Pacific Regional Solar Heating Handbook

This handbook has been prepared for ERDA,
San Francisco Operations Office,

by

Los Alamos Scientific Laboratory
Solar Energy Group
University of California
Los Alamos, New Mexico 87545

1st edition: March, 1976
2nd edition: November, 1976

Acknowledgements

This handbook has been prepared by the Solar Energy Group at the Los Alamos Scientific Laboratory, Los Alamos, New Mexico. The general direction of the project, as well as most of the technical writing, has been by J. Douglas Balcomb. The greater part of the technical analysis was performed by James C. Hedstrom with the assistance of Hugh S. Murray. Rewriting: Stanley G. Crawford. ERDA San Francisco Operations Office liaison: Sandra Chalton. Illustration: Michael A. Garcia. Editing and Layout: Merily H. Keller. Cover Design: Violet McFall.

An edition of the handbook was published for limited distribution as LA-6242-MS in March, 1976, with editing and layout by Derrick P. Grimmer and Kenneth C. Herr. Benjamin T. Rogers contributed to the section on passive systems.

Abstract

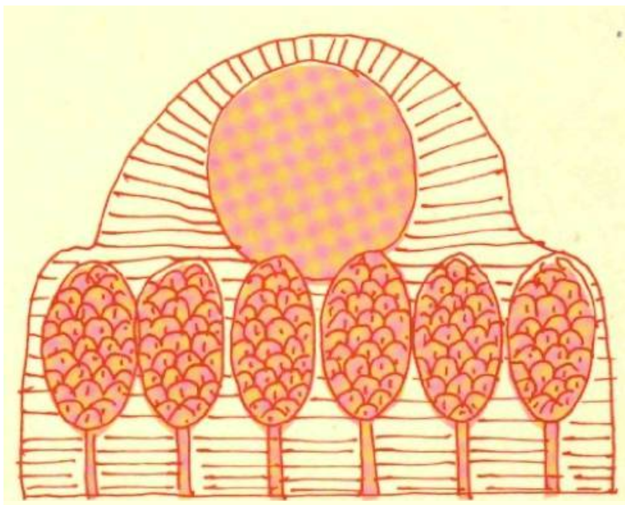
This handbook is intended as a guide for engineers, architects, and individuals familiar with heating and ventilating applications who wish to design a solar heating system for a residential or small commercial building in the Pacific Coast Region.

The climate of the region is discussed by selected cities in terms of the effect of climate on solar heating requirements. The four states covered are California, Oregon, Washington, and Arizona.

Presented in detail are the design parameters and performance characteristics of (1) liquid space heating systems of the active type using flat plate collectors, water heat storage, and forced air distribution; (2) active air space heating systems using flat plate collectors, rock-bed storage, and forced air distribution; and (3) domestic hot water heaters of the active type using liquid-cooled flat plate collectors. The analyses are based on hour-by-hour computer simulations using actual weather data from six selected cities. The collector area required for the three types of active systems has been determined as well as the effect of changes in all the design parameters. A simple, empirical, monthly solar-load ratio method for determining collector area is described and results are presented for 13 additional locations within the region.

Swimming pool heaters are described briefly, and passive space heating systems are discussed qualitatively in terms of a number of successful designs.

Contents



Introduction

I. Introduction

The aim of this handbook is to aid in the proper choice of collector size, storage size, and other important design parameters for a solar space heating or domestic hot water system of the active type in the Pacific Region.

Much of the information presented here is relatively technical and will be of use primarily to architects, contractors, and consulting engineers in heating, ventilating, and air-conditioning. It may be of use to the "do-it-yourself" owner-builder with a sufficient technical background.

A great deal of design information on solar heating systems is currently available. New "how-to" books are appearing all the time. Solar hardware is being manufactured by a number of companies in the United States and abroad. We estimate that there will be 1,000 to 1,500 solar heated buildings in this country alone by the end of 1976.

Although design principles of solar heating are now widely understood, current literature does not provide quantitative descriptions of the thermal interaction of the solar heating system components. This type of analysis is relatively complicated and is highly dependent on the weather patterns in the locality where the system is to be used and involves simulation of solar performance from hourly solar radiation records on a digital computer. This has hardly been a practical approach for the individual builder.

The handbook supplies this type of quantitative analysis for the Pacific Region. With it, the designer can determine collector and storage size and other important design parameters proper to the building's locality without an individualized computer simulation.

The emphasis of the handbook is on what have come to be called "active" systems. No quantitative analyses are offered here for "passive" systems. This is in part because quantitative analysis of active systems is reasonably well understood, while with passive systems much work remains to be done. It is also a matter of what can be acceptably defined as a "standard" system of either kind during this early period of the development of solar technology. An active system is generally considered to consist of a liquid-or-air-cooled metal flat plate collector with a storage tank or rock bed and with forced air or liquid distribution systems connecting the solar collector to the heat storage area and living space. Certain elements of hardware, notably collectors, are commercially available for active systems and speak for a degree of standardization at the present time. It is upon these notions that the term "standard" used in the handbook is based.

By contrast, the passive approach to solar heating is almost as varied as the individual buildings themselves, and it is not yet possible—if even desirable—to call any one passive approach "standard." However, a section on the passive use of solar energy has been included in the handbook in the interests of comprehensiveness. It is hoped thereby that liquid and air systems of the active type will not be taken as the standard for all solar energy systems, simply because they are more easily subject to quantitative analysis, or more closely resemble conventional heating systems in certain respects.

For similar reasons, brief discussions of swimming pool heaters, heat pumps, reflectors, etc., have been included in the handbook. These are economically feasible uses of solar energy at the present time, although detailed quantitative analyses of performance data are not yet available.

The greater bulk of information presented here will therefore concern the designer faced with a specific problem in an active space heating or domestic hot water system, and some indication at this point needs to be given concerning the use of the handbook for this purpose.

In the section of Space Heating-Active Systems, the effects of varying any one of the fifteen parameters of the nominal standard liquid system are presented in detail for six locations within the region. The standard air space heating system is presented in the same manner for those parameters it does not share with the liquid system, and the domestic hot water system also for those parameters it does not share with either.

Thus the designer interested in the effect of varying any one of the design parameters on the performance of the whole system should first consult the list of nominal design parameters on page 17, and then turn to the detailed discussions of individual parameters that follow.

It should be noted here that the design parameters were varied singly in the simulations, holding all others constant. The designer should, therefore, use care in varying two or more design parameters simultaneously, since some of these may be coupled non-linearly. Any individual system will, of course, deviate somewhat from the standard system as well. Furthermore, the effect of parameter changes has been studied in the handbook for only six locations where hourly weather and solar radiation have been recorded. The magnitude of the effect will be different for other localities within the region. However, the results for the six cities studied should encompass the range of climate within the Pacific Region.

One of the main purposes of the handbook is to enable the designer to determine the size of the collector array for an active space heating system anywhere in the region. The subject is covered for space heating systems for the six cities beginning on page 52, where it is possible for the designer to size a solar heating system simply by using the tables on pages 53 and 61. For other areas within the region, it will be necessary to use the Monthly Solar-Load Ratio Method, an explanation of which begins on page 59, with case studies exemplifying the use of the method in the Appendix.

Similar tables and calculations for determining the size of domestic hot water heating systems are presented at the end of the hot water heating section.

Some last words of caution. The apparent simplicity and grass-roots appeal of solar heating can trap the unwary into a poor installation. Anyone who would not consider designing a home heating system should not blithely launch into a solar heating installation, particularly of a complex type. Adequate preparation is advised and should include a basic working knowledge of the conventional heating system to be employed in conjunction with the solar unit. Many books are available to the individual wishing to design his own system. This handbook is not intended to overlap the practical details and hardware considerations in the now numerous "how-to" books and catalogs, some of which are listed in the reference section.

It should be noted that because solar heating is a relatively new commercial field, certification procedures are still in the process of being developed for solar equipment. Thus, the builder must be especially attentive in his choice of equipment to determine that it will last the required time and be easily maintained and repaired. Solar heating systems of the active type are relatively expensive, and their cost effectiveness depends on these factors.

The Pacific Region

II. The Pacific Region

Solar energy can be used to provide a major fraction of the domestic hot water and space heating requirements in buildings throughout most of the continental United States. Although solar radiation is a relatively diffuse and weak source of energy at the earth's surface, it is precisely this type of low-temperature energy which is best suited for space heating and hot water heating on a small residential scale. Given the fact that present-day collectors can retain only one-third to one-half of the total solar radiation that falls on them, they are still able to raise the temperature of considerable quantities of water to useful levels (e.g. 130-150°F) in the course of a sunny day. Such temperatures will not power turbines or smelt iron ore but will serve admirably for heating a house.

Climate is a critical factor in the performance of any solar heating system. The Pacific Region includes some areas ideally suited for solar energy systems as well as others that present a variety of problems. For the purposes of this handbook, the region is defined as the three Pacific Coast states, California, Oregon, and Washington. Arizona, however, has been included, owing to similarities between its climate and that of Southern California. It also contains large population centers for which hour-by-hour weather data are available. A city outside the region, Bismarck, North Dakota, has been included in order to provide an example of a cold, northern climate similar to the eastern parts of Washington and Oregon for which detailed weather data were not yet available.

The Pacific Region encompasses such a tremendous variety of climates that it cannot be considered a homogeneous zone. In terms of solar heating, for example, the collector area required to reduce the consumption of conventional fuels by 50% will vary by a factor of nearly seven throughout the region. Moreover, boundaries of climatic zones within the region are not clearcut; the climate near the coastline can change markedly within a few miles or a few hundred feet in altitude. These and some inland areas are subject to foggy conditions at various times throughout the year. Highly variable conditions dependent on altitudes, slope, etc., also affect mountainous areas. Such factors should be taken into account in estimating the performance of a solar heating system. They imply a detailed knowledge of not only the characteristics of the climate of the general area where the solar heating system is to be constructed, but of the peculiarities of the microclimate as well.

A useful way, however, to discuss the climate of the region is in terms of solar and weather data from representative cities. LASL has studied the data of 18 cities which can serve as guides to the varieties of climate within the region. Of these cities, six were studied in detail at LASL; they are indicated by circled dots on the map on page 4. The hourly records of solar radiation and weather data available for these six cities were used in the computer simulations of parametric variations for solar space heating and domestic hot water systems, the results of which are discussed beginning on page 19. Their general climatic characteristics can be summarized as follows:

Phoenix, Arizona. The climate is arid southwestern, although significant irrigation has increased the humidity within the Phoenix basin. The winter heating season is short with ample sunshine and large diurnal fluctuations in temperature.

Santa Maria, California. The climate of this coastal city is cool but relatively sunny. Although the heating season is mild, it extends over most of the year due to the influence of sea breezes from the cold Pacific stream.

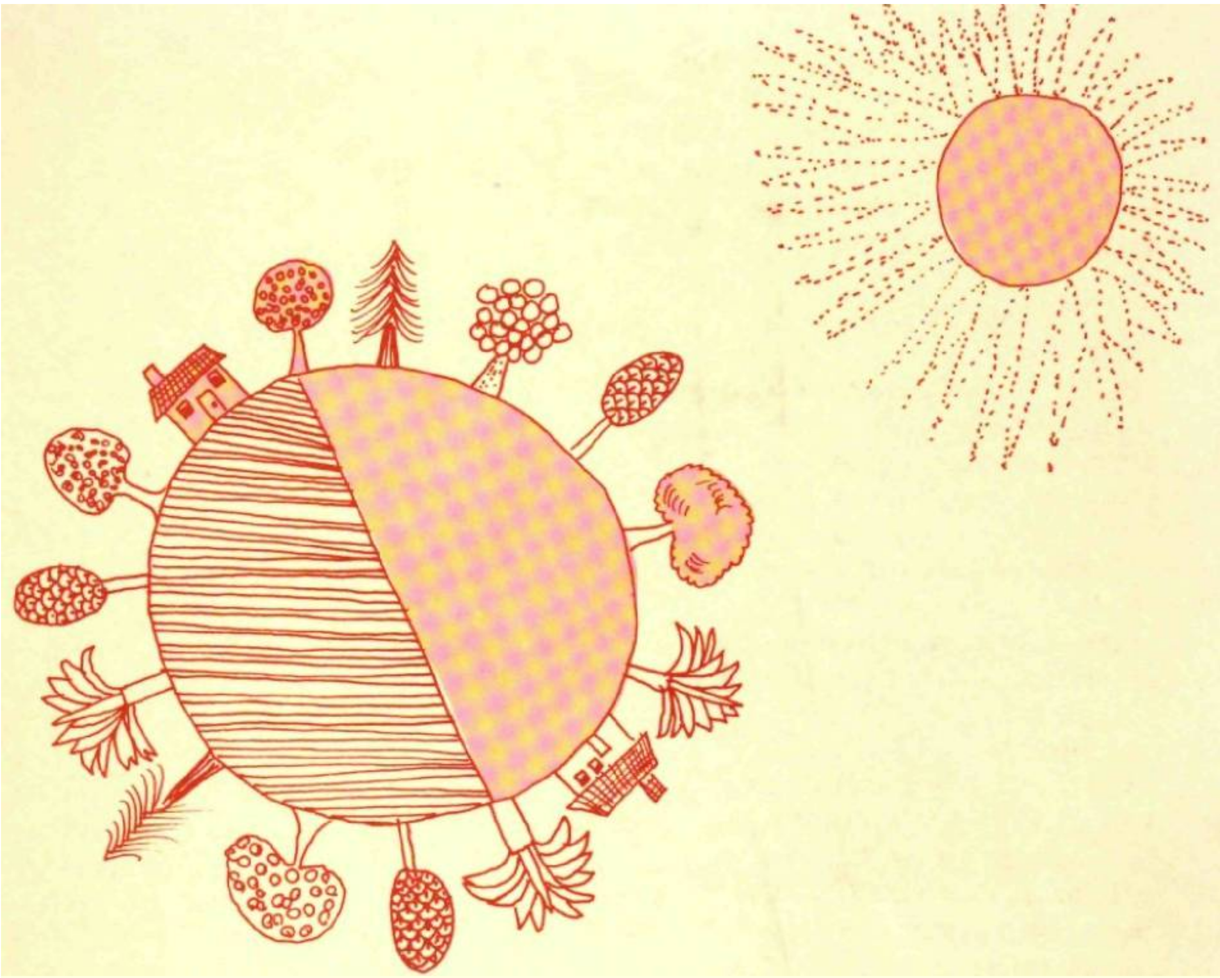
Fresno, California. The climate is typical of the inland basin, with the heating season concentrated in the two-month period of December-January when the weather is generally foggy but not severely cold.

Medford, Oregon. The climate is moderate, perhaps the most average of those studied in the region. There is considerable rainfall in the winter months.

Seattle, Washington. The foggy coastal climate of Seattle is dominated by the cold Pacific stream, which results in a heating season that is long and cool without being severely cold. Little direct-beam solar energy penetrates the humid, foggy atmosphere.

Bismarck, North Dakota. In this central continental climate, the heating season extends from September into June and is severe with average daily temperatures of 10°F in January. Although there is appreciable snowfall and blizzards are frequent, the potential for solar heating is good on account of the high percentage of possible sunshine in the winter.

The remaining 13 cities were studied in terms of monthly weather and solar data. They are: Page and Tucson, Arizona; El Centro, Inyokern, Los Angeles, and Riverside, California; Astoria and Corvallis, Oregon; and Prosser, Pullman, and Spokane, Washington. They are indicated by dots on the map on this page , and the tables giving data for these cities are to be found on page 61 in the section on estimating the size of solar collectors for active systems.



General Design Considerations

III. Design Considerations

Some climatic zones within the Pacific Region are clearly more suited for solar heating than others, either in terms of overall solar radiation or in terms of seasonal or even daily patterns. In some areas the goal can be, at best, to reduce the costs of conventional heating by enough to justify the additional cost of a supplementary system. In the other sunny locations a solar heating system can be expected to supply the major source of heat with gas, propane, electricity, or wood providing the rest.

Site will also have a bearing on which heating system is to be the primary one. In extreme cases it will determine whether a solar installation should be considered at all. Obscuration by buildings, trees, hills, or mountains should be studied to ascertain when and how much solar radiation is lost. Until legal sun rights are established by legislative or judicial processes, the builder should anticipate whether the site might fall at some future time in the shadow of a nearby high-rise construction or a growth of trees on adjoining land. Such considerations, as well as those of cost and technological limitation, suggest that solar space heating installations are best suited for residential or small commercial buildings in rural or suburban areas, and not yet for densely populated high-rise inner city areas. Domestic hot water systems, by contrast, are both more compact and cost effective, allowing a greater flexibility of installation and application.

While climate and site will have an effect on the percentage of solar space heating designed for, the type of building construction will have a bearing on the most suitable solar design. Unfortunately for the builder the sun cannot be turned on or off at will in any climate, thus thermal storage of solar radiation will be a feature of all heating systems intended to carry through night-time hours and periods of cloudiness. Thermal storage depends on mass. In stone, cement block, concrete or adobe structures, some useful thermal storage is already in the structure. This mass is particularly effective in smoothing out variations in inside temperature at locations where the ambient day-night temperature changes are large. Building mass is also advantageous for storing solar energy which enters the structure through windows.

