flow in opaque solids. The basic relation for heat transfer by conduction states that the rate of heat flow (flux) by conduction q_k in a material is equal to the product of the following three quantities:

- 1- k , the thermal conductivity of material, in Btu/hrft²°K or W/m²°K
 - 2- A , the area of the medium e.g., a building wall, through which heat flows by conduction, in ft² or m²
 - 3- dT/dx , the temperature gradient through the medium, in °F/ft or °C/m
- In equation form we have,

$$q_k = - k A \frac{dT}{dx} \quad (3.1)$$

Equation (3.1) can be integrated for heat flow through a wall of thickness b having a temperature T_1 over one surface and T_2 over the other to yield the relation.

$$q_k = \frac{kA}{b} (T_1 - T_2) \quad (3.2)$$

In equation (3.2) the temperature difference between the higher temperature T_1 and the lower temperature T_2 is the driving potential which causes the flow of heat. The value b/kA is equivalent to a thermal "resistance" R_k the wall offers to the flow of heat by conduction, and therefore

$$q_k = \frac{T_1 - T_2}{R_k} \quad (3.3)$$

The subscript k indicates the transfer mechanism is conduction. The concept of resistance will be found helpful in the analysis of thermal systems where several modes of heat transfer occur simultaneously.

3.2 RADIATION

Radiation is a process by which heat flows from a body at higher temperature to a body at a lower temperature when the bodies are separated in space or even when a vacuum exists between them. The term radiation is generally applied to all kinds of electromagnetic wave phenomena, but in heat transfer only those phenomena that are the result of temperature and can transport energy through a medium such as air or space are of interest. The energy transmitted in this fashion is called radiant heat. Radiation is the mode of heat transfer by which the sun transfers energy to the earth.

A perfect radiator, as so-called black body, emits radiant energy from its surface at a rate q_r given by

$$q_r = A \sigma T^4 \quad (3.4)$$

When two bodies exchange heat by radiation, the heat exchange is then proportional to the difference in T^4 . Thus,

$$q_r = \sigma A (T_1^4 - T_2^4) \quad (3.5)$$

where σ is the proportionality constant and is called the Stefan-Boltzmann constant with the value of $5.669 \times 10^{-8} \text{ w/m}^2 \text{ K}^4$.

Equation (3.5) is called the Stefan-Boltzmann law of thermal radiation, and it applies only to black bodies. It is important to note that this equation is valid only for thermal radiation; other types of electromagnetic radiation may not be treated so simply.

3.3 CONVECTION

Convection is a process that transfers heat from one region to another by motion of a liquid. The rate of heat transferred by convection q_c between a surface and a fluid can be calculated from the relation.

$$q_c = h_c A (T_s - T_f) \quad (3.6)$$

where q_c = rate of heat flow by convection, W

A = base area of heat transfer by convection, m^2

T_s = surface temperature, $^{\circ}\text{C}$

T_f = fluid temperature, $^{\circ}\text{C}$

h_c = convection heat-transfer coefficient, W/m^2

The thermal resistance to convection heat transfer R_c is given by

$$R_c = \frac{1}{h_c A} \quad (3.7)$$

3.4 COMBINED HEAT TRANSFER MECHANISMS

In most engineering situations, heat is transferred by two or three of the mechanisms acting simultaneously. For example, the surface of a solar-energy collector loses heat simultaneously by convection as well as by radiation to environment. For heat loss through a combination of radiation and convection, the total rate of heat transfer q from the surface of the collector is given by the equation

$$q = q_r + q_c = \frac{T_{\text{coll}} - T_{\text{out}}}{R_{\text{cr}}} \quad (3.8)$$

where R_{cr} = combined resistance for the two mechanisms, convection and radiation acting in parallel

T_{coll} = average collector-surface temperature

T_{out} = outside air temperature

IV. METHODS OF SOLAR ENERGY COLLECTION AND USE (2)

4.1 CONVERSION OF SOLAR ENERGY TO HEAT

Solar energy is transmitted from the sun through space to earth by electromagnetic radiation. It must be converted to heat before it can be used in practical heating or cooling systems. Since solar energy is relatively dilute when it reaches the earth, the size of a system to convert it to heat on a practical scale must be relatively large. Solar-energy collectors, the devices used to convert the sun's radiation to heat, usually consist of a surface that efficiently absorbs radiation and converts this incident flux to heat which raises the temperature of the absorbing material. A part of this energy is then removed from the absorbing surface by means of a heat-transfer fluid that may be either liquid or gaseous.

Since 1900 at least 20 solar-collector designs have been demonstrated as functional. These designs are separated into two generic classes: concentrating and nonconcentrating. Nonconcentrating, or flat plate collectors intercept solar radiation on a metal or glass absorber

plate from which heat is transferred and used in the required thermal application. Since the absorber plate has a temperature greater than its environment, unrecoverable heat losses occur from the entire absorbing surface of the collector to the environment. Consequently, 100 percent collector efficiency can not be realized in practice.

Concentrating collectors attempt to reduce heat loss by using an absorber plate area smaller than the area that intercepts the sun's rays, called the aperture area. This performance improvement is accomplished by reflecting the sun's rays from the large aperture area to the small absorbing area by shaped mirrors or other reflecting surfaces. Since only the direct or collimated portion of solar radiation is amenable to effective radiation, most concentrators must move to track the sun and can not collect as much as diffuse radiation as flat plate collectors. These two negative aspects of concentrators can be offsets by their greater efficiency, particularly in sunny regions of the world.

Since solar energy is available only during daylight hours and during periods when the sun is not significantly obscured by clouds, a means of providing heat on a continuous basis from this intermittent source is required. In nearly all applications a form of thermal energy storage is used for this purpose. Three practical storage media have shown acceptable performance. Sensible heat storage (storage in a material by virtue of a temperature rise) in pebble or rock beds and water have been the two most widely used means of storage; the first is used with air-cooled solar conversion devices, the second, with liquid-cooled devices.

The third storage method is by the use of latent heat of materials undergoing a phase change. Such materials are capable of larger energy-storage per unit volume than are sensible heat storage but to data have shown unacceptable reliability in repeated freeze-thaw cycles. Since the heating and cooling loads on a building do not occur in phase with useful solar collection, all practical, solar building thermal systems require storage.

4.2 FLAT PLATE COLLECTOR FUNDAMENTAL

Method of solar-energy collection range from active, mechanical, solar thermal collectors to photoelectrical devices to windmills to fully passive collection systems. In this section, primary emphasis is placed on the performance characteristics of solar thermal collectors that can be used for building heating and cooling.

A solar collector is a special kind of heat exchanger that transforms solar radiant energy into heat. A solar collector differs in several respects from more conventional heat exchangers. The latter usually accomplish a fluid-to-fluid exchange with high heat transfer rates and with radiation as an unimportant factor. In the solar collector, energy transfer is from a distant source of radiant energy to fluid. Without optical concentration, the flux of incident radiation is, at best, approximately 1100 W/m^2 and variable. The wave length range is from 0.29 to $2.5 \mu\text{m}$, which is considerably shorter than that of the emitted radiation from most energy-absorbing surfaces. Thus, the analysis of solar collectors presented unique problems of low and variable

energy fluxes and the relatively large importance of radiation.

Flat-plate collectors can be designed for applications requiring energy delivery at moderate temperatures, up to perhaps 100°C above ambient temperature. They use both beam and diffuse solar radiation, do not require tracking of the sun, and they require little maintenance. They are mechanically simpler than concentrating collectors. The major applications of these units currently are in solar water heating and building heating, whereas potential uses include building air conditioning and industrial process heat. Passively heated buildings can be viewed as special cases of flat-plate collectors with the room or storage wall as the absorber.

4.3 GENERAL DESCRIPTION OF FLAT PLATE COLLECTORS

The important parts of a typical liquid heating flat-plate solar collector, as shown in Figure 4.1 are the "black" solar energy-absorbing surface, with means for transferring the absorbed energy to a fluid; envelopes transparent to solar radiation over the solar absorber surface that reduce convection and radiation losses to the atmosphere; and back insulation reduce conduction losses as the geometry of the system permits. Figure 4.1 depicts a water heater and air heater are fundamentally the same except that the fluid tubes are replaced by a duct.

Flat-plate collectors are almost always mounted in a stationary position (e.g. as an integral part of a wall or roof structure) with an orientation optimized for the particular location in question for the time of year in which the solar device is intended to operate.

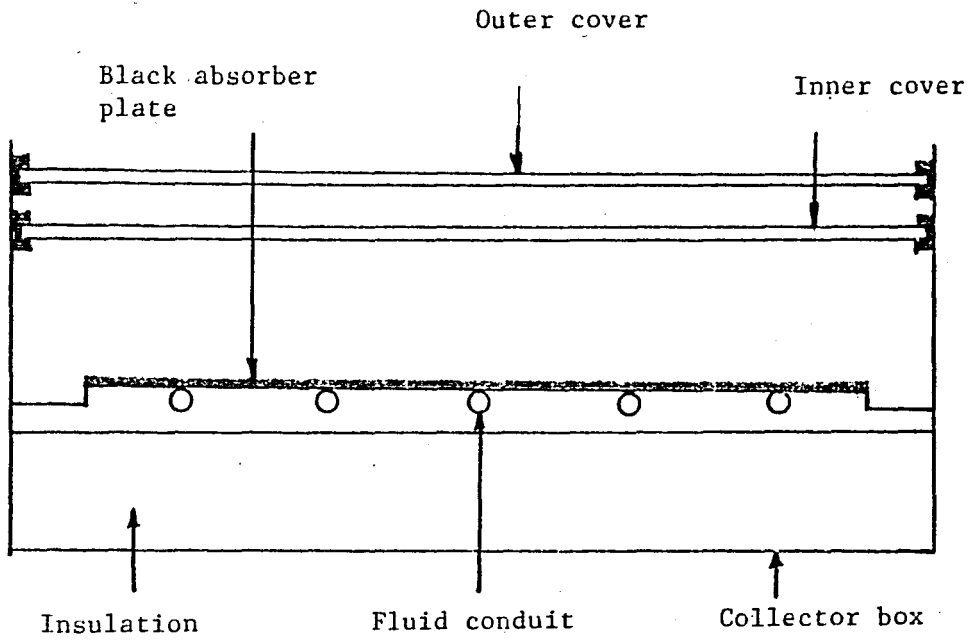


Figure 4.1 Cross-section of a basic flat plate solar collector

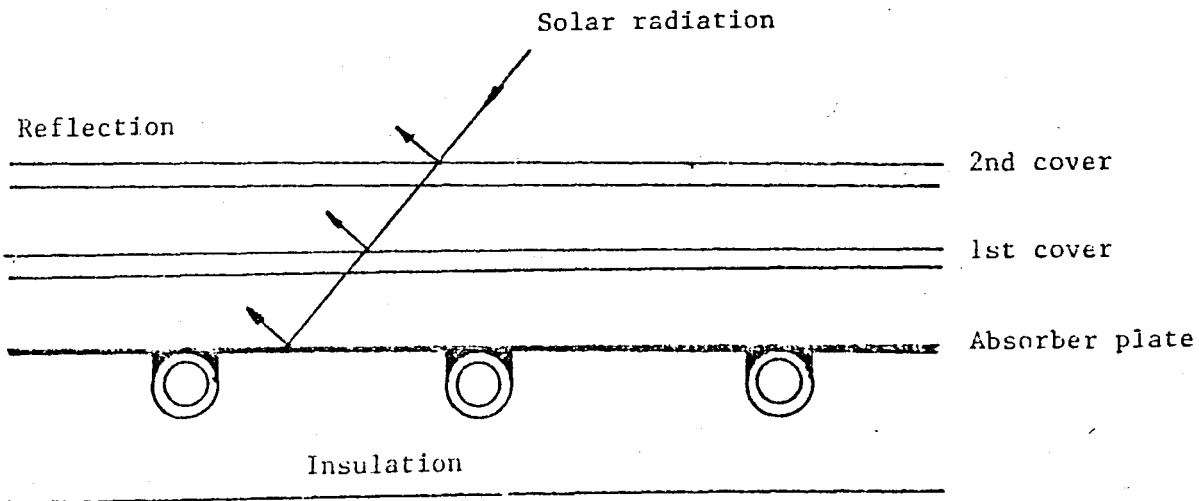


Figure 4.2 Sheet and tube solar collector

In their most common forms, they are air or liquid heaters or low-pressure steam generators.

4.4 THE BASIC FLAT - PLATE ENERGY BALANCE EQUATION

In steady state, the performance of a solar collector is described by an energy balance that indicates the distribution of incident solar energy into useful energy gain, thermal losses, and optical losses. The solar radiation absorbed by a collector, S , is equal to the difference between the incident solar radiation and the optical losses and is defined by equation

$$S = I_b R_b (TA)_b + I_d (TA)_d \frac{(1 + \cos \phi)}{2} + g_r (I_b + I_d) (TA)_g \frac{(1 - \cos \phi)}{2} \quad (4.1)$$

where,

I = Insolation for an hour (J/m^2)

R = The ratio of radiation on the tilted surface to that on a horizontal surface at any time.

TA = Transmittance - absorptance product

g_r = Ground reflectance

ϕ = Slope, that is, the angle between the plane surface in question and the horizontal $0 \leq \phi \leq 180$ ($\phi > 90^\circ$ means that the surface has a downward facing component). This angle can be shortly named collector tilt angle ($1 + \cos \phi$) / 2 and ($1 - \cos \phi$) / 2 are view factors from the collector to the ground, respectively. The subscripts b, d and g represent beam, diffuse and ground. The thermal energy lost

from the collector to the surroundings by conduction, convection, and infrared radiation can be represented by a heat transfer coefficient U_L , times the difference between the mean absorber plate temperature, $T_{p,m}$ and the ambient temperature, T_a . In steady state the useful energy output of a collector is then the difference between the absorbed solar radiation and the thermal loss.

$$Q_u = A_c (S - U_L (T_{p,m} - T_a)) \quad (4.2)$$

where

A_c = Collector area

The problem with this equation is that the mean absorber plate temperature is difficult to calculate or measure since it is a function of the collector design, the incident solar radiation, and the entering fluid conditions. So that the useful energy gain can be expressed in terms of the inlet fluid temperature and a parameter, called the collector heat removal factor, which can be evaluated analytically from basic principles or measured experimentally.

Equation (4.2) is an energy rate equation and in SI units, yields the useful energy gain in watts when S is expressed in W/m^2 and U_L in $W/m^2 \cdot ^\circ C$. Often, the most convenient time base for solar radiation is hours rather than seconds since this is the normal period of reporting meteorological data. S can be considered as an average energy rate over a one-hour period with units of $J/m^2 \cdot hr$ in which case the thermal loss term, $U_L (T_{p,m} - T_a)$ must be multiplied by 3600s/hr to obtain numerical values of the useful energy gain in J/hr.

Alternatively, Equation (4.2) can be integrated over a one-hour period. Since we seldom have data over time periods less than one hour, this integration can be performed only by assuming that S , $T_{p,m}$ and T_a remain constant over the hour.

A measure of collector performance is the collection efficiency, defined as the ratio of the useful gain over the some specified time period to the incident solar energy over the same time period. (2)

$$\eta = \frac{Q_u dt}{A_c \int G_T dt} \quad (4.3)$$

where

G_T = Total instantaneous solar radiation (W/m^2)

Q_u = Useful energy output of a collector (W)

The design of a solar energy system is concerned with obtaining minimum cost energy. Thus, it may be desirable to design a collector with an efficiency lower than is technologically possible if the cost is significantly reduced.

4.5 TEMPERATURE DISTRIBUTIONS IN FLAT-PLATE SOLAR COLLECTORS

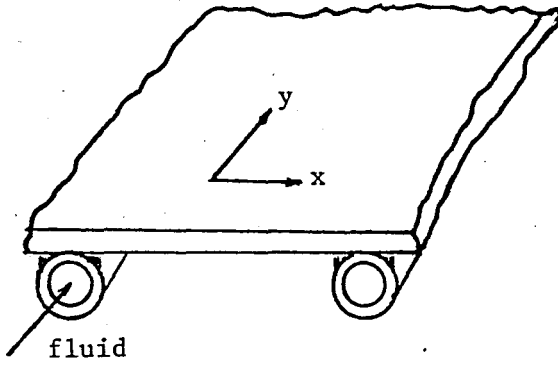
The detailed analysis of a solar collector is a complicated problem. Fortunately, a relatively simple analysis will yield very useful results. These results show the important variables, how they are related, and how they affect the performance of a solar collector. To illustrate these basic principles, a liquid heating collector, as

shown in Figure 4.2 will be examined first. The analysis presented follows the basic derivation by Whillier(1953,1977) and Hottel and Whillier(1958)

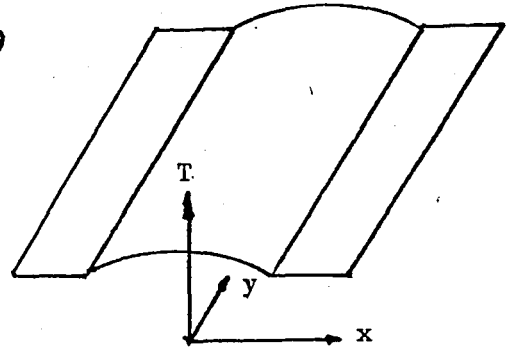
To appreciate the development that follows, it is desirable to have an understanding of the temperature distribution that exists in a solar collector constructed as shown in Figure 4.2. It shows a region between two tubes. Some of the solar energy absorbed by the plate must be conducted along the plate to the region of the tubes. Thus, the temperature midway between the tubes will be higher than the temperature in the vicinity of the tubes. The temperature above the tubes will be nearly uniform because of the presence of the tube and weld metal.

The energy transferred to the fluid will heat up the fluid causing a temperature gradient to exist in the direction of flow. Since in any region of the collector the general temperature level is governed by the local temperature level of the fluid, a situation as shown in Figure 4.3.b is expected. At any location y , the general temperature distribution in the x direction is as shown in Figure 4.3.c, and at any location x , the temperature distribution in the y direction will look like Figure 4.3.d (2).

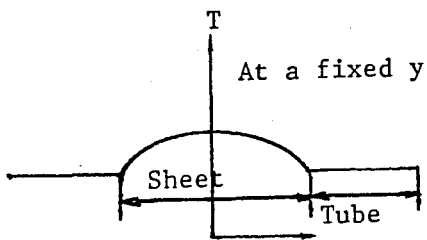
To model the situation shown in Figure 4.3 a number of simplifying assumptions can be made to lay the foundations without obscuring the basic physical situation. These important assumptions are as follows:



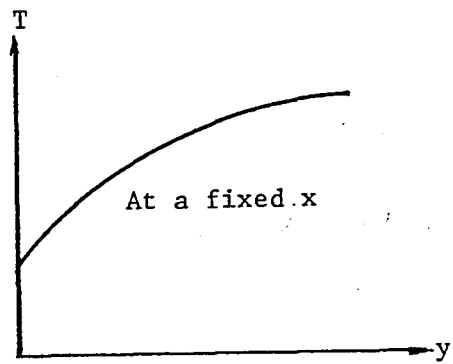
(a)



(b)



(c)



(d)

Figure 4.3 Temperature distribution on an absorber plate

- 1- Performance is steady-state.
- 2- Construction is of sheet and parallel tube type.
- 3- The headers cover a small area of collector and can be neglected.
- 4- The headers provide uniform flow to tubes.
- 5- There is no absorption of solar energy by covers insofar as it affects losses from the collector.
- 6- There is one-dimensional heat flow through covers.
- 7- There is a negligible temperature drop through a cover.
- 8- The covers are opaque to infrared radiation.
- 9- There is one-dimensional heat flow through back insulation.
- 10- The sky can be considered as a black body for long-wavelength radiation at an equivalent sky temperature.
- 11- Temperature gradients around tubes can be neglected.
- 12- The temperature gradients in the direction of flow and between the tubes can be treated independently.
- 13- Properties are independent of temperature.
- 14- Loss through front and back are to the same ambient temperature.
- 15- Dust and dirt on the collector are negligible.
- 16- Shading of the collector absorber plate is negligible.

