PACIFIC REGIONAL SOLAR HEATING HANDBOOK

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Informal Report

los alamos
scientific laboratory
of the University of California

ERRATA

Pg. 9 - Last sentence:"---air system and can be used without modification."

Pgs. 21-22 - are reversed

Pg. 23 - Footnote should read: "The load of an existing building can be estimated---"

Pg. 29 - The left arrow in the collector loop of the two-tank system diagram is inverted.

Pg. 34 - Bottom caption should read: Forced Air Distribution.

Pgs. 42,60,105 - The time scale on these graphs is in error by two months. It should read July 1957 to June 1958, as on pages 58 and 104.

Pg. 43 - BTU/Mo./ft_c^2 is for a surface tilted 46.8°.

Pg. 52 - Change Thermia 15 to Thermia 33.
Change Viscosity 28.5 to 121

Pg. 71 - 48 BTU/DD ft_c^2 should read 96 BTU/DD ft_c^2.

Pg. 126 - The last expression should read:
.083 BTU/hr°F ft^2. (no subscript c)

Pg. 152 - Last sentence:"---reflects visible radiation and radiates heat in the infrared."

Pg. 167 - 2nd to last sentence:"Summer sun" should be "winter sun."

Pg. 170 - Pacific Time Zone should be 120°W

Pg. 171 - "Solar Atitude" should be "Solar Altitude."

If additional serious errors are found, the authors would appreciate being notified about them.

This Handbook prepared for ERDA,
San Francisco Operations Office,

by

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PACIFIC REGIONAL SOLAR HEATING HANDBOOK

ABSTRACT

This handbook is a guide for engineers, architects, and individuals familiar with heating and ventilating applications, who wish to design a solar heating system for a residential or small commercial building in the Pacific Coast Region. Both air and liquid space heating systems and domestic hot water heating applications are presented in detail. Passive space heating and swimming pool heating systems are discussed qualitatively; a quantitative analysis is not available at this time. Sources of information are listed, and design considerations, collector types, and collector coolants are discussed. The active systems described and studied are as follows: 1) domestic hot water heating using liquid heating collectors; 2) space heating using liquid heating collectors, water heat storage, and forced air distribution; and 3) space heating using air heating collectors, rock-bed heat storage, and forced air distribution. Performance charts are presented for the three types of active solar heating systems. The analyses are based on hour-by-hour computer simulations using actual weather data from seven Pacific Regional cities. The required collector area has been determined for these cities in the four-state region, Arizona, California, Oregon, and Washington. A simplified approach to collector sizing is

presented based on monthly solar radiation and temperature data. The effects of variations in important design parameters have been studied extensively for Fresno, California, and are presented in graphical form. However, much of the information in the Handbook is generally applicable.

CONTENTS

INTRODUCTION

The solar energy incident on the outside of a building can be used to provide a major fraction of the domestic hot water and space heating requirements of the building in most of the continental United States. An "active" solar heating system generally consists of solar collectors to absorb the sun's heat energy, a separate heat storage medium to hold excess heat for release during periods when the sun does not shine, and a heat transfer loop to transport heat from the collectors to the storage medium. "Active" systems tend to be relatively expensive - a home installation may cost from $4,000 to $10,000.

Heating a building by the "passive" use of solar energy is an important aspect of solar utilization. Solar gains through windows, walls, modified walls, skylights, clerestory windows, or roof sections provide an opportunity to dramatically reduce the total heating energy requirements of a building. Thermal storage in building materials is frequently employed. Many such passive solar heating elements have been used and are available to a designer. These represent a large number of possible building designs. Passive cooling utilizing evaporation and night sky radiation can also reduce energy consumption.

Although the operation of a solar heating system can be readily understood in a qualitative fashion, the quantitative analysis of a system (e.g., sizing of collector array) involves computer simulation of solar performance using actual weather data, and is considerably more difficult.

This Handbook is intended to supply this type of quantitative analysis.

Since the weather and solar insolation pattern varies significantly from place to place, it is desirable to base the design of a solar heating system on the local situation. The weather pattern varies widely throughout the four states covered by this handbook and a fairly general method is given to cover the range of conditions encountered. Since the weather varies from year to year, one must expect some deviation in the overall performance from year to year.

Solar heating is a technology which is considered to be available for general use. A growing number of companies are marketing components and whole systems. As fuel prices rise the extra cost associated with the installation of a solar heating system becomes less prohobitive. The growing market, mass production of components, and experience in installation of systems can be expected to bring costs down somewhat as solar heating becomes widespread.

Solar energy can also be used in an active system for cooling a building, to provide process steam heat, or for electrical power generation. These technologies are not yet considered to be economical and are not discussed in this handbook.

PURPOSE OF THIS HANDBOOK

This Handbook is designed for an individual interested in the installation of a solar heating system, and is intended to aid in the proper choice of collector size, storage size and other important design parameters to meet the requirements imposed by the individual. The Handbook is intended to be used primarily by an architect, contractor, or heating, ventilating, and air-conditioning consulting engineer. It may be of use to the "do-it-yourself" owner/builder with a sufficient technical background.

The Handbook is not intended to overlap coverage of other sources of design information such as details of collector design, conventional heating system design or building thermal load analysis. References to some of these sources of information are given on pages 16 - 20. Although many texts and other "how to" books have been written on solar heating, they are generally oriented toward the hardware used in the system. Few, if any, provide a quantitative description of the thermal interaction of the components. This is because the analysis is relatively complicated and is highly dependent on the weather pattern in the locality where the system is to be used. The problem can be readily solved on a digital computer if hourly records of solar radiation and weather are known. This is hardly a practical approach for the individual builder.

By using the charts and simplified method described in this Handbook, the designer can size the system components without an individualized computer simulation.

GEOGRAPHICAL AREA COVERED BY THIS HANDBOOK

The geographical area covered by this Solar Heating Handbook comprises the four state region of Washington, Oregon, California, and Arizona. Arizona is nominally included in the "West Coast" region because its climate is quite similar to parts of southern California, and because it contains large population centers for which hour-by-hour weather data are available. Vancouver, BC is included as a city for Washington state, because weather data is available.

The seven cities which have been studied in detail are the only ones in the Pacific region for which hourly records of solar radiation and weather data are available at LASL.

The seven cities so investigated are: Phoenix, AZ; Los Angeles, Ca; Santa Maria, CA; Fresno, CA; Medford, OR; Seattle, WA; and Vancouver, BC.

GEOGRAPHICAL AREA COVERED
BY THIS HANDBOOK

HOW TO USE THIS HANDBOOK

One should first scan through this Handbook and become generally familiar with the contents.

In order to use the information to choose the size of the solar collector array, it is first necessary to determine the building thermal load (page 22). The size of a "standard" system, either of the liquid or air type, can then be determined for any one of the seven cities listed simply by using the appropriate design chart (page 64 or 65). For other locations, the "simplified method" can be used to obtain a reasonable estimate of the required collector array size. To use the simplified method, it will be necessary to obtain monthly average data for solar radiation and for heating degree-days. The heating degree-day information can frequently be obtained from the Weather Service. Solar radiation data are not so widely available and may have to be estimated from data for a neighboring location with a similar climate or by some other estimating process such as observed sunshine hours. The simplified method is explained starting on page 140.

Each case will deviate somewhat from the "standard" system designs which have been chosen for the above estimates. The effects of departures from the chosen parameters can be estimated by studying the many graphs. The effect of changes in collector design were calculated for the liquid system but are similar for the air system to be used without modification.

The effect of parameter changes have been studied only for Fresno, Ca. The magnitude of the effect would be different for different localities and the results for Fresno should be used only as a guide to trends.

NOTES OF CAUTION

The apparent simplicity and grass-roots appeal of solar heating can trap the unwary into a poor installation. Many who would not have considered designing their own home heating system blithely launch into a solar heating installation without adequate preparation. A basic working knowledge of the conventional heating system to be employed with the solar heat supply system is essential if a good design is to result. Many books are available to the individual wishing to design his own system. Alternatively, contractors and competent heating, ventilating, and air-conditioning consulting engineers are available in any city. They are trained and licensed in design of conventional systems and are usually quite willing to consult on the design of a solar heating installation.

Since solar heating is a relatively new commercial field, certification procedures for solar equipment have not yet been developed. Thus the individual must be especially cautious in his choice of equipment - to determine that it will operate as advertised and <u>especially</u> to determine that it will last the required time.

A last note of caution concerns obscuration of the sun. Trees, buildings, even telephone poles or air pollution can cause the incident solar energy to be signigicantly less than the solar radiation values used for the performance estimates in this Handbook. These solar radiation measurements were generally made in the 1950's. In mountainous or coastal regions the climate can be quite different at locations only a few miles apart.

WHAT TO EXPECT FROM SOLAR HEAT

Solar energy is a practical source for heating swimming pools, domestic hot water and buildings throughout the U.S. Since solar radiation is a relatively diffuse and weak source of energy at the earth's surface, large collection areas are required to obtain a large energy. There are no miracles to solar energy utilization - just straightforward use of basic engineering principles. A certain area of solar collectors can collect and utilize only a portion of the solar radiation which falls on it - usually no more than one-third to one-half of the total.*

It is uneconomical to design solar heating to provide for 100% of the heating requirements for most locations because of the necessarily large collector area and storage volume that would be required. Prolonged cloudy cold weather situations are most economically handled by an auxiliary heating system.

The use of an efficient wood-burning stove has appealed to many because wood is a relatively inexpensive energy source in many locations, is easily stored, and is a

*Estimates of the total annual energy which can be collected and utilized by a solar heating system are tabulated on page 15.

renewable resource - in fact a form of stored solar energy.

COST OF SOLAR HEAT

While it is true that the solar power received at the earth's surface is free for the collecting, the installation cost of a solar heating system is certainly not free.

Solar heating economics deals basically with the tradeoff between solar heating system installation cost and the future cost of the fuel saved by the solar heating system over its anticipated lifetime (e.g. 20 years). One must also account for any maintenance, operating, and repair costs.

The approach described below is a fairly useful way of estimating the cost of solar heat.

$$\text{Cost of Solar Heat} = \frac{\text{system installation cost} \times \left[\text{annual capital fixed charge rate} + \text{annual operating, maintenance, and repair cost}\right]}{\text{annual energy delivered by solar energy system}}$$

The annual capital fixed charge rate is simply the value of capital to the individual. It may be the interest rate that the capital could be earning if it were invested rather than used to install the solar heating system or it may be the annual cost of a loan made to finance the system installation. For example, the annual fixed charge rate of an 8-1/2%, 20-year loan is 0.1041.

An example calculation is shown below with the following assumptions:

system installation costs = $15 per sq ft of collector

annual capital fixed charge rate = 0.08

maintenance, operating, and repair cost = 2% of installation cost per year

annual energy delivered by solar energy system = 135,000 BTU per year per sq ft of collector area

Then, from the formula:

$$\text{Cost of solar heat} = \frac{(\$15)(0.08 + 0.02)}{(0.135)}$$

$$= \$11.11 \text{ per million BTU}$$

The cost of competing energy such as natural gas or electricity varies very widely and at present is generally less than $11.11 per million BTU. For example, if natural gas is $1.50 per 1000 standard cubic feet and is burned in a furnace at 60% efficiency, then the delivered cost of the heat is roughly $2.50 per million BTU.

If electricity costs 2.5¢/kwh then the cost of electrical resistance heating is $7.32 per million BTU.

A compelling argument in favor of solar heating is the knowledge that the cost of the solar heat will remain constant in the future since all the parameters in the equation should remain fixed. It is certain that the cost of competing energy will rise. The question is only - how much?

The installation cost of active solar heating systems is fairly high at present. Values of $20 to $30 per sq ft of collector are common for professional installations. Repetitive installations of mass produced components and competition should bring the costs down some. Do-it-yourself installations can reduce the labor costs. Typically the cost of solar collectors accounts for 40% to 60% of the total installation costs of the system.

The annual energy delivered by the solar energy system can be estimated fairly closely using the simulation code. For convenience, these simulated results are tabulated on the following page.

SIMULATED ANNUAL ENERGY DELIVERED BY THE SOLAR HEATING SYSTEM FOR THE PACIFIC REGIONAL CITIES STUDIED

City	Annual net energy collected per sq ft of collector, in millions of BTU per year		
	Fraction of the total heat delivered by the solar heating system		
	25%	50%	75%
Santa Maria	0.376	0.354	0.293
Los Angeles	0.327	0.264	0.228
Phoenix	0.235	0.190	0.131
Fresno	0.203	0.140	0.095
Medford	0.226	0.145	0.073
Vancouver	0.174	0.122	0.066
Seattle	0.168	0.108	0.056

SOURCES OF INFORMATION

Solar Energy Books

F. Daniels, "Direct Use of the Sun's Energy", Ballantine, New York, NY, 1974, 271 pp, about $2.

J. Duffie and W. Beckman, "Solar Energy Thermal Processes", Wiley-Interscience, New York, NY, 1974, 386 pp, about $17.

D. S. Halacy, Jr., "The Coming Age of Solar Energy", Harper and Row, New York, NY, 1973, 231 pp, about $9.

W. H. Rankins, III and D. A. Wilson, "Practical Sun Power", Lorien House, New York, NY, 1974, 52 pp, about $4.

P. A. Fleck, "Solar Energy Handbook", Time-Wise Publication, Pasadena, CA, 1975, 92 pp, about $4.50.

J. R. Williams, "Solar Energy Technology and Applications", Ann Arbor Science, Ann Arbor, MI, 1975; 120 pp, about $7.

J. Sands, "Solar Heating Systems", Solar Systems, Box 110, Danbury, NH, 46 pp.

S. V. Szokolay, "Solar Energy and Building", John Wiley and Sons, New York, NY, 1975, 148 pp, about $18.50.

"Solar Energy in Building Design", Total Environmental Action, unpublished but available from TEA. Church Hill, Harrisville, NH.

SOURCES OF INFORMATION

Heating System and House Thermal Design

ASHRAE, Handbook of Fundamentals, 1975, published by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, New York, NY, 688 pp.

ASHRAE, Handbook and Product Directory; on Systems (1973); Applications (1974); Equipment (1975).

Climatic and Solar Radiation Data

U.S. Climatic Atlas, U.S. Dept. of the Interior, Geological Survey, Washington, D. C., 417 pp, about $4.95.

J. Keyes, "Harnessing the Sun to Heat your House", Morgan & Morgan, Dobbs Ferry, NY, 1975, about $3.00.

List of Equipment Manufacturers

Solar Energy Heating and Cooling Products, ERDA, Division of Solar Energy, ERDA-75, 1975.

Survey of Solar Energy Products and Services - May 1975, Committee on Science and Technology, U.S. House of Representatives, 94th Congress, 1st Session, 545 pp.

"Solar Energy Industry Directory and Buyer's Guide, 1975", Solar Energy Industries Assoc., Inc., 1001 Connecticut Ave., NW, Washington, D. C. 20036, 42 pp.

SOURCES OF INFORMATION

Bibliographies

Solar Thermal Energy Utilization, 1957-74
Technology Applications Center
University of New Mexico (1974), Two Volumes (updated periodically)

Solar Energy
A Bibliography
U. S. Atomic Energy Commission, TID-3351 (1974)

Journals

Solar Energy, International Solar Energy Society Journal

ASHRAE Journal

Directories

1975 Solar Directory - Compiled and distributed by:
Environmental Action of Colorado, University of Colorado, about 200 pp, and $15.
1100 14th St.
Denver, CO 80702

W. A. Shurcliff, "Informal Directory of the Organizations and People involved in the Solar Heating of Buildings", 1975, 118 pp, about $5.

W. A. Shurcliff, "Solar Heated Buildings, a Brief Survey", (Updated periodically, 11th edition, 1975), 172 pp, about $7.
(Both publications may be purchased at: 19 Appleton St., Cambridge, MA 02138.)

Directories (Continued)

"Solar Oriented Architecture", Arizona State University, College of Architecture, 1975 Research Report prepared for the AIA Research Corp. and the National Bureau of Standards; 142 pp.

Proceedings of Workshops on the Solar Heating and Cooling of Buildings

"Solar Energy Storage Subsystems for the Heating and Cooling of Buildings", Charlottesville, VA, April 16-18, 1975; published by ASHRAE, 191 pp.

"Solar Cooling for Buildings", Los Angeles, CA, Feb. 6-8, 1975; U.S. Government Printing Office, Stock No. 3800-00189, 231 pp.

"Workshops on Solar Collectors for Heating and Cooling of Buildings", New York City, NY, Nov. 21-23, 1974; National Science Foundation and RANN Document NSF-RA-N-75-019; 507 pp.

Alternate Energy Resources and Home Design

P. Clegg, "New Low-Cost Sources of Energy for the Home", Garden-Way, Charlotte, VT, 1975, 252 pp, about $6.

K. Kern, "The Owner-Built Home", Owner-Builder Publications, 1972, Oakhurst, CA, about $7.50.

"Solar Energy Home Design in Four Climates", Total Environmental Action, Church Hill, Harrisville, NH, 198 pp.

SOURCES OF INFORMATION

Alternate Energy Resources and Home Design (Continued)

R. Schoen et al, "New Energy Technologies for Buildings", Ballinger, New York, NY, 1975, 217 pp.

P. Steadman, "Energy, Environment, and Building", Cambridge, MA, 1975, 287 pp.

W. Clark, "Energy for Survival", Anchor Books, New York, NY, 1975, 652 pp, about $5.

"Energy Primer", Portola Institute, Menlo Park, CA, 1974, 200 pp, about $4.50.

V. G. Olgyay, "Design with Climate", Princeton, University Press, 1963.

The amateur is advised to be very well informed or to seek the service of a consultant before committing to a major investment.

BUILDING THERMAL LOAD

In order to use the graphs and charts in this Handbook, it is necessary to calculate the thermal load of the building to be solar heated and express the load in units of BTU/Degree-Day. This is the total heat required by the building per day per degree Fahrenheit temperature difference between the inside temperature and the outside temperature. The major contributors to building thermal load are thermal conduction through the materials comprising the outer fabric of the building, and the energy required to heat the outside air which infiltrates the building through cracks, porous walls, open windows, and doors. Methods of calculating heat losses are to be found in handbooks of the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE). Basically, the procedure consists of adding up the area of each type of external building fabric (walls, windows, doors, ceiling, floor, etc.) times the appropriate thermal conduction coefficient (U-value, in BTU/ft^2 °F hr), and adding to this the infiltration load determined by multiplying the building volume times the number of air changes per hour times the heat capacity of the air (0.018 BTU/°F ft^3). The infiltration rate is usually in the range of one-half to one air change per hour. Less infiltration than this will lead to stuffiness and more may be required for some en-

DESIGN CONSIDERATIONS

The first consideration in the thermal design of the building should be to minimize the load within the constraints imposed by economics and architecture. Adequate insulation, double glazing, control of infiltration, and passive control of the solar gains are usually more cost effective than active solar heating. After the loads have been minimized by these techniques, then active systems might be considered to satisfy the bulk of the remaining load.

The system should always be designed with a full-capacity auxiliary heating unit for periods of extended cloudiness.

Design optimization usually involves a tradeoff between cost and performance. At some point the extra performance which can be achieved by adding on more equipment or material will exceed the savings incurred. This is true at some point of extra insulation, extra collector area, and many other design variables. In a few cases a true performance optimum exists (for example, collector tilt and orientation).

Component lifetime and maintenance should receive major consideration in the design. The solar energy which falls on a building may be free but the equipment involved can represent 5 to 15% of the building cost and must have a lifetime of 15 to 30 years to warrant the investment. Corrosion, ultraviolet degradation, weathering, and fouling are areas which deserve special consideration.

vironments like a meeting room.

For a small, single story, well insulated building, the thermal load should be in the range of 8 to 10 BTU per degree-day per sq ft of floor area* It is important to minimize the building load since the area of solar collector required to obtain a given solar heating fraction is directly proportional to the building load.

*The exact building load can be determined from past monthly and annual heating bills and degree day values. Fuel consumption corrections should be made for furnace efficiency (typically 0.6) and for non-space-heating energy uses. The latter can be determined from summer fuel bills.

TYPES OF SOLAR HEATING SYSTEMS

Solar heating of swimming pools is particularly cost effective because relatively inexpensive collectors can be used. This is because glazing is not required at low temperatures typical of pool water. LASL has not made performance calculations. However, manufacturers recommend a collector area of roughly one-half the pool area being heated. Pools should be covered at night to reduce evaporative and radiation cooling.

Domestic solar water heaters are attractive because they work year around and can usually be retrofit to existing dwellings because the total collector area requirements are small (50-200 ft^2). Tank type systems which do not require circulation have been designed. More commonly, circulation of collector coolant is achieved either with a pump or by placing the storage tank above the collector and utilizing natural circulation.

Many kinds of space heating systems have been devised. The more common types utilize a liquid or air to cool separate solar collectors, water or a rock bed for heat storage, and forced air, convectors, or radiant panels for heat distribution. Domestic water heating is frequently combined with space heating.

