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Municipal Environmental Research Laboratory
Cincinnati OH 45268

EPA-600/2-78-114
July 1978

Use of Solar Energy to Heat Anaerobic Digesters

Part I Technical and Economic Feasibility Study

Part II Economic Feasibility Throughout the United States

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July 1978

USE OF SOLAR ENERGY TO HEAT ANAEROBIC DIGESTERS

Part I
Technical and Economic Feasibility Study

Part II
Economic Feasibility Throughout the United States

by

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David E. Cassel
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Part I: Contract No. 68-03-2356
Part II: Order No. CA-6-99-3499-A

Project Officer

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FOREWORD

The Environmental Protection Agency was created because of increasing public and government concern about the dangers of pollution to the health and welfare of the American people. Noxious air, foul water, and spoiled land are tragic testimony to the deterioration of our natural environment. The complexity of that environment and the interplay between its components require a concentrated and integrated attack on the problem.

Research and development is that necessary first step in problem solution and it involves defining the problem, measuring its impact, and searching for solutions. The Municipal Environmental Research Laboratory develops new and improved technology and systems for the prevention, treatment, and management of wastewater and solid and hazardous waste pollutant discharges from municipal and community sources, for the preservation and treatment of public drinking water supplies, and to minimize the adverse economic, social, health, and aesthetic effects of pollution. This publication is one of the products of that research; a most vital communications link between the researcher and the user community.

This report contains the results of two studies in which the use of solar energy to heat anaerobic digesters was proven to be technically and economically feasible at Annapolis, Maryland and economically feasible at all other locations in the United States. Economic justification for using solar heat for anaerobic digestion was based on the value of the methane gas produced.

Francis T. Mayo
Director
Municipal Environmental Research
Laboratory - Cincinnati

ABSTRACT

Part I of this report was prepared as a result of a study based on the premise that heat requirements for anaerobic sludge digestion represent a large proportion of the energy used in most conventional wastewater treatment plants.

Digestor gas, consisting principally of methane, is commonly used as fuel for digester heating. Recognizing that if solar energy could be substituted as the prime heat source for anaerobic digestion, then the methane produced could be freed for higher grade energy requirements elsewhere. The technical and economic feasibility of providing this alternative heat source was evaluated by Environmental Systems, Incorporated. Detailed plans and specifications were also prepared for the addition of a solar heating system to the municipal wastewater treatment plant at Annapolis, Maryland.

To optimize the design, a computer program simulated the operation of the digester heating requirements over the annual cycle. The inclusion of all operating parameters and economic factors allowed precise determination of both the size and specific design of the solar heating system. The present worth of the solar heating system over the 25-year project life was compared with the present worth of the digester gas conserved over the same period. The system size and design were chosen as those which would provide the maximum value of gas conserved relative to the cost of the solar heat.

For an anaerobic digester maintained between 32 and 38 degrees Celsius, at Annapolis, Maryland, the optimum size of the solar heating system is that which will supply approximately 90 percent of the annual heat load. Flat-plate solar heat collectors having two glass covers proved to be the most cost-effective for this digester temperature range and location.

Part I of the report was submitted in fulfillment of Contract Number 68-03-2356, by Environmental Systems, Incorporated, under the sponsorship of the U.S. Environmental Protection Agency. Work was completed as of June 1976.

Part II of this report was prepared as the result of a study to apply the principles developed in Part I to other locations throughout the United States.

Solar digester heating is economically feasible at all locations in the nation. The degree of economic attractiveness at any given location is directly proportional to the average annual solar radiation multiplied by the difference between the digester design operating temperature (35°C) and the average annual air temperature.

The study shows that optimum-sized flat plate solar collectors can provide from 82 to 97 percent of the total annual digester heat load, the higher percentages being applicable to areas of higher solar radiation intensity. Specific guidelines are given for determining the optimum size and design of solar heating system for any size of sludge digester at any location.

Part II of this report was submitted in fulfillment of Order Number CA-6-99-3499-A by Environmental Systems, Incorporated, under the sponsorship of the U.S. Environmental Protection Agency. Work was completed as of December 1976.

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The authors wish to thank officials of the City of Annapolis, particularly William R. Jackson, former Director of Public Works, for records, plans, data and access to the Annapolis Wastewater Treatment Plant, without which Part I of this study could not have been conducted.

Special acknowledgement is extended to Dr. Bringham H. Van Dyke, Jr. of Van Dyke Associates, for aid in design of solar collector supply and return manifolds and collector mounting arrangment; Arnold F. Richter, for assistance and structural steel design and conceptulization of solar heat collection system; James Pluzinski of National Computer Network, for use of his time-sharing computer terminal and aid in programming; Dr. J. Taylor Beard of the University of Virginia, for providing advance performance data on the Solaris collector panels; Dr. Daniel Lufkin, Solar Consultant, for advice regarding solar collector designs; Daniel Fisher of the Carolina Solar Equipment Company, for the loan of a model Solaris collector panel; James Mazzullo of Solar Heating Services, Incorporated, for demonstration of the Revere collector panel; Dr. William McEver of Intertechnology Corporation, for inspection of the Fauquier High School solar energy system, Warrenton, Virginia; James Larrimore of Polytech, Incorporated, for advice and specifications regarding insulation for the solar heat storage tank; Leon Slavon of the Federal Power Commission, for information on natural gas prices, and John H. Koehnlein, Jr. of the Portland Cement Association for concrete reactor design information.

PART I

TECHNICAL AND ECONOMIC FEASIBILITY STUDY

by

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Contract No. 68-03-2356

SECTION 1

INTRODUCTION, CONCLUSIONS, AND RECOMMENDATIONS

INTRODUCTION

Anaerobic sewage sludge digesters are usually heated by burning digester gas, and in some cases by burning another fuel such as oil. To present an alternative to this use of "high grade" fuels, and to free the digester gas for other purposes, a six-month technical and economic feasibility study was performed to evaluate the use of solar energy for heating anaerobic digesters. There are many facets of energy use and conservation associated with sewage treatment, but this study is limited to analyzing direct solar heat as a heat source for the digesters.

CONCLUSIONS

It is technologically and economically feasible to heat digesters with solar energy. All of the necessary components are available as "off-the-shelf" items. Solar heat collector technology is changing rapidly, but there are collectors now available that are quite adequate.

Under the requirement of this study, that is, maintaining the digester temperature in the upper mesophilic range (32-38°C) year round and for weather conditions similar to those at Annapolis, Maryland, the lowest-cost method of heating the digester is to supply about 90 percent of the annual heat load with solar energy.

RECOMMENDATIONS

As a result of this feasibility study, we make the following recommendations:

1. The research plan contained herein should be carried out to positively demonstrate that the solar heating system will work as expected;

2. A further study should be undertaken, using the computer techniques and equations developed herein, to evaluate solar digester heating under different weather conditions for application throughout the United States; and

3. The role of solar energy in wastewater treatment should be evaluated as a source of energy for other processes (for example, sludge drying). The cost-effectiveness of solar energy should be compared with that of other energy conservation measures such as insulation.

SECTION 2

DESCRIPTION OF SOLAR HEATING SYSTEM

PRINCIPLES OF OPERATION

Solar Sludge Preheater

The solar heating system for the anaerobic digester is essentially a solar preheater for the raw sludge. Raw sludge from the primary clarifiers passes through the solar heat storage tank/heat exchanger where it is warmed before entering the digester. The sludge is preheated to the temperature required to provide all of the digester heat whenever possible. Any additional heat input to the digester enters via the conventional heating coils, which will be called the auxiliary heating system for the solar heated digester.

Heat Storage Tank/Heat Exchanger

The combination solar heat storage tank and sludge preheater consists of a tank of solar-heated water through which the raw sludge pipe passes. The water in the tank is kept as hot as possible in the winter, but the temperature of the water is limited in the summer to avoid overheating the raw sludge.

Solar Heat Collection System

When the solar collector control system calls for heat, the solar collector pump pumps water from the bottom of the heat storage tank to the flat-plate solar heat collectors, which are at a higher elevation than the tank. The water runs through the collectors and drains by gravity to the top of the tank. When the pump turns off, all water drains back into the tank.

Control System

The control system is completely automatic and very simple. Temperatures are sensed at three places in the system to control the on/off condition of the solar collector pump: 1) in one of the solar collectors; 2) at the heat storage tank; and, 3) in the digester. A differential thermostat turns the collector pump on when the temperature of the collectors is a set number of degrees above that in the heat storage tank. A high-limit thermostat prevents the collector pump from operating when the temperature in the digester reaches a set high temperature during summer operation. If the temperature in the digester reaches a set low limit, the auxiliary heat system adds heat directly to the digester.

DETAILED DESCRIPTION

Solar Heat Collection

The solar collector panels are commercially available flat-plate panels. They have two layers of tempered glass over the copper absorber sheet. This copper sheet has soldered to it copper tubes through which the water passes. The tubes are joined at top and bottom by larger tubes running horizontally. Inlet and outlet tubes join these tubes and penetrate the panel casing. The copper absorber sheet is backed by insulation. The panels are individually encased and are each self-contained units.

One hundred and thirty panels, comprising an effective collector area of about 230 square meters, are arranged in two planes facing seven degrees west of south, at an angle of 60 degrees from the horizontal. Each plane is two collector panels high. A reflector, consisting of a horizontal concrete slab painted white, equal in width to the height of the panels, is positioned at the base of and parallel to each array of panels. The concrete reflectors substantially increase the solar radiation on the panels during the critical winter period.

Pre-insulated plastic pipe carries the water from the heat storage tank to the collectors and back again. This piping consists generally of schedule 80 PVC or CPVC pipe plus a layer of urethane foam (rigid) insulation, with an outer shell of PVC pipe. Sections of this pipe run along the top and bottom of each collector plane as supply and return manifolds. Pipe nipples are tapped into the manifold for each connection to the collector panels. Steel frameworks with concrete footings support each row of panels.

The solar collector pump is a close-coupled centrifugal pump with a nominal 3/4 horsepower (560 W) electric motor. Water can flow back through the pump freely when it is not operating.

A filter in the discharge line of the pump prevents any solid particles from reaching the collectors. A pressure gage at the pump discharge indicates whether the pump is operating normally.

Solar Heat Storage and Heat Transfer

The solar heat storage tank is a 75 m3 (20,000 gallon) steel tank. The tank is placed horizontally. It is 9.45 m (31 feet) long and 3.2 m (10.5 feet) in diameter. For convenience at the Annapolis plant, the tank is placed in the empty Number Three digester, previously a floating-cover digester from which the cover has been removed. The nominal six-inch diameter steel sludge pipe enters near the bottom of one end of the tank, makes five full-length passes through the tank, and exits near the top of the other end. From there, it enters the Number Two digester through the digester wall. The warmed sludge is introduced into Digester Number Two in the center near the bottom. The five passes of six-inch steel pipe within the heat storage tank are supported by vertical steel rods welded to the pipe and tank.

The tank is insulated with sprayed-on urethane foam insulation to a thickness of 6.4 cm ($2\frac{1}{2}$ inches). A coating of hydrocide elastomeric roofing (HER) is sprayed onto the urethane for waterproofing and to protect the urethane from the sun's rays. This coating is 1.5 mm (0.060 inch) thick.

A sight glass is supplied with the tank to give a visual indication of the water level. Water can be added via a manual valve when necessary. The water level should be within 10 cm of the top of the tank when the pump is not operating.

A sump pump is placed under the tank in the bottom of the Number Three digester to remove any accumulated rainwater or seepage from the ground.

Control System

The differential thermostat and high-limit controller are located near the solar collector pump in the pump room adjacent to the Number Three digester. Wires run to the solar collector, the heat storage tank, and to the temperature sensor in the digester. The dual element temperature-sensing resistance bulb is located in a well in the digester wall. A #14 gage solid copper wire connects one element of the bulb to a controller in the pump room. The other element of the bulb is used to send a signal to the other controller for the auxiliary heat system. Rho Sigma model 106 differential thermostat and Rho Sigma model STH sensors will be used. Honeywell Model HP7E11-20-3A dual element resistance bulb with stainless steel well and housing will serve to measure digester temperature. The controllers will be Honeywell model 7351 Dialatrol.

Auxiliary Heat System

No additional components will be required for the auxiliary heat system since the solar heating design is completely independent of that system. The existing system will not be operated at all in the summer, and only at a fraction of its normal capacity in the winter. One optional control may be used with the auxiliary heating system to turn on the boiler when the digester temperature lowers to near the point where auxiliary heat may be needed. This is not necessary for adequate operation, however, and is not included in this design.

DETAILED OPERATION

Heat transfer rates (power) for various parts of the system over an annual cycle are shown graphically in Figure 1. Corresponding temperatures at various points are shown in Figure 2. All of the assumptions underlying these graphs will be explained in a later section of this report, but basically they represent average and typical conditions. The power given in Figure 1 can be considered to be five-day averages for typical conditions.

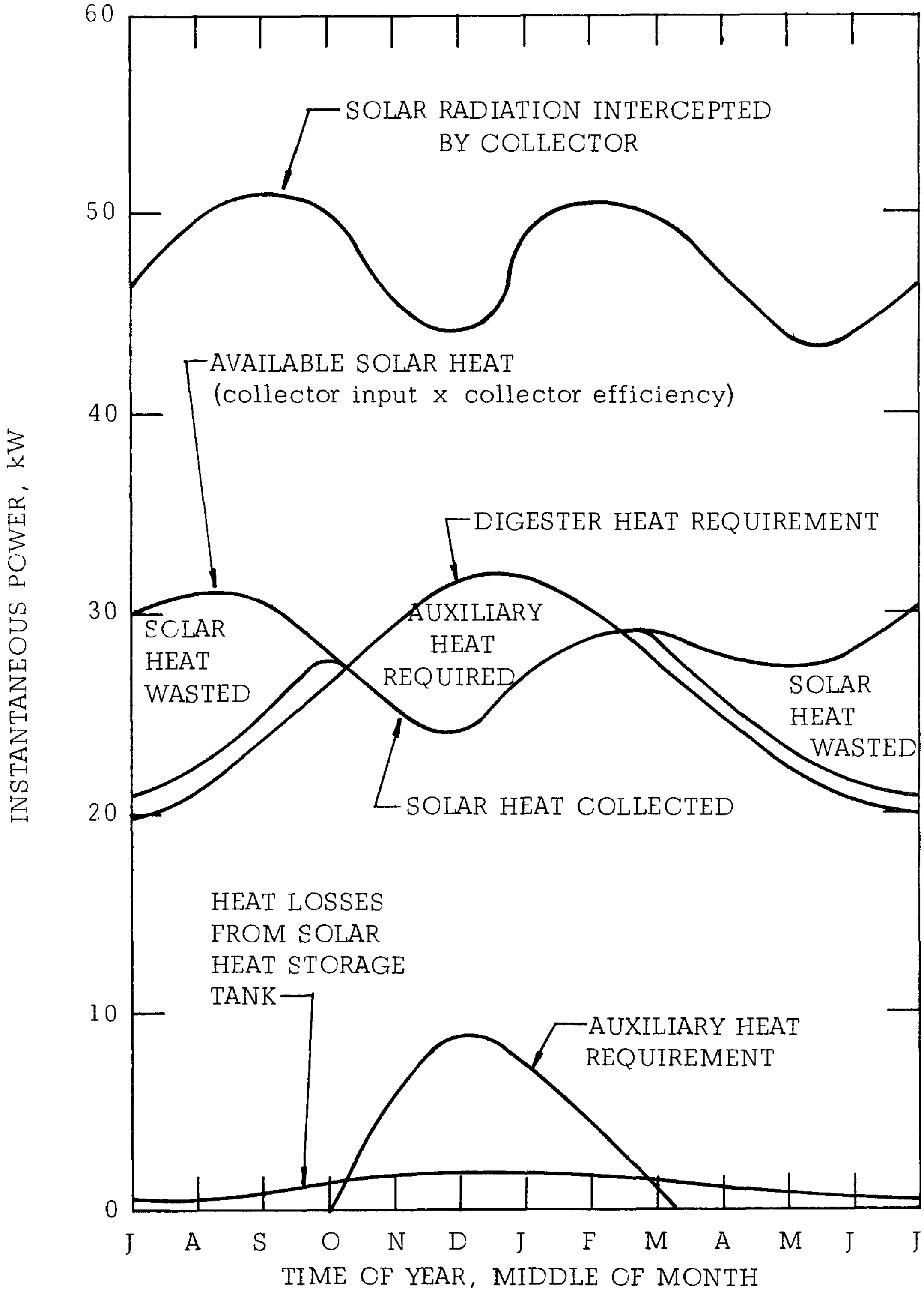


Figure 1. Operation of 90% solar heating system over annual cycle

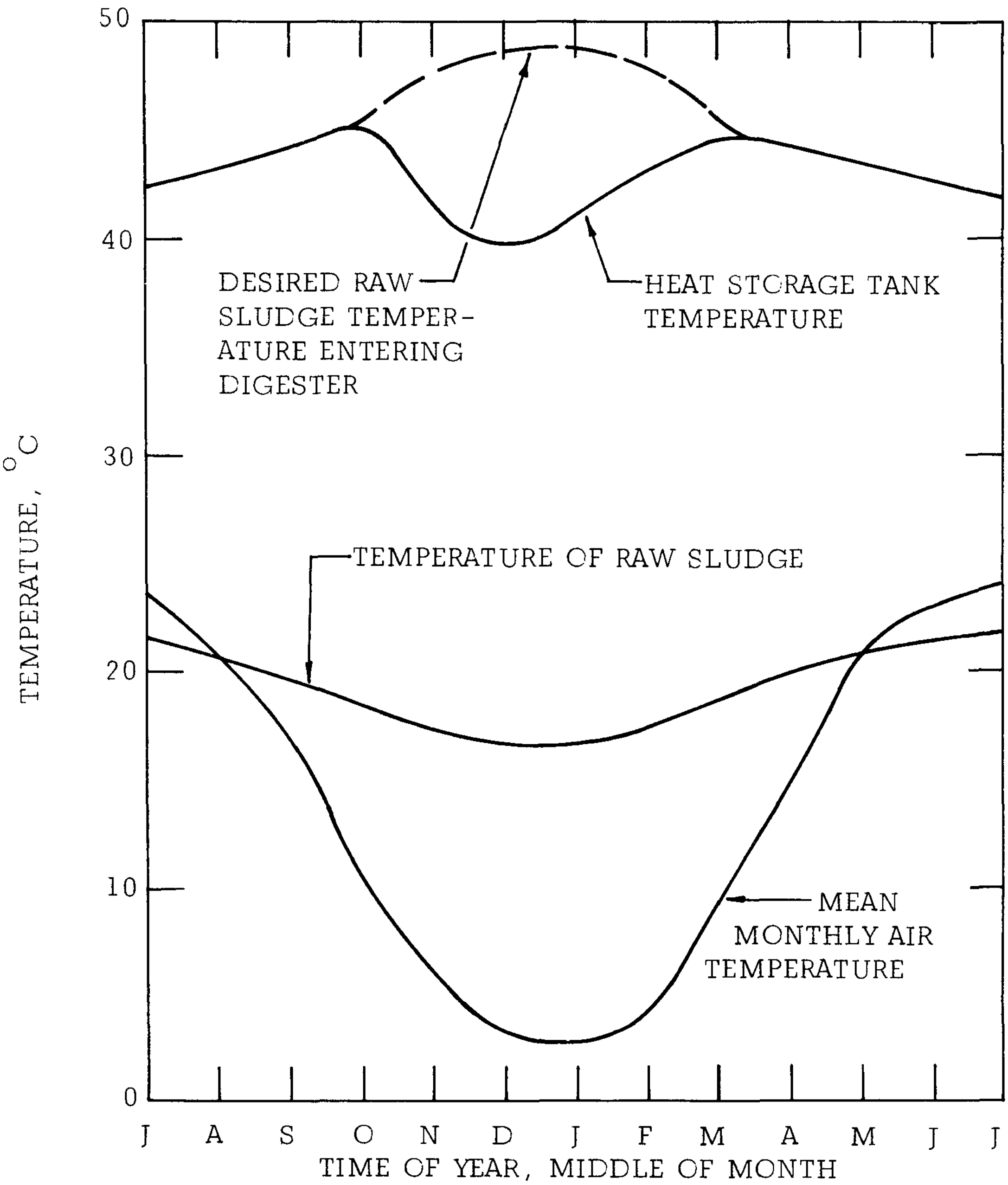


Figure 2. Temperature curves for the system

Starting at the top graph in Figure 1, the solar radiation intercepted by the collector is given. Multiplying this by the collector efficiency for each time of year (Figure 3) gives the available solar power. From about the middle of March through the middle of October, not all of the available energy is transferred to the heat storage tank because it is not needed. From October through March, however, all of the available solar energy from the collector is transferred to the heat storage tank. The total digester heat requirement varies approximately sinusoidally over the annual cycle, reaching the maximum of 31.6 kW in the middle of the winter.

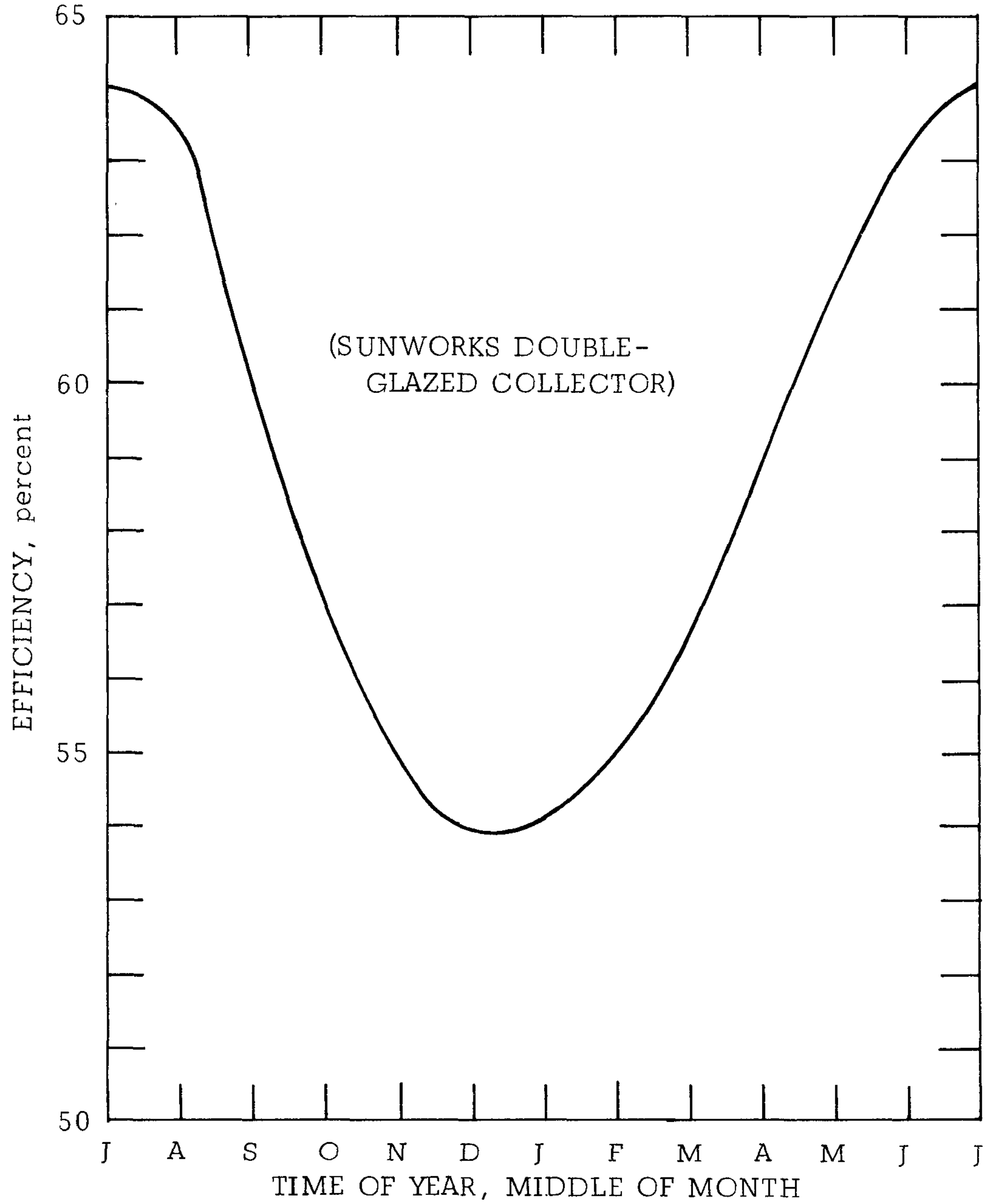


Figure 3. Efficiency of solar heat collector versus time of year

Auxiliary heat is required from October through March when the digester heat requirement exceeds the available solar energy. Maximum rate of auxiliary heat occurs in December, at 8.6 kW. Heat losses from solar heat storage tank are faily constant throughout the year, but they are slightly higher in winter than in summer. The rate of heat input to the heat storage tank any given time is approximately equal to the sum of the heat losses from the tank plus heat transferred to the sludge. When the temperature of the tank is changing, these rates differ slightly, but by not more than about 1 kW at most.