V. SOLAR COOLING

5.1 SOLAR COOLING OF BUILDINGS

Solar cooling of buildings represents a potentially significant application of solar energy for building air conditioning in most sunny regions of Turkey. However, solar cooling technology is presently not as advanced as solar-heating technology. It is expected that research works will close the gap between two by 1980.

Several of all the possible applications of solar energy, its use for cooling seems the most fitting. This is one of the few uses in which the supply of energy and the demand for it are closely matched. In the distillation of water for drinking and the pumping of water for irrigation, the greatest need also occurs during periods of greatest insolation, but with cooling requirements, the matching extends in many situations even to the variation within the day.

In developed countries, a society has grown up that is largely

urban and is sustained by an industry having high productivity, based on the intensive use of energy. The refrigeration of foodstuffs and many other commodities is a normal part of its commercial and domestic life. Air conditioning of homes, factories and offices has become sufficiently extensive to have disturbed the usual pattern of demand for electricity, causing the peaks to occur on summer afternoons and evenings, and even at weekends. Cooling is thus beginning to appear as a significant element in the energy economy. The need to conserve fuels in the face of their rapid depletion has come a new interest in the utilization of solar energy, with the prospect of solar cooling tempting prize.

5.2 DESIGN OF A COLLECTOR FOR A SOLAR-COOLED BUILDING

Solar energy can be an economical source of energy for building temperature control in areas where the local climate requires both air conditioning and heating, because use of solar energy for heating and air conditioning increases the system usage factor. The combined system shown in Figure 5.1 is presently installed in the Solar Energy Application Laboratory, Colorado State University, Fort Collins, Colo. The same solar collector and storage system can be used for combined, year-round heating and cooling operation. Löf and Tybout performed an optimization study of combined systems for the same geographical locations as their solar-heating study. The general conclusions pertinent to collector design for combined systems used in climates with hot summers are

++++ Heating
 ---- Air conditioning

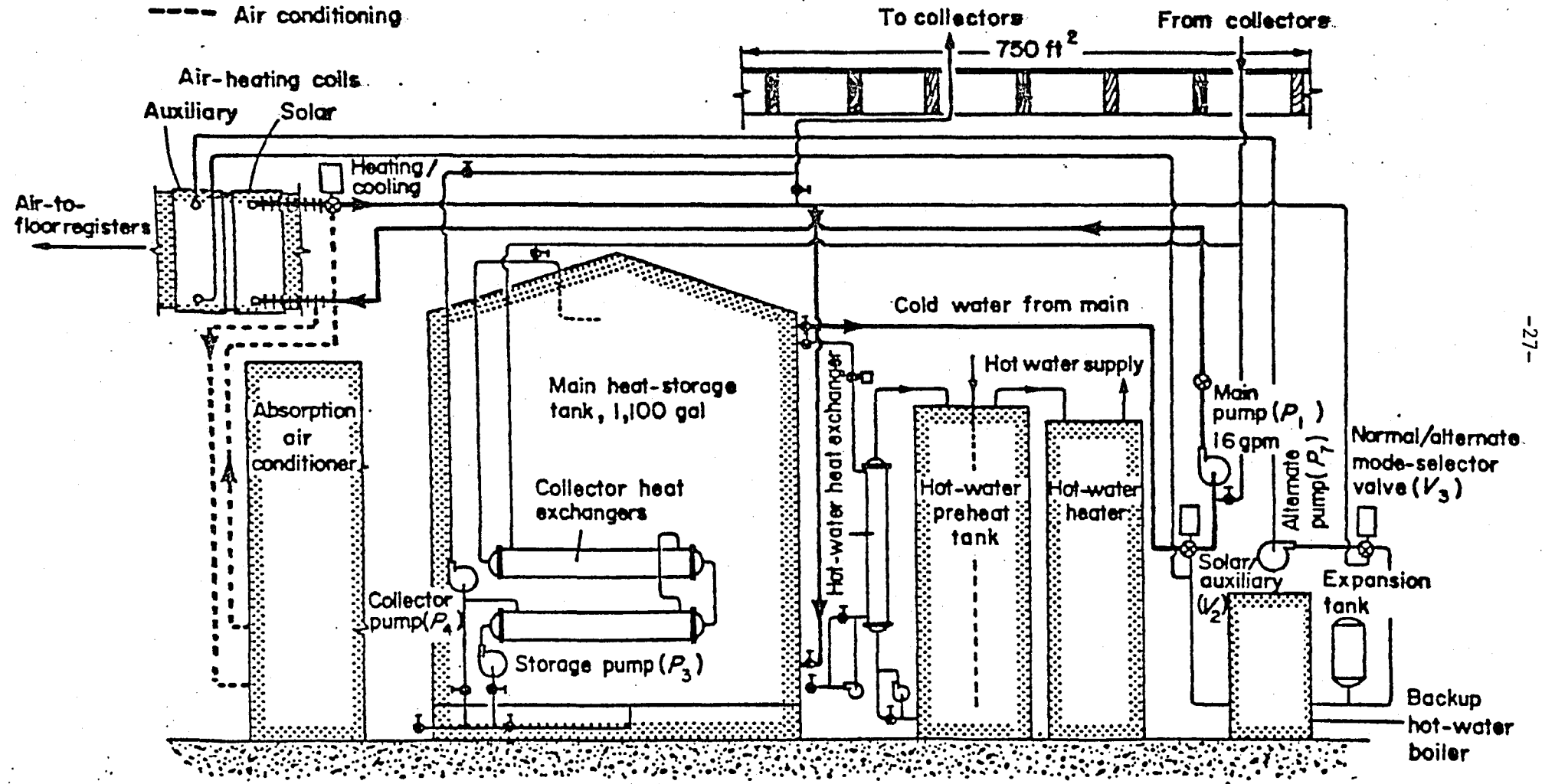


Figure 5.1 Combined solar heating and cooling system at the Solar Energy Applications Laboratory, Colorado State University, Fort Collins (Courtesy of G.O.Löf)

- 1- Optimal collector tilt ϕ is equal to latitude λ except in the South where $\phi = \lambda - 10^\circ$
- 2- Optimal collector area is always greater than needed for heating only.
- 3- Three glass covers are desirable due to higher temperature needs of air conditioning systems; in subtropical steppe climate, two are optimal. (1)

5.3 STORAGE OF ENERGY AT HIGH AND LOW TEMPERATURES

In an air-conditioning system it is also possible to cool a tank of water or other storage medium if excess cooling capacity is available during operational hours. One of the attractions of this approach is that the temperature difference between cold storage (say, 7°C at the coldest) and building temperature ($\sim 21^\circ\text{C}$) is less than the temperature difference between hot-storage and room temperature. As a result, less cooling effect is lost from cold storage than from hot storage. Moreover, cold storage is preferable to hot storage because losses from hot storage in a building add to the summer air-conditioning load. (1)

In addition, 1 Btu of cold storage is 1 Btu of cooling; 1 Btu of hot storage is equivalent to less than 1 Btu of cooling since the air conditioner is not 100 percent efficient. These advantages of cold storage are offset by the additional cost and upkeep required to install and maintain a second storage unit and the associated plumbing and ducting.

The decision to use hot, cold, or combined storage depends on the building location. In a climate where heating loads are small (only 1,500 degree-days per year) cold storage should be designed since the investment in hot storage is uneconomical. On the other hand, in a climate where cooling loads are small hot storage should be designed. It should be noted that the hot-storage tank used for winter heating can be used for summer cold storage by a proper valve arrangement.

5.4 VAPOR - COMPRESSION SYSTEMS AND HEAT PUMPS

5.4.1 INTRODUCTION

The periodicity and intermittence of solar energy incident on a collector require the use of a conventional backup (auxiliary) system. The size and cost of the total system depend not only on the Btu's collected but also on storage facilities; any concept that can reduce the collection and storage requirements can improve the economics of solar-energy use. One attractive method to achieve a fuller use factor is to utilize a solar-assisted heat pump-air conditioner for all-year climate control. From an energy standpoint the efficiency of a conventional heat-pump system operating in northern climate is reduced as a result of low ambient temperatures; owing to the reduced efficiency of the heat pump, reliance of the system of electric-resistance heating is necessary. However, solar energy can be used at low ambient temperatures to reduce or often eliminate the need for supplemental resistance heating in such systems.

A heat pump may be defined as a system that absorbs energy at low temperature and delivers energy at higher energy through a vapor-compression or absorption cycle. However, no absorption-cycle heat pumps are presently available. Available heat-pumps use a standard vapor-compression refrigeration cycle operated in reverse: the evaporator is usually placed outdoors and the condenser indoors. The mechanical energy input to the compressor during vapor compression raises the internal energy absorbed at low ambient temperatures to a temperature level useful for space heating. The quantity of heat extracted from such a system can be several times larger than the energy required by the compressor. The basic advantage of a heat pump is this "heating multiplication".

In a solar-assisted heat-pump designed both to heat and to cool a building, two general modes of operation are possible. As shown in Figure 5.2.a, the solar heating system and the heat-pump system may be separate. In Figure 5.2.b the two cycles are shown connected, and a solar collector assists in reducing the amount of heat that must be supplied by the compressor when heating the building. Obviously, the second arrangement is more efficient. (1)

5.4.2 SOLAR POWERED HEAT PUMP CYCLE (1)

The heat pump can theoretically be used with any thermodynamic cooling cycle, such as an absorption, a jet compression, and a mechanical vapor compression cycle. Fig 5.3 illustrates the basic heat flows in a conceptual solar-powered heat pump system in which the sun's energy directly provides shaft work instead of acting as a

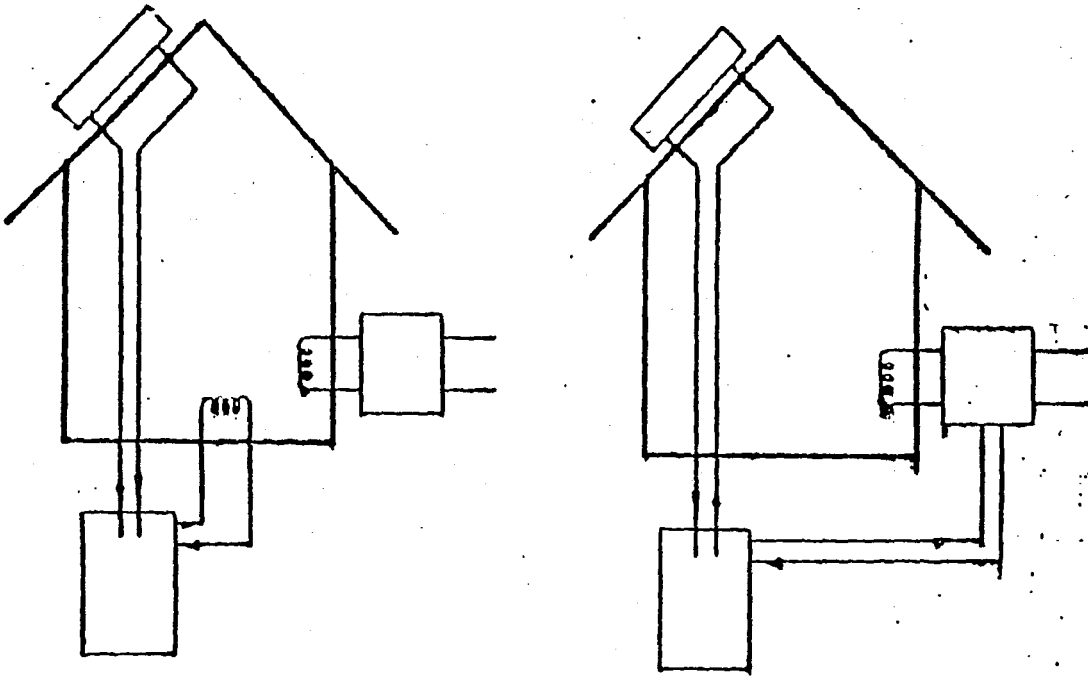


Figure 5.2 (a) Direct solar heating system
 (b) Solar-assisted heat pump

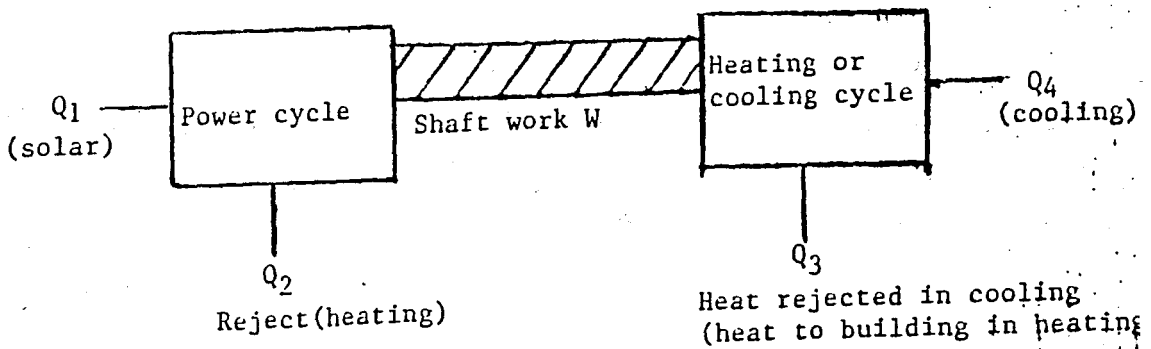


Figure 5.3 Basic solar-powered heat-pump system

booster as in Fig 5.3. Q_4 is the amount of heat removed from the environment during the heating mode. A heat balance of the cycle in Fig 5.3 gives,

$$Q_1 + Q_4 = Q_2 + Q_3 \quad (5.1)$$

where Q_1 = solar heat input

Q_2 = heat rejected in the power cycle

Q_3 = heat rejected in the cooling cycle

The efficiency in cooling, i.e., the heat transferred from a building divided by the energy required to drive the system, is called the coefficient of performance (COP) and is given as,

$$\text{COP} = \frac{\text{Cooling effect}}{\text{Heat input}} = \frac{Q_4}{Q_1} \quad (5.2)$$

When the system is acting as a heat pump to heat a building, Q_2 and Q_3 , the so-called heat rejected are rejected into the building and serve to heat the interior. When the system is acting as an air conditioner, the refrigerant flow is reversed, and heat is removed from the building and rejected to the environment.

5.5 MECHANICAL REFRIGERATOR SYSTEMS

5.5.1 IDEAL REFRIGERATOR

The ideal refrigerator is a reversed heat engine with reversible heat and work transfer. For the expenditure of work W , heat Q_1 is withdrawn from the cold space at temperature T_1 ,

and Q_2 is rejected into the hot space at temperature T_2 . The effectiveness of the device is defined by the coefficient of performance COP, where

$$\text{COP} = \frac{Q_1}{W} \quad (5.3)$$

As there is no net change energy

$$W + Q_1 - Q_2 = 0 \quad (5.4)$$

and there is no net change of entropy and the heat transfers are reversible

$$\frac{Q_1}{T_1} - \frac{Q_2}{T_2} = 0 \quad (5.5)$$

To give chilled water in the range of temperature required, the cold space temperature might have to be 3 °C, say, so that the heat exchange area would not be too large. The warm space temperature will be determined by that of the heat sink available. For a pond or cooling tower sink, the temperature might be 30 °C, and for an ambient sink, perhaps 50 °C. For these conditions, the COP is found to be 10.2 and 5.87, respectively. (3)

If the machine had been driven by an ideal heat engine operating from a source at temperature T_h and rejecting heat at the same temperature T_2 as the refrigerator, the work W would be

obtained by the supply of heat Q_h given by

$$\frac{W}{Q_h} = \frac{T_h - T_2}{T_h} \quad (5.6)$$

Then the thermal ratio (TR) for the combined engine/refrigerator is

$$TR = \frac{Q_1}{Q_h} = \frac{T_1 (T_h - T_2)}{T_h (T_2 - T_1)} \quad (5.7)$$

Although these figures are for ideal systems, they illustrate clearly the central problem in the effective use of solar energy for cooling. A refrigerator requires little power to drive it, but if this power is provided by a solar engine, the overall thermal ratio is poor except when the input can be provided at high temperature.

5.5.2 VAPOR-COMPRESSION REFRIGERATOR

Many practical refrigeration machines operate on a simple vapor-compression cycle. Heat is extracted from the cold space to evaporate the refrigerant fluid at low pressure. A compressor provides the pressure rise between evaporator and condensor, but the drop is obtained in a throttle valve or capillary tube. Departure of the T-S diagram from the rectangular form of the ideal cycle indicates that the COP will be somewhat lowered. It may usefully be expressed in terms of the enthalpy at extremities of the cycle.

$$\text{COP} = \frac{h_1 - h_4}{h_2 - h_1} \quad (5.8)$$

In practice, the value obtained from this equation will usually be between 0.75 and 0.85, the factor depending largely upon the properties of the refrigerant fluid used. In many respects, water is a good refrigerant, but it has very low vapor pressure (0.0076 bar or 0.76 kN/m^2 at 3°C) so that the system has to be large and physically robust and is difficult to keep free from leaks. Moreover, temperatures below 0°C can not be obtained. Many machines use one of the fluorinated hydrocarbons often called by the trade name Freon 12 or Freon 22. Whatever fluid is employed, the overall COP based on the work supplied to the machine is further reduced by the mechanical efficiency of the compressor and unavoidable heat transfers to and from the surroundings. For planning purposes, a value of about 50 % of the Carnot COP may usually be assumed. (3)

VI. ABSORPTION REFRIGERATION

6.1 ELEMENTARY PROPERTIES OF BINARY MIXTURES

Before discussing absorption refrigeration cycles, some fundamental characteristics of binary mixtures will be studied. Mixtures are formed by combining two or more pure substances. A mixture may be classified as either homogeneous or heterogeneous. A homogeneous mixture is uniform in composition, regardless of how small the particles are. The various properties such as density, pressure and temperature are uniform throughout the mixture. A homogeneous mixture can not be separated into its constituents by pure mechanical means such as settling or centrifuging. Practically all gaseous mixtures are homogeneous because it is the nature of gases to diffuse into each other. An example is moist air which is a homogeneous mixture of dry air and water vapor.

A heterogeneous mixture is non-uniform in composition. It can be separated by ordinary mechanical means. An example of a heterogeneous mixture is a fog or cloud, which is a mixture of liquid water and saturated air.

In general, substances may be combined to form heterogeneous or homogeneous mixtures in solid, liquid, or vapor phases. This study will be concerned only with homogeneous liquid and homogeneous vapor mixtures, and it will be limited to a study of binary mixtures.

The thermodynamic state of a mixture can not be established by pressure and temperature alone as may be done with a pure substance. We will consider the quantitative composition in terms of the concentration X which is the mass of one arbitrary constituent divided by the mass of the mixture. Knowledge of pressure p , temperature T , and X enables one to

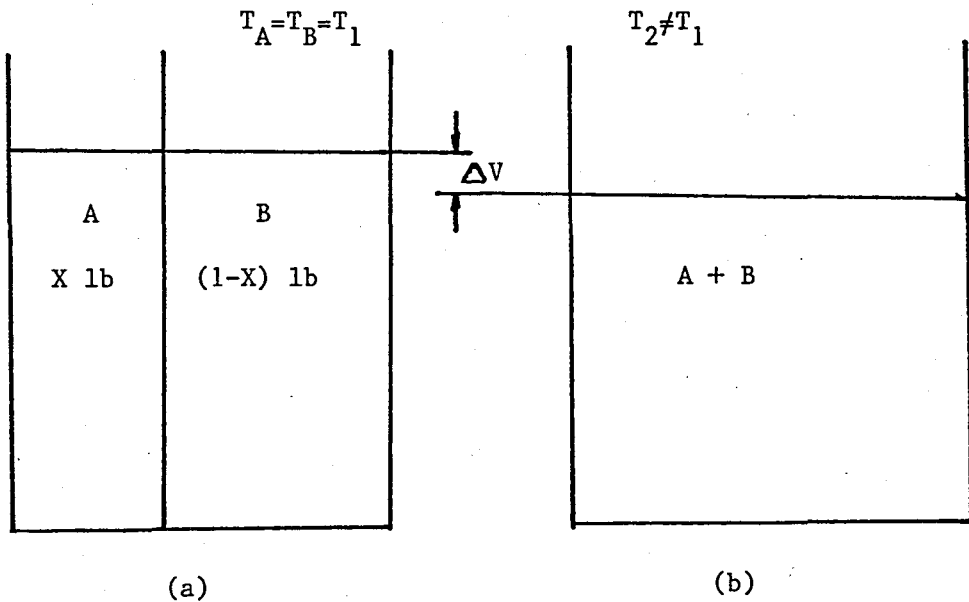


Figure 6.1 Schematic illustration of change of volume and change of temperature in the mixing of two liquids.

establish the thermodynamic state of the mixture.