In passive designs, one attempts to utilize the architecture of a building to maximize solar gains during the cold season, minimize heat losses, and provide thermal storage within the building. These

TYPES OF SYSTEMS

- SWIMMING POOL HEATERS

  UNGLAZED PLASTIC COLLECTORS

- DOMESTIC HOT WATER HEATERS

  NON-CIRCULATING
  OR
  CIRCULATING — THERMOSIPHON (NATURAL CONVECTION)
  — PUMPED

- SPACE HEATING (ACTIVE SYSTEMS)

  LIQUID COOLED COLLECTORS / WATER STORAGE / FORCED AIR DISTRIBUTION OR CONVECTOR DISTRIBUTION

  AIR COOLED COLLECTORS / ROCK BED STORAGE / FORCED AIR DISTRIBUTION

  } USUALLY COMBINED WITH DOMESTIC HOT WATER HEATING

  OR
  ABOVE TWO TYPES INCORPORATING HEAT PUMPS

- SPACE HEATING (PASSIVE SYSTEMS)

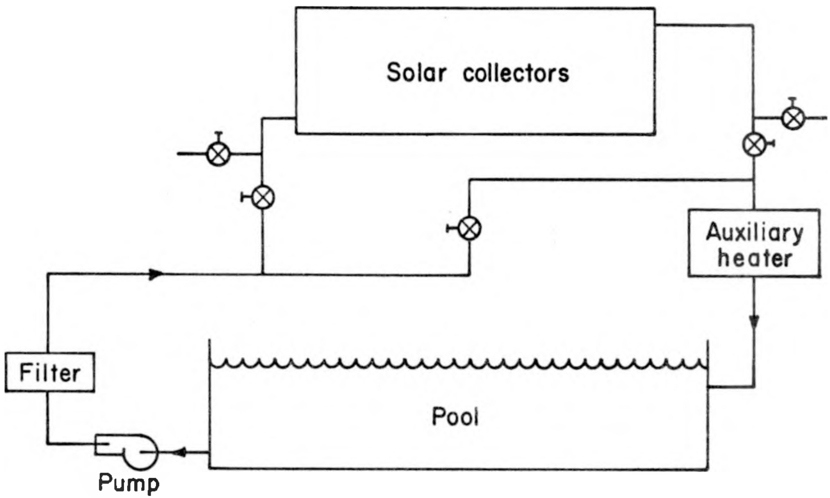
systems are described on the following pages.

SWIMMING POOL HEATERS

An unheated swimming pool has a natural yearly temperature cycle that varies with climate and geography. For northern California, a comfortable three to four month swimming season can be stretched out to five or six months when a pool heater is added; for southern California or Arizona the three to four month swimming season can be stretched out even further. The key to effectively heating a pool with solar energy is a very large solar panel area to collect sufficient amounts of solar energy. A minimum of one-half the pool's surface area in solar panels must be used and often the full area of the pool is duplicated in solar panels.

The operating regime of a solar swimming pool heater is different from either a solar hot water heater or space heater, both of which commonly produce 140°F water and for which the surface temperature of the collector can exceed 200°F. The pool heater is trying to raise several thousand gallons of water to around 80°F while the hot water heater, for example, is trying to raise less than 100 gallons to 140°F. By circulating a large volume of low temperature water through the solar collector, thereby keeping the collector cool, the solar pool heater can take full advantage of the fact that solar collectors are much more efficient working at low temperatures. A solar pool heater can deliver a large volume of water only several degrees warmer than it was when it entered the collector.

SWIMMING POOL SCHEMATIC



Solar pool heaters can operate at efficiencies of 70-80% whereas solar hot water and space heaters commonly operate in the 30-50% range.

Swimming pool collectors are normally unglazed and therefore much less expensive than collectors for water heating or space heating which must be glazed to achieve the desired higher temperatures.

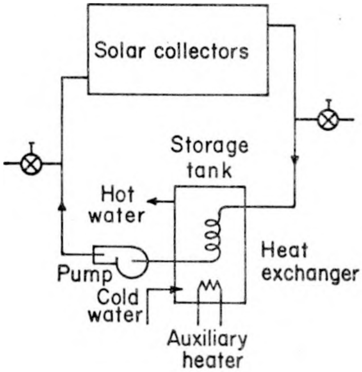
There are also passive means of heating swimming pools. The sides and bottom of the pool can be painted a dark color (e.g., dark blue) to increase the absorption of the solar radiation incident on the water. Also, the pool can be covered by a glazed enclosure to trap incident solar radiation and reduce convective and evaporative cooling. The pool water will act as effective thermal storage.

DOMESTIC HOT WATER HEATERS

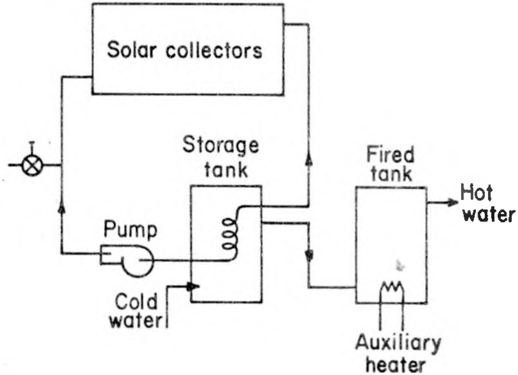
Solar hot water heaters are common in many parts of the world, particularly the warmer regions where collector freezing is not a problem. Of the many different types available, the simplest is a large plastic pillow filled with water which can be installed on the roof of the building to collect whatever solar energy is available. More durable containers use four or six inch diameter pipes of steel, plastic or glass to hold the water.

A regular liquid-heating solar collector is often used as a solar hot water heater. In this case the absorbed heat travels by thermal convection or electric pump to a storage tank. Both these collectors heat the domestic water directly.

DOMESTIC HOT WATER SCHEMATIC



ONE-TANK SYSTEM



TWO-TANK SYSTEM

To avoid the possibility of freezing in colder climates, an indirect heating system may be used in which a liquid coolant is circulated between the collector and a heat exchanger in the storage tank. One must always worry about contamination of the storage water, particularly if the collector coolant is toxic.

Since solar water-heating systems only work when the sun shines, auxiliary heating may be needed from time to time in order to maintain a desired minimum temperature in the domestic water tank. A thermostatically-controlled electric resistance coil or gas burner would suffice.

Two basic types of domestic hot water solar heaters have been studied: one-tank and two-tank. In the one-tank system, already described above, the solar heated water storage tank serves directly as the tank for domestic hot water needs. Auxiliary heat is applied directly to this tank. In the two-tank system, the solar storage tank serves as a supply of preheated water for the regular, fired tank.

SPACE HEATING SYSTEMS

LIQUID HEATING COLLECTORS

In a solar space heating system employing a liquid heating collector, a liquid heat transfer fluid passes through channels in a solar collector that is blackened and oriented to be heated by the incident solar energy. The fluid is heated in the collector and the collected heat is transferred to a heat storage system. The common thermal storage medium for liquid heating collectors is water, which is heated directly if water is also the heat transfer fluid, or indirectly heated through a heat exchanger if a non-aqueous heat transfer fluid is used. Space heating is accomplished by pumping the heated water from storage through a water-air heat exchanger. Cooler air is drawn from the room, forced through the heat exchanger, and the heated air discharged into the room through registers. If the room temperature is too low, an auxiliary heater is used to add additional heat to the air.

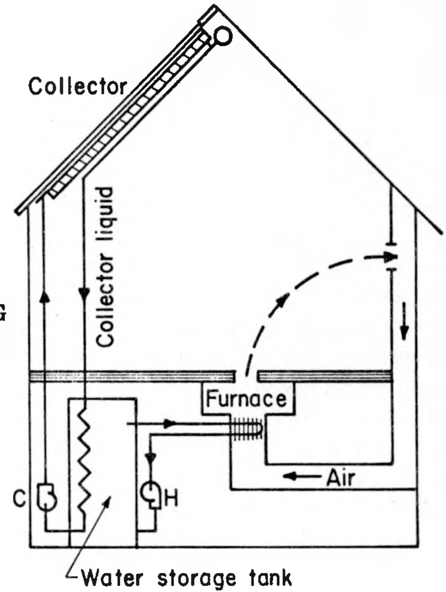
LIQUID HEATING COLLECTORS - SYSTEM OPERATION

Operation

When the collector temperature exceeds the water tank storage temperature. Pump C is operated to transfer solar heat to storage (see figure on next page).

When the building requires heat, Pump H is operated to heat the air upstream of the furnace through a finned-tube coil. If the energy added is insufficient to maintain room temperature, the furnace is operated.

SPACE HEATING SYSTEM USING LIQUID HEATING COLLECTORS



Liquid Heating Collectors / Water Storage / Forced Air Distribution

Heat Exchanger

The heat exchanger to transfer heat from the collector liquid to the water tank can be as simple as a coiled piece of tubing, usually copper, immersed in the water storage tank through which the collector liquid passes. Alternatively, an external heat exchanger can be used.

Heat Transfer to House

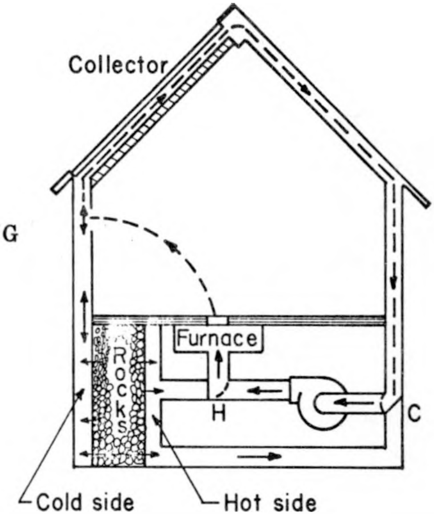
To transfer heat to the building, heated water is pumped from storage through the finned-tube coil. Cooler air drawn from the room is forced over the coil, and the heated air is discharged into the room through registers. Alternatively, a convector can be used, although the performance is usually reduced.

AIR HEATING COLLECTORS

In a system employing an air heating collector, air passes through channels in the collector, is heated, and the heat is transferred to a heat storage system. Air has a low heat transfer coefficient, but air-heating collectors can be properly designed to have adequate performance.

The common thermal storage medium for air heating collectors is a bin full of rocks which are heated by the hot air from the collector. Space heating is accomplished by blowing cooler air from the room through either the heated rock-bin or directly through the collector itself. If room temperature falls below a chosen level, an auxiliary space heater is used to

SPACE HEATING SYSTEM USING AIR HEATING COLLECTORS



Air Heating Collectors / Rock Bed Storage / Forced Air Distribtion maintain room temperature.

Operation

The system requires only one fan and two double-dampers.

When the collector temperature exceeds the rock bed exit temperature (left side), the collector is on and Damper C is in the position shown. Otherwise the collector is off and Damper C is in the upper position.

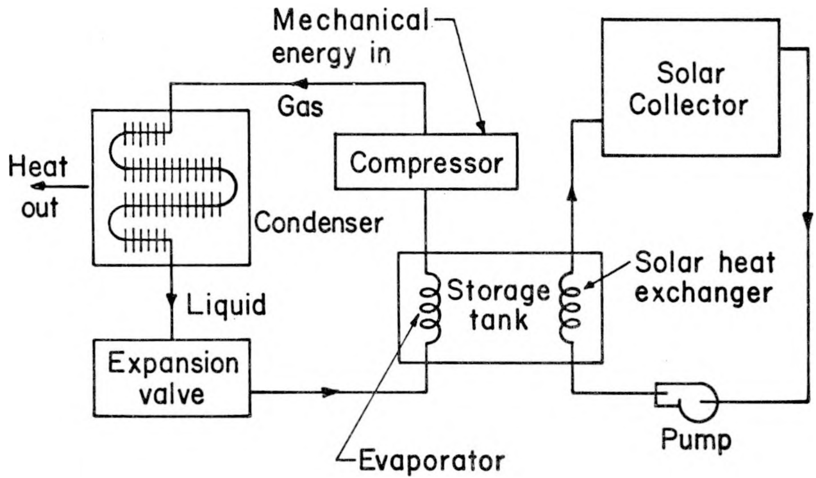
When the building requires heat the Damper H is in the position shown. Otherwise the Damper H is in the upper position. The furnace is operated as necessary to satisfy the building load. The fan is on when either the collector is on or the building needs heat.

When the collector is on, the solar heated air is routed either to the building space directly (when the building needs heat) or to the rock bed. When the collector is off, the building is heated by blowing air through the rock bed in the reverse direction and directly into the building space.

HEAT PUMP AUXILIARY

A heat pump auxiliary can be used to improve the performance of either a liquid or air heating collector system. Essentially a heat pump adds thermal energy using an electric motor. It consists of a closed loop of a refrigerant, which evaporates and condenses continuously (as in a refrigerator). Low-grade solar heat is applied to one side of the heat pump and

HEAT PUMP SCHEMATIC



causes the refrigerant to evaporate. An electrically driven compressor then raises the pressure and temperature of the vapor and when condensed gives off heat at a higher temperature than was provided. When the temperature difference is less than 30-40 °F, a good heat pump can provide around 3.5 BTUs of high grade heat (160-180°F) for every BTU of energy the compressor uses (i.e., has a coefficient of performance, or COP, of 3.5).

There are many possible ways to utilize a heat pump in a solar heating system, one of which is shown in the preceding figure.

PASSIVE USE OF SOLAR ENERGY

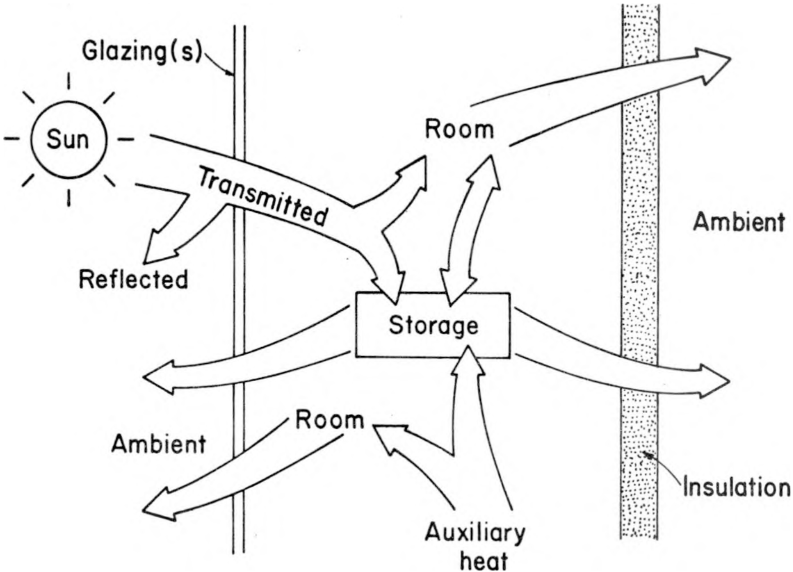
The conventional approach to solar heating of buildings discussed on the previous pages utilizes separate thermal storage. Thermal energy is transported from the collectors to storage and subsequently from storage to the building thermal control system in a completely regulated way using liquid or air as a heat transport fluid. This approach has come to be known as an "active" solar energy system due to the requirement for pumping the heat transport fluid.

An alternative approach is frequently referred to as "passive" solar energy utilization. Solar gains through windows or onto walls of the building are utilized to effect a net reduction in the total building energy requirement. Generally there is no separate collector as such. Generally there are no continuously pumped fluids. Heat is transported by natural convection or radiation. Since solar gains are present in every building each is passively solar heated to some extent. It is when solar energy utilization becomes a major objective of the architectural design and solar energy supplies a major fraction of the building energy that one would refer to the building as being a solar building. Frequently these buildings are referred to as sun-tempered structures.

The term "passive" has been adopted because it is in widespread use. However, it has the unfortunate connotation of a completely hands-off, self-running design. For the purposes of this handbook the definition of sun tempering or passive solar energy utilization should be taken in its widest context. It certainly might include any system which collects, stores, and controls solar gains by any means other than the use of a separate solar collector cooled by a forced-convection coolant. If the use of a small amount of auxiliary energy can materially decrease the overall auxiliary thermal requirement, then this arrangement should not automatically be ruled out, although it may not be strictly passive. Examples would be the use of a small fan to control air circulation, or movable insulation or shading devices although they may be motor driven. In short, the purpose is not to make a fetish of the passive nature of the approach but rather to maximize the effectiveness of the design while attempting to minimize the requirement for equipment and mechanical energy.

A number of examples of passive solar heating concepts have been built into structures and have received widespread attention. Some of these are described later on. Most use extensive south exposure

PASSIVE SCHEMATIC



glass to admit low-angle winter sunshine to the building and extensive use of mass heat capacity inside the thermal envelope of the building to store heat energy. Despite the publicity given to these projects and their apparent success in saving energy, they have not been widely adopted. This is probably due to a combination of skepticism as to their effectiveness and a lack of engineering criteria.

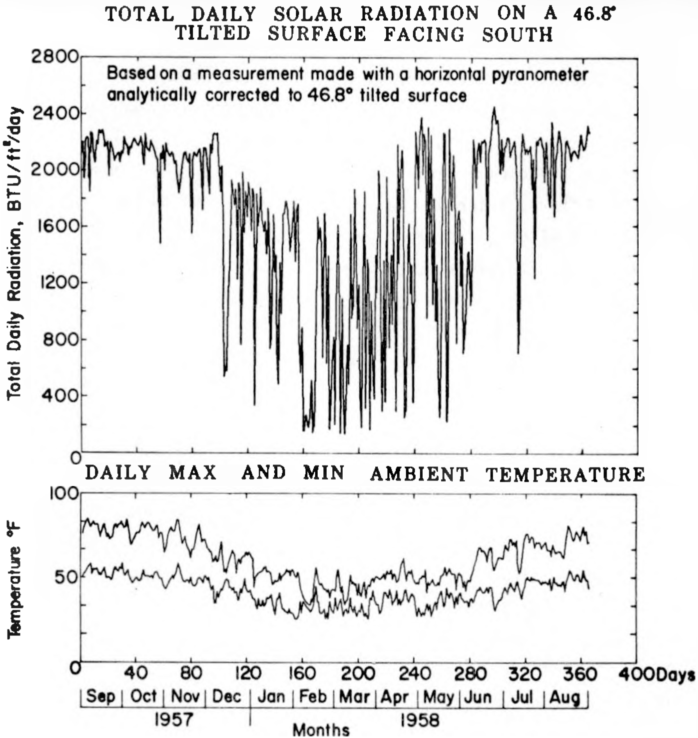
The principle problem has been lack of quantitative basis for incorporation of the basic concepts into architectural design. Although the calculation of solar gains through fenestration is well understood and documented, the storage and control of this energy is not within the grasp of the building designer. The challenge confronting passive solar energy design is not one of heat collection, since a window is an efficient solar collector, but one of storage and control of the heat to maintain suitable comfort standards within the building. When used in conjunction with a conventional heating system, temperature variations in the building can be reduced to those normally designed.

It is possible with proper building design to maximize collection of solar energy using passive heating concepts. Adequate solar energy collection combined with the proper conservation measures can provide for a large fraction of the heating load of a building. In passive systems the surface receiving the solar flux can be combined with the thermal storage mass in a more or less integrated unit (e.g., drum wall collection/storage); or the storage can be separate from the directly irradiated surface (e.g., massive internal walls out of direct sunlight); or there can be a combination of storage masses directly and indirectly heated by the solar radiation -- the usual case of a house with large south facing windows, furniture, internal walls, and floor in direct sunlight, and other internal walls in shadow. In general, the arrangement where the thermal storage medium is directly heated by the solar flux will provide for the highest fraction of passive solar heating.

SOLAR ENERGY AVAILABILITY AND HEATING LOADS

The daily solar energy availability for the year analyzed for Fresno is shown in the following figure, along with the corresponding daily temperature variations. The graph of the daily radiation is based on insolation data for radiation on a horizontal surface that has been analytically corrected to a 46.8° tilted surface. For the year analyzed, the total solar radiation incident on a flat surface tilted 46.8° and facing due south was 614,067 BTU per year per square foot. The corresponding space heating load for Fresno for the same period was 2622 degree-days, 90 more than the normal value of 2532 degree-days. The month-to-month integrated variation in solar radiation, along with the corresponding degree-day heating load, is given in the table on Page 43.

Typical yearly total sunlight incident on a horizontal surface and typical heating loads at the chosen seven Pacific Regional cities are given in the table on Page 44.


TOTAL DAILY SOLAR RADIATION ON A 46.8° TILTED SURFACE FACING SOUTH
Based on a measurement made with a horizontal pyranometer analytically corrected to 46.8° tilted surface
Total Daily Radiation, BTU/ft²/day
0
400
800
1200
1600
2000
2400
2800
DAILY MAX AND MIN AMBIENT TEMPERATURE
Temperature °F
0
50
100
0
40
80
120
160
200
240
280
320
360
400 Days
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
Jun
Jul
Aug
1957
1958
Months

SOLAR ENERGY AVAILABILITY AND HEATING LOADS FOR THE SEVEN PACIFIC REGIONAL CITIES FOR SELECTED YEARS

City	Year	$E_{Horizontal}$ BTU/FT^2/Yr*	DD/Yr
Fresno	1957-58	560946	2622
Phoenix	1962-63	686521	1278
Santa Maria	1956-57	649922	3065
Los Angeles	1963-64	620994	1700
Medford	1961-62	527449	5275
Seattle	1963-64	387602	5204
Vancouver	1970	385672	5909

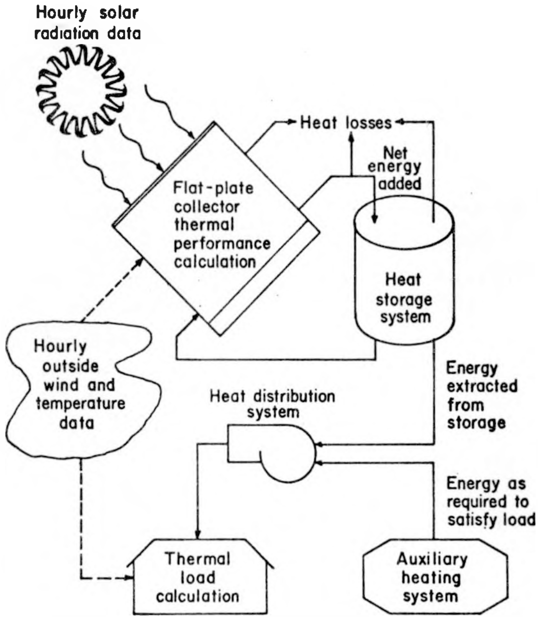
*Measured on a horizontal surface.

DETERMINATION OF PERFORMANCE BY SIMULATION ANALYSIS

The performance of a solar heating system is predicted by simulation analysis using the observed weather and solar data. The actual system is simulated by a digital computer code on an hour-by-hour basis. The simulation model is shown schematically, on the following page.