The average monthly ambient air temperature and raw sludge temperature vary approximately sinusoidally throughout the year, as shown in Figure 2.

The temperature to which the sludge would have to be preheated to satisfy the total digester heat requirement reaches a maximum in the winter. The heat storage temperature, however, cannot be maintained at that temperature during the winter, and dips severely from October through March, reaching a minimum of 39.5^{o}C in December. It is assumed that the sludge exiting from the heat transfer tank will be at the average temperature of the water in the tank. It is planned that sludge will be pumped steadily during twenty minutes out of each hour. The volume of sludge pumped during each pumping cycle is equal in amount to that required for one complete replacement of sludge in the heat exchanger.

The digester temperature is shown as a constant 35^{o}C. Again, these graphs represent average and typical conditions. In actual practice, the digester temperature would vary between 32 and 38^{o}C, still well within the upper mesophilic digestion range.

The areas under the power curves of Figure 1 represent total annual energy consumed or transferred. The area under the auxiliary heat curve, for example, represents the total annual auxiliary heat used, which is about tem percent of the area under the "digest heat requirement" curve. Similarly, the area between the "available solar power" curve and the "solar heat storage input" curve represents solar heat wasted.

SECTION 3

SYSTEM DESIGN

PRELIMINARY DESIGN CONSIDERATIONS

Approach

It is impossible to separate the technical design from economic consideration. The process of technical design consists of optimizing the system design; that is, finding the most economical hardware to do the job. The overall approach is to express the total present worth of the cost of the solar heating system as a function of percent solar heat (total annual heat supplied to the digester from the solar heating system divided by total annual heat required, expressed as a percentage), and to compare it with the present worth of digester gas saved, also as a function of percent solar heat. It is only after examining the cost data in this way that the most economical size solar heating system can be established.

Characteristics of the Raw Sludge

The total wastewater flow rate into the plant will average 0.197 m^3/s (4.5 mgd). Because half of the raw sludge will enter each of the two digesters, and only one digester will be heated by solar energy, all calculations will be based on a wastewater flow rate of half of the total, or 0.0985 m^3/s (2.25 mgd).

Average suspended solids content of the wastewater will be 0.160 kg/m^3 (160 mg/l). Additional suspended solids in the form of ferric chloride ($Fe\ Cl_3$) will be added at the rate of 0.040 kg/m^3. The total suspended solids concentration of the wastewater entering the clarifiers will therefore be 0.200 kg/m^3. About 90 percent of the suspended solids are expected to be removed in the clarifiers, giving, per digester, a mass flow rate of solid material in the sludge of:

$$(0.200\ kg/m^3) \times (0.0985\ m^3/s) \times 0.90 = 0.0177\ kg/s.$$

The raw sludge will be about 7.5 percent solids by mass. Therefore, the mass flow rate of the sludge is:

$$(0.0177\ kg/s)\ /\ 0.075 = 0.236\ kg/s.$$

Using the equations on pages 582 - 583 of Metcalf and Eddy(1), the specific gravity of sludge having 7.5 percent solids, with one third of the solid mass fixed and the other two-thirds volatile, is 1.02. This corresponds to a density of 1,020 kg/m^3. The average volumetric flow rate of the sludge to each digester is:

$$(0.236 \text{ kg/s}) \times (\text{m}^3/1{,}020 \text{ kg}) = 2.31 \times 10^{-4} \text{ m}^3/\text{s} \ (3.66 \text{ gpm}).$$

The temperature of the raw sludge at the Annapolis plant varies from a low of about 16°C in January to a high of 22°C in July, the average annual temperature being 19°C.

The five-day biochemical oxygen demand (BOD$_5$-20°C) of the influent wastewater will be about 0.190 kg/m^3. On a mass flow basis, this is:

$$(0.190 \text{ kg/m}^3) \times (0.0985 \text{ m}^3/\text{s}) = 0.0187 \text{ kg/s}.$$

About 60 percent of the BOD will be removed by the clarifiers and become part of the sludge:

$$0.0187 \text{ kg/s} \times 0.60 = 0.0112 \text{ kg/s}.$$

Or, on a volumetric basis, the BOD content of raw sludge is:

$$(0.112 \text{ kg BOD/s}) \times (\text{s}/2.31 \times 10^{-4} \text{ m}^3) = 48.5 \text{ kg BOD/m}^3.$$

Characteristics of the raw sludge are summarized in Table 1.

Digester Heat Requirements

The heat requirement of the anaerobic digester can be considered to consist of two parts: 1) heat necessary to raise the temperature of the incoming sludge to 35°C; and, 2) heat necessary to offset heat lost from the digester to the surrounding ground.

The average annual temperature of the raw sludge is 19°C, and the flow into one digester is 2.31 x 10^{-4} m^3/s. The heat capacity of sludge will be assumed to be the same as that of water, 4.19 x 10^6 J/m^3-C. The average annual power required is therefore:

$$(2.31 \times 10^{-4} \text{ m}^3/\text{s}) \times (4.19 \times 10^6 \text{J/m}^3\ {}^\circ\text{C}) \times (35 - 19)^\circ\text{C}$$
$$= 15{,}500 \text{ J/s} = 15{,}500 \text{ W, or } 15.5 \text{ kW}.$$

The power required in July when the incoming sludge temperature is 22°C is:

$$(2.31 \times 10^{-4} \text{ m}^3/\text{s}) \times (4.19 \times 10^6 \text{ J/m}^3\ {}^\circ\text{C}) \times (35 - 22)^\circ\text{C} = 12.6 \text{ kW}.$$

The power required in January when the incoming sludge temperature is 16°C is:

$$(2.31 \times 10^{-4} \text{ m}^3/\text{s}) \times (4.19 \times 10^6 \text{J/m}^3\ {}^\circ\text{C}) \times (35 - 16)^\circ\text{C} = 18.4 \text{ kW}.$$

The annual variation is sinusoidal.

Table 1. CHARACTERISTICS OF RAW SLUDGE

Flow rate, m^3/s (each digester)		2.31×10^{-4}
Flow rate, kg/s (each digester)		0.236
Density, kg/m^3		1,020
Percent solids by mass		7.5
Percent water by mass		92.5
Volatile solids, % by mass of total solids		67
Fixed solids, % by mass of total solids		33
Temperature, $^{\circ}C$:	Max. (July)	22
	Min. (January)	16
	Ave. annual	19
BOD content (BOD_5 - $20^{\circ}C$), kg/m^3		48.5
BOD (BOD_5 - $20^{\circ}C$) mass flow rate, kg/s		0.0112
Density of fixed solids, kg/m^3		2,500
Density of volatile solids, kg/m^3		1,000

The digester is of approximately cylindrical shape, 12.2 m (40 feet) in diameter and 6.1 m (20 feet) high, and is surrounded on all sides by earth. The wall areas are as follows:

$$\text{Roof area} = \pi(6.1\ m)^2 = 117\ m^2$$
$$\text{Side area} = \pi(12.2\ m)\ (6.1\ m) = 234\ m^2$$
$$\text{Floor area} = \pi(6.1\ m)^2 = 117\ m^2$$

Heat transfer coefficients (overall) are approximately as follows:

Roof--- $U = 1.9\ W/m^2C$
Concrete walls below grade,
dry earth----------------------------------- $U = 0.68\ W/m^2C$
Floor, concrete in contact
with moist earth---------------------------- $U = 0.77\ W/m^2C$.

Temperatures in the soil outside the roof, side, and floor of the digester are assumed to be as in Table 2.

Table 2. TEMPERATURE IN SOIL OUTSIDE DIGESTER
(degrees Celsius)

	Roof	Sides	Floor
January	6	7	10
July	20	19	16
Ave. annual	13	13	13

Heat transfer rates out of the digester are calculated using the following equation:

$$q = UA\ (\Delta T)$$

where q = heat transfer rate, W
U = overall heat transfer coefficient, W/m^2C
ΔT = temperature differential, ^oC

January:
roof $q = (1.9)\ (117)\ (35 - 6) = 6.4$ kW
sides $q = (0.68)\ (234)\ (35 - 7) = 4.5$
floor $q = (0.77)\ (117)\ (35 - 10) = 2.3$
Total 13.2 kW

July:
roof $q = (1.9)\ (117)\ (35 - 20) = 3.3$ kW
sides $q = (0.68)\ (234)\ (35 - 19) = 2.6$
floor $q = (0.77)\ (117)\ (35 - 16) = 1.7$
Total 7.6 kW

Average annual:
roof $q = (1.9)\ (117)\ (35 - 13) = 4.9$ kW
sides $q = (0.68)\ (234)\ (35 - 13) = 3.5$
floor $q = (0.77)\ (117)\ (35 - 13) = 2.0$
Total 10.4 kW

The annual variation is sinusoidal.

The total power needed to supply the digester heat requirement is the sum of that required to preheat the sludge and that required to offset losses from the digester. It varies sinusoidally over a year's time from a low of 20.2 kW in July to a high of 31.6 kW in January. The average annual power required is 25.9 kW. The total energy required to heat the digester for one year is:

$$(25.9 \times 10^3\ J/s) \times (3.15 \times 10^7\ s) = 816 \times 10^9\ J = 816\ GJ.$$

The variation of total digester heat requirements over a year's time can be expressed by the following equation:

$$p = 25.9 + 5.7 \sin (t - 210)$$

where p = average monthly power required, kW
t = time of year, degrees (March = 0, April = 30, etc.).

The average required to supply the total heat requirements of one digester for each month is shown in Table 3.

Table 3. AVERAGE MONTHLY POWER REQUIRED TO SUPPLY ALL HEAT FOR ONE DIGESTER

Month	Power, kW	Month	Power, kW
January	31.6	July	20.2
February	30.8	August	21.0
March	28.8	September	23.1
April	25.9	October	25.9
May	23.1	November	28.8
June	21.0	December	30.8
		Total =	311

Total annual heat = 311 kW x 2.63 x 10^6s = 8.18 x 10^{11} J

Digester Gas Production

Digester gas is composed of about 65 percent methane (CH_4) and 35 percent carbon dioxide (CO_2) by volume, or 40 percent methane and 60 percent carbon dioxide by mass. Any gas under standard conditions occupies a volume of 22.4 m^3 per kg-mole. Hence, the density of CH_4 at standard conditions (0^oC and 1.01 x 10^5 Pa) is 16/22.4 = 0.714 kg/m^3, and that of CO_2 is 44/22.4 = 1.96 kg/m^3. Since digester gas is 65 percent CH_4 and 35 percent CO_2 by volume, the density of digester gas is:

$$(0.714)\ (0.65) + (1.96)\ (0.35) = 1.15\ kg/m^3$$

Air at standard conditons has a density of 1.29 kg/m^3, so that the specific gravity of digester gas is 1.15/1.29, or 0.89 relative to air.

Because the density of gases changes appreciably with temperature and pressure, rates of gas production will be calculated on a mass rather than a volumetric basis.

Digester gas production can be estimated from volatile solids loading. Metcalf and Eddy[1], p. 606, give a typical value of 0.62 m^3 of digester gas produced per kg of volatile solids added, at standard conditions. (This is also equal to 0.93 m^3 of digester gas per kg of volatile solids destroyed, assuming that on the average 2/3 of the volatile solids added are destroyed.) The mass of digester gas per unit mass of volatile solids added is:

$$0.62\ m^3 \times (1.15\ kg/m^3) = 0.713\ \text{kg gas per kg volatile solids added.}$$

The mass flow rate of the raw sludge is 0.236 kg/s. The raw sludge is about 7.5 percent solids, and about 2/3 of the mass of the solids is volatile. Therefore, the volatile solids flow rate is:

$$0.0236 \times 0.075 \times 0.67 = 0.0119\ kg/s.$$

The digester gas production rate is, therefore,

$$0.0119\ kg/s \times 0.713\ \text{kg gas/kg volatile solids} = 0.00849\ kg/s.$$

This is 3.60 percent of the mass flow rate of the raw sludge.

Digester gas production can also be estimated on the basis of population. Metcalf and Eddy, p. 606, state that in secondary treatment plants digester gas production can be estimated to be about $3.28 \times 10^{-7}\ m^3/s$ (1.0 ft^3/day) per person at standard conditions. For the City of Annapolis, therefore, with a population of about 35,000 people, the estimated digester gas production rate is 0.0115 m^3/s. Since one half of the sludge enters each digester, the production rate of each digester is 0.0115/2, or $5.75 \times 10^{-3} m^3/s$, which is equal to 0.00661 kg/s. This is slightly lower than the rate based on volatile solids reduction. The chemical treatment process at Annapolis probably warrants the higher figure of 8.49×10^{-3} kg/s.

Since 40 percent of the mass of the digester gas is methane, the production rate of CH_4 is (0.40) x (0.00849 kg/s) = 0.00340 kg/s. The low heating value of methane is 49.9×10^6 J/kg. Therefore, energy is being produced in the form of burnable methane gas at the rate of (0.00340 kg/s) x (49.9×10^6 J/kg) = 170 kW. Assuming a combustion efficiency of 66 percent, 112 kW would be available for digester heating. This is 112/25.9, or 4.32 times the average power required to heat one digester, and 112/31.6, or 3.54 times the maximum power requirement of one digester in January. Twenty-three percent (1/4.32) of the total annual gas produced would be required to supply 100 percent of the digester heating requirements. Thus 23 percent of the gas produced would be conserved by using a 100 percent solar heating system.

Mass Flow Rates Into And Out Of Digester

Bacteria in the anaerobic digester convert a large percentage of the volatile (organic) solids to liquid (supernatant) and digester gas. It is desirable to know how much digested sludge, supernatant, and gas are produced per inlet flow rate of raw sludge. The following assumptions will be made in addition to those concerning the characteristics of the raw sludge previously mentioned:

1. Sixty-seven percent of the volatile solids are destroyed during anaerobic digestion;

2. The absolute mass flow rate of the fixed solids out of the digester (in the digested sludge) is equal to that into the digester (in the raw sludge);

3. The digested sludge contains 12 percent solids by mass; and

4. The density of the supernatant is the same as that of water, 1,000 kg/m^3.

The following calculations will be based on a batch of raw sludge having a total mass of 1,000 kg. Of this, 75 kg are solid material and 925 kg are water. Of the solids, (0.67) (75), or 50.2 kg are volatile (organic) and 24.8 kg are fixed.

After digestion, the remaining mass of volatile solids is (0.33) (50.2), or 16.6 kg. The percent that the volatile solids are of the total solids is therefore:

$$16.6 \text{ kg} / (16.6 \text{ kg} + 24.8 \text{ kg}) = 40.1\%.$$

The density of the fixed solids remains 2,5000 kg/m^3, which corresponds to a specific volume of 4×10^{-4} m^3/kg. The density of the volatile solids is 1,000 kg/m^3, or a specific volume of 10^{-3} m^3/kg. The average specific volume of all the solids in the digested sludge is:

$$(0.401)\ (10^{-3}) + (0.599)\ (4 \times 10^{-4}) = 6.41 \times 10^{-4} \text{ m}^3/\text{kg}.$$

The average density of all the solids is $1/(6.41 \times 10^{-4})$, or 1,560 kg/m^3. Since the digested sludge is 12 percent solids by mass, its specific volume is:

$$(0.12)\ (6.41 \times 10^{-4}) + (0.88)\ (10^{-3}) = 9.57 \times 10^{-4} \text{ m}^3/\text{kg}.$$

The average density of the digested sludge is $1/(9.57 \times 10^{-4})$, or 1,040 kg/m^3. The mass of the digested sludge is equal to the total mass of the solids divided by the percent solids:

$$(16.6 \text{ kg} + 24.8 \text{ kg}) / 0.12 = 345 \text{ kg}.$$

This is 34.5 percent of the mass of the raw sludge.

The mass flow rate of digester gas was calculated previously to be 8.49×10^{-3} kg/s, or 3.60 percent of the mass flow rate of the raw sludge. The balance is assumed to be supernatant. These results are summarized in Table 4.

Table 4. AVERAGE MASS FLOW RATES
INTO AND OUT OF DIGESTER

	Mass flow rate, kg/s	Percent by mass of total
Raw sludge	0.236	100.0
Digested sludge	0.814	34.5
Supernatant	0.146	61.9
Digester gas	0.00849	3.6

Alternate Energy Considerations

The scope of this study was limited to the technical and economic feasibility of substituting solar heat for digester gas to heat anaerobic digesters. Other energy considerations have been brought up in the course of this study, however, which should be mentioned at this time.

Considering the cost of all energy, including solar, consideration should be given to insulating the digester as well as practical, thus reducing the heat losses to the surrounding ground, which for the Annapolis Plant amount to 40 percent of the total digester heat requirement.

Depending on what is to be done with the supernatant and digested sludge, much heat could be extracted from them. The mass of the supernatant is 62 percent of the mass of the incoming raw sludge. It is theoretically possible, therefore, to supply 62 percent of the digester heating requirements simply by efficient heat exchange to reduce the temperature of the supernatant to that of the raw sludge. Similarly, the digested sludge is 35 percent by mass of the raw sludge. If this digested sludge were to be trucked away, a large percentage of the digester heat requirement could be fulfilled by transferring heat from the digested sludge to the cold, raw sludge.

Due to the small mass of digester gas produced and low heat transfer coefficients for gas, less than one percent of the digester heat requirements could be fulfilled by cooling the gas from 35 to 20^{o}C. Extracting sensible heat from the digester gas is therefore impractical.

General Data Sheet

Some of the more important values and constants used throughout this study are summarized in Table 5.

Table 5. GENERAL DATA SHEET

Heat capacity of water		4.19×10^6 J/m^3C
		4.19×10^3 J/kg C
Density of water		1,000 kg/m^3
Density of air	(20^oC, 10^5 Pa)	1.29 kg/m^3
Heat capacity of digester gas		1,510 J/kg C
Density of methane gas	(0^oC, 10^5 Pa)	0.714 kg/m^3
	(20^oC, 10^5 Pa)	0.668 kg/m^3
Density of carbon dioxide gas	(0^oC, 10^5 Pa)	1.96 kg/m^3
	(20^oC, 10^5 Pa)	1.83 kg/m^3
Heat capacity of methane		2,480 J/kg C
Heat capacity of carbon dioxide		859 J/kg C
Low heat value of methane	(0^oC, 10^5 Pa)	33.6×10^6 J/m^3
		49.9×10^6 J/kg
Composition of digester gas	by volume:	65% methane
		35% carbon dioxide
	by mass:	40% methane
		60% carbon dioxide
Average raw sludge flow rate	(per digester)	0.236 kg/s
		2.31×10^{-4} m^3/s
Average wastewater flow rate, total		0.197 m^3/s
Influent suspended solids		0.160 kg/m^3
Ferric Chloride feeding rate		0.40 kg/m^3
Suspended solids reduction in clarifiers		90%
BOD reduction in clarifiers		60%
Raw sludge, percent solids		7.5%
Volume of one digester		679 m^3

DESIGN CONCEPTUALIZATION

Additon of Solar Heat to Digester

Many methods were considered for adding the solar heat to the digester. Sludge, either raw or digesting, could be pumped through a solar collector to be heated directly. Solar-heated water could be circulated in coils within the digester. The cold, raw sludge could be preheated by passing it through a solar heat exchanger before it enters the digester. Sludge could be recycled from the digester to a solar heat exchanger. Or, any combination of the above methods could be used. The criteria used to evaluate the various alternatives were as follows:

1. Efficiency of solar heat collection - solar heat must be collected quickly when available and stored for future use. The temperature of the solar heat storage unit must be as low as possible for maximum collection efficiency;

2. Cost - expensive valves or controls should be avoided;

3. Temperature shock - the bacteria should not be exposed to extremes of temperature or rapid changes between temperature ranges;

4. Operation of system during periods of little or no sunlight;

5. Ease of operation and ability to control conditions positively; and

6. Adequacy to adapt to changing sludge rates, temperatures, and weather conditions.

Preheating the raw sludge was chosen based on the above criteria. Heating the sludge directly by passing it through a solar collector would overheat it or require costly controls. Circulating solar-heated water through coils in the digester would be inefficient due to the desired low temperature of that water. Also, this would tend to promote uneven temperatures in the digester. Recycling hot, digesting sludge through a solar heat exchanger would be less efficient than passing cold, raw sludge through the same heat exchanger. Recycling of digesting sludge through the solar heat exchanger in addition to preheating the raw sludge would not be advantageous because this would only lower the temperature of the solar heat storage unit, reducing the amount of heat available for preheating the raw sludge.

There are additional advantages of sludge preheating as the sole means of adding solar heat to the digester. This method would work equally well for stratified or for high-rate, mixed digesters. The solar heating system is kept completely separate from the auxiliary heat system, desirable for two reasons: 1) adding solar heat to existing digesters is simplified; and 2) conventional heating systems can be used as auxiliary heat, operating independently of the solar heating system.

If sludge pre-heating is used as the sole method of digester heating, the sludge must be heated to a temperature greater than the design temperature of the digester, 35^{o} C. The additional temperature rise required above $35^{o}C$ must compensate for the heat lost from the digester to the surrounding ground. The sludge flow rate per digester is 0.236 kg/s and the heat capacity of the sludge is assumed to be equal to that of water, 4.19×10^{3} J/kg - ^{o}C. In January, when the heat loss from the digester is 13.3 kW, the additional temperature rise is:

$$(13.3 \times 10^{3}\ J/s) \times (kg - {}^{o}C/4.19 \times 10^{3} J) \times (s/0.236\ kg) = 13.5^{o}C.$$

Therefore, the sludge must be preheated in January to 35 + 13.5, or $48.5^{o}C$. In July, when the heat loss from the digester is 7.6 kW, the additonal temperature rise is:

$$(7.6 \times 10^{3}\ J/s) \times (kg - {}^{o}C/4.19 \times 10^{3} J) \times (s/0.236\ kg) = 7.7^{o}C.$$

The sludge must be preheated in July to 35 + 7.7, or 42.7°C (see Figure 2). When the solar heat storage tank cannot be maintained at or above these temperatures, as will be the case during the critical winter period, auxiliary heat must be added to the digester.

Type of Solar Collectors

The temperatures required for heating the sludge are in the range of those produced efficiently by standard flat-plate solar heat collectors. Therefore, this is the basic type of collector considered for this application.

Heat Storage

Plain water has proven to be the most economical and most easily used medium for both solar heat collection and storage. The heat storage tank was placed at a lower elevation than the collectors to allow draining of the collectors when not in use. This has the dual advantages of preventing freezing (and not requiring antifreeze) and of keeping the heat capacity of the collectors low. Rocks, which are sometimes used for transfer of solar heat to air, were ruled out. Air as a heat-collection fluid is less efficient than water.

System Operation

Simplicity was attained by using water as both heat collection and storage medium. To make the system very simple to operate, a single centrifugal pump was used to circulate water from heat storage to collectors. The differential thermostat pump control for selecting the operating times of the pump is widely used in solar building heating systems. This system was used with the addition of another thermostat in series with the differential thermostat to limit the amount of solar heat transferred to the storage tank in the summer.

The other option that was considered was to collect solar heat year round, allowing the temperature of the stored water to increase very high in the summer and to drop down to a reasonable level in the winter. This would perhaps increase the percent solar heat for a given size solar heating system, storing surplus heat in the summer for use in the winter. This was considered impractical for the following reasons. If the temperature of the solar heat storage were to increase higher than necessary in the summer, the heat transfer rate between solar heat storage and sludge would have to be decreased during that period. In fact, means would have to be provided to very the heat transfer rate depending on the temperature of the stored water and the requirements of the digester. This would involve either valving, changing flow rates, or providing auxiliary heat exchangers outside of the solar heat storage tank. Sophisticated, expensive control systems would be required. Storing heat at such high temperatures would increase heat losses from the storage tank, decreasing the efficiency of the system. Also, the higher the temperature of the heat storage, the lower the collector efficiency, so that less heat would be collected than may be expected based on normal collector efficiencies. The potential difference between year-round collection and storage of surplus heat and "wasting" excess heat in the summer is only about 11 percent. (This is partly due to the orientation of the collectors and use of reflector, both of which maximize solar heat collection when needed most: in the winter.) Collector inefficiencies and greater heat losses reduce the amount considerably.

Sensing the temperature of the digester and limiting the collection of solar heat when not required (when the digester temperature increases to a given value) provides the most practical and economical summertime control system.

Collector Orientation

Flat-plate, stationary solar heat collectors are normally positioned facing south at an angle from the horizontal. The physical arrangement of the Annapolis plant allows placing the collectors parallel to a fence, facing them seven degrees west of due south. This is ideal because facing the collectors west of south allows maximum heat collection in early afternoon when ambient temperatures are usually highest, promoting the highest possible collector efficiencies at that time.