An important characteristics of a mixture is its miscibility. A mixture is miscible if through any arbitrary range of concentration values, a homogeneous mixture is formed. Thus, a non-miscible mixture

is a heterogeneous mixture. Some mixtures are miscible under certain conditions but otherwise non-miscible. Miscibility is materially influenced by temperature.

Binary mixtures suitable for an absorption system must be completely miscible in both the liquid and vapor phases. There can be no miscibility gap or range of concentration values where a heterogeneous mixtures would exist.

Two important phenomena occurring with the mixing of two liquids are the change of volume and change of temperature of the constituents during mixing. Figure 6.1 shows schematically a divided container holding X pound of liquid A and $(1-X)$ pound of liquid B. Each liquid is at the same temperature T_1 . The volume of the kg of constituents is

$$V_1 = XV_A + (1 - X) V_B \quad (6.1)$$

It is assumed that the dividing wall is removed and the two liquids are thoroughly mixed. It is generally observed that $V_2 \neq V_1$. If the experiment was repeated with different liquids and with various concentration values, it would be found that, in some cases, contraction occurred. In other cases, ΔV would be positive. It would be found that no general rule was followed, and that experimentation was necessary to find the result.

Another important observation in the experiment of Figure 6.1 would be that, in general, $T_2 \neq T_1$. In some cases, it would be found that a warming effect occurred; in others, it might be observed a decrease in temperature. The effect may be expressed in terms of a

heat of solution ΔH_x . In the experiment it could be measured that the heat which had to be removed or added in order to keep the temperature constant. If the mixing were carried out at constant pressure, ΔH_x would be strictly related to the enthalpy of the mixture. For the original components,

$$h_1 = Xh_A + (1-X) h_B \quad (6.2)$$

and after mixing

$$h_2 = h_1 + \Delta H_x = Xh_A + (1-X) h_B + \Delta H_x \quad (6.3)$$

Equation (6.3) allows calculation of the specific enthalpy in Btu per pound of mixture for a solution of known concentration at some fixed temperature and pressure, if the enthalpies of the pure components and the isothermal heat of solution are known.

The pressure-temperature relationships for a boiling-liquid binary mixture or a condensing-vapor binary mixture are of especial importance in absorption refrigeration.

Figure 6.2 schematically shows the experiment and the observed results on T-X coordinates. It is started with a liquid solution (state 1) and slowly add heat to it, at all times keeping the pressure constant. (4) (It is imagined a glass cylinder so that the solution inside is viewed). Until the temperature T_2 is reached it is observed that the solution remains entirely liquid. However, upon further addition of heat, it is found that the piston rises above the liquid indicating the vaporization occurs. If the experiment

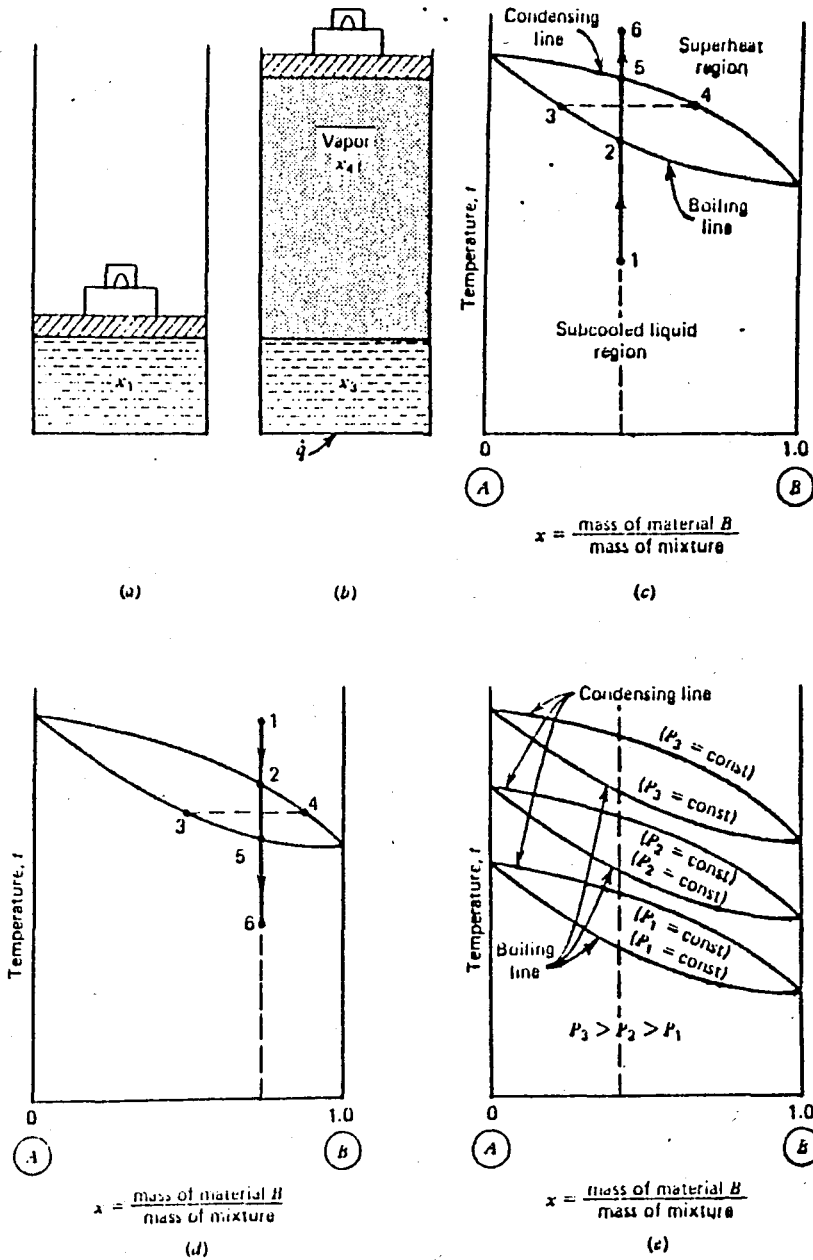


Figure 6.2 Evaporation and condensation characteristics for a homogeneous binary mixture

would be stopped at the condition shown in Fig 6.2.b and chemically analyze the vapor and liquid components, an interesting discovery would be different, and furthermore, both concentrations would be different from the original concentration X_1 . It would be found that the liquid concentration $X_3 < X_1$, while the vapor concentration $X_4 > X_1$. If more heat was added, it would be noticed that the liquid gradually disappear. It would also be noticed that both the liquid concentration and the vapor concentration had decreased. When the state-point (5) was reached, only vapor would remain, here it would be observed that $X_5 = X_1$. Further heating would result in superheating of the vapor at constant concentration.

If the experiment was repeated with different initial concentration values, but with the same pressure, experimental results would give the equilibrium-boiling and equilibrium-condensing lines shown in Figure 6.2.c. If the experiments were repeated with different pressures, the results shown in Figure 6.2.d would be obtained. If the experiments were reversed, started with a superheated vapor, and then heat was removed, the results shown in Figure 6.2.e would be obtained.

Thus, binary mixtures, as contrasted with pure substances, do not have a single boiling or condensing temperature for a given pressure. The equilibrium or saturation temperature is also dependent on the concentration. These relationships must be determined experimentally.

The enthalpy-concentration diagram (h - X coordinates) is the most useful diagram of properties for a binary including liquid and

vapor regions for a homogeneous binary mixture(Figure 6.3)

Enthalpy-concentration diagrams for two actual binary mixtures for ammonia-water and water-lithium bromide have been used in the design of many absorption-refrigeration systems.

Enthalpy-concentration diagram of ammonia-water combinations is drawn for saturated or equilibrium solutions of ammonia and water. The liquid region is the lower part of the diagram. Equilibrium liquid lines (boiling lines) are shown for various pressures. Isothermal lines are shown in the liquid region. If a liquid is known to be saturated, the state-point may be located on Figure 6.4 by the intersection of the temperature line with the saturated liquid line for the appropriate pressure. If the liquid solution is subcooled, the state-point may be located approximately if the temperature and concentration are known. (4) :

No temperature lines are shown in the vapor region of Figure 6.4. However, a saturated vapor state in equilibrium with a known saturated liquid state may be located through use of the equilibrium construction lines. If pressure and temperature are known a vertical line is drawn upward from the saturated liquid point to the equilibrium construction line for the same pressure from this intersection, a horizontal line is drawn to the saturated vapor line for the same pressure. This intersection is the vapor state-point. Furthermore, a straight line drawn from the saturated liquid state to saturated vapor state is an isotherm for the liquid-vapor region.

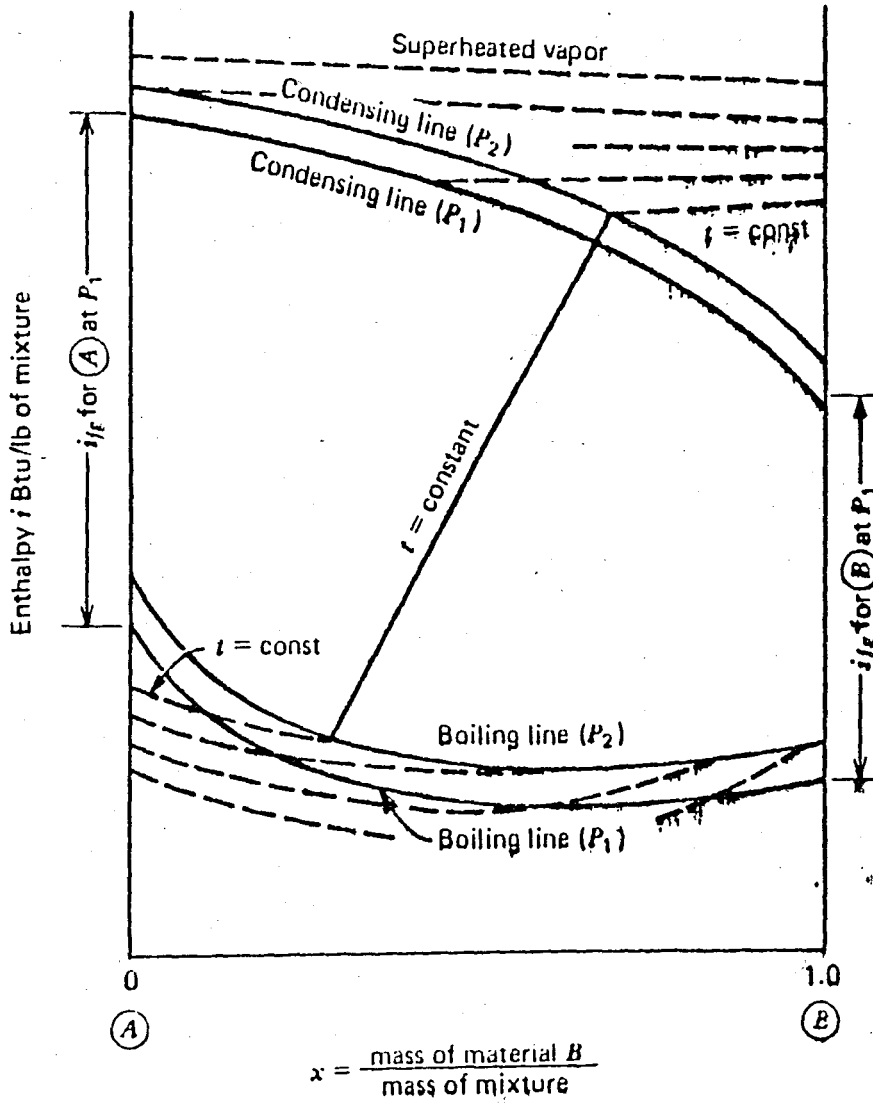
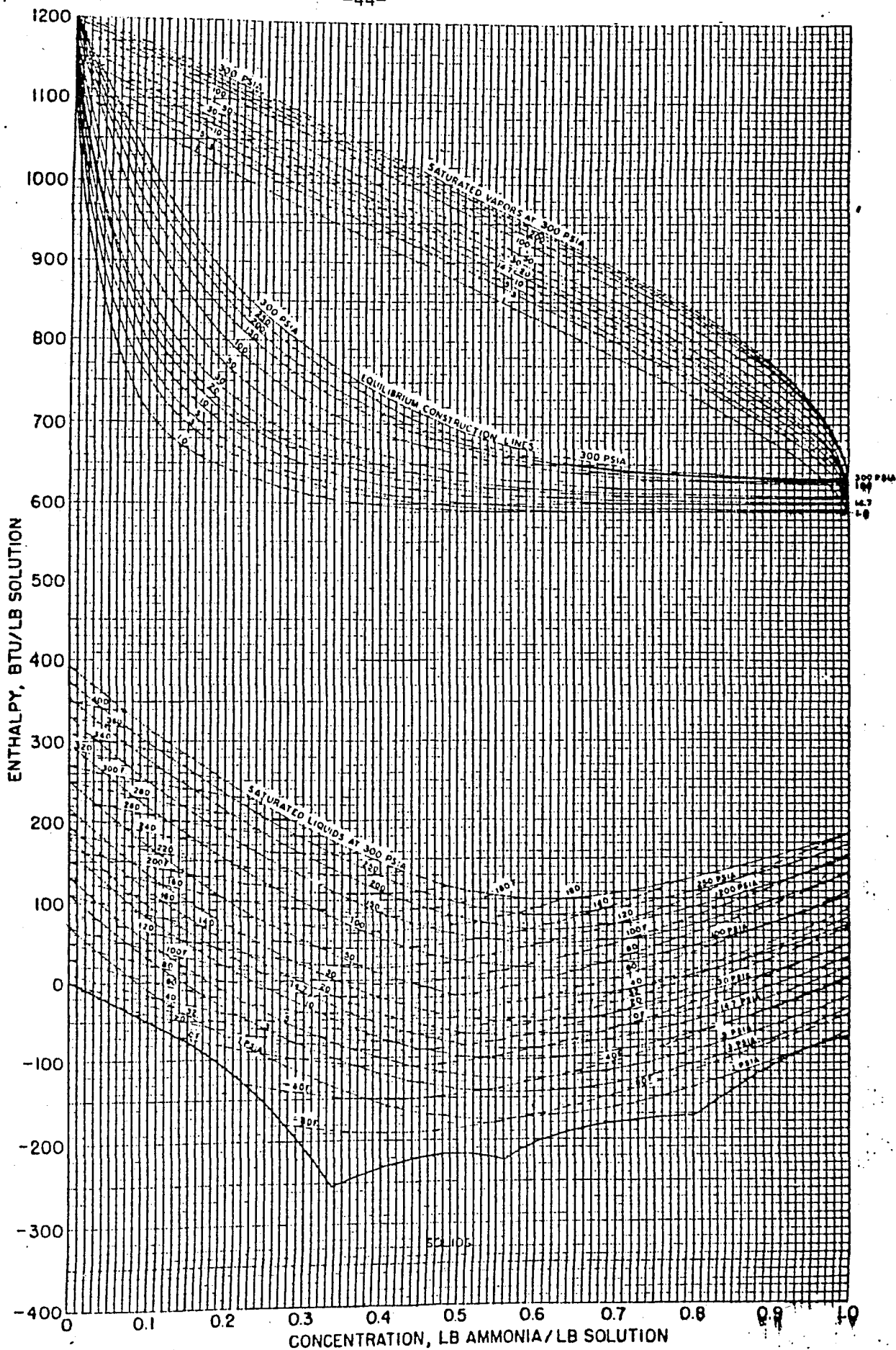


Figure 6.3 enthalpy(i)-concentration(x) diagram for liquid and vapor regions for a homogeneous binary mixture



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 Figure 6.4 h-x diagram for $\text{NH}_3\text{-H}_2\text{O}$ binary mixtures

6.2 ABSORPTION REFRIGERATION SYSTEM

The mechanical vapor compression refrigerant system is an efficient and practical method. However, the required energy is shaft work or power, which is high grade energy and expensive. The relatively large amount of work required is because of the compression of the vapor that undergoes a large decrease in volume.

It is possible to replace the vapor compression process with a series of processes where the refrigerant vapor is absorbed by a liquid and then the liquid solution is pumped to the higher pressure. Although the shaft work required is much less, large quantities of the heat must be transferred to the system. Figure 6.5 illustrates the essential features of the absorption cycle. Note that the compressor of the vapor compression cycle has been replaced by the generator, absorber, and pump, and that the solution of refrigerant and absorbent circulates through these elements. The refrigerant alone flows through the condenser, expansion valve, and evaporator. Heat is transferred to the solution in the generator, and the refrigerant vapor is separated from the solution while opposite process occurs in the absorber. The overall performance of the absorption cycle in terms of refrigerating effect per unit of energy input is generally poor; however waste heat such as that rejected from a power plant can often be used to achieve better overall energy conservation.

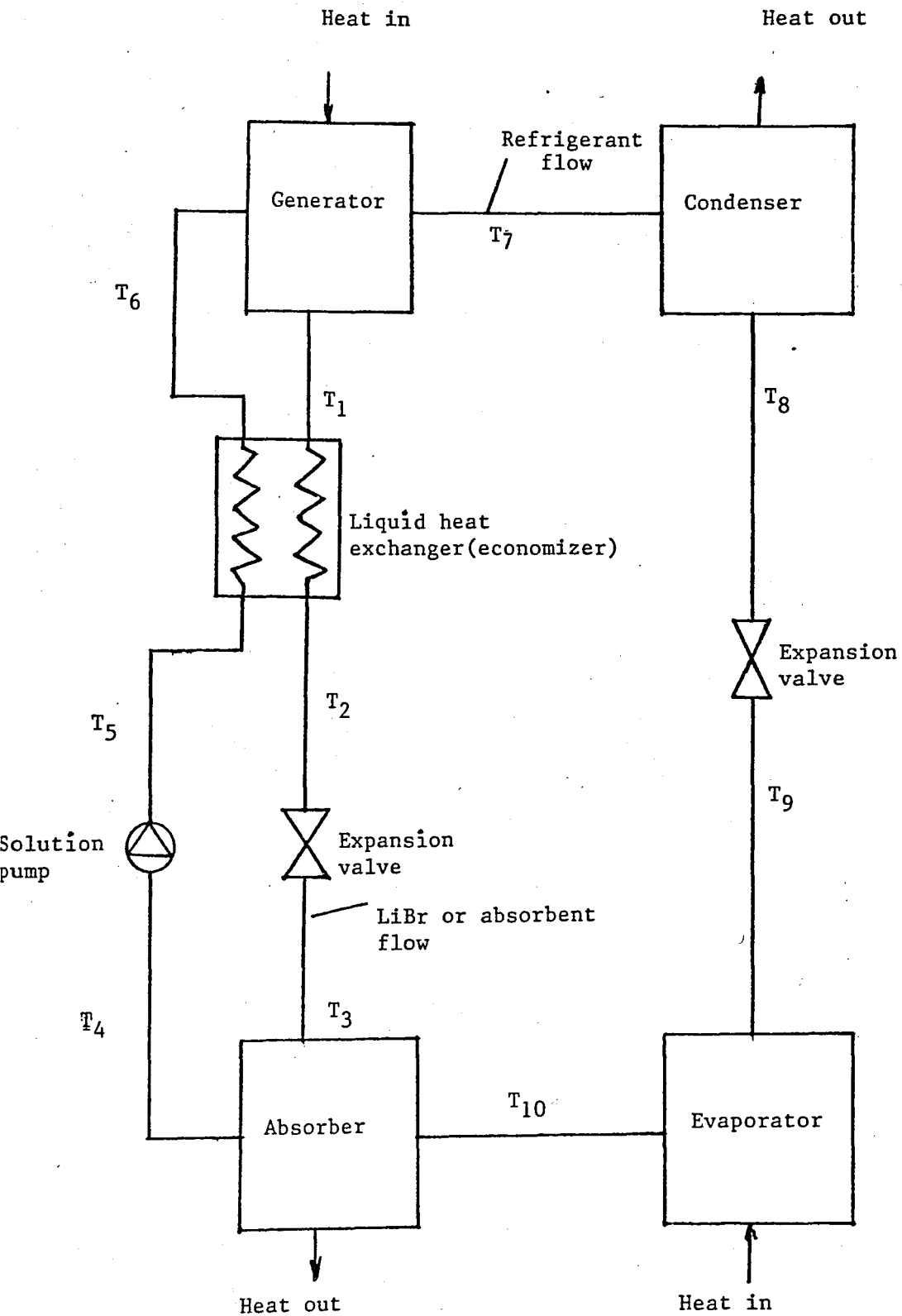


Figure 6.5 Simple absorption refrigeration system

6.3 THE THEORETICAL ABSORPTION REFRIGERATION SYSTEM

Figure 6.5 shows a schematic arrangement of equipment. For simplicity it is assumed that absorbent does not vaporize in the generator. Thus only pure refrigerant flows through the condensor and evaporator, and these two components may be identical to those used in a mechanical vapor-compression system. The vapor leaving the evaporator is mixed with a weak liquid solution in the absorber. By the removal of heat the refrigerant vapor could be absorbed by the weak liquid producing a liquid solution stronger in the refrigerant. The pressure of the liquid is then raised to the generator pressure by the pump. By the addition of heat in the generator, refrigerant vapor is driven out of the liquid solution. A heat exchanger is placed in the liquid solution circuit between the generator and absorber. The generator and condensor are on the high-pressure side of the system, while the evaporator and absorber are on the low-pressure side.