At each hour the net energy which can be extracted from the collector is calculated. This is determined from the solar radiation, the collector design, the outside temperature and wind condition, and the inlet fluid temperature from storage. If this energy is positive it is added to storage. The thermal load is calculated either from the outside temperature (for space heating) or a fixed schedule (for water heating). This energy is extracted from storage by the heat distribution system to satisfy the load. If the load cannot be totally satisfied from storage then auxiliary heat is added as required to make up the difference. The change in storage temperature over the hour is the net energy added from the collector minus storage heat losses minus the energy extracted by the thermal load, divided by the storage heat capacity.

This calculation is repeated for each of the 8760 hours of the year. All energy flows are summed hour-by-hour and both monthly and yearly summaries are printed out. A typical year-long calculation requires only 34 seconds on the Los Alamos



SIMULATION SCHEMATIC

CDC 6600 computer and thus it is feasible to study the effect of changes in many design parameters.

FLAT PLATE COLLECTOR DESIGN

At the low temperatures used for water or space heating, flat plate types work with reasonable efficiency and are rela-easy to construct.

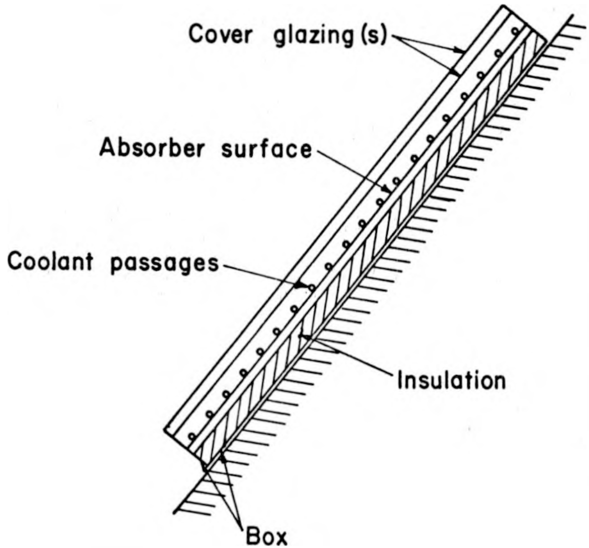
The major functional parts of a collector are the absorber surface, coolant passages, cover glazing, and back insulation.

Collectors are designed to maximize absorption of solar radiation and minimize heat losses. The predominant heat loss mechanisms from the front face are by convection and radiation. Convection losses are controlled by the use of one or more transparent covers. Radiation losses are sometimes reduced by means of a "selective" coating on the absorber surface which has a high absorptance for the solar spectrum and a low emittance for the infrared re-radiation spectrum. Other heat loss mechanisms are conduction to the collector backside and edges.

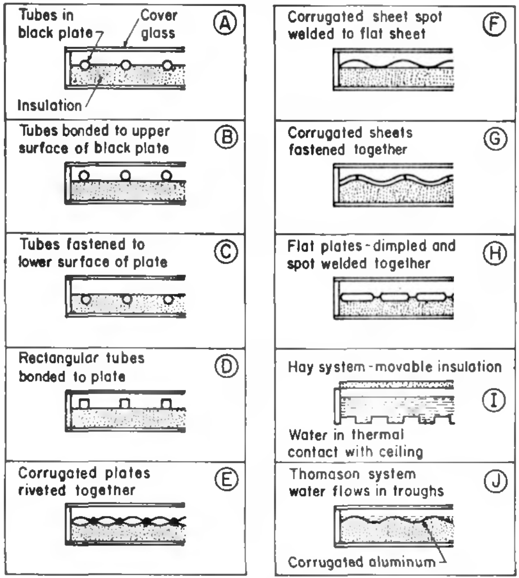
COLLECTOR PERFORMANCE

Collector efficiency is defined as the ratio of the heat removed by the coolant divided by the incident solar energy. It is not a single number but is dependent on a variety of conditions such as the coolant

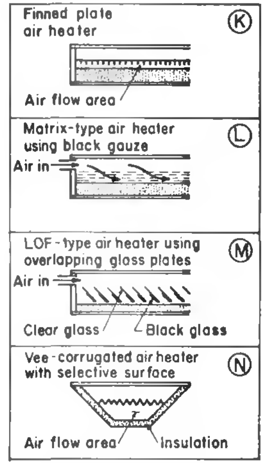
COLLECTOR FUNCTIONAL PARTS



COLLECTOR TYPES

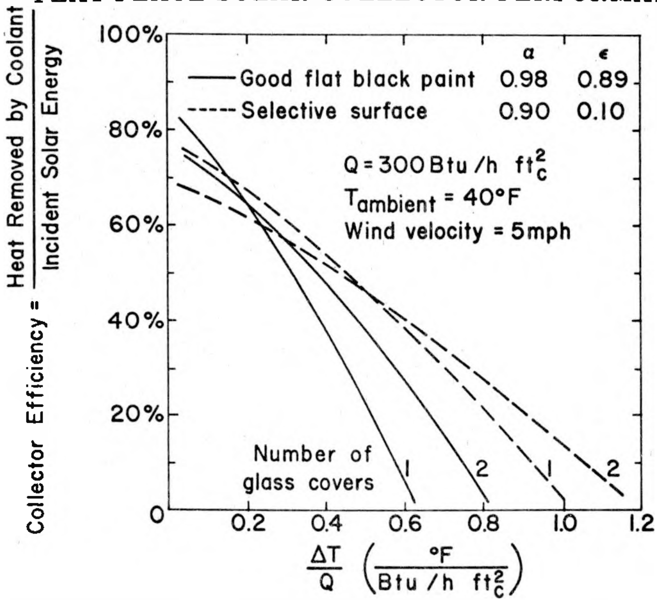


WATER HEATERS



AIR HEATERS

FLAT-PLATE SOLAR COLLECTOR PERFORMANCE MAP



temperature and flow rate, the angle of the incident sunlight, and the wind velocity. It is also dependent on collector design parameters.

Collector performance is calculated by performing a detailed heat balance on the absorber surface and on each sheet of glazing, accounting for solar and infrared absorption and radiation, convection and conduction heat flows.

Collector efficiency is shown at the left for one set of conditions. It has become standard practice to plot collector efficiency as a function of the parameter $\Delta T/Q$:

$$\frac{\Delta T}{Q} = \frac{\text{Average Collector Surface Temperature} - \text{Ambient Temperature}}{\text{Incident Solar Radiation}}$$

LIQUID COLLECTOR FLUIDS

The selection of a liquid collector coolant is a choice between conflicting requirements. One would desire a stable non-freezing, non-boiling, non-corroding, non-flammable, inexpensive, high-specific-heat, non-viscous, non-toxic, and non-energy-intensive fluid. Many liquids have been tried and all have been found deficient in some category. Ethylene glycol/water mixtures are often used. LASL is currently evaluating a number of fluids among which a class of light paraffinic oils such as Dowtherm HP and Thermia 33 show promise.

The use of a collector coolant other than water implies that a heat exchanger is required between the collector coolant circuit and the water in the heat storage tank.

LIQUID COLLECTOR FLUIDS

	WATER	85% ETHYLENE GLYCOL/WATER	THERMIA 15 PARAFFINIC OIL	UCON (POLYGLYCOL) 50-HB-280-X
FREEZING POINT	32°F	-33°F	--	--
POUR POINT	--	--	10°F	-35°F
BOILING POINT (@ ATM. PRESS.)	212°F	265°F	700°F	600°F
CORROSION	(CORROSIVE TO Fe OR Al, REQUIRES INHIBITORS)		NON-CORROSIVE	NON-CORROSIVE
FLUID STABILITY	(REQUIRES pH OR INHIBITOR MONITORING)		GOOD*	GOOD**
FLASH POINT	NONE	NONE	455°F	500°F
SPECIFIC HEAT	1.0	0.65	0.46	0.46
BULK COST ($/GAL)	--	2.35	1.00	4.40
THERMAL CONDUCTIVITY (BTU/HR FT °F @ 100°F)	0.359	0.18	0.76	0.119
HEAT CAPACITY (BTU/LB °F @ 100°F)	1.0	0.66	0.46	0.45
VISCOSITY (LB/FT HR @ 100°F)	1.66	15.7	28.5	143.1

*REQUIRES AN ISOLATED COLD EXPANSION TANK OR NITROGEN CONTAINING HOT EXPANSION TANK TO PREVENT SLUDGE FORMATION.

**CONTAINS A SLUDGE FORMATION INHIBITOR.

DESIGN PARAMETERS OF THE STANDARD LIQUID SYSTEM

Values of parameters used for the "standard" solar heating system using liquid heating solar collectors, a heat exchanger, water tank thermal storage, and forced air heat distribution system to the building. The values are normalized to one square foot of collector (ft_c^2).

PARAMETER:	NOMINAL VALUE:
Solar Collector	
Number of glazings	1
Glass transmissivity (at normal incidence)	0.86 (6% absorption* 8% reflection)
Surface absorptance (solar)	0.98
Surface emittance (IR)	0.89
Back insulation U-value	0.083 BTU/hr °F ft_c^2
Coolant flow rate	20 BTU/hr °F ft_c^2**
Heat capacity	1 BTU/°F ft_c^2
Heat transfer coefficient to liquid coolant	30 BTU/hr °F ft_c^2
Tilt (from horizontal)	Latitude + 10°
Orientation	Due south
Collector Plumbing	
Heat loss coefficient (to ambient)	0.04 BTU/hr °F ft_c^2
Heat Exchanger	
Heat transfer effectiveness	10 BTU/hr °F ft_c^2

DESIGN PARAMETERS
(continued)

Thermal Storage

Heat capacity	15	BTU/°F ft_c^2
Heat loss coefficient (i.e. assuming all heat loss is to heated space)	0	BTU/hr °F ft_c^2

Heat Distribution System

Design water distribution temperature***	133°F	

Controls

Building maintained at 68°F	36	BTU/DD ft_c^2
Collectors on when advantageous	Standard building load	

---

*These values apply for normal incidence on ordinary double strength glass (1/8 in.). For other angles of incidence the Fresnel equation is used.

**See p. 86 for discussion of units of flow rate.

***The coil and air circulation are sized to meet the building load with an outside temperature of -2°F with 133°F water and an air flow rate adequate to make up the space heat losses at an air discharge temperature of 120°F. This corresponds to a finned-tube coil effectiveness of 80%.

LIQUID SPACE-HEATING SYSTEM

The nominal design parameters of what shall be called the "standard" liquid-heating collector system are shown on the preceding pages. The nominal values of the parameters were chosen using the computer simulation: the parameters were varied singly in order to obtain a value for which solar performance is optimized (or near optimized), as measured by percent solar heating. The results of varying these parameters for the liquid collector system will be shown later.

CHOICE OF FRESNO, CA, FOR THE TRADEOFF STUDIES

Fresno was chosen for the great bulk of the analysis presented in this Handbook. This choice was made on the basis of the curves presented on page 65. Fresno is intermediate between the colder three cities in the north and the warmer three cities in the south. No choice of a single city would be representative of the entire region.

Unfortunately, Fresno has a climate in which the heating load is concentrated in a two-month period when there is little sun. The climate during this period is foggy but not severely cold. The chart on page 58. indicates that the distribution of solar heating is very nonuniform. The effect of this climate characteristic is to desensitize the effect of some of the parameters on the total heat collected. Basically, during the two difficult months no active solar heating system works very well, and during the remainder of the year any solar heating system would work pretty well. Therefore the overall effect of a parametric change is averaged between two widely different operating regimes.

A similar tradeoff study has been done for a very different climate -- in Los Alamos, NM, where the winter climate is cold but sunny, with appreciable snowfall. These results are published in the "Los Alamos Solar Heating Handbook," LA-5967-MS, May 1975. (This is available through the National Technical Information Service, USDOC.) The study was not as comprehensive as the one done for Fresno. The tradeoffs are different in detail but the trends are the same.

SIMULATION RESULTS FOR THE "STANDARD" LIQUID SYSTEM IN FRESNO, 1957-58

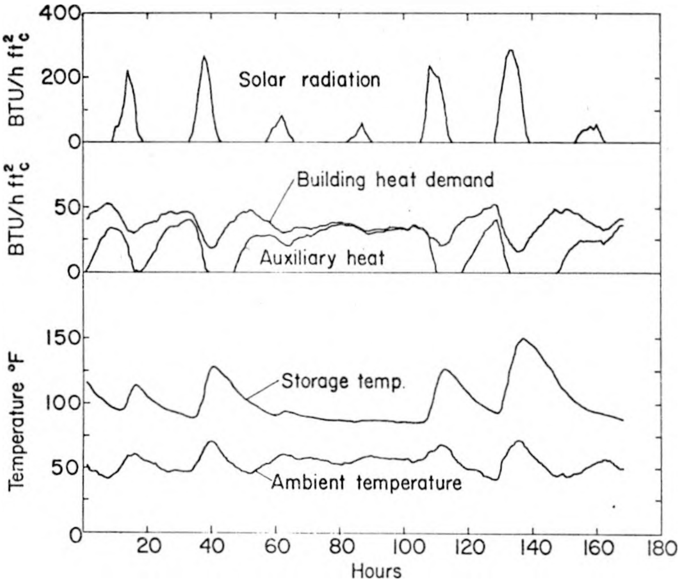
The computer simulation results for the "Standard" liquid system in Fresno for the year 1957-58 are analyzed in the next several pages.

On the following page is shown the response of the solar heating system to weather variations for seven consecutive days.

The simulation results for the entire year are shown on page 60 in response to weather conditions shown on page 42.

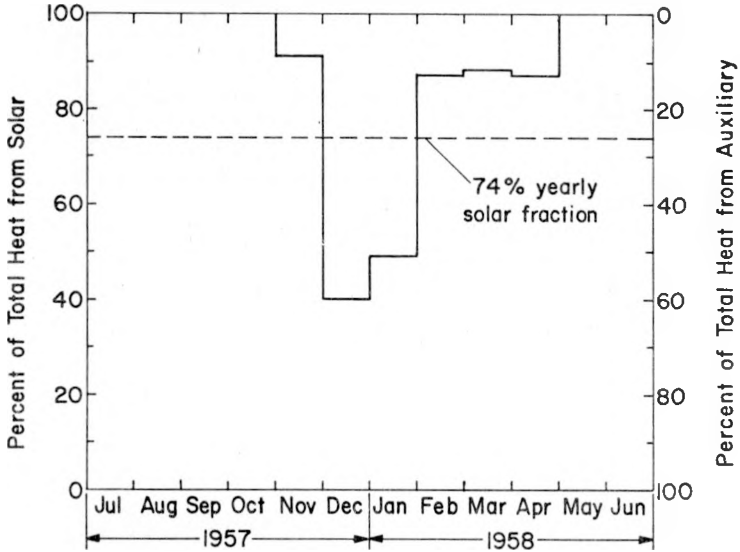
Monthly energy balances are shown on pages 58 and 59.

"Standard" Liquid System in Fresno
SEVEN DAYS IN WINTER
Jan. 15-21, 1958



"Standard" Liquid System in Fresno

SOLAR HEATING FRACTION BY MONTH



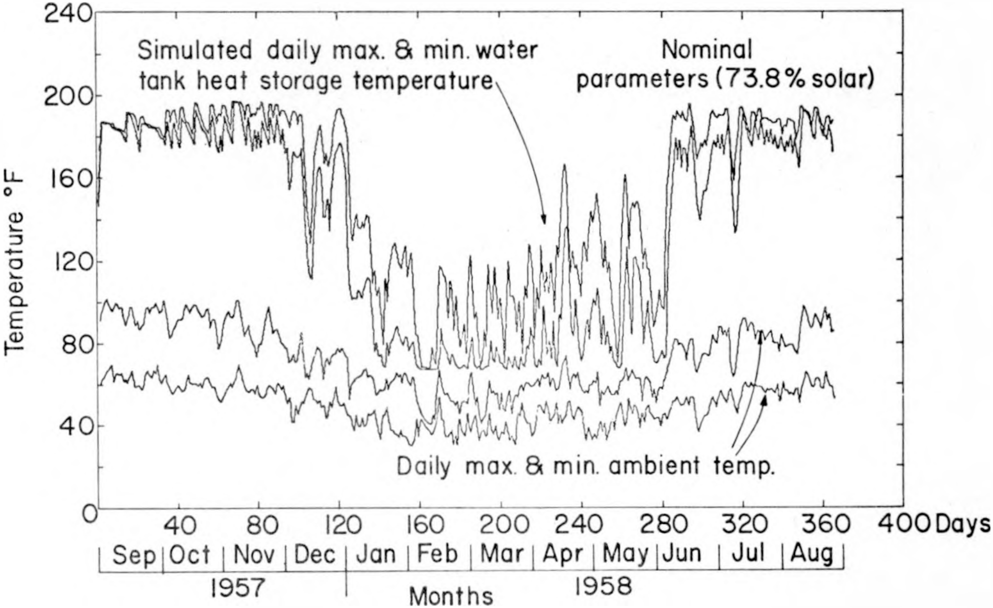
MONTHLY ENERGY BALANCE FOR FRESNO

Total energy flows for each month are shown below, for one ft^2 of collector. The sum of the solar heat collected plus the auxiliary heat used does not necessarily add up to equal the building load, because there is some carryover from the heat in storage from month to month. Initial storage temperature is set at 150°F for the simulation code. The percent solar heating for each month is calculated by dividing the building load minus the auxiliary heat by the building load.

Year	Month	Building Load BTU/Mo	Solar Heat Collected BTU/Mo	Auxiliary Heat Used BTU/Mo	Solar Heating %
1957	July	655	1262	0	100.00
	Aug.	1954	2146	0	100.00
	Sept.	2484	2781	0	100.00
	Oct.	8289	8876	0	100.00
	Nov.	18819	16534	1631	91.33
	Dec.	26043	10439	15575	40.20
1958	Jan.	24879	12857	12597	49.37
	Feb.	15247	13490	1939	87.28
	March	19029	17308	2185	88.52
	April	11104	11380	1434	87.09
	May	4176	4742	0	100.00
	June	2487	2885	0	100.00

"Standard" Liquid System in Fresno

SIMULATION RESULTS



YEAR TO YEAR VARIATIONS IN PERFORMANCE:

SIMULATION RESULTS FOR THE "STANDARD" LIQUID SYSTEM FOR SEVERAL YEARS IN FRESNO

Shown in the table are computer simulation results for the "standard" liquid-heating collector system for several years in Fresno. The table can be used to estimate year-to-year variation in solar performance and the average solar performance. The average percent solar heating is 75.2% with a standard deviation of 2.47%.

The year 1957-58 was chosen as a "test year" for detailed study because it is closest to the 11 year average.

SIMULATION RESULTS FOR THE "STANDARD" LIQUID SYSTEM FOR SEVERAL YEARS IN FRESNO

YEAR	BTU/yr ft^{2*}	BTU/yr ft^{2**}	DEG/DAY	% SOL
1952	688422	793966	2589	87.74
1953	672515	774268	2550	84.88
1954	627851	698473	3092	69.42
1955	643081	731670	2650	80.57
1956	591320	664500	2723	77.52
1957	560946	614067	2622	73.79
1958	578257	639456	2097	78.77
1959	566323	626296	2467	78.53
1960	532529	577985	2817	66.41
1961	545994	587526	2988	59.76
1962	636553	711595	2567	80.49
1963	606957	663720	3111	64.50
AVERAGES	604229	673626.83	2689.42	75.20

*Measured on a horizontal surface.

**Calculated for a tilted collector surface.

SIMULATION RESULTS FOR THE "STANDARD" LIQUID SYSTEM IN OTHER LOCATIONS AND FOR OTHER COLLECTOR AREAS

The computer simulations of solar performance for the "standard" liquid system are given below for the seven West Coast cities, where the square feet of collector per unit thermal load (i.e., DD ft_c^2/BTU) is varied. The calculations are made assuming a collector tilt of latitude-plus-ten-degrees.

The results of the hour-by-hour computer simulations are given in the following figure for the seven Pacific Regional cities studied. The year chosen for each city is a "test" year which was selected by examining the results of hour-by-hour computer simulations for several years (usually 10 to 12 years) for each city. The weather data used are for the period from 1950 to 1962.

The data on pages 64 and 65 are presented in terms of the ratio of building load to collector area to achieve a desired fraction of total heat from the solar heating system for the "standard" liquid system defined on pages 53 and 54. For example: for a 1500 square foot house in Fresno, CA with a building thermal load of 10 BTU/degree-day/ft^2 of house, the building load is 15,000 BTU/degree-day. To achieve a 75% solar heating fraction in Fresno, the table gives a value of 34 BTU/degree-day/ft^2 of collector. Therefore the required collector area is

$$\frac{15000}{34} = 441 \text{ ft}^2 \text{ of solar collector}$$

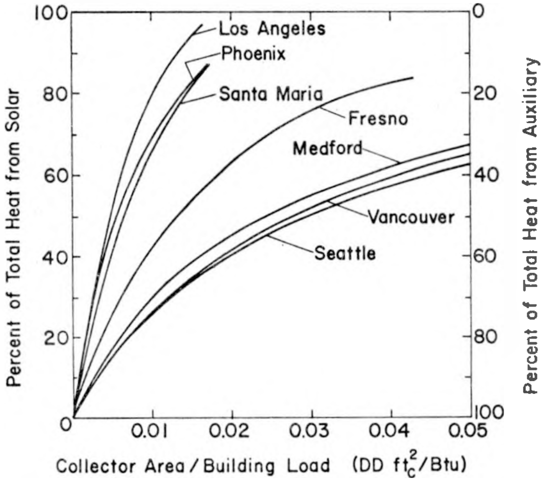
RATIO OF BUILDING LOAD TO COLLECTOR AREA FOR THE "STANDARD" LIQUID SYSTEM

($BTU/DD\ FT_c^2$)

	Percent Solar		
	25	50	75
Los Angeles	400	200	111
Phoenix	400	175	83
Santa Maria	310	152	76
Fresno	213	74	34
Medford	125	41	15
Vancouver	109	36	13
Seattle	106	33	12

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR AREA FOR VARIOUS LOCATIONS



EFFECT OF CHANGES IN DESIGN PARAMETERS ON THE PERCENT SOLAR HEATING USING THE "STANDARD" LIQUID SYSTEM

On the next several pages, the effect of changes in the design parameters on the computer simulated solar performance will be examined for the "standard" liquid system in Fresno for 1957 to 1958. The design parameters that are examined include the collector area, the number of glazings, the collector tilt angle, the collector orientation angle, the collector surface solar absorption, the glass transmissivity, the collector surface emittance, the collector back insulation, the collector coolant flow rate, the collector heat capacity, the coolant-to-collector heat transfer coefficient, pipe heat losses, heat exchanger effectiveness, water storage mass heat capacity, heat losses from storage and the design air distribution temperature. The design points for the "standard" liquid system parameters are indicated on each figure by a black dot.

