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THE DEVELOPMENT OF A RESIDENTIAL HEATING AND COOLING SYSTEM USING NASA-DERIVED TECHNOLOGY

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FOREWORD

This report represents the results of work performed by the Lockheed-Huntsville Research & Engineering Center for the NASA-Marshall Space Flight Center, Alabama, under Exhibit A of Contract NAS8-25986 (Mod. 5).

The NASA contract monitor for this study was Mr. R.L. Middleton of the MSFC Astronautics Laboratory.

SUMMARY

This report contains the results of a study to determine the technical and economic feasibility of a solar-powered space heating, air-conditioning, and hot water heating system for residential applications. The basic system utilizes a flat-plate solar collector to process incident solar radiation, a thermal energy storage system to store the collected energy for use during night and heavily overcast periods, and an absorption cycle heat pump for actually heating and cooling the residence. In addition, heat from the energy storage system is used to provide domestic hot water.

The basic system represents a direct extension of technology developed during several space-related engineering studies conducted by Lockheed and funded by NASA-Marshall Space Flight Center. In one such study, Lockheed conducted extensive analytical and empirical investigations of an absorption cycle environmental control system for the Space Station. In other studies, Lockheed thoroughly investigated phase change materials (PCMs) for high-capacity energy storage systems. In still other NASA-funded studies, Lockheed has developed sophisticated computational tools for thermodynamic and energy transfer analyses of thermal control systems. Thus, the solar-powered residential system is a direct and natural spin-off from these previous space-related projects.

This solar-powered system for residential application offers several significant economic and ecological benefits to the nation as a whole and to the average American citizen. Several of these benefits are summarized below.

- Because the system is powered almost entirely by solar energy, the energy expenditures in this nation for space heating, air-conditioning, and water heating will be greatly reduced when the system is widely adopted. Currently, 25% of all the energy used

in the United States is expended to meet these requirements. Thus, the system will have an important, favorable impact on the national energy crisis.

- Since widespread adoption of the system will reduce the need for conventional energy sources for providing heating, cooling, and water heating services, the pollution byproducts of conventional energy production will be reduced. These pollution byproducts include air (gaseous, particulate, thermal), water, radiation, solid waste and radioactive waste pollution.
- The replacement of conventional energy requirements by solar energy utilization will help preserve fossil fuel reserves, will help prevent environmental destruction due to mining, and will reduce this nation's dependence on foreign nations for petroleum.
- Since solar energy is free, the American citizen will realize tremendous economic savings in heating, cooling, and water heating for his home and business.

This report contains a detailed description of the solar-powered system. The analyses of the three major components of the system (the solar collector, the energy storage system, and the heat pump package) are discussed and results are presented. The total system analysis is discussed in detail, including the technical performance of the solar-powered system and a cost comparison between the solar-powered system and a conventional system. The projected applicability of the system to different regions of the nation is described.

The primary conclusion of the study to date is that the system is technically and economically feasible, and that a prototype demonstration unit should be fabricated and tested as quickly as possible. Further conclusions and detailed recommendations are also presented.

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NOMENCLATURE

Symbol	Description
A	area
Area	collector area
d	distance between plates
DME-TEG	dimethyl ether of tetraethylene glycol
E_{max}, $E_{max\ stored}$	maximum energy storage
g	acceleration of gravity
$\bar{h}$	convective heat transfer coefficient
$\Delta h_{fg_{refrigerant}}$	latent heat of vaporization of refrigerant
k	thermal conductivity
L	length
$\dot{m}$	mass flow rate
ΔP	pressure difference
Pr	Prandtl number
Q	energy exchange
$\dot{Q}$	energy exchange rate
R-21	refrigerant 21
R-22	refrigerant 22
Re_L	Reynolds number based on length
T	temperature
ΔT	temperature difference or change
U	unit conductance
UA	overall heat transfer coefficient
v	specific volume
X	refrigerant concentration in solution with absorbent

Greek	Description
α	absorptivity
α_s	solar absorptivity
β	coefficient of performance ($\dot{Q}_{cooling}/\dot{Q}_{gen}$)
ϵ	emissivity
ϵ_{IR}	infrared emissivity
η	efficiency ($\dot{Q}_{heating}/\dot{Q}_{gen}$)
ν	kinematic viscosity
ϕ	tilt angle
σ	Stefan-Boltzmann constant

Subscripts

a	absorber
aux	auxiliary
c	condenser
e	evaporator
g	generator
gen, gen exit	generator exit
IR	infrared
max	maximum
mean	mean value
opt	optimum
s	solar

Section 1
INTRODUCTION

One of the most pressing national problems today is the increasingly publicized energy crisis. America's ever-growing demand for energy is outpacing the nation's ability to produce energy. Yearly, the threat of fuel shortages and electrical power shortages (brownouts and blackouts) becomes more real in nearly every region of the country. The byproducts of the energy crisis are themselves critical national problems:

- Increased air, water, thermal, radiation, and solid waste pollution;
- Natural resource problems: rapid depletion of fossil fuel reserves, environmental destruction due to mining, and dependence upon foreign nations for oil; and
- Constantly rising costs for the consumer of energy.

In the midst of this energy crisis, it is somewhat ironic that there remains a virtually untapped energy source which is totally pollution-free, impossible to deplete, available in quantities sufficient to power the world indefinitely, and free of charge. This energy source is, of course, solar energy.

The present lack of exploitation of solar energy is attributable to two basic characteristics of solar radiation:

- The relatively low concentration of solar radiation reaching the Earth's surface makes its conversion to useful energy difficult and expensive.
- The intermittency and inconstancy of the solar radiation reaching the Earth's surface makes energy storage necessary in any application where continuous output is required. Such energy storage is difficult and expensive.

In the past, the widespread availability of conventional energy sources and their nominal costs made solar energy exploitation unnecessary and economically unattractive. However, several recent developments have substantially altered this situation:

- Conventional energy sources are hard-pressed to meet the growing national demands for energy. Thus, an energy crisis has emerged.
- The public has become increasingly alarmed over the pollution which accompanies conventional energy production. Ecological concerns are now as important as economical concerns to many Americans.
- The limited nature of natural resources is finally being realized by a large segment of the populace. The predicted lifetimes of American natural gas and oil reserves and the growing public bitterness over practices such as the strip-mining of coal attest to the necessity of finding alternate energy sources.
- The cost of conventional energy is soaring at an unprecedented rate, making solar energy exploitation economically attractive for the first time in history.
- The vast store of new technology currently available makes the efficient and economical exploitation of solar energy more realizable today than ever before. A great deal of this new technology is directly attributable to the American space program.

Thus the solar energy question is no longer whether or not it should be exploited, but rather how it should be exploited. There are two basic avenues of solar energy exploitation, as described below:

- Large scale solar-powered facilities must be developed to produce electricity and high temperature thermal energy for industrial processes. Such facilities could directly replace conventional facilities. These large scale solar-powered installations will require large investments of time and money until they ultimately emerge as practical, everyday solutions to the nation's energy problems.
- Small scale solar-powered installations must be developed to meet the nation's large demand for moderate temperature thermal energy to provide residential and commercial heating, air-conditioning, and hot water heating. Such small scale systems can be developed quickly and economically and would

have a large effect on the nation's energy troubles, since 25% of all the energy consumed in this country is used to provide space heating, air-conditioning, and water heating.

Lockheed-Huntsville began investigating small scale solar-powered systems for residential applications in 1971. These preliminary studies were direct and natural extensions of NASA-funded, space-related studies conducted by Lockheed. In one such study, Lockheed has conducted extensive analytical and experimental investigations of an absorption cycle environmental control system for the Space Station. An absorption cycle system is powered primarily by thermal energy rather than electrical energy, which is required by most other environmental control systems. In the Space Station application, the input thermal energy requirement will be met by waste heat from onboard equipment. A natural extrapolation of this concept would be to use solar energy to power an environmental control system for a home. Thus, Lockheed used the tools developed to analyze and design the Space Station system to investigate a residential solar-powered heating, cooling, and water heating system. The results of these preliminary analyses were so favorable that NASA-MSFC awarded Lockheed a contract in June 1972, to perform more detailed analyses of this solar-powered system.

The objectives of this study were to determine the economical and technical feasibility of the concept and, based upon the results of this feasibility study, to decide whether or not a full-scale prototype system should be fabricated and tested. Additional objectives of the study were to perform preliminary system design, to initiate system optimization, and to formulate plans for the full-scale demonstration program if the system proved feasible.

Throughout the entire study, NASA-MSFC made significant technical contributions through in-house efforts. These efforts were concentrated in the following areas:

- Development of a selective coating for the absorber plate of the solar collector. (The result of this effort was an economical coating with excellent thermal properties.)

- Experimental evaluation of thermal energy storage materials, including effects of additives on melt temperatures of phase change materials.
- Fabrication of a 4-foot square solar collector for testing in the near future.
- Widespread literature surveys and data compilation in all areas of solar energy research.

This report contains the results of all contractual efforts to date.

Section 2

OVERALL DESCRIPTION OF SOLAR-POWERED SPACE HEATING, AIR-CONDITIONING, AND HOT WATER HEATING SYSTEM

In November 1971,* Lockheed-Huntsville began investigating the feasibility of developing a system which could collect solar energy economically and efficiently, store this energy for use during the night and on extremely cloudy days, and use this energy to drive an absorption cycle heat pump and also to provide hot water for domestic use. A schematic of the system is presented in Fig. 2-1. Solar energy is collected by a flat-plate solar collector which is capable of utilizing both direct and diffuse solar radiation incident upon it. The collected energy is transferred to a heat transfer fluid which transports the energy to the thermal energy storage system. Here, the energy is stored either as latent heat in a phase change material (PCM) or as sensible heat in a liquid or solid. When heating or air-conditioning is required, energy is transferred from the storage system to the absorption cycle heat pump, which can either heat the inside air or cool and dehumidify the inside air. Also, heat from the thermal energy storage sytem is used to provide hot water for domestic use.

The system depicted in Fig. 2-1, when eventually constructed, will represent several advances over previous solar-powered installations. Some of these advances are listed below.

- The system will utilize solar energy to provide nearly all of the heating, air-conditioning, and water heating requirements of the residence, with the small remainder of these requirements being met with auxiliary energy. In the literature, no current or past system has been found which provides the major portion of all three of these residential requirements by utilizing solar energy. By using the same collector and energy storage system year-round for all three functions, the economy of the new system is greatly improved over previous systems.

* These investigations started as an in-house project and a NASA contract was received in June 1972.

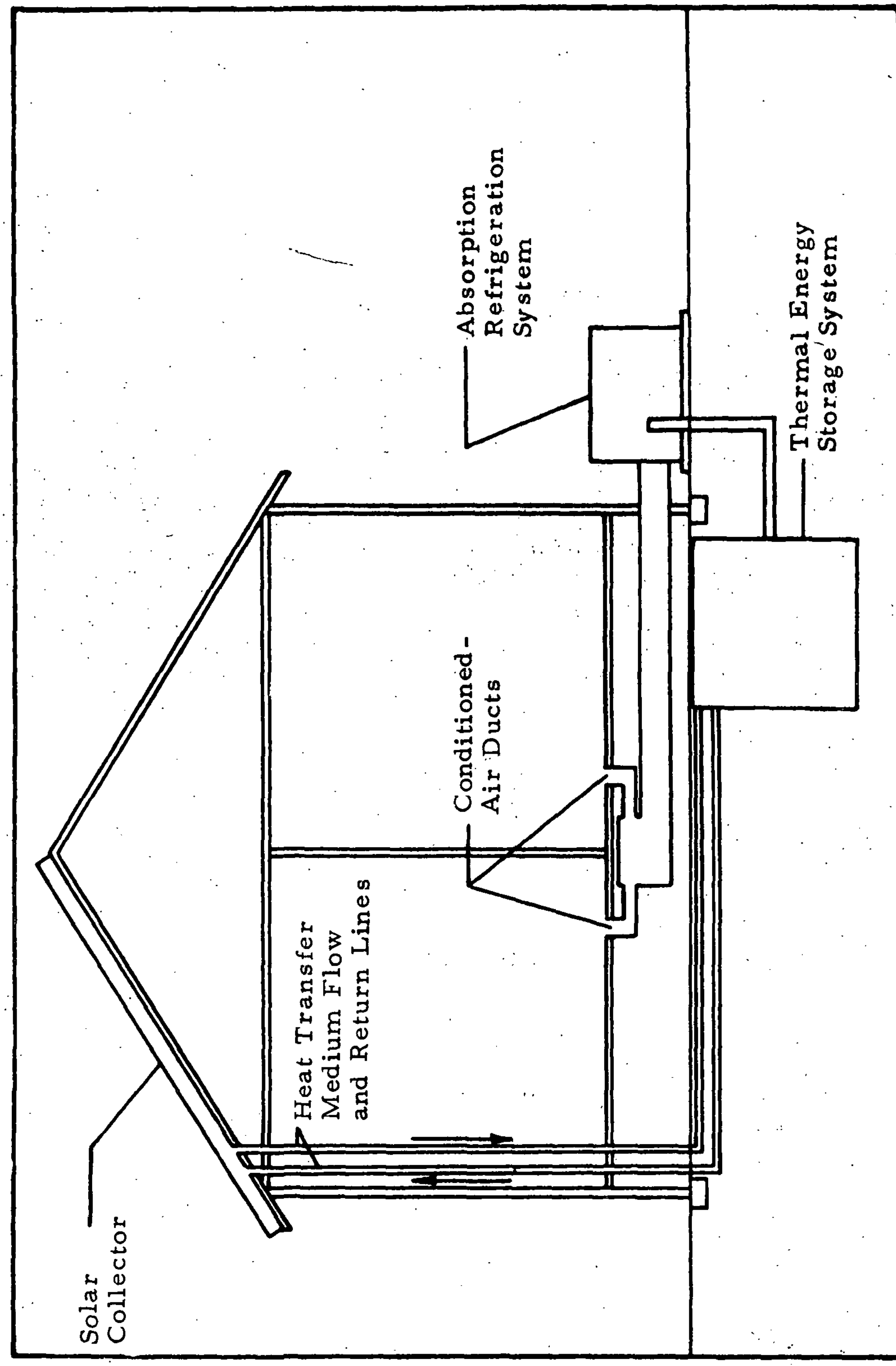


Fig. 2-1 - Schematic of Solar-Powered Space Heating, Air-Conditioning and Hot Water Heating System

- The system will utilize the absorption cycle for both heating and cooling. Current and past systems found in the literature use the absorption cycle only for cooling, while heating is done directly. The benefit of using the absorption cycle for heating is the ability to absorb heat from the ambient air to supplement the solar energy collected, thereby reducing required collector area and energy storage. The reduced requirements for collector area and energy storage will improve the economy of the system.
- The system will be cost-optimized for the particular locale where it will operate. This optimization will be accomplished using the latest analytical techniques, including transient computer thermal analysis of the collector, experimental solar radiation data, and daily inventories of all energy quantities. The optimization will identify the optimum collector area, energy storage system capacity, and operating temperature of the collector for a particular application. This optimization will obviously increase the economy of the system.

From Fig. 2-1, it is apparent that the total system is comprised of three major subsystems* which include the:

- Solar collector,
- Thermal energy storage system, and
- Absorption cycle heat pump.

Each of these three systems was analyzed in detail during the current study. The following sections of this document discuss the analytical studies in detail.

*The hot water heating system will merely be a heat exchanger with the thermal energy storage system and is not considered a major subsystem.

Section 3
SOLAR COLLECTOR ANALYSIS

The function of the solar collector is to efficiently collect solar radiation incident upon it, and to transfer the collected energy to a heat transfer fluid for delivery to the energy storage system. There are two basic types of solar collectors: (1) those which concentrate the incident solar radiation, and (2) those which do not concentrate the incident solar radiation. The relative advantages and disadvantages of each type are given below.

- Concentrating Solar Collectors

Advantages

a. Since the incident solar radiation is concentrated on a small absorber area, thermal losses are small and collection efficiency ($\dot{Q}_{collected}/\dot{Q}_{incident}$) is high

b. Capable of high temperature operation due to high efficiency

Disadvantages

c. Limited to collection of direct component of solar radiation since diffuse component cannot be concentrated

d. No collection on cloudy days

e. Collector must track the sun's movement across the sky

f. Collector design is complex due to mirrors or lenses and tracking system

g. Expensive due to complexity

- Non-Concentrating Solar Collectors

Advantages

a. Able to collect direct and diffuse components of solar radiation

b. Capable of collection on cloudy days

c. Collector can operate in fixed orientation (no tracking necessary)

d. Collector design is simple (insulated flat plate with transparent covers)

e. Cheap due to simplicity.

Disadvantages

a. Since the absorber area is larger for a flat-plate collector than for a concentrating collector, thermal losses are higher and collection efficiency is smaller.

b. Limited to relatively low temperature operation because efficiency drops rapidly at higher temperatures.

For the present application, relatively low temperature operation is acceptable and cost minimization is of primary importance. Therefore, non-concentrating flat plate solar collectors were selected as superior to concentrating solar collectors for the current application. A detailed analytical study of flat plate solar collectors was conducted, as described in the following sections of this report.

3.1 COMPUTER ANALYTICAL MODEL

Solar collectors have been analyzed by many investigators for several decades. The classical methods of analysis rely heavily upon approximations and shortcuts because these methods were developed before computers became available. In the current study, numerous previous analyses were surveyed and all were found unacceptable by comparison to current analytical techniques. The shortcomings of past analyses will be described in later sections of this report.

There are six different energy exchange mechanisms which must be included in the thermal analysis of a flat plate solar collector, as shown in Fig. 3-1. They are listed on page 3-4.

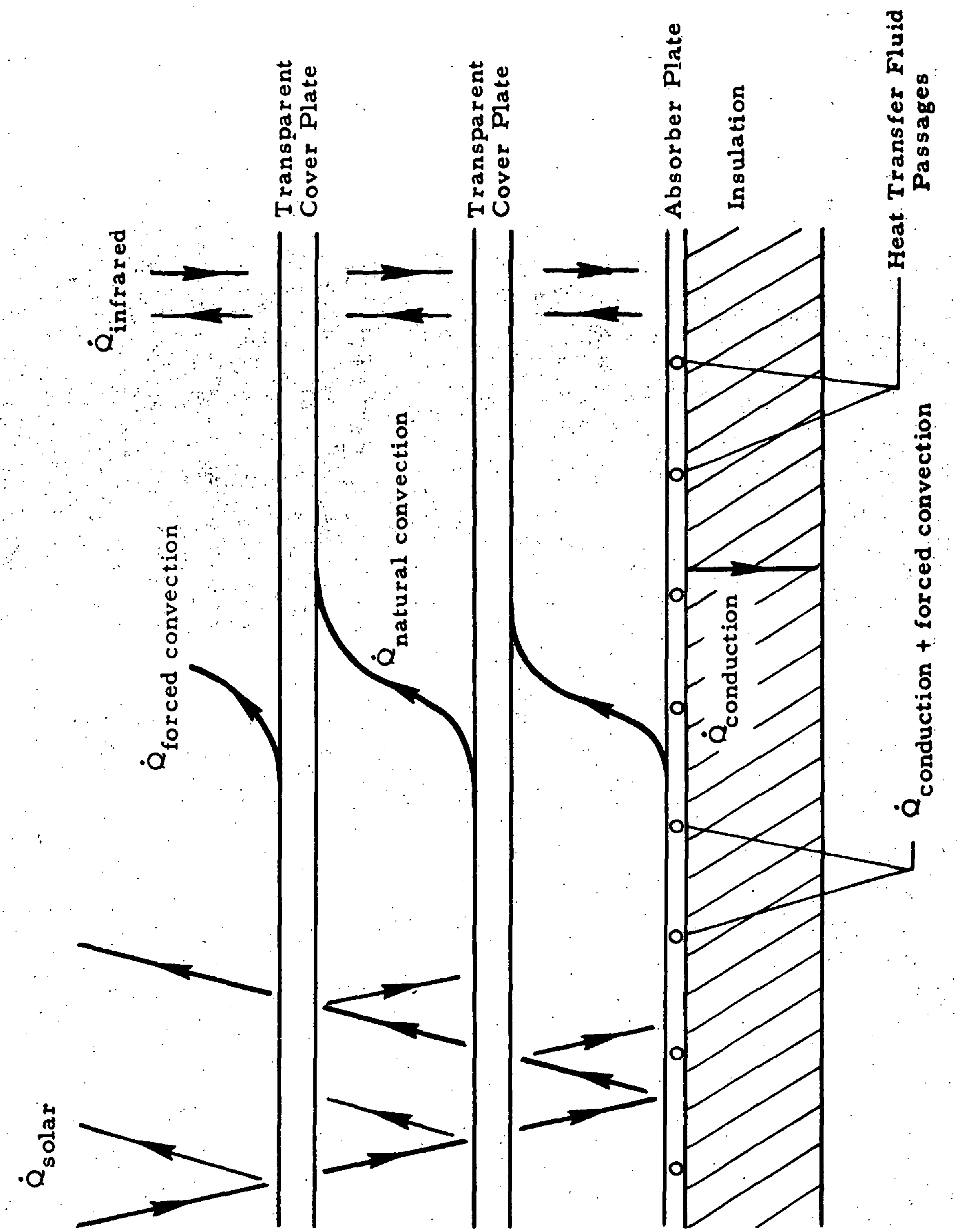


Fig. 3-1 - Energy Exchange Mechanisms in Flat Plate Solar Collector

- Solar radiation energy exchanges ($\dot{Q}_{solar}$), including absorption, transmission, and reflection of this short wavelength radiation
- Infrared radiation energy exchanges ($\dot{Q}_{infrared}$), including emission, absorption, transmission (through plastic covers), and reflection of this long wavelength radiation
- Forced convection between the top transparent cover and the environment ($\dot{Q}_{forced\ convection}$)
- Natural convection between plates ($\dot{Q}_{natural\ convection}$)
- Combined conduction and forced convection of heat from the absorber plate surface to the heat transfer fluid within the passages ($\dot{Q}_{conduction\ +\ forced\ convection}$)
- Conduction in the backside insulation.

Each of these energy exchange mechanisms is discussed separately in the following sections of this report.

3.1.1 Solar Radiation Energy Exchanges

In a classical paper (Ref. 1), Stokes derived the governing equations for solar radiation energy exchanges between multiple transparent plates. These equations treat variations in incident radiation magnitude, angle of incidence, index of refraction, and extinction coefficient for any number of plates. The effects of polarization, which become extremely important when the angle of incidence approaches Brewster's angle, are also treated. The results of Stokes' analysis were incorporated into the solar collector model. It is surprising that other solar collector analysts have avoided using these results, since they represent exact analytical solutions to the governing physical equations; however, most prior analyses have

relied upon simplistic approximations to these results. One consequence of using the approximations often found in the solar energy literature is that transient analysis becomes impractical, and only steady-state calculations can be made.

3.1.2 Infrared Radiation Energy Exchanges

For glass covers, the infrared radiation equations between plates can be easily determined since glass is opaque to this long wavelength radiation. For example, between any two parallel plates, the following equation can be used when edge effects are negligible:

$$(\dot{Q}/A)_{net_{1-2}} = \frac{\sigma(T_1^4 - T_2^4)}{1/\epsilon_1 + 1/\epsilon_2 - 1} .$$

However, for plastic film covers, some of the infrared radiation is transmitted and the above equation is no longer valid. In such cases, the solar collector model utilized a matrix solution of the flux equations to determine all net energy exchanges between plates due to infrared radiation.

3.1.3 Convection Between Top Cover and Environment

The heat transfer coefficient between the top cover and ambient air is a function of plate temperature, air temperature, wind speed, plate dimensions, and plate orientation. For normal wind conditions and solar collector dimensions, the convection process is laminar. The following equation from Ref. 2 was used in the solar collector model:

$$\bar{h}_{external} = \frac{k}{L}\left[0.664 \, Pr^{1/3} \, Re_L^{\ 1/2}\right]$$

where k, Pr and Re_L were treated as temperature-dependent. (The preceeding equation is usually called the Pohlhausen solution.)

An average wind speed of 10 mph was assumed and a dimension of 30 feet was used to describe the collector.

3.1.4 Natural Convection Between Plates

In most previous analyses, some constant unit conductance was assumed between parallel plates. In the current study, however, the actual variations in unit conductance caused by air temperature, plate spacing and plate orientation were included by using curve fits for empirical data, as shown in Fig. 3-2. A plate spacing of 1 inch was assumed and thermodynamic properties of air were treated as temperature-dependent.

3.1.5 Heat Transfer from Absorber Surface to Fluid in Passages

This heat transfer process is a classic example of the fin-tube radiator problem which has been investigated widely. Preliminary calculations revealed that proper tube sizing and spacing could result in negligible temperature gradients over the collector, and, therefore, this heat transfer process was omitted from the model.