An absorption refrigeration system may be properly called a vapor compression system. Several components are required to perform the function of the compressor in a mechanical vapor-compression system.

The system shown in Figure 6.5 is not intended to represent an ideal type of absorption system. The ideal system would be completely reversible; this is not theoretically possible in Figure 6.5. To establish the maximum attainable C.O.P. for an absorption system

Bossjakovic's method will be used. Figure 6.7 shows the energy transfers to and from the fluids of an absorption system. The generator heating medium adds heat Q_G to the system. The pump adds work W_p . The substance to be cooled in the evaporator adds heat Q_E to the absorption system. The system rejects heat to the environment (cooling water or atmospheric air) in the absorber (Q_A) and in the condensor (Q_C). These last quantities will be combined into the waste heat

$$Q_0 = Q_A + Q_C \quad (6.4)$$

By the first law of thermodynamics

$$Q_0 = Q_G + Q_E + W_p \quad (6.5)$$

It will be assumed that the generator heating medium temperature T_G , the refrigerated substance temperature T_E , and the environmental temperature T_0 are constants.

The fluids within the absorption system circulate in a closed cycle for steady-state operation, the total entropy change of the fluids is zero. The changes of entropy for the overall system are only those occurring externally to the fluids of the absorption system. The entropy change for the generator heating medium is $\Delta S_G = -Q_G / T_G$, for the refrigerated substance

$\Delta S_E = -Q_E / T_E$ and for the environment $\Delta S_0 = Q_0 / T_0$. By the second law of thermodynamics, we must have

$$\begin{aligned} \Delta \dot{S} &= \Delta S_G + \Delta S_E + \Delta S_0 \geq 0 \\ \Delta S &= -\frac{Q_G}{T_G} - \frac{Q_E}{T_E} - \frac{Q_0}{T_0} \geq 0 \end{aligned} \quad (6.6)$$

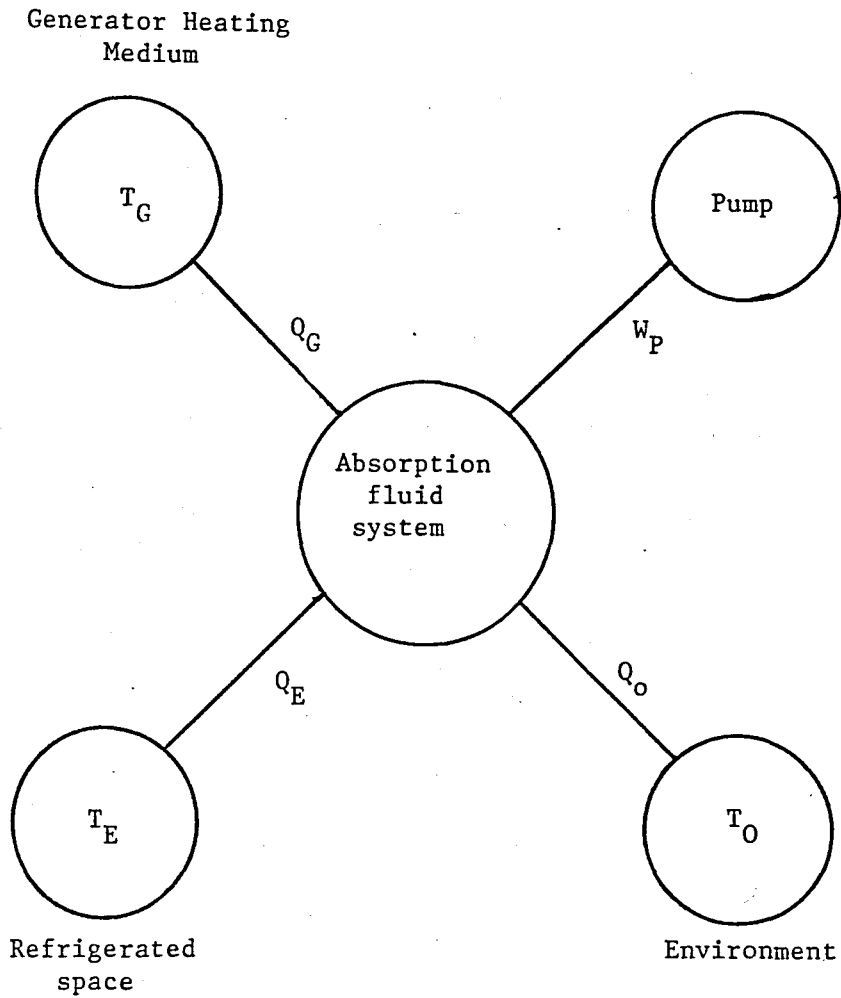


Table 6.7 External energy transfers for an absorption refrigeration system.

Using equations (6.5) and (6.6) it is obtained

$$Q_G \frac{(T_G - T_0)}{T_G} \geq Q_E \frac{(T_0 - T_E)}{T_E} - W_P \quad (6.7)$$

If it is that W_P is negligible, we have

$$\text{C.O.P} = \frac{Q_E}{Q_G} \leq \frac{T_E}{T_G} \frac{(T_G - T_0)}{(T_0 - T_E)} \quad (6.8)$$

and for a completely reversible system

$$(\text{C.O.P.})_{\max} = \frac{T_E (T_G - T_0)}{T_G (T_0 - T_E)} \quad (6.9)$$

Equation (6.9) shows an interesting result: the maximum attainable coefficient of performance for an absorption system is equal to the coefficient of performance for a carnot refrigerating cycle working between the temperatures T_E and T_0 multiplied by the efficiency of a carnot engine working between the temperatures T_G and T_0 .

Equation (6.9) also shows that for a given environmental temperature T_0 , the COP will increase with increase of the generator heating medium temperature T_G and with increase of temperature of the refrigerated region T_E . In practice, however the COP is much less than that given by Eq. 6.9 (4.)

6.4 THE ADVANTAGE OF ABSORPTION REFRIGERATION SYSTEM (5)

Since heat in the form of steam or hot water is generally the operating force of an absorption machine, the following situations are favorable to the application of absorption refrigeration machines:

1- Where low cost fuel is available, as in natural gas regions.

2- Where electric rates are high. Whenever the cost of steam in dollars per thousand pounds is less than fifty times the cost of electricity in dollars per kilowatt, a lower operating cost can be expected for the absorption machine. This is approximately the break-even point in operating cost (at design) between this machine and the electrically-driven compressor. The cost of steam is shown graphically in Figure 6.6a for different fuels. Comparison curves in Figure 6.6b indicate the operating cost of refrigeration for various costs of steam and electricity when applied to an absorption and centrifugal machine respectively. Demand charges should be included in the average electric cost. When comparing the operating cost of steam turbine-driven centrifugal and an absorption machine, a straight steam rate per ton of refrigeration is not a proper criterion. To obtain a correct analysis, the total system heat input should be used.

3- Where steam or gas utilities are desirous of promoting summer loads.

4- Where low pressure heating boiler capacity is largely or wholly unused during the cooling season.

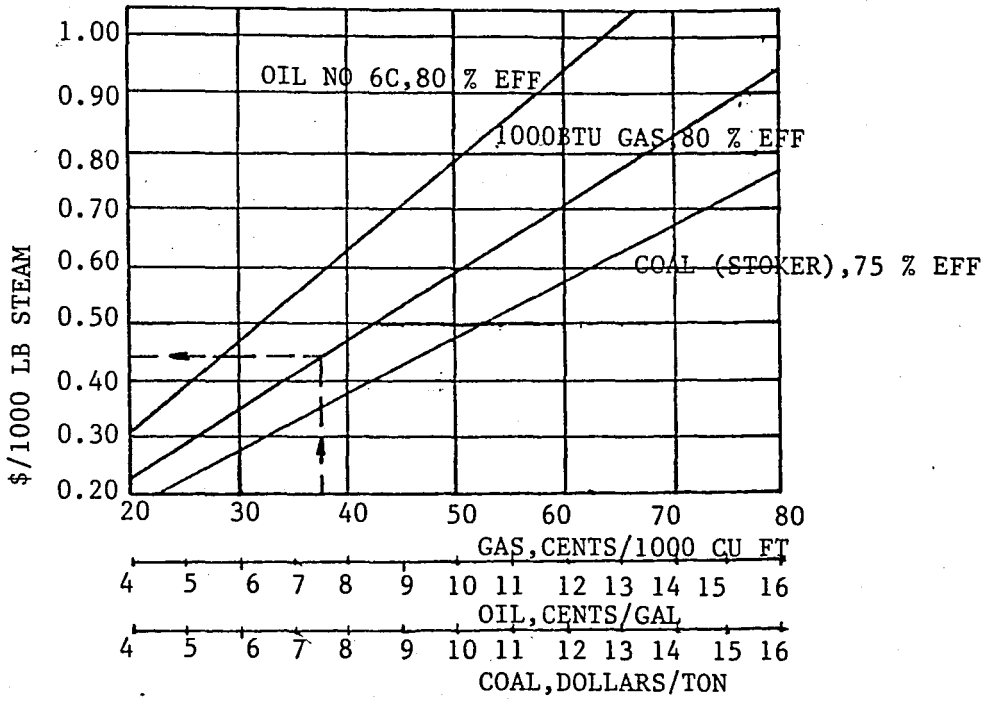


Figure 6.6.a. Comparative Costs of Steam

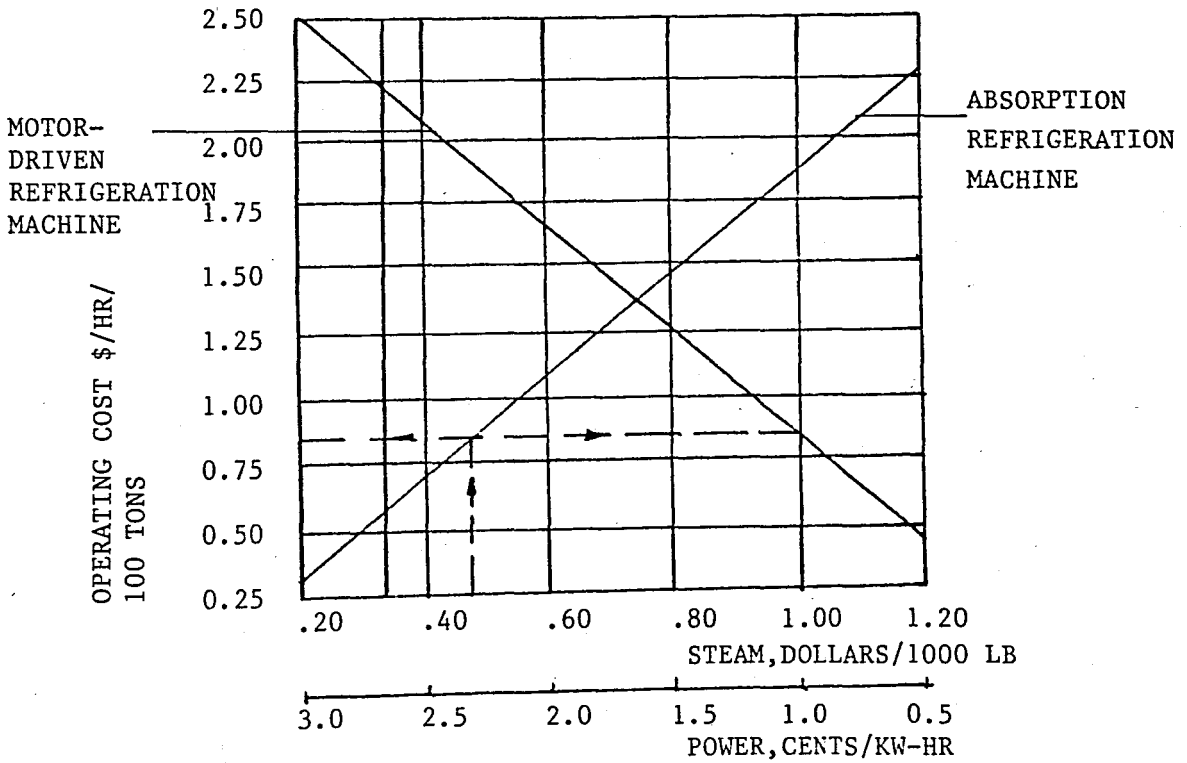


Figure 6.6.b. Steam Costs vs Power Costs

5- When waste steam is available such as that rejected from a power plant or from geothermal sources.

6- Where there is a lack of electric facilities for installing a conventional compression machine. Since the absorption machine uses only 2-9 % of the electric power required by compression type equipment, its use becomes attractive where emergency stand-by power is required, as in hospitals.

The absorption machine can be installed in practically any location in a building where the floor is of adequate strength and reasonably level. The absence of heavy moving parts practically eliminates vibrations and reduces the noise to a minimum.

Absorption machines may be applied also in conjunction with gas engines or turbines and with centrifugal machines as combination systems. The absorption machine can use as its heat source the steam or hot water made in a waste heat boiler or the jacket cooling water from a gas engine (120°C or higher). These temperatures can be obtained by solar collectors. For this reason, absorption machines are suitable for solar applications and used either for refrigeration or for heat -pump purposes. In the first case the heat from the solar collectors is used to operate or to assist the vapor generator, and it is required to be at a high temperature level. In the second case the solar heat delivered to the evaporator has a low temperature level.

6.5 THE AQUA - AMMONIA ABSORPTION REFRIGERATION SYSTEM

The Aqua-ammonia absorption system is one of the oldest methods of refrigeration. Ammonia is the refrigerant and water

is the absorbent. The system may be applicable where ammonia is a suitable refrigerant. Since the absorbent, water, is volatile, the simple system of must be modified. If aqua-ammonia were used in the system of Fig. 6.5, the vapor leaving the generator would contain too much water. Means must be provided to rectify the generator vapor and increase the mechanical concentration.

Figure 6.8 schematically shows an industrial aqua-ammonia system. The principal difference between Figures 6.5 and 6.8 is that a double rectifying column and a deplegator are added in Fig 6.8. A second heat exchanger is also added. The vapor leaving the generator may contain from five to ten percent of water vapor. Through the use of the rectifying equipment, the concentration of the vapor entering the condensor can be made almost equal to unity.

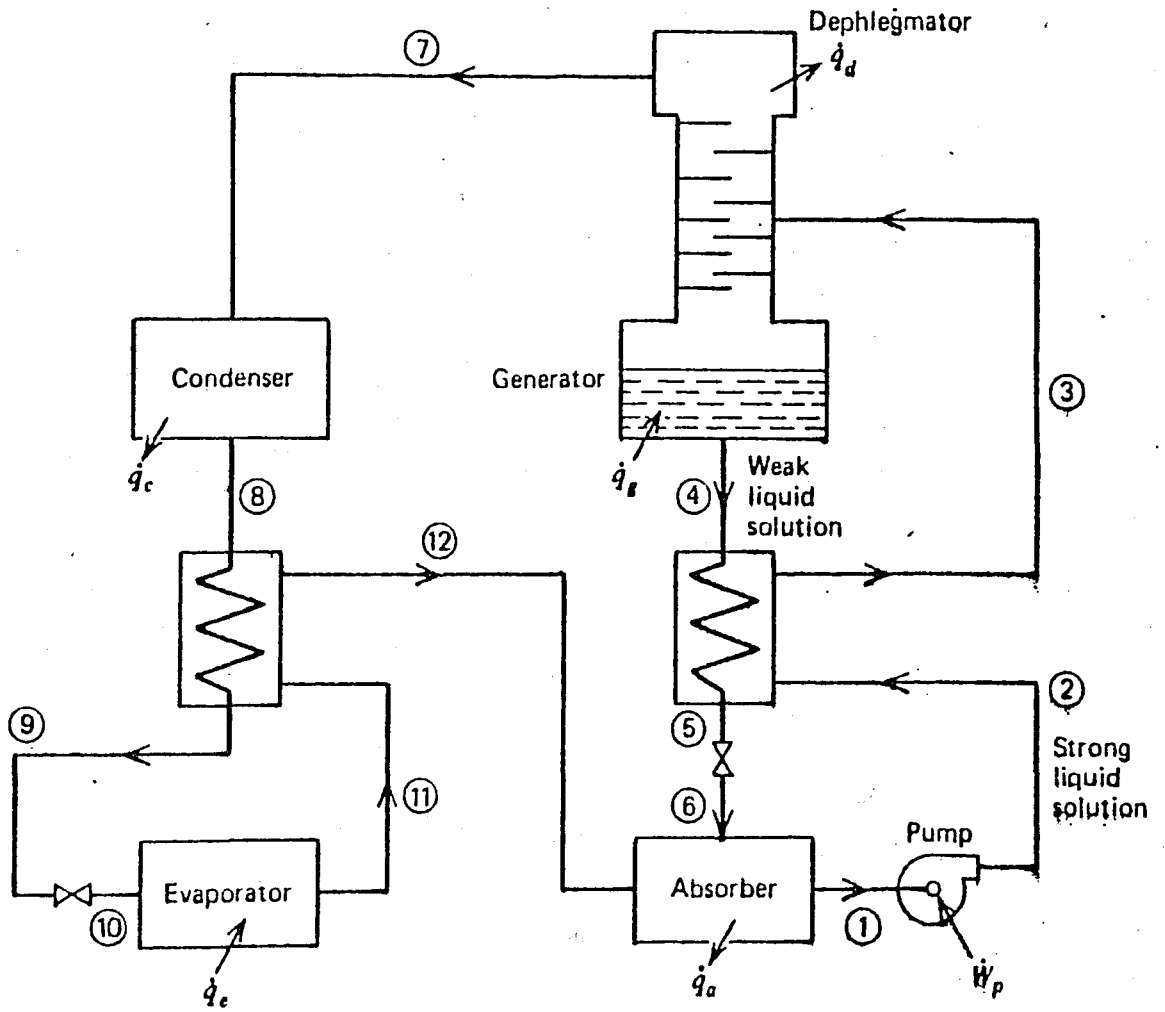


Figure 6.8 Industrial aqua-ammonia absorption system

6.6 THE LITHIUM BROMIDE-WATER ABSORPTION SYSTEM

In recent years the lithium bromide-water absorption system has become prominent in refrigeration for air conditioning. Water is the refrigerant while the absorbent is lithium bromide. Pure lithium bromide is a solid, but when it is mixed with sufficient water, homogeneous liquid solutions are formed. The outstanding feature of the system is the non-volatility of the lithium bromide. In the generator, only water vapor is driven off. Therefore no rectifying equipment is required. Compared to the aqua-ammonia system, the lithium bromide-water system is simpler and operates with a higher coefficient of performance. The primary disadvantage of the system is its limitation to relatively high evaporating temperatures since the refrigerant is water.