If the collector angle could be changed during the year to allow maximum heat collection at all times, the collector angles (for Annapolis, Maryland, 39°N Latitude) would be as shown in Figure 4. Figure 4 also gives monthly normals of temperature at Annapolis, Maryland, averaged from 1941 through 1970. The coldest day of the year occurs in mid-January. Heat losses from the digester would be greatest then, and raw sludge temperatures would be lowest. Also, collector efficiency would be lowest at that time. The collector angle that allows maximum heat collection when needed most is 60 degress from the horizontal. As Figure 1 shows, even at this angle excess solar heat is available in June. Therefore, the collector angle was chosen as 60 degress from the horizontal.

In January with the collector 60 degrees from the horizontal and the sun's rays normal to the collector at 30 degrees from the horizontal, the amount of radiation incident on the collector can be increased by using a horizontal reflector in front of the collector. The cost of the reflector per unit area is a small fraction of the cost of the collector. Wind load considerations prohibited additonal reflector above the collectors.

Method of Adding Auxillary Heat

Several methods were considered for adding auxiliary heat to the digester. A heat pump could be used to "pump" heat from the relatively cool solar heat storage tank to the digester. This method was discarded because cooling the solar heat storage tank would reduce the heat available for preheating sludge. In addition, it is wasteful to use electricity for partial auxiliary heat when digester gas is available. Either digester gas or other fuels could be used to add heat either directly to the digester or to the solar heat storage tank. It is desirable to keep the temperature of the solar heat storage tank

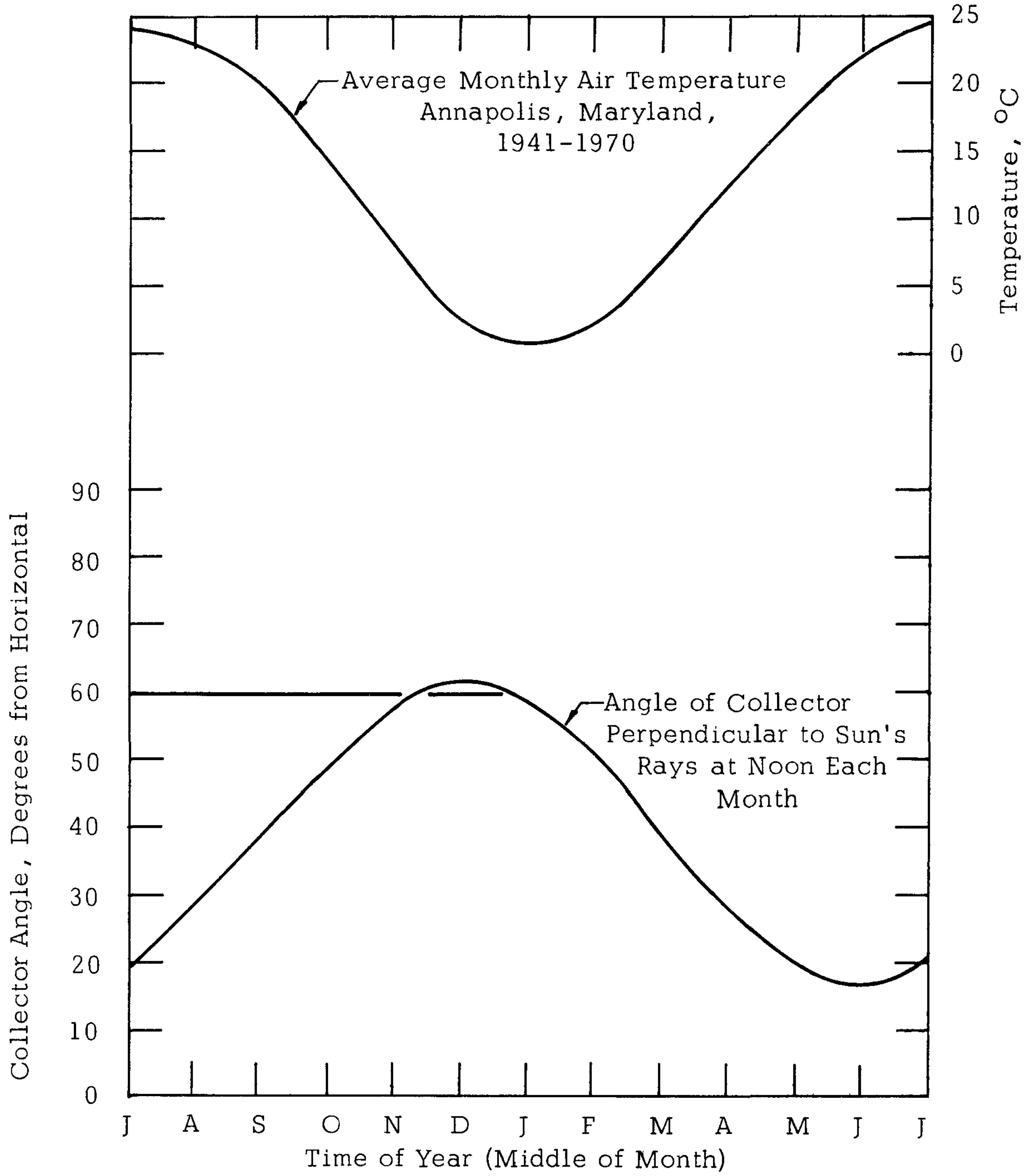


Figure 4. Determination of Optimum Solar Collector Angle for Winter Heating, Annapolis, Maryland

as low as possible, so the auxiliary heat should be added directly to the digester. This is most easily done by using the existing digester heating system at a lower capacity to provide the auxiliary heat. Either preheating the raw sludge after it is first preheated by solar energy, recycling digesting sludge through external heat exchangers, or circulating hot water in heating coils within the digester would be satisfactory methods of adding the auxiliary heat. At Annapolis, the heating coils are already in place, so the method chosen was to heat water by burning digester gas and circulate the hot water in the coils within the digester.

HEAT STORAGE AND TRANSFER

Size of Solar Heat Storage Tank

The size of the solar heat storage tank is not critical. Unlike the effective solar collector area, the size of heat storage has virtually no direct relationship to the annual percent solar heat. It does, however, affect the system stability, that is to say temperature variations in the digester and heat storage tank.

The parameter R (the ratio of volume of water stored in m^3 to area of solar collectors in m^2) was varied from 0.1 to 1.5 as a variable in one of the earlier computer programs of this study. It was found that the annual percent solar heat was independent of R for all R's over 0.2. Under 0.2 the system became unstable with fluctuating temperatures in the heat storage tank. These results were based on heat inputs and withdrawals from the tank every five days. For a collector area of 400 m^2 (proposed at that time, based on less efficient collectors than those of the final design) a storage volume of 80 m^3 appeared to be optimum. The standard size steel tank of 75 m^3 (20,000 gal) was chosen. The value of R was kept at 0.2 for all subsequent computer runs.

Heat Capacities Within the System

The heat capacity of the 75-m^3 heat storage tank is 11 percent of that of the 680-m^3 digester. The allowable temperature range of the digester is from 32 to 38°C, six degrees Celsius. With no heat input to the digester in the winter when the total heat requirement is at the rate of 3.16 kW, the digester temperature would drop 0.91°C each day, giving about 6½ days' heat storage within the digester itself. Using the same temperature drop in the heat storage tank, less than one additional day's storage would be available from that tank. When heat must be stored and not added to the digester during the winter, the 75-m^3 tank of water could store about 4½ days' worth of heat input from the sun, assuming a 30-degree Celsius temperature rise:

$$75\ m^3 \times (4.19 \times 10^6\ J/m^3 \cdot C) \times 30^oC \times (1/25\ kW) = 377\ ks\ (4.36\ days).$$

Heat Transfer Unit

Various methods of construction were considered for the combination solar heat storage and heat transfer unit. Of the more common materials (wood, concrete, steel, and plastic), steel was chosen because of cost, ease of construction, and availability. The tank could be buried in the ground or supported above ground. Number three digester at the Annapolis plant was convenient for the tank, located between the solar collector site and the digester to be heated by solar energy. Sprayed-on urethane insulation with a sprayed-on protective coating of hydrocide elastomeric roofing (HER) was the easiest and least expensive method of insulating the tank.

Unfortunately, the heat transfer rate between the solar-heated water and raw sludge is very low due to the laminar flow of the sludge. Therefore, the volume within the sludge pipe in the storage tank was made equal to the volume of sludge pumped during each pumping cycle, 20 minutes out of each hour. The volume pumped during each cycle is:

$$(2.31 \times 10^{-4} m^3/s) \times 3 \times 1200 \text{ s} = 0.832 \text{ m}^3.$$

The volume contained within five passes of 6-inch pipe is:

$$(3.14 \times (0.152)^2/4 \times 9.45 \text{ m} \times 5 = 0.857 \text{ m}^3.$$

The raw sludge will remain in the heat storage tank for more than 40 minutes, so that it should be at virtually the same temperature as the tank upon entering the digester.

CONTROL SYSTEM

Figure 5 is a schematic diagram of the control system. Several control systems were proposed, but the one shown proved to be the best and simplest, and the lowest in cost. There are two different operating periods during the year: winter and summer. At the beginning of the winter period, about mid-October, the auxiliary gas boiler is fired up, and it remains on until no more auxiliary heat is needed, about the end of March. Turning the boiler on once a year is the only manual operation. As much solar heat is collected as possible except when the digester temperature exceeds 37^oC. When the temperature of the digester is about 37^oC, the solar collector pump is prevented from operating.

When the digester temperature is below 37^oC, the collector pump is turned on when the temperature on the surface of the copper plate in one of the collectors rises to about 11 degrees Celsius above the temperature at the bottom of the heat storage tank. When the temperature of the collector is reduced to about 1.7 degrees Celsius above the storage temperature, the pump is turned off. These on/off temperature differentials are adjustable on the differential thermostat to provide optimum system operation, that is, minimum cycling of the pump and maximum solar heat collected.

If the digester temperature falls to 33^oC, the auxiliary heat circulating pump is turned on until the digester temperature rises above 33^oC.

Three cases of adverse conditions will be described to demonstrate the operation of the automatic control system.

1. During summer operation when the digester temperature is 37^oC and when the temperature of the heat storage tank is very high (50^oC), will the digester overheat? The collector pump will be off, so no heat will enter the heat storage tank. If the sludge were heated to the storage temperature of 50^oC, after one day, the digester

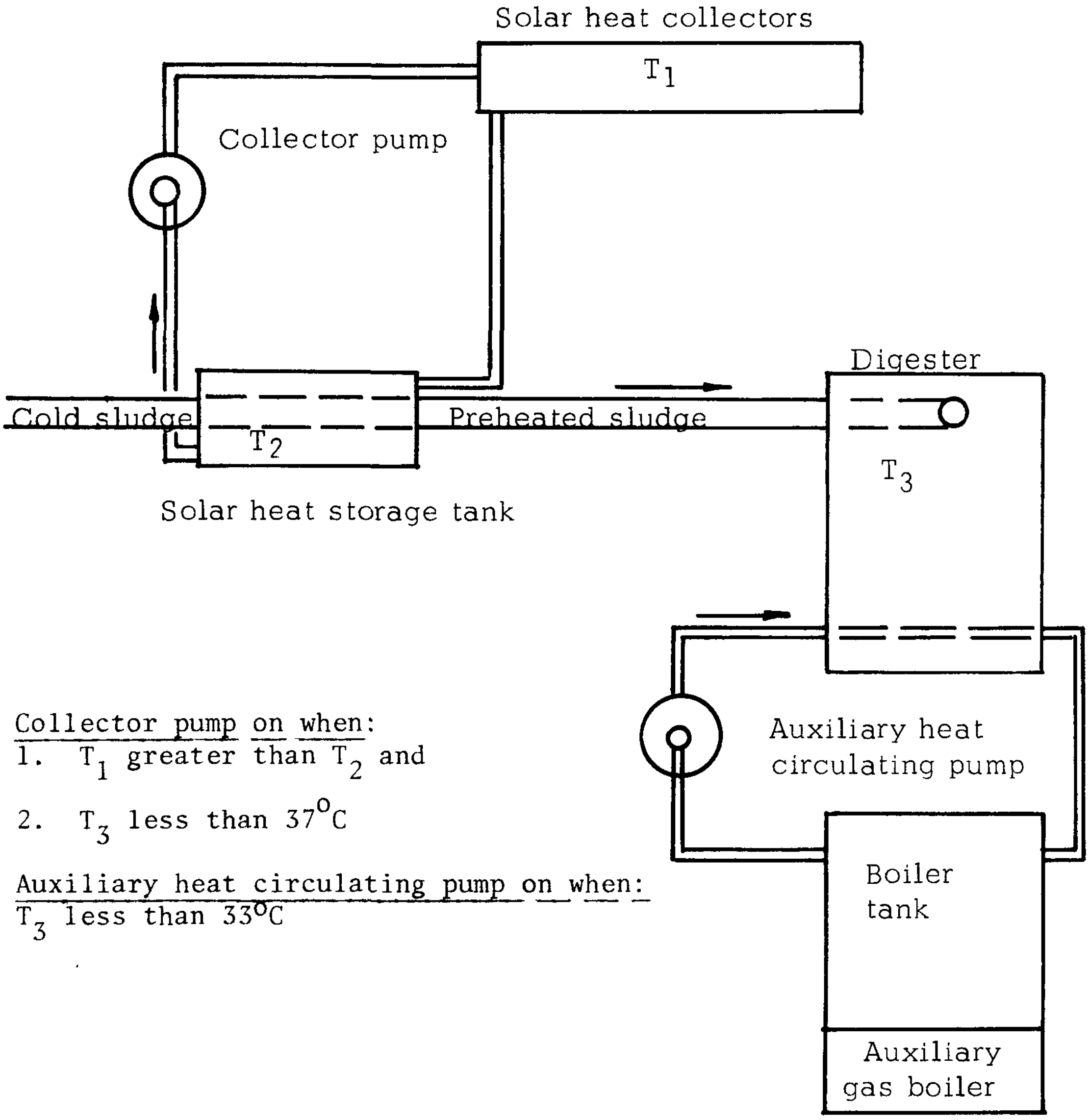


Figure 5. Schematic diagram of automatic control system

temperature would increase to 37.2°C. The storage temperature would drop to 43°C. After the second day, the digester temperature would still be at 37.2°C because heat losses would equal heat added via the raw sludge. The temperature of the storage would be 37°C. After the third day, the digester temperature would drop to 37.0°C. and the storage temperature would be 33°C. Therefore, the digester will not overheat in the summer with this control system.

2. During the critical winter period, what would happen if the solar input became zero for five days and then returned to "normal" for the next five days, starting with the digester at 33°C, heat storage at 35°C, and heat losses of 32 kW? This is a severe test because if the solar input were zero for five days it should be twice normal for the next five days. Results of the calculations show that the temperature of the heat storage would decrease from 35°C to 18.3°C on the fifth day and increase to 35.1°C by the tenth day. The digester temperature would remain at about 33°C throughout the 10-day period. Auxiliary heat input would increase to a maximum of about 29 kW on the fifth day, and then decrease to 15 kW by the tenth day. During winter conditions such as this, the auxiliary heating system would operate continually to maintain 33°C in the digester, the solar heat storage temperature would vary depending on the amount of solar radiation available, and the raw sludge would be preheated to the temperature of the heat storage at any given time.

3. During the summer, with the digester temperature at 37°C, the heat storage at 43°C, and if solar radiation stopped for five days, the digester temperature would drop to between 34 and 35°C after the fifth day, and no auxiliary heat would be needed.

The control system is adequate to maintain the digester temperature between 32 and 38°C during all days of all seasons of the year. As shown by Figure 6, the curve of digestion period versus digester temperature is very flat within this range. Note that this is the upper part of the mesophillic temperature range, which extends from about 10 to 38°C.

COMPUTER SIMULATION

Need for Computer Simulation

Two essential parameters, the efficiency of the solar heat collectors, and the amount of heat transferred from the solar heat storage tank to the raw sludge, depend on the temperature of the heat storage tank. Heat losses from the tank, less important but significant, also depend on the temperature of the heat storage tank. Heat inputs and outputs from the tank, however, determine its temperature at any given time. The only way to determine the system operation, therefore, is to actually simulate the system operation over the annual cycle using equations for the various heat transfers, heat losses, and efficiencies that contain heat storage temperature and time of year as independent variables.

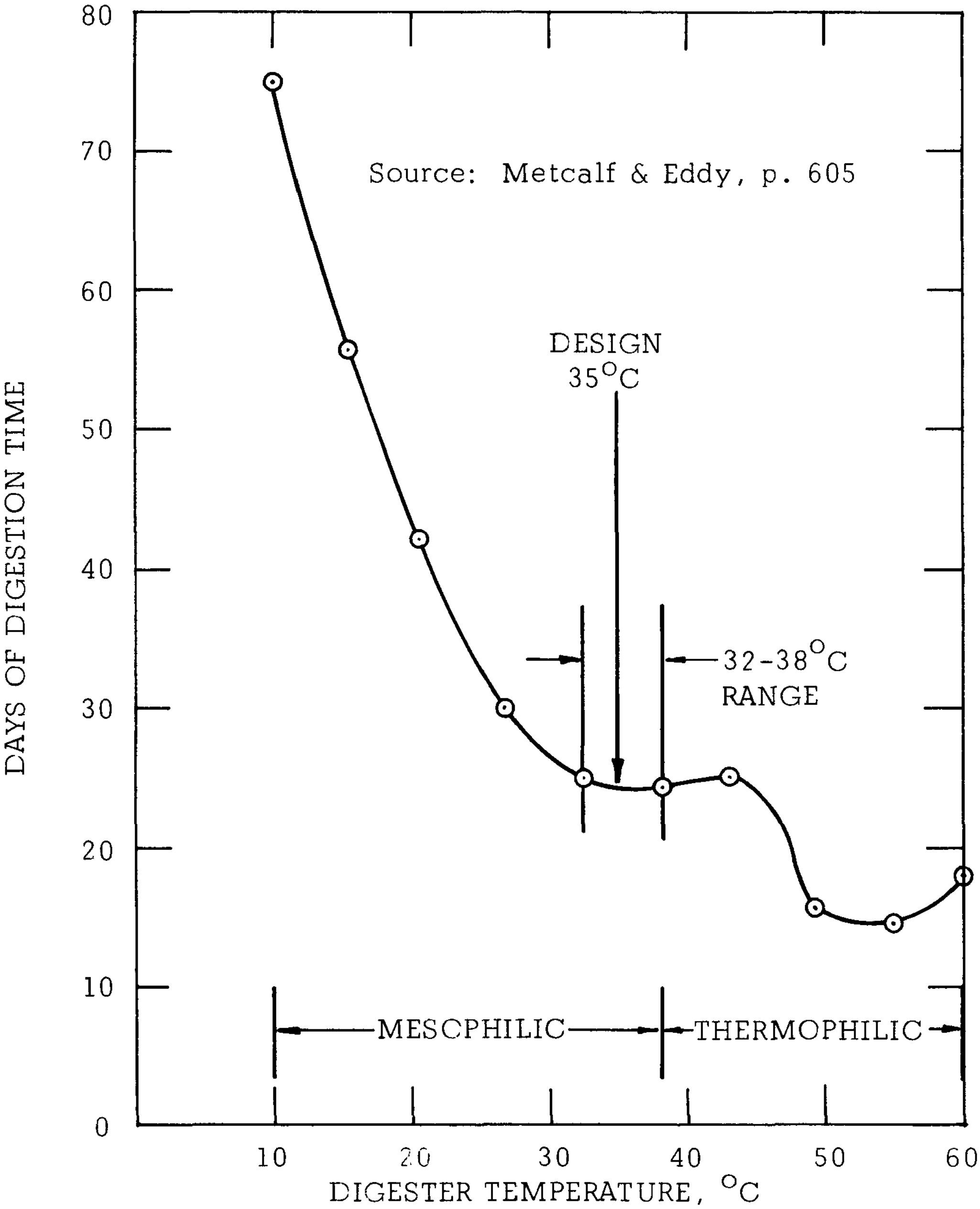


Figure 6. Digestion time versus digester temperature

Description of the Computer Program

The final form of the basic computer program used in this study, designated "SOL 4," is given in the appendix of this report. This program simulates operation of the digester heating system on a daily basis over the annual cycle, and prints an annual summary of all significant parameters. A modification of this program, designated "SOL 5," also found in the appendix, prints all heat transfer rates and temperatures every five days throughout the year, as well as the annual summary. Both programs were written in standard FORTRAN IV computer language and were run on the Honeywell 1648 Series time-sharing computers of National Computer Network of Chicago, Inc., via teletype from Annapolis, Maryland.

The programs were run separately for each solar collector design. The annual summary allowed plotting percent solar heat versus effective collector area. Multiplying by cost of collectors per unit effective collector area gave curves of collector cost versus percent solar heat. Thus the most cost-effective collectors for this particular heating application could be chosen.

For any given set of conditions, the computer program runs through all calculations for one year without printing any results. This assures that the system has been completely stablized for that particular set of conditions. Values are recorded during simulation of the second year.

The computer program is centered around heat inputs to and outputs from the solar heat storage tank. After each day of heat transfers, a new heat storage temperature is calculated for use in determining heat transfers for the next day.

Results are recorded with and without the horizontal reflector to evaluate the benefit of the reflector.

Equations for Use in Computer Program

Equations for most parameters were expressed as a function of time of year, heat storage temperature, collector area, or some combination of these variables. Parameters expressed as a function of time of year generally very sinusoidally over the annual cycle. The variable "TY," time of year in radians, starting with TY as zero at the spring equinox, was used in all such equations.

Several sources of solar radiation data were analyzed for completeness, accuracy, and capability of expression in the form required. The final equation used was derived from data given in the 1975 EI&I Associates guide, "Determining the Availability of Solar Energy within the Continguous United States."[2] The variable F is an empirical factor that, when plotted versus time of year, follows approximately a sine curve:

F = 0.725 + 0.175 *SIN (TY - 3.1416).

It is defined by the following equation:

F = CI/(SEXH*COS(DA))

where CI = Average terrestrial solar energy received on south-facing surface (60 degree angle from horizontal), W/m^2

SEXH = Average extraterrestrial solar energy received on a horizontal surface, W/m^2

DA = Deviation angle = absolute value of difference between angle of collector perpendicular to sun's rays at solar noon and collector 60 degrees from horizontal, radians.

The fact that F and SEXH vary approximately sinusoidally over the annual cycle allows CI to be expressed mathematically as a function of TY. Data for the calculations are given in Table 6.

Table 6. DATA FOR DERIVATION OF EMPIRICAL EQUATION FOR INCIDENT SOLAR RADIATION

Month	SEH W/m^2	SEXH W/m^2	RA (Dimensionless)	CI W/m^2	DA, degrees	F (Dimensionless)
Jul.	262	466	0.76	199	41.4	0.569
Aug.	227	418	1.00	227	32.8	0.645
Sept.	185	342	1.13	209	21.0	0.655
Oct.	142	258	1.44	205	9.2	0.804
Nov.	92	189	1.67	153	0.6	0.809
Dec.	75	157	1.93	145	2.5	0.924
Jan.	85	174	1.81	153	0.6	0.879
Feb.	118	235	1.57	185	9.2	0.797
Mar.	165	313	1.24	205	21.0	0.702
Apr.	203	396	0.96	195	32.8	0.586
May	236	454	0.82	194	41.4	0.569
June	270	478	0.76	205	44.5	0.601

SEH = Average terrestrial solar energy received on a horizontal surface, Annapolis, Md. (Table 2 of Ref. 2).
SEXH = Average extraterrestrial solar energy received on a horizontal surface, latitude 40°N (Table 1 of Ref. 2).
RA = CI/SEH ratio, derived from Figure 10c of Ref. 2, given latitude, month, and SEH/SEXH ratio.
CI = (SEH) x (RA)
DA,F see text

Computations for the effect of the horizontal concrete reflector are given in Table 7. The following equation, used only when TY is greater than 3.1416, was derived as a best fit to the reflector data of Table 7.

$$REFL = 1.0 + 0.36 * SIN (TY - 3.1416).$$

The factor REFL is multiplied by CI to obtain the total radiation on the collector when the reflector is used.

The equations for the efficiencies of the various solar collectors compared in this study are equations for the best straight-line fit of data for the collectors as shown in Figure 7. Where the experimental data points or curves given by the manufacturers did not follow a straight line closely, the best fit was chosen in the range of solar radiation and temperatures expected in this application.

SOLAR COLLECTOR AND STRUCTURE

Benefit of Reflector

The effective collector area required versus percent solar heat is plotted in Figure 8 with and without the reflector. The effect of the refector is to increase the radiation incident on the collector during the critical winter period, and therefore to reduce the amount of auxiliary heat required. As can be seen in the graph, the reflector becomes increasingly more effective as the percent solar heat increases. At 50 percent solar heat, for example, the reflector eliminates the need for about 10 square meters of collector area. At 100 percent solar heat, the reflector replaces about 240 square meters of collector area.