3.1.6 Insulation Losses

The heat losses from the back of the collector are minimized by insulating the entire backside area of the collector. Initial calculations revealed that these losses could be made negligible by using a fairly thick volume of cheap, lightweight, loose-fill insulation over the back of the collector. Therefore, these losses were omitted from the analytical model of the collector.

Note: Curves below are curve fits to experimental data of Ref. 3.

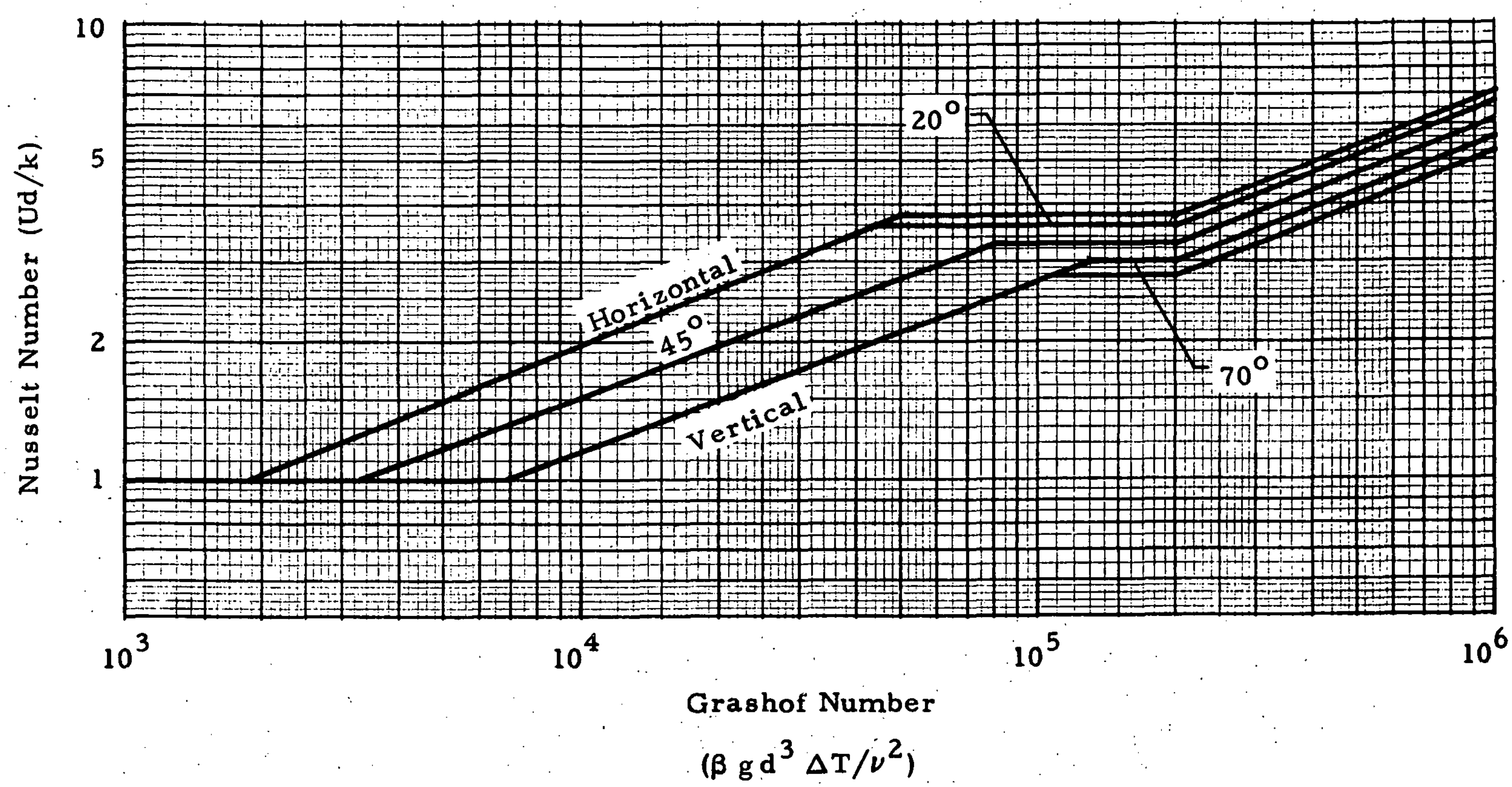


Fig. 3-2 - Natural Convection Between Parallel Plates at Different Tilt Angles for Air

3.1.7 Model Options

The analytical computer model was designed for use in either of the following modes:

- Steady-State Conditions

In this mode of operation, an iterative procedure was used to solve the simultaneous energy transfer equations to yield the temperatures of all collector components and the net energy transfer rates between components. Constant values were input for collector temperature, ambient temperature, incident solar radiation intensity and angle of incidence.

- Transient Conditions

This was the most important mode of operation and is felt to represent an advancement of the state of the art in solar collector analysis. In this mode, the collector operating temperature, ambient temperature, collector orientation, and experimental solar radiation data were input. The program then conducted a transient thermal analysis of the collector treating variations in solar radiation intensity and direction based upon experimental solar data and the astronomy of the sun-earth system. All components of the collector were assumed to be at ambient temperature at sunrise, and the computer program solved numerically the transient energy transfer equations to determine all temperatures and energy transfer rates as functions of time. Since the actual collector will physically operate in the transient mode, only a transient analytical treatment should be used as the basis for designing a solar collector. However, nowhere in the solar energy literature was there found a treatment similar to the current transient analysis.

3.2 PARAMETRIC SOLAR COLLECTOR ANALYSIS

A parametric analysis was conducted to determine the effect on collector performance of the following variables:

- Collector orientation
- Collector temperature
- Selective absorber surface coatings
- Number and type of transparent covers
- Solar environment
- Ambient environment.

The important results of the parametric analysis are discussed in the following sections.

3.2.1 Optimum Collector Orientation

For a fixed orientation flat-plate solar collector operating in the northern hemisphere, the best orientation for receiving a maximum amount of incident solar radiation is southward-facing, with the collector tilted back from the vertical by an angle, ϕ_{opt}, (see Fig. 3-3). This optimum tilt angle (ϕ_{opt}) is a strong function of the time of year since it depends strongly on the earth-sun astronomy. The optimum tilt angle for a solar collector operating at the latitude of Huntsville, Alabama, is shown in Fig. 3-3 as a function of time of year. This curve was generated by integrating the total solar radiation received per day per unit area, and then differentiating this function of ϕ, setting the derivative equal to zero, and solving for the ϕ_{opt} which maximizes radiation received. Several interesting points are presented by this curve. The variation between ϕ_{opt} for 22 June and for 22 December is large, over 70^{o}. This immediately suggests that shifting orientation twice per year, one to yield summer optimal performance and the other to yield winter optimal performance, might improve performance

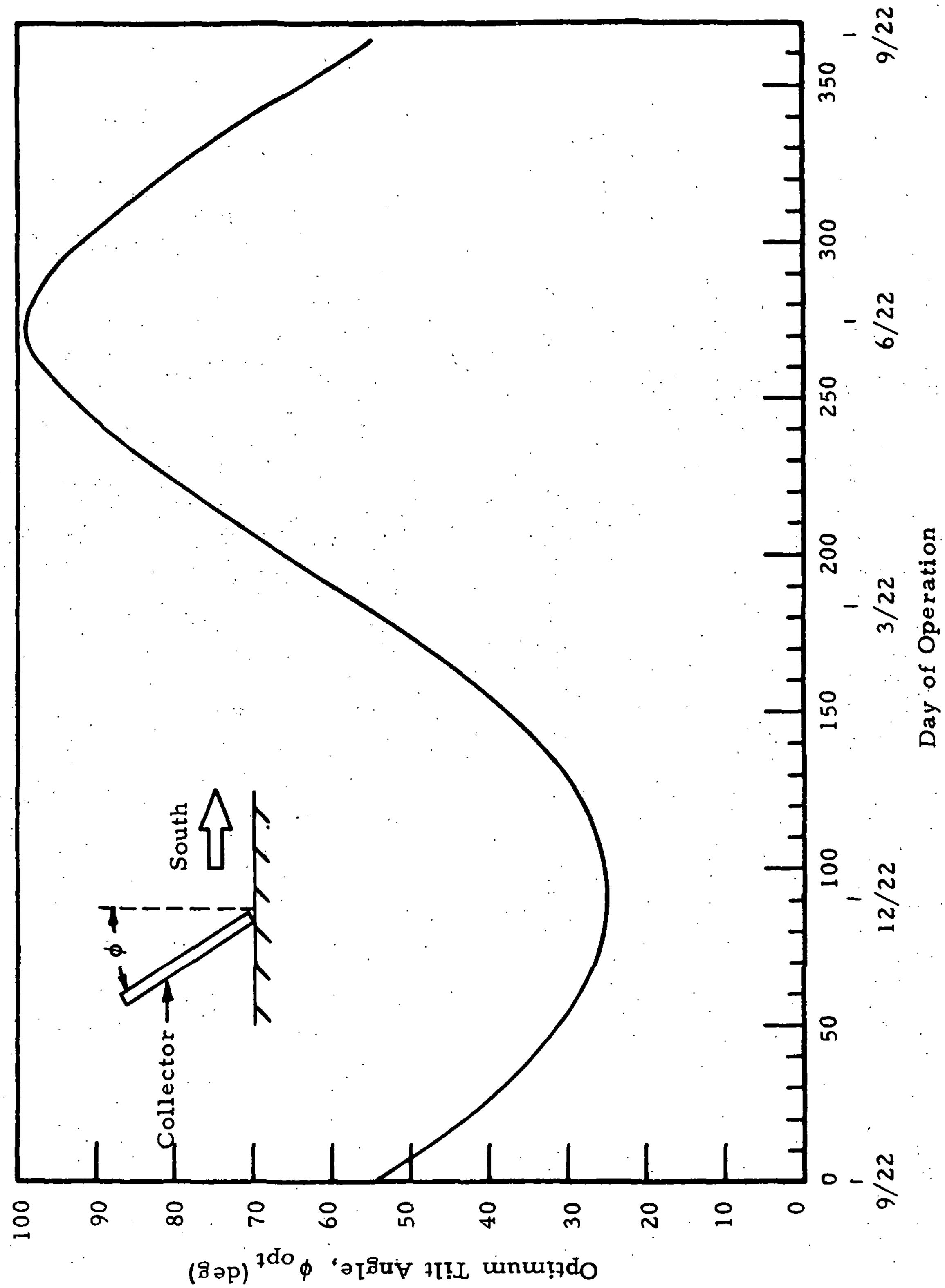


Fig. 3-3 - Optimum Tilt Angle for Solar-Collector at Latitude of Huntsville, Alabama

appreciably. However, in the current study, a single fixed orientation was assumed. Another interesting point is that on 22 June a ϕ_{opt} value greater than 90° is observed. This results from the fact that the sun rises and sets about 23.5° north of east and west, respectively, on this date. Thus the total insolation for the day is greater for a slight northern tilt than for any other value. However, this is somewhat misleading since the atmospheric attenuation of solar radiation is greatest in the early morning and late evening, and the atmospheric absorption was not considered in calculating ϕ_{opt}.

To determine the best ϕ for the entire year from the curve in Fig. 3-3 is difficult. The following points must be considered.

- More energy collection is needed in winter than in summer
- Cloud cover prevails more in winter than in summer
- Days are shorter in winter than in summer
- Heat losses from the collector are greater in winter than in summer due to the lower ambient temperatures.

After some deliberation, a collector tilt angle of 45 degrees was selected for the entire year. This angle is biased in favor of winter collection, but not enough to preclude adequate summer performance. In the total system analysis, described in Section 6, this selection was found to be a good one.

3.2.3 Collector Temperature and Selective Coatings

The surface which ultimately absorbs the solar radiation trapped by the collector can merely be blackened, or it can be treated to selectively absorb solar radiation while emitting very little infrared radiation. The latter surface should perform better, especially at higher temperatures. During the course of this study, NASA-MSFC Materials Laboratory developed a selective coating with excellent properties. Figure 3-4 compares

NOTES:

- Steady State Conditions
- Zero Angle of Incidence
- Solar Radiation = 300 Btu/hr-ft^2
- Two Tedlar Cover Sheets

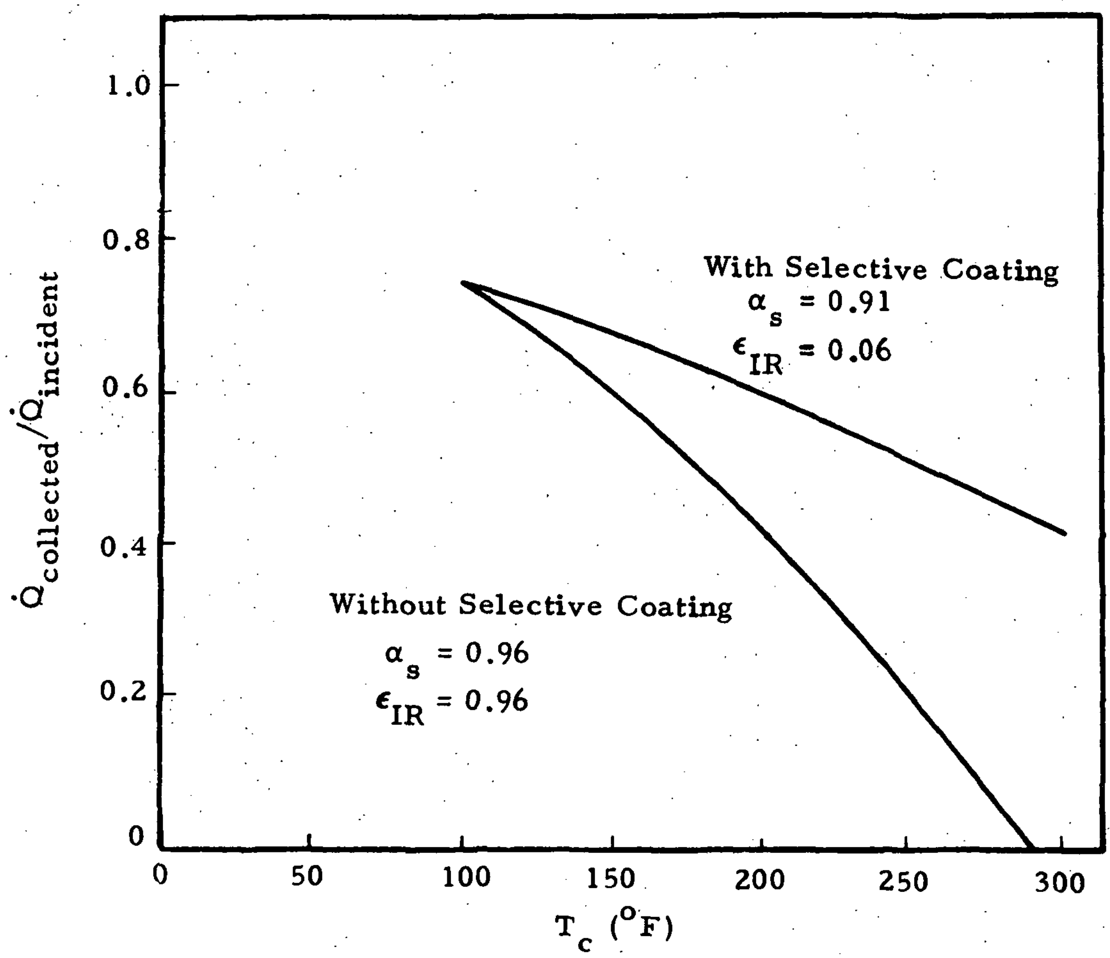


Fig. 3-4 - Effect of NASA-MSFC Developed Selective Coating on Collector Efficiency

a collector utilizing this coating with one utilizing a black surface. The collector efficiency ($\dot{Q}_{collected}/\dot{Q}_{incident}$) is plotted for each collector as a function of collector temperature. In the current application, collector temperatures between 220 and 250°F will be required, as discussed further in Section 6. The benefits of using the selective coating in this temperature range are obvious, since the efficiency is nearly doubled. In addition to the excellent thermal performance of this NASA-developed coating, the cost of applying the coating will be small because of the simple application process developed by Materials Laboratory during the course of this study.

Several other points should be made about Fig. 3-4. The curves were generated using the steady-state option of the computer program. The constant conditions are specified on the figure. Because the conditions do not correspond to the actual transient operation of the collector, these data provide little information about how much energy could be collected on a particular day or about how large a collector should be used to drive the heat pump/water heater system. Thus, these data cannot be used to actually design a solar collector. Ironically, this steady-state presentation of data is all that is found in most of the solar energy literature. This method of presenting data is valuable, however, for comparing different design modifications such as different coatings. Also, this type of data can be generated on a computer for a minimal amount of computer time, while a transient analysis of a collector over the entire year takes several hours of digital computer time. Therefore, in the current study, steady-state analysis was used to compare the effects of different design parameters and transient analysis was used to generate design data for the optimum collector concept.

Because of the obvious benefits of the selective coating and because NASA-MSFC personnel are confident that the coating cost can be reduced to a totally acceptable level, all further collector designs discussed utilize this coating.

3.2.3 Collector Temperature and Transparent Covers

The type and number of transparent covers used in a solar collector greatly influence its performance. Figure 3-5 presents the results of a parametric study of transparent covers. The most important result of this study is that Tedlar, a DuPont polyvinyl flouride plastic film, is superior to glass for any number of covers and any temperature. Fortunately, Tedlar is also cheaper than glass. Thus, Tedlar was chosen as the transparent cover material. The Tedlar considered in this analysis is 0.004 inches thick.

Another important point made by Fig. 3-5 is that two Tedlar covers are superior to one or three such covers for the temperature range of interest (220°F and slightly higher). Thus, the solar collector design from this point on is based upon using two Tedlar cover sheets.

3.2.4 Selected Collector Concept

Based upon the parametric studies described previously, the collector concept presented in Fig. 3-6 was selected as optimum. This collector utilizes the selective coating and two Tedlar covers. The most economical method of manufacturing this type of collector will probably be to modularize panels as shown. The heat transfer fluid passages will probably be integrated into the plate, rather than being tubes as shown.

To demonstrate the performance of the selected collector concept, NASA-MSFC is currently constructing a 4-foot square test model. This model will be tested in the near future to evaluate the accuracy of the analytical performance predictions. The model will utilize the NASA-developed selective coating and two Tedlar covers, as shown in Fig. 3-6.

For the remainder of the collector analysis, a fixed-orientation flat-plate collector designed as shown in Fig. 3-6 was used. The tilt angle was set at 45 degrees, for reasons previously discussed. The effects of solar environment and ambient environment were determined in the transient analysis described in the following section.

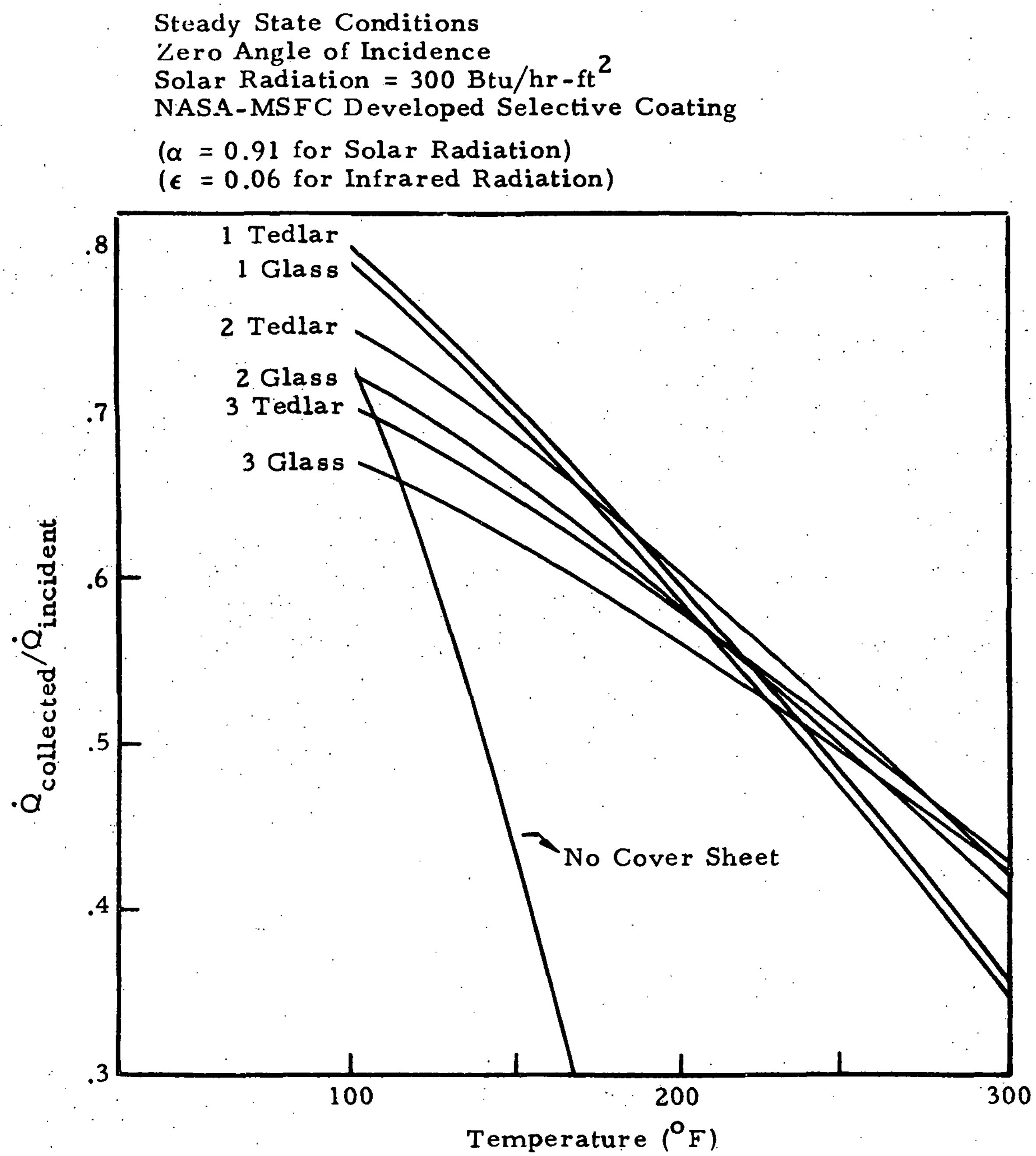


Fig. 3-5 - Solar Collector Performance for Different Cover Materials

NOTE: Collector panels are "modularized" in sections.