In the Pacific Region the majority of residential building construction is of the frame type—and mobile homes can be included here—in which the structure contains relatively little mass. The thermal storage in such a building will be almost entirely that associated with the active solar heating system. As a water storage tank or rock bed, it will usually be insulated from the living space so that heat stored in it can be distributed according to demand. To some extent, the design of active solar heating systems and of conventional systems is conditioned by the economics of lightweight building practices and the need to maintain even temperatures within a structural envelope that provides little thermal flywheel effect of its own.

The first consideration in the thermal design of a building is to minimize the heating load. For a small, single-story, well-insulated building, the thermal load ought to be in the range of 8-10 BTU per degree-day, per square foot of floor area.* The major contributors to building thermal load are thermal conduction through the materials comprising the outer fabric, and the energy required to heat the outside air which infiltrates the building through cracks,

* Detailed methods of calculating heat losses (the building thermal load) are to be found in the ASHRAE Handbooks, listed in the references. A summary of the methods is to be found on page (52) below, for both planned buildings and existing buildings.

porous walls, open windows, and doors. The infiltration rate can be effectively reduced by the use of weatherstripping around doors and windows and with vapor barriers built into walls, ceilings, and floors. A permissible infiltration rate is usually in the range of one-half to one air change per hour. Less than this will lead to stuffiness in normal living spaces. In areas of greater activity such as meeting rooms, the infiltration rate will have to be increased. Adequate insulation of doors, walls, floors, and ceilings will reduce thermal conduction; and the use of double-glazed windows, storm doors, draperies, and insulated window covers at night will help conserve the positive solar gains through south-facing windows during daytime hours.

The importance of these load-reducing measures must be stressed. First, they are usually more cost-effective than active solar heating. Second, the area of solar collector needed to obtain a given solar heating fraction is directly proportional to the building load. Collectors, storage tanks, and rock beds occupy more space within a building than conventional heating systems, and the design should seek to hold this down to a minimum. And, in some cases, the size of collector array might bear on aesthetic considerations. For these reasons, only after the heating load has been reduced by more cost effective measures should an active heating system be considered for handling a portion of what remains.

Any design optimization usually involves a tradeoff between cost and performance. At some point the extra performance that can be achieved by adding more insulation to a wall or ceiling or increasing a solar collector area will exceed the savings incurred. In a few cases, a true performance optimum does exist—with collector tilt and orientation, for example.

For most locations it is uneconomical to design a solar heating system to provide for 100% of the heating requirement, since collector areas and storage volumes needed to collect and store enough heat for prolonged cloudy weather would be large and unwieldy. Thus in general, a solar heating system should be designed with a full capacity auxiliary heating unit for periods of extended cloudiness. This applies, in particular, to northern areas where interior temperatures must be maintained in order to protect water lines. Where the solar heating system is the primary one, the use of an efficient wood-burning space heater has appealed to many as wood is a relatively inexpensive energy source in many locations and is easily stored. It is a renewable resource as well—being, in fact, a form of stored solar energy.

Once the type of solar heating system has been decided on, questions of maintenance and lifetime should have a direct bearing on the selection and construction of the system components. Solar equipment can represent from 5-15% of the building cost and, therefore, must have a lifetime of 15-30 years to warrant the investment. The designer should pay particular attention to potential problems in the following areas:

- Accessibility to glazed areas, fluid lines, water storage tanks.
- Ultraviolet degradation and weathering, fouling of exterior surfaces, and painted surfaces.
- Corrosion in liquid systems.

In the case of a complex solar heating system, the amateur is advised to be very well-informed or to seek the service of a consultant with experience in the field before committing himself to a major investment. In addition, he may do well to have a frank talk with the owner or resident of a solar-heated building. Most people involved in using solar energy are very willing to talk over problems and share their knowledge.

Cost Effectiveness

IV. Cost Effectiveness

The do-it-yourself homeowner can build a greenhouse for as little as $2.50 per square foot. A lean-to greenhouse on a south-facing wall will provide a significant quantity of heat for the adjacent house. This sort of system may not afford a very high degree of temperature control or storage but is, at the present time, a very economical way to trap fairly large quantities of solar radiation for space heating purposes, and the construction techniques are within the range of anyone moderately skilled in carpentry.

The installation costs of active solar heating systems are relatively high but allow greater flexibility of temperature control. Values of $20-$30 per square foot of collector area are common for professional installations; the entire active heating system for a house may cost from $2,000 to $10,000 at the present time. Mass production and increased competition should bring the costs down somewhat in the future and do-it-yourself installations will reduce labor costs.

Passive systems with good temperature control capability will probably fall somewhere in between the costs cited above. However, neither their cost nor performance characteristics are well delineated at the present time.

The major consideration is the tradeoff between the system's installation cost and the future cost of the fuel saved by the solar heating system during its anticipated lifetime. One must also account for operating, maintenance, and repair costs. A fairly useful way of estimating the cost of solar heating is by the following formula:

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

The annual capital fixed charge rate is simply the value of the capital to the individual. It may be the interest rate that the capital would 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 eight percent twenty-year loan is 0.1004, as determined from banker's tables or formulas.

The annual energy delivered by a "standard" active liquid or air space heating system can be estimated from the data presented in the table on page 10 for 18 cities in the region. It will be noticed that the smaller the solar heating system, the more heat it will deliver per square foot of collector, largely because the smaller system will be running at full capacity more of the time.

A sample calculation will help il[1] lustrate the use of the formula. In the sample, let the following assumptions be made:

- System installed costs at $15 per square foot of collector. This is a fairly low figure for an active system.
- Annual capital fixed charge rate at 0.10, representing a 20-year loan at 8% interest.
- Operating, maintenance and repair costs at 2% per year of installed cost per year.
- Annual energy delivered by the solar heating system equal to 162,000 BTU per year per square foot of collector area, a figure that represents the anticipated performance of a 50% space heating system in Los Angeles, California.

Then, from the formula:

$$\text{Cost of solar heat} = \frac{\$15 \times 0.10 + (\$15 \times .02)}{0.162}$$

we obtain

= \$11.11 per million BTU for the cost of solar heat.

How does this hypothetical figure compare with competing energy sources? Natural gas and electricity costs vary widely throughout the country but, at present, they are generally lower. For example, if natural gas is \$1.50 per 1,000 standard cubic feet and is burned in a furnace at 60% efficiency, then the delivered cost of heat is roughly \$2.50 per million BTU. If electricity costs 3.54 per kilowatt hour, then the cost of electrical resistance heating is \$10.25 per million BTU.

In many instances, solar heating systems of the active type are not as competitive with natural gas as they are with propane or electric resistance heating systems. However, natural gas is not available in many areas and is becoming scarce in others reducing its use as a competing alternative fuel source. It is reasonable to assume that the cost of all sources of fossil energy will rise. The uncertainty is by how much. This, perhaps, is the most compelling argument in favor of solar heating.

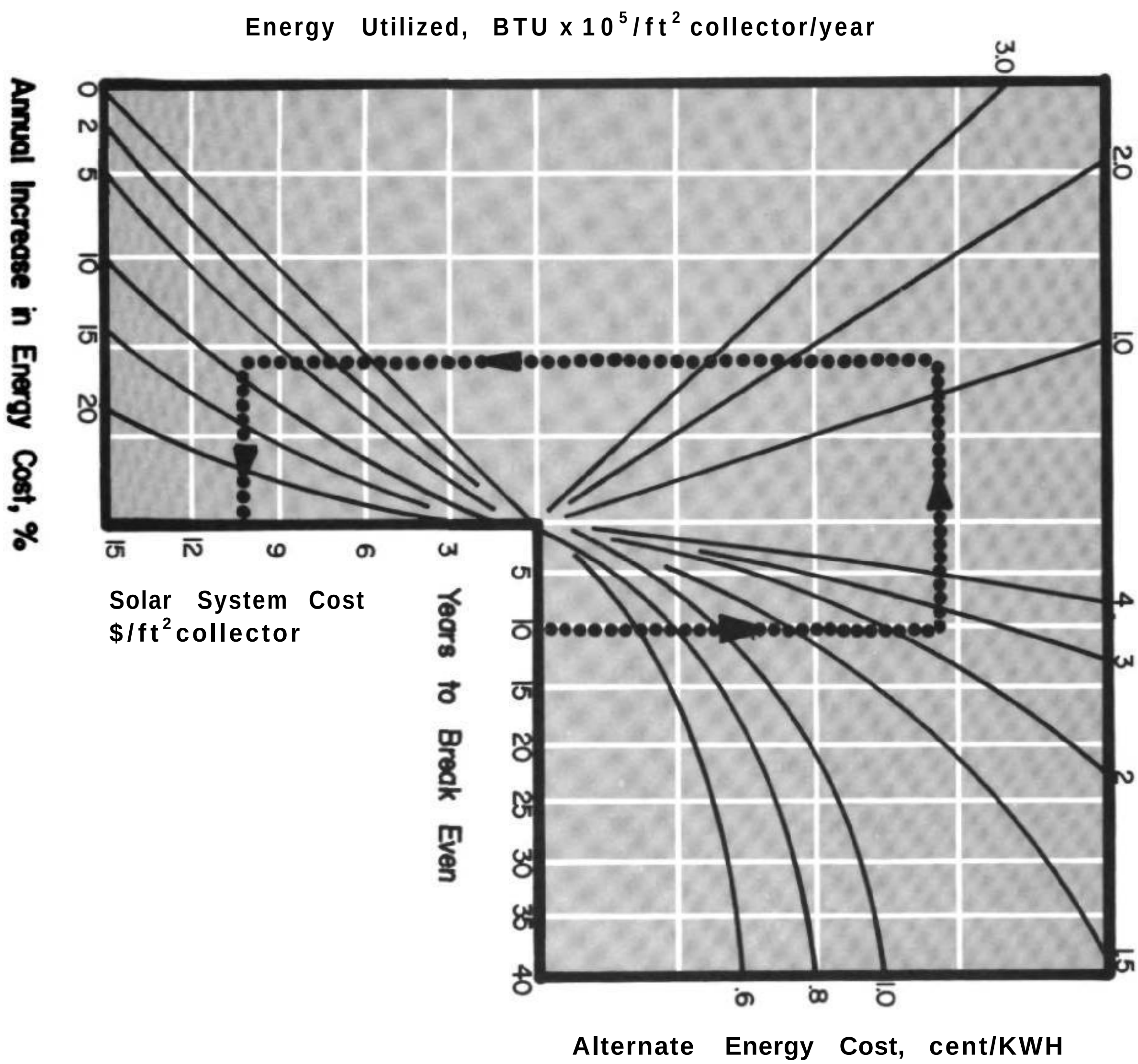
The formula given above does not take into account a number of other considerations which can affect life-cycle costs. The net result of these considerations can be either positive or negative and will vary widely from state to state. They will usually be different for a business than for an individual. Some positive considerations are direct tax incentives or grants which may be enacted by state or federal law, deduction of interest payments in calculating state and federal income taxes, and business depreciation of equipment. Some negative considerations are property tax evaluations of the solar installation, and the ability to deduct fuel costs as a business expense.

A method similar to calculation of the cost of solar heat but which allows one to include a forecast of the annual increase in conventional fuel costs is the nomograph on page (9).* With it, one can select a break-even time for the solar heating system, the type and cost of conventional fuel with the forecasted annual increase, and the energy output of the solar heating system. From these figures, one can derive what the system must cost in order to provide the desired breakeven time. An example is plotted on the nomograph by the dotted line.

- The solar heating system is to break even at ten years.
- Electricity is assumed to be priced at 2.4¢ per kilowatt hour, with an 8% annual increase.
- Useful solar energy is assumed to be 120,000 BTU/yr per square foot of collector area.
- The cost of money is assumed to be 8%.
- The solar heating system maintenance cost is assumed to be 1%.

From these figures one derives a \$10 per square foot system cost goal on the nomograph. The nomograph is intended to be only illustrative of the role of the various costs, prices, and performance factors that determine the economics of solar heating and should be used accordingly.

* From "Energy Conservation Design Guidelines for New Office Buildings," GSA, as quoted by P. D. Maycock in "Solar Energy: The Outlook for Widespread Commercialization of Solar Heating and Cooling," ERDA.


Energy Utilized, BTU x 10^5/ft^2 collector/year
3.0
2.0
1.0
Annual Increase in Energy Cost, %
0 2 5 10 15 20
15 12 9 6 3
Solar System Cost
$/ft^2 collector
Years to Break Even
5 10 15 20 25 30 35 40
4
3
2
1.5
.6 .8 1.0
Alternate Energy Cost, cent/KWH

Annual Energy Delivered by a Space Heating Solar Energy System*

Site	Annual net energy delivered per square foot of collector, in millions of BTU/year ft_c^2		
	Solar Heating Fraction		
	25%	50%	75%
Arizona			
Page	0.212	0.160	0.115
Phoenix	0.132	0.104	0.078
Tucson	0.136	0.107	0.080
California			
Davis	0.124	0.090	0.062
El Centro	0.199	0.150	0.106
Fresno	0.122	0.087	0.060
Inyokern	0.205	0.154	0.110
Los Angeles	0.214	0.162	0.116
Riverside	0.176	0.137	0.100
Santa Maria	0.262	0.211	0.149
Oregon			
Astoria	0.164	0.117	0.073
Corvallis	0.142	0.098	0.064
Medford	0.134	0.095	0.061
Washington			
Prasser	0.140	0.098	0.064
Pullman	0.139	0.098	0.066
Richland	0.148	0.104	0.068
Seattle	0.121	0.081	0.051
Spokane	0.150	0.104	,0.069

*It should be noted that these values apply to a space-heating-only installation. They have been derived based on the Monthly Solar-Load Ratio method—an explanation of which begins on page 59. If a domestic hot water system is added to a space heating system, as described on page 66, then the annual energy delivered may be substantially increased since the solar collectors are in use year around. In one example (for Albuquerque, NM) the yield was increased from 0.225 to 0.281 million BTU/yr-ft_c^2. In another example (for Madison, WS) the yield was increased from 0.134 to 0.180 million BTU/yr-ft_c^2. These results depend, of course, on the relative size of the two loads.

Solar Space Heating, Active Systems

V. Solar Space Heating, Active Systems

Flat Plate Collectors. A flat surface painted black, enclosed in a glazed box, and oriented and tilted properly in relation to the sun will trap a considerable amount of solar radiation. But solar collectors must do more than collect. They must also move the heat out of the collector; and it is the nature of this heat transfer function, and specifically whether the heat transfer medium is air or a liquid, that will dictate the form of much of the remainder of the heating system.

All of the collector types shown on this page, with one exception, use black-painted metal as the collector surface to absorb the solar radiation. In liquid collectors, the heat is removed from the surface by a liquid pumped through tubes or ducts attached to the surface or dribbled down corrugated metal troughs. Similarly in air collectors, the air is blown across the collector surface or through laminations of metal gauze.

At the low temperatures used for space heating or domestic water heating, flat plate collectors of either type, liquid or air, work with reasonable efficiency and are relatively easy to construct. Convective losses are easily reduced by a transparent cover, usually one or two layers of glass, and conduction losses through the collector back and sides can be cut down by insulating these surfaces. It may be of use to reduce radiation losses by means of a selective coating on the absorber surface, that is, a coating with a high absorptance for the solar spectrum and a low emittance for the infrared reradiation spectrum.

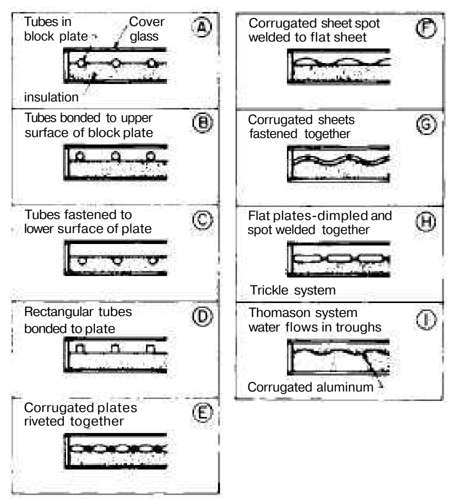
Collector efficiency can be defined as the ratio of heat removed from the collector by the heat transfer medium, the collector coolant, divided by the incident solar radiation. No single number results, however, as the ratio is dependent on a variety of conditions, such as coolant temperature and flow rate, the magnitude and angle of incident solar radiation, and the wind velocity. It is also dependent on the design parameters of the collector itself.

But collector efficiency can be calculated by performing a detailed heat balance on the absorber surface and on each sheet of glazing, which will account for solar and infrared absorption and radiation, convection, and conduction heat flows. Collector efficiency for one set of conditions is shown on page 12. It is convenient to plot efficiency as a function of the parameter $\Delta T/I$:

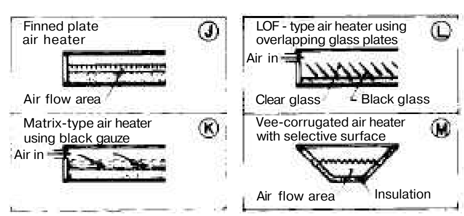
$$\frac{\Delta T}{I} = \frac{\text{Average Collector Surface Temperature} - \text{Ambient Temperature}}{\text{Incident Solar Radiation}}$$

COLLECTOR TYPES

WATER HEATERS



AIR HEATERS



* This potpourri of collector types is taken from Chapter 59 of the ASHRAE Handbook, 1974, "Solar Energy Utilization for Heating and Cooling," by John I. Yellott.