6.7 ABSORPTION AIR CONDITIONING AND FLAT PLATE COLLECTORS

Absorption air conditioning is only the air conditioning system compatible with the upper collection-temperature limits imposed by currently available flat-plate collector. Home-sized absorption air conditioning units are more expensive than vapor-compression air conditioning units, but to date only absorption air conditioning has been operated successfully in a full-scale installation. (1)

Presently, two types of absorption air conditioning systems are widely marketed: the lithium bromide-water system and the ammonia-water system. The main principals of these

system were explained before.

Of the two common absorption air conditioning systems, the $\text{LiBr-H}_2\text{O}$ is simpler since rectifying column is not needed: in the $\text{NH}_3\text{-H}_2\text{O}$ system a rectifying column assures that no water vapor, mixed with NH_3 , enters the evaporator where it could freeze. In addition, the $\text{NH}_3\text{-H}_2\text{O}$ system requires higher generator temperatures ($120\text{-}150^\circ\text{C}$) than a flat-plate solar collector can provide without special techniques. The $\text{LiBr-H}_2\text{O}$ system separates at a generator temperature of 85 to 92°C , achievable by a flat-plate collector; also, the $\text{LiBr-H}_2\text{O}$ system has a larger COP than the $\text{NH}_3\text{-H}_2\text{O}$ system. The disadvantage of $\text{LiBr-H}_2\text{O}$ systems is that evaporators can not operate at temperatures much below 4.44°C since the refrigerant is water vapor. However, for air conditioning this is not a so important disadvantage since evaporator temperatures between $5\text{-}10^\circ\text{C}$ are sufficient.

The effective performance of an absorption cycle depends on the two materials that comprise the refrigerant-absorbent pair. Desirable characteristics for the refrigerant-absorbent pair are,

- 1- The absence of a solid-phase absorbent.
- 2- A refrigerant more volatile than the absorbent in order to be separated from the absorbent easily in the generator.
- 3- An absorbent that has a small affinity for the refrigerant.
- 4- A high degree of stability for long-term operations.
- 5- A refrigerant that has a large latent heat so that the circulation

rate can be kept at the minimum.

6- A low corrosion rate and nontoxicity for safety reasons.

The only disadvantage of the LiBr-H₂O pair is the possible problem with crystallization in the generator. Actual COP of an absorption air conditioner can be calculated by

$$\text{COP} = \frac{\text{cooling effect}}{\text{heat input}} \quad (6.10)$$

or
$$\text{COP} = \frac{Q_E}{Q_G}$$

The pump work has been neglected since it is quite small or relatively small; in some 3 and 5 ton (10.5-17.5 kW) units the pump can even be eliminated entirely and a percolation principle employed instead.

The COP values for absorption air conditioning range from 0.5 for a small, single-stage unit to 0.85 for a double-stage, steam-fired unit. These values that can be achieved by a vapor-compression air-conditioner, COP for vapor-compression system changes from 3-6. It is incorrect to compare the COP of an air-conditioner with that of a vapor-compression air-conditioner, however, because the efficiency of electric power generation or transmission is not included in mechanical system is multiplied by the thermal efficiency of the power plant and the efficiency of the transmission network, it can be shown that the vapor-compression air-conditioner has little or no thermal performance advantage over the absorption air-conditioning system.

Now, an analytical evaluation of the LiBr-H₂O cycle. This system such as that shown in Figure 6.9 is to be analyzed for the following requirements.

- 1- The machine is to provide 25 kW of refrigeration with an evaporator temperature of 40°F (~4.44°C), an absorber outlet temperature of 90°F (~32°C), and a condensor temperature of 110°F (44°C).
- 2- The approach at the low temperature end of the liquid heat exchanger is to be 10°F (~5.55°C).
- 3- The generator is heated by a flat plate solar collector capable of providing a temperature level of 192°F (89°C) for the evaporation of the refrigerant.

Analytical evaluation of the LiBr-H₂O cycle requires that some simplifying factors be assumed, such as,

- 1- At those points in the cycle for which temperatures are specified, the refrigerant and absorbent phases are in equilibrium.
- 2- With the exception of pressure reductions across the expansion device between points 8 and 9 in Figure 6.9, pressure reductions in the lines and heat exchanger are neglected.
- 3- The temperature difference at the inlet to the liquid heat exchanger is 10°F.
- 4- Pressures at the evaporator and condensor are equal to the vapor pressure of the refrigerant, i.e., water, as found in the steam tables.
- 5- Enthalpies for LiBr-H₂O mixtures are given in Figure 6.10.
- 6- States 4, 6 and 1 are equilibrium states.

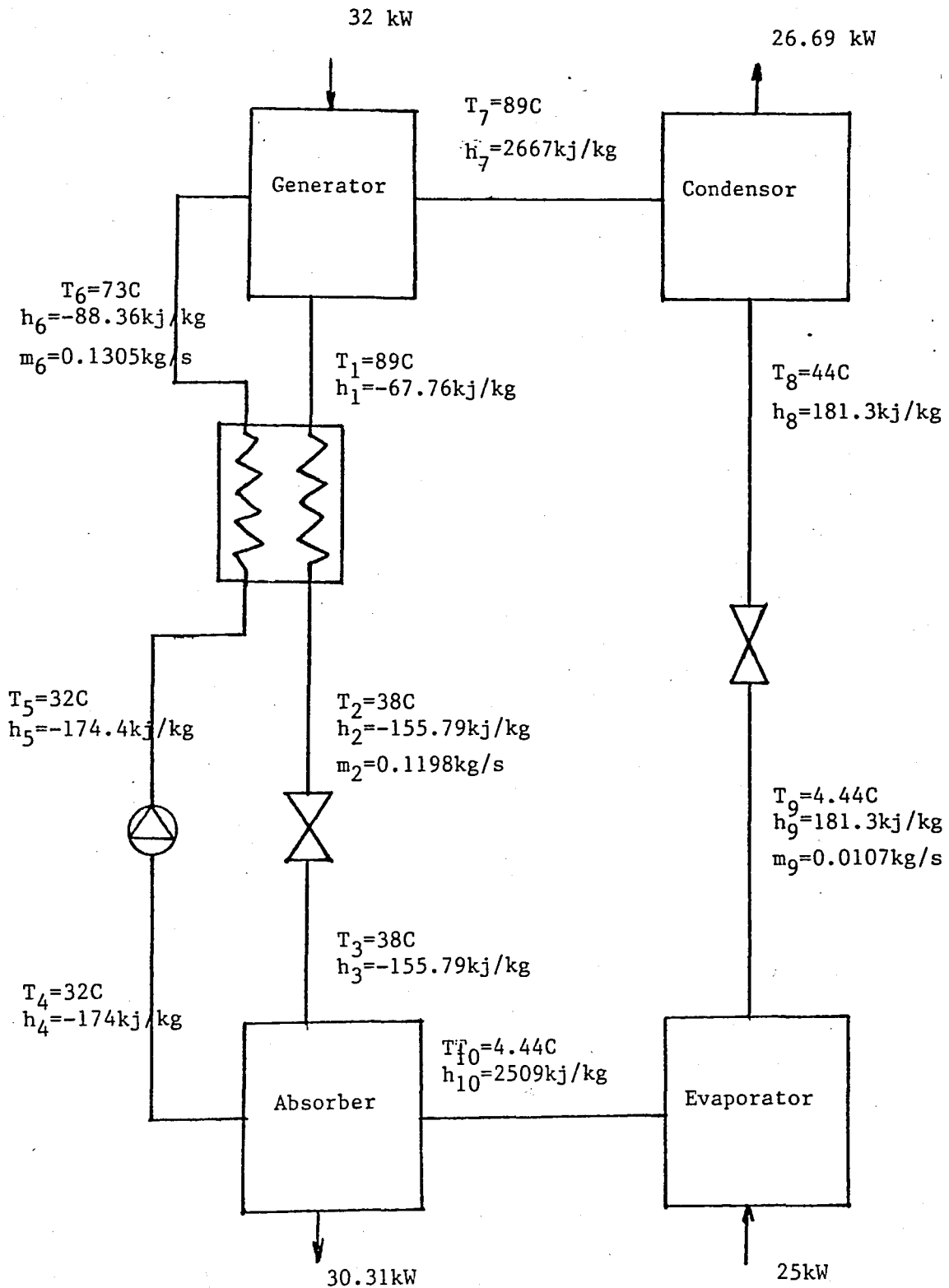


Figure 6.9 LiBr-Water absorption refrigeration cycle for 25kW of cooling.

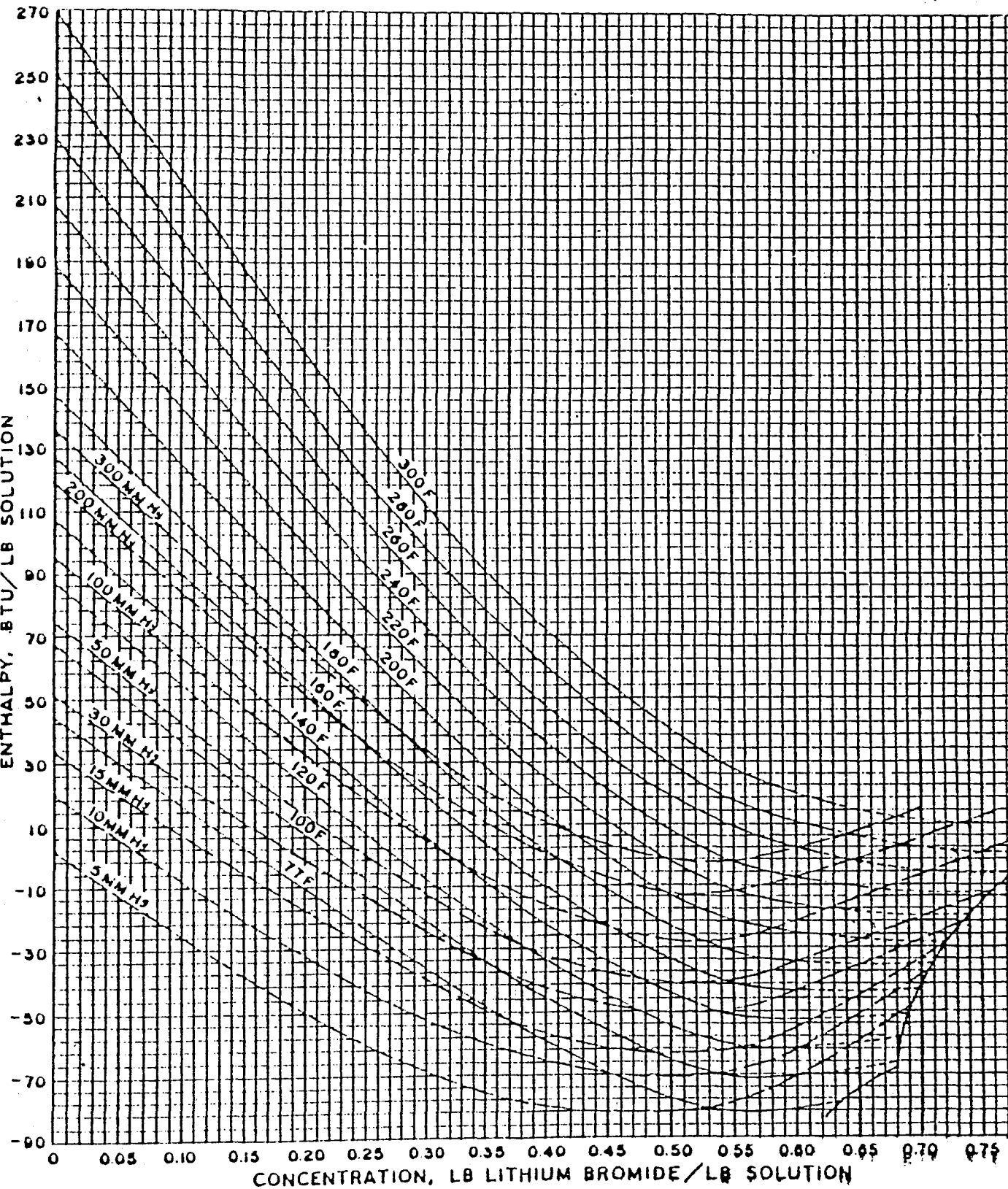
The first step in solving the problems is to find vapor pressures of steam or water, at points 8 and 9. The vapor pressure of steam or water corresponding to 44°C and 4.44°C are approximately 66 mm Hg (8.8 kPa) and 6.3 mm Hg (0.839 kPa).

For temperature 90°F and pressure 6.3 mm Hg, and, temperature 192°F and pressure 66 mm Hg, concentrations can be found as 0.56 and 0.61 respectively (from Figure 6.10)

Now a table can be prepared to show thermodynamic properties of refrigerant absorbent for Figure 6.9.

Table 6.1 Thermodynamic properties of refrigerant and absorbent for Figure 6.9

Condition no	Temperature $^{\circ}\text{F}$	Pressure mm Hg	LiBr weight fraction, %	Flow lb/lb H_2O	Enthalpy Btu/lb
T_1	192	66.0	0.61	11.2	-30
T_2	100	66.0	0.61	11.2	-67
T_3	100	6.3	0.61	11.2	-67
T_4	90	6.3	0.56	12.2	-75
T_5	90	66.0	0.56	12.2	-75
T_6	163	66.0	0.56	12.2	-38
T_7	192	66.0	0	1.0	1147
T_8	110	66.0	0	1.0	78
T_9	40	6.3	0	1.0	78
T_{10}	40	6.3	0	1.0	1079



Courtesy of Institute of Gas Technology, Chicago, IL
Figure 6.10 h-x diagram for LiBr-H₂O binary mixture

The enthalpy of states 2 and 3 are found by using the energy balance equations. The difference between h_6 and h_5 must be approximately equal to the difference between h_1 and h_2 . It must be added that $h_3=h_2$, $h_9=h_8$, and $h_4=h_5$.

Table 6.2 Thermodynamic properties of refrigerant and absorbent for Figure 6.9 in S.I. units.

Condition no	Temperature °C	Pressure KPa	LiBr weight fraction, %	Flow kg/kg H ₂ O	Enthalpy kJ/kg
1	89	8.8	0.61	11.2	-69.760
2	38	8.8	0.61	11.2	-155.79
3	38	0.839	0.61	11.2	-155.79
4	32.2	0.839	0.56	12.2	-174.400
5	32.2	8.8	0.56	12.2	-174.400
6	73	8.8	0.56	12.2	-88.36
7	89	8.8	0	1.0	2667.000
8	44	8.8	0	1.0	181.300
9	4.44	0.839	0	1.0	181.300
10	4.44	0.839	0	1.0	2509.000

Mass balance equations:

Relative flow rates for the absorbent (LiBr) and the refrigerant (H₂O) are obtained from material balances. A total material balance on the generator gives,

$$\dot{m}_6 = \dot{m}_1 + \dot{m}_7 \quad (6.11)$$

while a LiBr balance gives

$$\dot{m}_6 X_r = \dot{m}_1 X_{ab} \quad (6.12)$$

where X_{ab} = concentration of LiBr in absorbent, kg/kg of solution

X_r = concentration of LiBr in refrigerant, kg/kg of solution.

Substituting $(\dot{m}_1 + \dot{m}_7)$ for $\dot{m}_6$ gives

$$m_1 X_r + m_7 X_r = m_1 X_{ab} \quad (6.13)$$

Since the fluid entering the condenser is pure refrigerant, i.e.,

water, m_7 is the same as the flow rate of the refrigerant m_r

$$\frac{\dot{m}_1}{\dot{m}_7} = \frac{X_r}{X_{ab} - X_r} = \frac{\dot{m}_{ab}}{\dot{m}_r} \quad (6.14)$$

where $\dot{m}_{ab}$ = flow rate of absorbent, kg/sec

$\dot{m}_r$ = flow rate of refrigerant, kg/sec

substituting for X_r and X_{ab} from the table gives the ratio of absorbent-to-refrigerant flow rate.

$$\frac{\dot{m}_{ab}}{\dot{m}_r} = \frac{0.56}{0.61-0.56} = 11.2$$

The ratio of the refrigerant-absorbent solution flow rate $\dot{m}_s$ to the refrigerant-solution flow rate $\dot{m}_r$ is

$$\frac{\dot{m}_s}{\dot{m}_r} = \frac{\dot{m}_{ab} + \dot{m}_r}{\dot{m}_r} = 11.2 + 1 = 12.2$$

The flow rate of refrigerant required to produce the desired 25 kW of refrigerant (equivalent to 85,300 Btu/hr) is obtained from an energy balance about the evaporator.

$$q_{ref} = \dot{m}_r (h_q - h_{10}) \quad (6.15)$$

where q_{ref} is the cooling effect supplied the refrigeration unit, and

$$\dot{m}_r = \frac{q_{ref}}{(h_q - h_{10})} = \frac{25}{(181.3 - 2509)} = 0.0107 \text{ kg/sec}$$

The flow rate of the absorbent is

$$\dot{m}_{ab} = \frac{\dot{m}_{ab}}{\dot{m}_r} \dot{m}_r = 0.1198 \text{ kg/sec}$$

while the flow rate of the solution is

$$\dot{m}_s = \dot{m}_{ab} + \dot{m}_r = 0.1305 \text{ kg/sec}$$

The rate at which heat must be supplied to the generator q_{sup} is obtained from the heat balance

$$q_{supp} = \dot{m}_r h_7 + \dot{m}_{ab} h_1 - \dot{m}_{ab} h_6 = 30.96 \text{ kW} \sim 31 \text{ kW} \quad (6.16)$$

This requirement, which determines the size of the solar collector, likely represents the maximum heat load the refrigeration unit must supply during the hottest part of the day.

The coefficient of performance, COP, is

$$\text{COP} = \frac{q_{\text{ref}}}{q_{\text{sup}}} = \frac{25}{31} = 0.8$$

The rate of heat transfer in the other three heat-exchanger units- the liquid heat exchanger, water condenser, and the generator- is obtained from heat balances. For the liquid heat exchanger this gives

$$\begin{aligned} q_{1-2} &= \dot{m}_{ab} (h_1 - h_2) \\ &= 11.22 \text{ kW} \end{aligned} \quad (6.17)$$

where q_{1-2} is heat transferred from the absorbent stream to the refrigerant/absorbent stream. For the water condenser the rate of heat transfer q_{7-8} rejected to the environment is

$$\begin{aligned} q_{7-8} &= \dot{m}_r (h_7 - h_8) \\ &= 26.59 \text{ kW} \end{aligned} \quad (6.18)$$

The rate of heat removal from the absorber can be calculated from an over-all heat balance on this system.

$$\begin{aligned} q_a &= q_{7-8} - q_{\text{sup}} - q_{\text{ref}} \\ &= -29.41 \text{ kW} \end{aligned} \quad (6.19)$$

6.8 OPTIMIZATION OF THE LITHIUM-BROMIDE ABSORPTION REFRIGERATION SYSTEM BY MEANS OF A COMPUTER PROGRAM

An analytical evaluation of the LiBr-H₂O cycle was studied in the previous section. To find optimum condensing, evaporating, and generator temperatures for maximum COP, a computer program was developed. In this case, the chart which gives the relationships between concentration X, temperature T, pressure P, and enthalpy h; and steam tables can not be used. Properties of the working fluids such as vaporization temperatures and enthalpies are described using second order algebraic equations with concentration and saturation temperature as the independent variables. (6)

Enthalpy of water at temperature T(°C)

$$h = 4190 (T - 25) \text{ (J/kg)} \quad (6.20)$$

Enthalpy of saturated steam at temperature T(°C)

$$h = 1800T + 2.4 \times 10^6 \text{ (J/kg)} \quad (6.21)$$

Enthalpy of superheated steam at temperature T and a pressure corresponding to saturation temperature T_s

$$h = 1925 T - 126 T_s + 2.4 \times 10^6 \quad (6.22)$$

where in our case, T_s is condensing temperature, T_g.

Enthalpy of lithium bromide solution at concentration X and temperature T

$$h = (1.74X^2 - 1.91X + 0.2851) \times 10^6 + (2.01X^2 - 5.15X + 4.23)(T - 25) \times 10^3 \quad (6.23)$$

The temperature T_b at which a solution of lithium bromide at concentration X will be boil when at a pressure corresponding to steam at saturation temperature T_s .