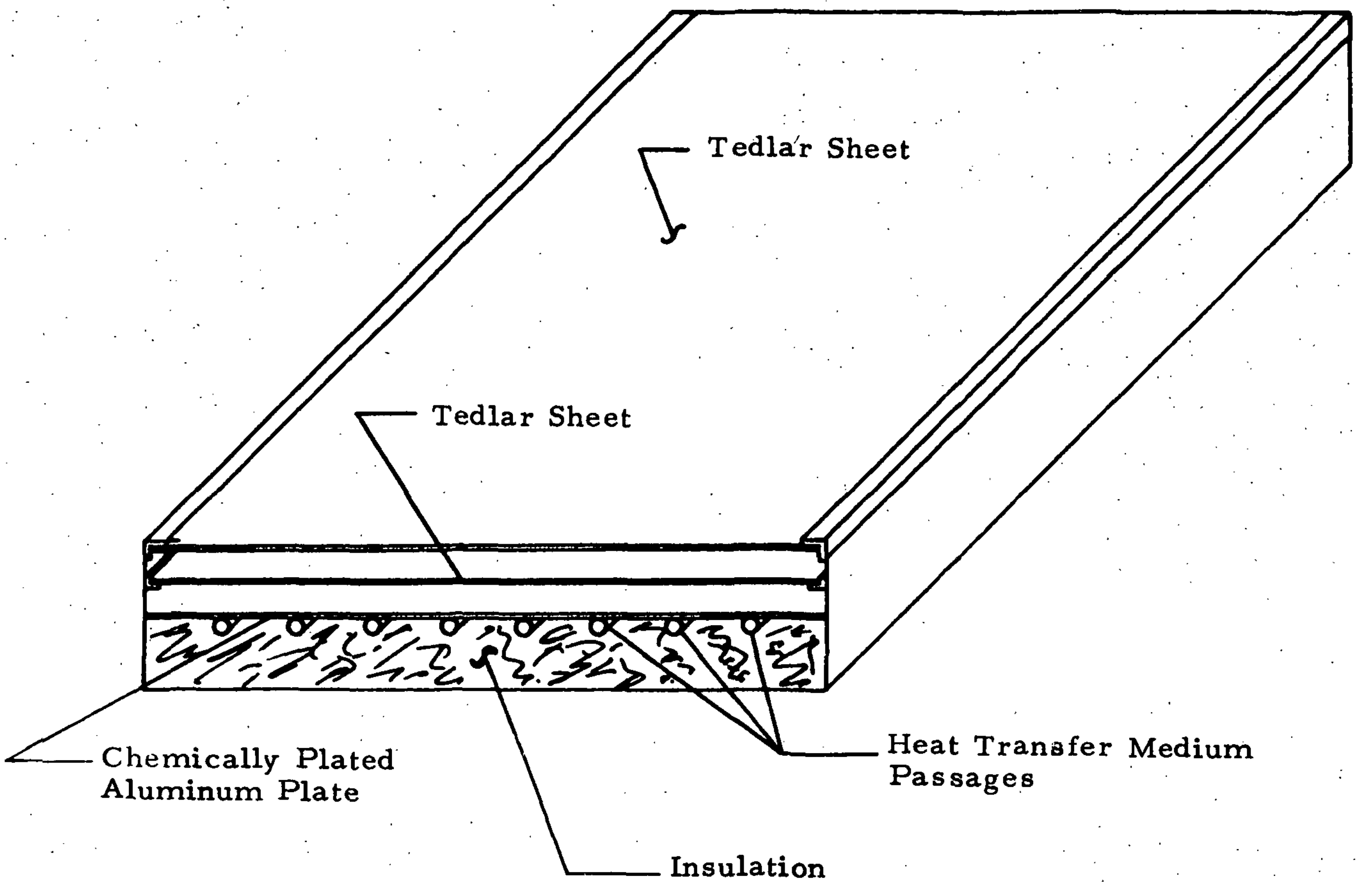


Fig. 3-6 - Selected Collector Concept

3.2.5 Transient Analysis of Selected Collector Concept

To actually design a solar-powered heat pump/water heating system using the selected solar collector concept, design data must be generated for the collector which provides the following information:

$$\left(\frac{\dot{Q}}{A}\right)_{collected} = \text{function (Day, } T_{collector}) .$$

The effects of solar radiation variations and ambient temperature variations must be included in generating these data. In the current study, a transient analysis was conducted for each day of the year using the computer program described previously. Experimental solar radiation measurements for each day of the year were obtained from the U.S. Weather Bureau for 1971 for Atlanta, Georgia. This is the closest location where the measurements are made which has a latitude nearly the same as Huntsville's latitude. These solar data are in the form of whole day totals for a horizontal flat plate. To make the data useful for a minute-by-minute transient analysis for a plate not horizontal in orientation, the data were recorrelated as follows:

- The theoretical whole-day total for a horizontal flat plate was calculated for each day of the year, based upon the solar constant unattenuated by atmospheric absorption.
- The measured whole-day total was divided by the theoretical whole-day total to obtain the atmospheric transmission factor for that day. Thus, the effects of cloud cover, atmospheric absorption, air pollution, etc., were lumped into this transmission factor which was different for each day of the year.
- The solar radiation incident upon the solar collector was calculated minute by minute, based upon the solar constant, and then multiplied by the atmospheric transmission factor for that day. Thus, time-dependent incident solar radiation was used in the transient analysis.

Seasonal variations in ambient temperature also were included in the analysis, as presented in Fig. 3-7. This ambient temperature was used for heat loss calculations, both radiative and convective, from the collector to the environment.

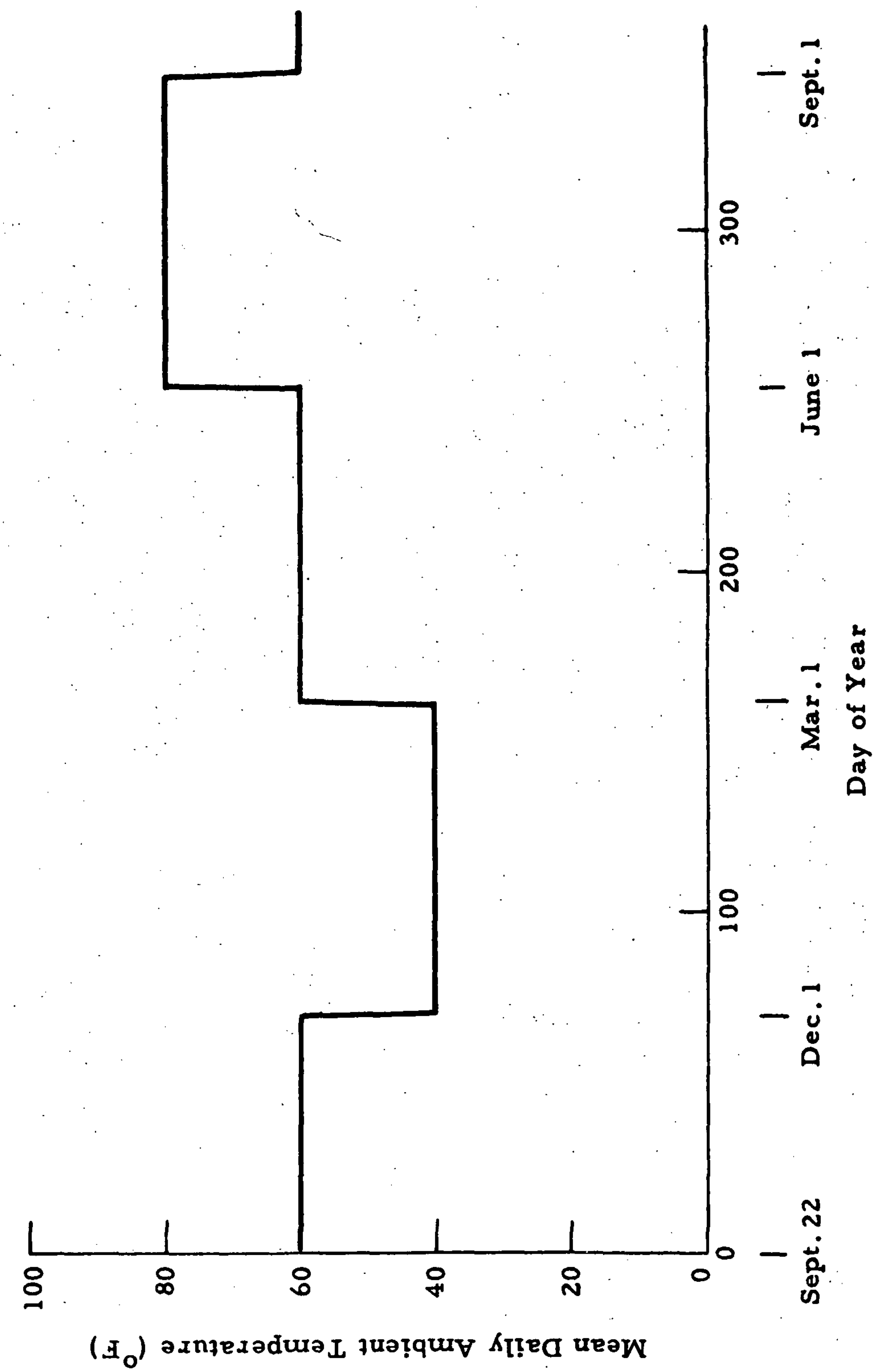


Fig. 3-7 - Ambient Temperature Variation

Using the solar and ambient data as described, the transient analysis was conducted for each day of the year and for several different collector temperatures to yield $(\dot{Q}/A)_{collected}$ as a function of Day and $T_{collector}$. Figure 3-8 presents an example of the transient analysis for one particular day and one particular collector temperature. Several interesting points are made by this figure. The entire collector is assumed to be at ambient temperature when the solar radiation first impinges upon it. The program then determines the transient temperature response during warmup. After the collector gets to its operating temperature, the heat transfer fluid begins circulating to maintain the operating temperature and net energy collection begins. Net energy collection continues until late afternoon when the losses overshadow the incident radiation. At this time, energy collection ceases and the collector begins to cool off. This same analysis was conducted for each day of the year and for collector operating temperatures from 100 to 300°F. The data thus generated were saved for use in the total system analysis discussed in Section 6.

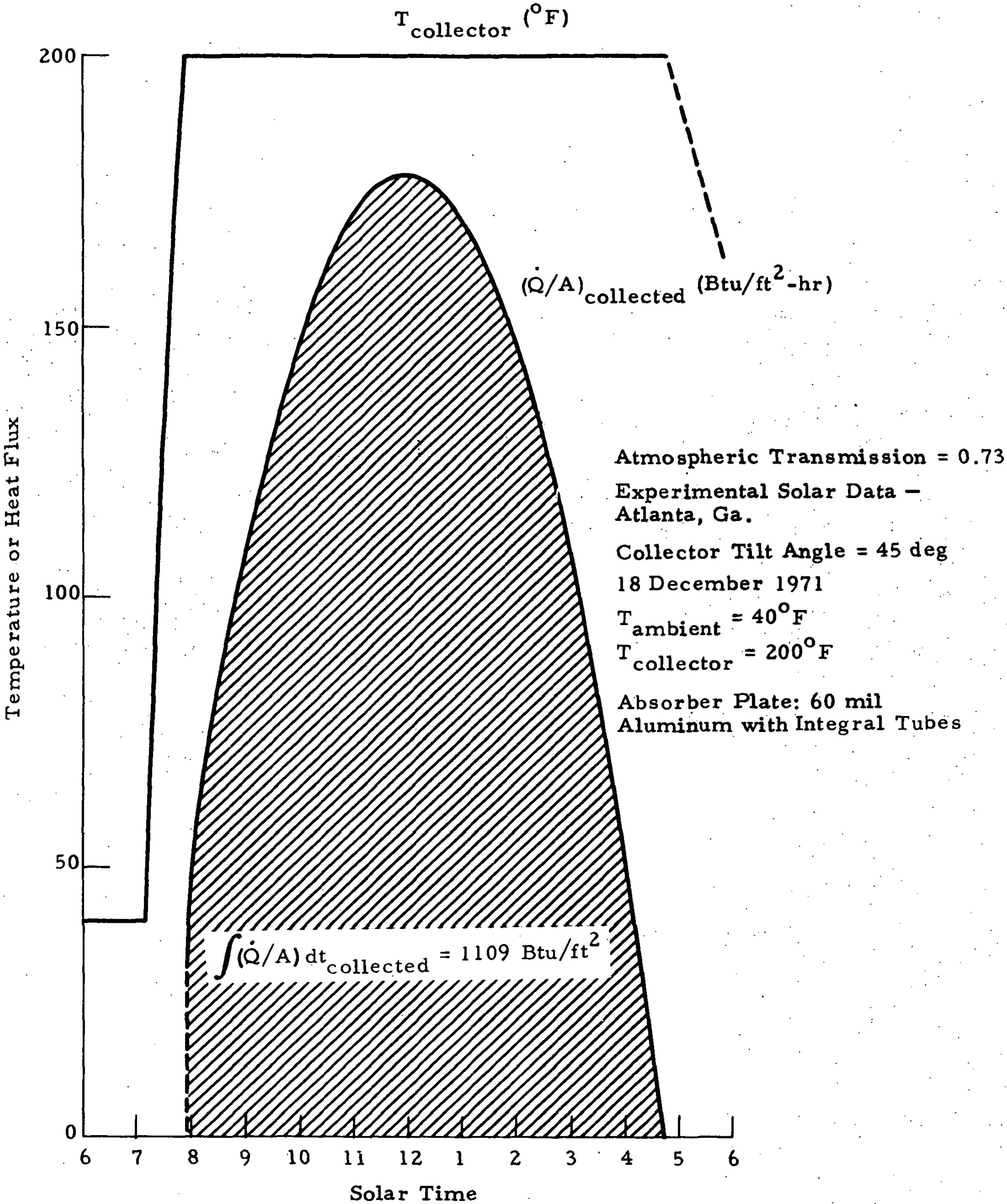


Fig. 3-8 - Daily Transient Solar Collector Performance

Section 4
THERMAL ENERGY STORAGE SYSTEM ANALYSIS

Since solar energy cannot be collected at night or on extremely overcast days, energy must be stored during collection periods for use in these non-collection periods. Although energy can be stored in many different forms, thermal energy storage is the most efficient form for the current application, since the collected solar energy is converted to thermal energy at the collector and reconversion to mechanical or electrical energy for storage would be highly inefficient. There are two primary means of storing thermal energy in a substance: (1) through a phase change, or (2) through a temperature rise (sensible heat storage). Both of these thermal energy storage mechanisms were investigated in the current study, as described in the following sections.

4.1 THERMAL ENERGY STORAGE SYSTEM USING PHASE CHANGE MATERIAL (PCM) AS THE ENERGY STORAGE SUBSTANCE

Energy can be stored in a substance as the latent heat of a phase change. This energy can be stored through solid-solid, solid-liquid, or liquid-vapor phase changes. However, in the current application, the quantity of energy to be stored is of the order of 10^6 Btu and the storage volume would be excessive for the liquid-vapor phase change. Therefore, only the solid-solid and solid-liquid phase changes were considered. Several hundred PCMs were surveyed to determine their applicability to the current system.

The desirable properties of a PCM are presented in Table 4-1. Each PCM surveyed was evaluated according to these properties. Several PCMs were found that would function adequately in the energy storage system, although none was ideal in every property. Several workable PCMs are listed in Table 4-2.

Table 4-1

DESIRABLE PCM PROPERTIES

- Low Cost per Btu Stored (low cost per lb_m and high heat of fusion)
- Melt Temperature: 200 – 230°F
- High Density
- High Thermal Conductivity
- Non-Hazardous, Non-Toxic
- Chemically Stable, Non-Corrosive to Container Materials.

Table 4-2

TYPICAL CANDIDATE PCM PROPERTIES

Material Name	Melt Temperature (°F)	Heat of Fusion (Btu/lb_m)
• Sulfur	240*	24
• α-Naphthol	203	70
• Methyl Fumarate	216	104
• Potassium Alum	196	79
• Oxidized Asphalt	180 - 220	—
• Micro-Crystalline Wax	170 - 220	60 - 100

*Melt temperature without additive. Additives to lower melt temperature have been tested by NASA-MSFC.

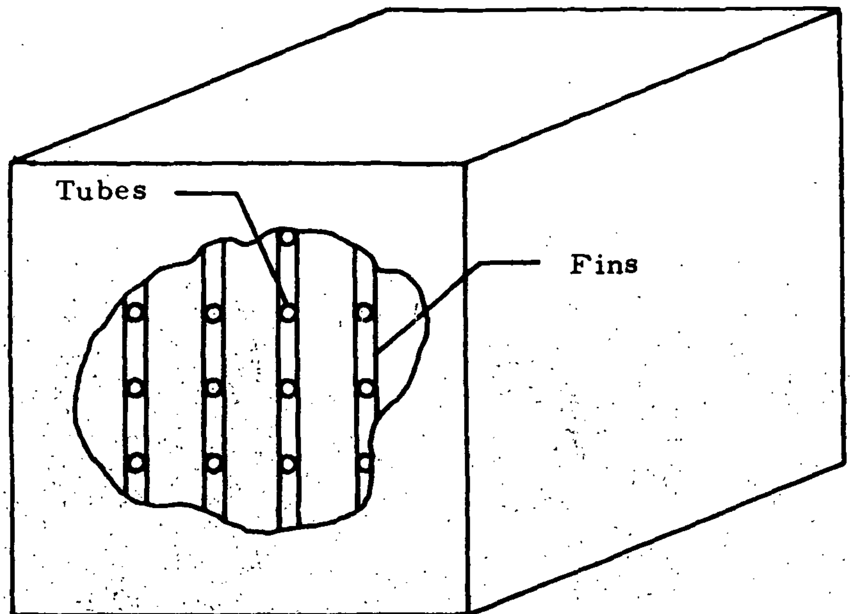
A thermal model of the PCM energy storage system was developed to determine the effects of fin arrangement, tube arrangement, and PCM properties upon overall ΔT ($T_{collector} - T_{generator\ exit}$). Typical results are presented for sulfur in Fig. 4-1. The container size, number of fins and tubes, and basic package design are seen to be reasonable for a ΔT of $20^{\circ}F$ and 10^6 Btu of energy storage.

For the PCMs thus far surveyed, the cost of PCM for about 10^6 Btu of energy storage will be $500 or more. The cost of the container, fins, tubes, plumbing, insulation and installation must be added to the cost of PCM to obtain total energy storage system cost. Thus, the total cost for a PCM system, while reasonable, will be greater than the cost of a similar size water container which can be used if sensible heat storage is utilized instead of phase change energy storage. The following section presents the results of the analysis of a thermal energy storage system using water as the sensible heat storage substance.

4.2 THERMAL ENERGY STORAGE SYSTEM USING WATER AS THE SENSIBLE HEAT STORAGE SUBSTANCE

A large quantity of energy can be stored as sensible heat in water. The mass of water required depends upon the quantity of energy to be stored and the temperature rise of the water. Since the cost of water is negligible, the container size is more important than the mass of water required. Figure 4-2 presents container size as a function of maximum energy storage and temperature rise. In the current application, about 1.1×10^6 Btu will need to be stored; this value is justified in Section 6 of this report. For a $20^{\circ}F$ temperature rise, a cubical container measuring 9.4 ft on a side will be needed to store 1.1×10^6 Btu, as seen from Fig. 4-2.

The basic water storage concept is shown in Fig. 4-3. The water tank will probably be placed below ground level for thermal and aesthetic reasons. As discussed further in Section 6, the generator exit temperature will be



Typical Results

- Energy Storage at 100% Melt: 1,000,000 Btu
- PCM: Sulfur
- Cube Dimensions: 7.2 x 7.2 x 7.2 ft
- $\dot{Q}_{solar\ in} = 100{,}000$ Btu/hr
- $\dot{Q}_{gen\ out} = 72{,}000$ Btu/hr
- Number of Fins: 49 (aluminum)
- Fin Spacing: 1.73 in.
- Fin Thickness: 0.150 in.

- Tube Spacing: 5.2 in. (Every other tube contains generator fluid; every other tube contains solar collector fluid.)
- $\Delta T_{in} = \Delta T_{fin_{in}} + \Delta T_{PCM\ (50\%\ melted)\ in} = 3^\circ F + 9^\circ F = 12^\circ F$
- $\Delta T_{out} = \Delta T_{fin_{out}} + \Delta T_{PCM\ (50\%\ frozen)\ out} = 2^\circ F + 6^\circ F = 8^\circ F$
- $\Delta T_{total} = \Delta T_{in} + \Delta T_{out} = T_{collector} - T_{generator\ exit} = 20^\circ F$

Fig. 4-1 - PCM Package Design

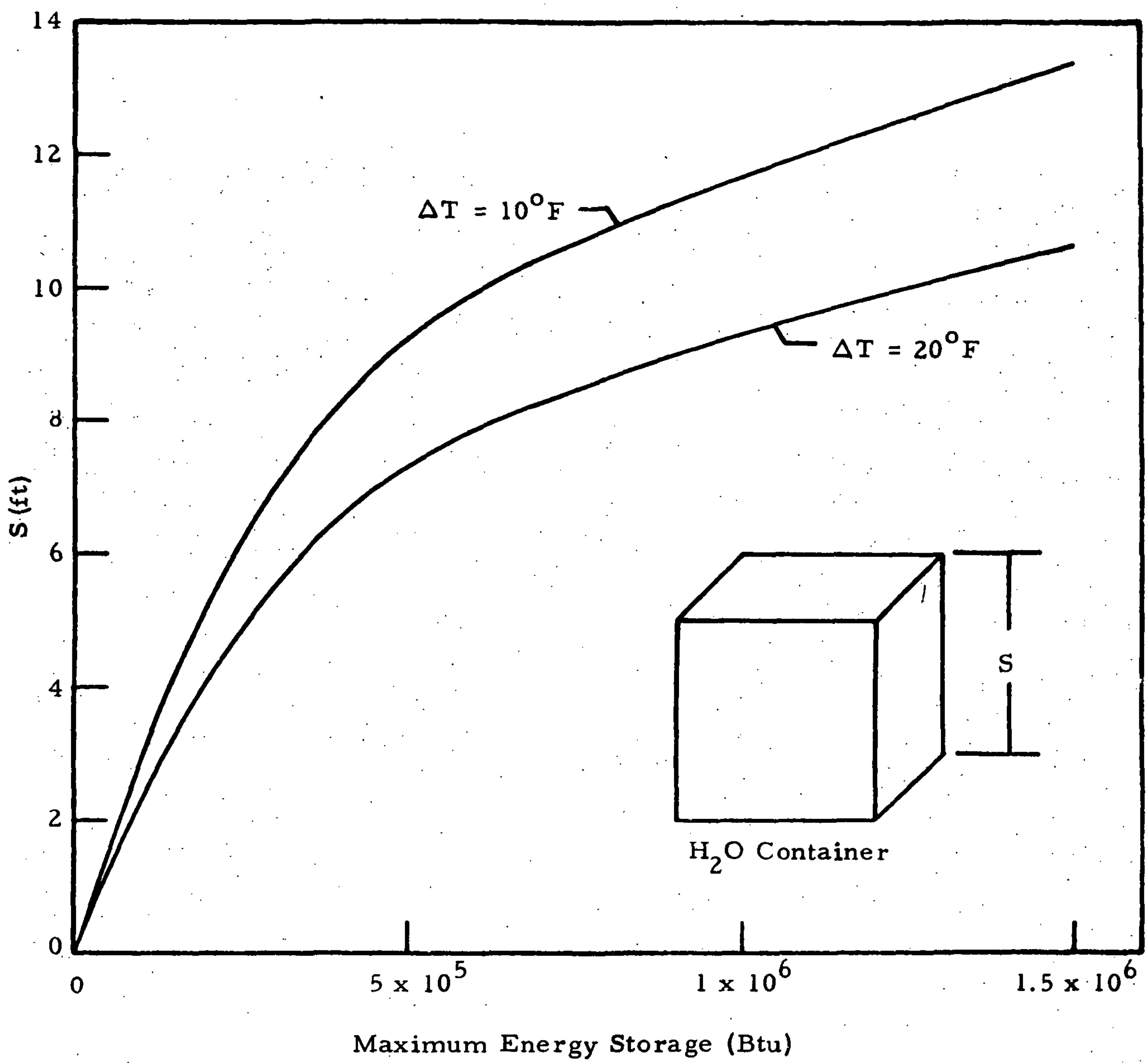


Fig. 4-2 - Container Size for H_2O Energy Storage System

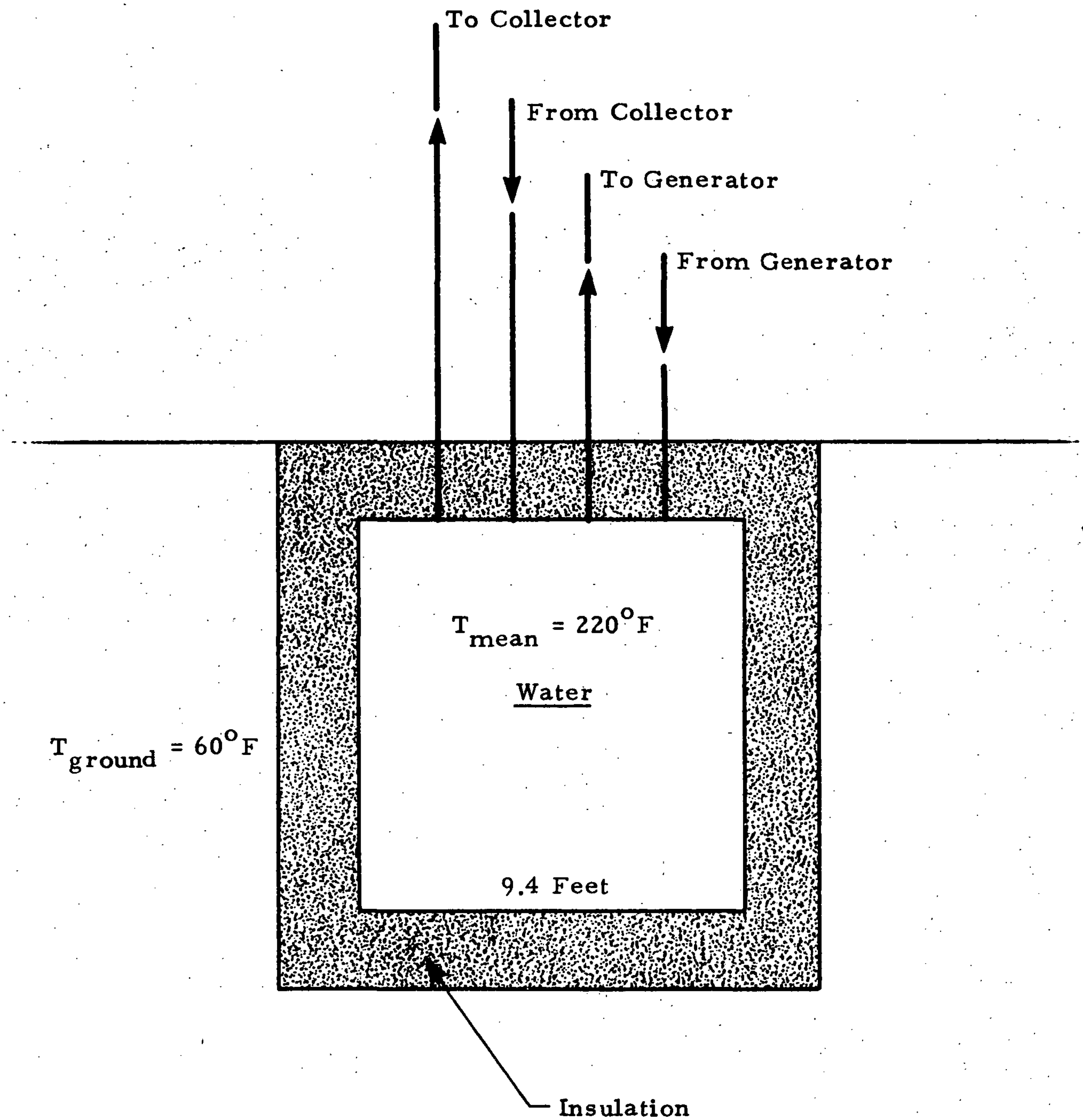


Fig. 4-3 - Thermal Energy Storage System Using Water

about 200°F. Thus, the water temperature will vary from about 210°F to about 230°F, for the 20°F temperature rise. To prevent boiling, a slight pressurization of the system will be required, about 6 psig. Non-volatile additives could be used instead of pressurization to elevate the boiling point, but the cost of the additives may be excessive for the large amounts involved. To prevent heat losses to the environment, some cheap loose-fill insulation will be placed between the tank and surroundings. The percent heat loss from the container versus insulation thickness is shown in Fig. 4-4 for a mean collector temperature of 220°F and for a mean water temperature of the same value.