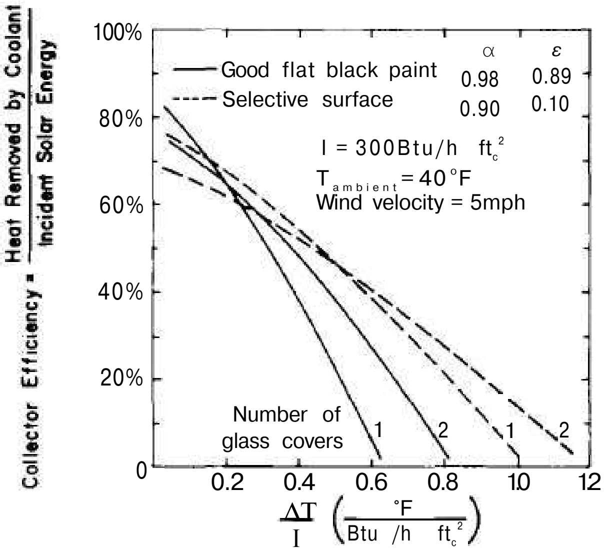
Reflectors. The amount of solar radiation striking a collector surface can be considerably increased by reflectors, thereby allowing one to reduce the size of the collector array. An area of white gravel in front of a collector, for example, will reflect a certain amount of radiation into the collector, as will snow, though more diffusely than sheet aluminum, or mirrored surfaces that can be mounted on movable panels.

Recent analysis indicates that the potential advantage of reflectors is substantial. A vertical solar collector with a horizontal reflector in front of it should perform substantially better than the same collector tilted at the optimum angle without a reflector. In addition, the seasonal variation in solar radiation incident on the collector is completely changed with a reflector.

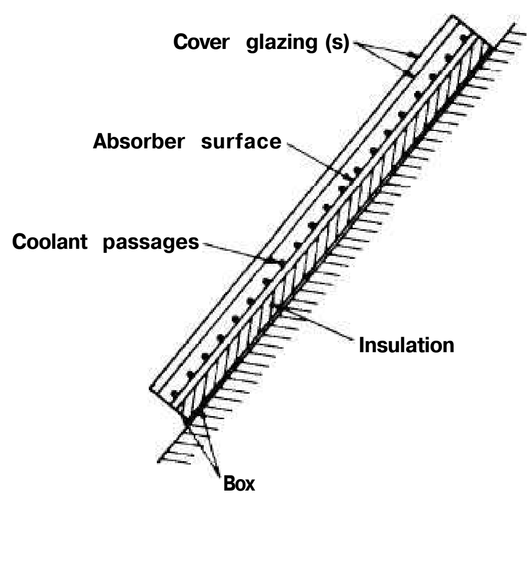
Heat Transfer, Collector to Storage. The most desirable medium to transfer heat from the collector surface to the heat storage area would be one that is stable, non-freezing, non-boiling, non-corrosive, non-flammable, inexpensive, non-viscous, non-toxic, non-energy intensive, and with a high specific heat and thermal conductivity. Air satisfies all these requirements except the last two, and minor leaks cause no inconvenience.

Water has many desirable characteristics as well. However, it freezes at relatively high temperature, boils at a relatively low one, and can cause problems with corrosion. Freezing can occur in almost all areas of the region and provision should be made for this, even in areas where freezing is infrequent or exceptional. Collectors can be designed to drain at night or whenever the temperature drops below a certain point, or they can be designed with coolant passages that can tolerate the expansion caused by freezing. Similarly, protection against boiling can be provided by draining the collectors, or by pressurizing the plumbing of the system. And corrosion, a serious problem, can be dealt with in a variety of ways using special materials and techniques.

FLAT-PLATE SOLAR COLLECTOR PERFORMANCE MAP



COLLECTOR FUNCTIONAL PARTS



Many different liquids have been tried, but all have been found deficient in some category. Ethylene glycol/water mixtures are now in common use, but like other fluids besides water, require a heat exchanger between the collector coolant circuit and the heat storage water tank. They also require corrosion protection. Expense is an additional factor to be considered as the volume of a typical liquid collector of 500 square feet plus the piping to and from storage can amount to as much as 30 gallons. Toxicity is of particular importance when a system includes domestic hot water heating. LASL is currently evaluating a number of fluids, among which a class of light paraffinic oils show promise. A summary of the characteristics of collector coolants is presented below.

COLLECTOR COOLANT CHARACTERISTICS

	Air	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	Non-Corrosive			Non-Corrosive	Non-Corrosive
Fluid Stability	-			Good*	Good**
Flash Point	-	None	None	455°F	500° F
Bulk Cost ($/gal)	-	-	2.35	1.00	4.40
Thermal Conductivity (BTU/HR°F@ 100°F)	.0154	0.359	0.18	0.76	0.119
Heat Capacity (BTU/LB°F@ 100°F)	.24	1.0	0.66	0.46	0.45
Viscosity (LB/FT HR@ 100°F)	.046	1.66	15.7	28.5	143.1

*Requires an isolated cold expansion tank or nitrogen containing hot expansion tank to prevent sludge formation.

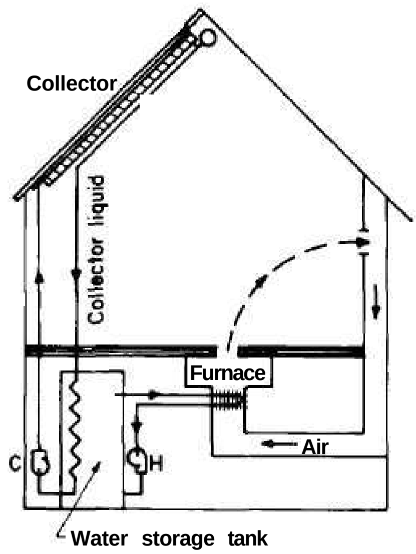
**Contains a sludge formation inhibitor.

Heat Storage and Distribution for Liquid Systems. The common thermal storage medium for liquid systems is water, usually held in a large basement tank and insulated from the living space of the building. The size of the tank is directly proportional to the collector area for the percentage of solar heating desired. For example, a system designed to supply 75% of the heating load of a 15,000 BTU/degree-day house in Medford, Oregon, would have a tank capacity of 1,680 gallons which would hold enough heat to carry over approximately 22 hours. If water is the heat transfer medium, the storage tank water will be heated directly, and indirectly through a heat exchanger if other fluids are used. The heat exchanger can be as simple as a coiled section of tubing, usually copper, immersed in the storage tank water or wrapped around the external surface of the tank; or it may be an external heat exchanger requiring an extra pump to move the tank water through the exchanger.

In order to transfer heat from storage to the building, heated water is pumped from the top of the storage tank through a second (liquid-to-air) heat exchanger, a finned tube coil, over which is blown cool return air drawn from the living space. The air re-enters the living space through registers shown in the diagram on page 15. Alternately, the heat can be distributed to the living space using conventional perimeter convectors, although usually at some loss in performance. A radiant heat distribution system could also be used.

Heat Storage and Distribution for Air Systems. For air systems, the common thermal storage medium is a bin full of fist-sized rocks, usually located (as with a water tank) in a basement area and insulated from the living space. A 75% solar system for the same 15,000 BTU/degree-day house in Medford, Oregon, would require a rock bin of 700 cubic feet, containing 32 tons of rocks. Problems of leakage and accessibility are not nearly as critical with rock beds as with water storage tanks, and they do not require maintenance. Rock beds are efficient heat transfer devices. The air gives up its heat quickly by flowing through the labyrinthine paths; as a result, the rocks near the air entry end of the rock bed can be at quite different temperatures than those near the exit end. This makes for a certain responsiveness in retrieving stored heat and compensates to a great degree for the one disadvantage air has as a heat transfer medium—its low heat-transfer coefficient.

An air system, of course, does not require heat exchangers of the sort used in liquid systems except in cases where domestic hot water is to be heated or preheated. For space heating, the hot air from the collector is blown directly into the living space; or for storage, into the rock bin as in the diagram on page 15. In terms of mechanical assistance, an active air system requires only one fan and two double dampers. As soon as the collector temperature exceeds the rock bed exit temperature (left side), the collector turns on (blower on) with damper C in the position shown. If the living space requires heat, damper H is in the position shown. When there is surplus heat from the collector, damper H moves to its upper position, directing the heat into the rock bed. When heat is needed from storage at night or during cloudy periods, damper H remains in the position shown with damper C in its upper position. This blows air through the rock bed in the reverse direction from storage, thus removing heat from the warm (right) side of the rock bed first. An auxiliary furnace, as with the liquid system above, is operated as necessary to satisfy the remainder of the heating load.



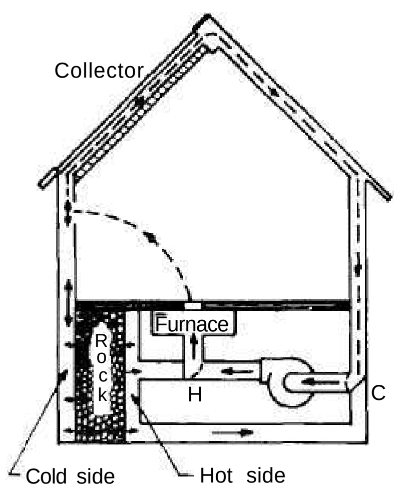
Liquid Heating Collectors / Water Storage / Forced Air Distribution

SPACE HEATING SYSTEM USING LIQUID HEATING COLLECTORS

Liquid or Air? In terms of performance there is little to choose from between liquid and air systems (Graph, below), based on the nominal parameters selected for the two "standard" systems. One could change the parameters slightly to give a small performance edge to either system, but the actual choice between competing design concepts will more likely rest on considerations other than those of performance alone.

Air systems may offer cost advantages in new installations by virtue of their greater simplicity and will probably be cheaper to service and repair. Unlike liquid systems, they need not be entirely leak-proof and there are no problems of liquid degradation or corrosion.

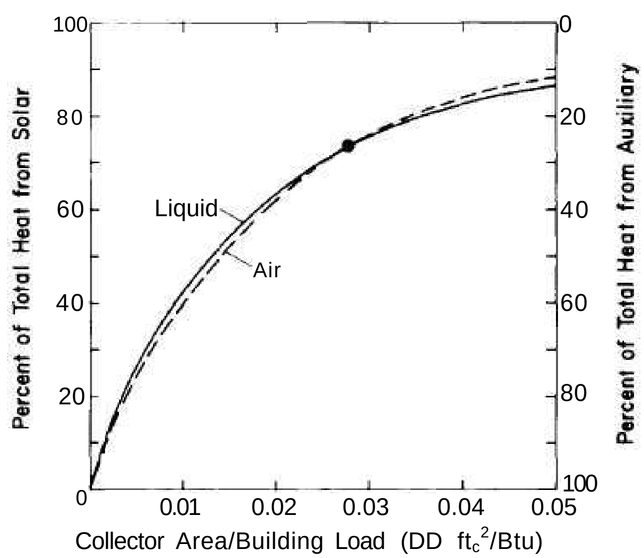
On the other hand, air ducts and rock beds are bulkier than pipes and water tanks and so may not be as adaptable to retrofit situations as the components of a liquid system. A 1,680 gallon water tank of 224 cubic feet, for example, will store as much heat as a 7 foot by 10 foot by 10 foot rock bed of 700 cubic feet.



Air Heating Collectors / Rock Bed Storage / Forced Air Distribtion

SPACE HEATING SYSTEM USING AIR HEATING COLLECTORS

COMPARISON OF AIR AND LIQUID SPACE HEATING SYSTEMS IN FRESNO



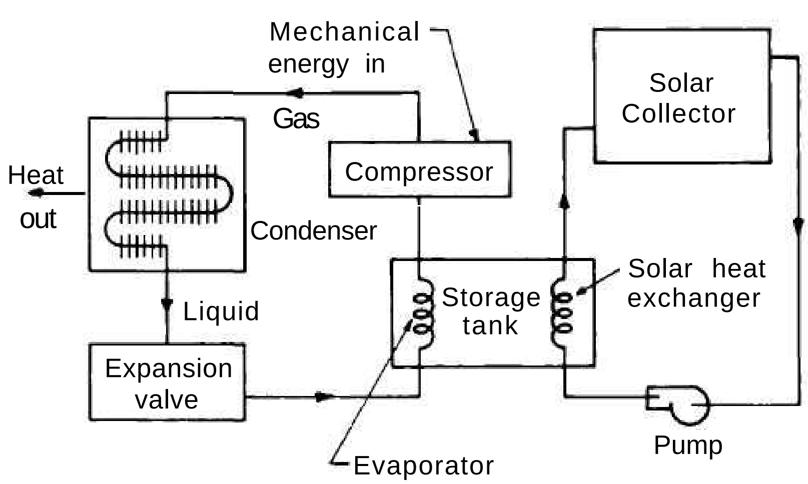
Liquid systems offer advantages in terms of integrating domestic hot water heaters and are, at the present time, more adaptable for solar cooling. In addition, because they have received greater attention by designers and by industry during the recent rebirth of interest in solar energy, liquid system components are currently more readily available. However, air-heating collectors and rock beds were among the earliest systems built, and they can be expected to proliferate as their good performance characteristics become more widely known.

Heat Pump Auxiliary. A heat pump auxiliary can be used to improve the performance of either a liquid or air system but adds complexity and expense. A heat pump works on the principle of the household refrigerator or air conditioner and consists of the same components—an electric motor, compressor, and a closed loop of a vapor that is pumped through cycles of condensation and evaporation. Essentially, a heat pump transfers thermal energy from a low temperature region to a higher temperature region using an electric motor.

In its solar heating application (see Diagram, below), low-temperature solar heat is applied to one side of the heat pump system and causes the condensed vapor to evaporate. The compressor then raises the pressure and temperature of the vapor which, when next 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 BTU of heat at a high temperature (160-180°F) for every BTU of energy the compressor uses—and thus has a coefficient of performance (COP) of 3.

The use of a heat pump is particularly attractive in a situation where summer cooling of a building would normally be handled by a conventional air-conditioner. In its place, a reversible air-conditioner can be installed at a small additional cost to function as a heat pump during the winter heating months.

HEAT PUMP SCHEMATIC



Design Parameters for Liquid Systems

Standard Liquid System. In order to determine the quantitative effect of any one component or feature of a solar heating system on the performance of the whole, it is first necessary to establish the design parameters for a reasonably good system consistent with cost and the present state of the art. Current design practices and manufacturing processes indicate what is possible, and within these conditions one can determine quantitatively the range of performance and the cost effective optimum within that range.

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

PARAMETER:	NOMINAL VALUE:
Solar Collector	
1. Orientation	Due south
2. Tilt (from horizontal)	Latitude + 10°
3. Number of glazings	1
4. Glass transmissivity (at normal incidence)	0.86 (6% absorption, 8% reflection) *
5. Surface absorptance (solar)	0.93
6. Surface emittance (IR)	0.89
7. Coolant flow rate x specific heat coolant	20 BTU/hr °F ft_c^2**
8. Heat transfer coefficient to liquid coolant	30 BTU/hr °F ft_c^2
9. Back insulation U-value	0.083 BTU/hr °F ft_c^2
10. Heat capacity	1 BTU/hr °F ft_c^2
Collector Plumbing	
11. Heat loss coefficient (to ambient)	0.04 BTU/hr °F ft_c^2
Heat Exchanger	
12. Heat transfer coefficient	10 BTU/hr °F ft_c^2
Thermal Storage	
13. Heat capacity	15 BTU/ °F ft_c^2
14. Heat loss coefficient	0 BTU/hr °F ft_c^2 (i.e. assuming all heat loss is to heated space)
Heat Distribution System	
15. Design water distribution temperature	> 133° F ***
Controls	
Building maintained at 68°F [&]	
Collectors on when advantageous	

* 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 page 33 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%

[&] It is assumed that internal sources such as people, lights, and equipment will be sufficient to raise the internal temperature by 4°F, i.e. up to the comfort standard of 72 °F.

The "standard" liquid system, whose parameters are listed on page 17, represents a good average solar heating system of its type. It consists of a metal flat plate collector, a water storage tank, a heat exchanger, and forced air distribution to the living space. The design parameters for these components and other features of the system were selected by means of computer simulations in which each parameter was varied singly in order to obtain a value for which the solar performance is optimized or nearly optimized.

Given, then, the nominal standard liquid system as a whole, one can proceed to vary each one of the parameters singly through simulated yearly heating loads based on actual weather data in order to determine the effect of such variations on the performance of the system.

This has been done for each design parameter for Seattle, Washington; Medford, Oregon; Fresno and Santa Maria, California; Phoenix, Arizona; and Bismarck, North Dakota, for "solar heating design years." The design year for each city was established by running hour-by-hour simulation analyses on 10 to 12 years of data for each of the cities. The design year for each is the year that most closely corresponds to the group average in terms of the solar heating fraction. The design years, the degree-day values and solar radiation data for these design years are shown in the tables following.

Each parameter variation was done for two different building loads which were selected to give 75% and 40% solar heating fractions for the nominal values of the parameters. Collector size, an important parameter, is treated separately in the section beginning on page 52.

For each calculation, only the parameter under study was varied; therefore all of the complex system interactions which result from changing that parameter have been taken into account.