The cost of the collectors, structure, and manifolds, including installation, is about \$195 per square meter of effective collector area. The cost of the reflector, however, is only about \$15 per square meter of effective collector area (the area of reflector is equal to the gross area of collector). For an installation supplying 90 percent solar heat, 220 square meters of reflector, costing \$3,300, would eliminate the need for 70 square meters of collectors and associated structure and piping, costing \$13,700. The net savings would be \$10,400.

White concrete was found to be unavailable, so it was decided to use a slab of regular concrete with a very smooth surface painted brilliant white.

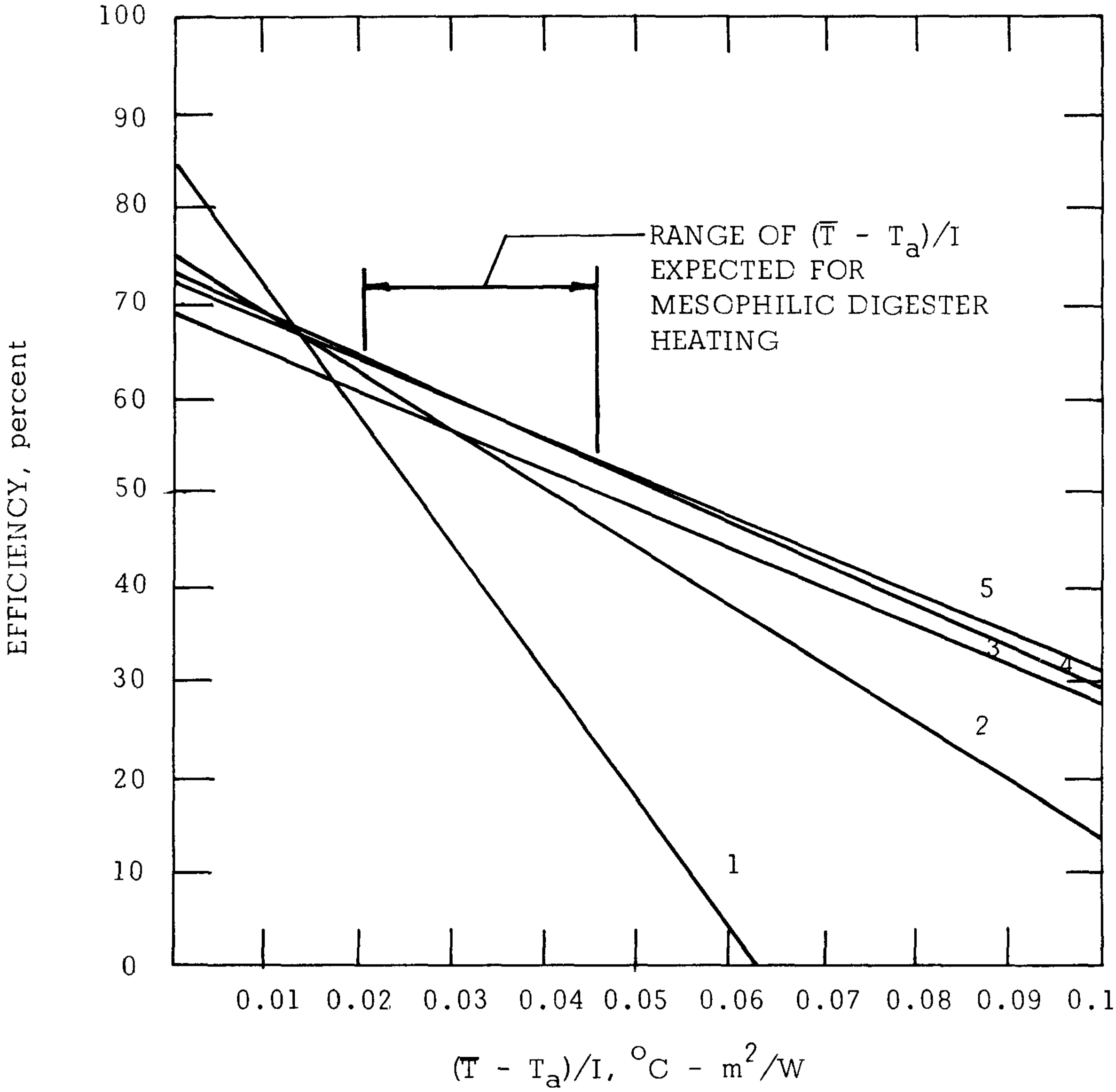
Selection of Collector

Costs per unit effective collector area for five collector designs are given in Table 8. Effective collector area is the "aperture area," or area of glass exclusive of the supporting framework. This is the area upon which collector efficiencies were calculated in all cases except for the Solaris collector for which only gross area data were available. Cost includes cost of delivery to Annapolis, Maryland, which was estimated in some cases.

Table 7. HORIZONTAL REFLECTOR FOR SOLAR PANELS

Line	Parameter	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.
1	Angle of sun's rays at solar noon, deg.	51	39	31	27	31	39	51
2	Angle of incidence of direct rays on 60^o collector, deg.	21	9	1	3	1	9	21
3	Cosine of angle of incidence of direct rays on 60^o collector, deg.	0.934	0.988	1.000	0.999	1.000	0.988	0.934
4	Angle of incidence of reflected rays on 60^o collector, deg. (Sun angle $+30^o$)	81	69	61	57	61	69	81
5	Transmittance of reflected rays through glass collector cover, relative to transmittance at 0^o incidence[a]	0.48	0.89	0.95	0.96	0.95	0.89	0.48
6	Increase in area of intercepted radiation due to reflector[b]	0.20	0.38	0.50	0.50	0.50	0.38	0.20
7	75% of above figure (assuming 75% reflectance of reflector)	0.15	0.29	0.38	0.38	0.38	0.29	0.15
8	Effective increase in intercepted radiation due to reflector[c]	0.072	0.26	0.36	0.36	0.36	0.26	0.072

[a] ASHRAE Handbook of Fundamentals (Ref. 3), data derived from Figure 5, p. 395.
[b] Graphical analysis.
[c] Line 5 times line 7.



Collector efficiency = $a - b(\overline{T} - T_a)/I$

	a	b
1. Solaris (trickle-type)	0.85	13.5
2. Sunworks, single-glaze	0.75	6.14
3. Revere, double-glaze, 4 tubes per 2-ft panel	0.69	4.14
4. PPG, double-glaze	0.73	4.31
5. Sunworks, double-glaze	0.72	4.00

Figure 7. Efficiencies of solar collectors evaluated in this study (Derived from published test data)

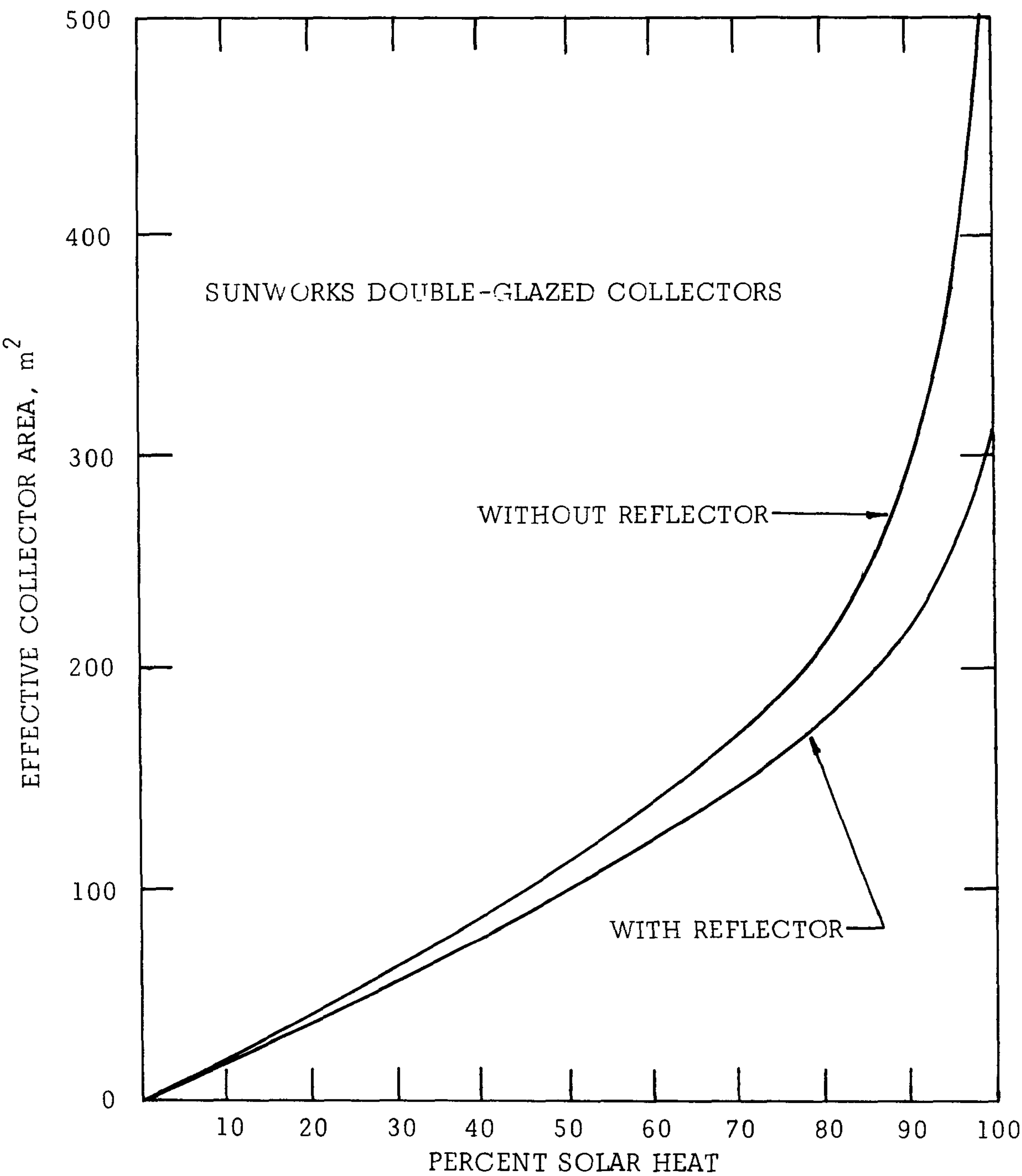


Figure 8. Solar collector area required to supply given percent of annual heat load, with and without horizontal reflector

Figure 9 shows graphs of effective collector area required versus percent solar heat for five collector designs. Multiplying the required areas by the cost per unit area of Table 8 gives, in Figure 10, the curves which show the cost of the delivered collectors versus percent solar heat.

Table 8. COST OF SOLAR COLLECTORS PER SQUARE METER OF EFFECTIVE AREA
(Costs as of spring 1976)

Solar Collector	Delivered cost per panel, $	Effective area per panel, m^2	Cost per effective area, $\$/m^2$
Sunworks, single-glaze	185	1.75	106
Sunworks, double-glaze	215	1.75	123
Revere, 4 tubes per 2-ft panel, double-glaze	200	1.62	124
PPG, copper, double-glaze	211	1.50	140
Solaris (efficiency calculated on basis of gross area)	315	4.46	71

Up to about 84 percent solar heat, the Solaris collectors are the least expensive, although the collector array would be larger (Figure 9), requiring greater piping and structure cost. Above 84 percent solar heat, the Sunworks collectors are least expensive. As will be shown later in this report, solar sludge heating at Annapolis appears most economical at about 90 percent solar heat. Therefore, the collector design will tentatively be chosen based on this value.

The cost of the Sunworks double-glazed collectors for a 90 percent solar heating system is about $800 more than the cost of the Sunworks single-glazed collectors. The cost of the structure, manifolds, reflector, and other components whose cost is directly proportional to the collector area is about $88 per square meter of effective collector area. Since 242 square meters of collector area are required for the single-glazed collectors, and 215 square meters are required for the double-glazed collectors, there is a total net savings of about $1,600 if double-glazed rather than the single-glazed collectors are used for a 90 percent system. As the percent solar heat is increased above 90 percent, the savings become greater. Therefore, as will be shown by the analyses which follow, the collector design that is most economical for this application is the Sunworks double-glazed collector.

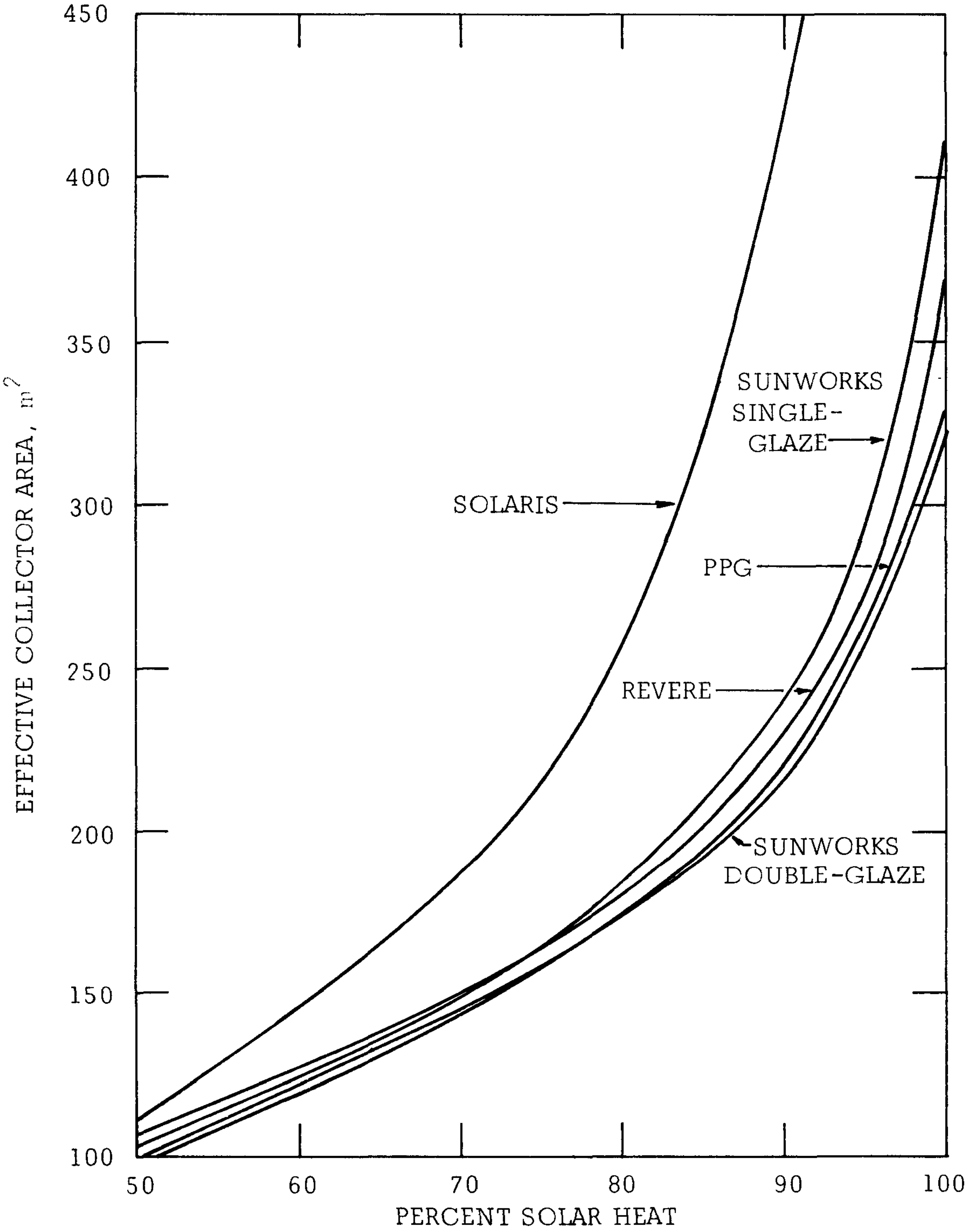


Figure 9. Effective collector area versus percent solar heat

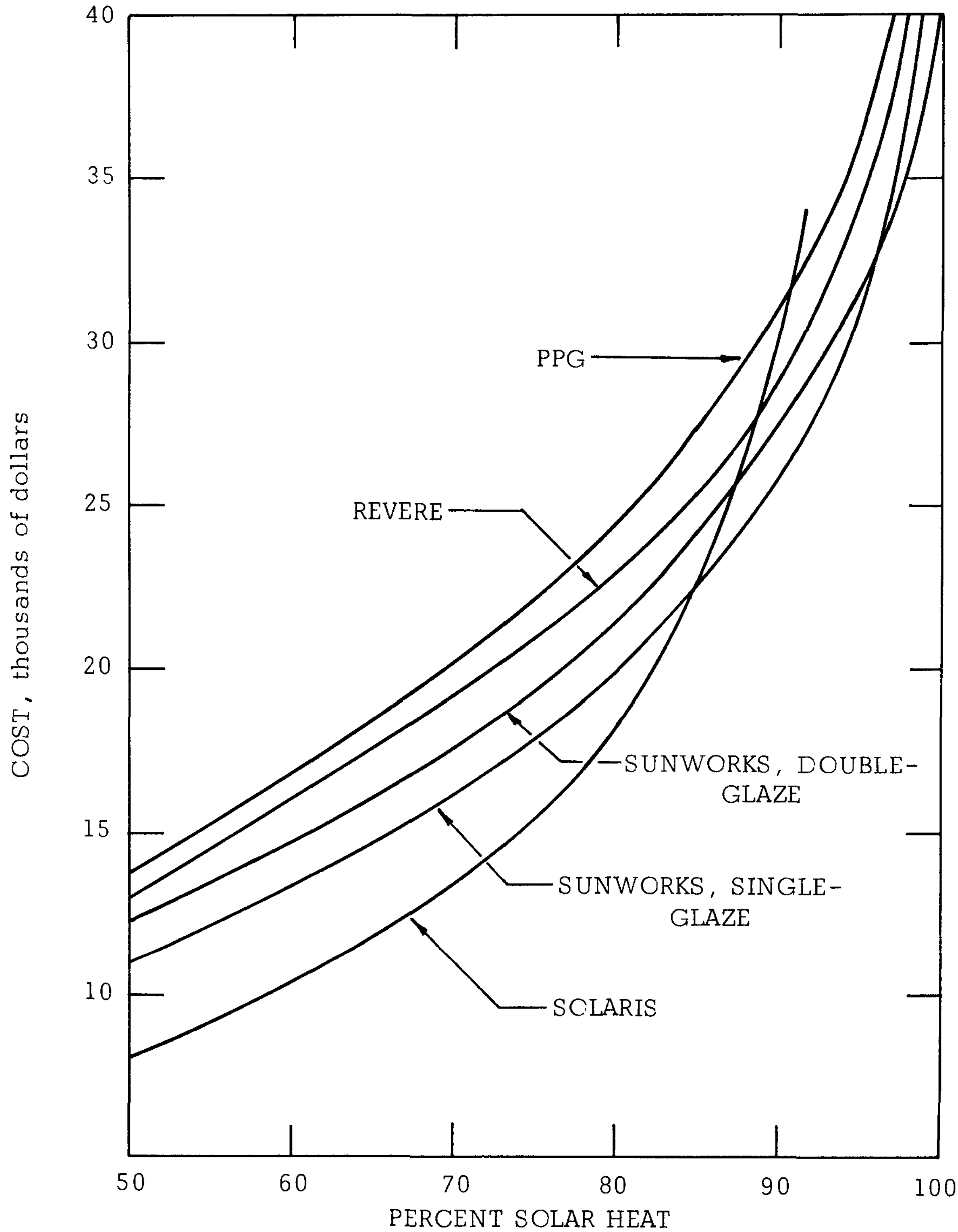


Figure 10. Cost of collectors, delivered to Annapolis

Solar Collector Mounting

Of the various considerations that entered into the choice of collector mounting arrangement, structural and manifold costs were prime considerations. Another design constraint was the fact that an access roadway had to be maintained through the construction site. Therefore, the chosen arrangment consists of two structures, each containing 65 panels in two rows. Each steel structure is 30 meters (100 feet) long and the maximum height of each is 4 meters (13 feet). Figure 11 is a sketch of the mounting arrangment. The entire plane is graded at a one percent slope so that all water in the supply and return manifolds drains toward the heat storage tank when the solar collector pump is off.

SOLAR COLLECTOR PIPING AND PUMP

Flow Rate Through Collectors

The flow rate through the collectors should be optimum for winter conditions when it is important that the solar heating system operate most efficiently. If the flow rate is very high, the average collector temperature approaches the heat storage temperature, resulting in the highest possible collector efficiency. The cost of the pump and piping, however, would be high. If the flow rate is very low, the average collector temperature is much higher than the heat storage temperature, producing low collector efficiency.

Figure 12 is a graph of collector efficiency and temperature rise through the collector versus flow rate, based on typical winter conditions and a typical double-glazed, flat-plate solar collector efficiency curve. The optimum flow rate is about 10×10^{-6} m^3/s per m^2 of effective collector area. For the design consisting of 230 m^2 of collector panels, the total optimum flow rate would be 0.0023 m^3/s (36.5 gpm).

Piping

The size of the inside manifold pipe was chosen as nominal 2-inch schedule 80 plastic pipe for the top, supply manifold and 3-inch for the bottom, return manifold. The flow will be reasonably well distributed to all panels. The pressure drop through each supply manifold will be less than 6000 Pa (2 feet), and the pressure drop through each set of two panels in series has been calculated to be about 10,000 Pa (3.3 feet). Float-actuated air vents at the highest point of each supply manifold will purge the system of air upon filling and allow air to enter when draining.

Pump

The total head of the pump will be about 120kPa (40 feet), including pressure drop through the supply piping, filter, and collectors, and elevation head. Using a pump efficiency of 60 percent and an electric motor efficiency of 90 percent, the input power to the pump motor is as follows:

Input power = 120,000 Pa x 0.0023 m^3/s (0.60 x 0.90) = 511 W (0.69 hp).