The cost of insulation, even for a thickness of 1.5 to 2.0 ft, should be small by comparison to the cost of the tank. The losses can therefore be reduced below 5% without excessive cost. The cost of the tank itself will rely heavily upon the ingenuity of the designer. One preliminary idea is to use a thin metal or plastic liner, supported by a metal, wood, or concrete structure, somewhat similar to current swimming pool construction techniques.

Two flow loops are attached to the container, one to the solar collector and one to the generator of the absorption machine. The collector loop is envisioned as an on/off controlled flow system, which will allow flow circulation whenever the collector temperature is slightly higher than the water temperature. The generator flow loop is envisioned as a variable flow loop, where the flow rate will depend upon the stored water temperature and upon the characteristics of the water/generator heat exchanger. For example, Fig. 4-5 presents water flow rate required to provide a constant 72,000 Btu/hr generator input at a generator exit temperature of 200°F for different heat exchanger characteristics (UA) and different water temperatures. This variable flow will be accomplished with cheap, commercially available mechanical valves which require no power. The flow variation will allow a constant generator heat input to be made regardless of the water temperature, thereby allowing the absorption machine to function in the steady-state, steady-flow condition.

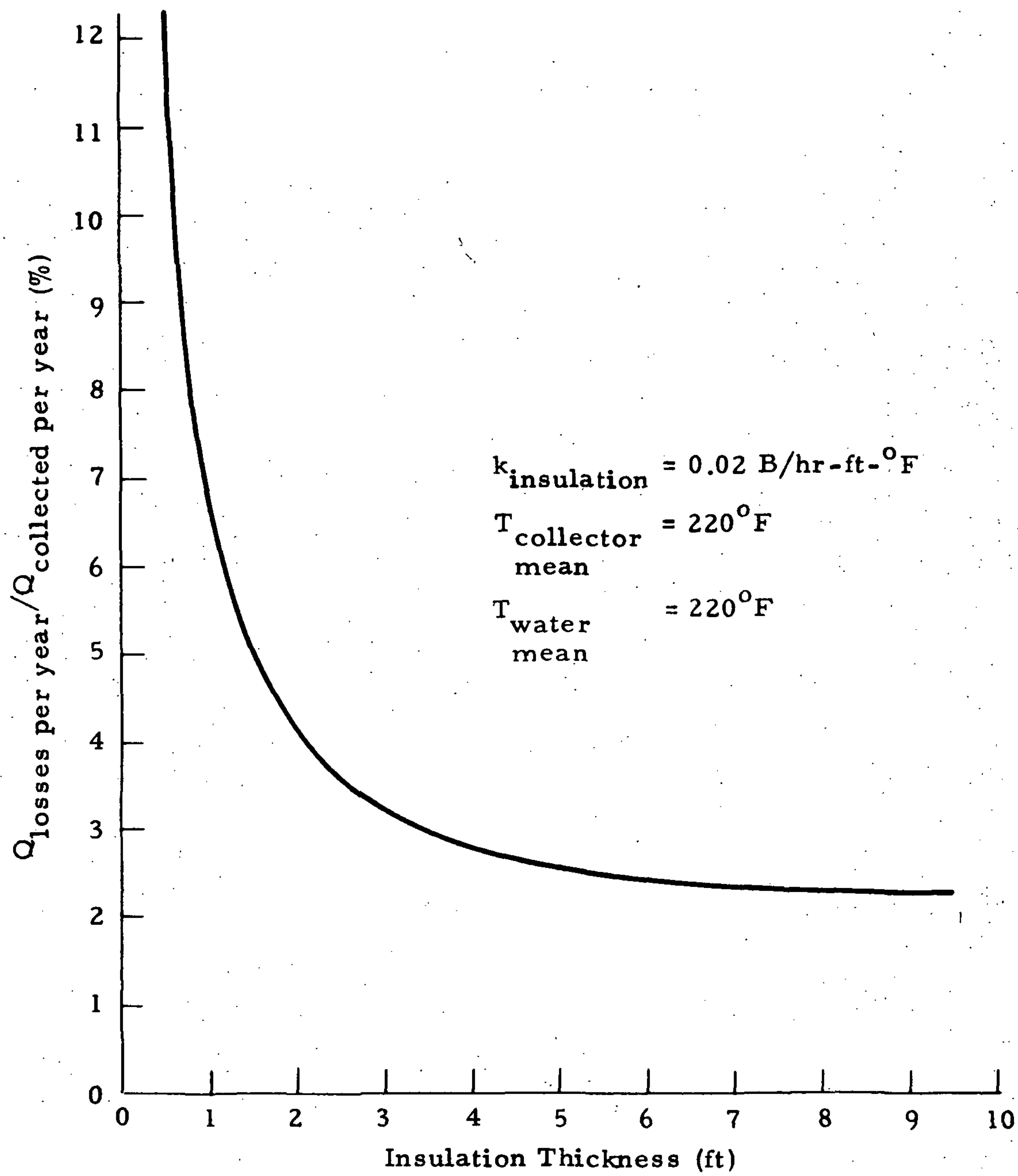


Fig. 4-4 - Heat Losses from Energy Storage System for Varying Insulation Thicknesses

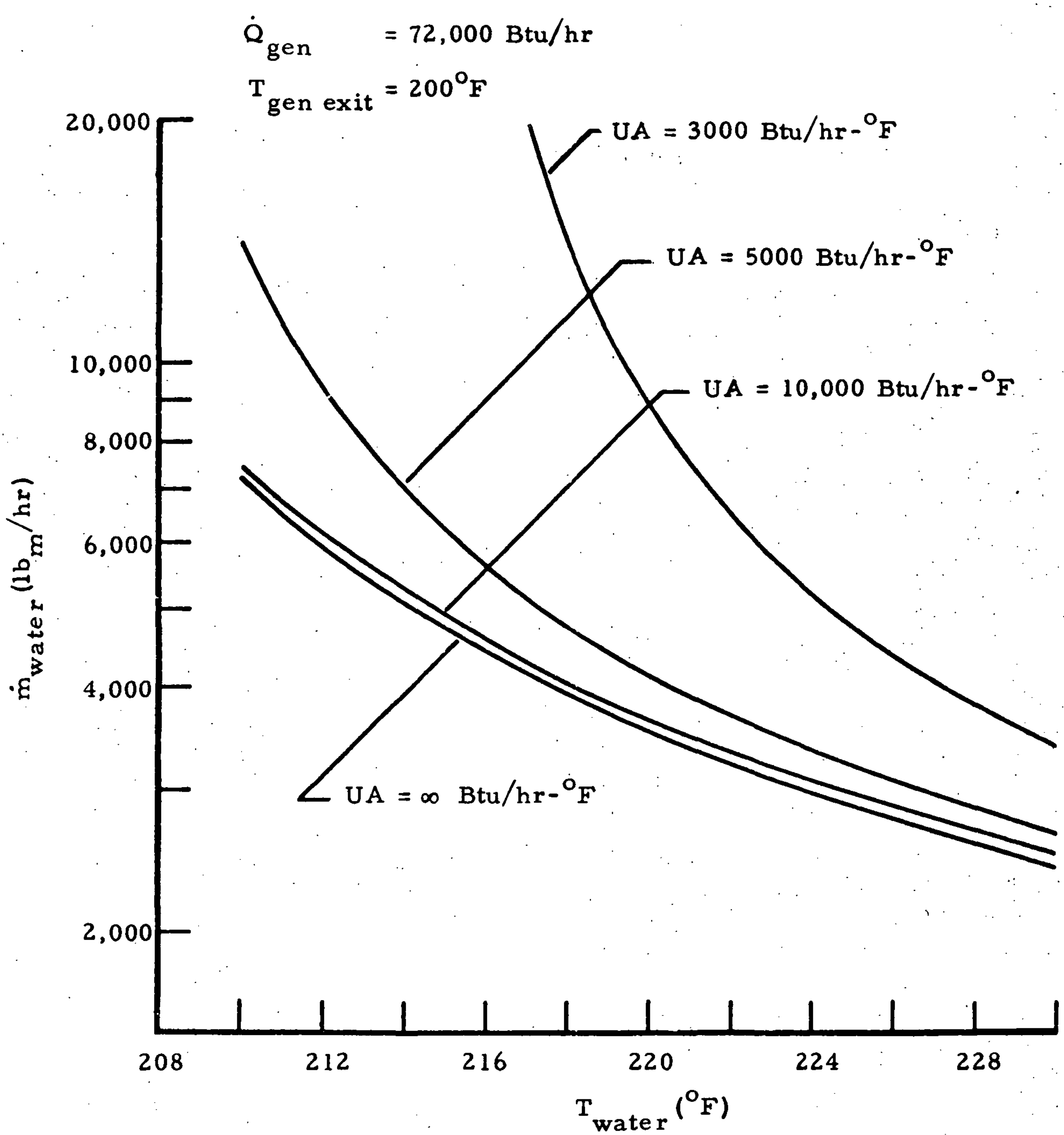


Fig. 4-5 - Water/Generator Heat Exchanger Requirements

The results of the thermal energy storage system analysis had to be incorporated into the total system analytical model discussed further in Section 6. Therefore, an overall ΔT ($T_{collector} - T_{generator\ exit}$) of $20^{\circ}F$ was assumed for the energy storage system, since this ΔT was found reasonable for either PCM or water energy storage. The cost of the energy storage system was more difficult to estimate. A value of 0.05 cents/Btu was finally assumed, based primarily upon the assumption of using water as the energy storage substance. This value corresponds to a storage system total cost of $550 for the 9.4-ft cubical system discussed previously.

Section 5
HEAT PUMP PACKAGE ANALYSIS

An absorption cycle heat pump is used for heating or cooling the air in the conditioned space. A schematic of the heat pump is presented in Fig. 5-1. Thermal energy ($\dot{Q}_g$) added at the generator represents the primary energy input for the machine. This energy is supplied by the energy storage system, which received this energy originally from the solar collector. A small amount of mechanical energy is added to the machine at the pump, but this energy is very small compared to the other principal energy exchanges of the machine, since the pumping is done on the liquid phase rather than on the vapor phase as in a conventional heat pump. Thermal energy is liberated at the condenser ($\dot{Q}_c$) and the absorber ($\dot{Q}_a$), and thermal energy is absorbed at the evaporator ($\dot{Q}_e$).

The heat pump operates in two basic modes: heating and cooling. In the heating mode, heat is absorbed from the outside air at the evaporator, which operates at a temperature below the outside temperature. Heat is supplied to the conditioned space at the condenser and at the absorber, both of which operate at temperatures above the temperature of the conditioned space. From the first law of thermodynamics,

$$\dot{Q}_{heating} = \dot{Q}_c + \dot{Q}_a = \dot{Q}_g + \dot{Q}_e.$$

Since $\dot{Q}_g$ represents collected solar energy, it is apparent that more heat is supplied to the conditioned space than must be collected. A quantitative measure of the efficiency of the heat pump operation is given by:

$$\eta = \frac{\dot{Q}_{heating}}{\dot{Q}_{solar}} = \frac{\dot{Q}_g + \dot{Q}_e}{\dot{Q}_g}.$$

In some instances, the outside air temperature will drop below the evaporator temperature. In such instances, the refrigerant flow will bypass the

T_{gen} is parameter
($T_{gen} \approx T_{collector} - 20^{\circ}F$)

(Note: Temperatures without brackets represent winter operating values. Temperatures inside brackets represent summer operating values.)

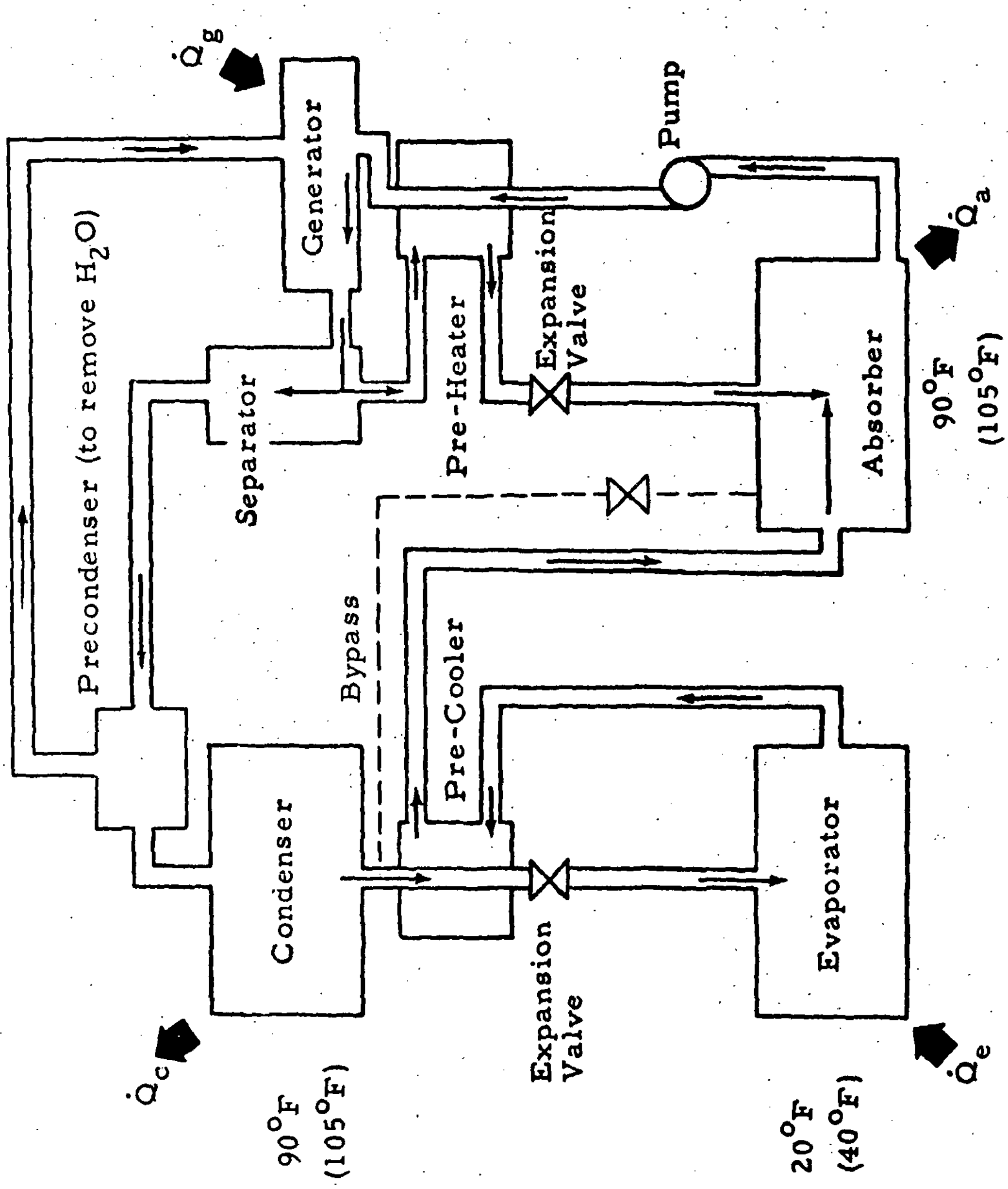


Fig. 5-1 — Absorption Refrigeration Cycle Schematic

evaporator as shown in Fig. 5-1. In this case, the lower limit of η is reached:

$$\eta = \frac{\dot{Q}_g}{\dot{Q}_g} = 1.0.$$

In the cooling mode, heat is absorbed from the inside air at the evaporator, which operates below the inside air temperature. This heat absorption cools and dehumidifies the conditioned space. Heat is dumped to the outside air at the condenser and at the absorber, both of which operate above the ambient air temperature. A quantitative measure of the performance of the heat pump under this mode of operation is given by:

$$\beta = \frac{\dot{Q}_{cooling}}{\dot{Q}_{solar}} = \frac{\dot{Q}_e}{\dot{Q}_g}.$$

Thus, it is seen that the most important parameter for the heat pump in the heating mode is η, and for the cooling mode it is β. In addition to η and β, the pump power required for operation in each mode is important since it represents an electrical energy input which must be paid for by the owner of the system. The magnitude of the pump power can be calculated under ideal thermodynamic conditions as:

$$\text{Ideal Power} = \dot{m} v \Delta P.$$

Now that the modes of operation of the heat pump have been described and the important parameters identified, the results of the heat pump analysis will be presented in the following sections.

5.1 SELECTION OF OPERATING TEMPERATURES

Before a performance analysis of the cycle could be conducted, the operating temperatures of the evaporator, condenser, and absorber had to be selected. For the heating mode of operation, the condenser and absorber

must transfer heat to the air in the conditioned space, which is about 70°F. Therefore, T_c and T_a were selected at 90°F, thus allowing a 20°F temperature differential to facilitate heat transfer. The evaporator temperature is more difficult to select. The lower the value of T_e, the fewer the instances when the outside temperature falls below T_e; however, the higher the value of T_e, the lower the pump power required for operating the heat pump. A value of 20°F for T_e was finally selected.

For the cooling mode of operation, the absorber and condenser temperatures were selected to be 105°F, thus allowing a reasonable temperature differential between these components and the outside air on hot summer days. This temperature is about the same as for a conventional heat pump. The evaporator temperature was selected as 40°F, thus allowing a 30°F temperature differential with the conditioned space for providing heat transfer and dehumidification.

The generator temperature was left as a parameter to be optimized, as discussed further in Section 5.4.

5.2 SELECTION OF DESIGN HEATING AND COOLING LOADS

The design heating and cooling loads for any conditioned space are functions of many variables, including building size, exposed area, windows, sun exposure, ambient temperature and humidity variations, insulation, construction materials, number of occupants, etc. In the current study, instead of determining design heating and cooling loads for a particular conditioned space, it was desired to define typical loads approximating the requirements of many houses. Thus, the following loads were defined:

- Maximum Heat Load = 72,000 Btu/hr
- Maximum Cooling Load = 36,000 Btu/hr.

These loads are fairly typical for a home in this region of the nation. They apply approximately to a poorly insulated 1500 ft^2 house or to a well-insulated 2500 ft^2 house. These loads were assumed in the remainder of the heat pump analysis to allow calculation of typical values of heat rates, flow rates, and power requirements. Any other loads could be met by the absorption cycle heat pump with appropriate linear adjustments in heat rates, flow rates, and powers.

5.3 SELECTION OF FLUIDS

The selection of the refrigerant and absorbent fluids for the absorption cycle heat pump greatly influences the performance of the machine. However, only a few fluid pairs are currently known to be practical for this application. Among those available, the parameters to be compared are basically two:

- the ability of the fluids to function at the selected temperatures,
- the effect of the fluid properties (primarily $\Delta h_{fg_{refrigerant}}$ and P-T-X properties) on the pump power requirement of the machine.

(The efficiencies (η and β) are also important in selecting fluid pairs; however, all current fluid combinations offer about the same η and β for the same operating temperatures.)

Table 5-1 presents a comparison between the most competitive fluid pairs. Obviously, ammonia and water are best for the current application.

5.4 HEAT PUMP PERFORMANCE

After determining the operating temperatures, the heating and cooling loads, and the best fluids for use in the absorption cycle heat pump, a performance analysis was conducted to determine the effect of generator temperature on η, β and pump power. The analysis was conducted by solving simultaneously the absorbent continuity equation, the refrigerant continuity

Table 5-1

COMPARISON OF REFRIGERANT-ABSORBENT FLUID COMBINATIONS

Refrigerant	Absorbent	Remarks
Ammonia	Water	• Able to function at selected temperatures • Low pump power
R-21	DME-TEG	• Able to function at selected temperatures • High pump power
R-22	DME-TEG	• Able to function at selected temperatures • High pump power
Water	Lithium Bromide (and similar salts)	• Unable to function at selected temperatures (freezing occurs in the evaporator which operates below 32°F) • Low pump power

equation, and the energy equation at each location around the cycle to meet the selected heating and cooling loads at the selected temperatures. For the heating mode, the analysis was based upon a generator input heat load of 72,000 Btu/hr rather than a heating output of 72,000 Btu/hr. This was done because the maximum heating load of the conditioned space will occur when the outside air temperature is below the evaporator temperature ($20^{\circ}F$). In such an instance, the refrigerant will bypass the evaporator and, from the first law of thermodynamics,

$$\dot{Q}_{heating} = \dot{Q}_a + \dot{Q}_c = \dot{Q}_g = 72{,}000 \text{ Btu/hr.}$$

When the outside air temperature is above the evaporator temperature, the evaporator will not be bypassed and:

$$\dot{Q}_{heating} = \dot{Q}_a + \dot{Q}_c = \dot{Q}_g + \dot{Q}_e = 72{,}000 \frac{\text{Btu}}{\text{hr}} + \dot{Q}_e.$$

Thus, the heating system will provide adequate maximum heating on very cold days without the evaporator and will provide more heating than is needed when the evaporator is used on normal or sunny days. The heating mode was analyzed as described above for two basic reasons:

- Conventional systems are designed to meet maximum heating loads with electric strip heaters which consume large quantities of electrical energy. Since the electrical power is to be minimized in the solar-powered system, strip heaters are undesirable and the system should be designed to deliver the maximum heating load without requiring strip heaters.
- Although the heating output will exceed the demand on normal days when the evaporator is not bypassed, this extra output will yield fast response and will require the same daily energy input at the pump as an undersized unit.

As stated previously, the generator temperature is the primary independent variable of interest in the heat pump analysis. For a given set of operating conditions, T_{gen} has a lower limit. This lower limit is, of course,

the boiling point of the solution. Figure 5-2 presents this lower limit for T_{gen} as a function of T_e for the heat pump in the heating mode. For the selected value of T_e ($20^{\circ}F$), the minimum generator temperature is about $175^{\circ}F$. Figure 5-3 presents this lower limit for T_{gen} as a function of absorber and condenser temperature ($T_a = T_c$) for the heat pump in the cooling mode. For the selected value of $105^{\circ}F$ for T_a and T_c, the minimum generator temperature is about $182^{\circ}F$. Therefore, only temperatures above these values must be considered in the performance analysis.

Figure 5-4 presents the results of the performance analysis for the heat pump in the heating mode. The ideal power decreases from infinity at about $175^{\circ}F$ to very low values above $200^{\circ}F$. The η decreases slightly with increasing T_{gen}, thus showing the benefits of relatively low temperature operation of the generator.

Figure 5-5 presents the results of the performance analysis for the heat pump in the cooling mode. The ideal power decreases from infinity at about $182^{\circ}F$ to very low values above $200^{\circ}F$, again indicating the need for a generator temperature exceeding $200^{\circ}F$. The β decreases with increasing T_{gen}, again showing the benefits of a fairly low temperature generator.

Both Fig. 5-4 and Fig. 5-5 imply that there is some optimum value of T_{gen}, depending upon the relative importance of maximizing η (or β) and of minimizing power. However, T_{gen} also affects solar collector efficiency since $T_{collector}$ must be greater than T_{gen}. Thus, an optimization of T_{gen} must include the effects of T_{gen} upon η, β, power and solar collector efficiency. Such an optimization was conducted for T_{gen}, as well as for several other total system design variables, and is discussed further in the following section of this report.