The results of these calculations are intended to give information on trends of solar performance as a function of parametric variation, rather then be taken as absolute results. No actual solar heating system will perform exactly as predicted. Furthermore, as the design parameters were varied singly to determine their individual effect on the overall performance of the system, care should be used in simultaneously varying two or more parameters, since some of them 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 system) will be different than the performance estimated by changing the

DESIGN YEAR DATA FOR SIX CITIES STUDIED

Design Year Site	Year Starting July 1	Solar Radiation on Horizontal Surface BTU/yr ft^2	Heating Deg-Days
Phoenix	1962	686521	1278
Santa Maria	1956	649922	3065
Fresno	1957	558516	2622
Medford	1961	524605	5275
Seattle	1963	385221	5202
Bismark	1954	479758	8484

parameters individually and adding up the effect of all changes. A separate computer simulation may be needed for an accurate performance prediction for this case.

The results of the computer simulations are to be found below in two forms. First, in summary form for each parameter; and, second, in detail, with the results plotted on graphs.

Since there are twelve curves on some of the graphs, that is, two curves for each of the six cities, one for the nominal 75% solar heating system and one for the 40% system, their order from top to bottom on the graph is indicated by the order of the list of cities on the graphs. For example, if the first city listed is "MD," then the top curve of the set is for Medford, and so forth. Fifteen parameters have been studied for the standard liquid system. Parameters one through six and nine are the same as for the standard air system, a discussion of which begins on page 45.

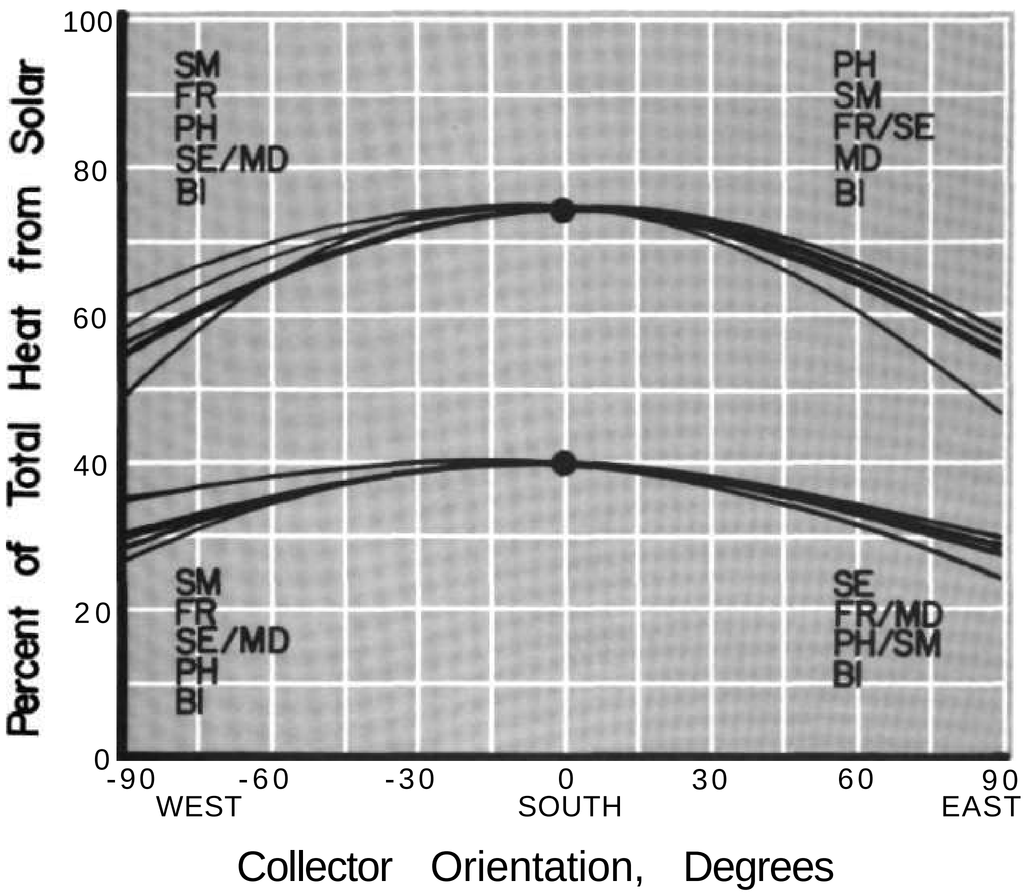
In brief, the effects of changes in design parameters for the standard liquid system can be summarized as follows.

1. **Collector Orientation:** Optimum orientation is usually due south. Variations east or west of up to 30° reduce performance by only 2.4% to 5%, but larger variations can reduce performance substantially. (Same for standard air system.)
2. **Collector Tilt:** The optimum tilt angle will range from latitude plus 10° up to latitude plus 25°, depending on climate. Like orientation, small variations from the optimum have a small effect. Variations of more than about 20° can substantially reduce performance. (Same for standard air system.)
3. **Number of Glazings:** Justification of the extra cost of double glazing will depend on climate. Collector area can be reduced by only 5% by the use of double glazing in Phoenix but by 25% in Bismarck. (Same for standard air system.)
4. **Collector Glass Transmissivity:** Improving glass transmissivity by 6% by the use of "water white" glass should increase the annual solar heat collected by 2% to 5%, depending on the site. Improving transmissivity by reducing glass reflectance should have an even greater effect. (Same for standard air system.)
5. **Collector Surface Absorptance:** Surface absorptance should not be less than 90%. (Same for standard air system.)
6. **Collector Surface Emittance:** A highly selective surface can reduce the required collector area by 15% in the warmer climates and by up to 38% in the colder climates such as Bismarck. Surface durability may be a problem. (Same for standard air system.)
7. **Collector Coolant Flow Rate:** Although not a critical design parameter, the flow rate should be designed to obtain a coolant temperature rise of about 20°F under peak conditions.
8. **Collector Heat Transfer Coefficient:** A net effective heat transfer coefficient greater than 10 BTU/hr °F per square foot of collector is adequate to achieve near maximum performance.
9. **Collector Back and Side Insulation:** U-values of no greater than 0.1 should be integrated into the collector system. (Same for standard air system.)
10. **Collector Heat Capacity:** The collector design should minimize the mass of metal and the fluid inventory in the collector.
11. **Distribution Pipe Insulation:** Collector-to-storage distribution

pipe heat losses can be kept acceptably low by insulating to a U-value of 0.2 BTU/hr °F ft_c^2.

12. **Heat Exchanger Effectiveness:** A heat transfer coefficient of at least 10 BTU/hr °F ft_c^2 should be achieved.
13. **Thermal Storage Heat Capacity:** Thermal storage for much more than overnight does not improve the yearly system performance very much, but fairly severe performance losses are predicted if the storage provides less than 10 BTU/ft_c^2 °F (1.2 gal of water/ft_c^2).
14. **Heat Losses From Storage:** If the storage tank is located within the living space, heat losses are far less detrimental than if the tank is located outside the living space, in which case losses can be severe without substantial insulation.
15. **Design Water Temperature:** As the design water temperature is decreased, the required air flow in the building is increased with an increase also in solar heating performance—but at higher capital and operating costs. Distribution by means of baseboard convectors is less effective than by forced air.

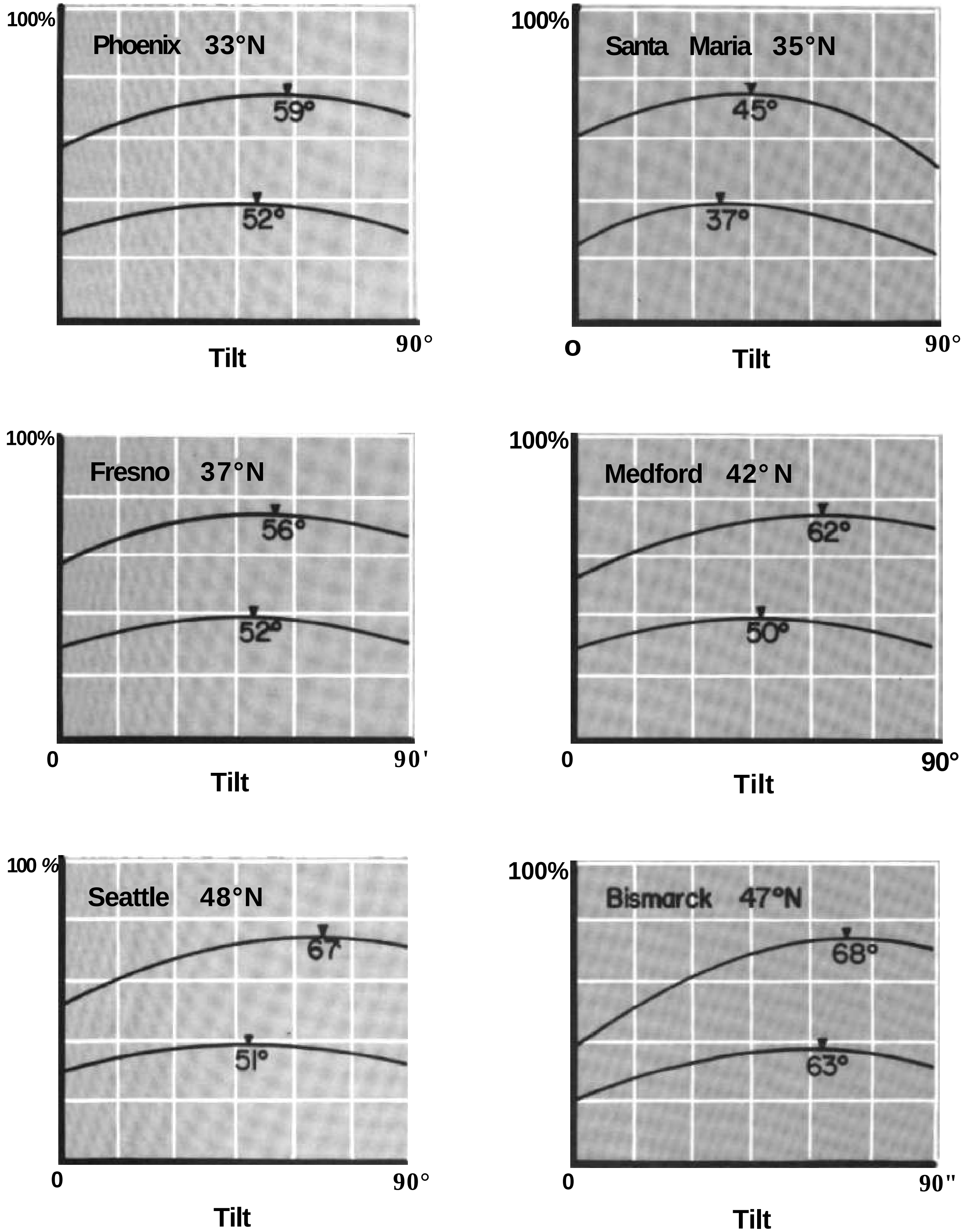
1. Collector Orientation. A due south orientation is near optimum for all six cities studied. The decrease in performance for variations from due south is somewhat greater for the colder northerly cities than for the warmer southerly ones, but variations of up to 30° east or west reduce performance by only 2.4% on the average, or 5% at the most.



2. Collector Tilt. Collector tilt angle is an important design consideration. Common practice is to use a tilt equal to the latitude of the site plus 10° or 15°. Clearly latitude is important; however, the optimum tilt will also be affected by the monthly distribution of the heating load—whether it is concentrated in a short period of two or three months or spread out over half the year—and by the solar heating fraction. It is important to note that the curves have a relatively flat maximum. This means that major deviations from the optimum tilt have only a minor effect on performance.

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

Percent of Total Heat from Solar



Degree of Tilt is taken from Horizontal

3. Collector Glazing. Collector glazing serves three functions: (1) it allows for transmission of solar radiation to the collector absorber surface; (2) it reduces convective heat losses to the surrounding air; and (3) it absorbs the infrared (long wavelength) radiation that would otherwise radiate back into the environmental from the absorber surface.* At least one sheet of glazing is essential in space heating and domestic hot water applications, while none is generally required for swimming pool heaters. Changing one or another characteristic of the glazing produces a net yearly effect on performance as the result of complicated interactions which depend not only on the glazing but on the rest of the system and its application, which together determine the operating regime of the collector. Intuitive judgements of performance based on collector performance charts alone, such as the one given on page 12, are often misleading or wrong, as they fail to take into account the interaction of all the other factors. In this and the following three sections, the effects of the parameter changes on the net yearly performance of space heating applications is discussed for the six sites.

The decision to use one or two sheets of glass depends on a trade-off between the added cost of the extra sheet and the increase in performance to be gained.

The added cost of a second sheet of glass will be determined, in part, by whether the glass procurement is small or large. Cost estimates should include additional parts required by the double-glazing, added glass installation costs, and the additional shipping and collector installation costs resulting from the heavier weight of the double-glazed collector. Estimates should also factor in extra costs associated with replacing broken glass, protecting against the higher operating temperatures to be expected during periods when the collector is not cooled, and dealing with the condensation of water vapor inside the collector.

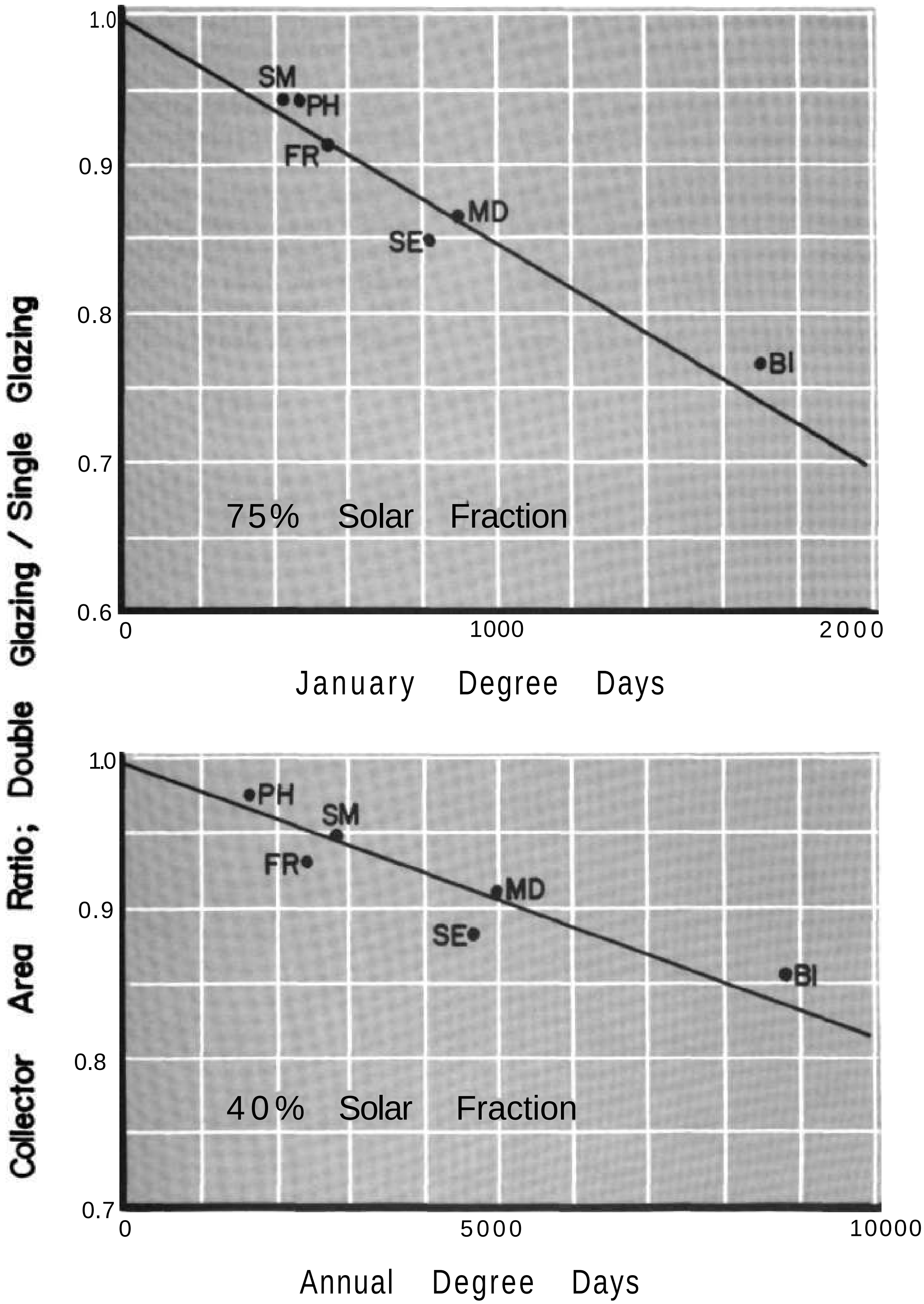
The performance to be gained by an additional sheet of glass, as shown in the table below, is predominantly the result of the difference in ambient temperature at the different sites. This suggests a correlation with heating degree-days. In the two graphs on page 25, it can be seen that the collector area ratio for a 75% solar system correlates well with the January degree-days, whereas a 40% solar system correlates better with the annual degree-days.

To illustrate how these graphs may be used, suppose that a solar heating system with single glazing will cost 10% less than one using double-glazed collectors. It can be seen, then, that it would be more economical to install a 75% solar heating system with double-glazed collectors at sites where the January heating load exceeds roughly 650 degree-days, such as Medford, Seattle, and Bismarck, and more economical to install single-glazed collectors in Phoenix, Santa Maria, and Fresno.