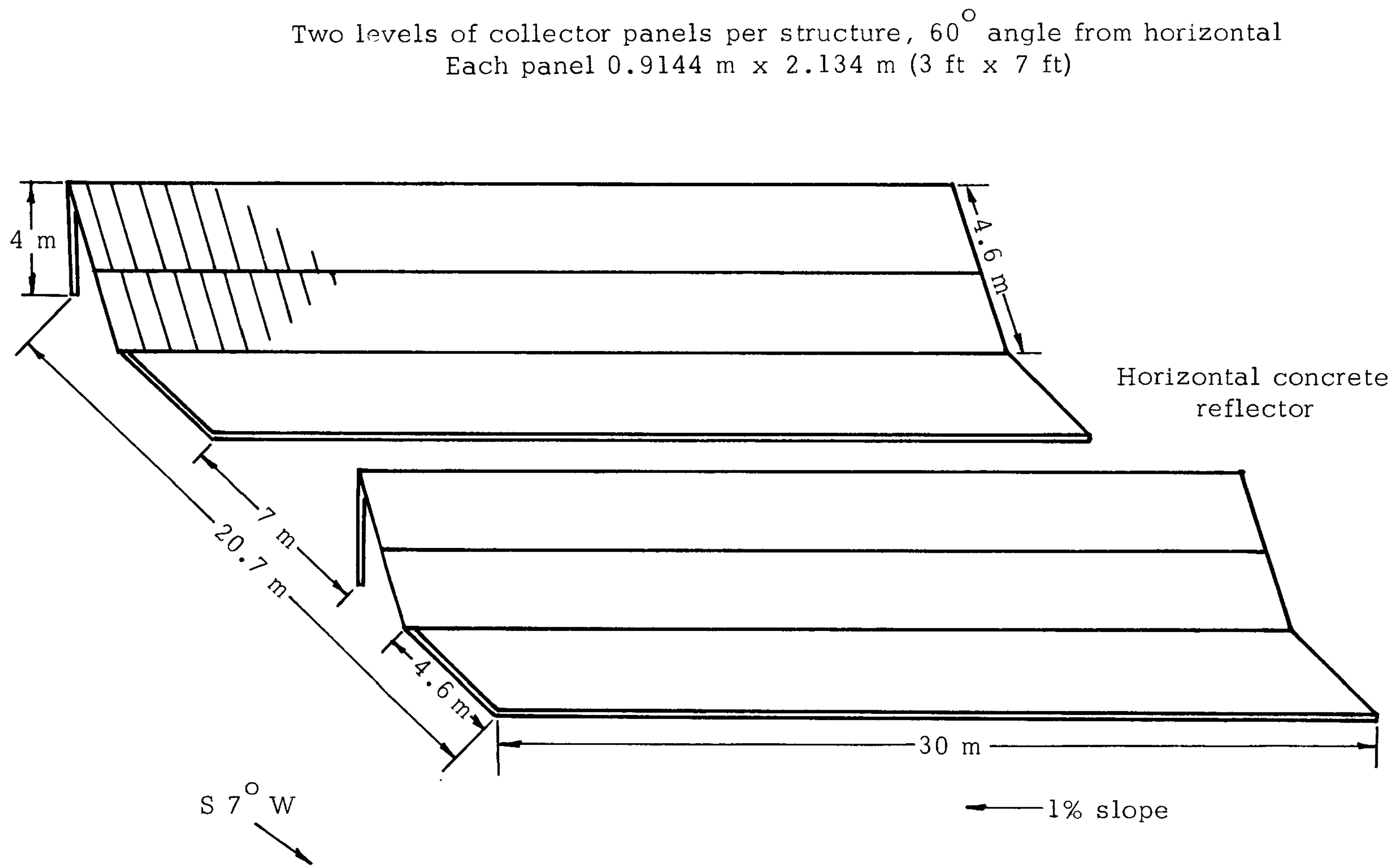
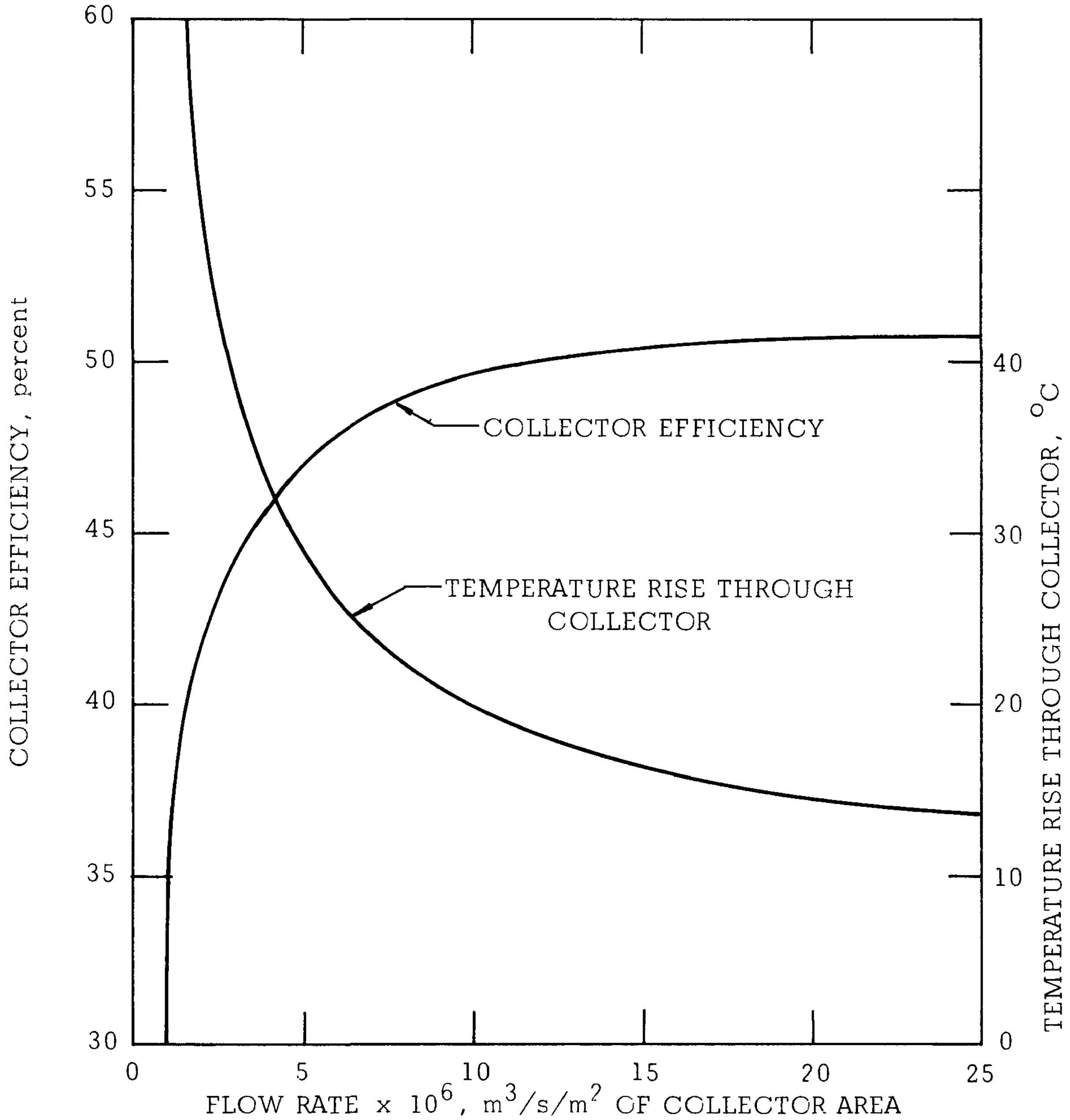


Figure 11. Solar collector mounting arrangement



January conditions

SOL = 825 W/m^2

TA = $6^{\circ}C$

T = $40^{\circ}C$

TC = $T + \frac{\Delta T}{2}$

CE = 0.69 - 4.14(TC - TA)/SOL

Figure 12. Collector efficiency and temperature rise through collector versus flow rate

A pump room is conveniently located adjacent to the Number Three digester in which the heat storage tank is located. It was decided to place a centrifugal pump, close-coupled to a 560-watt (3/4 hp) electric motor, in the pump room. The pump will be flooded at all times. An ordinary enclosed-impeller, single-volute type centrifugal pump will be used so that water can easily drain backwards through the pump when the pump is turned off. Backflow of water through the filter will present no problem.

ECONOMIC ANALYSIS

Cost of Solar Heating System

All costs associated with the solar heating system were categorized as to either fixed costs (independent of size of solar heating system) or variable costs (proportional to the size of solar heating system). The variable costs are expressed in terms of dollars per square meter of effective collector area. Because the solar heating system is completely automatic, operator costs were assumed to be zero. Maintenance costs consist primarily of repainting the reflector, structure, and heat storage tank; the present worth of these costs over the project life of 25 years was included. The cost of operating the solar collector pump (about $20 per year) was neglected. The total present worth of all costs associated with the solar heating system is summarized in Table 9. The cost of the solar heating system versus percent solar heat is shown graphically in Figure 13.

Cost of Auxiliary Heat

The present worth of the amount of gas necessary to fulfill 100 percent of the digester heat requirements over the project life of 25 years was calculated as follows. According to the U.S. Department of Commerce publication, "United States Energy Through the Year 2000,"[4] the price of all fuels will rise faster than other commodity prices. In addition, the rate of price increase for gaseous fuels is expected to be about 2.0 times the rate expected for coal. Using a rate of inflation of 6.0 percent per year, a conservative estimate for the rate of increase in the price of gas would be 12 percent per year.

Wholesale natural gas prices from 1960 through 1975 are plotted in Figure 14. These data conform very closely with the continuous interest equation:

$$p(n) = e^{rn}$$

where	$p(n)$	=	price in any year n
	r	=	annual interest rate
	n	=	year number
	e	=	exponential constant, 2.71828....

Table 9. SUMMARY OF SOLAR HEATING COSTS
(Present worth of all costs)

Item	Fixed cost $	Variable costs $/m^2 of collector area	Variable costs For 230 m^2 collector area, $
Solar heat storage tank, heat exchange, insulation, paint	5,000	49.10	11,300
Filter	50	1.10	250
Pump	50	1.10	250
Sump pump	200		
Sludge pipe	200		
Digester pipe support	500		
Controls	950		
Supply, return pipe, valves	1,000	5.57	1,280
Installation labor	2,000	23.70	5,450
Manifolds		19.70	4,530
Reflector		14.70	3,380
Structure		27.90	6,420
Site grading		2.10	500
Solar collectors		123.00	28,300
Maintenance		28.30	6,500
Total	9,950	296.00	68,200

Present worth of total solar system = \$9,950 + \$296 (CA)

where CA = effective collector area, m^2

A least-squares analysis has been made for two time periods, 1968--1975 and 1971--1975:

1. 1968--1975 r = 10.6% with a coefficient of determination of 0.877
2. 1971--1975 r = 16.52% with a coefficient of determination of 0.931

It is understood that the second figure may be reduced somewhat if the OPEC monopoly can be broken, but that is a contingency that, from a conservative point of view, should not be anticipated. In any event, as

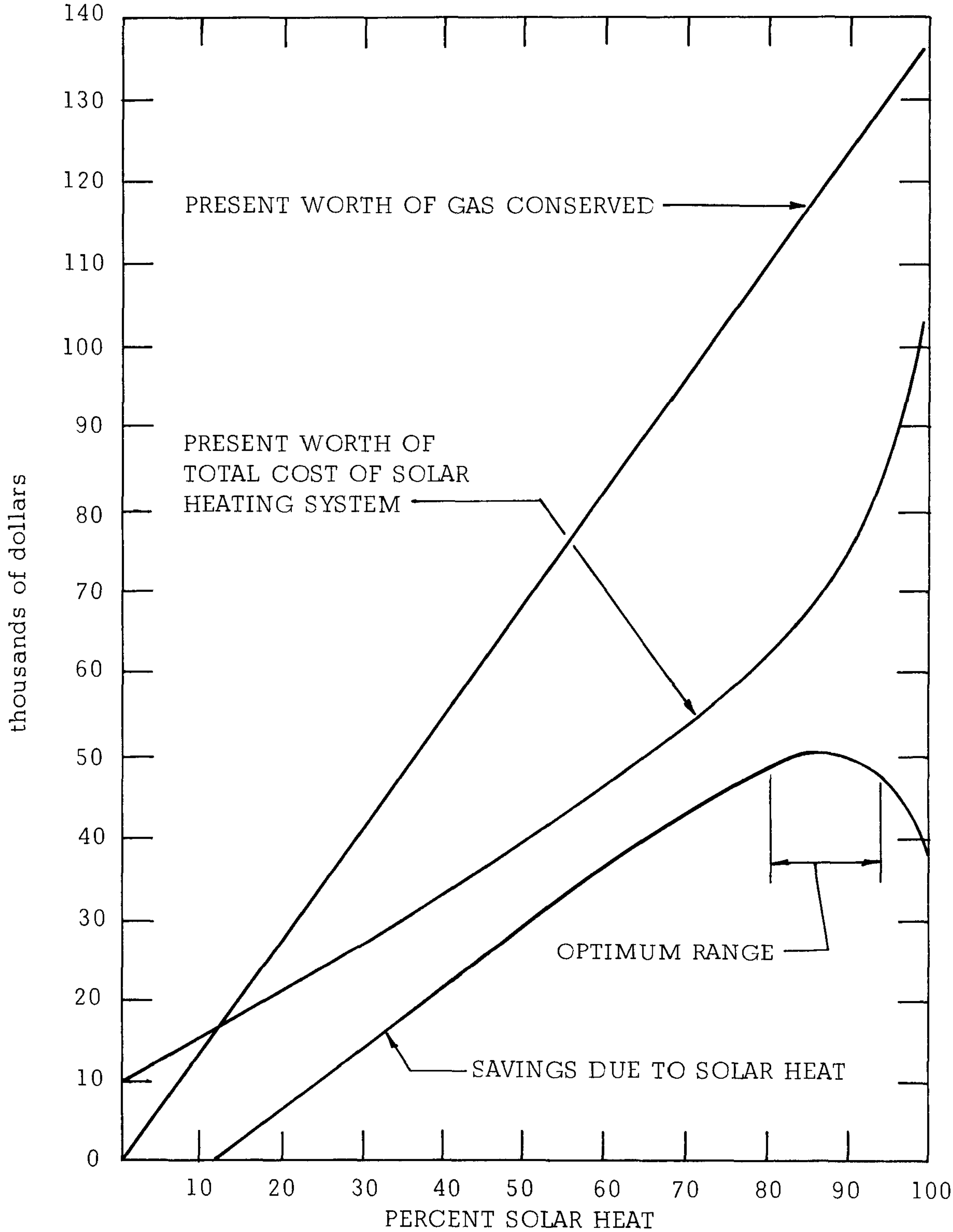


Figure 13. Economic analysis

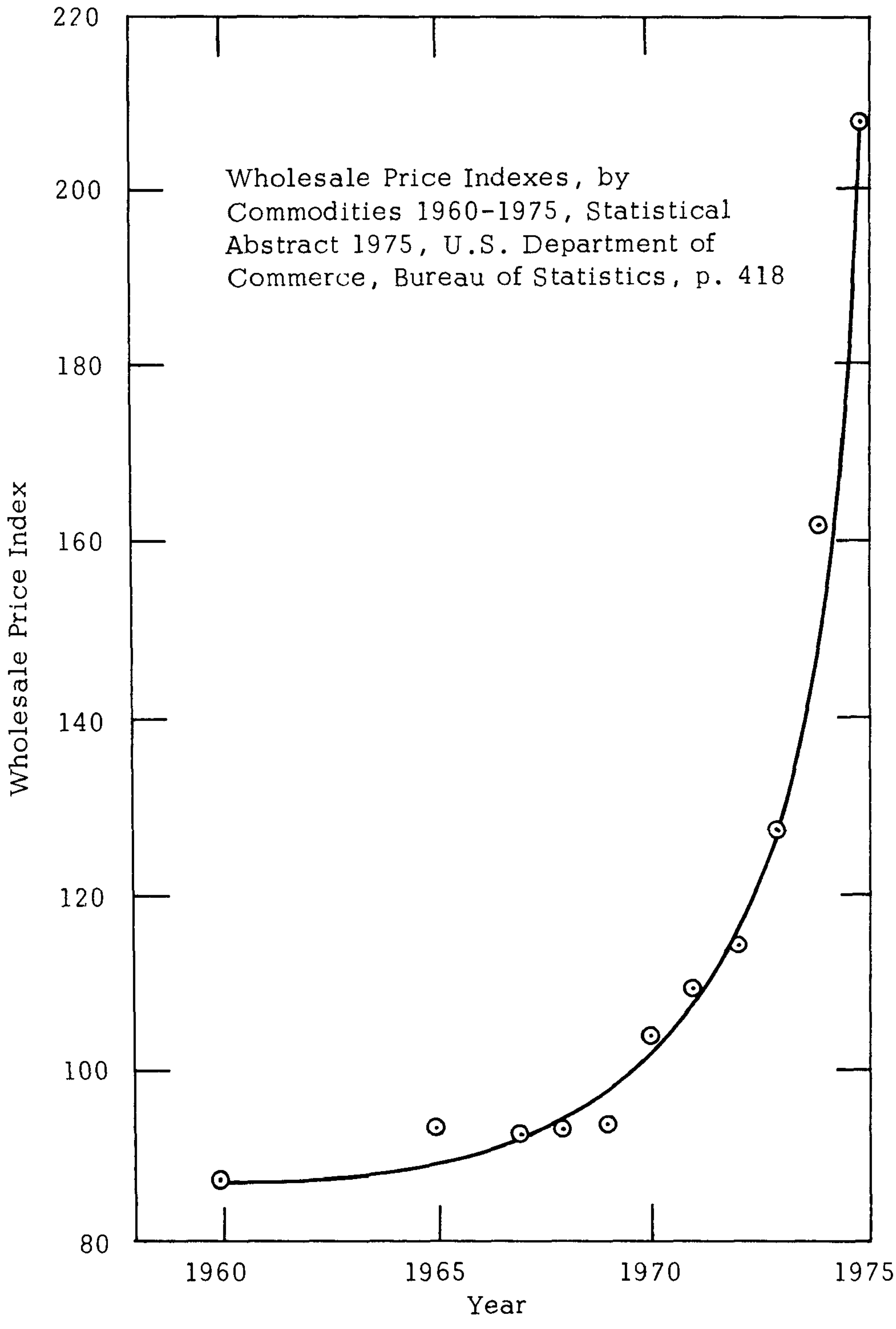


Figure 14. Wholesale Price, Natural Gas 1960-75

petroleum reserves on shore and at shallow depth are exhausted, the real costs of exploration can be expected to rise exponentially. An assumption of 12.0 percent increase in prices is not unreasonable for fossil fuels in general, and gas in particular.

The price of gas in any year n will be a factor of $(1.12)^n$ times the present price. The present worth of the cost in year n, using an inflation rate of 6.0 percent, is $(1.06)^{-n}$ times the cost in year n. Summing for values of n from 1 to 25 gives a factor, which, multiplied by the present annual cost of gas, will give the total present worth of gas heating costs over the 25-year period:

$$\text{Factor} = \sum_{n=1}^{25} (1.12)^n \times (1.06)^{-n} = \sum_{n=1}^{25} (1.057)^n = 55.3.$$

The total annual heat requirement of the digester is 8.18×10^{11} J. Assuming a combustion efficiency of 66 percent, an amount of gas having a net heating value of 8.18×10^{11} J/0.66, or 1.24×10^{12} J would be required annually.

Current prices of natural gas and other fuels are given in Table 10 along with the corresponding present annual cost of gas to fulfill 100 percent of the digester heat requirement. The prices for natural gas at the well-head and at the "city gate" are averages of the Federal Power Commission regulated prices, which are well below fair market prices. If the interstate gas prices are deregulated it is expected that the price of natural gas would be equivalent to that of fuel oil at the present time. Considering the long-term projections used herein, the most realistic present price of natural gas would be the fair market value of \$2.00/GJ, not an artificially low regulated price. This is equivalent to a present cost for digester heating of \$2,480 per year. Multiplying by the factor 55.3 gives a present worth for 100 percent gas heating of the digester over 25 years of \$137,000. The straight line on the graph (Figure 13) indicates the present worth of digester gas saved versus percent solar heat.

Optimum Size Solar Heating System

Subtracting the solar heating cost of Figure 13 from the auxiliary heat cost gives the present worth of the total savings as a result of using the solar heating system. The maximum savings occurs at 85 percent solar heat and is equal to about \$50,000. The savings is within 5 percent of this amount from values of percent solar heat from 80 to 94 percent. The solar digester heating system for the Annapolis wastewater treatment plant will be based on approximately 90 percent solar heat.

Table 10. CURRENT PRICES OF NATURAL GAS AND OTHER FUELS

Fuel source	Price, \$/GJ	Annual cost of 100% heating one digester, \$
Gas, wellhead, Jan. '76[a]	0.364	450
Gas, "city gate," Jan.'76[b]	0.837	1,040
Gas, intrastate, 1975[c]	1.80	2,230
Gas from solid waste[d]	1.97	2,440
Fuel oil	2.45	3,040
Gas, home heating[e]	2.53	3,140
Gasoline	4.98	6,170

[a] Federal Power Commission, interstate, average 34 major pipeline companies.

[b] Federal Power Commission, interstate, average 34 major pipeline companies.

[c] Standard & Poors Industry Surveys - Oil (Ref. 5), p. 71.

[d] Fuel Gas Production from Solid Waste (Ref. 6).

[e] Baltimore Gas & Electric Company residential gas rate, November, 1975 (Ref. 7).

PART II

ECONOMIC FEASIBILITY OF SOLAR DIGESTER HEATING THROUGHOUT THE UNITED STATES

by

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Order No. CA-6-99-3499-A

SECTION 4

INTRODUCTION, CONCLUSIONS AND RECOMMENDATIONS

INTRODUCTION

The use of solar energy to heat anaerobic sludge digesters was determined to be technically and economically feasible for Annapolis, Maryland. The results of this study were reported in Part I of this report. This nationwide study was undertaken to determine the economic feasibility of this concept at all locations in the United States. In addition, it was desirable to develop specific guidelines for sizing the most economical solar heating system at any given location.

The solar heating system considered for use throughout the country is identical to that described in detail in Part I for Annapolis, except as modified for other locations according to the guidelines presented in Part II.

CONCLUSIONS

Solar heating of anaerobic digesters is economically feasible at all locations in the United States. Areas of the country showing the greatest economic attractiveness include Alaska, the central northwest states of Montana, North Dakota, Nevada and southward to New Mexico, and also the eastern state of Maine. Least attractive areas include the southeastern United States, especially Florida and southern Texas, and the northwest Pacific coast. The degree of economic attractiveness of solar digester heating is, generally speaking, proportional to the average annual solar radiation intensity multiplied by the difference between digester temperature ($35^{o}C$) and average annual air temperature.

The optimum-size solar heating system, expressed as solar heat input to the digester as a percentage of the total annual heat load, varies with location from about 82 to 97 percent. In general, within this range, the optimum percent solar heat for a given location varies in direct proportion to the average annual solar radiation intensity. The optimum-size solar heating system, in terms of solar collector area and total cost, is higher for higher latitudes and lower for lower latitudes. However, since the digester heating requirements are greater at the higher latitudes, economic gains are also greater due to the comparatively low cost of solar heat compared to the higher cost of heating with conventional fuels.

Of the three principal types of liquid flat-plate solar collector designs evaluated, the most economical design for virtually all locations in the United States consists of copper tubes attached to copper sheet with selective, high energy absorbing black coating, covered with two layers of glass. Of the locations studies, a collector of the same design, but with only one glass cover, proved to be marginally economical only at Miami, Florida. The black coated, corrugated-aluminum, trickle-type collector with one glass cover is shown to be the least economical design for digester heating an any of the locations evaluated.

RECOMMENDATIONS

All existing anaerobic sludge digesters in the United States should be fitted with solar heating systems wherever it is physically possible to construct the system. New wastewater treatment plants should be planned to allow incorporation of solar-heated digesters as well as other energy conservation measures. Special effort should be made to use solar heat in areas where it is most economically attractive.

Before construction of a solar digester heating system at a particular location, the computer program contained herein should be run for that location, to determine the optimum solar heating system, using the most accurate design parameters available.

SECTION 5

GUIDELINES

GENERAL

The following guidelines constitute modifications of the Annapolis design for heating anaerobic digesters with solar heat at any location in the United States. The recommended procedure is to determine all major design factors, in the order given, for a sludge flow rate equal to that used at Annapolis (and in the computer model for all locations) and then scale certain factors to the given wastewater treatment plant size.

SPECIFIC GUIDELINES

Solar Collector Area

The optimum size of the solar heating system curve shown in Figure 15 is a function of the effective, or net solar collector only. The actual gross, or total collector area required would be about ten percent greater than the "net" surface area represented by the curve.

To estimate the optimum collector area for any location between 25 and 50 degrees north latitude in the United States, the following equation can be used:

$$CA = 8.0(Lat) - 110$$

where CA = effective collector area, m^2

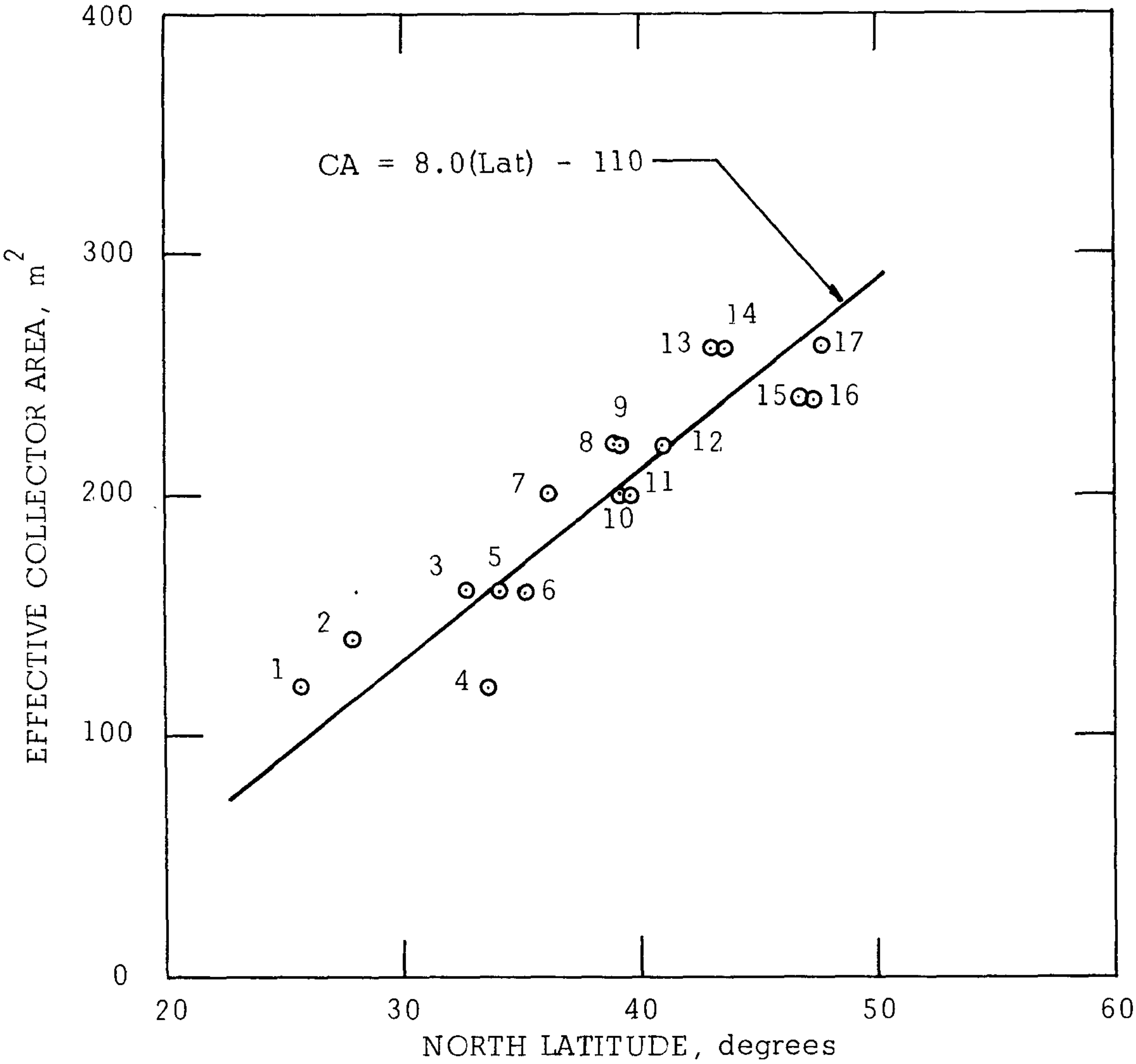
Lat = north latitude, degrees

At locations greater than 50 degrees north latitude, use of the above equation results in larger collector size than actually required for optimum operation. It is for this reason that a data point for Fairbanks, Alaska, also evaluated as part of the nationwide study, does not appear in Figure 15.