The inputs of this program are $T_1, T_4, T_7, T_8, T_9, X_1, X_2$ and cooling load. They are, respectively, $38^\circ\text{C}, 32^\circ\text{C}, 90^\circ\text{C}, 44^\circ\text{C}, 6^\circ\text{C}, 0.56, 0.61$ and 1kW . The change of COP with respect to change of T_9 (evaporating temperature), T_8 (condensing temperature), T_7 (generator temperature) and X_2 (concentration of LiBr in the binary mixture) is observed.

Calculations for the cooling cycle were performed for the range of generator temperatures of 60 to 95°C , evaporating temperatures of 0° to 20°C , condensing temperatures 20 to 50°C and higher concentration X_2 of 0.50 to 0.645 . The influence of T_7, T_8, T_9 and X_2 on system coefficient of performance (COP) was considered under generator temperature of 90°C , evaporating temperature of 6°C , condensing temperature 44°C , absorber temperature 32°C , weak concentration 0.56 , rich concentration 0.61 and the temperature of mixture leaving the heat exchanger (economizer) at higher pressure of 38°C .

Figures 6.11, 6.12, 6.13 and 6.14 show that COP does not change drastically depending upon the change of T_7, T_8, T_9 and X_2 . This is not a surprised result since some of the results have been compared with data from the literature (7). The correspondance the data is quite satisfactory.

If only figures 6.11, 6.12, 6.13 and 6.14 are taken into account while system design conditions are being selected, it can be easily said that evaporating temperature must be as high as possible, condensing temperature must be as low as possible. However, higher evaporating temperature is not

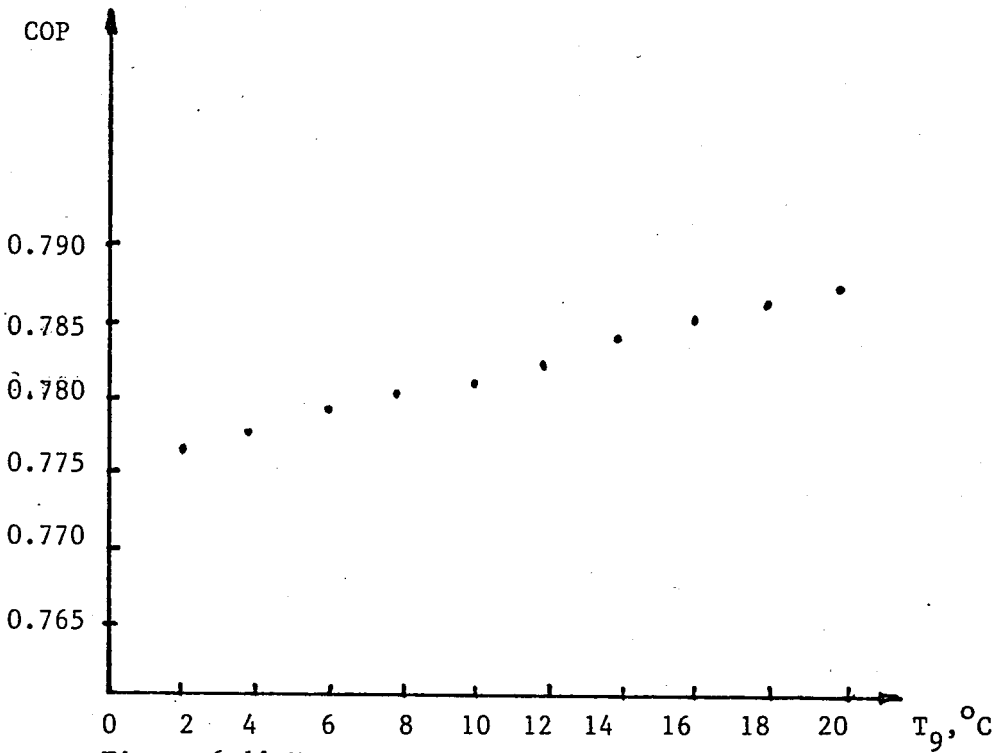


Figure 6.11 Variation of COP with evaporator temperature as a parameter

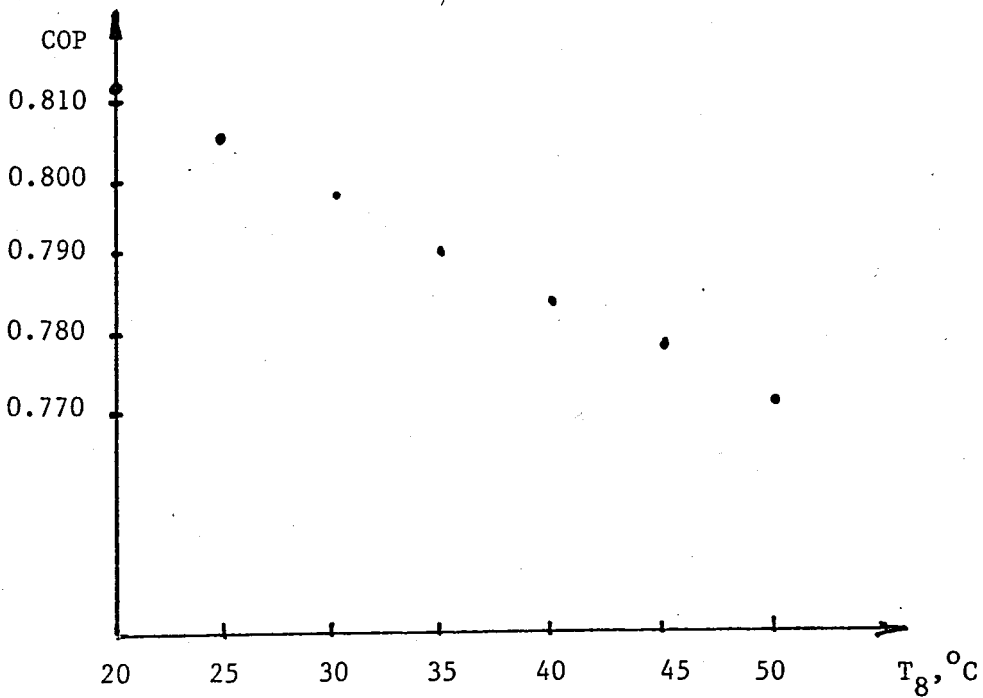


Figure 6.12 Variation of CPO with condensator temperature as a parameter

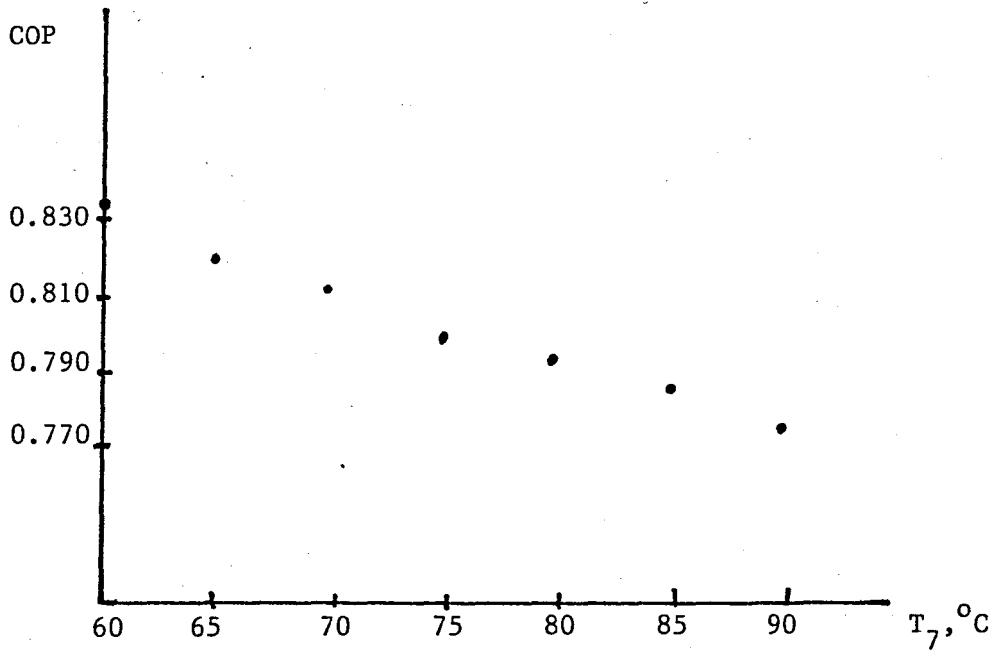


Figure 6.13 Variation of COP with generator temperature as a parameter

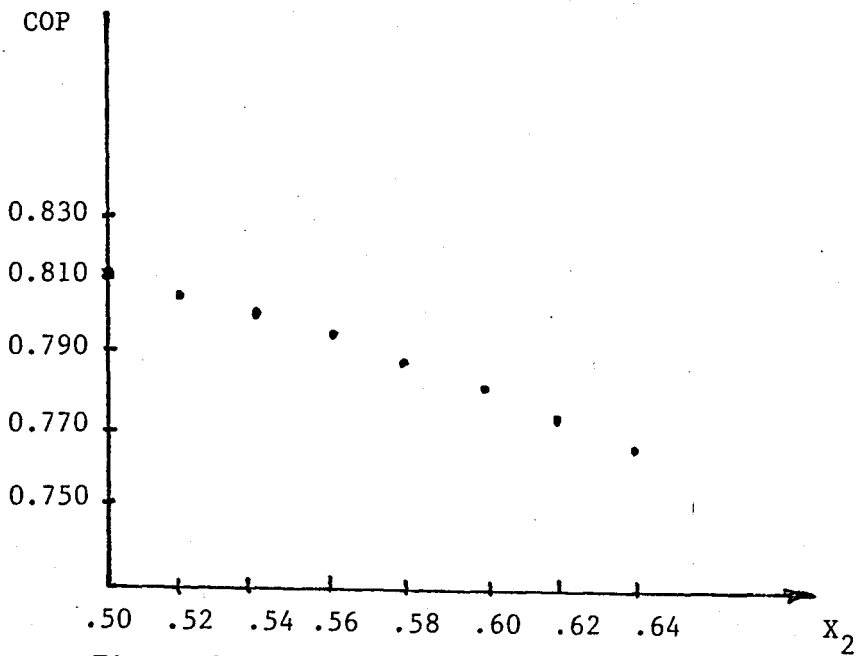


Figure 6.14 Variation of COP with concentration as a parameter

convenient for cooling purposes. So that it must be as low as possible - not lower than 0°C since water freezes at this temperature. Additionally, lower condensing temperature needs a cooling tower instead of air cooling. For this reason, condensing temperature is selected as high as possible. The condensor is proposed to be air cooled and would be made of finned tubes. It is not so easy to say that lower generator temperature means higher COP since experimental studies made by R.K. Suri and S. Ayyash (8') show that COP decreases drastically at temperatures below 85°C . This is valid for climatic conditions of Kuwait. There is no doubt that it will be different for Turkey's climatic conditions. However, it is impossible to predict the limit of generator temperature without any experimental study. It is not possible to keep the generator temperature constant without auxiliary energy. The utility of auxiliary energy input is highly attractive because of its high effective COP. To calculate the amount of auxiliary energy for a solar-cooled absorption system, an experimental study is necessary. In our study there is no auxiliary energy. It is assumed that all thermal energy is supplied by solar collectors.

Under the view of these explanations, the design conditions of our system can be determined as evaporating temperature 6°C , condensing temperature 44°C , generator temperature 90°C , rich concentration 0.61, weak concentration 0.56, absorber temperature 38°C , lower pressure 0.9345 kPa and higher pressure 9.1 kPa.

For this conditions COP is 0.77924, supply heat energy is 1.283 kW for 1 kW of cooling, the flow rate of refrigerant is 0.004 kg/sec, the flow

rate of rich solution is 0.0052 kg/sec, the flow rate of weak solution is 0.0048 kg/sec, the heat transfer to the outside from the absorber is 1.216 kW and 1.0673 kW from the condensor, and, lastly, the heat exchange in the economizer is 0.5322 kW.

VII. CALCULATION OF INSTANTANEOUS SOLAR RADIATION

7.1 SOLAR ANGLES

First of all, solar angles which will be used calculation of instantaneous solar radiation have to be defined.

Any point on the surface of the earth can be described in relation to the sun's rays at any instant if three fundamental quantities are known. Figure 7.1 shows a point P located on the surface of the earth in the northern hemisphere.

The latitude ϕ is the angle between the projection of OP on the equatorial plane. This is the same latitude that is commonly used on globes and maps to describe the location of a point with respect to the equator.

The hour angle w is the angle between the projection of OP on the equatorial plane and the projection on that plane of a line from the center of the sun to the center of the earth. Fifteen degrees of

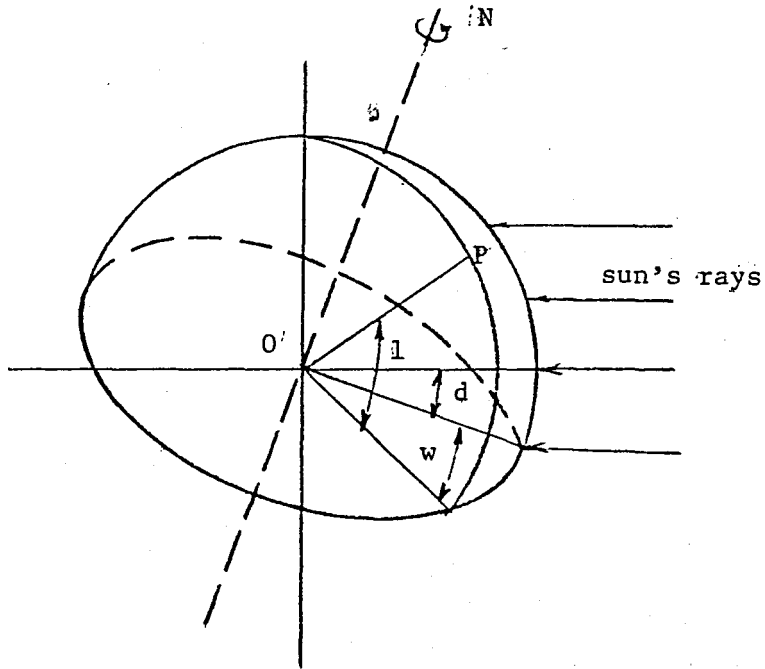


Figure 7.1 Latitude, hour angle and sun's declination

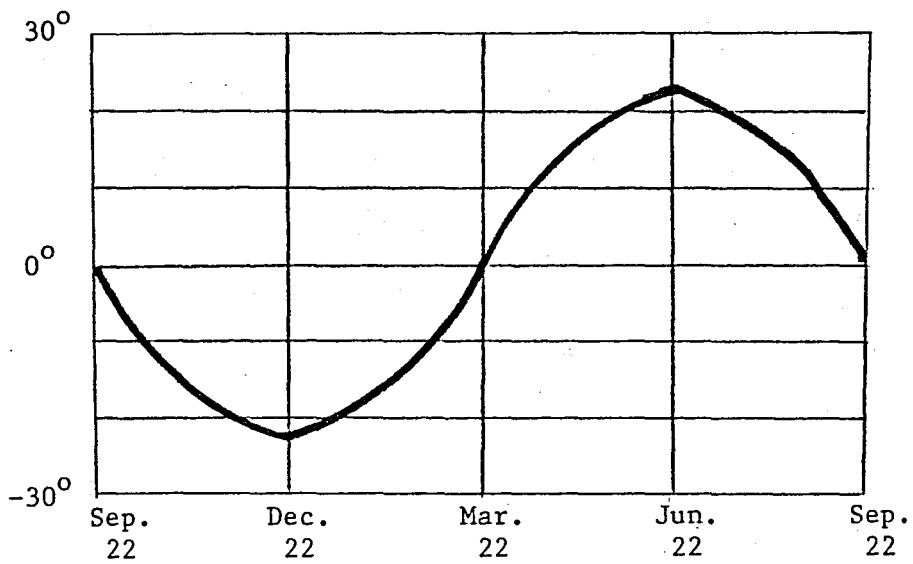


Figure 7.2 Variation of sun's declination

hour angle corresponds to one hour of time. The hour angle varies from zero at local solar noon to a maximum at sunrise or sunset. Solar noon occurs when the sun is at the highest point in the sky, and hour angles are symmetrical with respect to the solar noon. Thus the hour angles of sunrise and sunset on a given day are identical. It is negative in the morning and positive in the afternoon.

The sun's declination d is the angle between a line connecting the center of the sun and earth and the projection of that line on the equatorial plane. Figure 7.2 shows how the sun's declination varies throughout a typical year. On a given day in the year the declination varies slightly from year to year, but for typical HVAC calculations the values of any year are sufficiently accurate. The declination d can be found from the equation Cooper(1969)(2)

$$d = 23.45 \sin\left(360 \frac{284 + n}{365}\right) \quad (7.1)$$

where n is the day of the year; n can be conveniently obtained with the help of Table 7.1.

The solar altitude b_e (sun's altitude angle) is the angle between the sun's ray and the projection of that ray on a horizontal surface, Figure 7.3. It is the angle of the sun above the horizon. It can be shown by analytic geometry that the following relationship is true

$$\sin b_e = \cos l \cos w \cos d + \sin l \sin d \quad (7.2)$$

Table 7.1 Recommended Average Day for each Month and Values of n by Months

Month	n for ith Day of Month*	For the Average Day of the Month		
		Date	n, Day of Year**	Declination
January	i	17	17	-20.9
February	31+i	16	47	-13.0
March	59+i	16	75	- 2.4
April	90+i	15	105	9.4
May	120+i	15	135	18.8
June	151+i	11	162	23.1
July	181+i	17	198	21.2
August	212+i	16	228	13.5
September	243+i	15	258	2.2
October	273+i	15	288	- 9.6
November	304+i	14	318	-18.9
December	334+i	10	334	-23.0

* The average day is that day which has the extraterrestrial radiation closest to the average for the month

** These do not account for leap year; values of n from March onward for leap years can be corrected by adding 1. Declination values will also shift slightly.