Each curve on each graph has been plotted by making several complete computer simulations through the 1957-1958 year using the Fresno solar and weather data. For each calculation, only the parameter being studied was varied. Therefore all the complex system interactions which result from changing that variable have been taken into account.

The computer calculations that have been made to determine the effect of various design parameters on the overall performance of the solar heating system are intended to give information on trends of solar performance as a function of parametric variation rather than be taken as absolute results.

One word of caution to the Handbook user. The design parameters were varied singly, holding all other design parameters constant, to determine their individual effect on the overall performance of the solar heating system. Care should be used in simultaneously varying two or more design parameters, since some of these may be coupled non-linearly. If one simultaneously varies two or more design parameters to any significant extent, it may be that the net change in performance (compared to the "standard case") will be different than the performance estimated by changing the parameters individually and adding up the effect of all of the changes. A separate computer simulation may be needed for an accurate performance prediction for this case.

THE EFFECT OF COLLECTOR AREA AND THE EFFECT OF NUMBER OF SHEETS OF GLASS

The chart shows the effect of changing the collector area for three different cities, Los Angeles, Fresno and Seattle. The plotting parameter is collector area divided by the building load in BTU per degree-day.

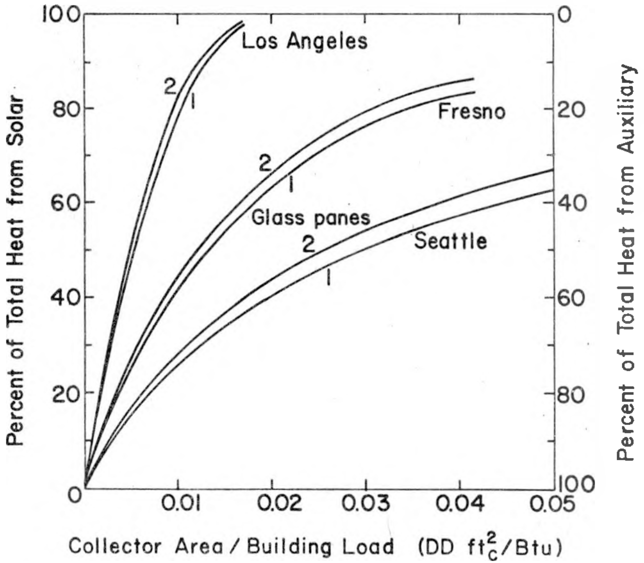
The effect of double glazing is shown on the graph. The major effect of the glazing of two sheets of glass is to reduce the collector heat losses. However, it also reduces the sunlight reaching the absorber surface and increases the collector cost. For heating applications, it is doubtful that two sheets of glass would be cost effective.

An example of the use of this graph can be made for the case of a house with a heating load of 20,000 BTU per degree-day. If it is desired to achieve 75% solar heating fraction in Fresno, then one can read from the graph for one sheet of glass in Fresno, a requirement of 0.0285 as the ratio of the collector area to the building load, therefore the required collector area is equal to 0.0285 x 20,000 = 570 sq ft.

This graph indicates the importance of location on the collector area required to achieve a given solar heating fraction and also indicates the importance of minimizing the building load, since the collector area required is directly proportional to the building load.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR AREA AND THE NUMBER OF SHEETS OF GLASS FOR VARIOUS LOCATIONS



THE EFFECT OF COLLECTOR TILT

Collector tilt angle is an important design consideration. The optimum angle is roughly 55° for Fresno, somewhat greater than the oft-quoted rule-of-thumb of latitude plus 15°. Fresno is at a latitude of 36.8°. It is important to note that the curve has a relatively flat maximum. This means that relatively major deviations from the optimum tilt have only a minor effect on the performance.

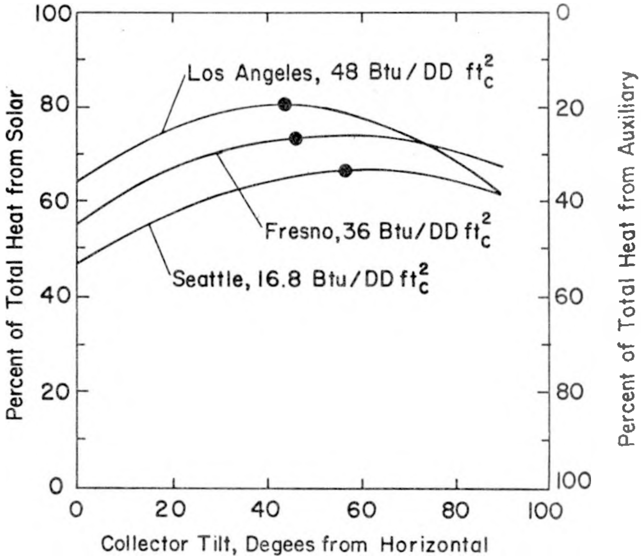
Many other considerations may play a more important role than maximizing the performance--such as ease of assembly and repair, shedding of snow and rain, architectural integration, and difficulty of overheating in summer. A vertical collector may well be best in many situations.

REFLECTORS

A reflector may be used to increase the solar radiation incident on the collector. Los Alamos has started to analyse reflectors (following on the work of McDaniels) but the results are not yet in publishable form. Clearly the potential advantage of reflectors is substantial. For example, a vertical collector with a horizontal reflector in front should outperform the same collector tilted at the optimum angle but without a reflector. Of course, the seasonal variation of incident energy is completely changed.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR TILT FOR VARIOUS LOCATIONS AND BUILDING LOADS



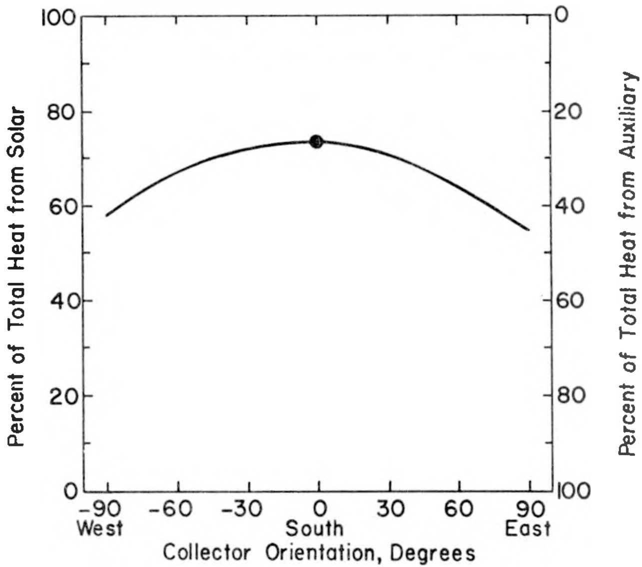
THE EFFECT OF COLLECTOR ORIENTATION

Collector orientation is also important. As expected, a due south orientation is optimum; however, variations of 30° east or west only reduce performance about 3%.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR ORIENTATION

(LATITUDE + 10° = 46.8° TILT)



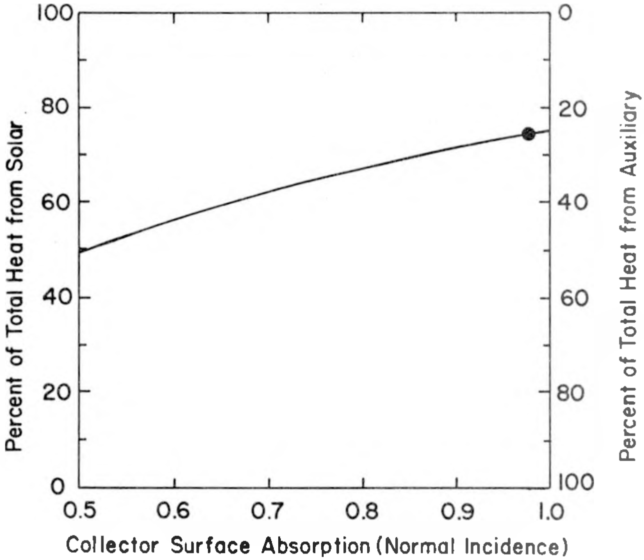
THE EFFECT OF COLLECTOR SURFACE ABSORPTANCE

The effect of collector surface absorptance is very important to system performance because solar energy which is not absorbed by the surface is not available for heat energy in the system. One should simply not accept a surface absorptance less than about 90%. It is interesting to note that a percent decrease in surface absorptance only decreases the overall system performance by about 0.54%. The reason that this decrease is less than one to one is because the system runs somewhat cooler with the lower absorptance and this decreases heat losses in the rest of the system.

Because of the strong dependence on surface absorptance, nearly all solar collectors are painted black. Even among black paints, one finds variations in absorptance from 0.92 to 0.98. The differences between these two values can be easily discerned by the naked eye. In a surface with absorptance of 0.98 it is almost impossible to discern any relief variations in the surface, whereas it is relatively easily to see such relief variations in a surface with an absorptance of 0.92. Of course one must be very concerned about the durability of the surface as well as its absorptance.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR SURFACE ABSORPTANCE



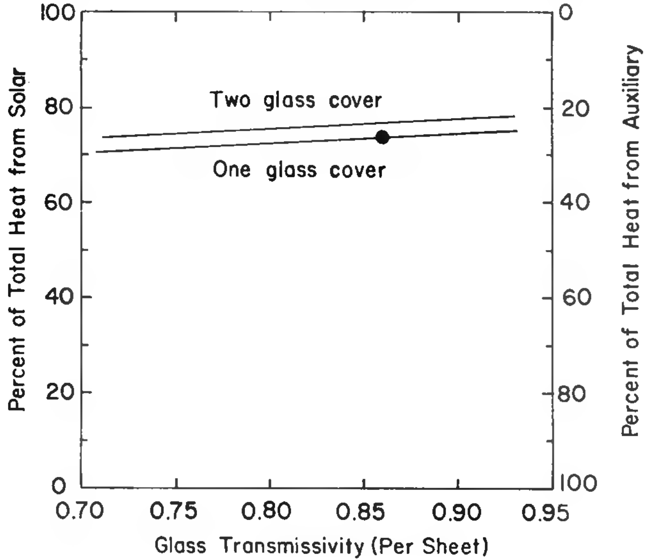
EFFECT OF GLASS TRANSMISSIVITY

Solar energy is lost in the collector glazing assembly due to two effects. One effect is the reflection of sunlight from the surfaces of the glass. The second effect is absorption of solar energy in the glass. In normal window glass, these effects result in degradations of comparable magnitude. The effect of reflections is very dependent on the angle of incidence of the solar radiation, and to a lesser extent, the absorptance is also dependent on this angle of incidence.

It has become customary to lump these effects into a single variable called "glass transmissivity" which is the total energy transmitted for sunlight which is perpendicular to the surface. This parameter is relatively easy to measure. For standard plate glass the reflection accounts for a reduction in transmissivity of about 8% at normal incidence and the absorption accounts for a reduction in transmissivity of about 6%. However, the latter is highly dependent on the iron content of the glass, and so-called white glass may have a much lower absorption.

Techniques for making glass with a total transmissivity in excess of 95% are known. Surface reflection can be reduced by etching the surface of glass to provide a gradation in density at the glass surface. This etching is hardly visible to the naked eye. The absorption is reduced by reducing the iron content of the glass. These processes are not in commercial production at the present time and it is not possible to buy glass with this high performance.

"Standard" Liquid System in Fresno
EFFECT OF GLASS TRANSMISSIVITY
FOR ONE-AND TWO-GLASS COVERS



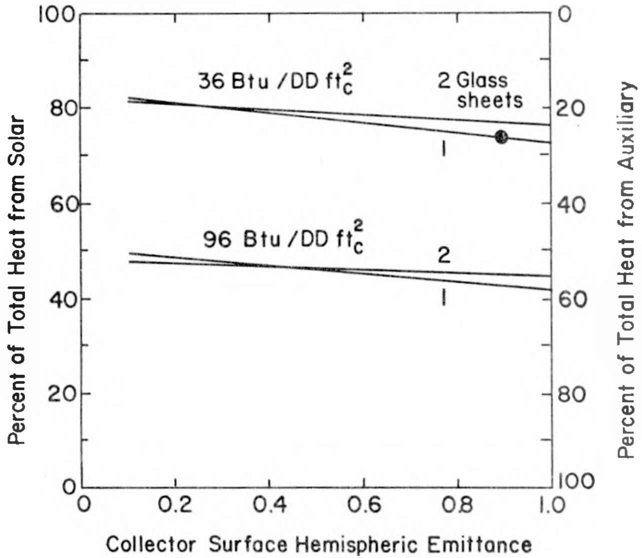
THE EFFECT OF COLLECTOR SURFACE EMITTANCE

A major heat loss in flat-plate collectors is the long-wave radiation from the collector surface. Normal collector coatings have a high effective emittance for long-wave or infrared radiation from the surface. A value of 0.89 is typical for a good black paint with a high absorptance.

There has been intense interest in the development of so-called "selective surfaces." These coatings have the characteristics of having a very high absorptance for the visible and UV solar radiation and a low emittance for the long-wave, infrared radiation. Most selective surfaces are either chemical coatings or electrodeposited coatings. Thin coatings of metal oxides tend to be selective when applied over a high reflectance substrate. The oxide coating is opaque for short-wave radiation, having therefore a high absorptance. and simultaneously transparent for infrared radiation. Therefore the effective emittance of the surface is the same as the effective emittance of the substrate, which is low since the surface is highly reflective. As can be seen from the graph, the effectiveness of a selective surface is higher for a single-glazed collector than for a double-glazed collector. For a single glazed collector a substantial performance increase can be obtained using a highly selective surface. For selective surfaces currently available effective emittance in the range of 0.1 can be obtained with an absorptance in the range of 0.95.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR SURFACE EMITTANCE FOR DIFFERENT LOADS AND NUMBER OF SHEETS OF GLASS



Whether or not a selective surface should be used depends on several factors. Cost of course is a major consideration. For example, a selective surface with an absorptance of 0.92, and an emittance of 0.1 will decrease the required collector area by about 22% compared to the use of a black-surface with an absorptance of 0.98, and an emittance of 0.89 (for a single-glazed collector).* The question then becomes whether the array of black surface collectors represents a larger total installed system cost than an array of selective surface collectors of 78% of the area of the black surface collectors. Other considerations may be the available roof area and the appearance of the collectors. Another major consideration in the use of a selective surface should be the durability of the surface. Some selective surfaces have shown a tendency to both mechanical and chemical degradation over a period of time.

It is interesting to note that an extra sheet of glass offers no performance advantage for the case of the collector with a selective surface.

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*The nominal operating point is above the knee of the collector performance curve. Because of this a small change in percent solar translates into a large change in collector area.

THE EFFECT OF HEAT EXCHANGER HEAT TRANSFER EFFECTIVENESS

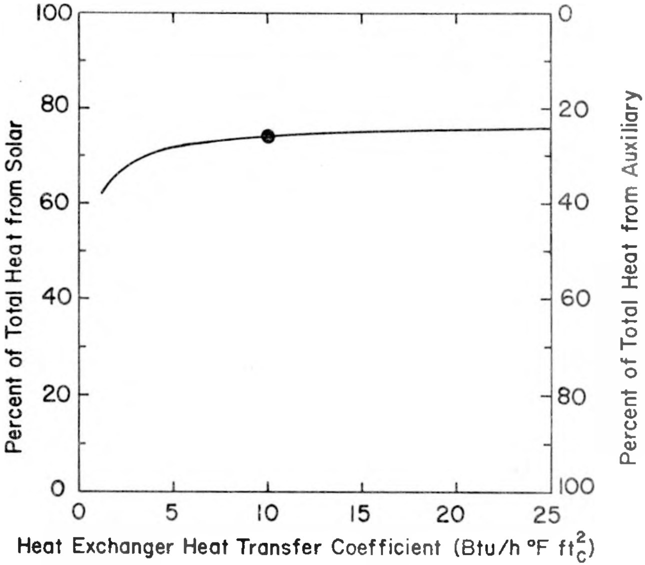
In the standard liquid system, a heat exchanger is used between the primary collector coolant loop and the water tank storage. There are two possible ways of doing this heat exchange. The most common method is to use a separate heat exchanger. In this case, the collector coolant is pumped through one side of the heat exchanger and the water is pumped with a separate pump out of the storage tank through the other side of the heat exchanger and back into the storage tank. Another common approach is to immerse a heat exchanger inside the main storage tank and rely on natural convection of the storage water around the heat exchanger coils to promote heat exchange between the coil surface and the water in the storage tank. Since heat exchangers are expensive the sizing of the heat exchanger is an important design parameter.

The graph indicates that a heat exchanger effectiveness of at least 10 BTU/hr °F ft_c^2 should be achieved. For example, if the total amount of energy being collected at a particular time is equal to 100 BTU/hr ft_c^2, then this heat exchanger temperature difference from one side of the heat exchanger to the other side of 10°F.

The use of some collector coolants with poor heat transfer characteristics such as the paraffinic oils can result in relatively difficult problems in achieving this kind of heat exchanger effectiveness. On the other hand, water is a relatively good heat transfer fluid and it is not very difficult to achieve good heat transfer on the water side of the heat exchanger.

"Standard" Liquid System in Fresno

EFFECT OF HEAT EXCHANGER, HEAT TRANSFER EFFECTIVENESS

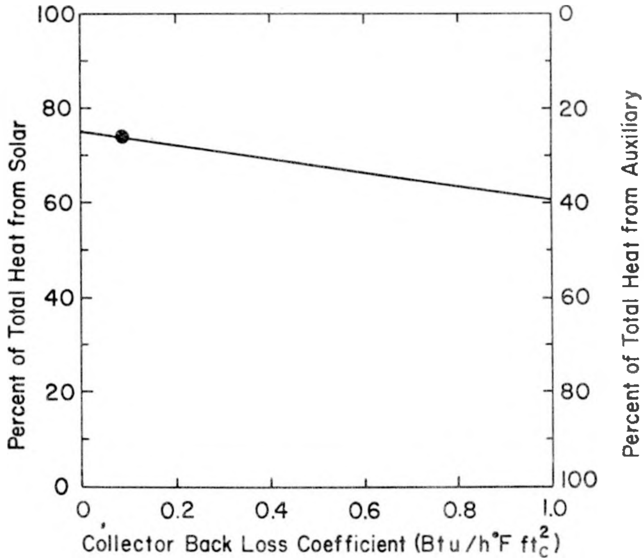


THE EFFECT OF COLLECTOR BACK INSULATION

Heat can be lost from the collector surface by conduction out through insulation on the back of the collector as well as conduction heat losses out to the edges and ends of the collector. It is good design practice to insulate with the equivalent of one to two inches of foam insulation on the collector back and insulating around all edges of the collector. The net effect of back insulation is shown graphically. These data indicate that U values 0.2 should be integrated into the collector design for optimum performance.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR BACK INSULATION



THE EFFECT OF COLLECTOR COOLANT FLOW RATE

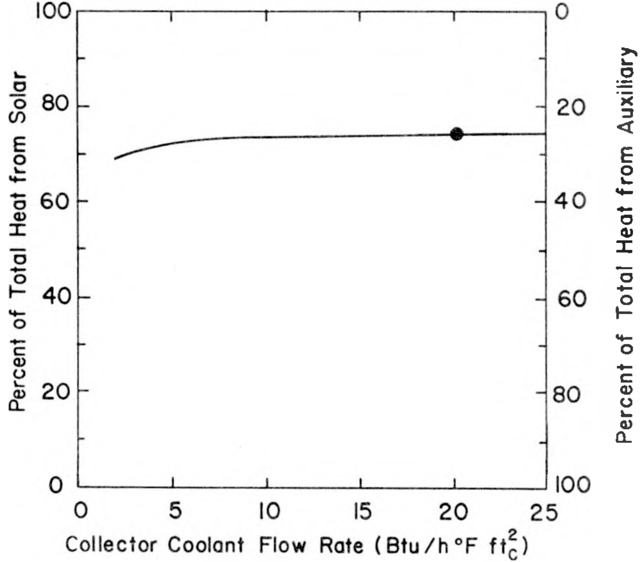
Collector coolant flow rate is important in a liquid cooled collector primarily because increased collector flow rate reduces the collector temperature rise and thereby reduces the amount of heat which is lost from the collector for a given inlet fluid temperature.

The curve is plotted as a function of the coolant flow rate in units of BTU/°F ft_c^2. The reason this is chosen is so that the curve can be used for different fluids with different specific heats. For example, if water is the coolant, which has a specific heat of 1 BTU per lb, then a value of 15 on the plot corresponds to 15 lb per hr which is equal to 1.8 gallons of water/hr ft_c^2. A fluid with a specific heat of one half that of water would require a flow rate of 30 lb/hr ft_c^2.