Collector Area Ratio: $\frac{\textbf{Double Glazed}}{\textbf{Single Glazed}}$

Solar Heating Fraction / **Site**	75%	40%
Phoenix	0.94	0.97
Santa Maria	0.94	0.94
Fresno	0.91	0.95
Medford	0.86	0.91
Seattle	0.84	0.88
Bismarck	0.74	0.85

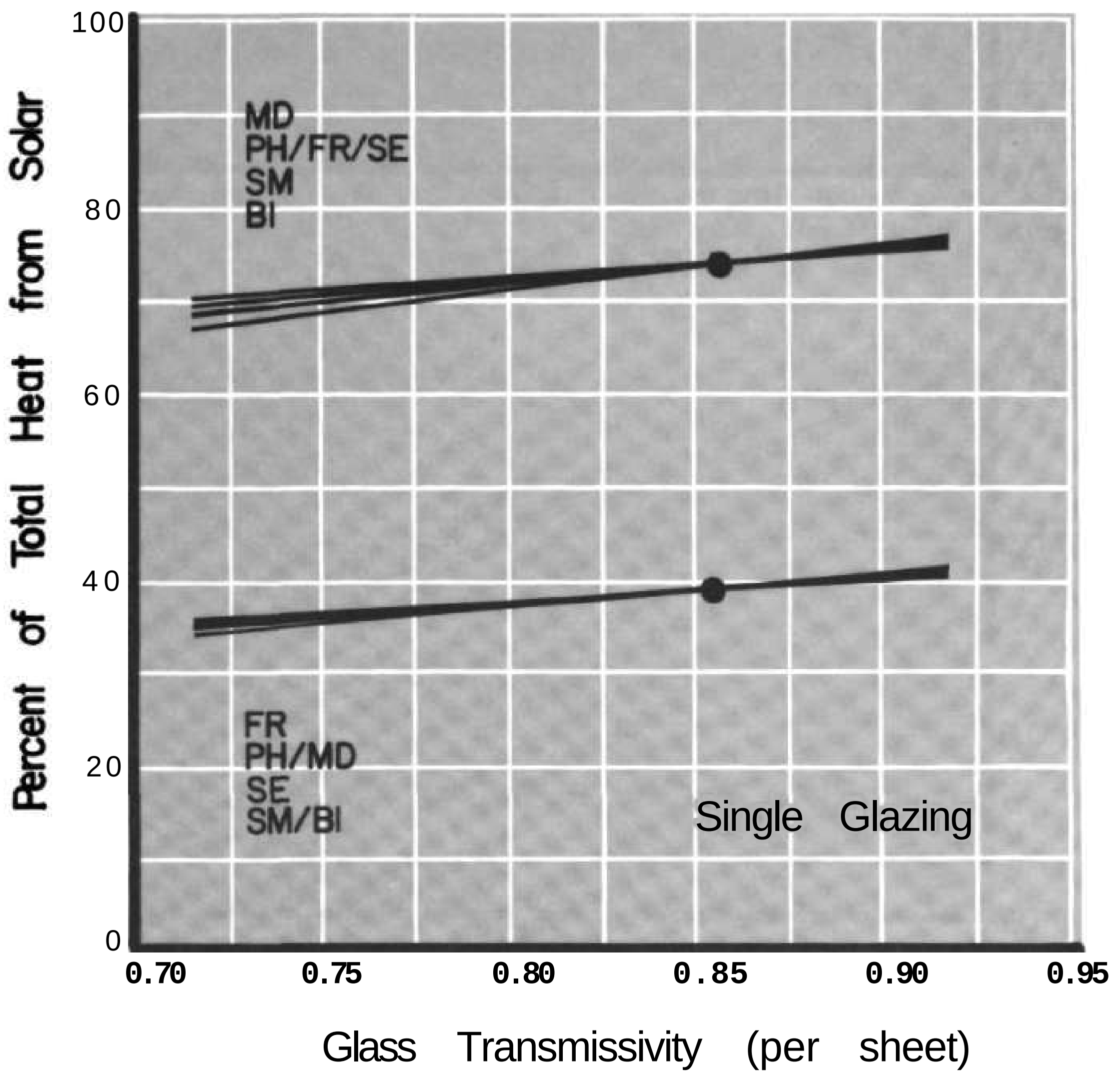
*This is not to imply that radiation losses from the collector are thereby eliminated. Glass has a high emittance and thus also radiates energy—but at the lower temperature of the glass.


Collector Area Ratio; Double Glazing / Single Glazing
1.0
0.9
0.8
0.7
0.6
SM
PH
FR
MD
SE
BI
75% Solar Fraction
0
1000
2000
January Degree Days
1.0
0.9
0.8
0.7
PH
SM
FR
MD
SE
BI
40% Solar Fraction
0
5000
10000
Annual Degree Days

4. Glass Transmissivity. Solar radiation is diverted in the collector glazing assembly through two effects. The first is the reflection of sunlight from the surface of the glass. The second is the absorption of solar energy by the glass itself. In normal window glass, these effects are nearly equal. Reflection is very dependent on the angle of incidence of the solar radiation, and the absorptance is also to a lesser extent.

It has become customary to lump these effects into a single variable called "glass transmissivity," which is the total energy transmitted through the glass for sunlight striking it perpendicularly. This parameter is relatively easy to measure. For standard plate glass, reflection accounts for a reduction in transmissivity of about 8% and absorption for about 6% at normal incidence, resulting in a transmissivity of about 86%.

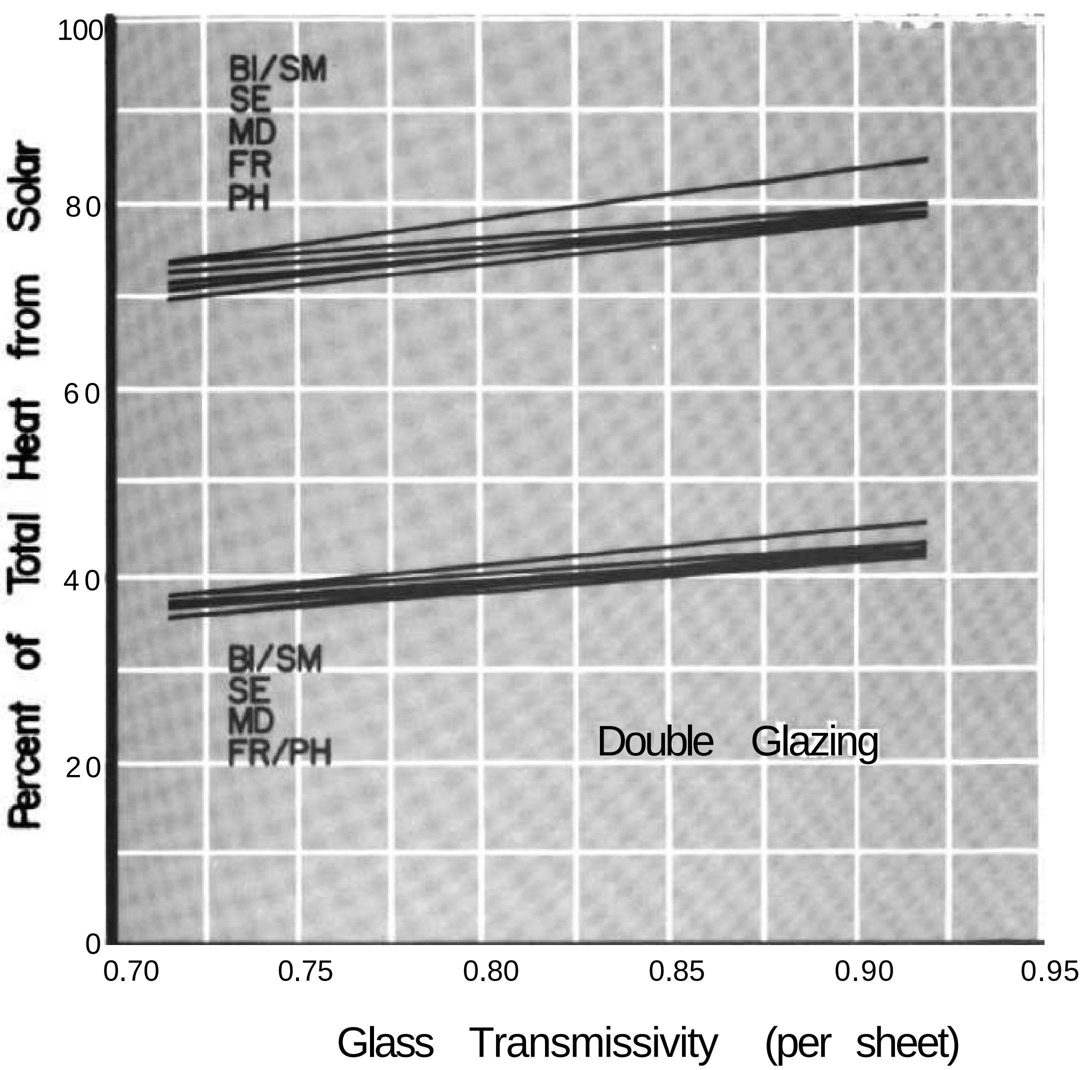
There are techniques for making glass with a total transmissivity in excess of 95%. These involve lowering the iron content in order to reduce absorption, and etching the surface to provide a gradation in density in order to reduce



reflection. The etching is barely visible to the naked eye. Coatings can also be used to reduce reflection. Low iron or "white glass" is in commercial production at the present time. Low-reflective glass, although now also in production, is not yet available at prices which would warrant its use in solar collectors.

The curves on the graphs on pages 26 and 27 were determined by varying the absorption with the perpendicular reflectance assumed to be 8%.* They show that the performance decrease is less than the reduction in transmissivity, principally because energy absorbed in the glass is not lost from the collector—it raises the glass temperature and thereby reduces the convective and radiative losses from the absorber surface. A 1% decrease in glass transmissivity reduces the yearly solar energy collected by 0.36% to 0.94%, depending on the site, the number of glazings, and the solar fraction.

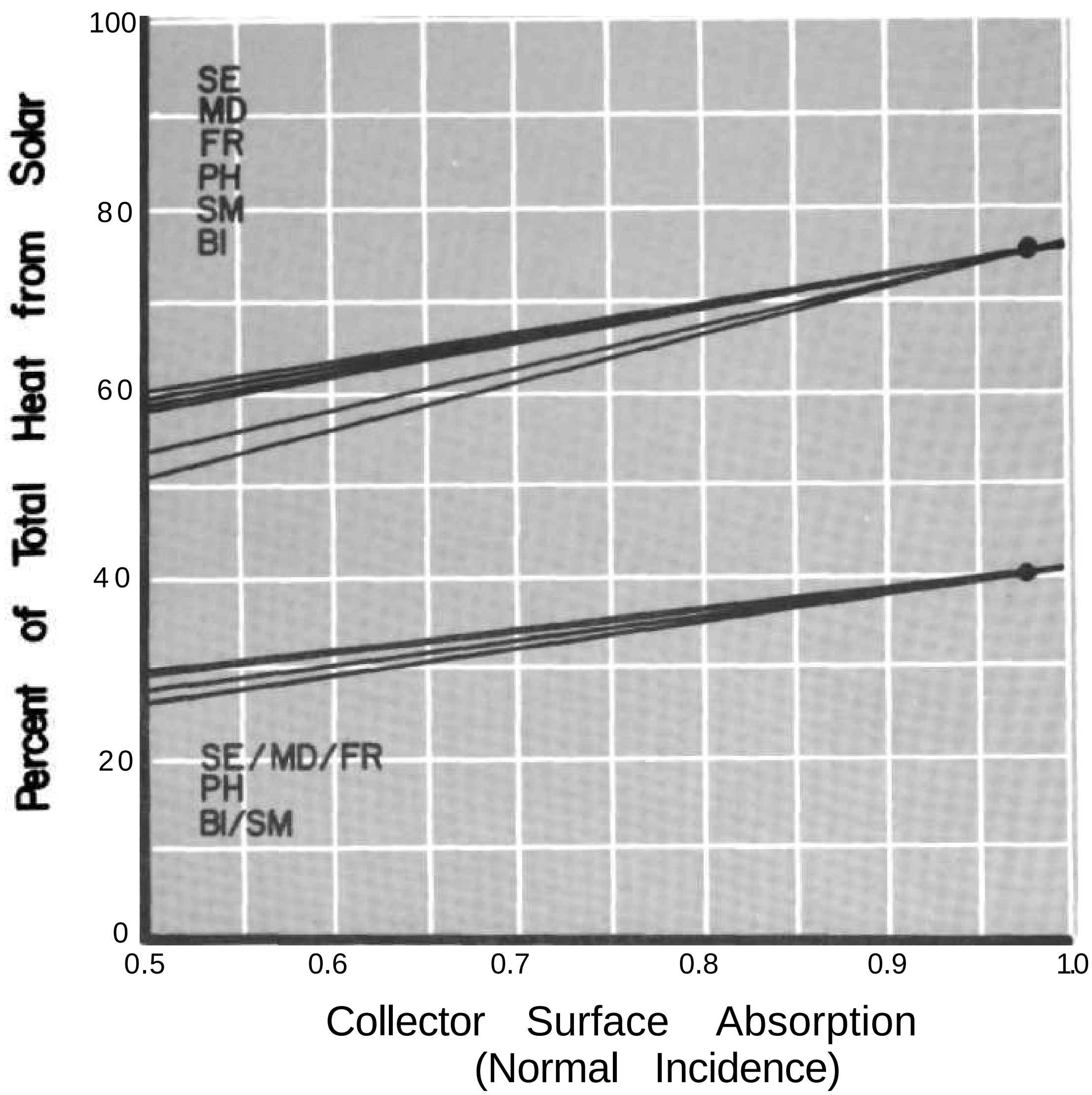
*Refractive index = 1.526.



5. Collector Surface Absorptance. Solar energy that is not absorbed by the surface is not available as heat energy in the rest of the system. Therefore, collector surface absorptance is a very important factor. Nearly all solar collectors are painted black, but even among black paints there will be variations in absorptance from 92% to 98%. The difference between these two values can be easily discerned by the naked eye. It is almost impossible to discern any relief variations in a surface with absorptance of 98% while it is relatively easy to see them on a 92% surface. In any case, collector surface absorptance should not be less than 90%.

It is interesting to note that a one percent decrease in surface absorptance decreases the overall system performance by only 0.44% to 0.67%. This may be due to the fact that with the lower absorptance the system runs somewhat cooler with correspondingly lower heat losses throughout.

Durability of the surface should be considered along with its absorptance. Dust or other deposits on the absorber surface may reduce its absorptance.



6. Collector Surface Emittance. A major heat loss in flat-plate collectors is long-wave radiation from the collector surface. Normal collector coatings have a high effective hemispheric emittance for long-wave or infrared radiation — 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 "selective surfaces"—coatings that have a very high absorptance for visible and ultraviolet solar radiation and a low emittance for infrared radiation. Most selective surfaces are either chemical or electro-deposited coatings. When deposited over a high reflectance substrate, a thin coating of metal oxide is opaque for short-wave radiation—and therefore has a high absorptance—and also transparent for infrared radiation. The effective emittance of the surface is the same as the effective emittance of the substrate, which is low since the substrate is highly reflective.

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.