Cost of Solar Heating System

The cost of the solar heating system is proportional to the effective solar collector area, according to the following equation (see Figure 16):

$$Cost = 9{,}950 + 296(CA)$$



1. Miami, FL
2. Corpus Christi, TX
3. Dallas, TX
4. Phoenix, AZ
5. Los Angeles, CA
6. Albuquerque, NM
7. Nashville, TN
8. Annapolis, MD
9. Topeka, KS
10. Grand Junction, CO
11. Reno, NV
12. Salt Lake City, UT
13. Madison, WI
14. Portland, ME
15. Bismark, ND
16. Great Falls, MT
17. Seattle, WA
18. Fairbanks, AK

Figure 15. Determination of optimum size solar heating system given latitude of loaction

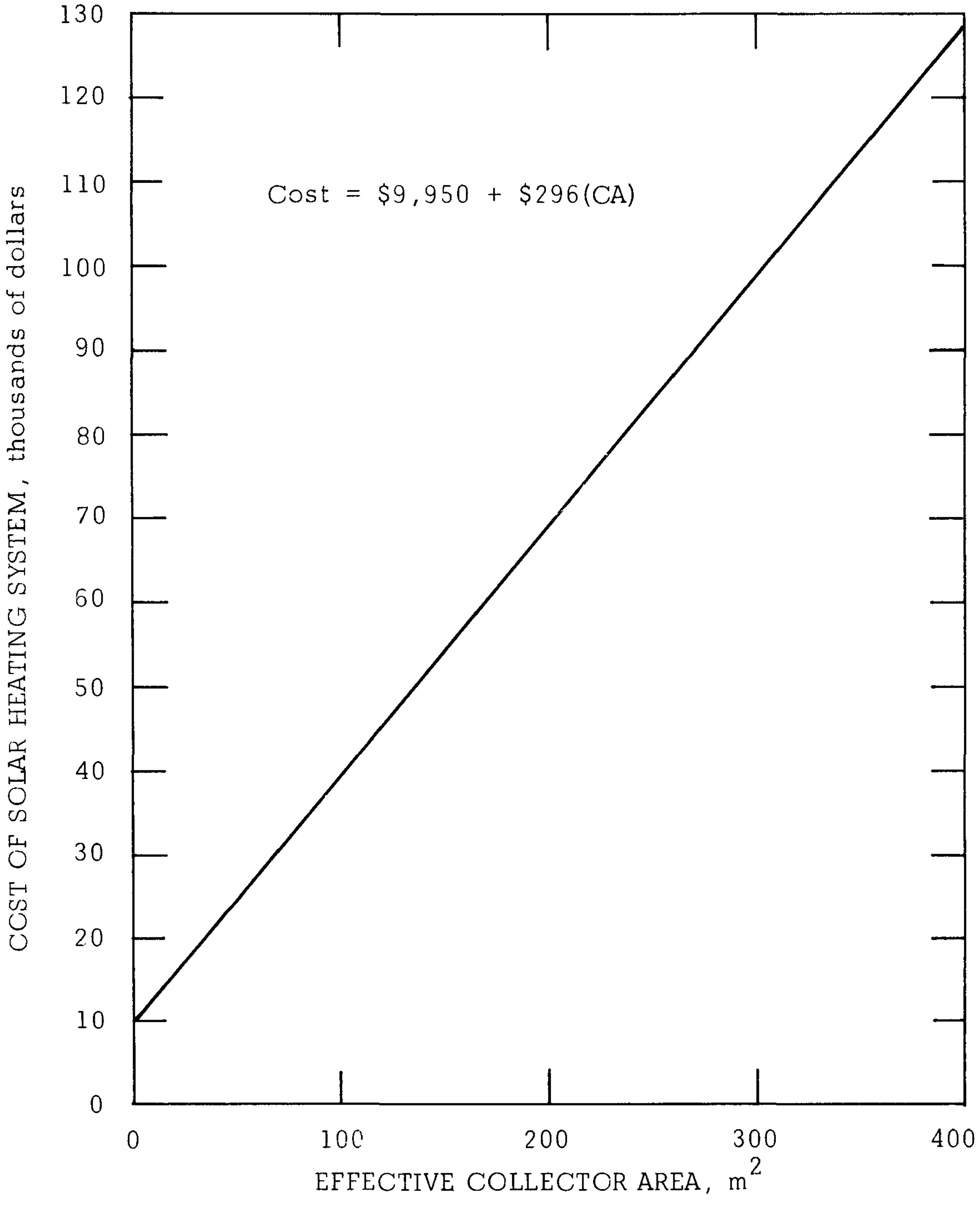


Figure 16. Determination of cost of optimum-size solar heating system given effective collector area

where Cost = present worth of total cost of solar heating system, dollars

CA = effective collector area, m^2

This represents the present worth of equipment, installation, operation and maintenance costs over the 25-year project life. Immediate cash outlay would amount to about 90 percent of the total cost.

Percent Solar Heat

Optimum percent solar heat, as shown in Figure 17, for a given location can be estimated by the equation:

$$PSOL = 0.138(SEH) + 65$$

where PSOL = optimum percent solar heat, %

SEH = average annual solar energy on a horizontal surface, W/m^2

Average annual solar energy for 125 cities in the United States is given in Table 11, condensed from Reference 2.

The size of solar heating system that was determined from latitude (Figure 15) should automatically provide the percent solar heat given by the above equation. By knowing the value of percent solar heat for the optimum design then the percent of the total annual heat load, and therefore the value of the digester gas conserved can be determined.

Savings Due to Solar Heat

The savings due to solar heat are representative of the economic attractiveness of a solar heated digester at any given location. "Savings" is defined as present worth of gas conserved, less present worth of the cost of the solar heating system, both over the project life of 25 years. As shown in Figure 18, savings can be estimated for a given location as follows:

$$\text{Savings} = 15.7(SEH)(35 - AAAT) - 4{,}290$$

where Savings = present worth of savings due to optimum-size solar heating system, $

SEH = average annual solar radiation, W/m^2

AAT = average annual air temperature, ^{o}C.

Collector Angle

The flat-plate solar collectors should face approximately due south, tilted at an angle of latitude plus 20 degrees from the horizontal as shown in Figure 19.

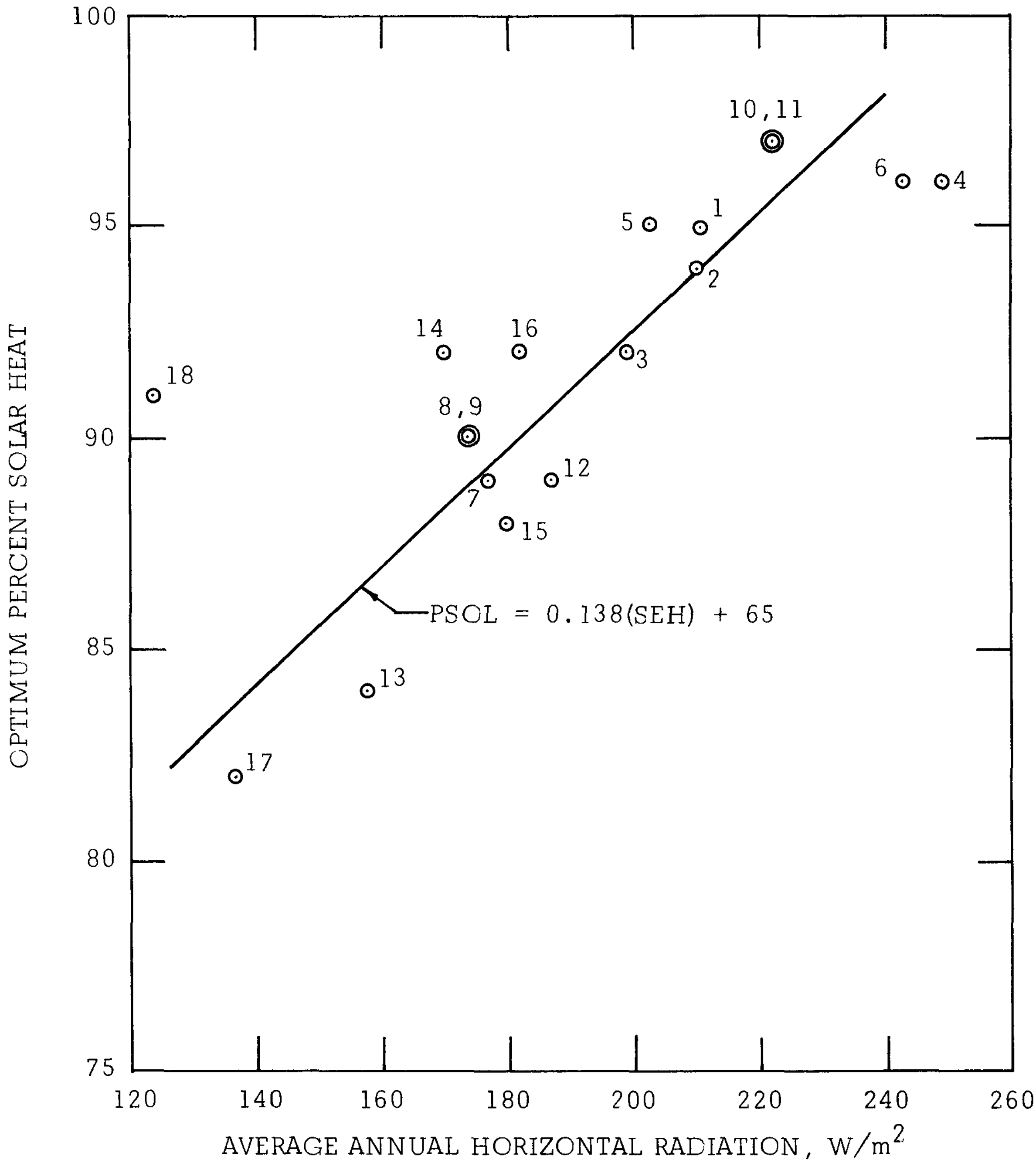


Figure 17. Determination of optimum percent solar heat given average solar radiation

Table 11. AVERAGE ANNUAL SOLAR ENERGY RECEIVED ON A HORIZONTAL SURFACE

State	City	Solar energy, W/m^2	State	City	Solar energy, W/m^2
AK	Annette	121	GA	Atlanta	191
	Barrow	100		Griffin	201
	Bethel	113	HI	Honolulu	250
	Fairbanks	108		Pearl Harbor	234
	Matanuska	108	ID	Boise	191
AR	Little Rock	186		Twin Falls	183
AZ	Page	240	IL	Chicago	132
	Phoenix	252		Lemont	166
	Tucson	251		Moline	170
	Yuma	245	IN	Indianapolis	167
CA	Davis	210	IA	Ames	167
	Eureka	152	KS	Dodge City	216
	Fresno	216		Kansas City	184
	Inyokern	280		Manhattan	180
	La Jolla	184		Topeka	180
	Los Angeles-WBAS	216	KY	Lexington	199
				Louisville	174
	Los Angeles-WBO	211	LA	Lake Charles	200
				New Orleans	193
	Pasadena	212		Shreveport	194
	Riverside	227	MA	Blue Hill	159
	San Mateo	192		Boston	151
	Santa Maria	233		Cambridge	156
	Soda Springs	222		East Wareham	156
CO	Boulder	178		Lynn	153
	Grand Junction	221	MD	Annapolis	172
	Grand Lake	201		Silver Hill	174
DC	Washington	161	ME	Caribou	153
FL	Aplachicola	215		Portland	170
	Belle Isle	192	MI	East Lansing	151
	Gainesville	209		Sault Ste Marie	161
	Jacksonville	196			
	Key West	219	MN	St. Cloud	168
	Miami	219	MO	Columbia	184
	Pensacola	201	MT	Glasgow	188
	Tallahassee	201		Great Falls	177
	Tampa	219		Summit	151

Table 11 (continued).

State	City	Solar energy, W/m^2	State	City	Solar energy, W/m^2
NB	Lincoln	176		El Paso	259
	North Omaha	183		Fort Worth	231
	North Platte	190		Midland	226
ND	Bismarck	179		San Antonio	214
NC	Cape Hatteras	216	UT	Flaming Gorge	206
	Greensboro	185		Salt Lake City	191
NJ	Sea Brook	165	VA	Norfolk	185
	Trenton	172	VT	Burlington	153
NM	Albuquerque	248	WA	Friday Harbor	155
NV	Ely	226		Pullman	180
	Las Vegas	246		Prosser	193
	Reno	232		Seattle	132
NY	Ithaca	146		Spokane	175
	New York	157		Tacoma	145
	Sayville	170	WI	Greenbay	158
	Schenectady	136		Madison	157
	Upton	172		Milwaukee	167
OH	Cleveland	155	WY	Lander	214
	Columbus	165		Laramie	198
	Put in Bay	161			
OK	Oklahoma City	211			
	Stillwater	196			
OR	Astoria	146			
	Medford	136			
PA	Philadelphia	172			
	State College	162			
RI	Newport	164			
SC	Charleston	197			
SD	Rapid City	190			
TN	Oak Ridge	176			
	Memphis	192			
	Nashville	179			
TX	Brownsville	211			
	Corpus Christi	211			
	Dallas	199			

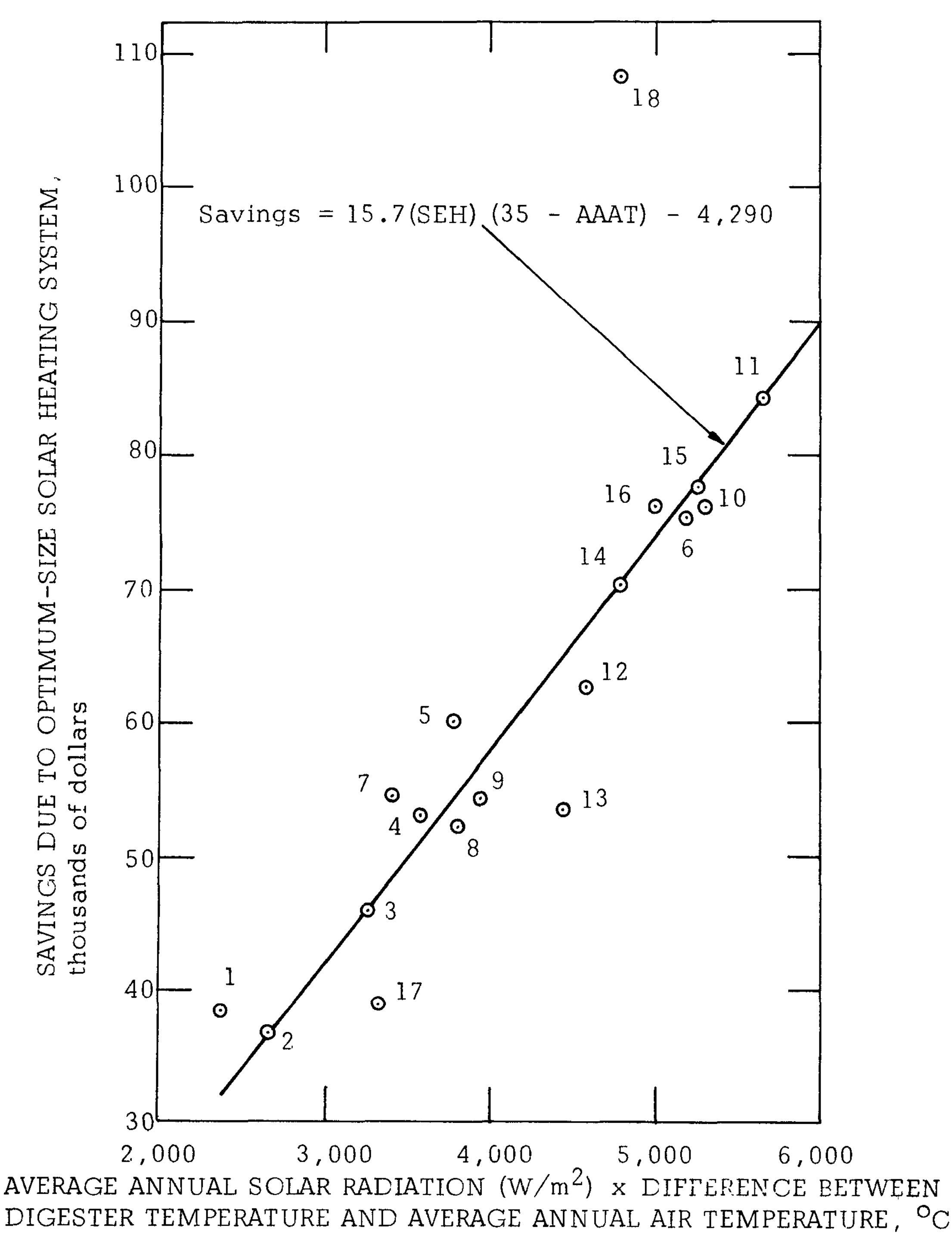


Figure 18. Determination of savings due to solar heat given average solar radiation and average annual air temperature

Reflector

Also as shown in Figure 19, the reflector should be positioned such that the included angle between collector and reflector is 120 degrees. Reflector width, "x" in Figure 19 should be equal to the length of the collector panels.

Collector Pump

A standard volute-type centrifugal pump should be used to circulate water from the solar heat storage tank to the collectors. Pump total head should be equal to the difference in elevation between the top of the collectors and water level in the storage tank plus friction loss through the supply piping and collectors. Flow rate through the collector should be about 10^{-5} m^3/s (0.16 gallons per minute) for each square meter of collector area.

Heat Storage Tank

Solar heat storage tank size is not critical, but the larger the better. The suggested minimum tank size in cubic meters is obtained by multiplying the solar collector area in square meters by 0.20.

SCALING TO PLANT SIZE

The optimum percent solar heat, collector angle, and reflector configuration are independent of the size of the wastewater treatment plant or sludge flow rate. Collector area, cost of the solar heating system, savings, pump size, and solar heat storage tank size are all proportional to the sludge flow rate. These five factors can be adjusted for plant size by multiplying the values obtained from "Specific Guidelines" above by the ratio of the actual average raw sludge flow rate divided by 0.236 kg/s. Alternately, they can be multiplied by the number of persons served by the plant divided by 17,500.

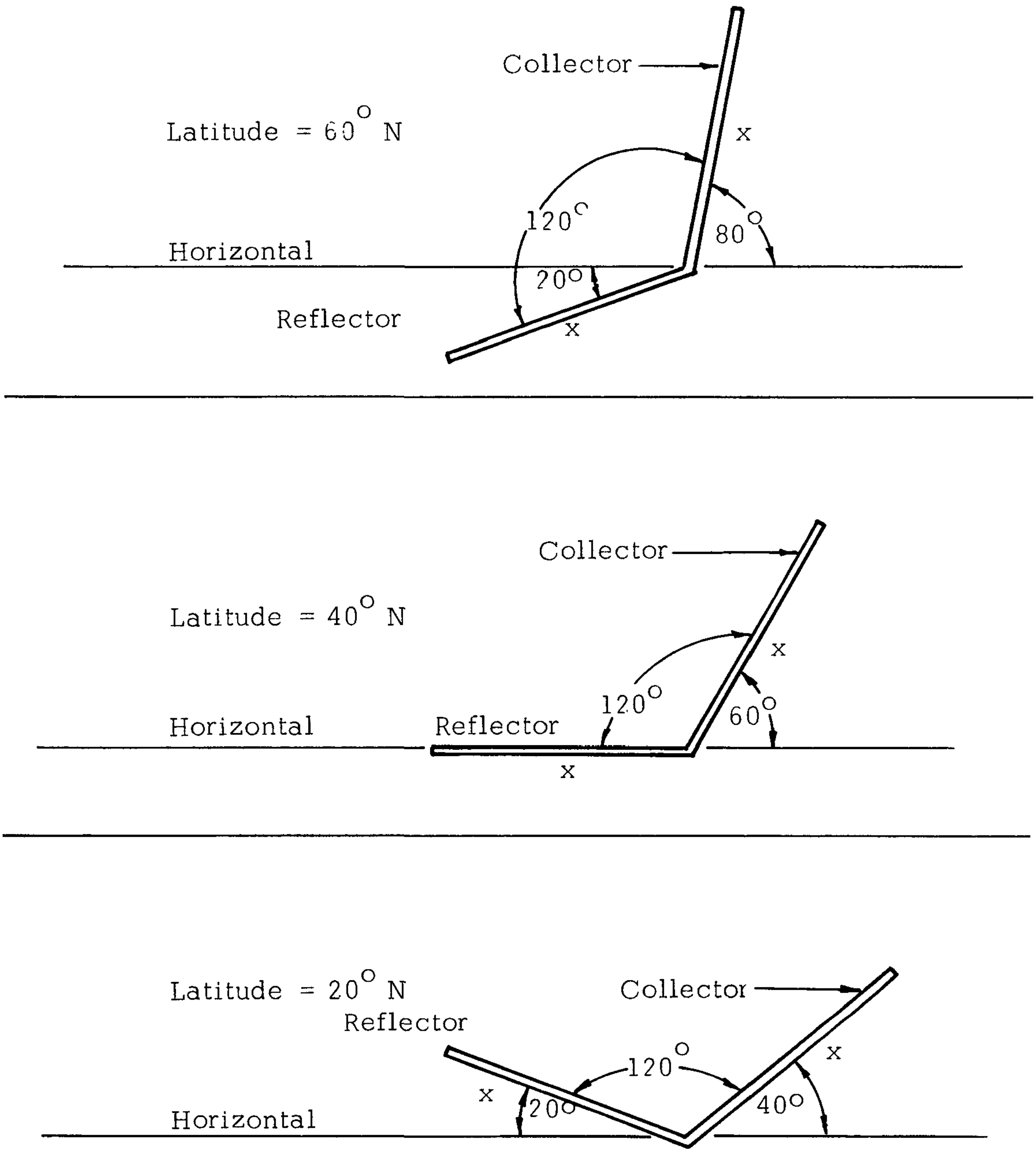


Figure 19. Optimum collector and reflector angles for three latitudes (Cross section view looking west)

SECTION 6

ASSUMPTIONS

ASSUMPTIONS USED IN ANNAPOLIS STUDY

Plant Operation

It is assumed that the raw sludge flow rate is an average of 0.236 kg/s, with a heat capacity equal to that of water. A constant temperature of 35°C is maintained in the digester. Digester gas consists of 40 percent methane and 60 percent carbon dioxide by mass. The low heating value of methane is used with a combustion efficiency of 66 percent. Heat loss from the digester varies sinusoidally throughout the year with a maximum heat loss in January and minimum in July.

Solar Heat Collection

Solar radiation varies gradually throughout the year based on monthly averages. Radiation intensity is constant throughout the period of availability each day. Air temperature during solar heat collection is equal to average daily temperature plus five degrees Celsius. Solar collector efficiency curves are straight-line approximations of published data.

Economic Factors

Present cost of methane is estimated to be $2.00/GJ for all areas of the United States, increasing at the rate of 12 percent per year throughout the project life of 25 years. The inflationary rate is constant at 6 percent per year. Cost of solar collectors is based on cost per unit effective area for all collectors.

ADDITIONAL NATIONWIDE ASSUMPTIONS

It was necessary to include the following additional assumptions to expand the Annapolis study for nationwide application.