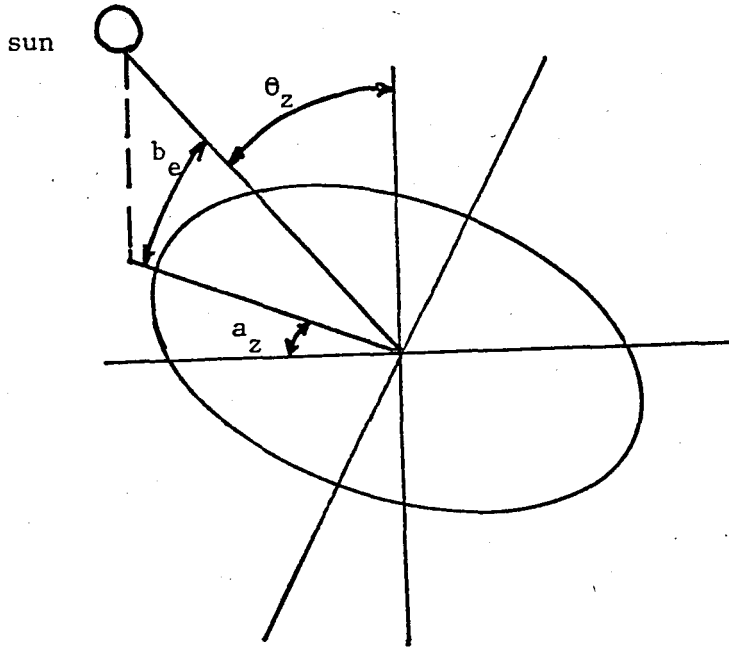


Figure 7.3 The solar altitude,zenith angle and azimuth angle

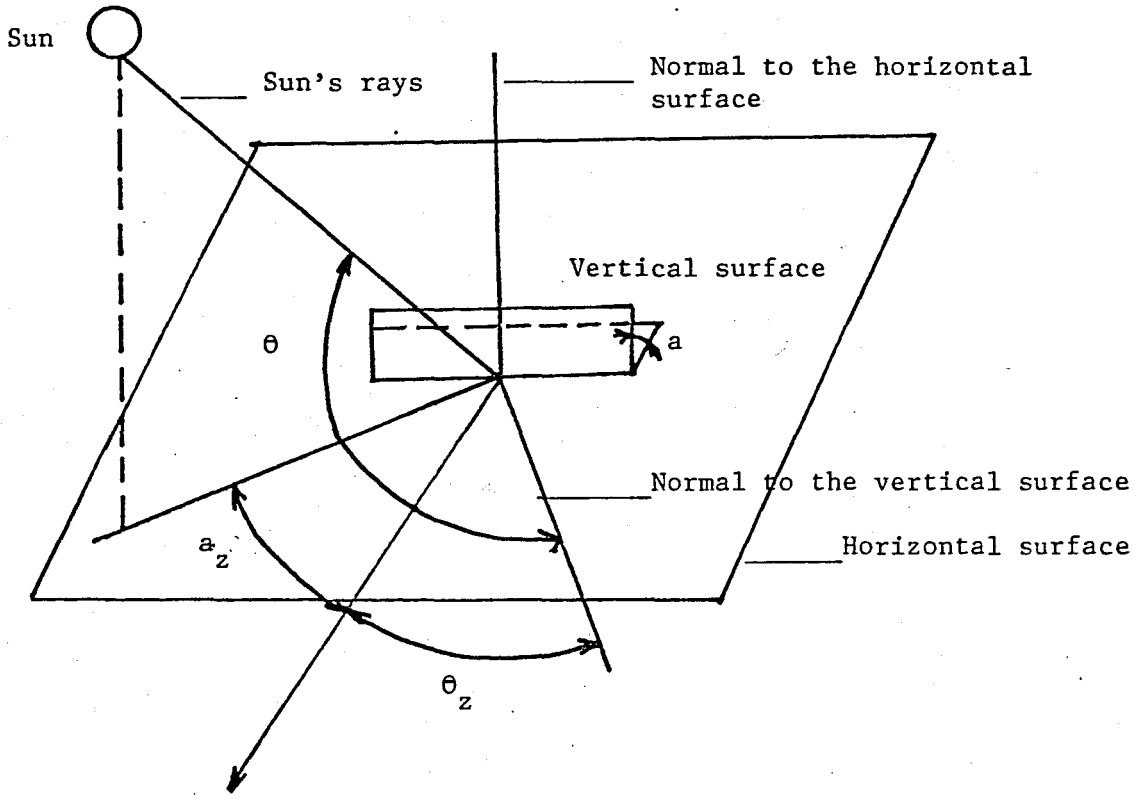


Figure 7.4 Wall solar azimuth angle of incidence and the angle of tilt

The sun's zenith angle θ_z is the angle between the sun's rays and a perpendicular to the horizontal plane at point P, Figure 7.3.

$$b_e + \theta_z = 90^\circ \quad (7.3)$$

The solar azimuth angle a_z is the angle in the horizontal plane measured between south and the projection of the sun's rays on that plane, Figure 7.3. Again by analytic geometry

$$\sin a_z = \frac{\cos \delta \sin w}{\cos b_e} = \cos \delta \sin w \sec b_e \quad (7.4)$$

Azimuth angle b_z is measured in degrees away from south. Figure 7.4 illustrates this quantity.

The angle of incidence θ is the angle between the sun's rays and the normal to the surface.

The angle of α is the angle between the surface and normal to the horizontal plane. The angle of ϕ is the angle between the surface and the horizontal plane ($90^\circ - \alpha$). The angle of incidence θ can be found by means equations (7.5) and (7.6)

$$\begin{aligned} \cos \theta = & \sin \delta \sin \alpha \cos a - \sin \delta \cos \alpha \sin a \cos b_z + \cos \delta \cos \alpha \cos a \cos w + \\ & \cos \delta \sin \alpha \sin a \cos b_z \cos w + \cos \delta \sin \alpha \sin b_z \sin w \end{aligned} \quad (7.5)$$

or

$$\cos \theta = \cos b_e \cos(a_z - b_z) \sin \phi + \sin b_e \cos \phi \quad (7.6)$$

7.2 SOLAR RADIATION ON THE EARTH'S SURFACE

A widely quoted value of the mean solar constant G_{sc} , the rate of irradiation on a surface normal to the sun's rays beyond the earth's atmosphere and at the mean earth-sun distance is 422 Btu/hrft^2 or 1394 kW/m^2 . According to a fairly recent study (M.P. Thekaekara and A.J. Drummond, "Standard Values for the Solar Constant and its Spectral Components"), the following value is used for G_{sc} :

$$G_{sc} = 428 \text{ Btu/hrft}^2 = 1353 \text{ W/m}^2 \quad (9)$$

Because of the inverse square law of irradiation from the sun varies about ± 3.5 percent because of the variation in distance between the sun and earth. On January 1 the radiation is the greatest and on July 1 the least. Because of the large amount of atmospheric absorption of this radiation, and because this absorption is variable and difficult to predict, precise values of the solar constant do not need to be used in most HVAC calculations.

Radiation arriving on the surface is dependent on

- a. Geometric considerations (positions of sun in the sky)
- b. Meteorological conditions (clouds, water vapor amount etc.)
- c. Local environment.

A part of the solar radiation entering the earth's atmosphere is scattered by gas and water vapor molecules and by cloud and dust particles. The blue color of the sky is a result of the scattering of

of some of the shorter wavelengths from the visible portion of the spectrum. The familiar red at sunset results from the scattering of longer wavelengths by dust or cloud particles near the earth. Some radiation (particularly ultraviolet) may be absorbed by ozone in the upper atmosphere and other radiation is absorbed by water vapor near the earth's surface. That part of the radiation is not scattered or absorbed and reaches the earth's surface is called direct radiation. It is accompanied by radiation that has been scattered or reemitted called diffuse radiation. Radiation may also be reflected onto a surface from nearby surfaces. The total radiation G_T on a surface normal to the sun's rays is thus made up of normal direct irradiation G_{ND} , diffuse irradiation G_d , and reflected irradiation G_R

$$G_T = G_D + G_d + G_R \quad (7.7)$$

where

$$G_D = G_{DN} \cos \theta \quad (7.8)$$

On a cloudy day all of the solar radiation is diffuse, on a clear day most (80 - 90%) of solar radiation is direct.

One of the methods to calculate direct and diffuse radiation is ASHRAE CLEAR SKY MODEL. The value of the solar irradiation at the surface of the earth on a clear day is given by (10)

$$G_{DN} = A / \exp(B/\cos \theta) \quad (7.9)$$

where

G_{DN} = the direct normal irradiation(W/m^2)

A = apparent solar irradiation at air mass equal zero(W/m^2)

B = atmospheric extinction coefficient

θ_z = zenith angle

$$G_d = CG_{DN}(1 + \cos\phi)/2 + g_r G_{dn}(\sin b_e + C)(1 - \cos\theta)/2 \quad (7.10)$$

where

G_d = the diffuse irradiation(W/m^2)

C = the ratio of diffuse to direct normal irradiation on a horizontal surface

g_r = ground reflectance, it is approximately equal to 0.7 for ground vegetation, 0.8 for snow cover, and 0.15 for a gravel roof.

ϕ = tilt angle of the collector.

If G_R , reflection from other buildings etc. is neglected, G_T can be written as

$$\begin{aligned} G_T &= G_D + G_d & (7.11) \\ &= G_{DN}\cos\theta + G_d \\ &= G_{DN}(\cos\theta + C(1 + \cos\phi)/2 + g_r(\sin b_e + C)((1 - \cos\theta)/2)) \end{aligned}$$

the values of A , B and C are given in Table 7.2.

Table 7.2 (10) The values of A,B,C and d on 21st day of each month each month.

Date	d	A	B	C
Jan 21	-20.00	1230	0.142	0.058
Feb 21	-10.00	1215	0.144	0.060
Mar 21	0.00	1186	0.156	0.071
Apr 21	11.60	1136	0.180	0.097
May 21	20.00	1104	0.196	0.121
Jun 21	23.45	1088	0.205	0.134
Jul 21	20.60	1085	0.207	0.136
Aug 21	12.30	1107	0.201	0.122
Sep 21	0.00	1151	0.177	0.092
Oct 21	-10.50	1192	0.160	0.073
Nov 21	-19.80	1221	0.149	0.063
Dec 21	-23.45	1233	0.149	0.057

7.3 CALCULATION OF USEFUL HEAT ENERGY

The basic method of measuring the amount of useful heat energy is to expose the operating collector to solar radiation and measure the fluid temperatures and the fluid flow rate.

The useful gain is (2) .

$$Q_u = \dot{m} C_p (T_o - T_i) \quad (7.12)$$

where,

Q_u = useful heat energy (kJ/s or kW)

$\dot{m}$ = fluid flow rate (kg/s)

T_o = fluid outlet temperature ($^{\circ}\text{C}$)

T_i = fluid inlet temperature ($^{\circ}\text{C}$)

C_p = specific heat factor (kJ/kg $^{\circ}\text{C}$)

The thermal performance of a collector operating under steady conditions can be written as

$$Q_u = A_c F_R (G_T(TA) - U_L (T_i - T_a)) \quad (7.13)$$

where

A_c = collector area (m^2)

F_R = heat removal factor

(TA)= transmissivity-absorptance product

U_L = loss coefficient (kW/ m^2 $^{\circ}\text{C}$ or W/ m^2 $^{\circ}\text{C}$)

T_a = ambient temperature

F_R is equivalent to a conventional heat exchanger effectiveness which is defined as the ratio of the actual heat

transfer to the maximum possible heat transfer. The maximum possible useful energy gain in a solar collector is at the inlet fluid temperature; heat losses to surrounding are then at a minimum.

U_L is the overall heat loss coefficient. It depends on many factors such as, ambient temperature, mean plate temperature, wind heat transfer coefficient, collector tilt angle etc. An empirical equation was developed by Klein (1979) (2.) : This relationship fits the graphs for U_L for plate temperatures between ambient and 200 °C to within $\pm 0.3 \text{ W/m}^2\text{°C}$.

$$U_L = \left[\frac{N}{\frac{F}{T_{p,m}} \cdot \frac{(T_{p,m} - T_a)}{(N + f)} + \frac{1}{h_w}} \right]^{-1} + \frac{\sigma (T_{p,m} + T_a) (T_{p,m}^2 + T_a^2)}{(e_p + 0.00591 N h_w)^{-1} + \frac{2N+f-1+0.133 e_p}{e_p} - N} \quad (7.14)$$

where,

N = number of glass covers

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

$F = 520 (1 - 0.00051 \phi^2)$ for $0^\circ < \phi < 70^\circ$ for $70^\circ < \phi < 90^\circ$, use $\phi = 70^\circ$

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

ϕ = collector tilt angle

e_g = emittance of glass(0.88)

e_p = emittance of plate

T_a = ambient temperature ($^{\circ}\text{K}$)

$T_{p,m}$ = mean plate temperature ($^{\circ}\text{K}$)

h_w = wind heat transfer coefficient ($\text{W/m}^2\text{ }^{\circ}\text{C}$)

Equation (7.14) is convenient both for hand calculations or for calculations on a digital computer. To use this empirical relationship or the more complicated exact equations to find U_L , it is necessary to know the mean plate temperature, $T_{p,m}$. For a theoretical study, it is very hard to calculate $T_{p,m}$. In this study, to find U_L Figure 7.5 and to find (TA) Tables 7.3 and 7.4 are used.

Table 7.3 (11)

Collector Type	(TA) _n	$U_L, \text{W/m}^2\text{ }^{\circ}\text{C}$
Black paint, single cover	0.84	7.4
Black paint, double covers	0.76	4.5
Selective surface (black nickel), single cover	0.80	5.0
Selective surface (black nickel), double covers	0.76	3.2

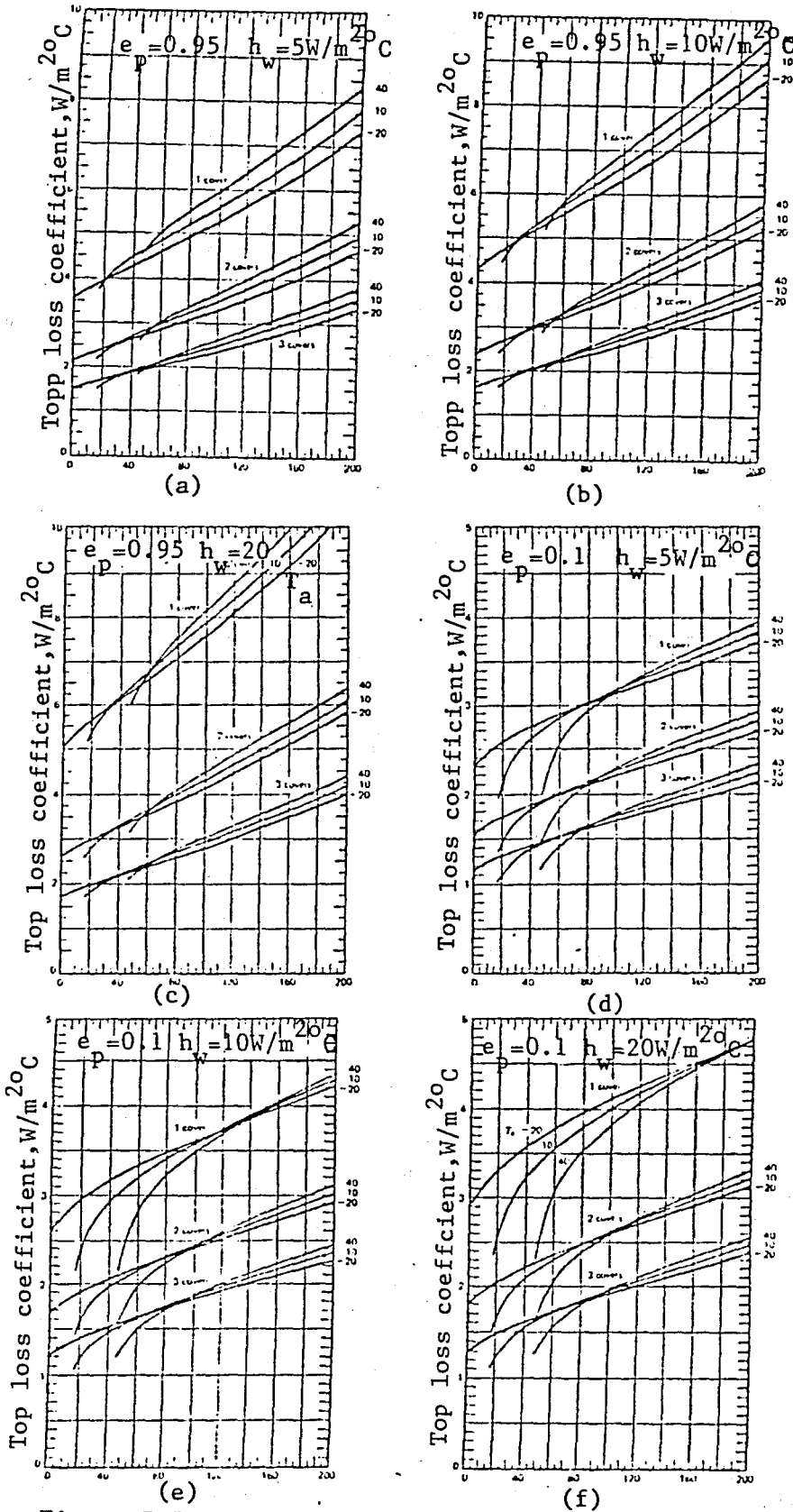


Figure 7.5 Top loss coefficient for slope of 45°

Table 7.4 (12) $(TA)/(TA)_n$ for black paint two glass cover collector,
 $l = 39^\circ$

	$\phi=1-15^\circ$			$\phi=1$			$\phi=1+15^\circ$			$\phi=90^\circ$		
HD/H	0.35	0.45	0.55	0.35	0.45	0.55	0.35	0.45	0.55	0.35	0.45	0.55
Jan	0.89	0.89	0.88	0.92	0.92	0.91	0.94	0.93	0.92	0.93	0.93	0.92
Feb	0.91	0.90	0.89	0.93	0.92	0.91	0.93	0.93	0.92	0.91	0.90	0.90
Mar	0.92	0.91	0.90	0.92	0.91	0.90	0.92	0.91	0.90	0.86	0.86	0.86
Apr	0.92	0.91	0.90	0.92	0.91	0.89	0.90	0.89	0.88	0.72	0.74	0.76
May	0.92	0.91	0.90	0.90	0.90	0.89	0.88	0.87	0.87	0.68	0.71	0.74
Jun	0.92	0.91	0.90	0.90	0.89	0.88	0.87	0.86	0.86	0.69	0.73	0.76
Jul	0.92	0.91	0.90	0.90	0.89	0.88	0.87	0.87	0.86	0.68	0.72	0.75
Aug	0.92	0.91	0.90	0.91	0.90	0.89	0.89	0.89	0.88	0.69	0.72	0.75
Sep	0.92	0.91	0.90	0.92	0.91	0.90	0.92	0.91	0.90	0.82	0.82	0.83
Oct	0.91	0.90	0.90	0.93	0.92	0.91	0.93	0.92	0.91	0.90	0.89	0.89
Nov	0.90	0.89	0.89	0.93	0.92	0.91	0.94	0.93	0.92	0.93	0.92	0.92
Dec	0.89	0.88	0.88	0.92	0.92	0.91	0.94	0.94	0.93	0.94	0.93	0.93

In Table 7.4, HD is the monthly average daily diffuse radiation on a horizontal surface and it is the monthly average daily radiation on a horizontal surface.

7.5 CALCULATION OF USEFUL HEAT ENERGY BY MEANS OF A COMPUTER PROGRAM

The inputs of this program shown in Appendix A2 are latitude angle A_1 , declination angle A_2 , the wall azimuth angle A_3 , collector tilt angle A_9 and collector area. This program can calculate the instantaneous heat gain between 7 a.m and 5 p.m. The wall azimuth angle A_3 and collector tilt angle A_9 are changed from 20° to 50° . The results show that tilt angle equals to latitude is the best for cooling, in other words, tilt angle equals to latitude. This is a good result since it matches the results of Kreith (1). Kreith states in Reference (1), optimal collector tilt ϕ or A_9 is equal to latitude (except in the South hemisphere where $\phi = 1 - 10^\circ$).

It is known that the wall azimuth angle 0° or near to 0° is the best for heating purposes if it is not changed during the day depending upon the sun's position. When the wall azimuth angle is 0° , the maximum heat energy is obtained at 12.00 a.m. However, for cooling, it is desired to move the peak point of useful heat gain to the afternoon since cooling loads are usually higher in afternoon. The results indicate that peak point moves to afternoon when azimuth angle is higher than 0° . In other words, collector is tilted to west. Unfortunately, in this case, total amount of collected heat energy decreases.

In the view of these explanations, the wall azimuth angle and collector tilt angle were selected 60° and 40° respectively, to move the peak point to the afternoon. The results are shown in Table 7.5.

Table 7.5 Useful Heat Gain for $\phi=40^\circ$ and $b_z=60^\circ$

Time	Useful Heat Gain(W/m^2)
6.30	17.574
7.00	72.683
7.30	140.048
8.00	212.368
8.30	286.555
9.00	360.532
9.30	432.569
10.00	501.103
10.30	564.687
11.00	621.970
11.30	671.702
12.00	712.729
12.30	743.994
13.00	764.528
13.30	773.418
14.00	769.764
14.30	752.772
15.00	720.565
15.30	671.802
16.00	602.851
16.30	506.831
17.00	368.398
17.30	155.107

VIII. CONCLUSION

It was found in Chapter 6 that coefficient of performance COP is 0.77924 for evaporating temperature 6°C , condensing temperature 44°C , generator temperature 90°C , rich concentration 0.61, weak concentration 0.56, absorber temperature 38°C , lower pressure 0.9345 kPa and higher pressure 9.1 kPa. As a result, 1283.3 Watts of heat supply is needed to obtain 1000 Watts of cooling with LiBr- H_2O absorption system.

Recommended cooling load per m^2 per hour in Reference (13) is $150 \text{ kcal/m}^2\text{hr}$. If a six-storey building with 1000 m^2 of roof area is taken into consideration, cooling load of this building is approximately 900,000 kcal/hr or 1047 kW.