Conditions

Heating Mode Operation
$T_a = T_c = 90^oF$
Aqua-ammonia Fluid Combination

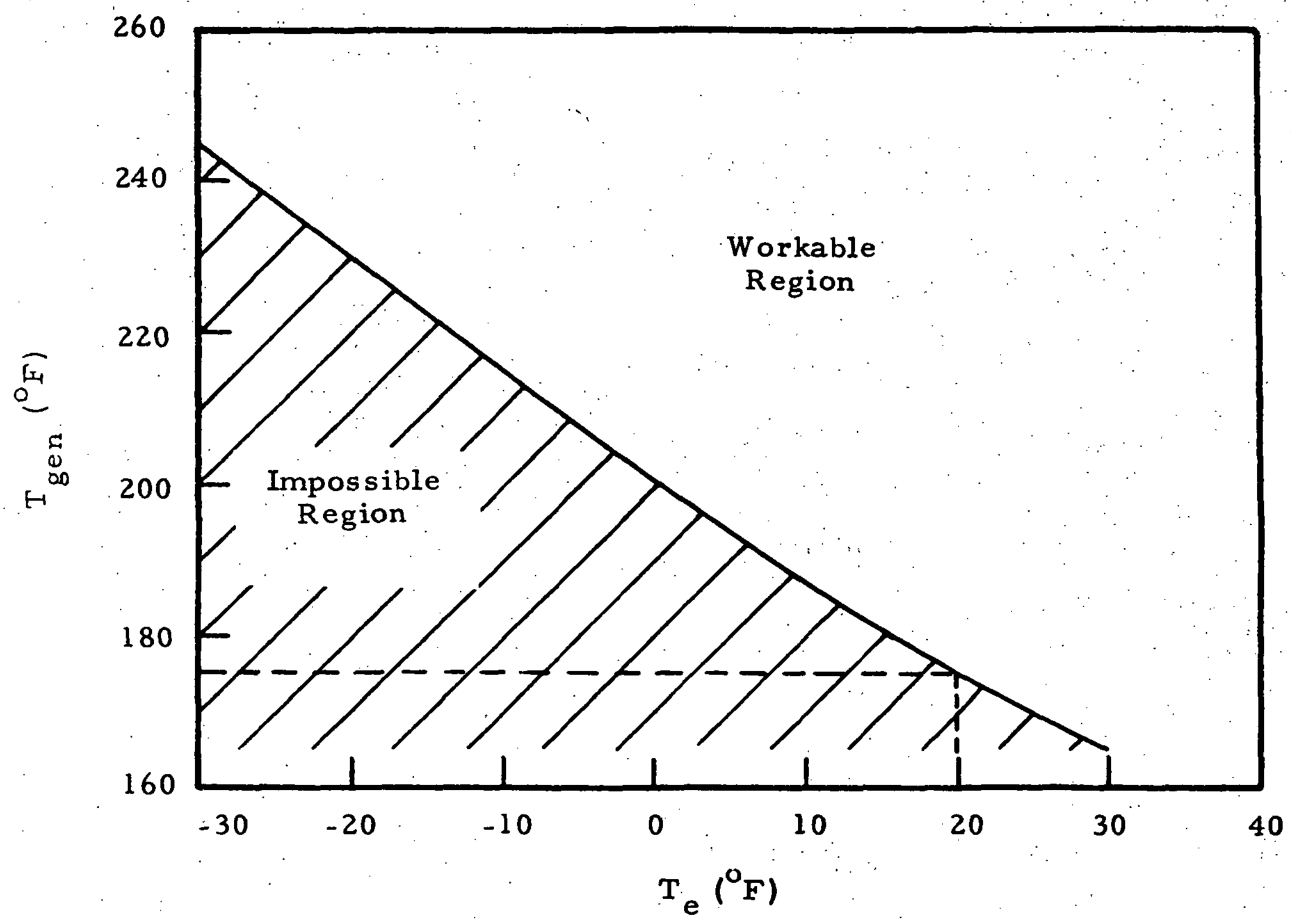


Fig. 5-2 - Required Generator Temperature for Different Evaporator Temperatures for Heating Mode Operation

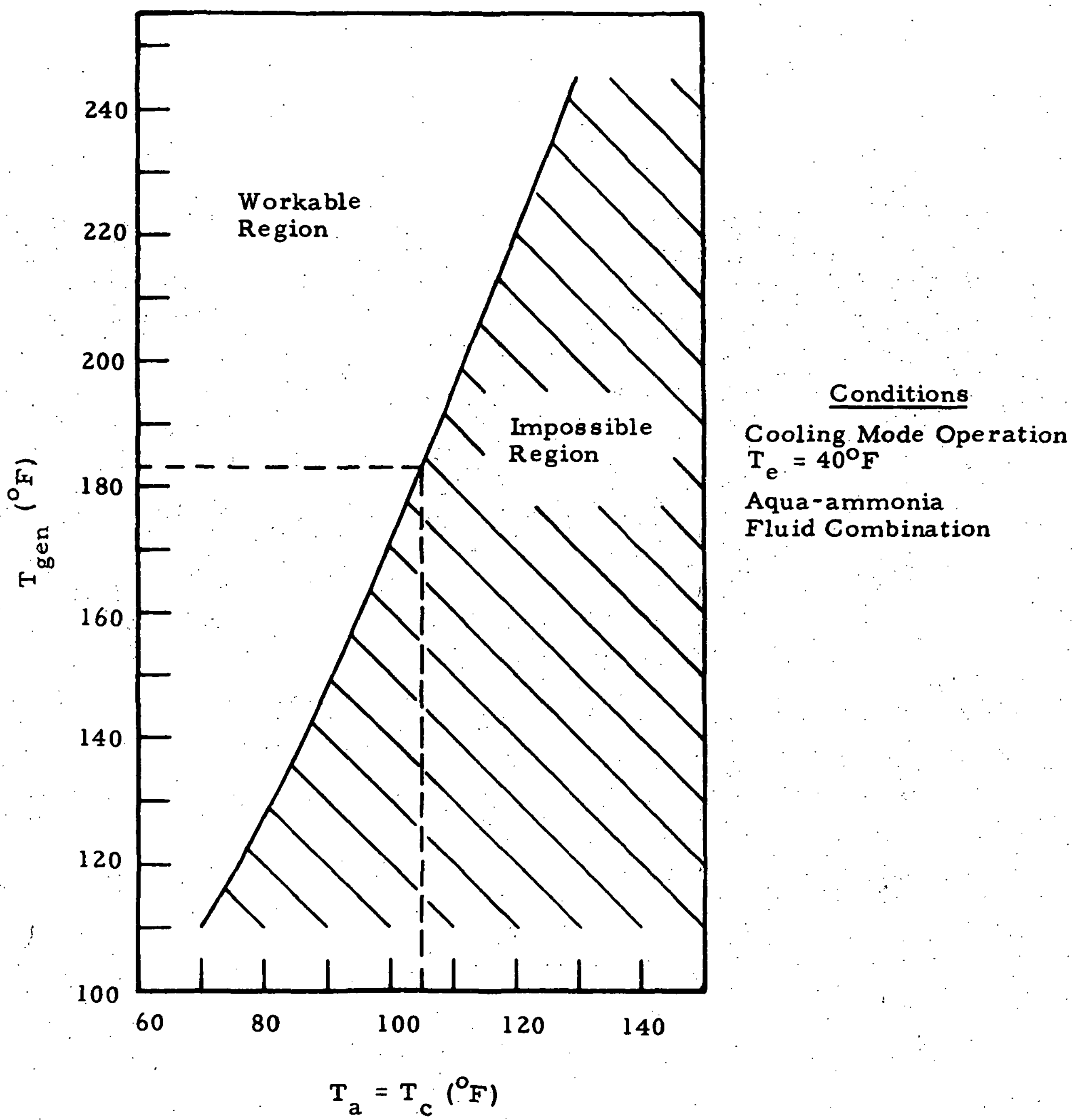


Fig. 5-3 - Required Generator Temperature for Different Absorber/Condenser Temperatures for Cooling Mode Operation

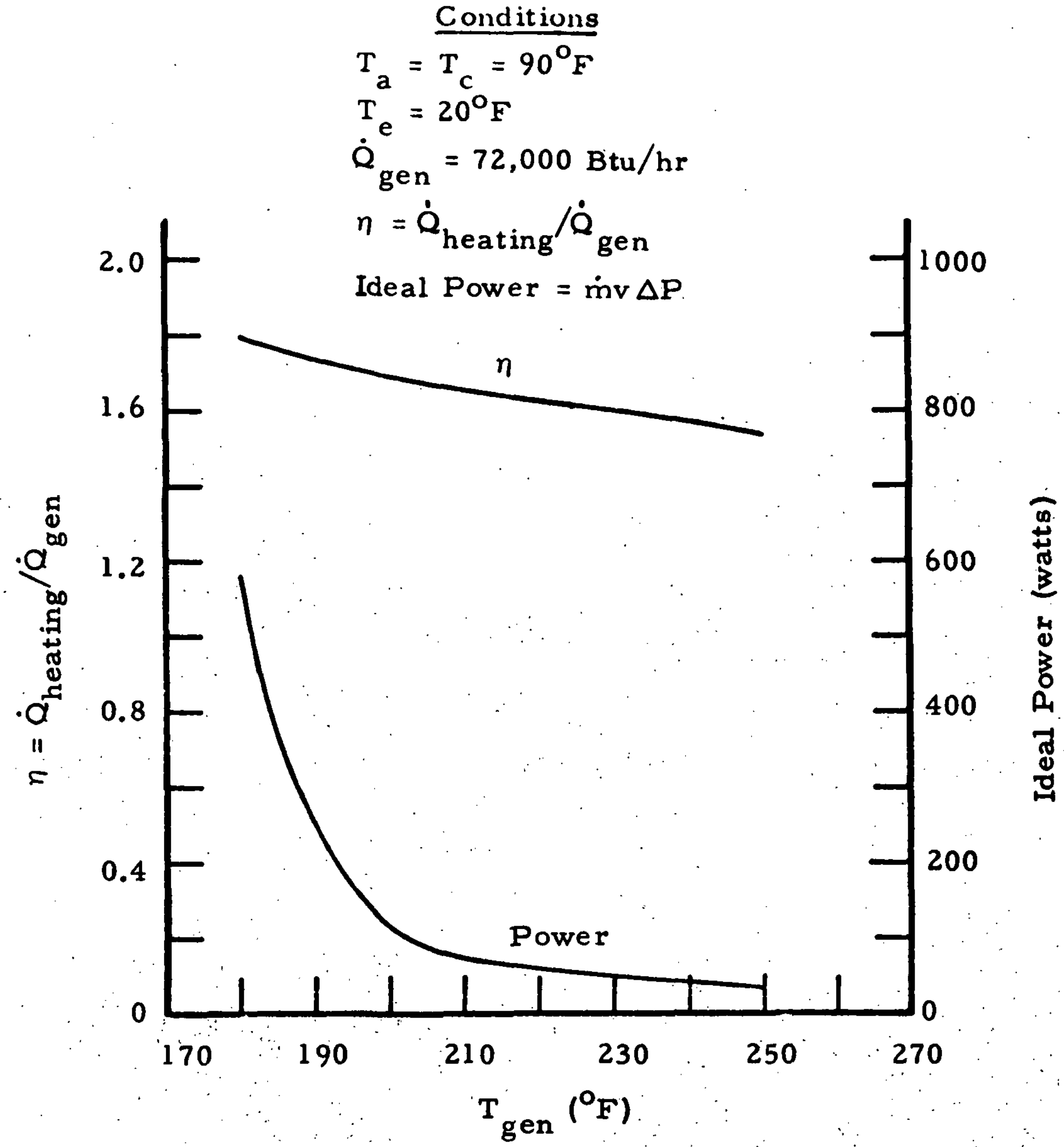


Fig. 5-4 - Heat Pump Performance in Heating Mode for Different Generator Temperatures

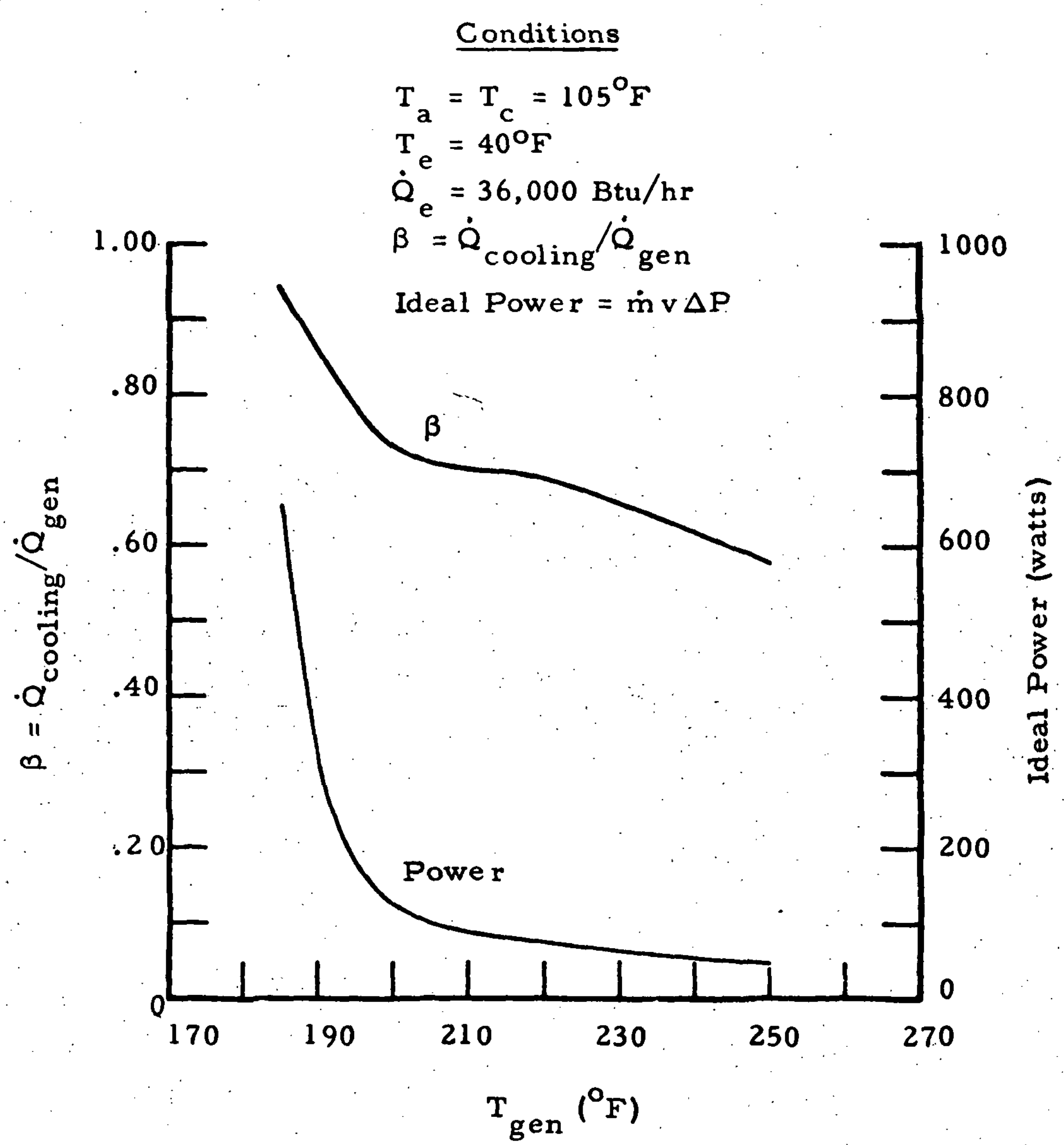


Fig. 5-5 - Heat Pump Performance in Cooling Mode for Different Generator Temperatures

5.5 CONTACT WITH MAJOR MANUFACTURER OF ABSORPTION CYCLE AIR-CONDITIONING SYSTEMS

To further establish the feasibility of large-scale manufacturing of the absorption cycle heat pump described in previous sections, Lockheed contacted a major manufacturer of commercial absorption cycle air conditioning systems, the Arkla Air Conditioning Company. The modes of operation, fluids, temperatures and performance of the heat pump were described to Arkla to obtain their critical opinion of the feasibility of the system. In response, Arkla personnel perceived no major problems with any aspect of the machine. In addition, they informed Lockheed that, from the preliminary description of the machine, they could see no reason why the mass-produced heat pump would cost much more than their current production systems, which cost less than $900 for residential sizes. It should be noted that these preliminary contacts were telephone conversations, and that the feasibility and cost estimates represent opinions rather than firm conclusions at this time. However, these positive opinions of a major manufacturer are considered important additional indications of the practicality of the absorption cycle heat pump.

Section 6
TOTAL SYSTEM ANALYSIS

The total system cost (including hardware, installation, maintenance, and operating costs) is a function of three primary independent variables:

- Solar collector area
- Energy storage system capacity
- Collector temperature.

To evaluate the economic feasibility of the solar-powered system, it is necessary to determine the optimum value for each of these three variables and then to compare the resultant total system cost to the cost of a conventional heating, cooling, and water heating system. To accomplish these objectives, a total system analysis was conducted during this study. To conduct this analysis, the performance and cost results of the solar collector analysis, the thermal energy storage system analysis, and the heat pump package analysis were incorporated into a total system computer analytical model. The analytical model and parametric analysis are described in the following sections. The performance of the preliminary optimized system is discussed, and the results of a detailed cost comparison are presented.

6.1 COMPUTER ANALYTICAL MODEL

As described in Section 3, the results of the solar collector analysis yielded the total energy collected per unit area as a function of collector temperature for each day of the year. These results were based upon transient analysis and empirical solar data. As further presented in Section 3, the collector concept, using two Tedlar cover sheets, was chosen as optimum from a cost and performance viewpoint. The best cost estimate was taken as $1.00 per square foot*. Thus, the following relations were incorporated into the total system computer analytical model:

*Estimated cost figures for system components are preliminary, and will be more firmly established during the Phase II program as discussed in Section 9 and the Appendix.

$$Q_{collected}(\text{Day}, T_{collector}) = Q/A(\text{Day}, T_{collector}) \cdot (\text{Area}),$$

$$\text{Cost}_{collector} = \$1/\text{ft}^2 \cdot \text{Area}.$$

As described in Section 4, the overall ΔT for the thermal energy storage system was found to be about $20^\circ F$ for a reasonable system design. The cost of the thermal energy storage system was estimated to be about 0.05 ¢/Btu, as also described in Section 4. Thus, in the total system computer analytical model, the following relations were utilized:

$$\text{Cost}_{\substack{\text{thermal energy}\\ \text{storage system}}} = (0.05\ ¢/\text{Btu}) \cdot (E_{max}),$$

$$T_{gen} = T_{collector} - 20^\circ F.$$

As described in Section 5, the efficiency and ideal power were determined as functions of generator temperature for both summer and winter operation. Thus, the following relations were used in the total system model:

$$\eta_{heating} = \eta(T_{gen}),$$

$$\beta_{cooling} = \beta(T_{gen}),$$

$$\text{Power}_{heating} = \text{Power}_H(T_{gen}),$$

$$\text{Power}_{cooling} = \text{Power}_C(T_{gen}).$$

A thermal model for the conditioned space was required in the total system analysis, and the chosen model is presented in Fig. 6-1. As discussed in Section 5, these heating and cooling loads are fairly typical for a house in the Huntsville area. The water heating requirement was taken to be the typical value of 4000 Btu/hr continuously throughout the year.

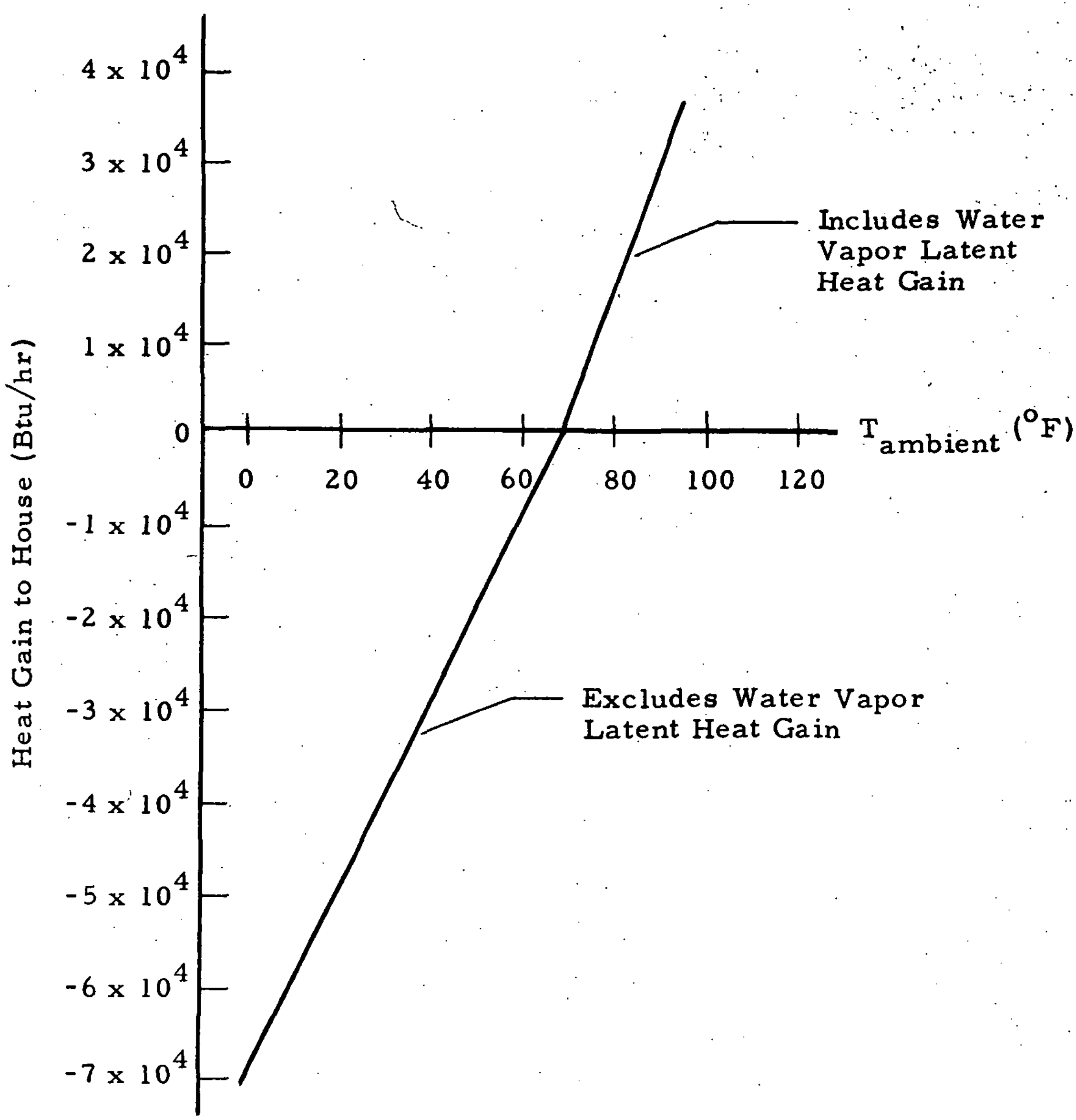


Fig. 6-1 - Heat Gain to House for Different Ambient Temperatures

A seasonal variation in ambient temperature was utilized in the total system analysis to determine daily heating and cooling requirements throughout the year. This ambient temperature variation is shown in Fig. 6-2, and is the same curve that was used in the solar collector analysis.

Using the relations and curves defined previously, the total system computer program was designed to calculate each energy quantity of importance for each day of the year for a given set of:

- Collector area (Area)
- Collector temperature ($T_{collector}$)
- Energy storage system capacity (E_{max}).

These important energy quantities calculated by the program were:

- $Q_{collected}$ (Day) = total energy collected on this day.
- $Q_{heating}$ (Day) = total heat added to house for space heating on this day.
- $Q_{cooling}$ (Day) = total heat removed from house for air conditioning on this day.
- Q_{H_2O} (Day) = total heat added to hot water heater on this day.
- Q_{pump} (Day) = total electrical energy used by pump on this day.
- E_{stored} (Day) = total energy left in energy storage system at end of this day.
- Q_{aux} (Day) = total auxiliary energy added to thermal energy storage system on this day to prevent E_{stored} from going below zero.
- Q_{waste} (Day) = total energy which was collected but could not be stored because the energy storage system was fully charged ($E_{stored} = E_{max}$) and could accept no further energy input on this day.