For a single-glazed collector, a substantial performance increase can be obtained by using a highly selective surface, thereby enabling one to decrease the collector area required to achieve a given solar fraction. For example, assuming a single-glazed collector with a selective surface with an absorptance of 0.95 and an emittance of 0.10 in comparison with a collector with a matte black surface (absorptance, 0.98; emittance, 0.89), the collector area of the first in relation to the second for two given solar heating fractions can be tabulated as follows:

$$\textbf{Collector Area Ratio: } \frac{\textbf{Selective Surface}}{\textbf{Matte Black Surface}}$$

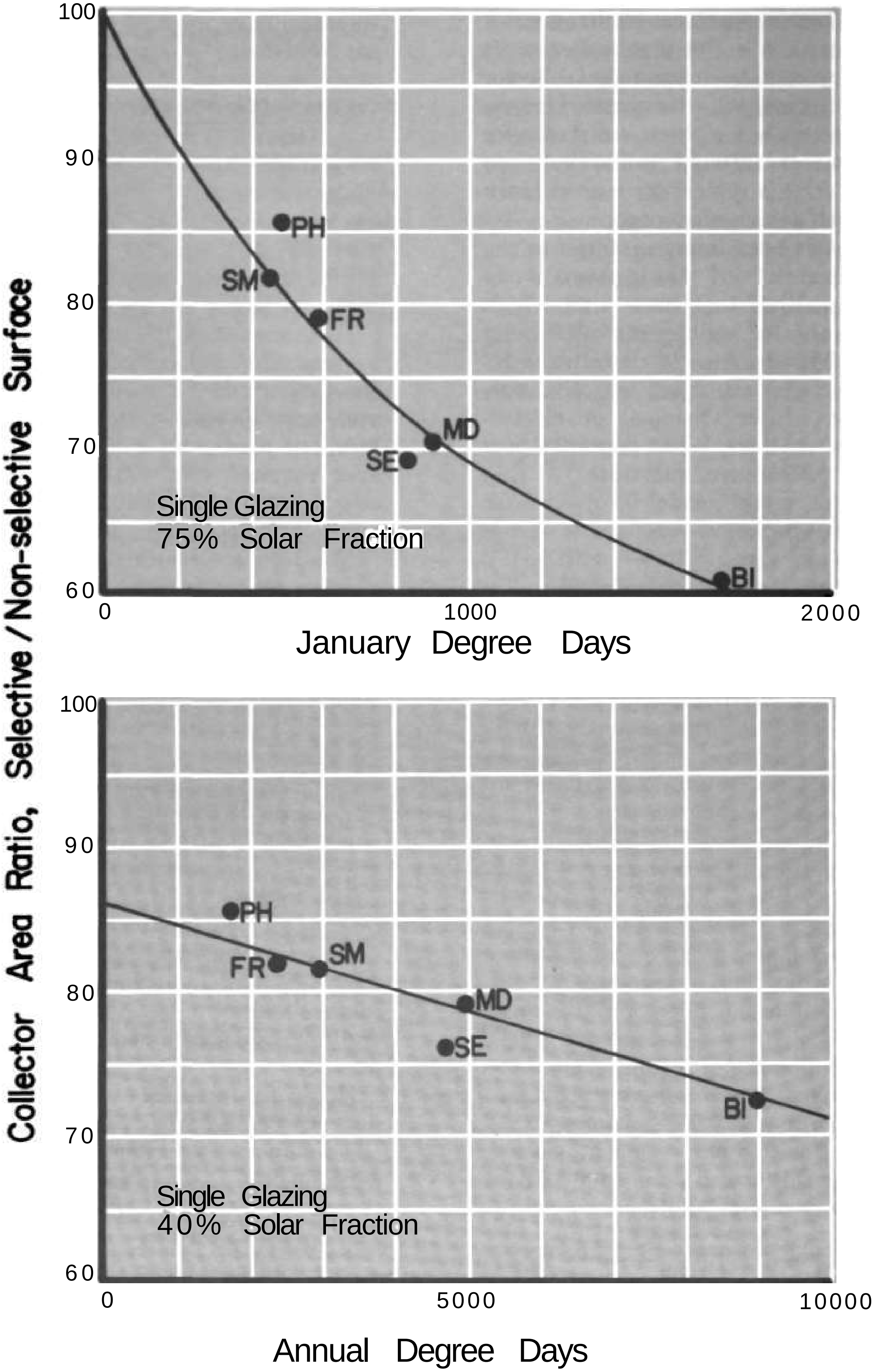
Solar Heating Fraction / **Site**	**75%**	**40%**
Phoenix	0.84	0.85
Santa Maria	0.82	0.82
Fresno	0.79	0.83
Medford	0.70	0.87
Seattle	0.69	0.75
Bismarck	0.62	0.73

In other words, for a 75% solar heating system in Phoenix with a selective surface, one needs only 84% of the collector area required for a system without a selective surface. The figures above were taken from detailed simulation analyses.

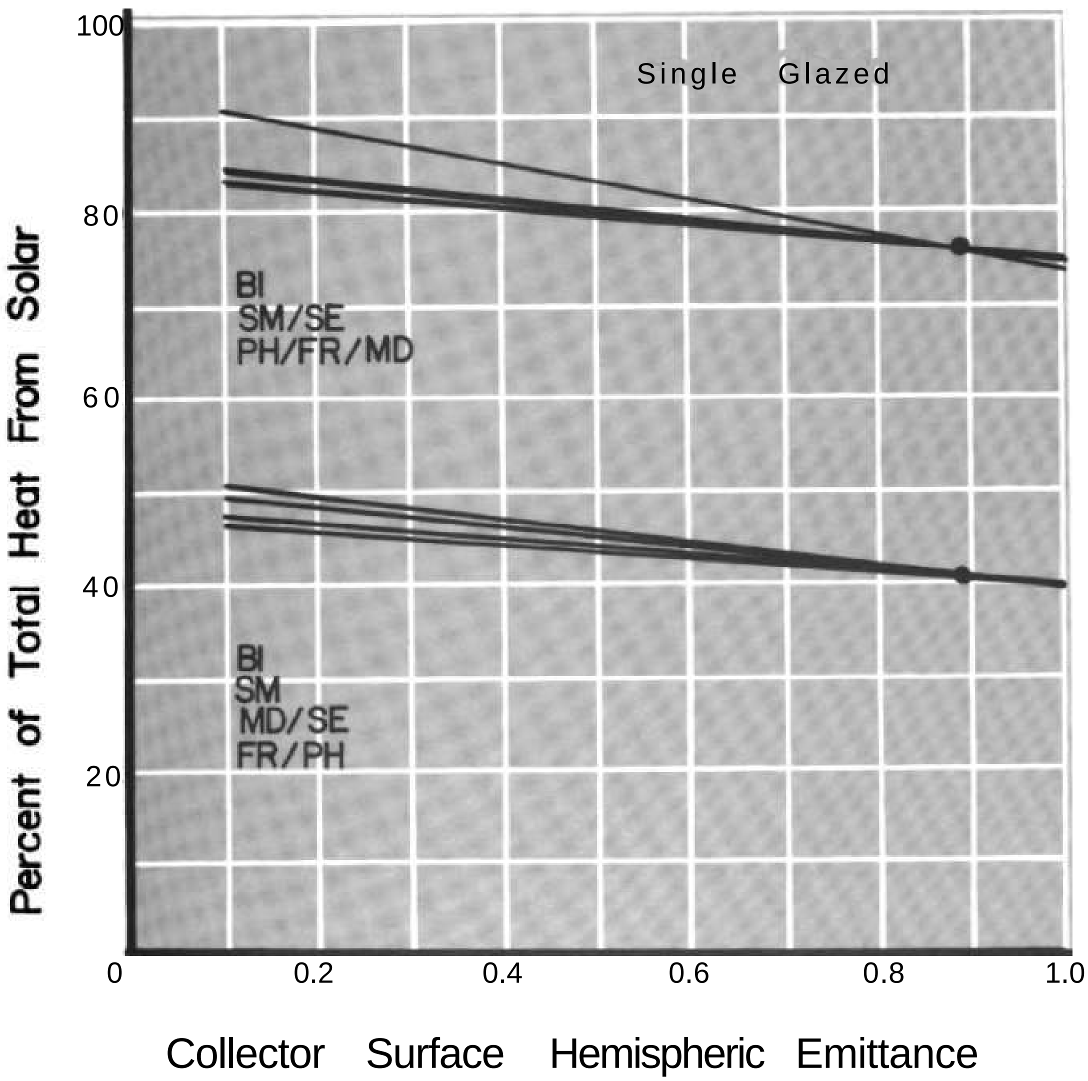
In all cases, performance was actually reduced by the use of double glazing over a selective surface compared to single glazing over the selective surface. One of the primary benefits of double glazing is its absorption of infrared radiation from the collector surface. With a selective surface, however, this effect is small; and the increased absorption and reflection caused by the second layer of glass is more of a loss than the gain resulting from decreased convective losses.

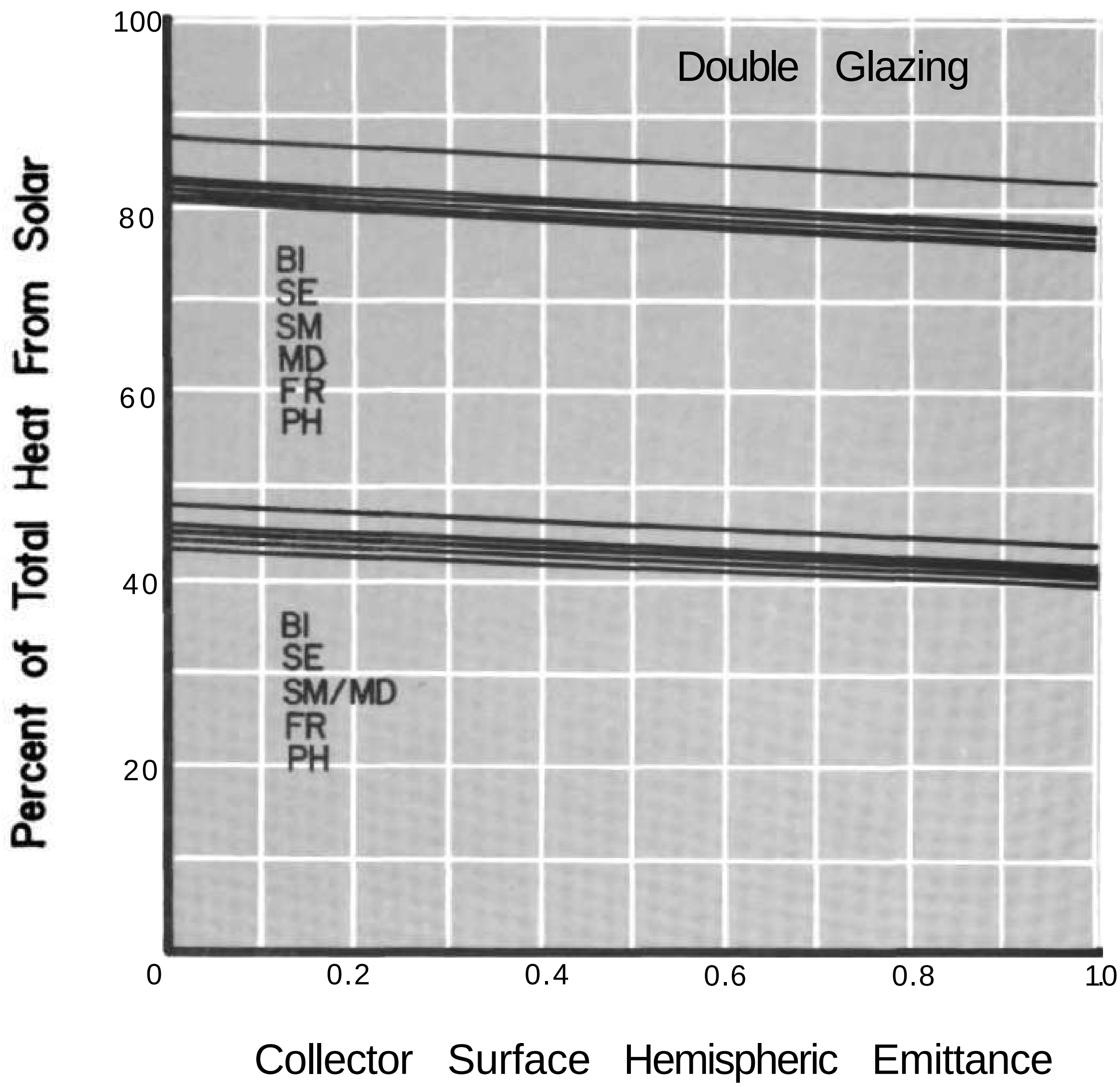
The figures in the table above show a reasonable correlation with heating degree-days indicated on the graphs of page 30. The improvement in performance is almost linear with changes in emittance, as can be seen in the graphs on pages 31 and 32.

However, the use of a selective surface depends on several other factors besides performance. Cost is one. For example, assuming the absorptance and emittance figures above, a selective surface will


Collector Area Ratio, Selective / Non-selective Surface
100
90
80
70
60
PH
SM
FR
MD
SE
BI
Single Glazing
75% Solar Fraction
0
1000
2000
January Degree Days
100
90
80
70
60
PH
FR
SM
MD
SE
BI
Single Glazing
40% Solar Fraction
0
5000
10000
Annual Degree Days

reduce the required collector area in Medford by about 30% over a collector with a non-selective surface. The question then becomes whether an array of collectors with a normal black surface will cost more than an array of selective surface collectors 30% smaller in area. The appearance of the building and the availability of roof area may have a bearing on this choice. Durability is also a factor—some selective surfaces have shown a tendency to break down both mechanically and chemically over a period of time. Other coatings, notably "black chrome," are relatively stable but are expensive.

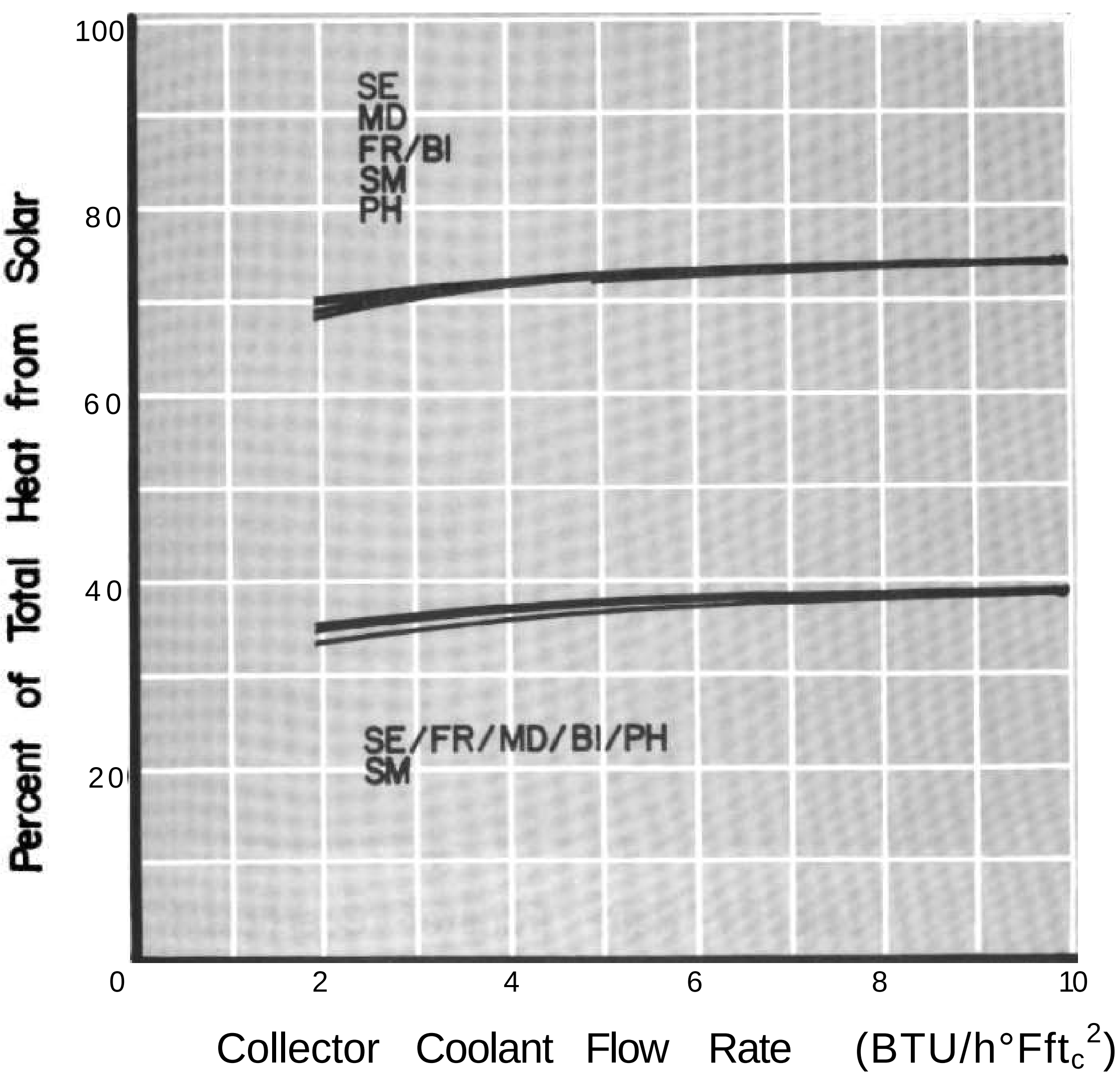



Double Glazing
Percent of Total Heat From Solar
100
80
60
40
20
0
0.2
0.4
0.6
0.8
1.0
BI
SE
SM
MD
FR
PH
BI
SE
SM/MD
FR
PH
Collector Surface Hemispheric Emittance

7. Collector Coolant Flow Rate. Collector coolant flow rate is important in a liquid collector primarily because increases in the flow rate reduce the collector temperature rise which in turn reduces the amount of heat lost from the collector for a given inlet fluid temperature.

The curve below is plotted as the function of the coolant flow rate in units of BTUh/°F ft_c^2, and can thus be used for fluids with different specific heats. For example, for water with a specific heat of 1 BTU per lb, a value of 15 on the plot corresponds to 15 lb/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.