The proper choice of coolant flow rate will depend on the pressure drop of the collector, the viscosity of the fluid, and the size and cost of pumps available to pump the fluid. Although higher coolant flow rates do increase the performance of the solar heating system, it also requires more energy to pump the fluid and therefore increases in flow rate may not represent a net increase overall in energy gain or cost savings for the system.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR COOLANT FLOW RATE

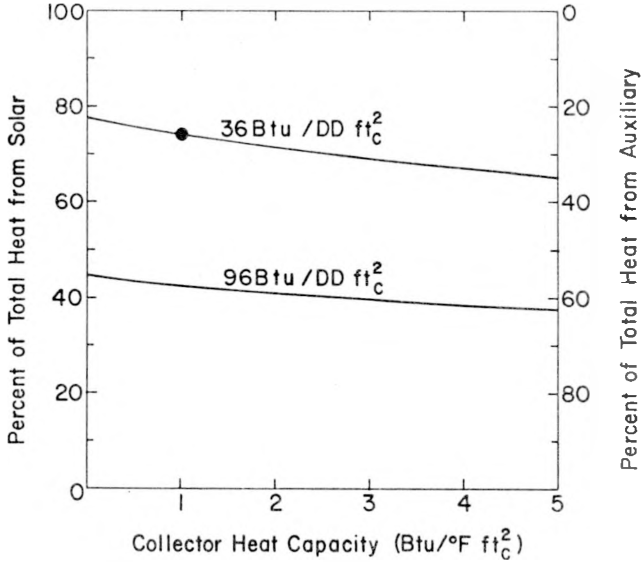


THE EFFECT OF COLLECTOR HEAT CAPACITY

Collector heat capacity decreases the performance of a collector array because solar energy is required to heat the collector up every morning before it is turned on. This energy is not recovered at the end of the day, but is lost to the environment. The same effect happens each time the sun goes behind a large cloud during the day if there is not sufficient diffuse energy to keep the collector at a temperature higher than the storage temperature. The major contributors to collector heat capacity are the fluid that is stored in the collector, the mass of metal in the collector, the mass of insulation associated with the collector surface, and the insulation connecting to the collector. The magnitude of this heat capacity depends on the collector design. The graph shows that this heat capacity can have a major effect in the performance of the overall system.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR HEAT CAPACITY FOR DIFFERENT LOADS



THE EFFECT OF COLLECTOR HEAT TRANSFER COEFFICIENT

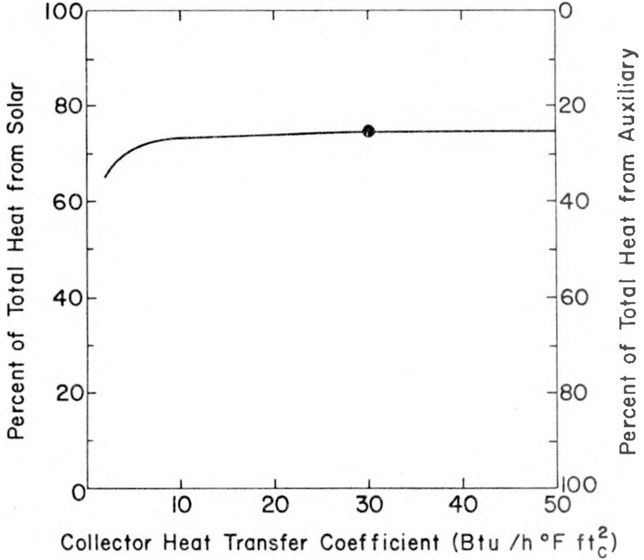
The average collector heat transfer coefficient is the effective heat transfer coefficient between the average surface temperature of the collector and the average coolant temperature. Due to the heat transfer properties of liquids the heat transfer coefficient of liquid cooled collectors is relatively high. Many liquid cooled collectors are designed with a thin member of metal between the collector tubes.

Conduction down this thin member represents a decrease in the effective transfer coefficient of the collector.

It can be seen from the chart that an effective heat transfer coefficient greater than 10 BTU/hr ^{o}F square foot of collector is adequate to achieve near-maximum performance from the solar heating system.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR HEAT TRANSFER COEFFICIENT

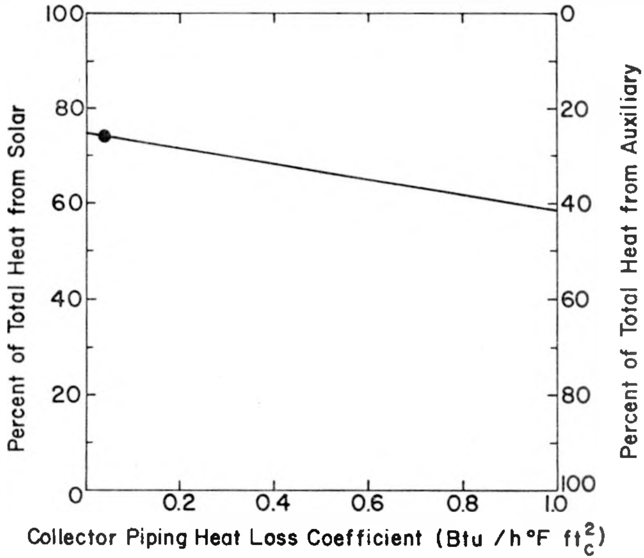


THE EFFECT OF COLLECTOR DISTRIBUTION PIPE INSULATION

The plumbing which connects the collector array to storage should be insulated. The minimization of heat losses from the pipe on the hot side of the collector are even more important than the minimization of heat losses from the collector itself since the hot side piping is at the highest temperature in the entire system. As in the case of the other parameters, the pipe insulation is put in terms of the total insulating characteristics per square foot of collector area. Since the distribution pipes are usually relatively small (2 inches in diameter or less), these pipe losses can be kept relatively low. A nominal value of only 0.04 BTU/hr/°F/sq ft of collector is chosen for the standard calculation.

"Standard" Liquid System in Fresno

EFFECT OF COLLECTOR DISTRIBUTION PIPE INSULATION



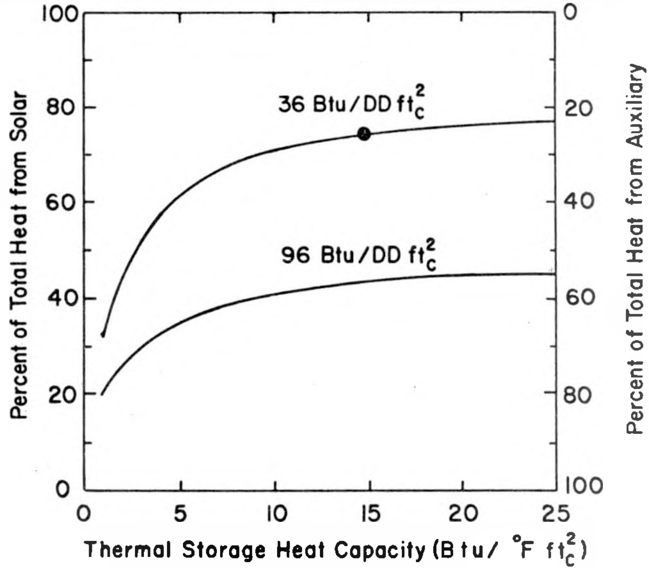
THE EFFECT OF THERMAL STORAGE HEAT CAPACITY

Thermal storage is desirable in a solar heating system to provide energy during those times when the sun is not shining - at night and during cloudy periods. The nominal thermal storage capacity assumed is 15 BTU/ft_c^2 °F (1.8 gal of water/ft_c^2). This is sufficient to fully heat the building only for 5.9 hrs assuming an initial storage temperature of 173°F and an outside temperature of 0°F. After this time energy would continue to be extracted from storage but an increasing amount of auxiliary heat would be required to maintain an inside temperature of 68°F.

The simulation analysis indicates that thermal storage for more than one day of poor weather does not improve the yearly system performance very much. However, fairly severe performance losses are predicted if the storage mass is less than 10 BTU/ft_c^2 °F (1.2 gal of water/ft_c^2). Extra storage mass is relatively inexpensive and provides some extra feeling of security.

"Standard" Liquid System in Fresno

EFFECT OF THERMAL STORAGE HEAT CAPACITY



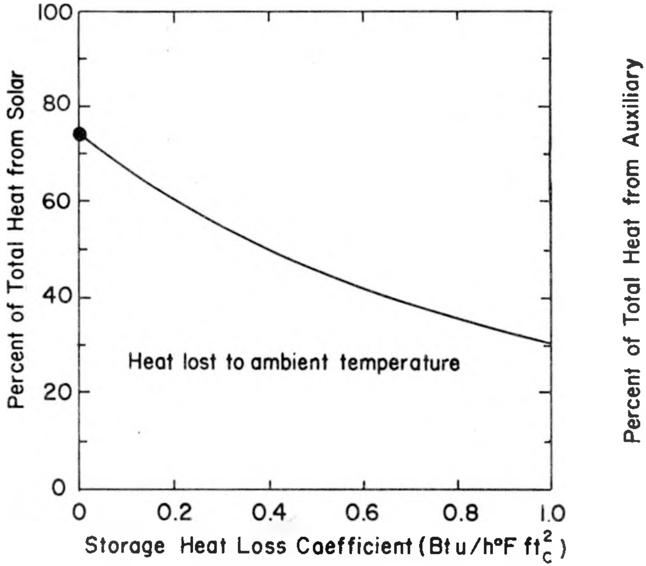
EFFECT OF HEAT LOSSES FROM STORAGE

Storage can be located in a variety of places with respect to the space to be heated. The best situation results when the storage is located within the space to be heated. In this case all heat lost from storage simply goes into the heated space and there is no net penalty to the system performance due to heat lost from storage. Even in this case the storage tank should be insulated because there will be times when the space does not require heat and heat lost from storage would be detrimental.

The second case is one in which the storage tank is located outside of the heated space and heat is lost to the ambient temperature by conduction through the tank insulation. In this case the performance degradation can be very severe because energy which is collected is simply lost to the environment. In order to utilize this graph, it is necessary to calculate the surface area of the tank, multiply that by the effective U-value of the tank insulation and divide by the collector area. This gives the storage heat loss coefficient.

"Standard" Liquid System in Fresno

EFFECT OF HEAT LOSSES FROM STORAGE



A finned-tube coil is used to transfer heat from the solar heated water to the building air. The air flow requirement of the building is a function of the design water temperature. As this design temperature is decreased, the required air flow is increased and the solar heating system performance is increased--at a higher capital and operating cost. The graph shows the performance part of this tradeoff.

The building air flow can be determined from the following equations, where CFM is the cubic feet per minute for the building. The building load is calculated as indicated on page 22, the design ΔT is the difference between 68°F and the minimum outside air temperature for which the building heating system is sized, and T_{conv} is the design air distribution temperature.

The coil effectiveness factor (CEFF) was assumed to be 0.80:

$$T_{conv} = T_{rm} + CEFF\ (T_{dw} - T_{rm})$$

$$CFM = \frac{(LOAD)(DESIGN\ \Delta T)}{(24)(1.08^{*})(T_{conv} - T_{rm})}$$

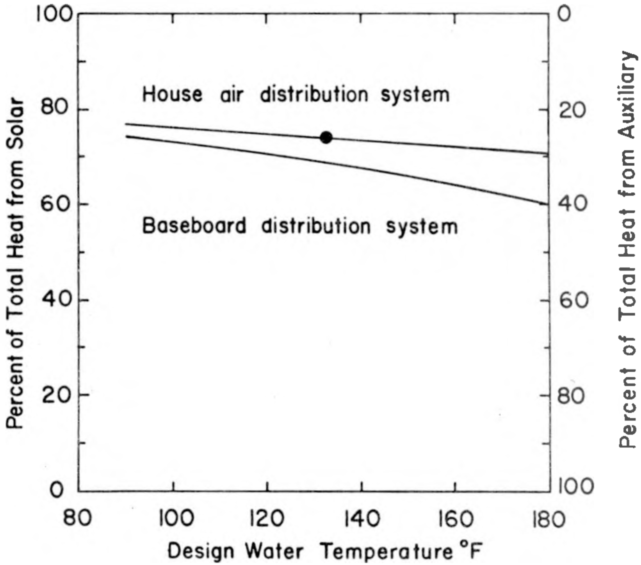
T_{rm} = room temperature

T_{dw} = design water temperature

*This is $\rho c_p \times 60 = 1.08$ BTU/CFM/°F/hr in Fresno, CA.

"Standard" Liquid System in Fresno

THE EFFECT OF DESIGN WATER TEMPERATURE



Note that a less efficient coil would increase the required house air flow but overall system performance is only a function of the design water temperature.

Calculations were also made with a baseboard hot water distribution system. Normal baseboard systems are designed with inlet water temperatures of 160 to 200°F. The baseboard convector system should be oversized to operate at lower temperatures available from the solar storage tank. The baseboard curve on the figure is a function of the design water temperature needed to meet the load at the design ΔT. At higher outside air temperatures a lower baseboard temperature could provide the load. The heat output of the baseboard heating system simulation is assumed to vary as the 1.176 power of the water-to-room temperature difference. This power law has been determined based on the manufacturer's data.

If the house heating load was higher than the amount of heat available at solar storage tank temperatures,then the system is switched to total auxiliary heat.

DESIGN PARAMETERS OF THE STANDARD AIR HEATING SYSTEM

Values of parameters used for the "standard" solar heating system using air-heating solar collectors, rock bed thermal storage, and a forced air heat distribution system to the building. The values are normalized to one square foot of collector (ft_c^2).

Solar Collector		
Number of glazings	1	
Glass transmissivity (at normal incidence) (at normal incidence)	0.86	(6% absorption, 8% reflection)
Surface absorptance (solar)	0.98	
Surface emittance (IR)	0.89	
Back insulation U-value	0.83	BTU/hr °F ft_c^2
Air flow rate	2	SCFM/ft_c^2
Heat capacity	0.5	BTU/°F ft_c^2
HA**	4	BTU/hr °F/ft_c^2
Collector Duct Work		
Heat loss coefficient (to ambient)	0.1	BTU/°F hr ft_c^2
Thermal Storage		
Heat capacity	15	BTU/°F ft_c^2
Heat loss coefficient (i.e. assuming all heat loss is to heated space)	0	BTU/hr °F ft_c^2

DESIGN PARAMETERS
(CONTINUED)

Heat Distribution System		
Air flow rate	2	$SCFM/ft_c^2$ forced air distribution
Controls		
Building maintained at 68°F	36	$BTU/DD\ ft_c^2$
Collectors on when advantageous	Standard	building load

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*These values apply for normal incidence on ordinary double strength glass (1/8 in.). For other angles of incidence the Fresnel equation is used.

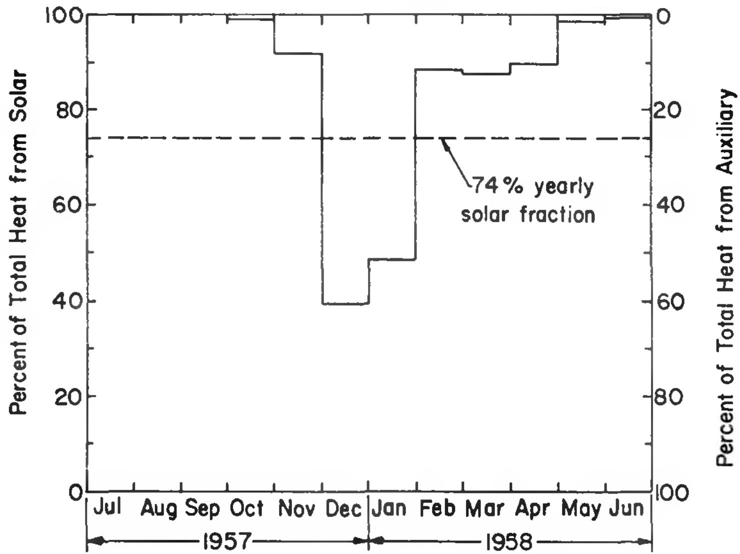
**The heat transfer effectiveness, HA, is the product of the effective heat transfer coefficient times the effective heat transfer area to the air coolant. It is normalized to 1 sq ft of collector (ft_c^2).

SIMULATION RESULTS FOR THE "STANDARD" AIR SYSTEM IN FRESNO

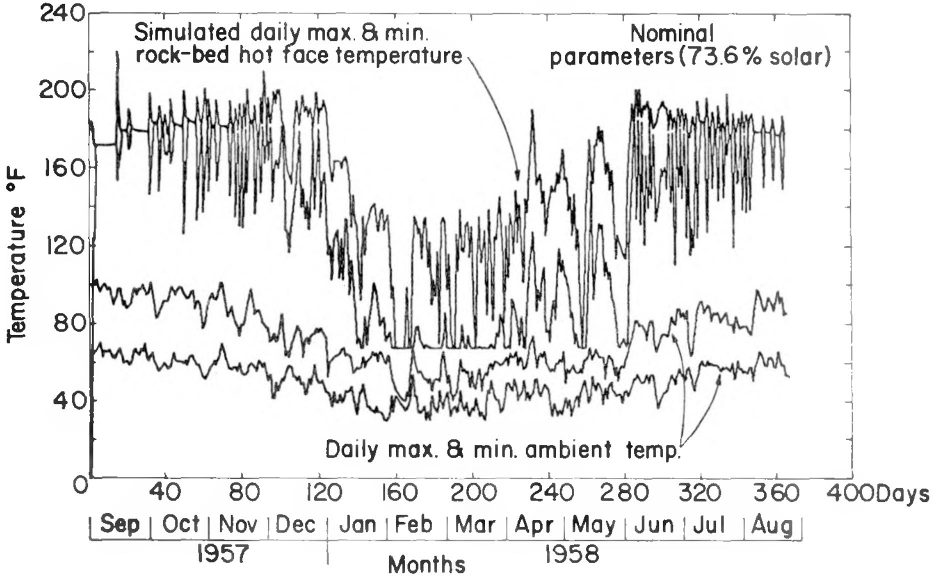
The nominal design parameters of what shall be called the "standard" air-heating collector system are shown on the preceding pages. The nominal values of the parameters were chosen using the computer simulation.

The computer simulation results for the "standard" air system in Fresno for the years 1957 to 1958 have been analyzed. On the following pages are shown a plot of the fraction of solar heating by month, and a plot of the daily rock bed storage temperature and ambient temperature fluctuations for the "standard" air heating collector system.

"Standard" Air System in Fresno
SOLAR HEATING FRACTION BY MONTH



"Standard" Air System in Fresno
SIMULATION RESULTS



EFFECT OF CHANGES IN DESIGN PARAMETERS ON THE PERCENT SOLAR HEATING USING THE STANDARD AIR SYSTEM

On the next several pages, the effect of changes in the design parameters on the computer simulated solar performance will be examined for the "standard" air system for Fresno in 1957-58. The design parameters that are examined include the collector area, the number of glazings, the collector air flow rate (with heat transfer constant), the collector heat transfer effectiveness, collector air flow rate (with collector geometry fixed), thermal storage heat capacity, rock-bed temperature distribution, and rock bed length. The design points for the "standard" air system parameters are indicated in each figure by a black dot.

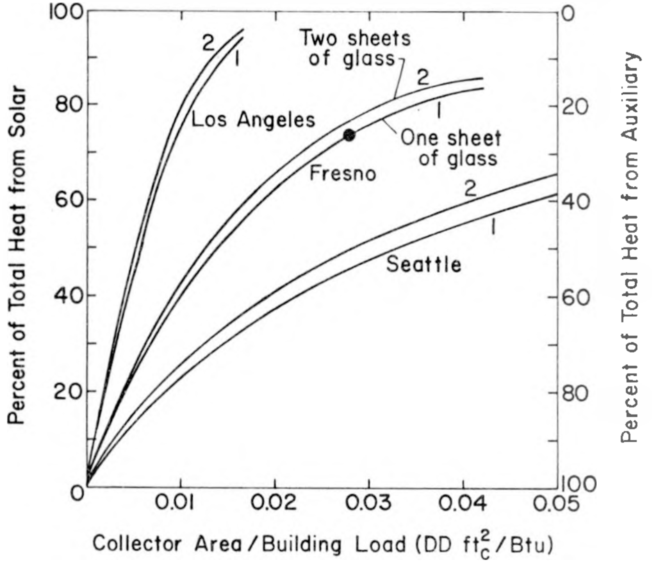
The effect of changes in design parameters that are common to both the liquid and air "standard" system (e.g. collector tilt and orientation) have been discussed already. The effect of changes in these parameters are similar for the standard air and liquid systems.

EFFECT OF COLLECTOR AREA AND EFFECT OF NUMBER OF SHEETS OF GLASS

The effects of collector area and of number of sheets of glass which were described on page 68 for the liquid system, are almost identifical for the air system.

"Standard" Air System in Fresno

EFFECT OF COLLECTOR AREA AND THE NUMBER OF SHEETS OF GLASS FOR VARIOUS LOCATIONS



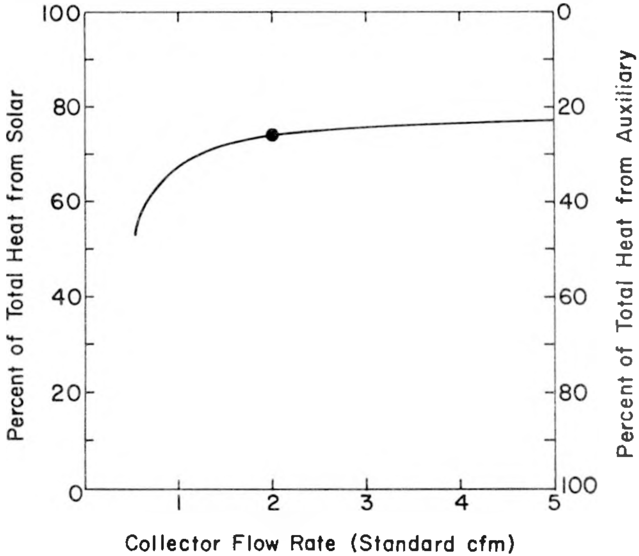
EFFECT OF AIR FLOW RATE, COLLECTOR HEAT TRANSFER HELD CONSTANT

Air flow rate is a very important parameter because as the air flow decreases the collector ΔT increases and collector efficiency decreases. A severe performance penalty will result at air flow rates below 1 $Scfm/ft_c^2$.