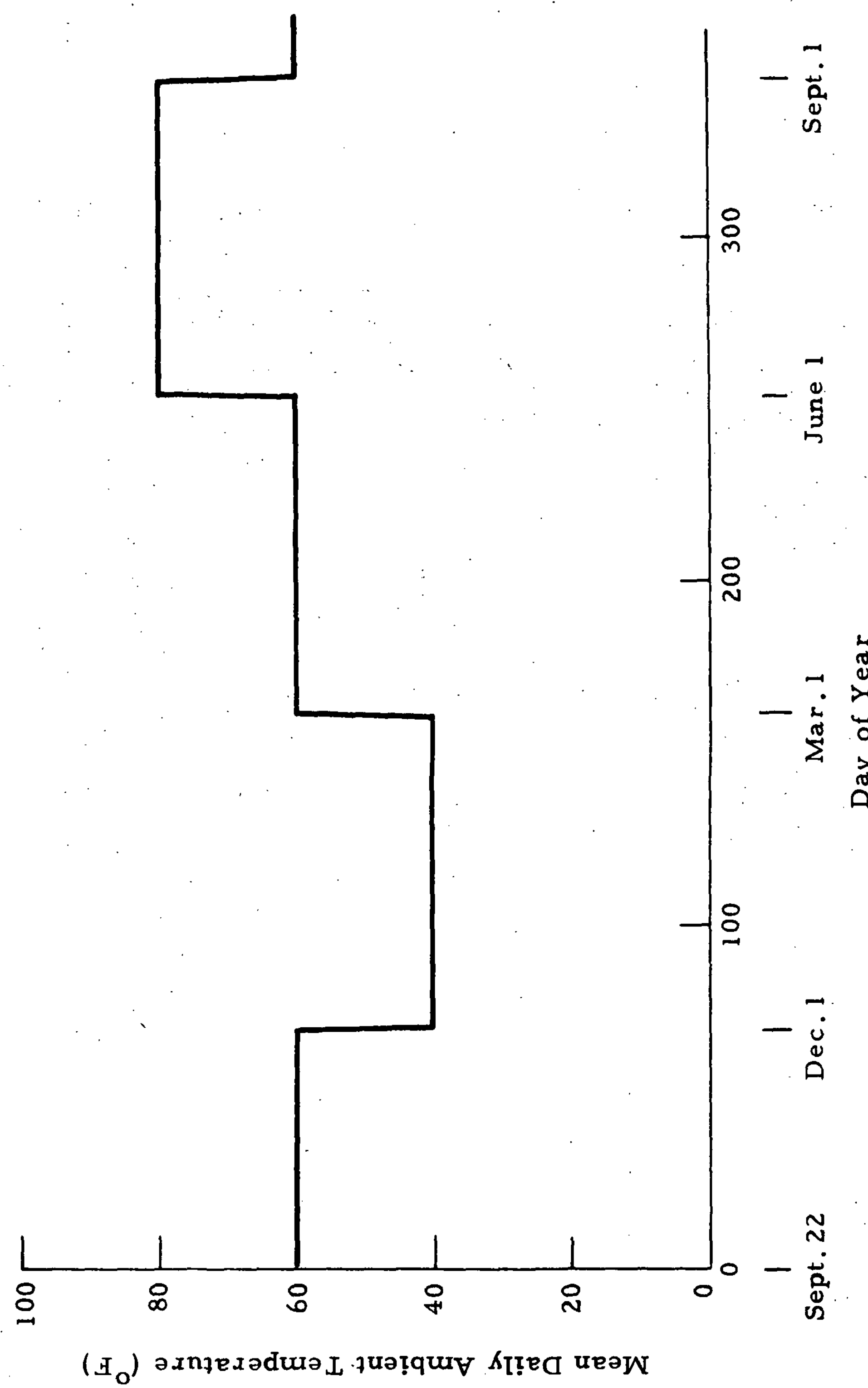


Fig. 6-2 - Ambient Temperature Variation

- $\sum_{Day=1}^{n} Q_{collected}(Day)$ = total energy collected from Day = 1 through this day.
- $\sum_{Day=1}^{n} Q_{heating}(Day)$ = total heat added to house for space heating from Day = 1 through this day.
- $\sum_{Day=1}^{n} Q_{cooling}(Day)$ = total heat removed from house for air conditioning from Day = 1 through this day.
- $\sum_{Day=1}^{n} Q_{H_2O}(Day)$ = total heat added to hot water heater from Day = 1 through this day.
- $\sum_{Day=1}^{n} Q_{pump}(Day)$ = total electrical energy used by pump from Day = 1 through this day.
- $\sum_{Day=1}^{n} Q_{aux}(Day)$ = total auxiliary energy added to thermal energy storage system from Day = 1 through this day.
- $\sum_{Day=1}^{n} Q_{waste}(Day)$ = total energy collected but not stored from Day = 1 through this day.

To determine the combination of Area, $T_{collector}$, and E_{max} which would yield the most economical total system, a total system cost equation was derived as presented in Fig. 6-3. The total cost per year was a function of only the three variables: Area, $T_{collector}$, and E_{max}. A parametric study was conducted to determine the optimum total system design, using the computer model previously discussed, and the results of this parametric study are presented in the next section.

$$\text{Total Cost/yr} = \frac{\text{Cost}_{\text{heat pump}}}{\text{Lifetime}} + \frac{(\text{Cost}_{\text{solar collector}}/\text{ft}^2)(\text{Area}_{\text{solar collector}})}{\text{Lifetime}}$$

$$+ \frac{(\text{Cost}_{\text{energy storage system}}/\text{Btu})(E_{\text{max}})}{\text{Lifetime}}$$

$$+ (\text{Cost}_{\text{aux power}}/\text{Btu})\left(\sum_{\text{Day}=1}^{365} Q_{\text{aux}}(\text{Day})\right)$$

$$+ (\text{Cost}_{\text{pump power}}/\text{Btu})\left(\sum_{\text{Day}=1}^{365} Q_{\text{pump}}(\text{Day})\right)$$

$$\text{Total Cost/yr} = \text{Function}(\text{Area}_{\text{solar collector}}, \text{Energy}_{\text{max stored}}, T_{\text{collector}})$$

Fig. 6-3 - Total System Cost Equation

6.2 PARAMETRIC TOTAL SYSTEM ANALYSIS

The three independent variables were varied in combination over the ranges shown below:

$$0 \leq \text{Area} \leq 2500 \text{ ft}^2$$

$$220^{\circ}\text{F} \leq T_{collector} \leq 250^{\circ}\text{F}$$

$$0 \leq E_{max} \leq 2.0 \times 10^6 \text{ Btu.}$$

Collector temperatures below 220°F were excluded from consideration because these temperatures correspond to generator temperatures below 200°F. At such low generator temperatures, the ideal pump power increases very rapidly with decreasing temperature, as shown in Section 5. Therefore, the deviation between actual power and ideal power could cause serious errors for these low temperatures. For collector temperatures above 220°F (generator temperatures above 200°F), however, the ideal pump power is so small that even a 100% deviation between actual power and ideal power would cause an error in total system cost per year of less than 2%.

The results of the total system optimization study are presented in Fig. 6-4. The collector temperature of 220°F was found to be superior to the other temperatures considered, and the best collector area and energy storage system capacity were found to be 1300 ft^2 and 1.1×10^6 Btu, respectively. The optimum design is seen to yield the minimum total cost per year in Fig. 6-4.

Several observations must be made about this optimum system design:

- Both the pump power and auxiliary power requirements were met with electrical energy at the current Huntsville minimum rate of 0.88 ¢/kW-hr. The auxiliary power requirement could be met with natural gas, fuel oil, or other thermal energy source., The rate of 0.88¢/kW-hr does not take cost increases into consideration over the 20-year lifetime of the system. (See Section 6.5 for more on this topic).

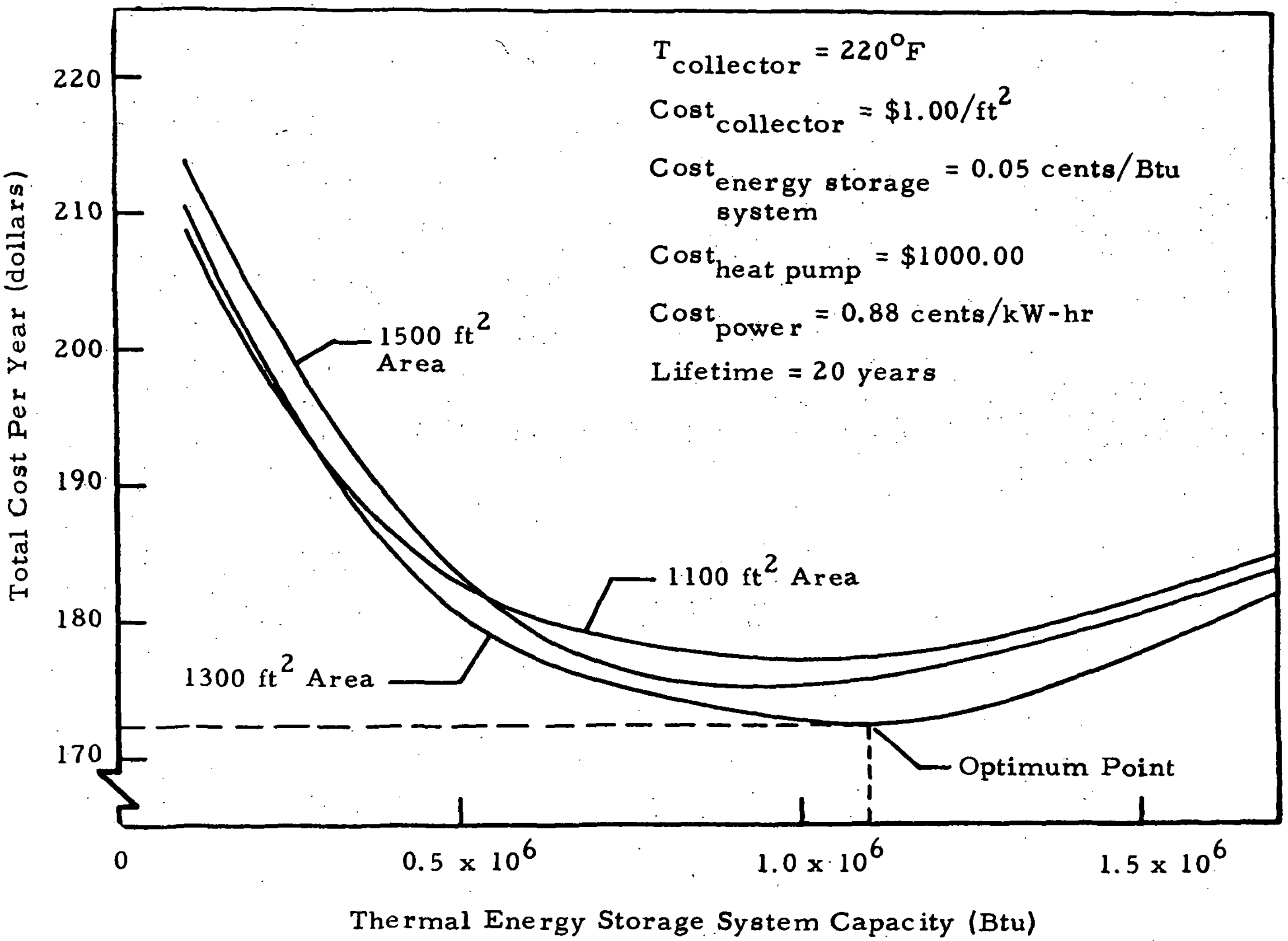


Fig. 6-4 - Total System Cost Optimization

- The cost of the heat pump (assumed $1000) does not affect the optimum point since this cost is not a function of the independent variables. However, it does affect the total system cost per year.
- The collector cost ($\$1/ft^2$), the energy storage system cost (0.05¢/Btu), and the lifetime of the system (20 years) all affect the optimum point and are all educated guesses at this stage of development. Therefore, the optimum point presented should not be considered the actual best possible system design. As better estimates of costs and lifetime become available, the system should be reoptimized to determine a new optimum design. The given optimization is meant only to illustrate the method available and to provide rough estimates of total cost per year.

6.3 PERFORMANCE OF PRELIMINARY OPTIMIZED SYSTEM

The performance of the system obtained from the preliminary optimization discussed previously is summarized in Table 6-1. Noteworthy in this table are the power requirements for heating and cooling, both of which represent lightbulb loads. The yearly operating cost ($30) is about an order of magnitude less than the operating cost for a conventional heat pump. When scanning the heating capacities, it should be remembered that no strip heaters are needed to deliver these energy outputs.

Figure 6-5 presents the important energy quantities for the same system through the entire year. Note that the outputs of the system, namely the total heating, cooling and water heating energy summations for the year greatly exceed the energy input

$$\left(\sum_{Day=1}^{n} (Q_{aux} + Q_{pump}) \, (Day) \right) ,$$

which must be paid for by the owner of the system. The total energy collected curve is somewhat misleading. About 50% of this collected energy was not utilized because the thermal energy storage system was fully charged while it was being collected. For example, on many spring and fall days, the solar energy collected far exceeds the requirements for providing heating, cooling,

Table 6-1

HEAT PUMP SUMMARY FOR $T_{collector} = 220^{o}F$ ($T_{gen} = 200^{o}F$)

Peak heating capacity — heat pump mode ($T_{amb} \geq 30^{o}F$)	121,000 B/hr
Peak heating capacity — direct heating mode ($T_{amb} \leq 20^{o}F$)	72,000 B/hr
Peak cooling capacity	36,000 B/hr
Continuous water heating	4,000 B/hr
Peak ideal pump power — heating	113 watts
Peak ideal pump power — cooling	126 watts
Collector area	1,300 ft^2
Energy storage system capacity	1,100,000 Btu
Total yearly cost*	$172
Yearly operating cost	$ 30

*Based on the following assumptions: (1) collector cost = $1.00 ft^2 (2 Tedlar covers and selective coating); (2) energy storage system cost = 0.05 cents/Btu; (3) heat pump cost = $1000.00; (4) electrical power cost = 0.88 cents/kW/hr (Huntsville minimum rate); and (5) lifetime = 20 years.

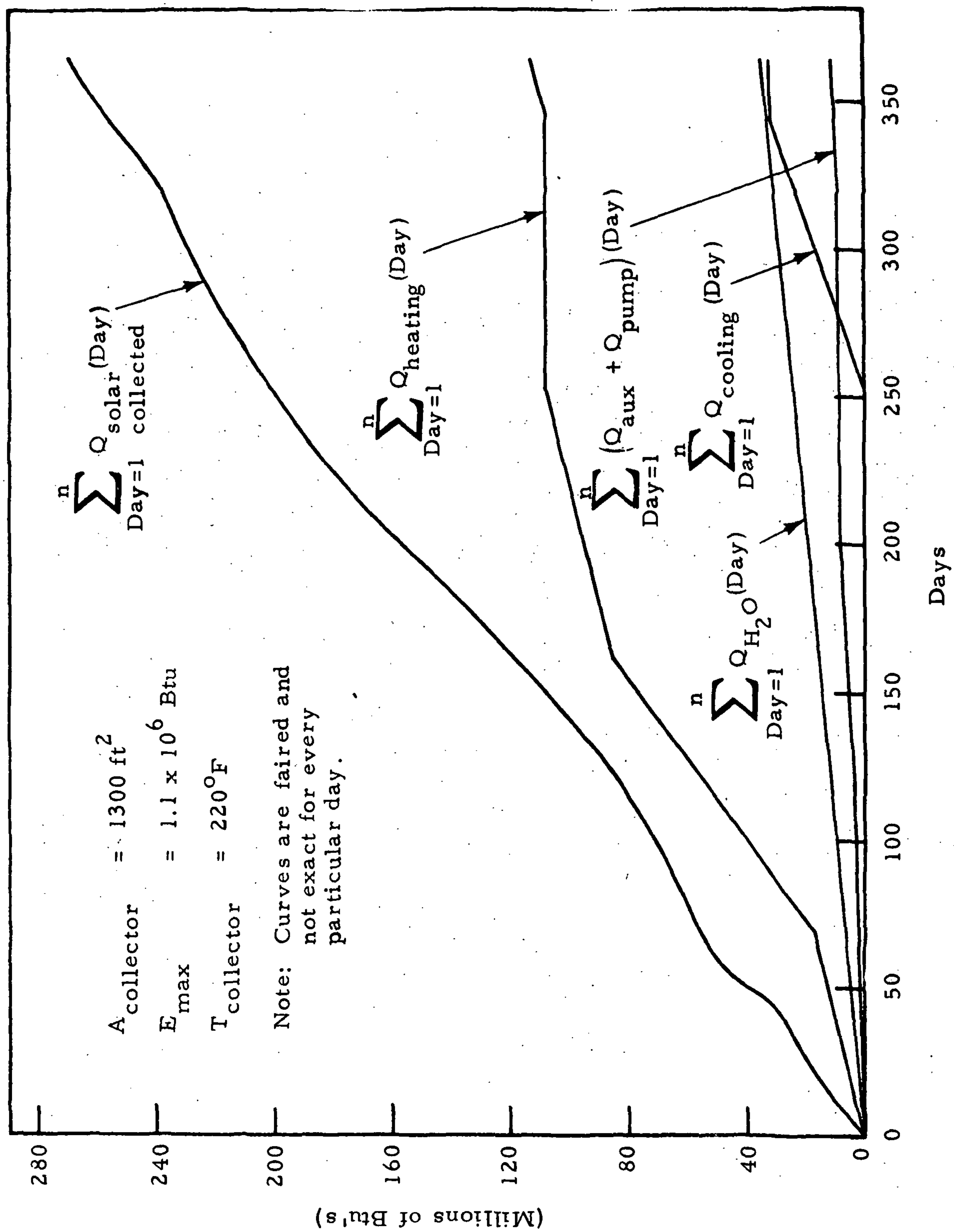


Fig. 6-5 - Yearly Performance of Solar Powered Heat Pump

and water heating. The excess energy collected is added to the storage system until it is fully charged with energy. Any additional solar energy collected must be wasted, since there is no place to store it. If water is used as the energy storage medium, the container could be designed to withstand pressures higher than the 6 psig required to maintain the liquid state at $230^{\circ}F$. If so, the water could be allowed to heat up to 240 or $250^{\circ}F$ on days when energy supply far exceeds its demand. For such a system, the auxiliary energy requirements would be substantially less than those depicted in Fig. 6-5.

The energy storage system performance is further illustrated by Fig. 6-6, which shows the daily variation in stored energy for one of the worst monthly periods of the year. On days 1, 4, 5, 6, 11, 12 and 30, more solar energy was collected than was needed. However, since the computer model will not allow the stored energy to exceed $E_{max} = 1.1 \times 10^6$ Btu, a large quantity of solar energy available was wasted for lack of a place to put it. And, unfortunately, on days 16, 19, 20, 21, 24, 25, 26, 27 and 28, more energy was required than available, and auxiliary energy had to be added to the system to make up the difference. Therefore, an obvious challenge to the inventiveness of the designer of such an energy storage system is to more effectively store the collected energy for longer periods economically. However, the optimized system did perform excellently in the total system analysis, and the following section shows that the solar-powered system offers significant economic superiority over a conventional heat pump system.

6.4 DETAILED COST COMPARISON OF SOLAR-POWERED SYSTEM WITH CONVENTIONAL SYSTEM

In spite of the obvious benefits of the solar powered heating, cooling and water heating system in the areas of energy, pollution and natural resources, the system will never be widely adopted unless it is proven to be economically competitive to conventional systems. To determine the economic competitiveness of the solar powered system, a detailed cost comparison of the two systems was made during the current study. The basis for comparison is presented in Table 6-2. The outputs, environment, power source, and lifetime were identical for both systems to make the comparison a fair one. The conventional heat pump

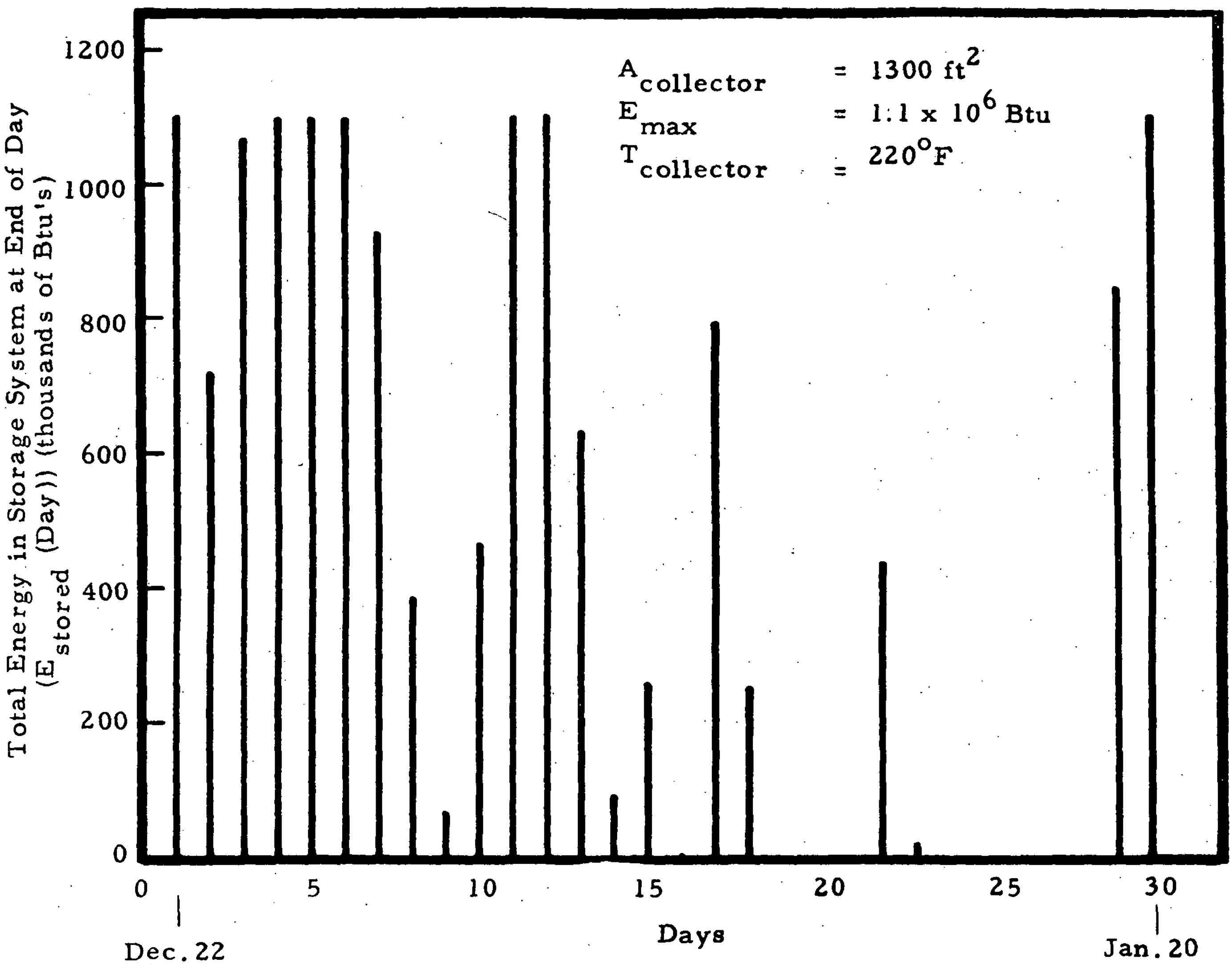


Fig. 6-6 - Daily Variation in Stored Energy

Table 6-2 - Basis of Comparison Between Solar-Powered and Conventional Systems

	Solar Powered System	Conventional System
Installation Date	1975	1975
Heating Requirements	Fig. 6-1	Fig. 6-1
Cooling Requirements	Fig. 6-1	Fig. 6-1
H_2O Heating Requirements	4000 Btu/hr	4000 Btu/hr
Environment	Fig. 6-2	Fig. 6-2
Heat Pump	(as described in Section 5)	Typical Commercial 3-Ton Unit
Water Heater	(as described in Section 2)	Electric
Initial Cost	Collector $1300 Energy Storage System 550 Heat Pump 1600 Total $3450	Heat Pump $1600 H_2O Heater 150 Total $1750
Power	Electrical for pump and auxiliary heating	Electrical for heat pump compressor and water heater
Maintenance	Tedlar replacement every 10 years	Compressor replacement every 5 years
Lifetime	20 years	20 years

cost and compressor replacement frequency and cost were obtained from heat pump manufacturers. To reflect unavoidable electricity cost increases over the 20-year period, a 1975 base cost estimate and a 5% estimated yearly increase in cost were based on conservative predictions from Huntsville Utilities.