Plant Operation

Based on known raw sludge temperatures for two cities, Annapolis and Boston[8], the raw sludge temperature for all locations is expressed as a function of ambient air temperature. Average annual sludge temperature was assumed to be five degrees Celsius higher than the average annual air temperature, with an annual variation amplitude equal to one-third of the air temperature variation. Annual variation in air temperature is sinusoidal, with maximum in July and minimum in January, the curve of each location approximating published monthly averages.[9]

Heat loss from the digester walls was assumed to have the same absolute value at all locations as at Annapolis. This requires only slightly more insulation in colder climates, and less in warmer climates. Heat loss from the solar heat storage tank, however, will vary according to air temperature for different regions, because the same value of thermal resistance of the tank insulation is used for all locations.

Solar Heat Collection

Average solar radiation on a horizontal surface varies sinusoidally throughout the year, with maximum radiation on June 21 and minimum on December 21. This value of radiation is adjusted by dividing by the cosine of the angle of the collector perpendicular to the sun's rays at solar noon for any given day, and multiplying by the cosine of the angle that the collector deviates from that theoretical perpendicular collector. Solar radiation intensity during collection is equal to the total energy received by the collector during the day divided by both the collector area and one-half the time between sunrise and sunset. The temperature of the water in the collector is calculated assuming steady-state heat-transfer conditions based on prevailing solar radiation intensity, water flow rate, and heat storage temperature.

SECTION 7

COMPUTER MODEL

COMPUTER PROGRAM

The computer program for this study, designated "SOL 6," is similar to that used for the Annapolis study. The computer program is given in Appendix B. Several modifications and improvements were made to the previous program "SOL 4" to facilitate input of data for various locations. The basic logic is described in the following steps:

1. Set values of constants, set initial values of variables.
2. Collector area = 20 m^2.
3. Set initial values of maximum and minimum heat storage temperatures.
4. Set cumulative annual heat transfers to zero.
5. Calculate temperatures of air and raw sludge, digester heat required, and heat loss, for given day.
6. Calculate actual heat transferred to sludge from solar heat storage, and auxiliary heat used, for given day.
7. Calculate solar heat storage temperature after heat is transferred to sludge, but before solar input from collectors.
8. Calculate available solar energy input to heat storage for given day.
9. Calculate heat storage temperature assuming all available solar energy is collected.
10. Reduce solar heat storage input to amount required by digester if necessary (summer operation); revise storage temperature.
11. Add all daily heat transfers to cumulative total to date.
12. Update maximum and minimum storage temperatures and their day of occurrence.
13. If not last day of year, increment TY (time of year) for next day and return to step #5; if last day of year, proceed to step #14.

14. If this was first time through year, return to step #3; if second time through year, proceed to step #15.

15. Calculate total annual percent solar heat.

16. Calculate present worth of total savings due to using given percent solar heat over 25-year period.

17. Print out collector area, percent solar heat, and savings.

18. Update values of all parameters at optimum percent solar heat.

19. If collector area is less that 400 m^2, increase collector area by 20 m^2 and return to step #3; if collector area = 400 m^2, proceed to step #20.

20. Print out annual summary of all parameters at optimum percent solar heat.

SPECIFIC MODIFICATIONS

Several of the more important modifications of "SOL 4" to produce "SOL 6" for this study are given below.

Addition of Economic Analysis

For each collector type and location, the revised computer program determines the most economical size of solar heating system. After running through the year's calculations for a given collector area, the percent solar heat, present worth of gas conserved, present worth of cost of solar heating system, and savings are calculated. If the savings are greater than for the greatest savings previously found, values for all annual summary parameters are revised to reflect the new optimum size heating system. Incorporating the economic analysis into the computer program allows quick and accurate analyses for all locations.

Solar Heat Collection

The horizontal solar radiation equation is expressed in the following form:

$$SEH = a + b * SIN(TY)$$

where SEH = average daily terrestrial solar radiation on a horizontal surface, W/m^2

a = average annual value, W/m^2

b = amplitude of variation throughout year, W/m^2

TY = time of year, radians, starting at spring equinox.

The value of "a" is taken as the average of the maximum and minimum monthly values from Table 12 of Reference 2. Figure 20 shows the average monthly values of "SEH" together with the smooth curve derived from the maximum and minimum values only for two typical locations evaluated in this study.

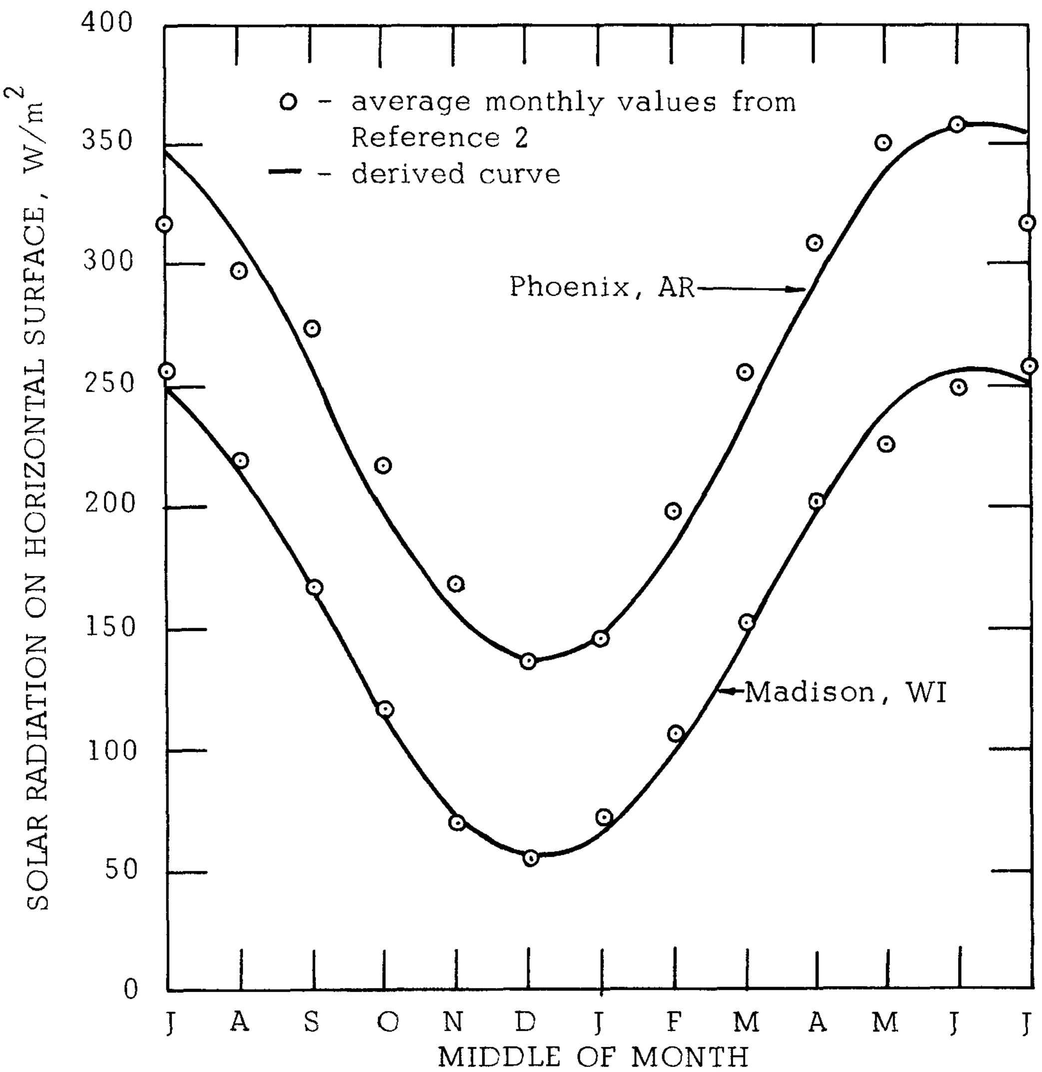


Figure 20. Comparison of measured average monthly solar radiation and derived curve for use in this study, two typical locations

Total solar energy input to the collectors during each day is expressed by:

$$CI = SEH * COS(DA) * P * CA/COS(PERP)$$

where CI = total solar energy intercepted by collector during day, J

SEH = average daily terrestrial solar radiation on a horizontal surface, W/m^2

DA = deviation angle between actual fixed collector and one that would be perpendicular to sun's rays at solar noon each day, rad

P = period, equal to 86,400s (one day)

CA = effective collector area, m^2

PERP = angle of collector (from horizontal) that would be perpendicular to sun's rays at solar noon each day, rad

The intensity of the solar radiation during collection is given by:

$$SOL = 2.0 * CI/(DAYL * CA)$$

where SOL = intensity of solar radiation during collection, W/m^2

CI = total energy intercepted by collector during day, J

DAYL = time between sunrise and sunset, s

CA = effective collector area, m^2.

The factor of 2.0 in the equation results from considering the solar radiation to be constant over one-half of the time from sunrise to sunset. Length of day is calculated each day based on latitude and time of year.

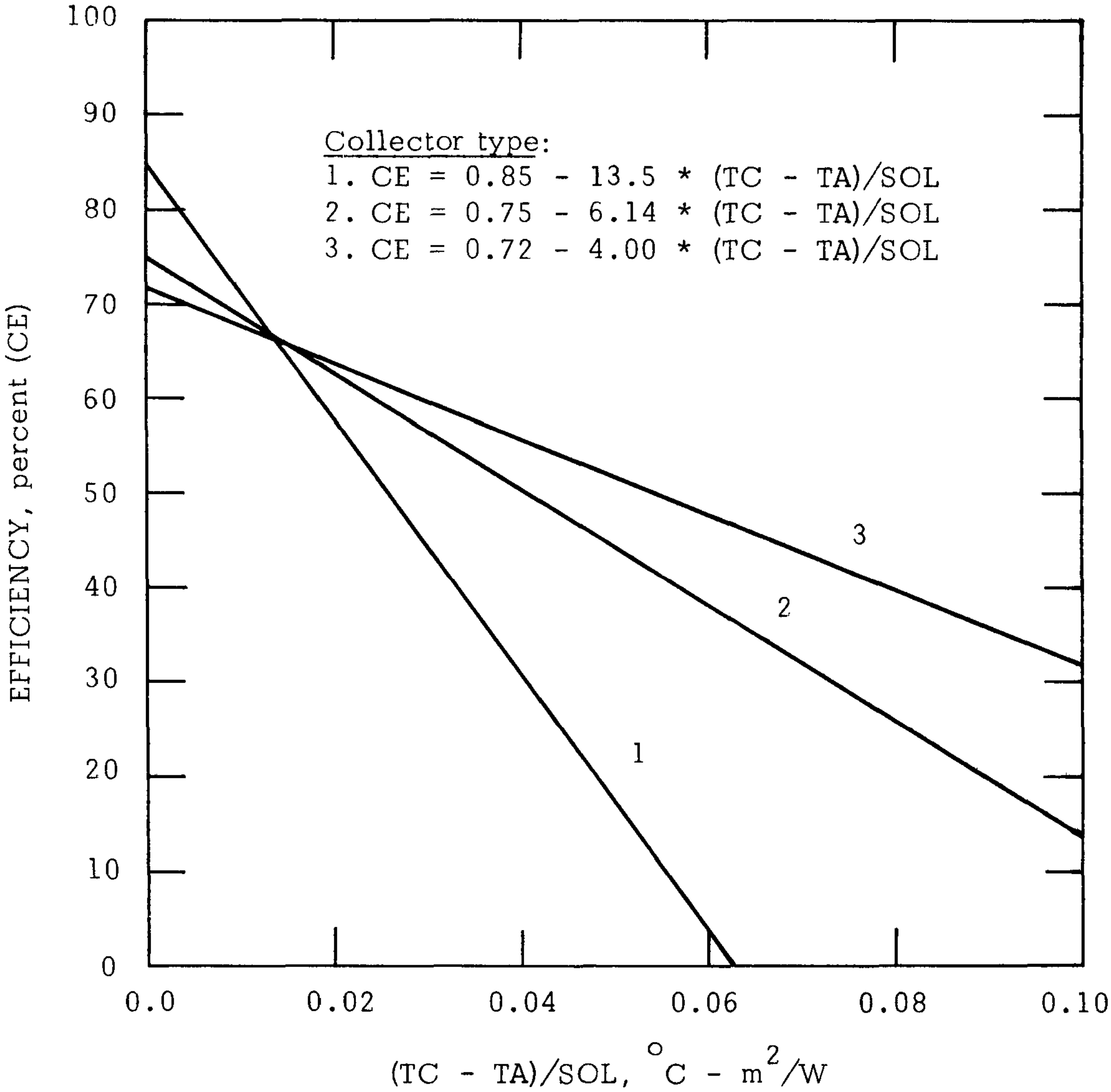
PROGRAM CHANGES FOR COLLECTOR DESIGN AND LOCATION

Collector Design

Three basic types of liquid flat-plate solar heat collectors were evaluated at each location. The efficiency curves for each type are shown in Figure 21. The specific variables in the program that were changed for each collector type are given in Table 12.

Location

Table 13 gives the computer program variables that were changed before running the program for each location. The locations were chosen to include all of the climatic regions of the TRW study[10] plus additional cities to cover all geographical regions of the United States. Temperature data were taken from World Climatic Data[9].



1. Solaris, trickle-type, single-glaze
2. Sunworks, tube-on-sheet, single-glaze
3. Sunworks, tube-on-sheet, double-glaze

Figure 21. Solar collector efficiency curves used in this study (Taken from published research or manufacturers' collector efficiency data)

Table 12. VARIABLES IN COMPUTER PROGRAM THAT CHANGE WITH COLLECTOR DESIGN

Collector type	Trade name	Design	CE a	CE b	C in PWS equation
1	Solaris	Trickle type, single-glaze	0.85	13.50	244
2	Sunworks	Tube-on-sheet, single-glaze	0.75	6.14	279
3	Sunworks	Tube-on-sheet, double-glaze	0.72	4.00	296

CE = collector efficiency = a - b * (TC - TA)/SOL
PWS = present worth of cost of solar heating system, $ = 9,950 + c * CA

Table 13. VARIABLES IN COMPUTER PROGRAM THAT CHANGE WITH LOCATION

Location number	City	State	AAAT °C	TAMPL °C	SEH a	SEH b	CLAT Deg.	CLAT Rad.
1	Miami	FL	23.9	4.6	211	56	25.8	0.451
2	Corpus Christi	TX	22.4	8.2	210	94	27.9	0.486
3	Dallas	TX	18.8	11.0	198	91	32.6	0.573
4	Phoenix	AZ	20.6	10.7	247	111	33.5	0.585
5	Los Angeles	CA	16.6	4.2	203	86	34.0	0.593
6	Albuquerque	NM	13.7	12.0	243	109	35.0	0.611
7	Nashville	TN	15.6	11.2	176	98	36.2	0.631
8	Annapolis	MD	13.1	11.7	173	98	39.0	0.681
9	Topeka	KS	12.5	14.7	174	94	39.0	0.681
10	Grand Junction	CO	11.4	14.4	223	120	39.2	0.684
11	Reno	NV	9.6	9.9	223	122	39.5	0.689
12	Salt Lake City	UT	10.5	13.2	186	115	40.9	0.713
13	Madison	WI	7.9	15.8	157	101	43.1	0.817
14	Portland	ME	7.2	12.9	170	102	43.6	0.762
15	Bismark	ND	5.7	17.9	179	119	46.8	0.817
16	Great Falls	MT	7.4	12.6	182	128	47.5	0.829
17	Seattle	WA	10.6	7.1	136	107	47.7	0.832
18	Fairbanks	AK	-3.6	20.3	124	121	64.9	1.132

AAAT = average annual air temperature
TAMPL = amplitude of annual air temperature variation
SEH = solar radiation on horizontal surface, W/m^2
(a = annual average, b = amplitude of variation throughout year)
CLAT = north latitude

SECTION 8

RESULTS

An example of the computer print-out data obtained for each location and collector design is given in Table 14. Each line of collector area, percent solar heat, and savings is printed after simulation of operation of the heating system for that particular collector area. The annual summary shows values of various temperatures, heat transfers, dates, costs, etc. for the optimum size system.

Graphs of savings versus effective collector area are plotted for the three collector types for two locations in Figures 22 and 23. At Phoenix, Arizona, the three collector types give almost the same maximum savings, whereas at Seattle, Washington, the difference is much greater. These two locations represent widely varying climatic conditions.

A summary of the results for all locations and collector types is given in Table 15. These results apply to the optimum-size solar heating system in each case.

The eighteen tested locations are listed in Table 16 in order of decreasing economic feasibility of solar digester heating. The most economical locations have both high solar radiation and large heat demand. A good correlation was found between savings and average annual solar radiation multiplied by difference between digester temperature ($35^{o}C$) and average annual air temperature (Figure 18).

Table 14. EXAMPLE OF COMPUTER PRINTOUT DATA OBTAINED FOR EACH LOCATION AND COLLECTOR TYPE (Annapolis, Maryland, Collector type # 3)

Collector area, m^2	Percent solar heat	Savings, $
20	11.6	500
40	21.0	7,900
60	30.3	15,200
80	39.6	22,400
100	48.5	29,000
120	57.1	35,200
140	65.3	41,000
160	73.2	46,200
180	80.2	50,200
200	85.8	52,200
220	90.2	52,500
240	93.7	51,500
260	96.6	49,700
280	98.8	46,900
300	99.9	42,600
320	99.9	36,700
340	99.9	30,700
360	99.9	24,800
380	99.9	18,900
400	99.9	13,000

Annual Summary for Most Economical Size System

CAO	= 220 m^2	CITA	= 1,500 GJ
PSOLO	= 90.2%	HSITO	= 787 GJ
VO	= 44.0 m^3	HSOTO	= 787 GJ
TMAXO	= 46.02°C	AUXTA	= 82.5 GJ
TYMAXO	= 216° (October 27th)	K	= 365
TMINO	= 40.68°C	PWSO	= $75,100
TYMINO	= 0° (March 21st)	PWGCO	= $127,600
TOP	= 40.74°C	SAVO	= $52,500
SHDTO	= 844 GJ	SOL	= 842 W/m^2
SHATO	= 761 GJ	DAYL	= 43,040s (11.96 hours)
HLTO	= 25.8 GJ		

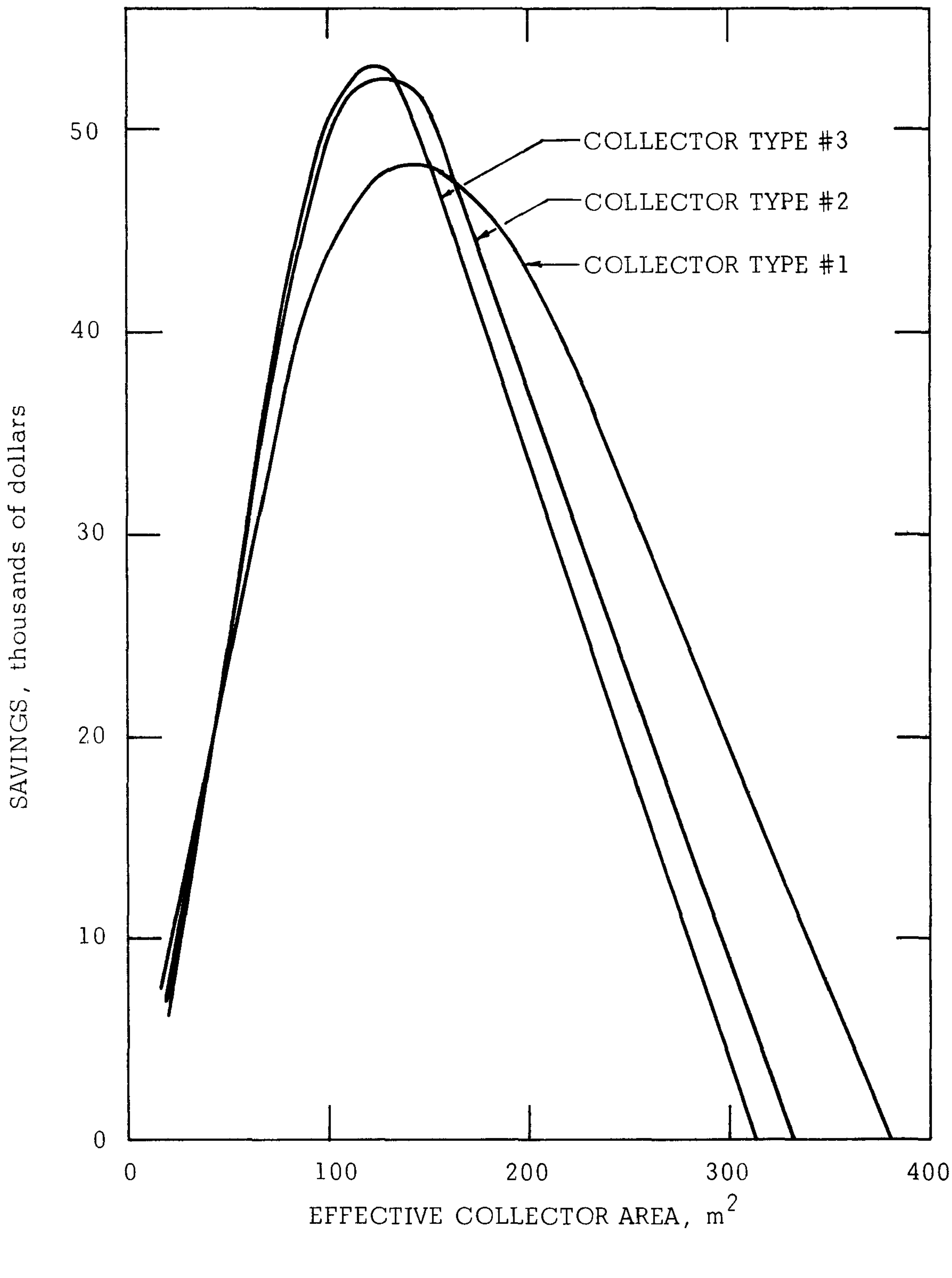


Figure 22. Savings versus collector area for three collector types, Phoenix, Arizona

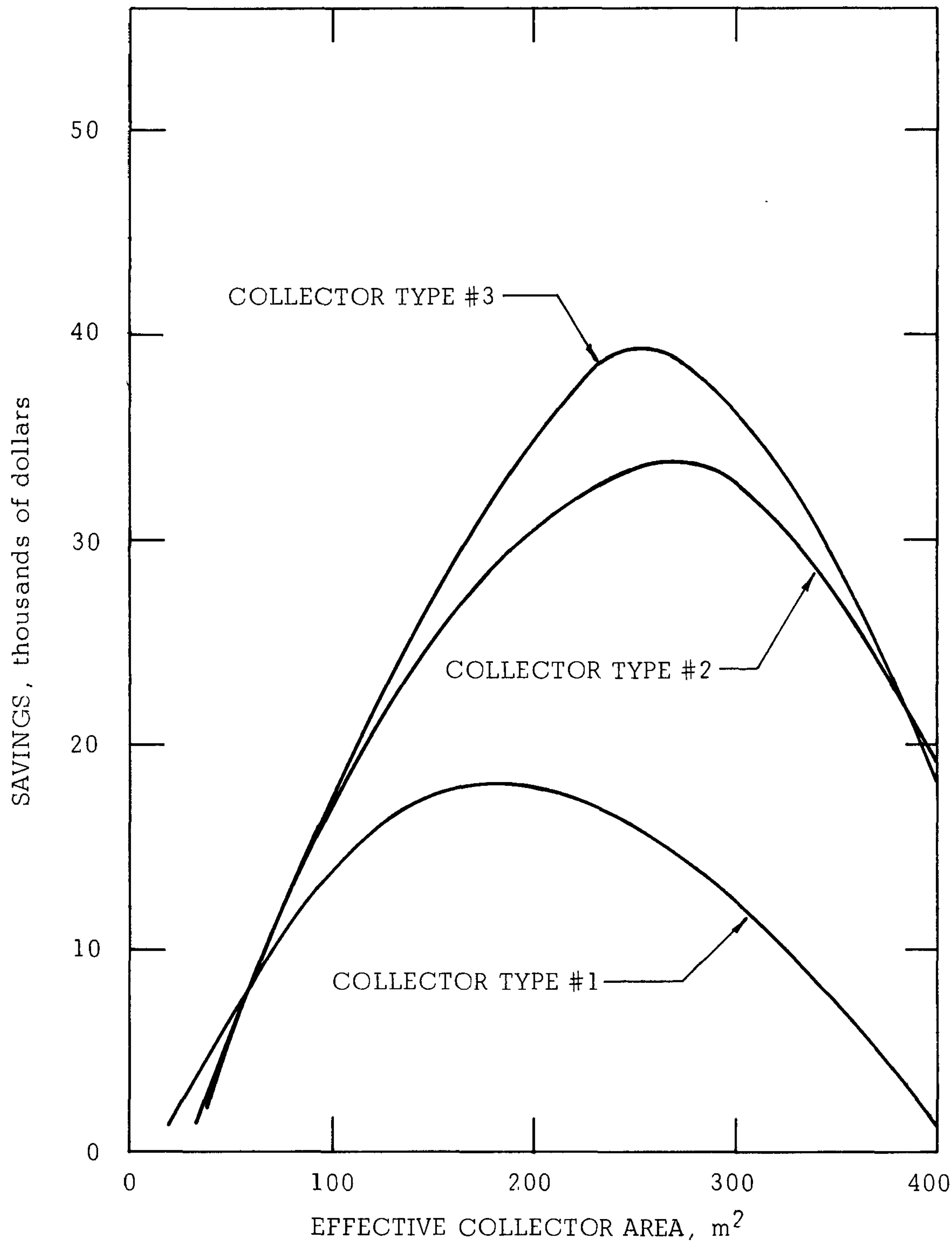


Figure 23. Savings versus collector area for three collector types, Seattle, Washington

Table 15. SUMMARY OF COMPUTER RESULTS

Location number	City	State	Collector type	PSOLO %	CAO m^2	PWSO $	SAVO $
1	Miami	FL	1	92.9	140	44,100	36,000
			2	95.2	120	43,400	38,600
			3	96.9	120	45,500	38,100
2	Corpus Christi	TX	1	81.1	140	44,100	32,000
			2	91.5	140	49,000	36,800
			3	94.5	140	51,400	37,300
3	Dallas	TX	1	80.6	180	53,900	36,600
			2	88.6	160	54,600	44,800
			3	92.3	160	57,300	46,300
4	Phoenix	AR	1	89.7	140	44,100	48,300
			2	93.1	120	43,400	52,500
			3	95.6	120	45,500	53,100
5	Los Angeles	CA	1	90.3	220	63,600	47,900
			2	96.0	180	60,200	58,400
			3	94.8	160	57,300	59,800
6	Albuquerque	NM	1	88.8	200	58,800	64,100
			2	96.9	180	60,200	73,900
			3	96.0	160	57,300	75,600
7	Nashville	TN	1	69.8	200	58,800	31,000
			2	84.1	200	65,800	42,400
			3	88.6	200	69,200	44,800
8	Annapolis	MD	1	69.1	220	63,600	34,100
			2	85 3	220	71,300	49,300
			3	90.2	220	75,100	52,500
9	Topeka	KS	1	68.9	220	63,600	35,900
			2	84.5	220	71,300	50,700
			3	89.6	220	75,100	54,300
10	Grand Junction	CO	1	82.7	220	63,600	60,500
			2	93.1	200	65,800	74,000
			3	97.0	200	69,200	76,400
11	Reno	NV	1	87.2	260	73,400	65,600
			2	96.2	220	71,300	81,900
			3	96.6	200	69,200	84,700
12	Salt Lake City	UT	1	68.7	220	63,600	42,700
			2	84.2	220	71,300	58,900
			3	89.0	220	75,100	62,600
13	Madison	WI	1	55.4	220	63,600	29,500
			2	77.8	260	82,500	48,200
			3	83.8	260	86,900	53,900
14	Portland	ME	1	63.7	240	68,500	40,700

Table 15 (continued).