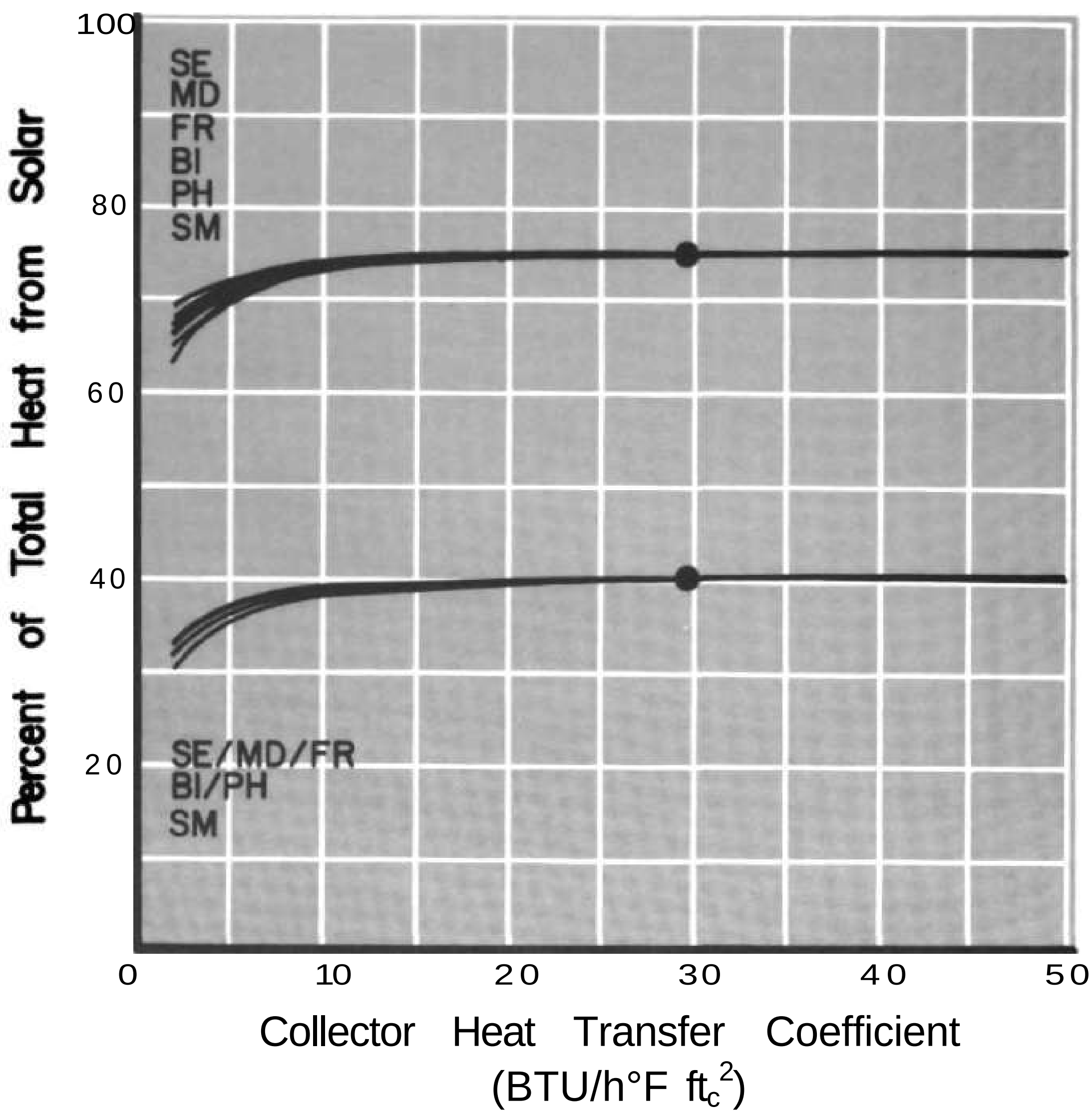
The proper choice of coolant flow rate will depend on the pressure drop of the collector, the viscosity of the fluid, and the size and cost of available pumps. Although higher coolant flow rates do increase the performance of the system, they also require more energy to pump at those rates. Therefore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



8. 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 collectors is relatively high.

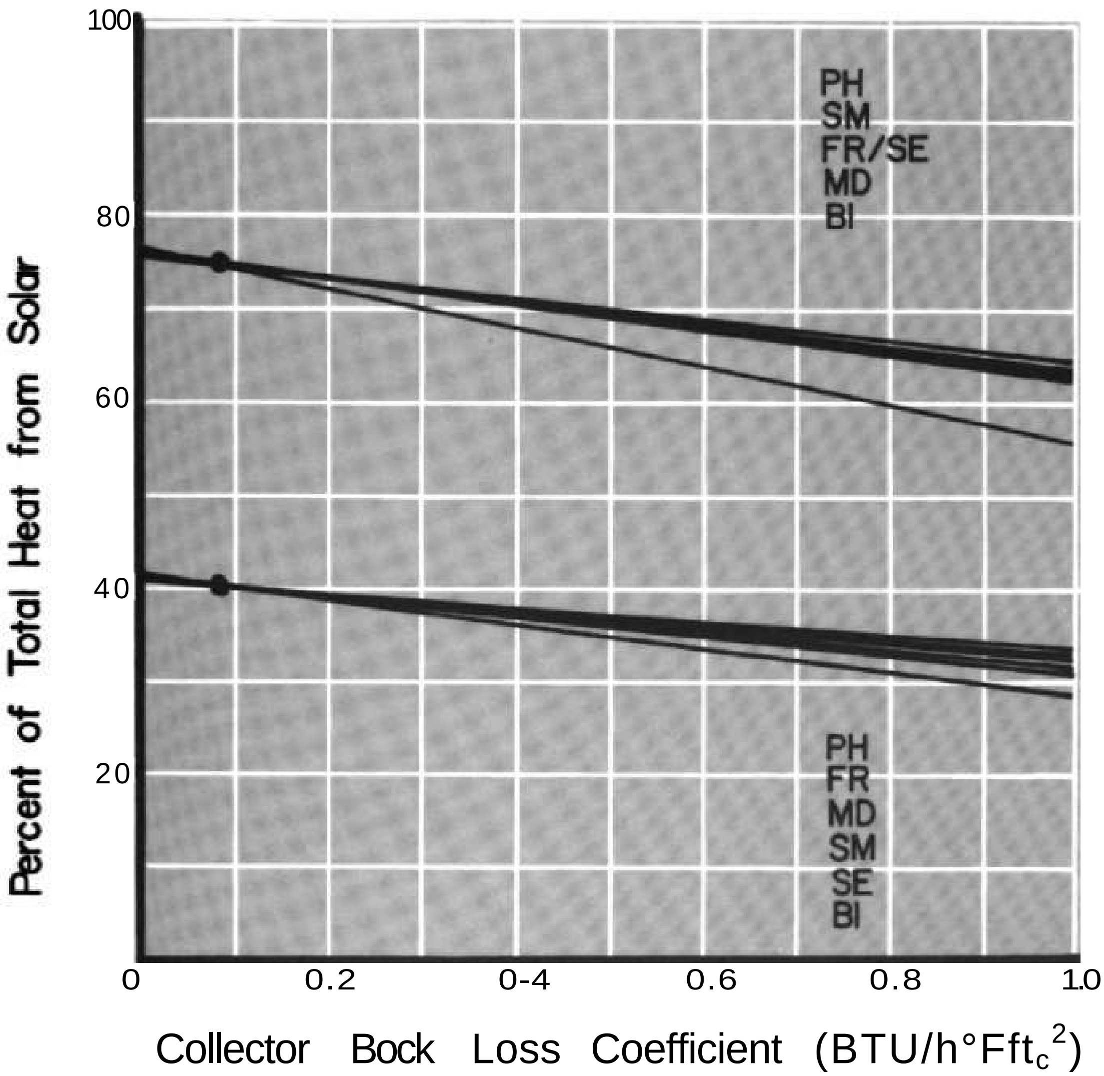
Many liquid collectors are designed with thin sheets of metal connecting the collector tubes. Conduction through this metal sheet represents a decrease in the effective transfer coefficient of the collector.

It can be seen from the graph below that an effective heat transfer coefficient greater than 10 BTU/hr °F per sq ft of collector is adequate to achieve near maximum performance from the solar heating system.



9. Collector Back and Side Insulation. Heat can be lost from the collector surface by conduction through the back and sides of the collector housing. It is good design practice to insulate with the equivalent of three to four inches of fiberglass insulation. The graph below indicates that a U-value of 0.1 should be integrated into the collector housing design for good performance.

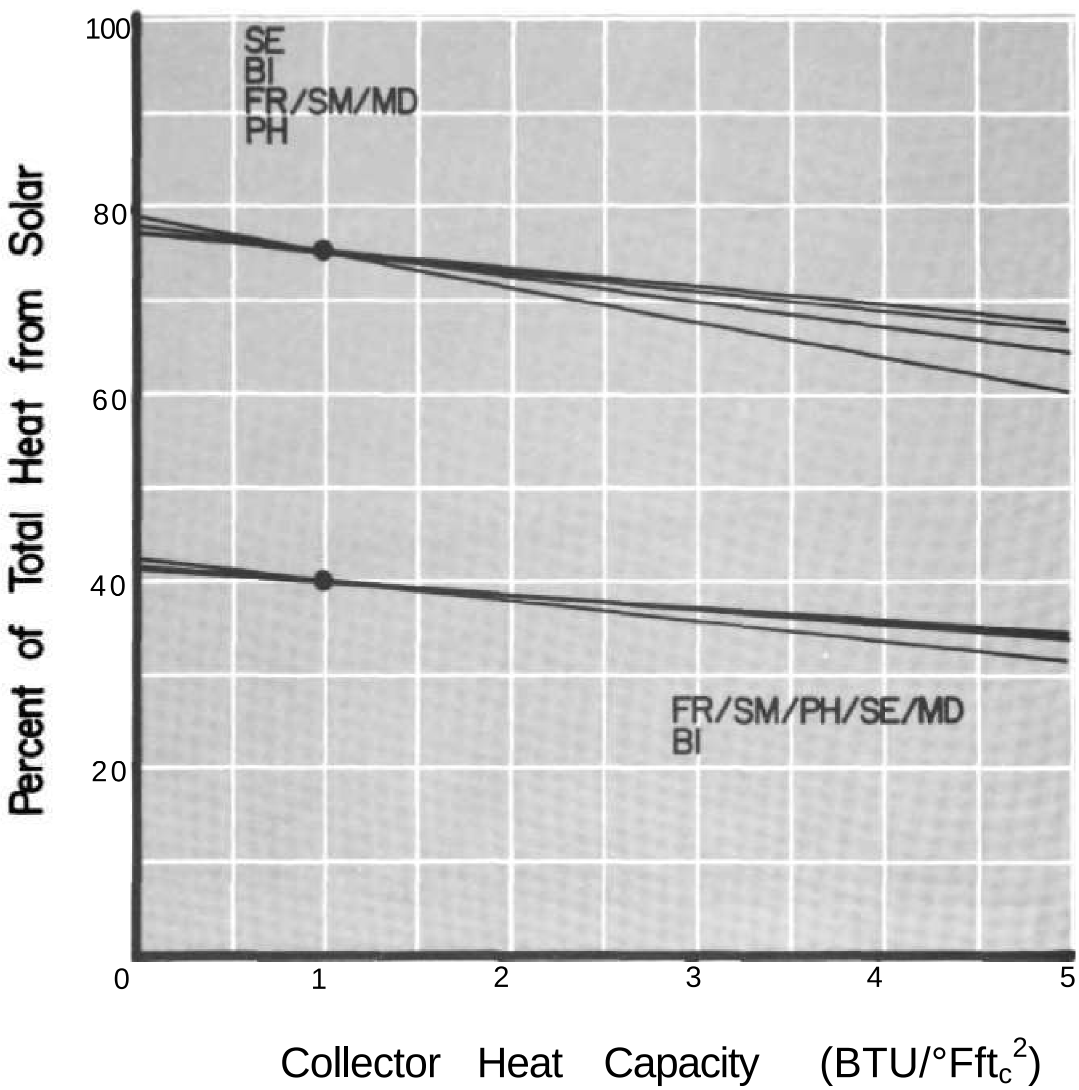
If foam insulation is to be used, care must be taken to select a type which is stable at the maximum temperature which the collector may achieve under no-flow conditions. Only a few of the foam insulation materials are suitable.



10. Collector Heat Capacity. A certain amount of solar energy is required to heat the collector up to operating temperature every morning—energy that is lost to the environment and not recovered at the end of the day. The same effect occurs each time the sun goes behind a large cloud and causes the collector temperature to drop below storage temperature,

The major contributor to collector heat capacity is the fluid stored in the collector and the mass of the metal in the collector and the collector insulation as well as the mass of all components of the heating system that cycle in temperature.

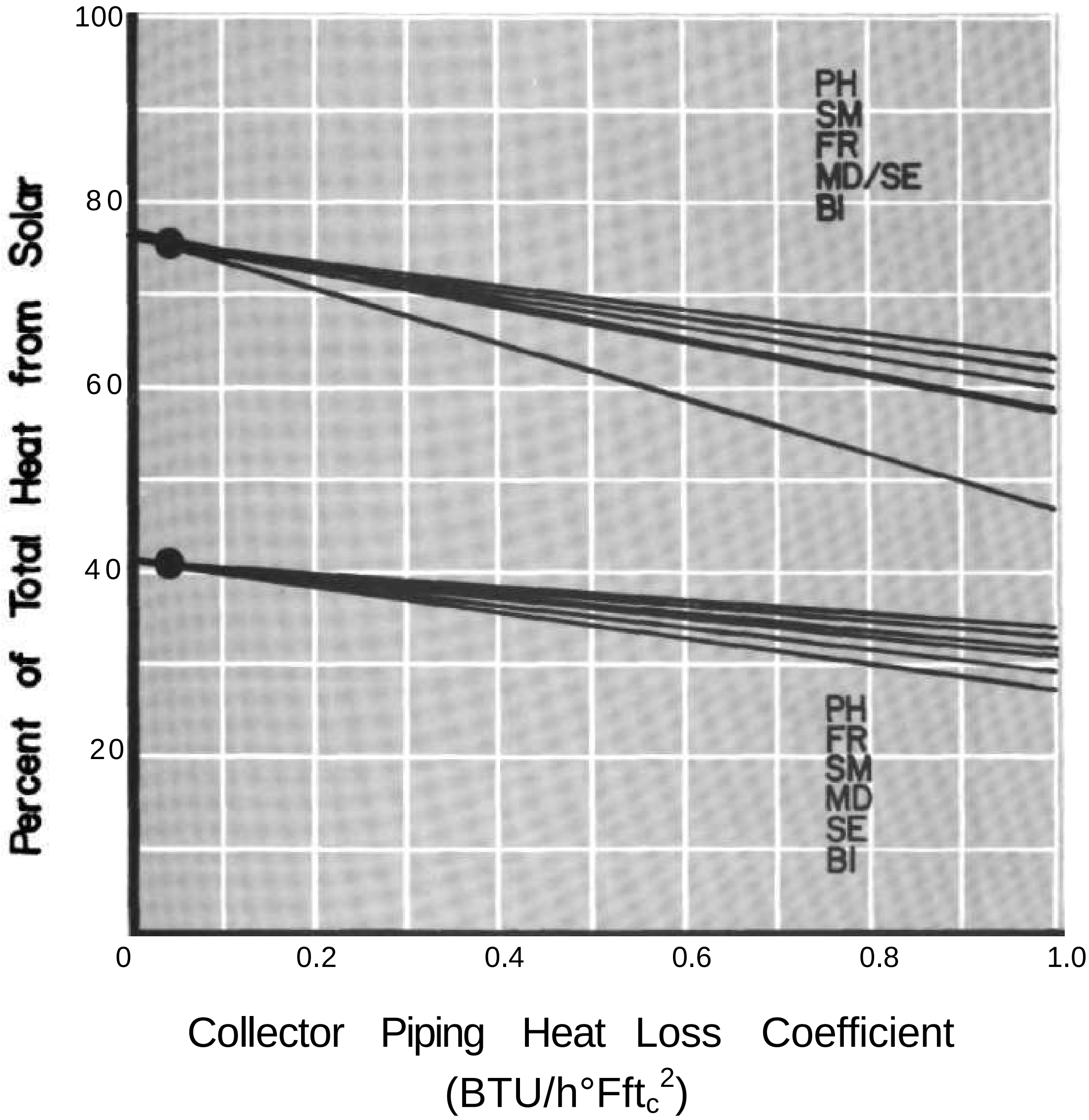
The magnitude of heat capacity depends on collector design, which ought to keep the collector mass and fluid inventory as low as possible. The graph shows that collector heat capacity can have a major effect in the performance of the overall system.



11. Distribution Pipe Insulation. The plumbing which connects the collector array to storage should be insulated. Minimizing heat losses from the pipe on the hot side of the collector is more important than minimizing losses from the collector itself, since the hot side piping may be the highest temperature of the entire system.

As in the case of other parameters, the pipe insulation is put in terms of the total insulating characteristics per square foot of collector area. Since distribution pipes are usually two inches in diameter or less, heat losses from them can be kept relatively low. Therefore a nominal value for distribution pipe heat losses of only 0.04 BTU/hr/°F sq ft of collector was chosen for the standard system calculation.

In order to use the graph below, it is necessary to calculate the total net thermal loss coefficient of all the distribution piping (BTU/hr °F) and divide this by the collector area (ft^2).