The results of the cost comparison are presented in Fig. 6-7. It must be stressed that these results are based upon rough cost estimates for the solar-powered system and, therefore, should not be construed as extremely accurate. However, the basic trends are apparent in Fig. 6-7. The solar-powered system will cost more initially than a conventional system, but will become economically superior in a relatively short period of time. Although accurate conclusions must await the actual fabrication and demonstration of a prototype solar-powered system, the following conclusions are made with reasonable confidence:

- Although the initial investment in the solar-powered system will be greater than for the conventional system, the solar-powered system will become superior in cost within a few years, probably 3 to 7 years.
- Over the lifetime of the solar-powered system, the owner will pay less than half what his neighbors with conventional systems will pay for heating, cooling, and water heating. This savings will amount to thousands of dollars, probably $8000 to $10,000.

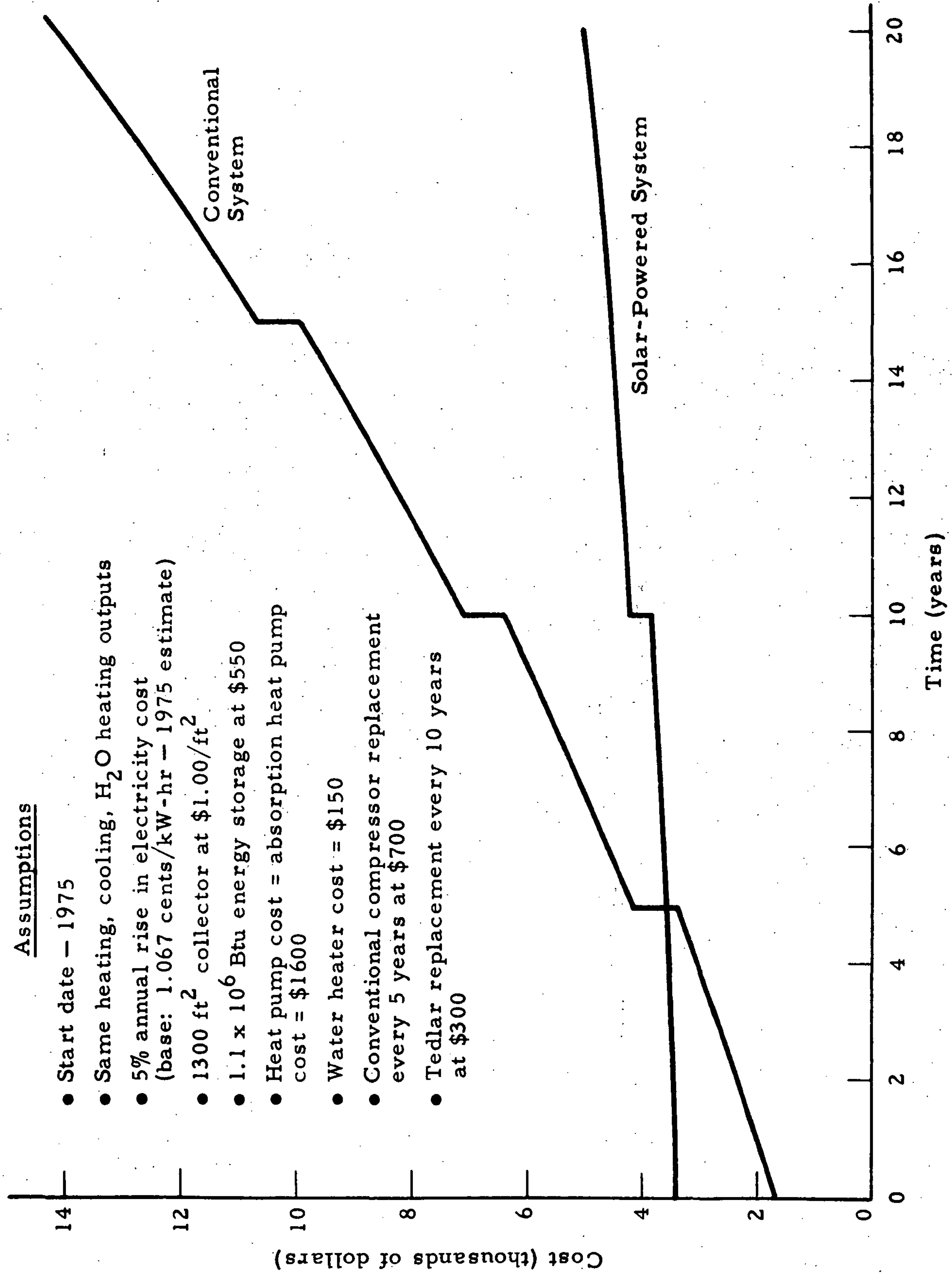


Fig. 6-7 - Cost Comparison Between Conventional and Solar-Powered Heat Pump/H_2O Heating Systems

Section 7
SYSTEM APPLICABILITY TO DIFFERENT REGIONS OF THE UNITED STATES

The applicability of the solar-powered system to different regions of the nation is a complex matter. The only accurate method of judging this applicability is to conduct a detailed analysis of the system for the actual weather conditions and solar conditions peculiar to a particular locale, as was done for Huntsville in the current study (Section 6). After such an analysis is conducted, a cost comparison between the solar-powered system and a conventional system will determine the applicability.

Under the current contract, detailed analyses for more than one location were beyond the scope of effort. Therefore, accurate estimates of system applicability for locations significantly removed from the Huntsville area have not been made. However, several significant facts strongly imply that the system should be applicable for most regions within the United States, as given below.

- The availability of solar radiation does not vary too greatly over most of the nation. Figure 7-1 shows the average daily total radiation received by a horizontal surface for each of the twelve months of the year (Ref. 4). Within the bands shown on the figure, all of the following cities are included:

  Atlanta, Ga.
  New York, N.Y.
  Miami, Fla.
  Los Angeles, Calif.
  Rapid City, S.D.
  Phoenix, Ariz.
  Portland, Me.
  Fort Worth, Texas
  Indianapolis, Ind.

  The points for Atlanta are shown on the figure since daily solar data from Atlanta were used in the detailed analyses conducted during this study. The Atlanta data are closer

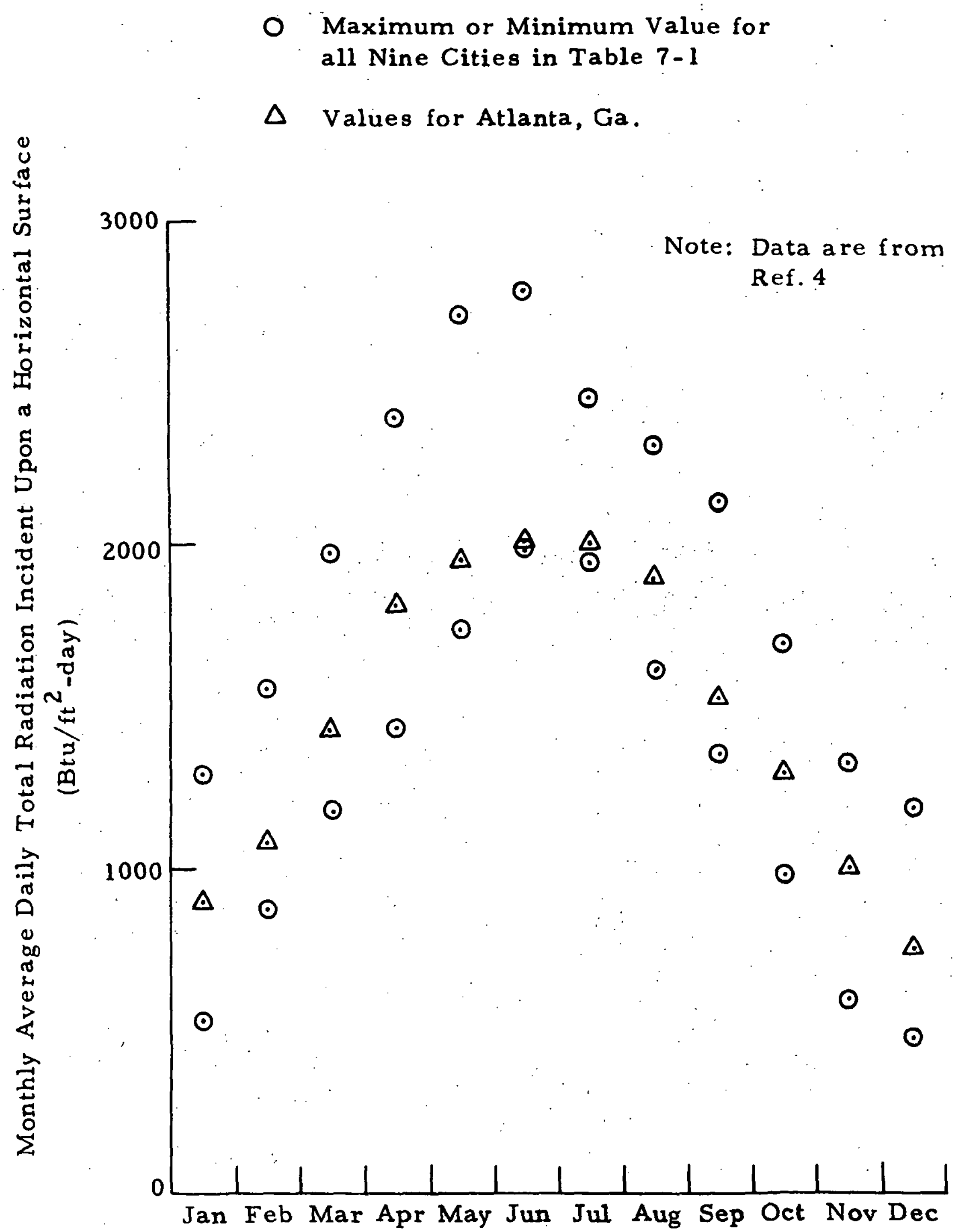


Fig. 7-1 - Monthly Average Daily Total Radiation Data for Nine U.S. Cities

to the minimum band of Fig. 7-1 than to the maximum band, thus indicating that the solar radiation assumed available in the current study is below the median available in these nine cities. The actual data for these cities is presented in Table 7-1 (Ref. 4).

- The cost of power for other regions of the nation is much higher than for Huntsville. The national average cost of electricity is more than double the Huntsville rate. Thus the potential savings in operating cost are greater for most regions of the nation than for Huntsville.

- In regions of the nation with more severe climates than Huntsville, the solar collector, energy storage system, and heat pump would have to be larger in capacity to meet the larger heating and/or cooling requirements. Thus, the initial investment in hardware would be greater. However, because greater heating and/or cooling requirements are needed, the potential savings in operating cost would also be greater than for Huntsville.

Therefore, based upon all indications discovered to date, the solar-powered heating, cooling and water heating system should be applicable to most regions of the nation with proper modifications in sizing and operating conditions.

Table 7-1

MONTHLY AVERAGE DAILY TOTAL SOLAR RADIATION INCIDENT UPON A HORIZONTAL SURFACE FOR NINE U. S. CITIES

City	Month											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Atlanta, Ga.	848*	1080	1427	1807	1953**	2003	2003	1898	1519	1291	998	752
New York, N.Y.	540	791	1180	1426	1738	1994	1939	1606	1349	978	598	476
Miami, Fla.	1292	1555	1829	2021	2069	1992	1993	1891	1647	1437	1321	1183
Los Angeles, Calif.	931	1284	1730	1948	2197	2272	2414	2155	1898	1373	1082	901
Rapid City, S.D.	688	1033	1504	1807	2028	2194	2236	2020	1628	1179	763	590
Phoenix, Ariz.	1127	1515	1967	2388	2710	2782	2451	2300	2131	1689	1290	1041
Portland, Me.	566	875	1330	1528	1923	2017	2095	1799	1429	1035	592	508
Fort Worth, Texas	936	1199	1598	1829	2105	2438	2293	2217	1881	1476	1148	914
Indianapolis, Ind.	526	797	1184	1184	1829	2042	2040	1832	1513	1094	622	491

Note: Data are from Ref. 4

*Btu/ft^2-day

** The value for May for Atlanta in Ref. 4 was apparently in error. The value shown is the average for 1969, 1970 and 1971 from U. S. Weather Bureau data.

Section 8
CONCLUSIONS

The basic conclusion of the study to date is that the solar-powered residential heating, air-conditioning, and hot water heating system is technically feasible and economically competitive with conventional systems. The following important conclusions were also drawn from the results of the study.

- The optimum solar collector design for the system consists of two transparent Tedlar covers over an aluminum plate treated with the NASA-developed selective coating. This coating offers excellent thermal performance at minimal cost and was developed by NASA-MSFC through in-house efforts during the current study. The collector should face southward with a tilt angle of 45 degrees for operation in the Huntsville, Alabama, area.
- At this time, water appears economically superior to available PCMs as the thermal energy storage substance, although PCMs may become superior to water through future research. An insulated, slightly pressurized container is envisioned for storing the water. Interfaces between the stored water and the collector and generator have been conceived and pose no serious design difficulties.
- The absorption cycle heat pump should utilize ammonia and water as the working fluids, and should operate at the conditions specified previously. The power requirements for the heat pump will be minimal and the performance adequate. No serious problems are foreseen in designing, fabricating and testing such a heat pump, since the operating conditions are similar to those available in current commercial units.
- During this study, the heat pump design was discussed with Arkla Air Conditioning Company, a major manufacturer of commercial absorption machines. They saw no obvious problems in achieving eventual mass production of such machines at reasonable costs.
- The total system has been optimized based upon preliminary cost estimates. From this optimization, 1300 ft^2 of collector, 1.1×10^6 Btu of energy storage, and a collector temperature of 220°F were found to be best for a typical home in Huntsville, Alabama. Before actual full-scale construction begins, the

system should be reoptimized for the actual test building, and updated cost estimates should be used if available.

- The optimized system will cost more than a conventional system for initial hardware and assembly. However, in about five years the solar powered system should demonstrate economic superiority over conventional systems. Over the lifetime of the system, the owner will save on the order of $10,000 in total system cost, compared to a conventional system.
- The system should be applicable to most regions of the United States, with proper modifications in collector area, energy storage system capacity, and heating and cooling outputs.
- If the system achieves widespread usage, dramatic effects will be observed in alleviating the national energy crisis, reducing pollution and preserving natural resources.
- When the solar-powered heating, cooling, and water heating system was originally conceived, uncertainties concerning its performance and cost precluded immediate construction and testing of the system. Therefore, this feasibility study was conducted to determine through extensive analysis the actual technical and economic feasibility. Every aspect of the system has been verified feasible to the extent possible through analysis. Thus, the technical risks involved in actually fabricating and testing a prototype system are currently considered minimal.

In light of the previous conclusions, efforts should begin immediately to design, fabricate and test a full-scale system to demonstrate its practicality. Specific recommendations for such a demonstration program are presented in the following section.

Section 9
RECOMMENDATIONS FOR PHASE II STUDY

The overall development of the solar-powered space heating, air-conditioning and hot water heating system should logically consist of three basic phases, as shown below.

- Phase I — Analytical Feasibility Study
- Phase II — Design and Fabrication of Full-Scale Demonstration Unit
- Phase III — Experimental Verification Program.

Phase I has been successfully completed and the feasibility has been verified analytically. Phase II is envisioned as a one-year program with a complete demonstration system as the major output. Phase III will require one calendar year of testing followed by several weeks of reducing and evaluating data. When the three phases are successfully completed, the system will be fully verified and ready for widespread application.

A recommended plan for the Phase II study was generated during the current study. The basic flow chart of efforts is presented in Fig. 9-1. A building will be selected and modified to simulate the heating, cooling and water heating requirements of a typical Huntsville residence. The three major systems will be designed concurrently and with a large amount of interaction to assure an optimum total system design. An instrumentation system will be designed to measure all pertinent quantities. The total system will be fabricated, and a detailed test plan will be developed. This flow chart represents an orderly manner of proceeding from the end of Phase I to the beginning of Phase III. The details of each effort in Fig. 9-1 are presented in the appendix.

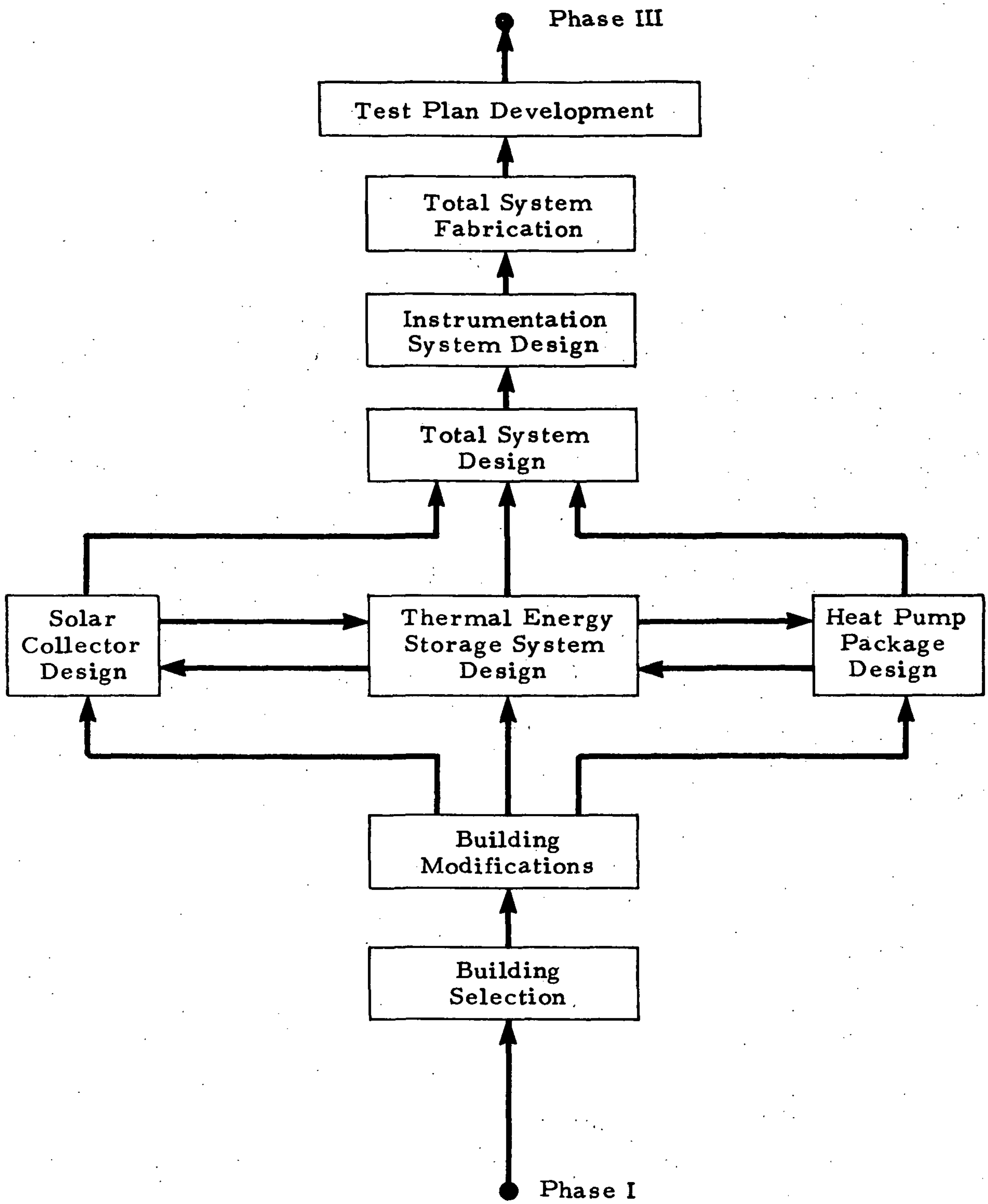


Fig. 9-1 - Flow Chart of Phase II Efforts

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Appendix
DETAILS OF PHASE II EFFORTS

Appendix

Each of the major efforts delineated in Fig. 9-1 is outlined below.

I. Building Selection

A. Cost Minimization

1. Use available NASA building and/or
2. Use prefabricated home or trailer

B. Location

1. Unshaded area
2. Non-protected area

C. Size

1. Should approximate typical Huntsville residence
2. Should allow collector installation on roof

II. Building Modifications

A. Modify Insulation if Necessary to Simulate Typical Huntsville Residence

B. Add Equipment to Simulate Typical Heat Loads, Cooling Loads, and Humidity Loads Produced by Occupants, Appliances, etc.

III-(i). Solar Collector Design

A. Structural Design

B. Thermal Design

1. Fluid loop design
2. Insulation design

C. Aesthetic Design

D. Cost Optimization Studies

1. Materials
2. Manufacturing processes
3. Installation techniques
4. Lifetime maximization

E. Control System Design

III-(ii). Thermal Energy Storage System Design

A. Structural Design

B. Thermal Design

1. Fluid loop to collector
2. Fluid loop to generator
3. Heat exchangers for hot water and generator
4. Insulation of tank

C. Aesthetic Design

D. Cost Optimization Studies

1. Materials
2. Manufacturing processes
3. Installation techniques
4. Lifetime maximization

E. Control System Design

III-(iii). Heat Pump Design

A. Work in Conjunction with Commercial Heat Pump Manufacturer

B. Heat Exchanger Designs

C. Air Distribution System Design

D. Blower Designs

E. Control System Design

F. Pump Design

G. Cost Optimization Studies
   1. Materials
   2. Manufacturing processes
   3. Installation techniques
   4. Lifetime maximization

IV. Total System Design

A. Develop More Detailed Cost and Performance Model of Total System
B. Use Model to Evaluate Each Proposed Design Detail for Collector, Energy Storage System, and Heat Pump to Determine Effect on Total System Cost and Performance
C. Determine Optimum Design from Total System Viewpoint
D. Determine Best Interface Between System and Test Building
E. Develop Sketches Suitable for Fabrication of All System Components

V. Instrumentation System Design

A. Measure all Heat Transfer Rates
B. Measure all Important Temperatures
C. Measure Weather Quantities
   1. Temperature
   2. Humidity
   3. Wind speed and direction
   4. Solar radiation
D. Measure Cycle Conditions
E. Measure Temperature and Humidity Distribution Within Building
F. Automate Data Acquisition System

VI. Total System Fabrication

VII. Test Plan Development

A. Explain all Measurements

B. How to Evaluate Measurements

C. Expected Variation in Measurements

D. Errors in Measurements

VIII. Phase II Final Report

A. Details of All Results

B. Plans for Phase III

132 units

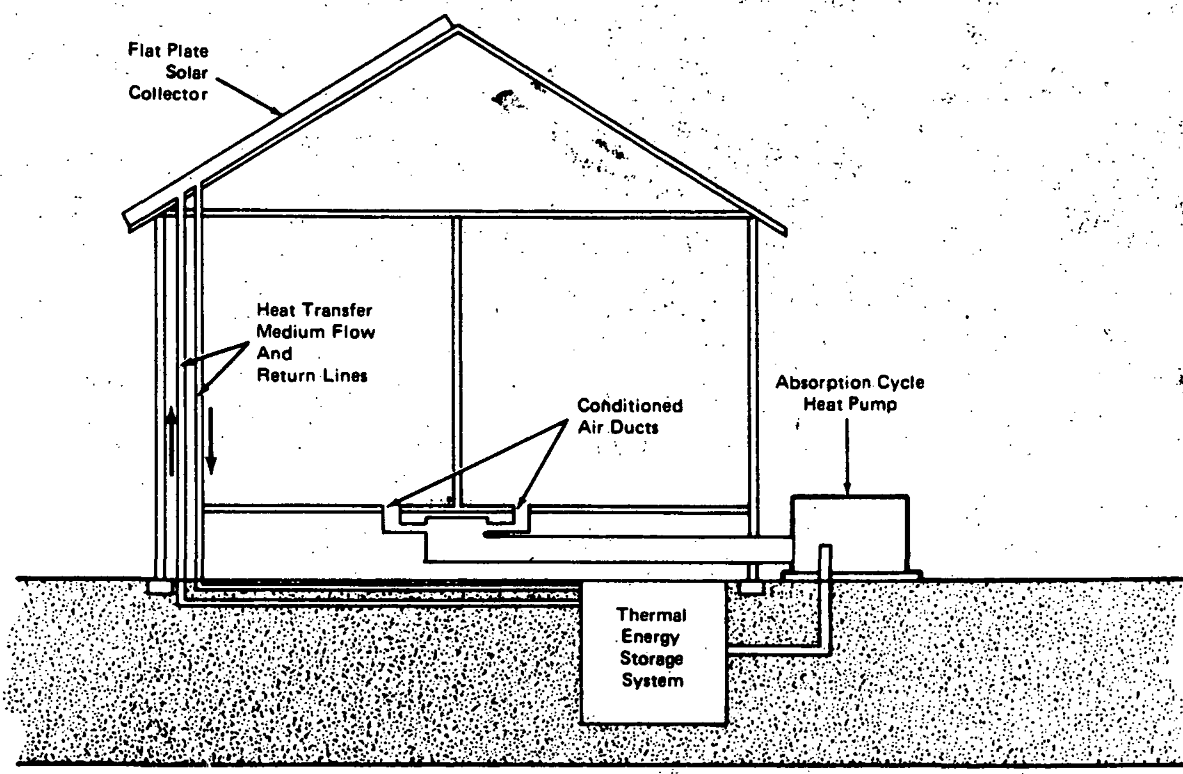
NASA—MSFC—C

May 1973

B73-10156

NASA TECH BRIEF
Marshall Space Flight Center

A Practical Solar Energy Heating and Cooling System



Schematic of Solar-Powered Space Heating, Air-Conditioning and Hot Water Heating System

A recent study has concluded that a solar-powered residential heating and cooling system is now technically and economically feasible. The proposed system provides space heating, air conditioning, and hot water. The illustration shows how the system could be used in a typical home. The major components are:

(a) a flat-plate solar collector to process solar radiation,
(b) a thermal-energy storage system to store the collected energy for use during night and heavily overcast periods.
(c) an absorption cycle heat pump for both heating and cooling the residence, and
(d) a hot water system (not shown) that uses heat from the energy storage system.