Note that the variation in air flow rate is made assuming that the parameter HA is constant at a nominal value of 4. Since H is dependent on flow rate in a collector of fixed geometry, holding HA constant implies changing collector geometry as air flow rate is changed.

The effect of changing the flow rate with a fixed collector geometry, which means that the type of heat transfer is changing with flow rate, is given on page 112.

"Standard" Air System in Fresno
EFFECT OF AIR FLOW RATE,
COLLECTOR HEAT TRANSFER HELD CONSTANT



EFFECT OF COLLECTOR HEAT TRANSFER EFFECTIVENESS

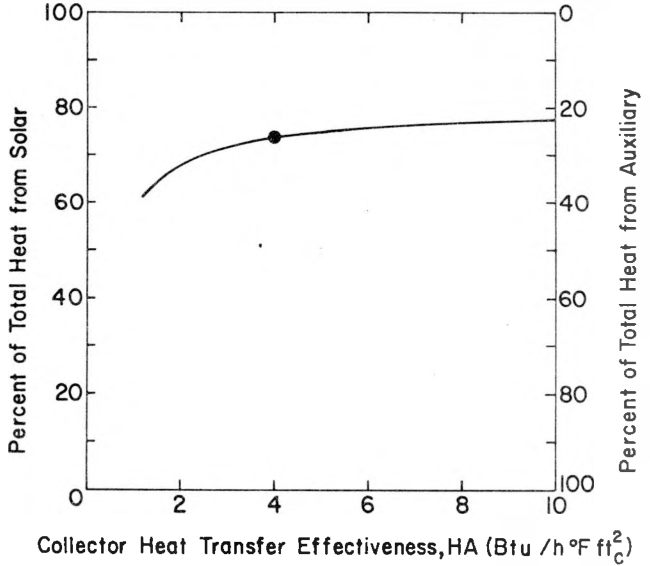
The other important collector parameter is the heat transfer effectiveness, HA. This is the product of the heat transfer coefficient times the heat transfer area divided by the collector area. The area, A, can be increased by adding fins or making all sides of the flow passage effective for heat transfer. The heat transfer coefficient can be increased by increasing the flow velocity and decreasing the flow channel size.

Note that the variation in HA is made assuming that air flow rate is constant at the nominal value of 2 cfm/ft_c^2.

The effect of collector heat transfer effectiveness in an air system is so important that it is extremely desirable to test the heat transfer characteristics of the collector before committing a final collector design for an installation. Too often systems have been built with inadequate heat transfer. Even with a heat transfer effectiveness, HA, equal to the nominal value of 4 BTU/h°F sq ft of collector, the net temperature difference between the collector surface and the average air temperature would be 25°F assuming a total collected energy of 100 BTU/hr/sq ft/°F.

"Standard" Air System in Fresno

EFFECT OF COLLECTOR HEAT TRANSFER EFFECTIVENESS



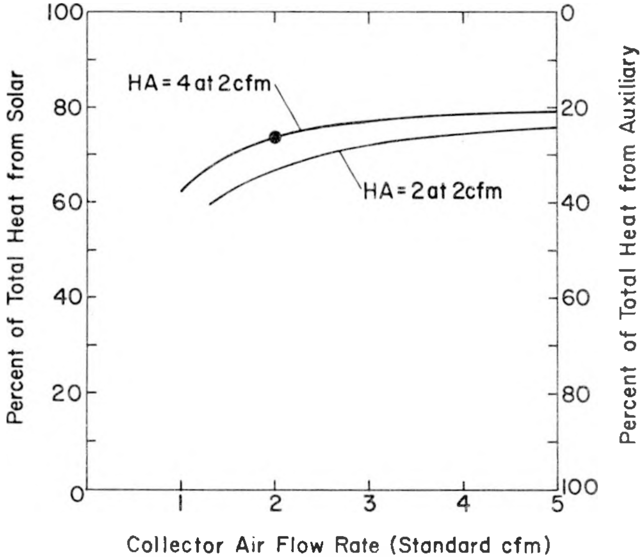
EFFECT OF AIR FLOW RATE, COLLECTOR GEOMETRY FIXED

In a collector of fixed geometry, the heat transfer coefficient H is dependent on flow rate, and so is the heat transfer effectiveness, HA. As air flow rate is changed, HA varies as $(CFM)^{0.8}$ for a fixed geometry.

The plot shows two curves of HA varying with air flow rate, with HA = 4 and 2, respectively, at an air flow rate value of 2 CFM.

"Standard" Air System in Fresno

EFFECT OF AIR FLOW RATE, COLLECTOR GEOMETRY FIXED

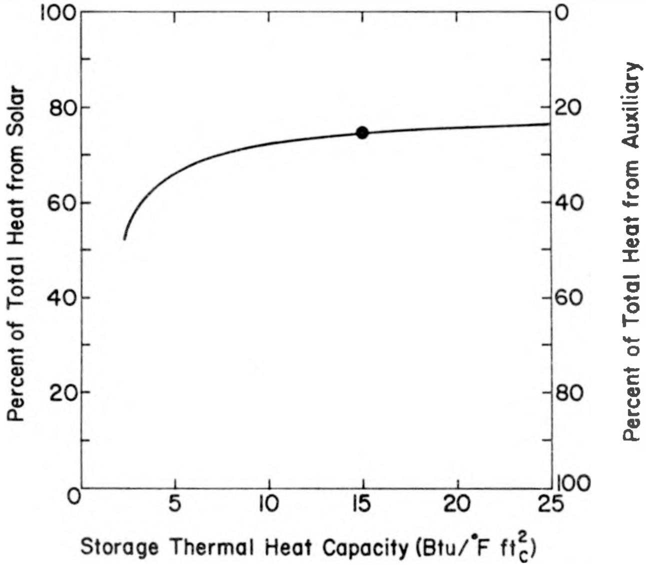


EFFECT OF STORAGE THERMAL HEAT CAPACITY

The effect of the size of the storage thermal capacity is not very important beyond a value of about 10 BTU/°F ft_c^2 corresponding to 25 lbs of rock/ft_c^2. The nominal value of 15 BTU/°F ft^2 corresponds to 71 lbs of rock/ft_c^2. This requires roughly three times the volume of water storage or about 1.0 ft^3/ft_c^2. Most rock has a specific heat of about 0.21 BTU/lb °F, a density of about 165 lb/ft^3, and packs with a void fraction of about 0.42 if the rocks are all roughly the same size.

"Standard" Air System in Fresno

EFFECT OF STORAGE THERMAL HEAT CAPACITY



OPERATION OF A ROCK BED

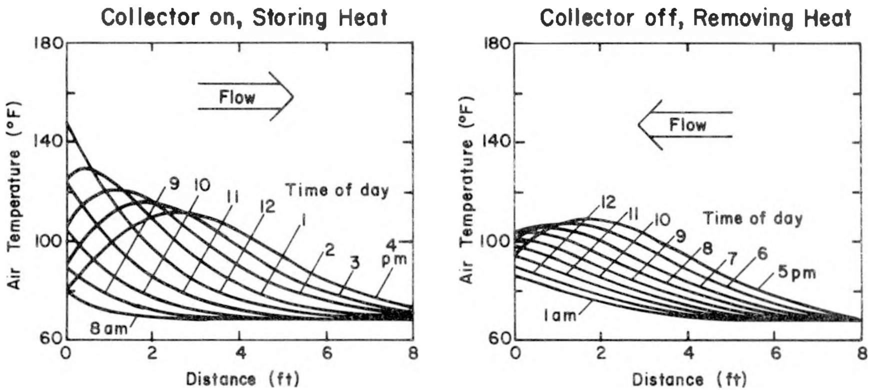
A rock bed is an efficient heat transfer device. The air quickly gives up its heat in flowing through the labyrinthine path. As a result the rocks near the air entry end of the bed can be at quite different temperature than rocks near the air exit end. This time dependent spatial temperature distribution must be accounted for in the simulation analysis.

A simplified illustration is given on the plot. We assume the bed is cold in the morning. As the day progresses the air temperature from the collector rises, peaks at noon, and decreases in the afternoon. The resulting air temperature distributions in the rock bed are shown at different times during the day. Note that the exit air from the bed is always cool and therefore the collector operates at high efficiency all day.

In the evening, when the collector is off and the building needs heat, the air flow direction through the bed is reversed so that the air exits from the hottest part of the bed. (If the air flow were not reversed, then one would have to wait hours to move the heat through the bed and even then would get only moderately warm air.) As the evening progresses the temperature out of the bed rises until 8:00 o'clock and then falls, resembling the time profile of the inlet temperature during the day, but in reverse.

In the simulation analysis, this detailed temperature distribution is determined each hour of the year by dividing the

ROCK BED TEMPERATURE DISTRIBUTIONS



rock bed into eleven axial zones and calculating the temperature of each zone. The hot side temperature plot on page 105 is for the air temperature at the bed frontal face corresponding to the right hand side of the rock bed shown in the diagram on page 34.

For a fixed bed volume the length can be varied by varying the frontal area. Generally speaking, a rock bed which has a short air flow path (and a large frontal area) is preferable to a bed which has a long air flow path (and a small frontal area) because of pressure drop considerations. If the air flow length is decreased below a value of about 12 rock diameters, then a performance penalty is incurred because the spatial temperature distribution benefits described above become ineffective.

ROCK BED PERFORMANCE MAP

The rock bed heat capacity and air flow rate, can be related to the real parameters of interest to the builder -- rock size, bed length, and pressure drop -- through a performance map.

Such a performance map is given on page 121. The bed pressure drop and the air pumping energy loss in the bed are plotted as a function of bed length divided by rock diameter. Lines of constant rock diameter and lines of constant bed length are drawn on this plot. Nominal parameters are chosen for the air flow rate and the storage mass heat capacity to plot these isovalue lines.

For a case of a 6 ft long rock bed (in the direction of air flow) and a rock bed length to rock diameter ratio of 36, the pressure drop is 0.03 in of water corresponding to a pumping energy loss in rock bed friction of 0.05 BTU/ft_c^2 hr or 0.02 watts/ft_c^2. This number is quite small even after factoring in fan and motor inefficiences.

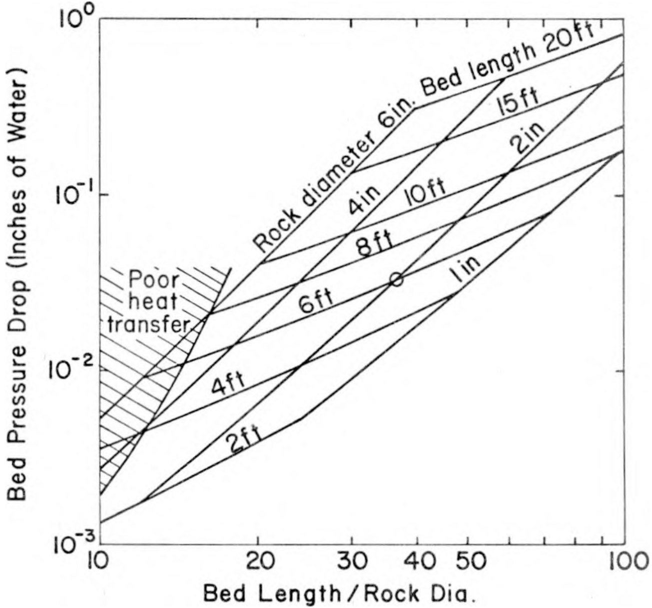
Rocks of uniform size should be used to avoid small rocks filling in the interstices between the large rocks and thereby increasing the bed resistance to air flow.

Since the bed length is not important beyond a value of about 12 rock diameters, the builder has great flexibility in arranging the rock bed within the structure.

The heat transfer coefficent and pressure drop correlations for rock beds were taken from the following reference:

R. V. Dunkle and W.M.J. Ellul "Randomly-Packed Particulate Bed Regenerators and Evaporative Coolers," Mech. & Chem. Eng. Trans. I.E. Aust., Vol MC8, No. 2, (1972), pp 117-121.

ROCK BED PERFORMANCE MAP

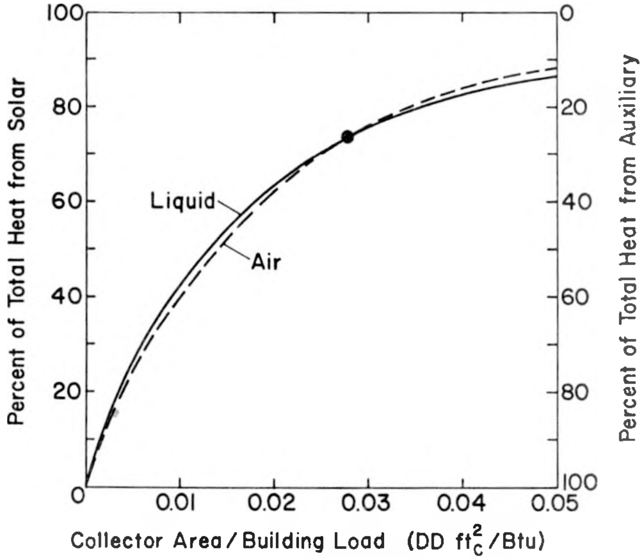


COMPARISON OF AIR AND LIQUID SPACE HEATING

This plot shows a comparison between the performance of the air/rock/air and a liquid/water/air system as a function of collector-to-building area ratio based on nominal parameter selections for each system. Clearly there is little to choose between the two, based on performance alone. One could change the nominal parameters slightly to give a small performance edge to either system.

Thus the actual choice between these competing system design concepts will probably rest on considerations other than pure performance. These other considerations may be cost, ease of installation, maintainability, convenience, availability, philosophy, or prejudice. In any case, the system should be designed in accordance with good engineering principles to yield an optimum, or near optimum installation based on all considerations.

COMPARISON OF AIR AND LIQUID SPACE HEATING SYSTEMS IN FRESNO



PROS AND CONS OF LIQUID AND AIR SPACE HEATING SYSTEMS

Some points of comparison between liquid and air systems will be raised and discussed here. The list is probably not complete, but perhaps will give an individual a basis for choosing between the two types of solar space heating systems:

Performance: As mentioned, when properly designed, air-heating collector systems and liquid heating collector systems have nearly identical performance.

Cost: Air systems may offer cost advantages over liquid solar heating systems. However, the reverse may be true in retrofit situations. The collectors and storage of liquid systems are more expensive because they must be leak-proof. Air is a free cooling fluid for air systems whereas the coolant in a liquid system adds a small cost. The duct work for air systems and the piping for liquid systems are both major expenses.

Ease of Installation: Air collectors are more easily built than liquid collectors with conventional construction techniques and materials. Air also avoids leak-proofing in the installation. The air system duct work size may cause installation difficulties, particularly in retrofit situations. The smaller storage volume required makes retrofitting liquid system installation easier than air/rock installation. Domestic hot water systems may be more easily integrated in a water system than an air system.

Maintainability: Air systems would have less maintenance associated with them than liquid systems, where for example, corrosion and liquid degradation would have to be monitored.

Convenience: The air system does not have the potential problem of messy leaks, cooling fluid degradation, or corrosion. However, the air system may create a noise problem, due to the air fans. Also, the air system is not as adaptable to a solar cooling system as is the liquid system.

Availability: Solar heating systems using air-heating collectors and rock-bin heat storage were among the earliest systems built, but have received little attention during the recent rebirth of interest in solar energy. Hence, liquid-heating systems and components are currently more readily available than air heating systems and components. Air systems can be expected to proliferate as their good performance becomes more widely known.

DOMESTIC HOT WATER HEATING

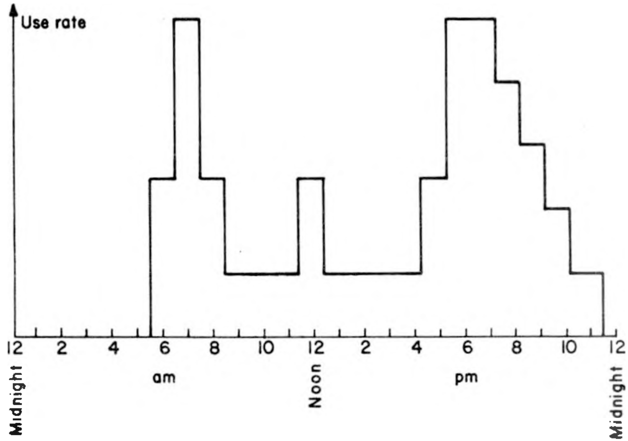
Compared to space heating, domestic hot water heating requires a relatively small amount of energy and therefore requires a correspondingly smaller solar collector array and storage tank. Although not explicitly considered in the simulation analysis, domestic hot water heating is a natural and almost universal add-on to a space heating system of either basic type.

Liquid cooled collector designs enjoy a natural advantage in a situation where only domestic hot water is generated: there is no air-to-liquid transfer required anywhere within the system. A liquid-to-water heat exchanger is considered desirable to avoid a water cooled collector and the associated problems of freezing, corrosion and scaling.

The thermal load is quite different for water heating than for space heating. An assumed profile of hot water demand shown on the opposite page was deduced based on personal experience and estimation. The simulations were run for this profile. The profile is assumed to be the same for every day of the year.

The nominal design parameters for the collector are the same as those given for the liquid system on Page 53 . Since the storage tank is relatively small, the heat loss from the tank surface is relatively larger than for a space heating system and is explicitly accounted for in the analysis. A tank surface of 0.5 ft^2/ft_c^2 is assumed with a tank insulation heat loss coefficient of .083 $BTU/h°Fft_c^2$.

ASSUMED USE PROFILE FOR DOMESTIC HOT WATER



ONE-TANK DOMESTIC HOT WATER SYSTEM

For the one-tank system, the solar-heated storage tank is fired by auxiliary sources to maintain the water temperature at a minimum of 120°F. A nominal storage was assumed equal to 15 lbs (1.8 gal) of water per ft^2 of collector.

TWO-TANK DOMESTIC HOT WATER SYSTEM

For the two-tank system, the solar-heated storage tank acts as a source of preheated water for the second tank, a conventional, fired, domestic hot water tank. A control scheme was adopted in which auxiliary heat is added to the second tank as necessary to maintain the storage temperature at 120°F.

As before, a nominal thermal storage heat capacity was chosen for the solar storage tank equal to 15 lbs (1.8 gal) of water per ft^2 of collector. For the second, auxiliary-fired tank, a nominal capacity equal to one-half the daily usage was chosen. For example, if the daily hot water usage is 80 gallons per day, the second storage tank would be 40 gallons.

The two-tank domestic hot-water system can be expected to perform at higher overall efficiency than the one-tank system. Since the auxiliary heat is supplied at the second tank, not at the solar heated tank, the solar-heated portion of the two-tank system can operate at a lower storage (and hence collector) temperature and higher heat collection efficiency. By proper adjustment of the solar storage tank temperature, the overall efficiency of the two-tank system can be raised over that of the one-tank system.

COMPUTER SIMULATION RESULTS

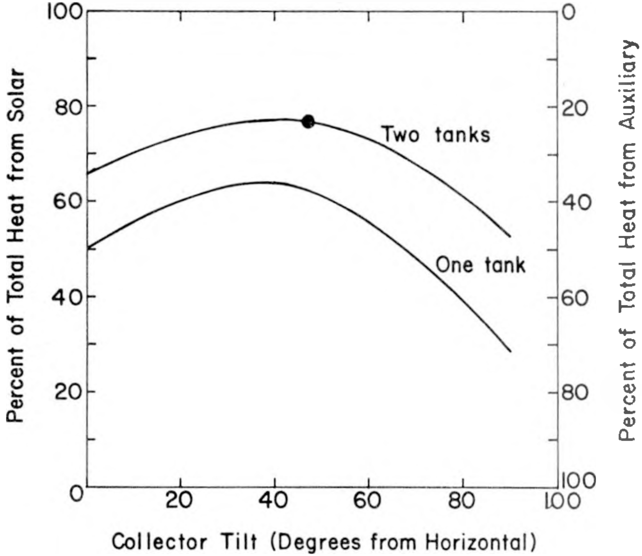
The plot shows the effect of changing the collector area for both the one- and two-tank domestic hot water systems using a liquid-heating solar collector. The plotting parameter is collector area divided by the heating load in gallons per day. The nominal parameters chosen were noted previously.

As discussed, the two-tank system outperforms the one-tank system because of the lowered operating temperatures possible for the solar-heated portion of the two-tank system.

THE EFFECT OF COLLECTOR TILT

The effect of collector tilt for domestic hot water heating is quite different than for liquid/water/air space heating because of the constancy of the load. The optimum tilt appears to be about 40°. The performance falls markedly at tilts below 20° or above 60°.