Location number	City	State	Collector type	PSOLO %	CAO m^2	PWSO $	SAVO $
			2	89.1	280	88,100	64,700
			3	91.7	260	86,900	70,500
15	Bismark	ND	1	69.7	260	73,400	51,500
			2	86.5	260	82,500	72,500
			3	88.5	240	81,000	77,600
16	Great Falls	MT	1	73.7	260	73,400	52,400
			2	87.6	240	76,900	72,500
			3	92.3	240	81,000	76,400
17	Seattle	WA	1	46.7	180	53,900	18,200
			2	75.5	260	82,500	33,900
			3	81.8	260	86,900	39,200
18	Fairbanks	AK	1	72.8	340	92,900	72,300
			2	85.9	300	93,700	101,100
			3	91.1	300	98,800	107,900

Collector type:

1. Solaris, trickle type, single-glaze
2. Sunworks, tube-on-sheet, single-glaze
3. Sunworks, tube-on-sheet, double-glaze

PSOLO = percent solar heat at optimum size solar heating system
CAO = collector area for optimum size solar heating system
PWSO = present worth of cost of optimum size solar heating system
SAVO = present worth of gas conserved less present worth of cost of solar heat, for optimum size solar heating system (greatest savings).

Table 16. LIST OF EIGHTEEN TESTED LOCATIONS IN ORDER OF DECREASING ECONOMIC FEASIBILITY OF SOLAR-HEATED DIGESTER

Location		SAVO,	Location		SAVO,
City	State	$	City	State	$
Fairbanks	AK	107,900	Topeka	KS	54,300
Reno	NV	84,700	Madison	WI	53,900
Bismark	ND	77,600	Phoenix	AR	53,100
Great Falls	MT	76,400	Annapolis	MD	52,500
Grand Junction	CO	76,400	Dallas	TX	46,300
Albuquerque	NM	75,600	Nashville	TN	44,800
Portland	ME	70,500	Seattle	WA	39,200
Salt Lake City	UT	62,600	Miami	FL	38,100
Los Angeles	CA	59,800	Corpus Christi	TX	37,300

SAVO = present worth of gas conserved less present worth of cost of optimum-size solar heating system, over 25-year project life

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APPENDIX A

COMPUTER PROGRAMS

Definition of Symbols, SOL 4

M	=	counter (M = 1 without reflector, M = 2 with reflector)
P	=	period = 86,400s (1 day), calculation interval
XK	=	thermal conductivity of storage tank insulation, assuming 4" thick fiberglass, = 0.353 W/m^2C
HC	=	heat capacity of water and sludge = 4.19 x 10^6 J/m^3C
SFR	=	sludge flow rate, m^3/s
T	=	temperature (vertical average) of water in solar heat storage tank, ^oC
R	=	ratio of storage volume, m^3, to collector area, m^2
CA	=	collector area, m^2
V	=	volume of solar heat storage tank, m^3
N	=	counter, year number of iterations (values of heat inputs, etc. are recorded only for N = 2)
K	=	counter, day number, should equal 365 at end of year
TY	=	time of year, radians, varies from 0 on March 21st to 2 π one year later
TMAX	=	maximum temperature reached in heat storage tank during year, ^oC
TYMAX	=	time of year of occurrence of maximum temperature in heat storage tank, degrees (March 21 = 0, April 21 = 30, etc.)
TMIN	=	minimum temperature reached in heat storage during year, ^oC

TYMIN	=	time of year (degrees) of occurrence of minimum temperature in heat storage tank
SHDT	=	sludge heat desired total = total annual heat input to digestr, J
SHAT	=	sludge heat actual total = total annual heat actually transferred from solar heat storage tank to raw sludge, J
HLT	=	heat loss total = total annual heat loss from heat storage tank to surrounding air, J
HSOT	=	heat storage output total = total annual heat output from heat storage tank, J
CIT	=	collector input total = total annual solar radiation incident on collector, J
AUXT	=	auxiliary heat total = total annual auxiliary heat used, J
HSIT	=	heat storage input total = total annual heat input to solar heat storage tank, J
SLIT	=	sludge inlet temperature = temperature of raw, cold sludge as it enters solar preheater, ^oC
SHD	=	sludge heat desired = required heat input to digester during period P, J
SLOTD	=	sludge outlet temperature desired - temperature of raw sludge as it exits solar preheater so that no auxiliary heat is needed, ^oC
TAM	=	temperature ambient mean = average daily outdoor temperature, ^oC
A	=	area = surface area of heat storage tank, m^2
HL	=	heat loss = heat loss from storage tank to surrounding air during period P, J
SHA	=	sludge heat actual = actual heat transferred from solar heat storage tank to raw, cold sludge during period P, J
AUX	=	auxiliary heat input to digester during period P, J
HSO	=	heat storage output = heat output of storage tank during period P, J
TI	=	temperature of heat storage after heat is removed, but before heat is added, ^oC (for system simulation purposes only)

F	=	factor = empirical factor relating horizontal extraterrestrial and terrestrial solar radiation, dimensionless
SEXH	=	solar energy extraterrestrial horizontal = average daily extraterrestrial solar energy received on a horizontal surface, W/m^2 (Annapolis, Maryland)
PERP	=	angle of collector perpendicular to sun's rays at solar noon, radians
DA	=	deviation angle = absolute value of difference between 60^o collector angle and angle of collector perpendicular to the sun's rays at solar noon, radians
CI	=	collector input = total radiation incident on collector during perido P, J
REFL	=	reflector = factor representing increased incident radiation on collector due to reflector, dimensionless
TA	=	ambient temperature during collection of solar energy, ^oC
TC	=	average temperature of collector during collection of solar energy, ^oC
SOL	=	solar insolation during collection, W/m^2
CE	=	collection efficiency of collectors, dimensionless
HSI	=	heat storage input = heat input to storage tank during period P, J
PSOL	=	percent solar = total annual solar heat actually transferred to sludge divided by total annual heat required by digester, dimensionless

Additional Symbols Used in SOL 5

J	=	counter that triggers printout every 5 days, dimensionless

SHD5 HL5 CI5 HSI5 SHA5 AUX5 HSO5	heat transfer rates, W, printed every 5 days
PERP5 DA5	angles, degrees, printed every 5 days

?LIST SOL4

```
    M = 1
    P = 86400.
    XK = 0.353
    HC = 4190000.
    SFR = 0.000231
    T = 35.
    R = 0.20
10  CA = 50.
15  V = CA*R
    N = 1
20  K =1
    TY = 0.
    TMAX = 0.
    TYMAX = 500.
    TMIN = 150.
    TYMIN = 500.
    SHDT = 0.
    SHAT = 0.
    HLT = 0.
    HSOT = 0.
    CIT = 0.
    AUXT = 0.
    HSIT = 0.
25  SLIT = 19. + 3.*SIN(TY-.5236)
    SHD = (25900. + 5700. * SIN(TY-3.665))*P
    SLOTD = SLIT + SHD/(SFR*HC*P)
    TAM = 13.1 + 11.7 * SIN(TY-0.5236)
    A = 6.2*V**0.6667
    HL = XK*A*(T-TAM)*P
    IF (T.LT.SLIT) GO TO 30
    SHA = (T-SLIT)*HC*SFR*P
    IF(SHA.LT.SHD) GO TO 40
    SHA = SHD
    GO TO 40
30  SHA   = 0.
40  AUX = SHD-SHA
    HSO = SHA + HL
    TI = T - HSO/(HC*V)
    F = 0.725 + 0.175*SIN(TY-3.1416)
    SEXH = 318. + 160.*SIN(TY)
    PERP = 0.6807 - 0.4102*SIN(TY)
    DA = ABS(1.047-PERP)
    CI = F*SEXH*COS(DA)*P*CA
    IF (M.EQ.2) GO TO 42
    GO TO 45
42  IF (TY.LT.3.1416) GO TO 45
```

SOL 4, continued

```
      REFL = 1.0 + 0.36*SIN(TY-3.1416)
      CI = CI * REFL
   45 TA = 18. + 11.7 * SIN(TY-0.5236)
      TC = T + 3.0
      SOL = CI * 4.0/(P*CA)
      CE = 0.73 - 4.31 * (TC-TA)/SOL
      HSI = CI*CE
      T = TI + HSI/(HC*V)
      IF (T.LT.SLOTD) GO TO 50
      T = SLOTD
      HSI = (SLOTD-TI)*HC*V
   50 SHDT = SHDT + SHD
      SHAT = SHAT + SHA
      HLT = HLT + HL
      HSOT = HSOT + HSO
      CIT = CIT + CI
      AUXT = AUXT + AUX
      HSIT = HSIT + HSI
      IF (T.LT.TMAX) GO TO 60
      TMAX = T
      TYMAX = TY*57.2958
   60 IF (T.GT.TMIN) GO TO 70
      TMIN = T
      TYMIN = TY*57.2958
   70 IF(TY.GT.6.2574) GO TO 80
      TY = TY + 0.0172142
      K = K + 1
      GO TO 25
   80 IF(N.EQ.2) GO TO 90
      N = N + 1
      GO TO 20
   90 PSOL = SHAT/SHDT
      WRITE (9,92) CA,PSOL,V,TMAX,TYMAX
      WRITE (9,92) TMIN, TYMIN, T,SHDT, SHAT
      WRITE (9,92) HLT, CIT, HSIT, HSOT, AUXT, K,M
   92 FORMAT (5E11.4,2I6)
      IF(CA.GT.580.0) GO TO 95
      CA = CA + 50.
      GO TO 15
   95 IF(M.EQ.2) GO TO 97
      M = M + 1
      GO TO 10
   97 STOP
      END
```

?LIST SOL5

```
      M = 1
      P = 86400.
      XK = 0.353
      HC = 4190000.
      SFR = 0.000231
      T = 35.
      R = 0.20
   10 CA = 230.
   15 V = 75.
      N = 1
   20 K =1
      J = 1
      TY = 0.
      TMAX = 0.
      TYMAX = 500.
      TMIN = 150.
      TYMIN = 500.
      SHDT = 0.
      SHAT = 0.
      HLT = 0.
      HSOT = 0.
      CIT = 0.
      AUXT = 0.
      HSIT = 0.
   25 SLIT = 19. + 3.*SIN(TY-.5236)
      SHD = (25900. + 5700. * SIN(TY-3.665))*P
      SLOTD = SLIT + SHD/(SFR*HC*P)
      TAM = 13.1 + 11.7 * SIN(TY-0.5236)
      A = 6.2*V**0.6667
      HL = XK*A*(T-TAM)*P
      IF (T.LT.SLIT) GO TO 30
      SHA = (T-SLIT)*HC*SFR*P
      IF(SHA.LT.SHD) GO TO 40
      SHA = SHD
      GO TO 40
   30 SHA   = 0.
   40 AUX = SHD-SHA
      HSO = SHA + HL
      T1 = T - HSO/(HC*V)
      F = 0.725 + 0.175*SIN(TY-3.1416)
      SEXH = 318. + 160.*SIN(TY)
      PERP = 0.6807 - 0.4102*SIN(TY)
      DA = ABS(1.047-PERP)
      CI = F*SEXH*COS(DA)*P*CA
      IF (M.EQ.2) GO TO 42
```

SCL 5, continued

```
      REFL = 100.
      GO TO 45
   42 IF (TY.LT.3.1416) GO TO 45
      REFL = 1.0 + 0.36*SIN(TY-3.1416)
      CI = CI * REFL
   45 TA = 18. + 11.7 * SIN(TY-0.5236)
      TC = T + 3.0
      SOL = CI * 4.0/(P*CA)
      CE = 0.72 -4.00*(TC-TA)/SOL
      HSI = CI*CE
      T = T1 + HSI/(HC*V)
      IF (T.LT.SLOTD) GO TO 50
      T = SLOTD
      HSI = (SLOTD-T1)*HC*V
   50 SHDT = SHDT + SHD
      SHAT = SHAT + SHA
      HLT = HLT + HL
      HSOT = HSOT + HSO
      CIT = CIT + CI
      AUXT = AUXT + AUX
      HSIT = HSIT + HSI
      IF (T.LT.TMAX) GO TO 60
      TMAX = T
      TYMAX = TY*57.2958
   60 IF (T.GT.TMIN) GO TO 62
      TMIN = T
      TYMIN = TY*57.2958
   62 IF (N.EQ.2) GO TO 63
      GO TO 70
   63 IF (J.EQ.5) GO TO 65
      GO TO 70
   65 SHD5 = SHD/P
      HL5 = HL/P
      CI5 = CI/P
      HSI5  = HSI/P
      SHA5 = SHA/P
      AUX5 = AUX/P
      HSO5 = HSO/P
      PERP5 = PERP*57.3
      DA5 = DA*57.3
      WRITE (9,67) SHD5,HL5,SHA5,AUX5,T,K,M
      WRITE (9,67) HSO5,CI5,HSI5,F,CE
      WRITE (9,67) SEXH,SOL,PERP5,DA5,REFL
      WRITE (9,67) SLOTD, SLIT, TAM, TA, TC
      J = 0
   67 FORMAT (5F11.4, 2I6)
   70 IF(TY.GT.6.2574) GO TO 80
      TY = TY + 0.0172142
```

SCL 5, continued

```
    K = K + 1
    J = J + 1
    GO TO 25
80  IF(N.EQ.2) GO TO 90
    N = N + 1
    GO TO 20
90  PSOL = SHAT/SHDT
    WRITE (9,92) CA,PSOL,V,TMAX,TYMAX
    WRITE (9,92) TMIN, TYMIN, T,SHDT, SHAT
    WRITE (9,92) HLT, CIT, HSIT, HSOT, AUXT, K,M
92  FORMAT (5E11.4,2I6)
95  IF(M.EQ.2) GO TO 97
    M = M + 1
    GO TO 10
97  STOP
    END
```

APPENDIX B

Computer Program "SOL 6" Symbols

The following list of symbols contains only those used in computer program "SOL 6" that did not appear in the previous computer program "SOL 4" of the Annapolis study. The number given is the computer program line number in which the symbol first appears.

160 SAVO	=	Present worth of gas conserved less percent worth of cost of solar heating system, for optimum-size solar heating system, $
330 AAAT	=	Average annual air temperature, ^oC
340 TAMPL	=	Amplitude of variation of air temperature throughout the year, ^oC
350 SLITM	=	Average annual raw sludge temperature, ^oC
360 SLITAM	=	Amplitude of variation of raw sludge temperature through-year, ^oC
380 HLD	=	Rate of heat loss from digester to surrounding ground or air, W
390 SLTH	=	Rate of heat input to raise temperature of raw sludge to 35^oC, W
540 SEH	=	Solar energy received on a horizontal surface, W/m^2
550 CLAT	=	Latitude, rad
630 D	=	Earth's declination angle from sun-earth plane, rad
640 X	=	Argument of arcos (arc cosine) in daylight equation
650 Y	=	Subroutine for arcos
660 ANG	=	Subroutine for ARCOS "X"
670 DAYL	=	Time between sunrise and sunset, s
690 Q	=	Water flow rate through collectors per unit collector area, $m^3/s \cdot m^2$

990 PWS	=	Present worth of cost of solar heating system, $
1000 EFF	=	Efficiency of combustion of methane, dimensionless
1010 PPRICE	=	Present price of methane based on low heating value, $/J
1020 FAC	=	Factor which, when multiplied by present annual cost of gas, will give the total present worth of gas heating costs over the 25-year period, dimensionless
1030 PWGC	=	Present worth of gas conserved, $
1040 SAVING	=	Present worth of gas conserved less present worth of solar heating system, $

1080 PSOLO
1090 CAO
1100 VO
1110 TMAXO
1120 TYMAXO
1130 TMINO
1140 TYMINO
1150 TOP
1160 SHDTO
1170 SHATO
1180 HLTO
1190 CITA
1200 HSITO
1210 HSOTO
1220 AUXTA
1230 PWSO
1240 PWGCO

Line numbers 1080 through 1240:
These symbols ending in "O" (or in some cases "A" or "OP") have the same meaning as corresponding, in the Annapolis study, parameters without the suffic; these symbols, however, refer to the value for the optimum-size solar heating system.

?LIST SOL6

```
       P = 86400.
       XK = 0.353
       HC = 4190000.
       SFR = 0.000231
       T = 35.
       R = 0.20
       SAVO = 0.
    10 CA = 20.
    15 V = CA*R
       N = 1
    20 K =1
       TY = 0.
       TMAX = 0.
       TYMAX = 500.
       TMIN = 150.
       TYMIN = 500.
       SHDT = 0.
       SHAT = 0.
       HLT = 0.
       HSOT = 0.
       CIT = 0.
       AUXT = 0.
       HSIT = 0.
       AAAT = 12.5
       TAMPL = 14.7
       SLITM = AAAT + 5.0
       SLITAM = TAMPL/3.0
    25 SLIT = SLITM + SLITAM*SIN(TY - 0.5236)
       HLD = 10400. + 2800.*SIN(TY - 3.665)
       SLTH = SFR * HC * (35.0 - SLIT)
       SHD = (HLD + SLTH)*P
       SLOTD = SLIT + SHD/(SFR*HC*P)
       TAM = AAAT + TAMPL*SIN(TY - 0.5236)
       A = 6.2*V**0.6667
       HL = XK*A*(T-TAM)*P
       IF (T.LT.SLIT) GO TO 30
       SHA = (T-SLIT)*HC*SFR*P
       IF(SHA.LT.SHD) GO TO 40
       SHA = SHD
       GO TO 40
    30 SHA   = 0.
    40 AUX = SHD-SHA
       HSO = SHA + HL
       TI = T - HSO/(HC*V)
       SEH = 174. + 94.*SIN(TY)
       CLAT = 0.681
```

```
      PERP = CLAT - 0.4102*SIN(TY)
      DA = ABS(0.3491 + 0.4102*SIN(TY))
      CI = SEH*COS(DA)*P*CA/COS(PERP)
   42 IF (TY.LT.3.1416) GO TO 45
      REFL = 1.0 + 0.36*SIN(TY-3.1416)
      CI = CI * REFL
   45 TA = TAM + 5.0
      D = 0.410*COS(TY-1.571)
      X = (-SIN(CLAT)/COS(CLAT)) * (SIN(D)/COS(D))
      Y = SQRT(1.0 - X*X)
      ANG = ATAN2(Y,X)
      DAYL = 27500. * ANG
      SOL = 2.0*CI/(DAYL * CA)
      Q = 0.000010
      TC = T + SOL/(HC*Q*2.0)
      CE = 0.75 - 6.14*(TC-TA)/SOL
      HSI = CI*CE
      T = TI + HSI/(HC*V)
      IF (T.LT.SLOTD) GO TO 50
      T = SLOTD
      HSI = (SLOTD-TI)*HC*V
   50 SHDT = SHDT + SHD
      SHAT = SHAT + SHA
      HLT = HLT + HL
      HSOT = HSOT + HSO
      CIT = CIT + CI
      AUXT = AUXT + AUX
      HSIT = HSIT + HSI
      IF (T.LT.TMAX) GO TO 60
      TMAX = T
      TYMAX = TY*57.2958
   60 IF (T.GT.TMIN) GO TO 70
      TMIN = T
      TYMIN = TY*57.2958
   70 IF(TY.GT.6.2574) GO TO 80
      TY = TY + 0.0172142
      K = K + 1
      GO TO 25
   80 IF(N.EQ.2) GO TO 90
      N = N + 1
      GO TO 20
   90 PSOL = SHAT/SHDT
   91 FORMAT(7H  CA = ,F5.0,9H  PSOL = ,F5.3,11H
      SAVING = ,E11.4)
      PWS = 9950. + 279.*CA
      EFF = 0.66
      PPRICE = 0.000000002
      FAC = 55.3
      PWGC = (SHDT/EFF)*PPRICE*FAC*PSOL
```

SOL 6, continued

```
      SAVING = PWGC - PWS
      WRITE(9,91)CA,PSOL,SAVING
      IF (SAVING.LT.SAVO) GO TO 94
      SAVO = SAVING
      PSOLO = PSOL
      CAO = CA
      VO = V
      TMAXO = TMAX
      TYMAXO = TYMAX
      TMINO = TMIN
      TYMINO = TYMIN
      TOP = T
      SHDTO = SHDT
      SHATO = SHAT
      HLTO = HLT
      CITA = CIT
      HSITO = HSIT
      HSOTO = HSOT
      AUXTA = AUXT
      PWSO = PWS
      PWGCO = PWGC
   94 IF(CA.GT.390.) GO TO 95
      CA = CA + 20.
      GO TO 15
   95 WRITE (9,96) CAO, PSOLO, VO, TMAXO, TYMAXO
      WRITE(9,96) TMINO,TYMINO,TOP,SHDTO,SHATO
      WRITE(9,96)HLTO,CITA,HSITO,HSOTO,AUXTA,K
      WRITE(9,96)PWSO,PWGCO,SAVO,SOL,DAYL
   96 FORMAT(5E11.4,2I6)
      STOP
      END
```

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(Please read Instructions on the reverse before completing)

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15. SUPPLEMENTARY NOTES

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16. ABSTRACT

Two distinct, yet related studies were conducted to determine the technical and economic feasibility of using solar energy as the source of heat for the anaerobic digestion process. Retrofitting a solar energy collection and heat transfer system to a digester at Annapolis, Maryland was proven feasible in the first part of the study and the concept of using solar energy for digester heating throughout the United States, including Fairbanks, Alaska, was shown to be economically feasible in the second part of the study.

The Part I study compared five (5) types of flat plate collectors and selected the cost effective design to supply approximately 90 percent of the heat load to maintain digester operating temperatures of 32 to 38 degrees Celsius. Three flat plate collectors of varying efficiencies were evaluated for use at numerous locations in the United States. The study showed that optimum-sized flat plate collectors can provide from 82 to 97 percent of the total annual digester heat, the higher percentages being applicable to areas of higher solar radiation.

The Part II study developed specific guidelines for determining the optimum size and conceptual design for a solar heating system for any size sludge digester at any location.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Solar Heating Sludge Digestion Solar Energy Waste Treatment Economic Analysis	Solar System Engineering Solar Heated Anaerobic Digesters Solar Energy Economy Design Guidelines	13B

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