The best solar collector was predicted to consist of two transparent covers over an aluminum thermal absorber plate treated with a special selective coating. The orientation of the collector can be optimized for a particular geographic location. The heat is transferred to a fluid that carries it to the energy storage system.

(continued overleaf)

The energy transfer fluid and the energy storage fluid may be water or a phase-change material. Economic and technical analyses of candidate fluids indicate that water is the best choice for the energy storage substance. In the system it is stored in an insulated, slightly pressurized container.

An ammonia and water mixture is the most efficient heat pump working fluid. Using this mixture no serious problems are envisioned in the design and fabrication of a heat pump, since the required pump will be similar to commercial units.

As part of the study, mathematical models have been constructed for the analysis and evaluation of all phases of the system. Both technical and economic criteria have been considered in the selection of an optimal system from among several alternatives.

The system should be usable in all parts of the United States. The costs of installation will be greater than for conventional heating systems, but this differential will be defrayed after a few years of service by the very low operating costs. In fact, in the long run, solar-energy heating and cooling will be less expensive than present methods.

Notes:

1. This study was a feasibility project, and an actual working model has not yet been tested. However, plans are underway for the construction and testing of prototypes.
2. Requests for further information may be directed to:
   Technology Utilization Office
   Marshall Space Flight Center
   Code A&PS-TU
   Marshall Space Flight Center, Alabama 35812
   Reference: B73-10156

Patent status:

NASA has decided not to apply for a patent.

Source: M. J. O'Neill, A. J. McDanal,
and W. H. Sims of
Lockheed Aircraft Corp.
under contract to
Marshall Space Flight Center
(MFS-22563)

73-10156
MFS-22567

Lockheed
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COMPANY

HUNTSVILLE RESEARCH & ENGINEERING CENTER • P. O. BOX 1103 WEST STATION • HUNTSVILLE, ALABAMA • 35807

6 December 1972

New Technology Utilization Program

Title: The Development of a Residential Heating and Cooling System Using NASA-Derived Technology," Contract NAS8-25986 (Mod. 5)

Brief Description: Under the subject contract, preliminary analytical development was completed and preliminary designs were generated for a system capable of providing space heating, air-conditioning, and hot water heating for a typical residence, using solar energy as the primary input. The system concept was proven to be technically feasible and cost-competitive with conventional systems. The major benefits of the system are in the areas of energy conservation, pollution abatement, natural resource preservation, and significant cost savings for the user.

Detailed Description: The general purpose of the item is to utilize available solar energy to provide heating, cooling, and hot water heating for a residence.

The solar powered system represents several advantages over conventional systems because the primary driving energy is solar energy rather than electricity, natural gas, or fuel oil. Some of these advantages are listed below.

- If the system receives widespread adoption, the national energy crisis will be favorably impacted, since one-fourth of the total energy consumed in the U.S. is used for space heating, air-conditioning, and water heating.
- Air, water, thermal, nuclear, and solid-waste pollution will be favorably impacted by the system because of reduced needs for electricity and fossil fuel combustion.
- Natural resource preservation will be aided by the system. Also, the environmental destruction due to mining and the dependence on foreign powers for petroleum will be reduced.

 11 Write /

- Since solar energy is free, the owners of the system will realize great savings in operational costs.

The principle of operation of the system is simple. Incident solar radiation is processed by a flat-plate solar collector located on the roof of the residence or elsewhere nearby. The collected energy is stored in a thermal energy storage system, using either the latent heat of fusion of a phase change material (PCM) or a sensible heat storage in water. Actual heating or cooling of the residence is obtained from an absorption cycle heat pump. An absorption cycle machine uses thermal energy rather than electrical energy as the primary input. In this case, thermal energy from the energy storage system powers the heat pump. The outside power required by the solar-powered system is similar in magnitude to a light bulb load. In addition, heat from the energy storage system is used to provide domestic hot water.

Several features of the system are believed to be new, including:

- The system will provide all heating, cooling, and water heating requirements for the residence with only minimal outside power.
- The system utilizes an absorption cycle heat pump for heating and cooling. During heating, therefore, more thermal energy is added to the conditioned space than is collected since additional heat is absorbed from the outside air.
- Methods and computer programs have been developed to optimize the system for a particular locale, thereby yielding required performance at minimum cost.
- The system offers significant economic savings to the user by comparison to conventional systems, on the order of $10,000 over a 20 year lifetime.
- The system utilizes new and improved designs for the solar collector, energy storage system, and absorption cycle machine.

Additional information about the solar-powered system is provided in the attached report.

Applications: The system should be applicable for residential usage throughout most of the U.S. with proper sizing of the collector, energy storage system, and heat pump. The major user will be the average American home owner.

Possible Extensions: The system should be extendable to heating, cooling, and water heating for apartment complexes, business facilities, factories, etc.

Degree of Development: Concept only.

Technological Significance: In relation to present state of technology, this reportable item is considered to be a major improvement.

Innovator: Mark J. O'Neill

Publications: Attached report

Technical Supervisor of Innovator: Juan K. Lovin, Dept. 54-01

D-03-6043

Disclosure of Invention (Cont'd)

M. J. O'Neill, E/N 668036 - Inventor   M. J. O'Neill 1/27/72
W. H. Sims, E/N 611014 - Co-Inventor   W. H. Sims 1/27/72

Title: Solar Collector/PCM/Absorption Refrigeration Home Heating and Cooling System

1. Collection of solar energy
2. Storage of this energy through heat of fusion in the PCM
3. Using the stored energy to drive an absorption refrigeration system
4. Using, in summer, the cooling capacity of the evaporator to cool and dehumidify the air in a building
5. Using, in winter, the heating capacity of the absorber and condenser to heat the air in a building.

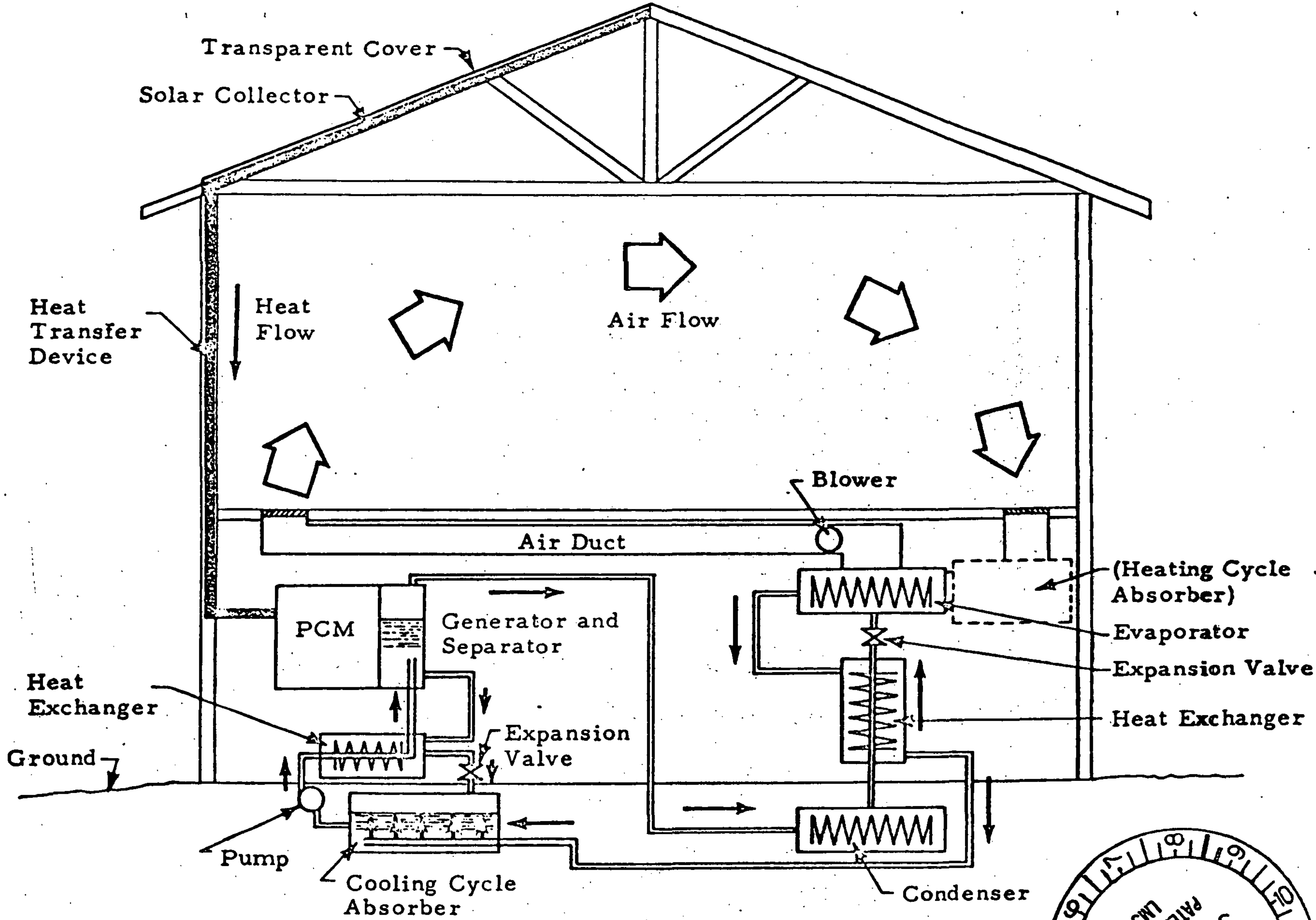

Fig. 1 - Schematic of Solar Home Air Conditioning System

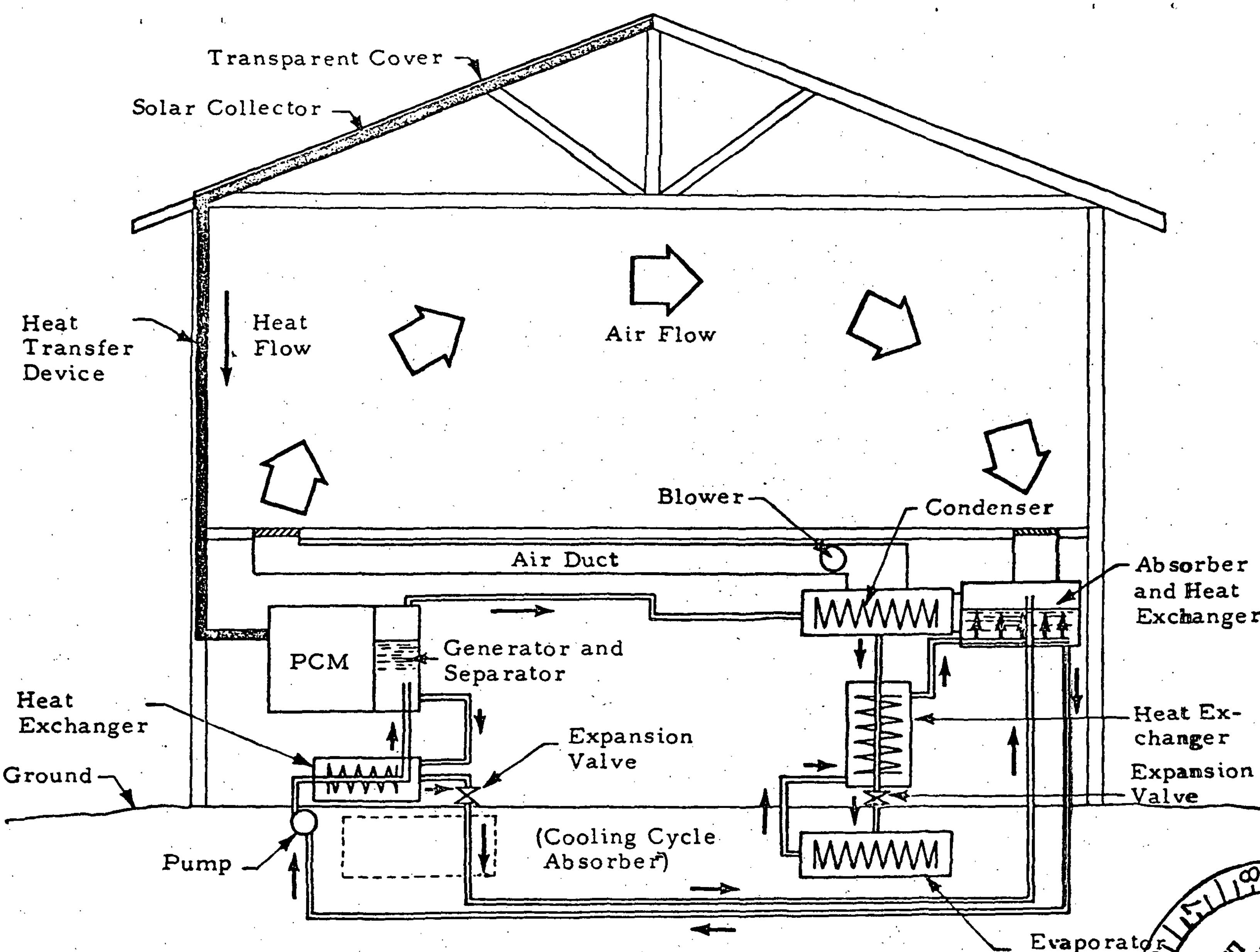

Fig. 2 - SCHEMATIC OF SOLAR HOME HEAT PUMP SYSTEM

D-03- 6043

RECEIVED JAN 31 1972 PATENT OFFICE LMSC

DISCLOSURE OF INVENTION

PATENT DEPARTMENT

LOCKHEED MISSILES & SPACE COMPANY

This disclosure of invention form sheet is for the purpose of securing a disclosure and record date of invention and it is important that it be made out and filed with the Patent Department as soon as possible after conception of the invention in order that priority rights to the invention may be secured. A separate sheet may be used for each invention or modification of the invention and each sheet should be signed and dated by the inventor and also signed and dated by witnesses, preferably two, by whom the contents of the disclosure have been read and understood.

IN THE SPACE BELOW, give a clear and concise explanation of the invention. If it is purely a process give a complete description of it including flow diagrams; if it is a method capable of being illustrated by a sketch or where it is an apparatus, circuit or mechanical device, the disclosure should consist of a sketch with the parts numbered with a description of the sketch and method of operation making reference to the numbered parts. If possible a specific illustrative example and operative description of the invention should be included.

If the space below is inadequate, attach separate drawings or prints and description (properly signed, witnessed and dated, if possible).

All of the following entries should be made preferably in ink or type.

1. TITLE OF INVENTION Solar Collector/PCM/Absorption Refrigeration Home Heating and Cooling System

2. SKETCH AND DESCRIPTION OF INVENTION

(Also list and identify herein all attached drawings and descriptions)

Fig. 1 - Schematic of Solar Home Air Conditioning System
Fig. 2 - Schematic of Solar Home Heat Pump System
Fig. 3 Schematic of Absorption Refrigeration Cycle

The concept of using solar energy for cooling and heating a home will employ solar collectors, phase change materials (PCM) and an absorption refrigeration system. The basic concept is as follows:

Solar energy, in both direct and diffuse radiation form, strikes the solar collector, which is an energy-absorbing system such as flat-black aluminum, with a glass cover to prevent re-radiation into the atmosphere. The collector is insulated on the bottom side to prevent conduction and radiation into the ceiling area of the home. Circulating through the collector is a heat transfer fluid, such as water or higher boiling temperature fluid. The fluid circulates through insulated lines to the PCM container. An appropriate heat-exchange mode transfers the thermal energy into the bulk of the PCM (such as paraffins or salts) which exhibit high energy storage capabilities during a phase change from solid to liquid. The heat transfer fluid then recirculates to the collector. The thermal energy stored in the PCM is then released to the generator component through conduction heat transfer. This energy is used to boil the solution in the generator in order to vaporize the refrigeration fluid (such as ammonia or freon). The refrigerant then cycles through the evaporator and condenser in basically the same process as found in a standard heat pump. The absorption fluid in the meanwhile is being circulated back to the absorber, in a weak solution form. Here the refrigerant is reabsorbed in the liquid carrier and re-enters the generator to begin the process over again. During a heating cycle, an auxiliary heat source is the absorber which is switched (electrically) into the system by thermostatic operation. During this process, heat is added to the home air by both the condenser and the absorber, increasing the efficiency of the operation. The extremely important point of the operation of the system is the almost total lack of external power sources (electrical) required for the system. The only input requirement is the liquid pump drive system which will consume only about 100-150 watts but not on a continuous basis. This represents a cost of a few pennies per month operational costs for both heating and cooling an average residence. The solar collector will be able to utilize both direct and diffuse solar radiation since no focusing or reflecting optical devices are employed. This will allow energy collection of significant quantity even on overcast days. Conservative estimates of operating performance indicate the solar collector/PCM/absorption cycle system can adequately heat and cool an average home anywhere in the U.S. with minimal auxiliary heating or cooling requirements, and with reasonable collector areas and PCM volumes. Although the prelimimary calculations have been based on ammonia and water as the refrigerant and absorbent, respectively, and on the temperatures and pressures given in Fig. 3, other fluids, temperatures, and pressures may represent improved system operation. Therefore, these fluids, temperatures, and pressures are presented here only to give a practical example of the solar/PCM/absorption concept and are not integral parts of the concept. The basic concept may be summarized as presented on the attached page. (See attached page)

FORM LMSC

(over)

3. PURPOSE of the invention To utilize solar energy for heating and cooling residential and office buildings, which will result in reduced power generation requirements, lower pollution products and much reduced cost for heating and cooling to the consumer.

4. PREVIOUS METHOD or apparatus Electrically driven heat pump, and forced air heating with central air conditioning.

5. INFORMATION on previous method or apparatus; known use, publication or patents None known to exist utilizing all concepts.

6. HOW does this invention differ from previous method or apparatus and what advantages does it offer? Major difference is mode of work input, major advantages include lower power generation requirements, lower environmental pollution and lower operational costs.

7. DATE OF CONCEPTION (when you first thought of the idea) December 15, 19 71

8. CHARGE NUMBER(S) or Work Order Number(s) to which each inventor was charging his time on the day the concept was completed. (Note: Charging time to a Government contract does not automatically entitle the Government to rights in an invention.) N1-RA10-5420 (NAS8 25986)

9. (a) First sketch or drawing made on December 17, 19 71

Where filed LMSC-HREC

(b) First written description made on January 15, 19 72

Where filed LMSC-HREC (LMSC D082102-812)

NOTE: Where possible the above sketches, drawings and descriptions should be attached to this sheet.

10. INVENTION was first disclosed to:

(1) J. K. Lovin Date 15 December, 19 71, How Verbal

(2) G. D. Reny Date 15 December, 19 71, How Verbal

11. FIRST APPARATUS (a) started on N/A, 19 (b) completed on N/A, 19

12. FIRST OPERATION of apparatus or process (a) started N/A, 19 (b) completed , 19

(c) Observed by N/A and

(d) Apparatus or result of process located at N/A

13. OTHER ACTS tending to prove conception, such as preparation of calculations, preparation of shop order for model, etc., giving dates and state where such data is filed:

Calculations
12/15/71 - 12/22/71,
filed LMSC-HREC

14. ~~I~~ (we) hereby certify that, to the best of ~~my~~ (our) knowledge, ~~I am~~ (we are) the first and original inventor(s) of the subject matter hereinbefore described.

this 27 day of January, 19 72

M. J. O'Neill
Inventor's Signature

Employee No. 668036

NAME (Please Print) M.J. O'Neill

Home Address 2601 Fantasia Dr., S.W., Huntsville, Ala.

Orgn. 54-20 Bldg. 401 Phone Extension 256

LMSC Division Name Huntsville Research & Engineering Center

W. H. Sims
Co-Inventor's Signature

Employee No. 611014

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Home Address 12112 Chicamauga Tr., S. E., Huntsville

Orgn. 54-20 Bldg. 401 Phone Extension 256

LMSC Division Name Huntsville Research & Engineering Center.

SHOWN and DESCRIBED to me/us

on this 28 day of Jan, 1972

[signature]

[signature]

WITNESSES' SIGNATURES

JAN 31 1972

FORM LMSC 473-1 (REVERSE)

D-03-6043

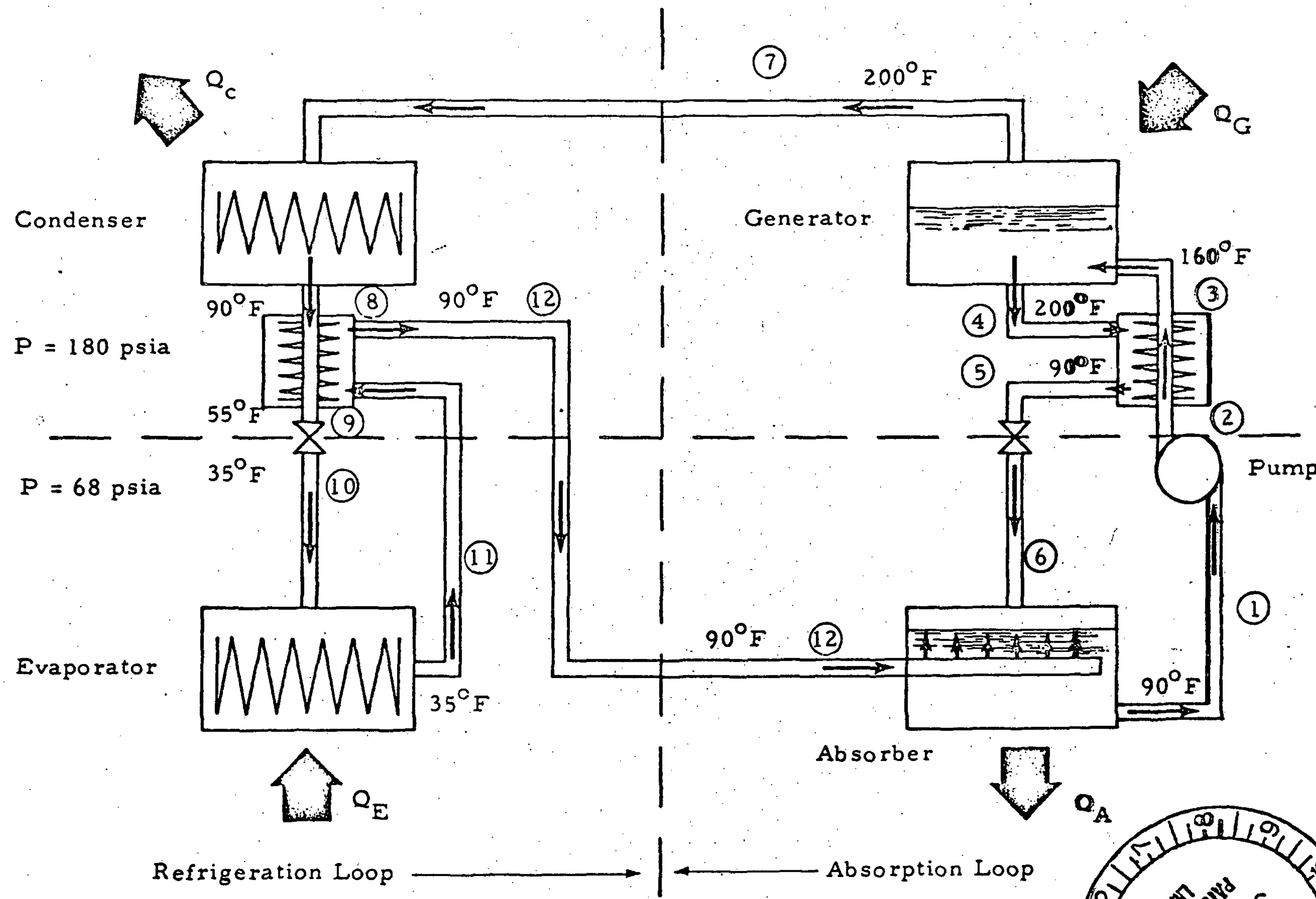

Fig. 3 – SCHEMATIC OF ABSORPTION REFRIGERATION CYCLE