Domestic Hot Water Systems in Fresno
EFFECT OF COLLECTOR TILT

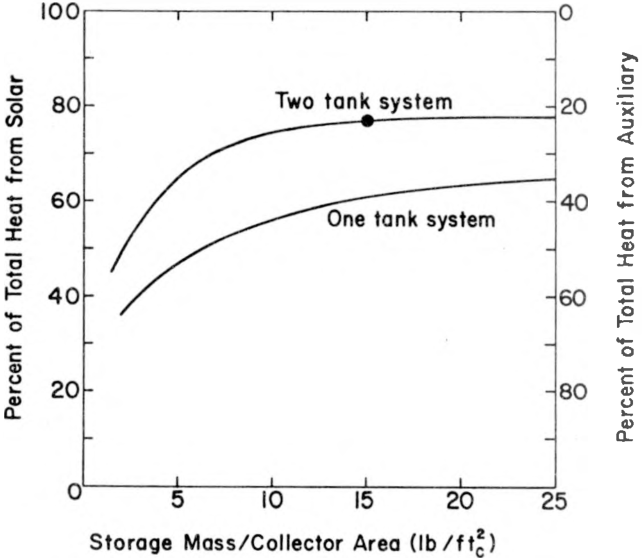


THE EFFECT OF WATER STORAGE MASS

The effect of storage mass for domestic hot water heating is very similar to the effect of storage mass in space heating system but drops off somewhat more rapidly at values below 5 BTU/^{o}F ft_c^2. At a value of 15 BTU/ft^2 ^{o}F there is nothing to be gained by further increases. The heat loss from both storage tanks is made proportional to the tank surface area by using the relation:

$$\frac{\text{Tank Surface Area}}{\text{Collector Area}} = 0.5 \left(\frac{\text{Storage } MC_p}{15}\right)^{2/3}$$

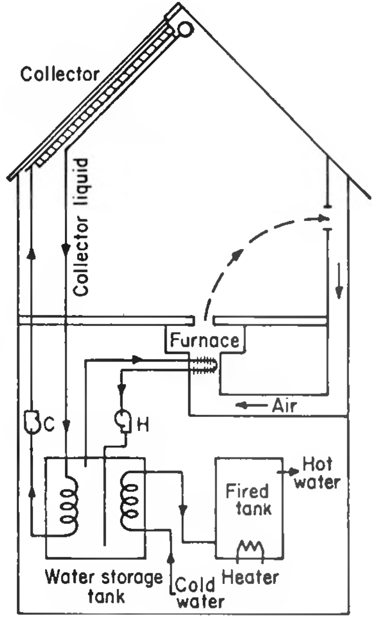
Domestic Hot Water Systems in Fresno
EFFECT OF WATER STORAGE MASS



CONNECTING DOMESTIC HOT WATER TO A SPACE HEATING SYSTEM EMPLOYING LIQUID HEATING COLLECTORS

A domestic hot water heater connected to a liquid space heating system could be of the one-tank variety, where the collector output goes through a separate heat exchanger in the domestic hot water supply tank. More feasible would be the two-tank variety, where the second tank would be a conventionally fired hot water tank using the main solar storage tank as a source of heat to preheat domestic water. The domestic water can be preheated by immersing another storage tank in the main storage tank, or by using a heat exchanger between the domestic and main storage tanks. Shown in the figure is a two-tank domestic hot water system, similar to the one on page 29, connected to a liquid-heating collector system.

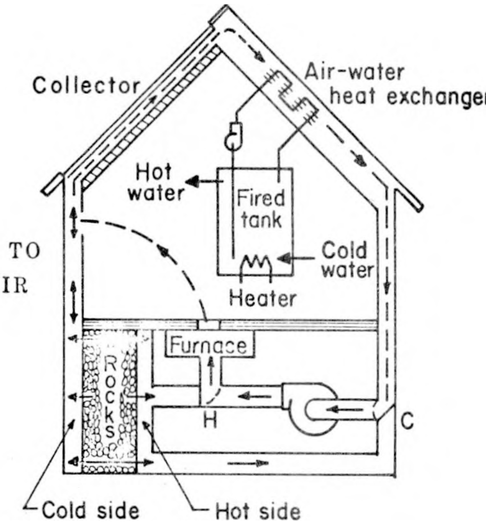
CONNECTING DOMESTIC HOT WATER TO A SPACE HEATING SYSTEM USING LIQUID HEATING COLLECTORS



CONNECTING DOMESTIC HOT WATER TO A SPACE HEATING SYSTEM EMPLOYING AIR HEATING COLLECTORS

Incorporation of domestic hot water heating into an air space-heating system would involve an air-to-liquid heat exchanger at the collector outlet, and a single-tank hot water system. The air-heating collector system would provide domestic hot-water at lower efficiencies than the liquid-heating collector system because of the collector-to-air and air-to-water heat-exchanges. On the other hand, air systems avoid the possibility of contamination of the domestic hot water system with the collector coolant liquid which may be toxic (e.g. ethylene glycol). Shown in the figure is a one-tank domestic hot water system, similar to the one on page 29, connected to an air-heating collector system.

CONNECTING DOMESTIC HOT WATER TO A SPACE HEATING SYSTEM USING AIR HEATING COLLECTORS



The required size of a solar collector array depends on several important factors:

1) The building thermal load;
2) The climate (solar radiation and ambient temperature);
3) The fraction of total building heat to be obtained from the solar heating system;
4) The design of the solar heating system.

Each of these must be determined before the solar collector size can be determined.

Usually the only reliable basis for predicting the yearly performance of a solar system is an hour-by-hour computer simulation analysis of the entire system. LASL has performed such computer simulation analyses for a total of 25 U.S. and Canadian cities for up to 12 years of weather data for each city. On the basis of these calculations, a simplified method has been devised which will allow a designer to predict the system performance based on only monthly data of horizontal solar radiation and heating degree-days. The method has been validated by comparison with the hour-by-hour computer simulations.

Two types of solar heating systems have been analyzed in detail: the first type employs the "standard" liquid-heating collector system; the second type employs the "standard" air- heating collector system. Initially the "standard" liquid system was chosen for analysis to devise a simplified method to predict system performance. The simulation results for the air system are so close to those for the liquid system that the same simplified method can be used for either one.

The simplified method is a five-step process which can be performed on a hand calculator with the aid of trigonometric and exponential tables (or a hand calculator which has these functions). The steps are as follows:

Step 1) Obtain the following data for each month of the year for the site where the solar heating system is to be located.

a) Heating degree-days, per month.

b) Total solar radiation on a horizontal surface, per month.

Step 2) Correct the solar radiation to a collector for tilt angle of latitude-plus-ten-degrees using the approximate formula on the following page:

$$\begin{pmatrix}\text{Total monthly} \\ \text{radiation on} \\ \text{tilted surface,} \\ \text{BTU/ft}^2\end{pmatrix} \approx 1.025\, Y - 8200,$$

where

$$Y = \frac{\text{total monthly radiation on horizontal surface, BTU/ft}^2}{\cos(\text{latitude - solar declination at mid-month})}$$

$$\begin{pmatrix}\text{solar declination} \\ \text{at mid-month}\end{pmatrix} \approx 23.45^{\circ} \cos\,(30M-187)$$

M = month, (January = 1, December = 12)

Step 3) Determine the building thermal load in units of BTU/degree-day. This is the total heat required by the building per day for a one degree Fahrenheit difference between the inside and outside temperatures. (See pp. 22-23.)

Step 4) Determine the "solar load ratio" (SLR) for each month from the following formula:

$$SLR = \frac{\begin{pmatrix}\text{Solar Collector}\\ \text{Area}\end{pmatrix} \times \begin{pmatrix}\text{Total Radiation on}\\ \text{Tilted Surface}\end{pmatrix}}{\begin{pmatrix}\text{Building Thermal}\\ \text{Load}\end{pmatrix} \times \begin{pmatrix}\text{Heating Degree-days}\\ \text{per month}\end{pmatrix}}$$

The solar load ratio is dimensionless. It is the ratio of the total solar energy incident on the collectors to the total energy required to heat the building.

Step 5) The annual solar heating fraction can then be estimated from the following formula:

$$\begin{matrix}\text{Annual}\\ \text{Solar}\\ \text{Heating}\\ \text{Fraction}\end{matrix} = \frac{\sum_{mo.=1}^{12} (\text{Degree-Days})(X)}{\sum_{mo.=1}^{12} (\text{Degree-Days})}$$

where

$$X = 1.06 - 1.366\, e^{-.55\, SLR} + .306\, e^{-1.05\, SLR}, \text{ (for } SLR < 5.66\text{)}$$

$$X = 1, \text{ (for } SLR > 5.66\text{)}$$

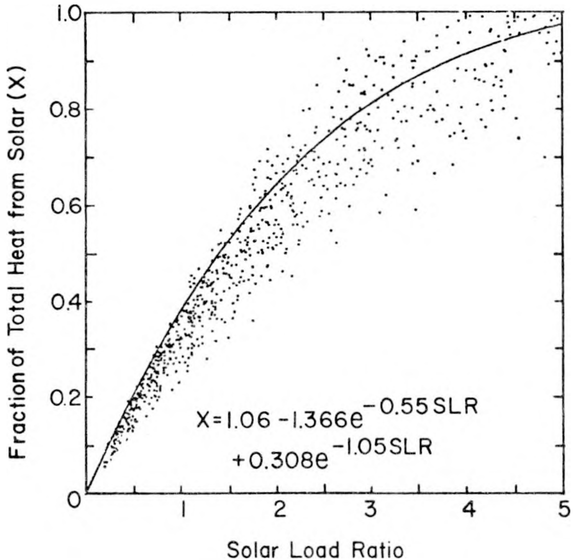
The function X is shown plotted on the opposite page. Note that X is not a very good estimate of the monthly solar heating fraction because it is designed to compensate (on an annual basis) for the fact that the heating load based on degree-days per month is lower than the heating load based on hourly calculations.

TEST OF THE SIMPLIFIED METHOD

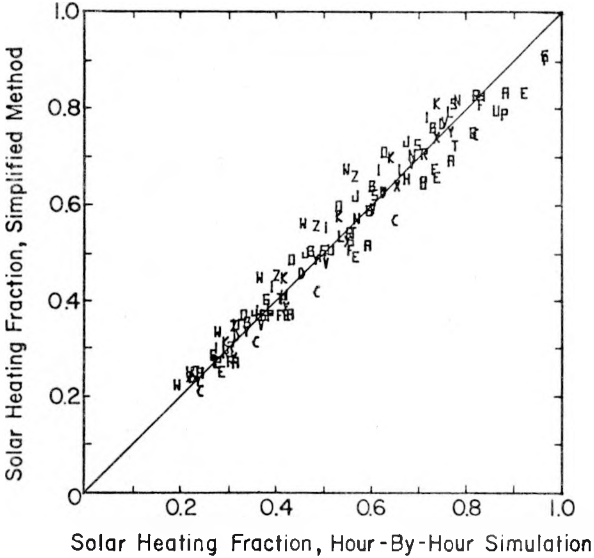
The key to the accuracy of the method is the determination of the function X used in Step 5. This function has been carefully determined so that the resulting error in predicting the solar heating fraction (compared to the simulation result) will be minimized without using a different function for each locality.

The determination of X is based on hour-by-hour computer simulations of a "test year" for 25 locations for five different collector sizes in each location. The average error resulting from the simplified method is essentially zero and the root-mean-square error (standard deviation of the prediction error) is 4.4 percent solar heating. The next figure shows the predicted versus simulated results for the 125 cases studied, and the following table identifies the cities corresponding to the symbols in the figure.

THE FUNCTION X FOR SIMPLIFIED METHOD



TEST OF SIMPLIFIED METHOD



PLOTTING SYMBOLS FOR CITIES EXAMINED IN THE TEST OF THE SIMPLIFIED METHOD

Plotting Symbol	City	Plotting Symbol	City
A	Albuquerque, NM	N	Nashville, TN
B	Boston, MA	O	Ottawa, Ontario
C	Charleston, W, VA	P	Phoenix, AZ
D	Dodge City, KS	R	Fort Worth, TX
E	El Paso, TX	S	Seattle, WA
F	Fresno, CA	T	Tallahassee, FL
G	Los Angeles, CA	U	Santa Maria, CA
H	Lake Charles, LA	V	Vancouver, BC
I	Madison, WS	W	Winnipeg, Manitoba
J	Fredericton, N.B.	X	Medford, OR
K	Bismark, ND	Y	New York City, NY
L	Lincoln, NE	Z	Edmonton, Alberta

As mentioned previously, the challenge confronting passive solar energy design is not one of heat collection, since a window is an efficient solar collector, but one of storage and control of the heat so as to maintain suitable comfort standards within the building.

The paradox is simply put. Sensible heat storage requires a temperature change in the storage medium and yet the objective of building thermal control is to maintain constant temperature. If the storage is to be a part of the living building, how can these be made compatible? One approach is to minimize the inconsistency by brute force - that is, to use such a large mass heat capacity that the temperature variations are tolerable. A second approach is to insulate storage from the room.

The control of the heat to the house is the other difficult problem in passively heated buildings although some moderately effective controls have been devised. Control of the incident sunshine can be achieved by movable shading devices. Control of night-time heat loss can be achieved by movable insulation. Control of heat flow from a storage wall can be achieved by opening natural ventilation ports.

The study of passive solar energy utilization brings together two disciplines - architecture and engineering. There is a great interest and level of activity in sun-tempered concepts among architects and new ideas have far outpaced the development of thermal criteria to guide their use. But there has been little quantitative assessment of the buildings which have been built.

Some elements of passively solar heated building design which should be considered from both the architectural and engineering viewpoints are tabulated on the following pages. The use of these building design elements requires both qualitative and quantitative evaluation.

ELEMENTS OF PASSIVELY SOLAR-HEATED BUILDING DESIGN

Element	Application
Radiation:	In cold periods solar radiation is absorbed directly or indirectly by the thermal storage mass of the passive system. The energy is removed by radiation and convection from the storage mass to heat the interior of the building when the sun is not shining. In warm periods solar radiation is blocked from the storage mass and heat from the air is transferred to the storage mass during the day. Radiation and convection are used to remove heat from the storage mass during the night time.
Thermal Insulation:	Static insulation is used in decreasing natural energy flow to maintain building interior comfort, e.g. thermal insulation retains building interior warmth in cold environment, and coolness in a warm environment. Movable insulation can be used to deminish natural energy flow through windows.
Heat of Fusion Storage Materials:	Thermal masses store sensible heat. Heat of fusion or phase change materials offer a promising energy storage mechanism in heating. Heat of fusion materials require smaller mass and volume requirements than sensible heat

	storage for the same heat capacity. In cooling, phase change materials offer a promising thermal heat sink.
Fenestration:	Windows act as an effective solar collector. Windows can be used to admit solar radiation either to warm the structure or for lighting.
Evaporation:	Static outdoor ponds can be cooled evaporatively, and the remaining water used to cool the space. Spraying or cascading the water increases the evaporative cooling, and can have a pleasing side effect which can be architecturally integrated.
Air Stratification:	Warm air can be stratified at the ceiling and removed through vents to reduce the air-conditioning load. Alternatively, stratification can be used to concentrate warm air for use elsewhere.
Building Structural or Added Mass:	Mass provides natural thermal storage of sensible heat. Materials of high mass and low thermal diffusivity, combined with the appropriate insulation, can delay the arrival of a thermal wave until the heat can be used effectively.

(CONTINUED)

Natural Convection:	In passively heated systems, natural convection of air can be used as a heat transport mechanism (along with radiation), and to produce air movement (ventilation). Natural convection in liquid systems can be used to transport heat also, as in a thermosiphon hot water heater.
Shading:	Roof overhangs and window awnings can be designed to admit the low winter sun and block out the high summer sun. Shading can be combined with insulation in the form of drapes and shutters for windows. Natural shading can also be provided by vegetation, both deciduous and evergreen.
Reflectors:	The use of horizontal reflectors signficantly improves the heating of passive storage walls by solar radiation. In the cooling mode, high emittance exterior surfaces reflect radiation and radiate heat.

SOME PASSIVELY SOLAR-HEATED BUILDING DESIGNS

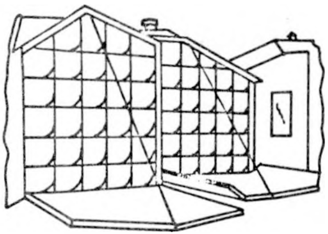
On the following pages a few passively heated, sun-tempered residences constructed by various individuals are described: the Steven Baer house, Corrales, NM; the Felix Trombe house, Odeillo, France: the David Wright house, Santa Fe, NM; and the Harold Hay house, Atascadero, CA. The operation of each of these houses is unique, and illustrates some of the alternative methods of passive solar heating and cooling that can be employed.

THE STEVEN BAER HOUSE, CORRALES, NM

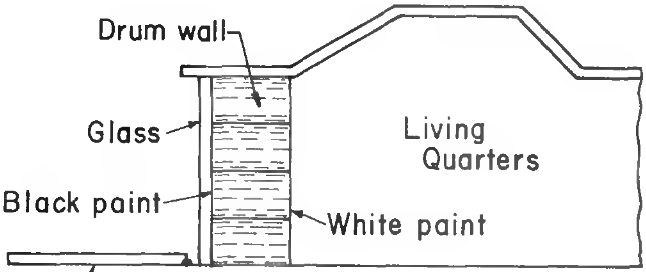
This house is located on a south facing slope in the sand hills northwest of Albuquerque, New Mexico.

The solar heating and storage system consists of "drum walls." On sunny days each south facing wall rotates about its lower edge exposing a glass wall; the inner reflecting surface of the wall (now in the horizontal postion) acts to enhance the incident radiation. Behind the glass is a floor-to-ceiling rack of 55-gal drums in a square close-packed horizontal array. The drums are filled with water and painted black on the side facing the glass. The inner portions of the drums are painted in light shades to complement the decorating scheme of the space they serve. Visually, from inside the building, the "drum walls" are interpreted as an architectural screen with the brightly lit spaces between the drums presenting a repetitive star shaped pattern. When the solar heating day is over, the wall is rotated into the closed position providing a thermal seal. The polished aluminum facing of the movable wall acts as reflective insulation to the thermal radiation.

In operation in the heating mode, the drums release heat to the space by radiation and convection. The capacity of the system is easily calculated. Allowing for filling volume, each drum will release about 418 BTU for 1^{o}F drop in the water temperature or about 7.9×10^{5} joules per degree Celsius.



South wall



Hinged insulation/reflector

THE BAER HOUSE

While the "drum wall" provides heating for most of the building, the kitchen is fitted with a Baer "sky dome," a large panel opening fitted with rotatable insulated louvers and covered by a clear plastic dome. A simple freon filled balanced control, that requires no external power, senses the direction of radiant energy flow and opens the louvers during the day and allows radiant energy to flow into the space. In the evening when the flow starts to reverse, the louvers close off the sky dome.

THE TROMBE HOUSE

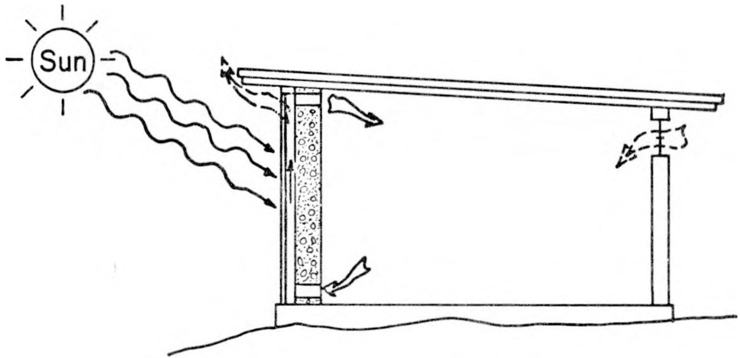
The Trombe house has its origins in the solar community near Odeillo in the French Pyrenees. Felix Trombe and his colleagues have constructed several houses following the general scheme sketched on page 159.

The basic passive element is a massive, south facing, concrete wall. The wall is painted black on the exterior surface and is provided with a double-glazed cover. There is a space provided between the wall and the glass that serves as an air passage. Other air passages are provided through the wall near the ceiling and floor levels.

In the heating mode, solar radiation strikes the black wall surface which becomes hot and starts convective air flow through the passage between the glass and the wall. Air is drawn from the floor level openings and discharged into the occupied space through the ceiling level openings. Thus, a convective loop is established, heating the occupied space during the day.

Meanwhile, the wall is slowly accumulating thermal energy as a portion of the solar heat diffuses into the concrete and is stored there. At night the convective loop is closed off and the concrete storage wall serves the occupied space as a low temperature radiant heating panel. As the wall is reradiating in the infrared spectrum, the exterior glass, which is opaque to this radiation, blocks radiant losses. There are, however, some convectional quasi-steady state transmission losses of heat to the outside environment.

In the summer the overhanging roof may be used to shade the fenestration from the direct rays of the sun. The top of the collection wall system may be vented and the convective circuit may be used to ventilate the building. This path is indicated by the dashed arrows on the sketch.



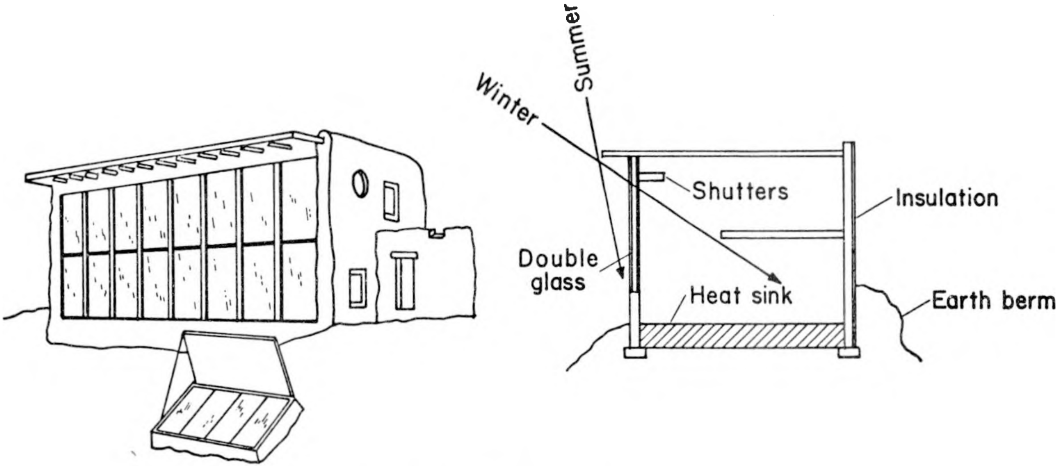
TROMBE HOUSE

THE DAVID WRIGHT HOUSE, SANTA FE, NM

This house presents a good example of a relatively extreme commitment to a total sun tempered design. Architecturally, the dwelling reads as a rather strong statement in the adobe idiom, counterbalanced by the harsh terrain of the pinon belt of the Sangre de Cristo range foothills.

The basic structure is D-shaped in plan with the flat-side facing south. This simple form is softened by additional elements at the east end that provide an entry and "air lock" that avoid excessive infiltration into the primary living space.

The building is two stories high with the living room a clerestory and the second floor characterized by rather open design. The south face of the building is a double glazed glass wall that rises from a few feet above the floor to the roof beam line and serves as the collector system. At night this glass area is insulated by an accordian-folded shutter lowered from the ceiling. The backup heating system consists of a wood-fired Franklin stove.