It can be easily said that 1343.6 kW of heat supply is required to obtain 1067 kW of cooling. Another assumption is that

1000 m² of solar collector can be constructed on the roof of building. Then 773.418 kW of required heat energy is supplied by these collectors at 13.30 for $b_z = 60^\circ$ and $\phi = 40^\circ$ (from Table A2.4). In other words, approximately 57 % of required heat is supplied by solar collectors. If the cooling load is constant during the day and there is no need for cooling between 17.30 and 6.30, the contribution of solar collectors is shown in Table 8.1.

It must be also noted that to find the amount of heat energy per hour, Q_{useful} (W/m²) at mid-hours is multiplied by 3600. The percent of heat energy supplied by solar collectors between 6.30 and 17.30 is 35.45 % for six-storey building. The results for five, four, three, two and single storey buildings are shown in Table 8.1. The percentages which are higher than 100 are accepted as 100 %. In other words, it is assumed that there is no heat storage.

There is no doubt that the rest of required heat energy will be supplied by another heat sources like fuel-oil, natural gas, etc. These results show that solar assisted LiBr-H₂O absorption system may provide energy savings in considerable amounts.

Table 8.1 The Contribution of Solar Collectors for Six, Five, Four, Three, Two and Single-Storey Buildings, respectively.

Time	%	%	%	%	%	%
6.30	1.30	1.56	1.95	2.60	3.90	7.80
7.00	5.40	6.48	8.10	10.80	16.20	32.40
7.30	10.42	12.50	15.63	20.84	31.26	62.52
8.00	15.80	18.96	23.70	31.60	47.40	94.80
8.30	21.32	25.58	31.98	42.64	63.96	127.90
9.00	26.83	32.19	40.24	53.66	80.49	160.98
9.30	32.19	38.62	48.28	64.38	96.57	193.14
10.00	37.29	44.74	55.93	74.58	111.87	223.74
10.30	42.02	50.42	63.03	84.04	126.06	252.12
11.00	46.29	55.54	69.43	92.58	138.87	277.74
11.30	49.99	59.98	74.98	99.98	149.97	299.94
12.00	53.04	63.04	79.56	106.08	159.12	318.24
12.30	55.37	66.44	83.05	110.74	166.11	332.22
13.00	56.90	68.28	85.35	113.80	170.70	341.40
13.30	57.56	69.07	86.34	115.12	172.68	345.36
14.00	57.29	68.74	85.93	114.58	171.87	343.74
14.30	55.98	67.17	83.97	111.96	167.94	335.88
15.00	53.62	64.34	80.43	107.24	160.86	321.72
15.30	50.00	60.00	75.00	100.00	150.00	300.00
16.00	44.86	53.83	67.29	89.72	134.58	269.16
16.30	37.72	45.26	56.58	75.44	113.16	226.32
17.00	27.41	32.89	41.11	54.82	82.23	164.46
17.30	11.54	13.84	17.31	23.08	34.62	69.24
6.30-17.30	35.45	42.53	53.17	70.90	77.52	86.63

APPENDIX A1

In this Appendix the results of a computer program which calculates the COP of LiBr-H₂O absorption system are listed. The description of symbols used in this Appendix are shown below.

T_9 : Evaporator temperature

Q_A : Heat transfer to the surrounding from the absorber

Q_C : Heat transfer to the surrounding from the condensor

Q_S : Heat transfer to the system in the generator

Q_E : Heat exchange in the economizer

m_3 : Mass flow rate of strong solution

m_4 : Mass flow rate of weak solution

m_{10} : Mass flow rate of refrigerant

COP: Coefficient of performance

APPENDIX A1. The Results of Computer Program to Calculate the COP of
LiBr-H₂O Absorption System

Table A1.1 The Variation of COP with Evaporator Temperature T₉ as a
Parameter

T ₉ °C	Q _A J	Q _C J	Q _E J	Q _S J	$\dot{m}_3$ kg/s	$\dot{m}_4$ kg/s	$\dot{m}_{10}$ kg/s	COP
00	1217.0	1072.3	525.6	1289.3	0.0053	0.0048	0.0004	0.77563
1	1216.8	1071.4	525.2	1288.3	0.0053	0.0048	0.0004	0.77624
2	1216.7	1070.6	524.8	1287.3	0.0052	0.0048	0.0004	0.77684
3	1216.5	1069.8	524.4	1286.3	0.0052	0.0048	0.0004	0.77744
4	1216.3	1069.0	524.0	1285.3	0.0052	0.0048	0.0004	0.77804
5	1216.2	1068.1	523.6	1284.3	0.0052	0.0048	0.0004	0.77864
6	1216.0	1067.3	523.2	1283.3	0.0052	0.0048	0.0004	0.77924
7	1215.8	1066.5	522.8	1282.3	0.0052	0.0048	0.0004	0.77985
8	1215.7	1065.7	522.3	1281.3	0.0052	0.0048	0.0004	0.78045
9	1215.5	1064.8	521.9	1280.3	0.0052	0.0048	0.0004	0.78105
10	1215.3	1064.0	521.5	1279.3	0.0052	0.0048	0.0004	0.78165
11	1215.2	1063.2	521.1	1278.4	0.0052	0.0048	0.0004	0.78225
12	1215.0	1062.4	520.7	1277.4	0.0052	0.0048	0.0004	0.78285
13	1214.8	1061.6	520.3	1276.4	0.0052	0.0048	0.0004	0.78346
14	1214.7	1060.8	519.9	1275.4	0.0052	0.0048	0.0004	0.78406
15	1214.5	1059.9	519.5	1274.4	0.0052	0.0048	0.0004	0.78466
16	1214.3	1059.1	519.1	1273.5	0.0052	0.0048	0.0004	0.78526
17	1214.2	1058.3	518.7	1272.5	0.0052	0.0048	0.0004	0.78586
18	1214.0	1057.5	518.4	1271.5	0.0052	0.0048	0.0004	0.78646
19	1213.8	1056.7	518.0	1270.5	0.0052	0.0048	0.0004	0.78707
20	1213.7	1055.9	517.6	1269.6	0.0052	0.0048	0.0004	0.78767

Table A1.2 The Variation of COP with Condensator Temperature T_8 as
a Parameter

T_8 °C	Q_A J	Q_C J	Q_E J	Q_S J	$\dot{m}_3$ kg/s	$\dot{m}_4$ kg/s	$\dot{m}_{10}$ kg/s	COP
20	1165.7	1065.8	501.5	1231.5	0.0050	0.0046	0.0004	0.81204
21	1167.7	1065.8	502.4	1233.5	0.0050	0.0046	0.0004	0.81067
22	1169.7	1065.9	503.3	1235.6	0.0050	0.0046	0.0004	0.80931
23	1171.8	1066.0	504.1	1237.7	0.0050	0.0046	0.0004	0.80794
24	1173.8	1066.0	505.0	1239.8	0.0051	0.0046	0.0004	0.80658
25	1175.8	1066.1	505.9	1241.9	0.0051	0.0046	0.0004	0.80521
26	1177.9	1066.1	506.8	1244.0	0.0051	0.0047	0.0004	0.80384
27	1179.9	1066.2	507.6	1246.1	0.0051	0.0047	0.0004	0.80248
28	1182.0	1066.3	508.5	1248.3	0.0051	0.0047	0.0004	0.80111
29	1184.1	1066.3	509.4	1250.4	0.0051	0.0047	0.0004	0.79975
30	1186.1	1066.4	510.3	1252.5	0.0051	0.0047	0.0004	0.79838
31	1188.2	1066.5	511.2	1254.7	0.0051	0.0047	0.0004	0.79701
32	1190.3	1066.5	512.1	1256.8	0.0051	0.0047	0.0004	0.79565
33	1192.4	1066.6	513.0	1259.0	0.0051	0.0047	0.0004	0.79428
34	1194.5	1066.6	513.9	1261.2	0.0051	0.0047	0.0004	0.79292
35	1196.6	1066.7	514.8	1263.3	0.0051	0.0047	0.0004	0.79155
36	1198.8	1066.8	515.7	1265.5	0.0051	0.0047	0.0004	0.79018
37	1200.9	1066.8	516.6	1267.7	0.0051	0.0047	0.0004	0.78882
38	1203.0	1066.9	517.6	1269.9	0.0051	0.0047	0.0004	0.78745
39	1205.2	1067.0	518.5	1272.1	0.0051	0.0047	0.0004	0.78608

Table A1.2 Continued

T_8 °C	Q_A J	Q_C J	Q_E J	Q_S J	$\dot{m}$ kg/s	$\dot{m}$ kg/s	$\dot{m}$ kg/s	COP
40	1207.3	1067.0	519.4	1274.4	0.0052	0.0048	0.0004	0.78471
41	1209.5	1067.1	520.3	1276.6	0.0052	0.0048	0.0004	0.78335
42	1211.6	1067.2	521.3	1278.8	0.0052	0.0048	0.0004	0.78198
43	1213.8	1067.2	522.2	1281.0	0.0052	0.0048	0.0004	0.78961
44	1216.0	1067.3	523.2	1283.3	0.0052	0.0048	0.0004	0.77924
45	1218.2	1067.4	524.1	1285.6	0.0052	0.0048	0.0004	0.77788
46	1220.4	1067.4	525.0	1287.8	0.0052	0.0048	0.0004	0.77651
47	1222.6	1067.5	526.0	1290.1	0.0052	0.0048	0.0004	0.77514
48	1224.8	1067.6	526.9	1292.4	0.0052	0.0048	0.0004	0.77377
49	1227.0	1067.6	527.9	1294.7	0.0052	0.0048	0.0004	0.77240
50	1229.2	1067.7	528.9	1297.0	0.0052	0.0048	0.0004	0.77103

Table A1.3 The Variation of COP with Generator Temperature as a
Parameter

T_7 °C	Q_A J	Q_C J	Q_E J	Q_S J	$\dot{m}_3$ kg/s	$\dot{m}_4$ kg/s	$\dot{m}_{10}$ kg/s	COP
60	1160.4	1042.5	234.8	1202.9	0.0052	0.0048	0.0004	0.83131
61	1162.2	1043.4	244.4	1205.6	0.0052	0.0048	0.0004	0.82946
62	1164.1	1044.2	254.1	1208.3	0.0052	0.0048	0.0004	0.82763
63	1165.9	1045.0	263.7	1211.0	0.0052	0.0048	0.0004	0.82579
64	1167.8	1045.8	273.3	1213.6	0.0052	0.0048	0.0004	0.82397
65	1169.7	1046.7	282.9	1216.3	0.0052	0.0048	0.0004	0.82216
66	1171.5	1047.5	292.5	1219.0	0.0052	0.0048	0.0004	0.82035
67	1173.4	1048.3	302.1	1221.7	0.0052	0.0048	0.0004	0.81855
68	1175.2	1049.1	311.7	1224.4	0.0052	0.0048	0.0004	0.81676
69	1177.1	1050.0	321.3	1227.0	0.0052	0.0048	0.0004	0.81498
70	1178.9	1050.8	330.9	1229.7	0.0052	0.0048	0.0004	0.81320
71	1180.8	1051.6	340.6	1232.4	0.0052	0.0048	0.0004	0.81143
72	1182.6	1052.4	350.2	1235.1	0.0052	0.0048	0.0004	0.80967
73	1184.5	1053.3	359.8	1237.7	0.0052	0.0048	0.0004	0.80792
74	1186.3	1054.1	369.4	1240.4	0.0052	0.0048	0.0004	0.80617
75	1188.2	1054.9	379.0	1243.1	0.0052	0.0048	0.0004	0.80444
76	1190.0	1055.7	388.6	1245.8	0.0052	0.0048	0.0004	0.80271
77	1191.9	1056.6	398.2	1248.5	0.0052	0.0048	0.0004	0.80098
78	1193.7	1057.4	407.8	1251.1	0.0052	0.0048	0.0004	0.79927
79	1195.6	1058.2	417.4	1253.8	0.0052	0.0048	0.0004	0.79756

Table A1.3 Continued

T_7 °C	Q_A J	Q_C J	Q_E J	Q_S J	$\dot{m}_3$ kg/s	$\dot{m}_4$ kg/s	$\dot{m}_{10}$ kg/s	COP
80	1197.5	1059.0	427.0	1256.5	0.0052	0.0048	0.0004	0.79586
81	1199.3	1059.9	436.7	1259.2	0.0052	0.0048	0.0004	0.79417
82	1201.2	1060.7	446.3	1261.9	0.0052	0.0048	0.0004	0.79248
83	1203.0	1061.5	455.9	1264.5	0.0052	0.0048	0.0004	0.79080
84	1204.9	1062.4	465.5	1267.2	0.0052	0.0048	0.0004	0.78913
85	1206.7	1063.2	475.1	1269.9	0.0052	0.0048	0.0004	0.78746
86	1208.6	1064.0	484.7	1272.6	0.0052	0.0048	0.0004	0.78581
87	1210.4	1064.8	494.3	1275.3	0.0052	0.0048	0.0004	0.78416
88	1212.3	1065.7	503.9	1277.9	0.0052	0.0048	0.0004	0.78251
89	1214.1	1066.5	513.5	1280.6	0.0052	0.0048	0.0004	0.78087
90	1216.0	1067.3	523.2	1283.3	0.0052	0.0048	0.0004	0.77924
91	1217.8	1068.1	532.8	1286.0	0.0052	0.0048	0.0004	0.77762
92	1219.7	1069.0	542.4	1288.7	0.0052	0.0048	0.0004	0.77600
93	1221.5	1069.8	552.0	1291.3	0.0052	0.0048	0.0004	0.77439
94	1223.4	1070.6	561.6	1294.0	0.0052	0.0048	0.0004	0.77279
95	1225.3	1071.4	571.2	1296.7	0.0052	0.0048	0.0004	0.77119

Table Al.4 The Variation of COP with Concentration X_2 as
a parameter

X_2	COP
0.50000	0.81146
0.50500	0.81053
0.51000	0.80955
0.51500	0.80851
0.52000	0.80742
0.52500	0.80627
0.53000	0.80508
0.53500	0.80383
0.54000	0.80253
0.54500	0.80118
0.55000	0.79978
0.55500	0.79833
0.56000	0.79683
0.56500	0.79529
0.57000	0.79369
0.57500	0.79205
0.58000	0.79036
0.58500	0.78863
0.59000	0.78685
0.59500	0.78503
0.60000	0.78316
0.60500	0.78125
0.61000	0.77929
0.61500	0.77730
0.62000	0.77526
0.62500	0.77319
0.63000	0.77107
0.63500	0.76891
0.64000	0.76672
0.64500	0.76449

APPENDIX A2

The results of a computer program which calculates the useful heat energy that can be obtained from flat plate collectors. The description of symbols used in Appendix are shown below.

ϕ :Collector tilt angle

b_z :The wall azimuth angle.

APPENDIX A2.The Resultsof Computer Program to Calculate Useful

Heat Energy from a Flat Plate Collector

Table A2.1 Useful Heat Gain (W/m^2) for $b_z=30^\circ$

Time	$\phi=20^\circ$	$\phi=30^\circ$	$\phi=40^\circ$	$\phi=50^\circ$
6.30	263.619	59.371	46.639	34.569
7.00	330.604	164.803	135.173	106.145
7.30	389.898	257.524	217.511	177.298
8.00	451.679	341.430	294.874	246.799
8.30	518.444	419.091	367.959	314.332
9.00	580.075	490.561	436.614	379.205
9.30	635.745	555.430	500.198	440.522
10.00	684.635	613.054	557.888	497.309
10.30	725.982	662.694	608.813	548.585
11.00	759.105	703.610	652.118	593.405
11.30	783.428	735.115	687.005	630.881
12.00	798.494	756.608	712.757	660.199
12.30	803.973	767.585	728.748	680.626
13.00	799.663	767.640	734.433	691.501
13.30	785.492	756.448	729.339	692.217
14.00	761.505	733.726	713.019	682.179
14.30	727.844	699.162	684.974	660.721
15.00	684.722	652.273	644.504	626.949
15.30	632.369	592.130	590.409	579.427
16.00	570.949	516.824	520.347	515.497
16.30	509.762	422.277	429.339	429.687
17.00	445.188	297.647	305.925	309.631
17.30	365.325	121.194	126.054	129.208

Table A2.2 Useful Heat Gain (W/m^2) for $b_z=40^\circ$

Time	$\phi=20^\circ$	$\phi=30^\circ$	$\phi=40^\circ$	$\phi=50^\circ$
6.30	65.451	50.244	35.295	21.057
7.00	179.694	145.428	110.786	77.093
7.30	279.105	233.726	187.309	141.265
8.00	367.773	316.338	262.682	208.437
8.30	448.602	394.155	336.195	276.485
9.00	522.238	467.000	406.924	343.835
9.30	588.474	534.211	473.802	409.084
10.00	646.747	594.932	535.721	470.915
10.30	696.375	648.255	591.582	528.079
11.00	736.662	693.297	640.339	579.399
11.30	766.968	729.236	681.017	623.778
12.00	786.738	755.341	712.729	660.198
12.30	795.522	770.979	734.679	687.728
13.00	792.970	775.612	746.159	705.505
13.30	778.821	768.781	746.523	712.723
14.00	752.865	750.062	735.147	708.572
14.30	714.871	718.987	711.339	692.158
15.00	664.446	674.890	674.174	662.319
15.30	600.770	616.605	622.162	617.273
16.00	522.009	541.812	552.537	553.858
16.30	423.980	445.514	459.570	465.721
17.00	296.768	316.180	330.294	338.682
17.30	119.775	129.659	137.386	142.719

Table A2.3 Useful Heat Gain (W/m^2) for $b_z=50^\circ$

Time	$\phi=20^\circ$	$\phi=30^\circ$	$\phi=40^\circ$	$\phi=50^\circ$
6.30	60.244	42.632	25.509	9.396
7.00	168.768	129.061	89.746	52.019
7.30	265.218	213.425	161.212	110.165
8.00	352.989	294.726	234.899	175.328
8.30	434.016	372.833	308.785	243.819
9.00	508.607	447.073	381.307	313.308
9.30	576.358	516.499	451.033	381.951
10.00	636.676	580.062	516.605	448.135
10.30	688.472	636.702	576.731	510.380
11.00	731.264	685.406	630.196	567.312
11.30	764.231	725.234	675.783	617.648
12.00	786.738	755.341	712.729	660.198
12.30	798.259	774.980	739.823	693.857
13.00	798.367	783.502	756.302	717.593
13.30	786.724	780.334	761.374	730.421
14.00	763.037	764.932	754.262	731.352
14.30	726.986	736.698	734.107	719.292
15.00	678.077	694.817	699.791	692.847
15.30	615.355	637.927	649.573	649.939
16.00	536.793	563.424	580.320	586.968
16.30	437.867	465.815	485.667	496.821
17.00	307.964	332.547	351.335	363.756
17.30	124.982	137.271	147.171	154.381

Table A2.4 Useful Heat Gain (W/m^2) for $b_z = 60^\circ$

Time	$\phi=20^\circ$	$\phi=30^\circ$	$\phi=40^\circ$	$\phi=50^\circ$
6.30	56.021	36.459	17.574	0.061
7.00	159.689	115.787	72.683	31.685
7.30	253.957	196.962	140.048	84.944
8.00	341.000	277.199	212.368	148.477
8.30	422.188	355.541	286.555	217.329
9.00	497.553	430.913	360.532	288.551
9.30	566.533	502.136	432.569	359.947
10.00	628.327	568.003	501.103	429.661
10.30	682.063	627.333	564.687	496.027
11.00	726.887	679.008	621.970	557.509
11.30	762.011	721.990	671.702	612.677
12.00	786.738	755.341	712.729	660.198
12.30	800.479	778.225	743.994	698.828
13.00	802.744	789.901	764.528	727.396
13.30	793.133	789.703	773.418	744.774
14.00	771.286	776.991	769.764	749.826
14.30	736.811	751.062	752.572	741.296
15.00	689.132	710.977	720.565	717.603
15.30	627.184	655.219	671.802	676.430
16.00	548.783	580.951	602.851	613.819
16.30	449.128	482.278	506.831	522.042
17.00	317.044	345.820	368.398	384.091
17.30	129.205	143.445	155.107	163.838

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