THE DAVID WRIGHT HOUSE

Thermal storage is provided by the adobe walls that are insulated by two inches of polyurethane applied to the outer surfaces and covered by a thick layer of plaster. The floor and perimeter are also insulated at depth, and there are some 55 gal drums filled with water embedded in the *banco* (a low seat, part of the structure, characteristic of Southwest architecture) under the south facing windows for additional mass. Domestic hot water is provided by a thermosiphon solar heater with the collector located on a south slope outside of the building.

The house has been occupied during the winter of 74/75 and is performing well. During the charging period it tends to overheat and this excess heat is disposed of by venting through windows on the second floor (included in the design for this purpose and for summer ventilation). Summer solar radiation is controlled by roof overhang on the south side.

HAROLD HAY HOUSE, ATASCADERO, CA

The Skytherm house invented by Harold Hay is a passive system consisting of shallow solar ponds lying on the metal ceiling of a flat roof house. The plastic-enclosed thermoponds (similar to waterbeds) lie between beams having trackways on which rigid polyurethane panels move horizontally to expose the thermoponds to winter sun for heating or to summer night sky for cooling. The present Skytherm system is designed primarily for new, one- to three-story flat-roof buildings in snow-free areas from West Texas to north of San Francisco. The system may be usable for other climates not having high humidity (dewpoint over 70°F).

The Skytherm concept has been tested in one-room structures in Phoenix, AZ, and more recently was evaluated in a full scale, occupied, test house in Atascadero, California expecially designed for the system. The California test house site has a more severe climate from the heating standpoint than the Arizona test room site, and indicated that it is possible to have economic solar heating and night sky cooling of houses integrally designed with the Skytherm system. It is also of significance that the occupants of the house claimed that the Skytherm system provided better comfort conditions than those generally obtained with standard gas heating and conventional home air conditioning.

The thermal performance of the Atascadero house was impressive. The system supplied 100% of the heating and cooling requirements of the building during the test months. During this time, the system was able to keep the indoor temperature between the extremes of 66°F and 74°F except during special test periods or times of prototype breakdown. Even during these exceptional periods the temperature never got higher than 79°F or lower than 62°F. The indoor temperature at the 5-ft level cycled less than 4°F daily. The vertical temperature stratification in the living space was usually less than 5°F in the winter, and less than 1°F in the summer.

The largest monthly average heating load handled by this application of the system was about 24,000 BTU/day in February. The largest monthly average cooling load handled was about 168,000 BTU/day during July. The experimental house had an overall heat transfer coefficient of about 9500 BTU/degree-day (excluding roof) and an equilibrium temperature (ambient temperature for which no heating or cooling is required) of about 62°F. The collector area was 1100 ft^2, about the same as the floor area. The roof pond average water depth was about 8.5 inches corresponding to about 6000 gal of water. The system was operated with both an unglazed and a single-glazed configuration by means of an inflatable plastic cover. Inflation was necessary in the winter months in order to keep the indoor temperature up to the reported levels. Without inflation it was estimated that the indoor temperature would have dropped to near 60°F in the early morning hours.

From a construction viewpoint the loads of the roof ponds created no major problems even on a site located in an earthquake area. The design of Skytherm is amenable to prefabrication which would allow minimal on-site labor. Problems with roof-leaks have been identified, corrected, and led to investigation of additional water-tight roof designs. The insulation panel system with an integrated automation system functioned well.

A sketch of the Skytherm-Southwest house is shown on p. 168.

SKYTHERM HOUSE, WINTER OPERATION

In the winter operating mode, the roof insulation panels are exposed to the sun on sunny days. The plastic cover of the thermoponds is inflated to act as a glazing cover to maximize solar heat collection. At night or in cloudy periods, the roof-insulation panels cover the roof thermoponds and prevent heat loss to the ambient. Heat is transferred to the interior of the house by radiation and convection from the metal ceiling in good thermal contact with the thermoponds.

In the winter mode, the thermoponds act as a large thermal storage mass with an average temperature above that of the average ambient temperature. The movable insulation behaves as a one-way thermal valve, or thermal diode, to keep the temperature of the thermoponds (and house interior) above that of the ambient.

SKYTHERM HOUSE, SUMMER OPERATION

In the summer operating mode, the roof insulation panels are closed during the day. In the summer, it was necessary to deflate the plastic cover in order to keep the living space temperature from approaching 80°F. However, better ventilation of the thermoponds and correction of panel air leaks would probably have allowed the system to have operated in the summer in the inflated configuration.

At night the roof insulation panels are exposed to the night sky for cooling and discharging the heat that was accumulated during the day. Interior heat built up during the summer day is transferred to the cooled storage thermoponds through the thermal diode effect of the movable insulation. Thus, in the summer mode, the thermoponds act as a large thermal storage mass with an average temperature below that of the ambient temperature.

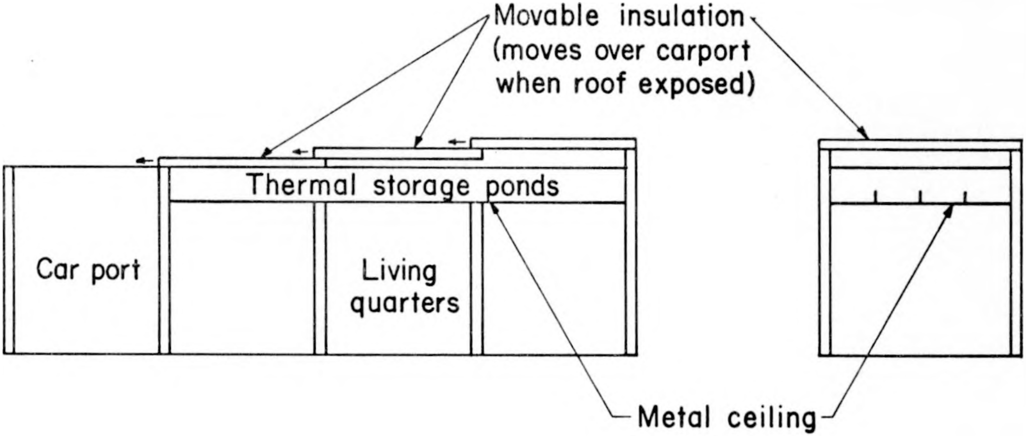
SKYTHERM HOUSE, ADAPTATIONS

The original (Southwest) version of the Skytherm principle was designed individually and primarily for new, one- to three-story flat roof buildings in snow-free areas from West Texas to north of San Francisco.

Architecturally, the system adapts well to a house typical of a majority of single-family detached houses in the U.S. The utilization of this system had no negative effect on interior space and the building aesthetics. No difficulties were encountered in getting a building permit, with inspections, or from mortgage and insurance companies in regard to construction of the test house.

Because the Skytherm system is integral with the building structure it is most suitable for new buildings designed with the system in mind. However, because of the flexbility of this system in regards to architectural design, its adaptation to other building types and needs has been explored in other designs, including two-story structures in climates requiring light thermal loads. Incorporation of the Southwest-Skytherm design into two- to three-story flat-roofed buildings would presumably involve the use of a fan to circulate interior air past thermal storage, for both heating and cooling purposes.

Skytherm designs are being developed for the south and north portions of the U.S., for places where the local humidity is not too high (dewpoint over 70°F). Presumably, high humidity would hamper radiation to the summer night-sky for the exposed thermoponds in the summer mode; this would be particularly important in the south. Also, for locations of high latitude in the north, some kind of vertical reflector would be needed to illuminate the roof thermoponds with the summer sun. Alternatively, vertical wall thermoponds could be used to supply interior heat for low-sun, winter conditions.



THE HAROLD HAY HOUSE

NOMENCLATURE

A = Area for heat transfer, square feet

CMF = Air volumetric flow rate, cubic feet per minute

C_p = Specific heat at constant pressure, BTU per (pound mass) (Fahrenheit degree)

H = Heat transfer coefficient, BTU per (hour) (square foot) (degree Fahrenheit)

L = Length, feet

M = Mass, pounds

Q = Incident solar radiation, BTU per (square foot) (hour)

T = Temperature (Fahrenheit degrees)

U = Thermal conductance, BTU per (hour) (degree Fahrenheit) (square foot)

$\dot{W}$ = Mass flow rate, pounds mass per hour

α = Solar absorptivity, dimensionless

ε = Infrared emissivity, dimensionless

λ = Rock bed relaxation length, feet

ρ = Density, pounds mass/cubic foot

Subscripts

c = collector

h = house

APPENDIX 1: LOCATION OF THE SUN IN THE SKY

It is important to be able to predict the location of the sun in the sky, relative to an observer on the ground, at different times of the day and in different seasons. This can be done with charts or with a set of straightforward trigonometric equations. The problem with the charts is that a different chart is needed for each latitude. Charts for latitudes of 32°N (roughly Tucson, AZ), 36°N (roughly Fresno, CA), 40°N (roughly Red Bluff, CA), 44°N (roughly Eugene, OR), and 48°N (roughly Seattle, WA) are given on the following pages. The charts show the position of the sun in the sky (azimuth and altitude) for different times of day and times of year.

The charts are given in terms of "sun time" which is measured relative to local solar noon -- the time when the sun crosses over the zenith. Local solar noon is usually different than local standard time. To make the correction use the equation:

SunTime = Standard Time + Equation of Time
+ 4 (Standard Meridan-Longitude)

The standard meridian for the U.S. time zones are as follows:

Eastern Time Zone	75°W
Central Time Zone	90°W
Mountain Time Zone	105°W
Pacific Time Zone	170°W

The "equation of time" is a correction for the non-uniformity of the earth rotation. It amounts to a correction of a few minutes. The "equation of time" values are as follows for mid-month (in minutes):

Jan	-8	May	+3	Sep	+5
Feb	-13	Jun	0	Oct	+14
Mar	-8	Jul	-5	Nov	+14
Apr	0	Aug	-4	Dec	+3

When calculating sun time, remember to subtract one hour from clock time to get standard time if daylight saving is in effect.

The equations used to calculate sun location for the charts use several terms which need to be defined. These are as follows:

Solar Atitude: The angle of the sun above the horizon,measured by an observer squarely facing the sun (the measurement made by a sextant).

Solar Azimuth: The rotational angle of an observer squarely facing the sun, measured from due south. (the measurement made by a compass, converted for magnetic variation)

Latitude: Latitude of the observer.

Declination: The angle between the plane of the earth's rotation around the sun and the earth's equatorial plane in a direction facing the sun.

The declination is easily calculated by one of the following equations:

$$\text{Declination} = 23.45^\circ \text{ sine } \left(360 \frac{284 + \text{day of year}}{365}\right)$$

or (less accurately) by:

$$\text{Declination} = 23.45^\circ \text{ sine } (30 \times \text{month} + \text{day} - 111)$$

The equations are as follows:

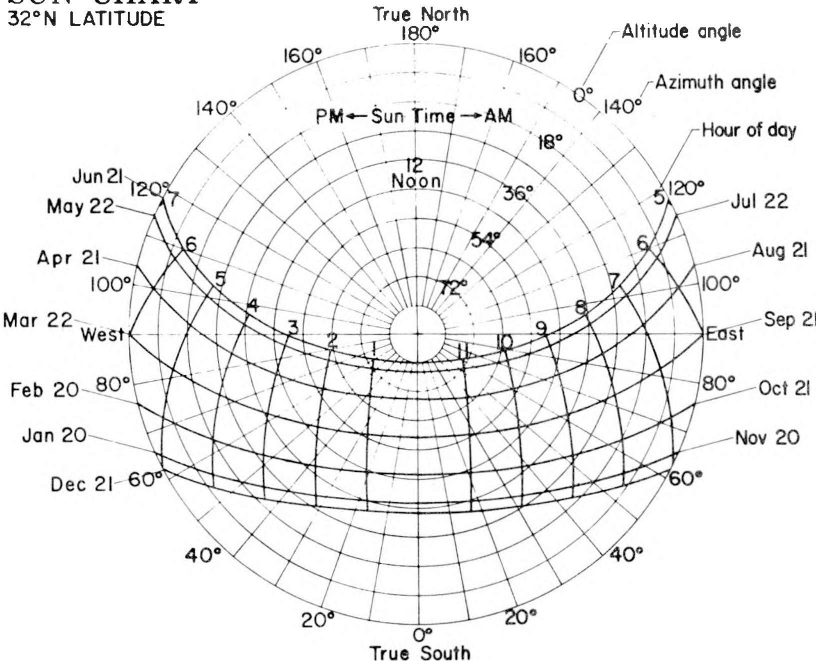
$$\text{sine (solar altitude)} = \text{cosine (declination)} \times \text{cosine (latitude)} \times \text{cosine } (15 \times \text{solar hour}) + \text{sine (declination)} \times \text{sine (latitude)}$$

$$\text{cosine (solar azimuth)} = \text{cosine (declination)} \times \text{sine } (15 \times \text{solar hour}) / \text{cosine(solar altitude)}$$

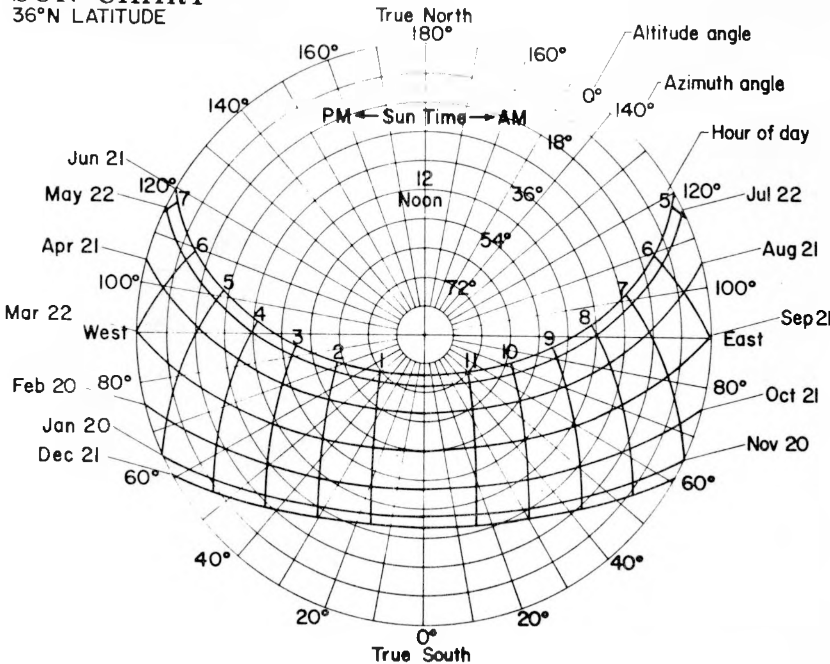
Other useful relations are as follows:

$$\text{cosine } (15 \times \text{sunset hour}) = \text{tangent (declination)} \times \text{tangent (latitude)}$$

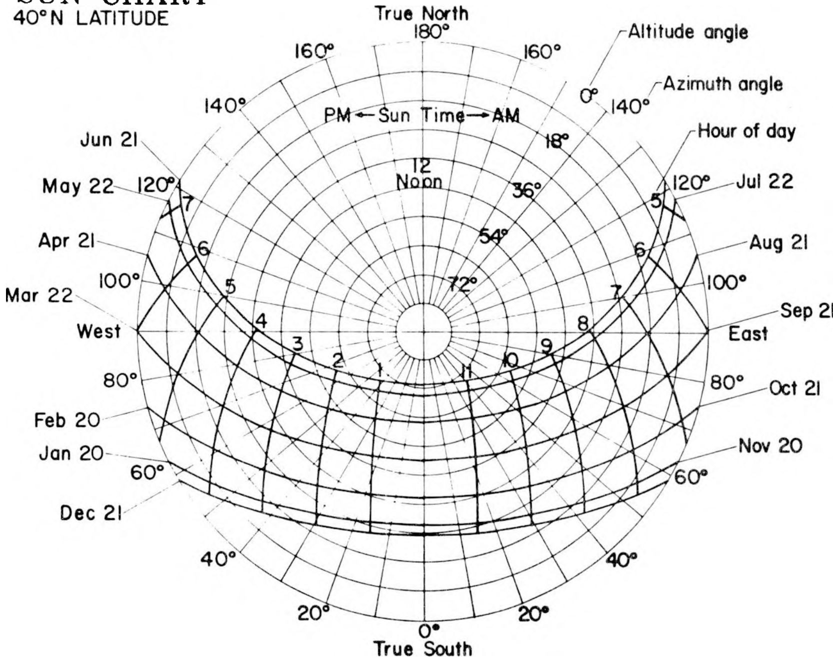
SUN CHART
32°N LATITUDE



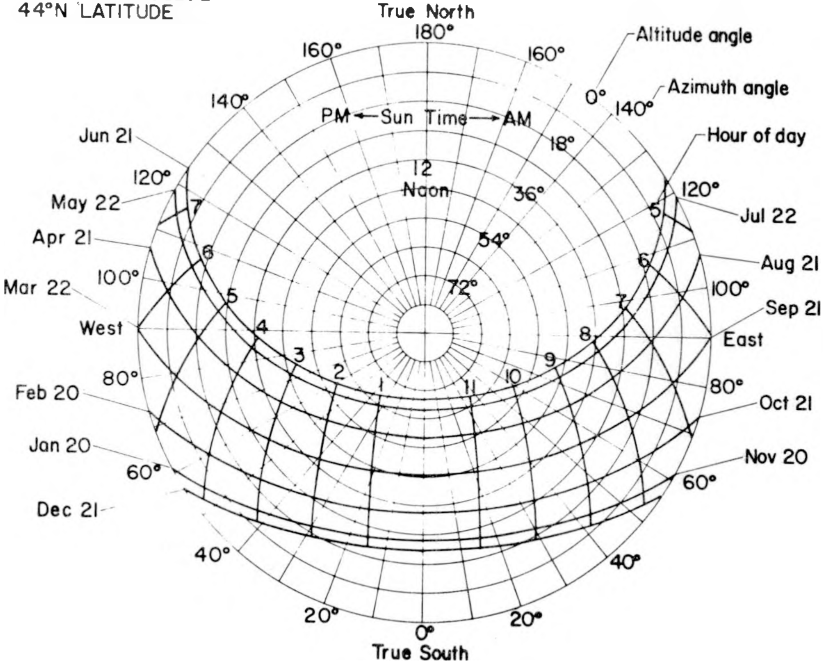
SUN CHART
36°N LATITUDE

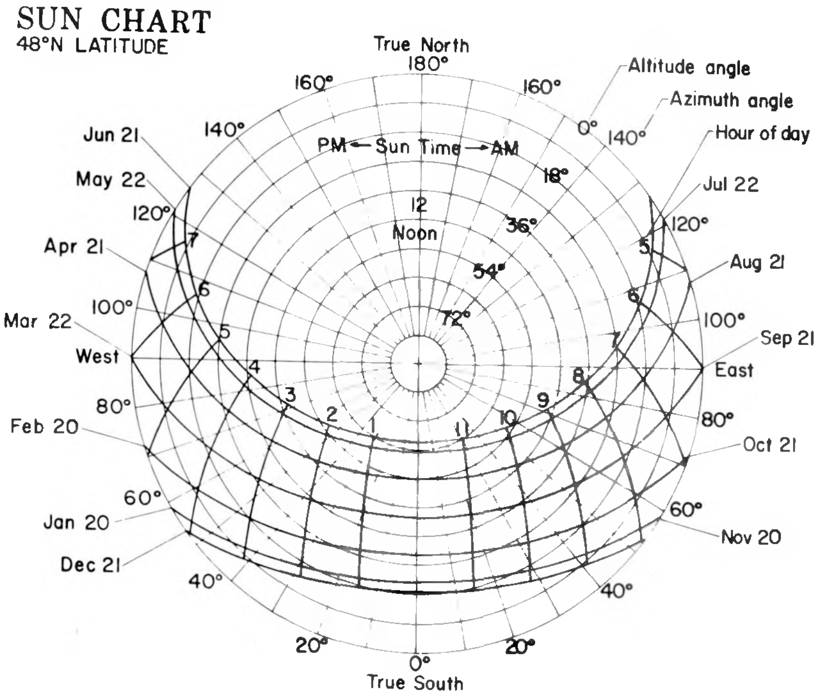


SUN CHART
40°N LATITUDE



SUN CHART
44°N LATITUDE




SUN CHART
48°N LATITUDE
True North
180°
160°
160°
Altitude angle
Azimuth angle
Hour of day
0°
140°
140°
PM ← Sun Time → AM
18°
Jun 21
May 22
Jul 22
12
Noon
36°
120°
120°
Apr 21
Aug 21
54°
100°
100°
72°
Mar 22
Sep 21
West
East
80°
80°
Feb 20
Oct 21
60°
60°
Jan 20
Nov 20
Dec 21
40°
40°
20°
20°
0°
True South

APPENDIX 2

CONVERSION TO SI UNITS

The following tables express the definitions of miscellaneous units of measure as exact numerical multiples of coherent SI units, and provide multiplying factors for converting numbers and miscellaneous units to corresponding new numbers and SI units:

To Convert from:	To Convert to:	Multiply by:
Foot^2 (ft^2)	Metre^2 (M^2)	0.09203
Pound (Mass)/Foot^3 (lbm/ft^3)	Kilogram/Metre^3 (kg/M^3)	16.01846
British Thermal Unit (BTU)	Joule (J)	1054.35
BTU/Foot^2·Hour ($BTU/ft^2 \cdot hr$)	Watt/Metre^2 (W/M^2)	3.15248
°F (Degrees Fahrenheit)	°C (Degrees Celsius)	$°C = \frac{5}{9}(°F-32)$
°C (Degrees Celsius)	°K (Degrees Kelvin)	$°K = °C + 273.2$
LBM	KG	0.45359
Inch of Water (Pressure)	Newton/Metre^2 (N/M^2)	248.84
Foot^3 (ft^3)	Metre^3 (M^3)	0.028317