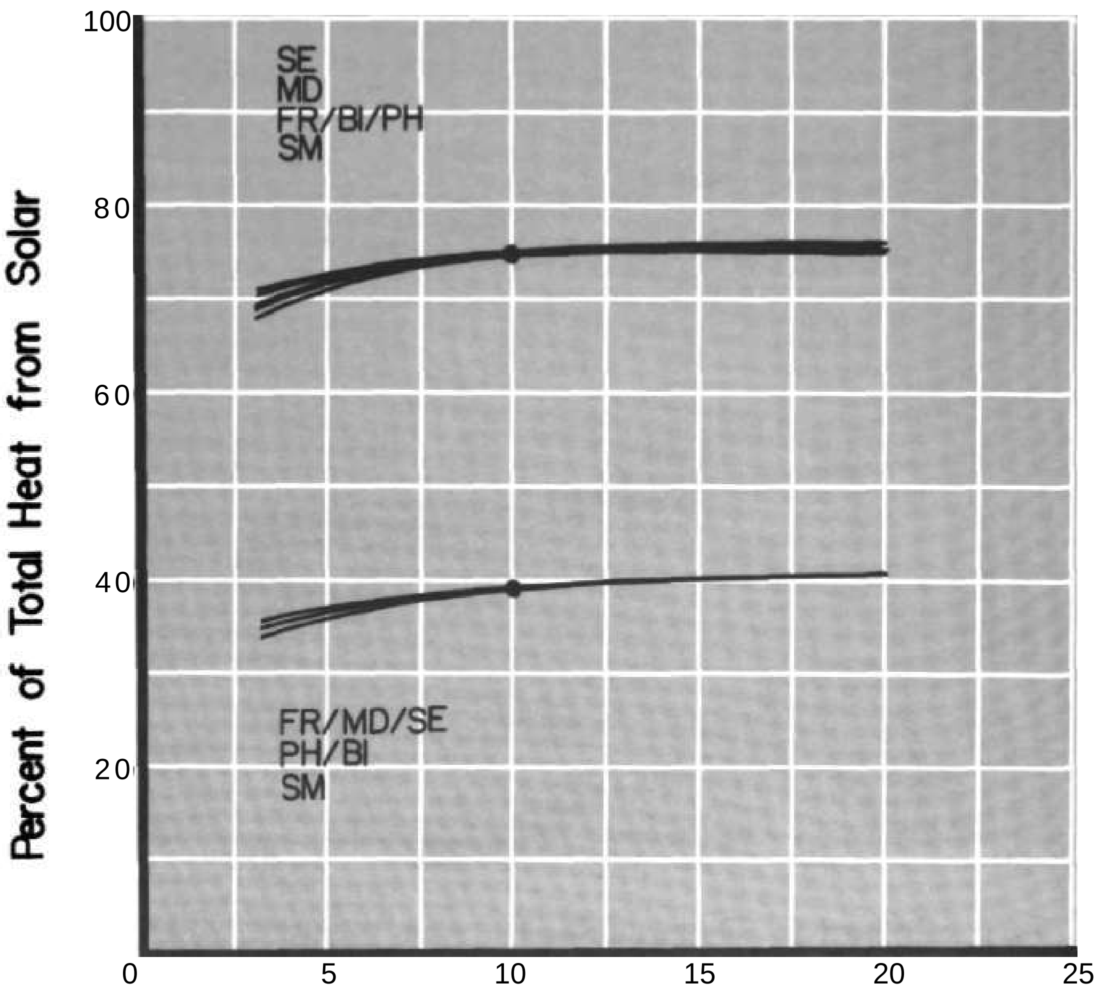


12. Heat Exchanger Heat Transfer Effectiveness. In the standard liquid system, the heat from the collector fluid is transferred to the water in the storage tank by means of a heat exchanger. A common method of implementing this heat exchanger is to pump water from the bottom of the storage tank through a heat exchanger where it picks up heat from the collector coolant and then returns the heated water to the top of the tank.

Another possible approach is, instead of pumping water out of the storage tank, to immerse a heat exchanger or coil of tubing near the bottom of the tank and rely on natural convection to transfer heat from the coil to the storage water.

A net heat exchanger/heat transfer coefficient of at least 10 BTU/hr°F ft_c^2, should be achieved, as can be seen from the graph below. For example, if the total amount of energy being collected at a particular time is equal to 100 BTU/hr ft_c^2, then the average temperature difference between the fluid on one side of the heat exchanger and the other should be no more than 10°F.

The use of some collector liquids with poor heat transfer characteristics, such as 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.



Heat Exchanger Heat Transfer Coefficient $(BTU/h°Fft_c^2)$

13. Thermal Storage Heat Capacity. A water storage tank for thermal storage is considered part of the standard liquid system. It provides energy for those times when the sun is not shining. The nominal thermal storage capacity is 15 BTU/ft_c^2 °F (1.8 gal of water/ft_c^2). This would be sufficient to heat a 75% solar building in Medford for 20 hours, assuming an initial storage temperature of 150°F and an outside temperature of 20°F.* After this time, heat 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 following general formula can be used to determine the carry-through time of storage, i.e., the time elapsed before auxiliary heat is needed to prevent the inside temperature from dropping:

$$t_{ct} = \frac{24S}{Load}\left[\frac{T_s - T_{rm}}{T_{rm} - T_a} - \frac{T_{dw} - 68}{68 - T_{da}}\right]$$

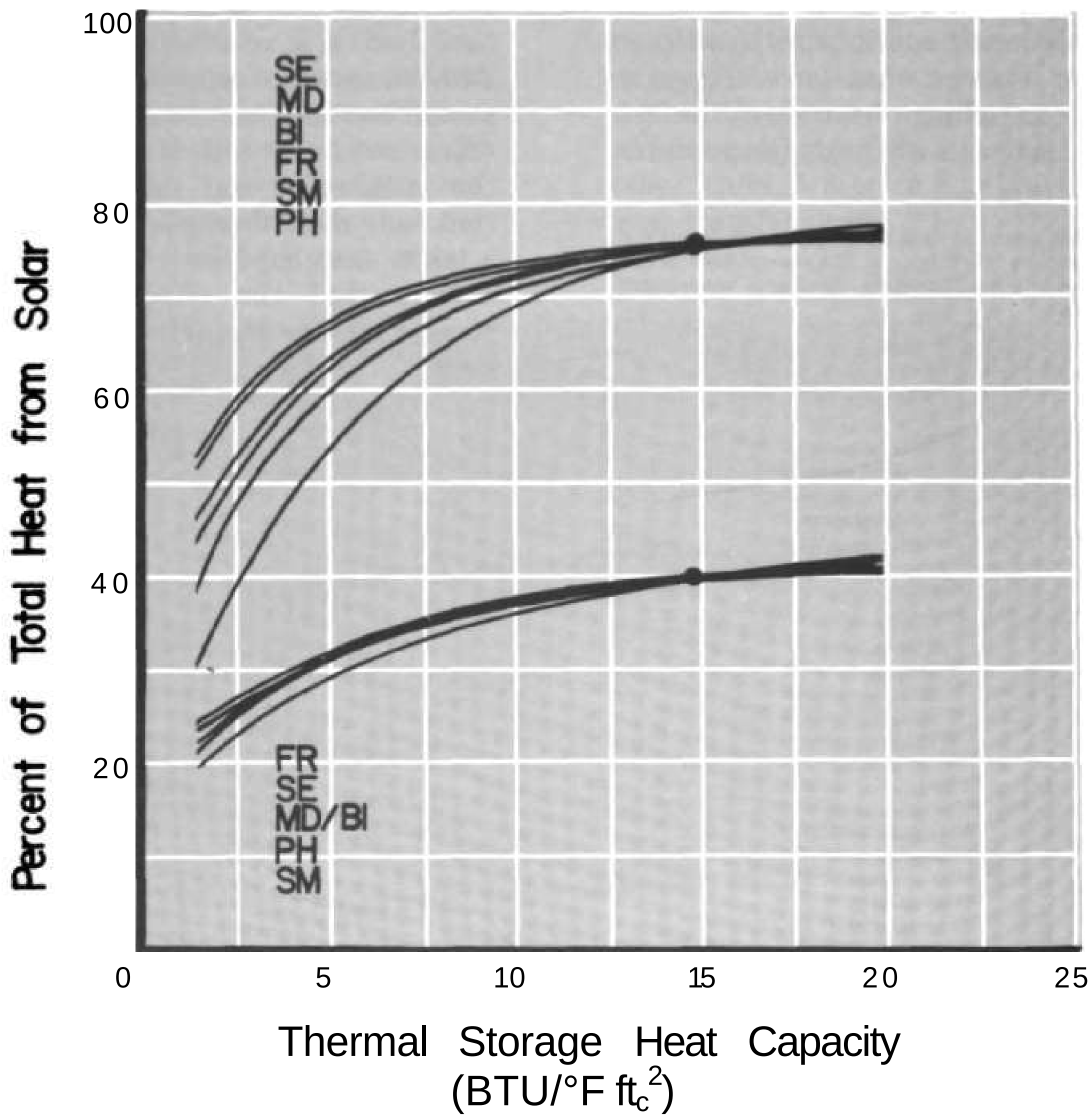
where

t_{ct} = carry through time, hours
T_S = storage tank initial temperature, °F
T_{rm} = room temperature maintained, °F
T_{dw} = design water temperature, °F
T_a = ambient temperature, °F. (assumed constant)
T_{da} = design ambient temperature, °F
Load = Building thermal load. BTU/DD. (see pp. 53)
S = Storage heat capacity, BTU/°F

The simulation analysis indicates that thermal storage in excess of 15 BTU/ft_c^2 does not improve the yearly 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). This means that the primary function of thermal storage is to carry over heat from the daytime hours to the night.

Once the minimal storage mass is established, it is relatively inexpensive to add extra storage capacity. However, one should consider heat losses from storage (described in the following section) since the increase in heat loss from a larger tank may more than offset the gain from a larger heat capacity.

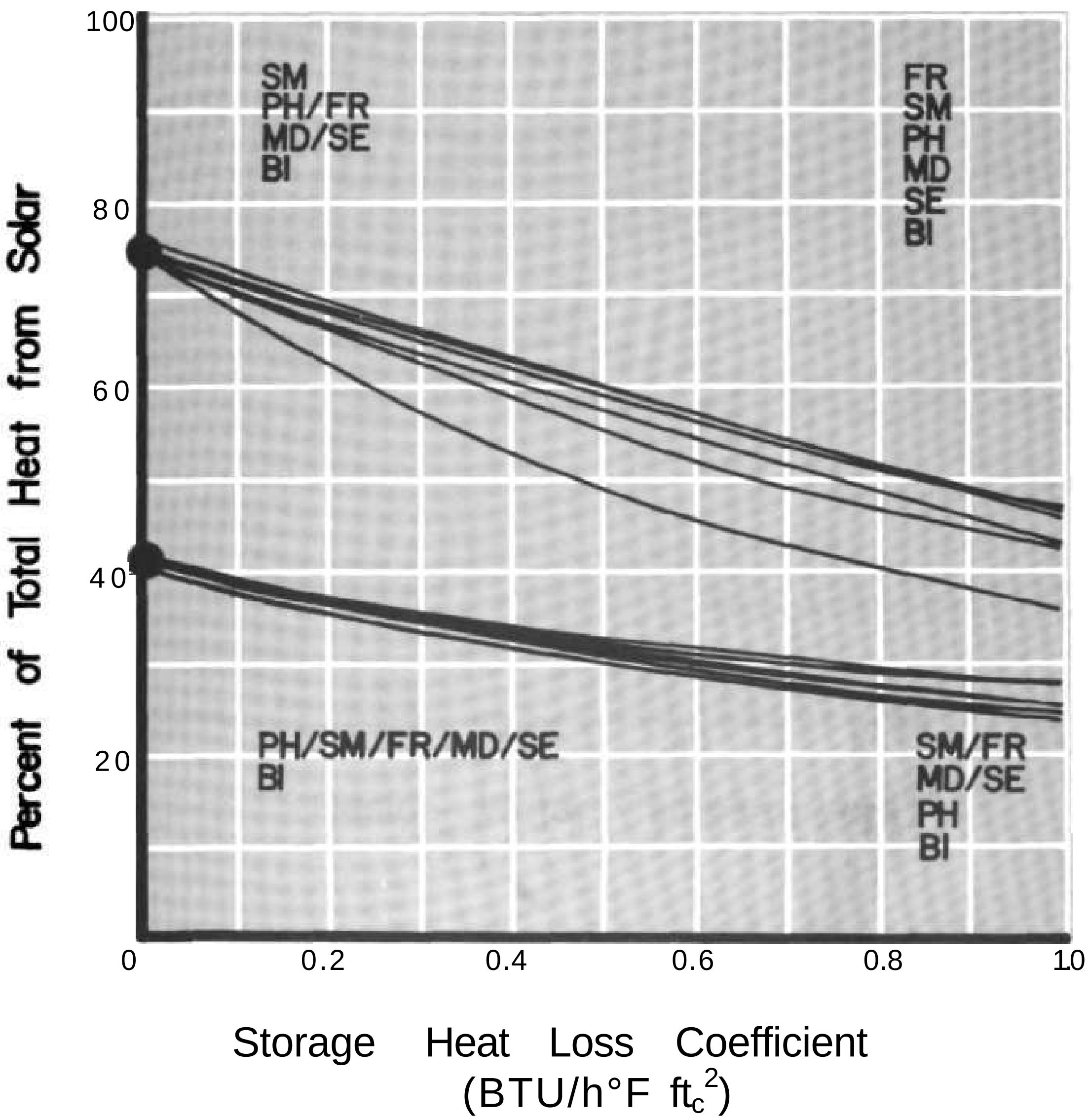
Note that the effect of changing storage mass is strongly location dependent.


100
80
60
40
20
0
5
10
15
20
25
SE
MD
BI
FR
SM
PH
FR
SE
MD/BI
PH
SM
Percent of Total Heat from Solar
Thermal Storage Heat Capacity
(BTU/°F ft$_c^2$)

14. Heat Losses From Storage. The best location for the heat storage tank is within or beneath the space to be heated. In this case any heat lost from storage simply goes into the heated space, and there is no net penalty to the system performance. But the tank should be insulated for times when the living space does not require heat and when, therefore, heat losses from the tank would contribute to overheating the building.

Where the storage tank is located outside the living space, heat is lost to environment by conduction through the tank insulation. Performance degradation here can be very severe.

In order to use the graph below, 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. The result is the storage heat loss coefficient.



15. Design Water Temperature. A finned-tube coil is used to transfer heat from the storage water tank to the air of the living space. The air flow requirement is a function of the design water temperature.

As the design water temperature is decreased, the required air flow is increased, resulting in an increase in solar heating system performance but at higher capital and operating costs.

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 according to the method on page 52. 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

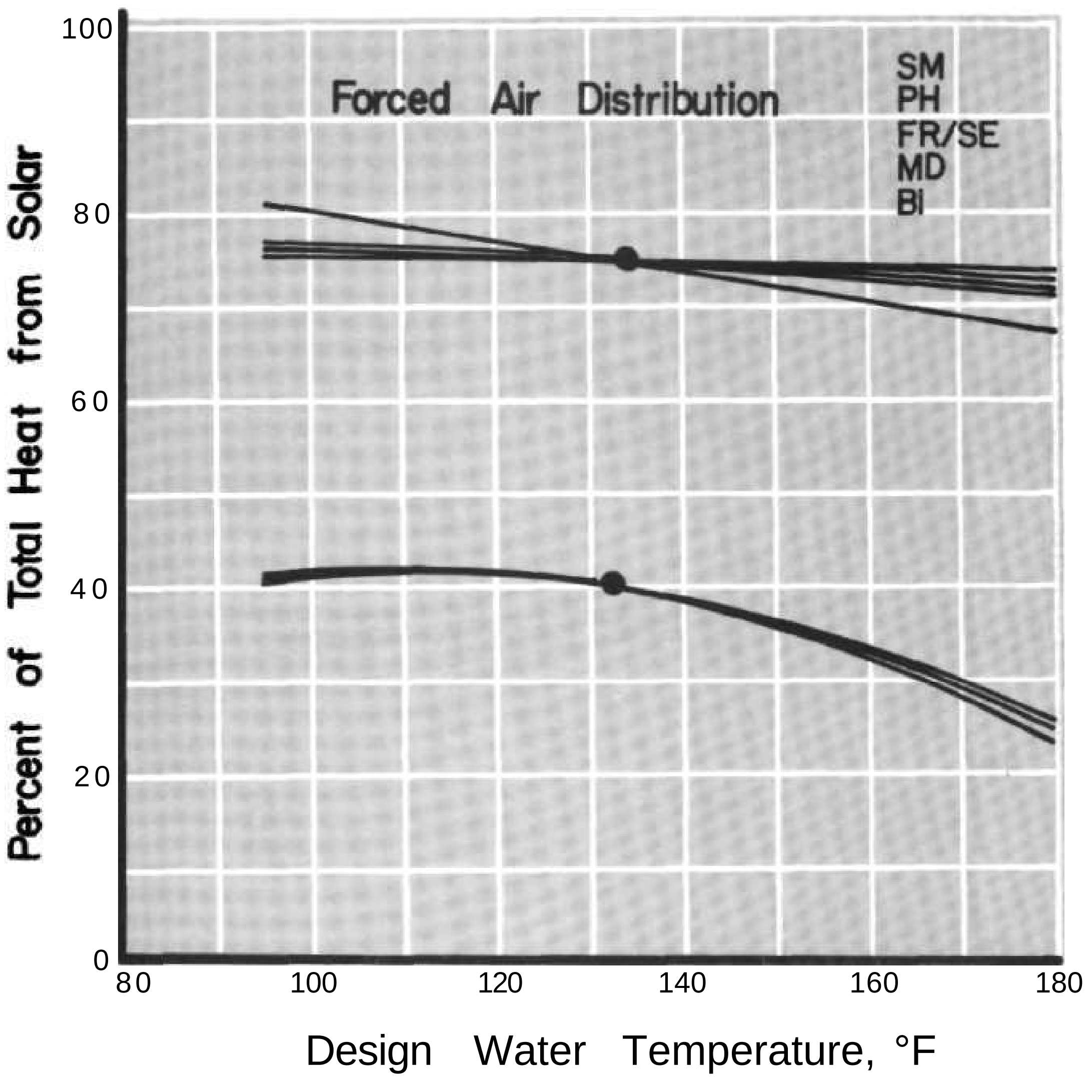
* This is ρc_p x 60 = 1.08 BTU/CFM/°F/hr at sea level.

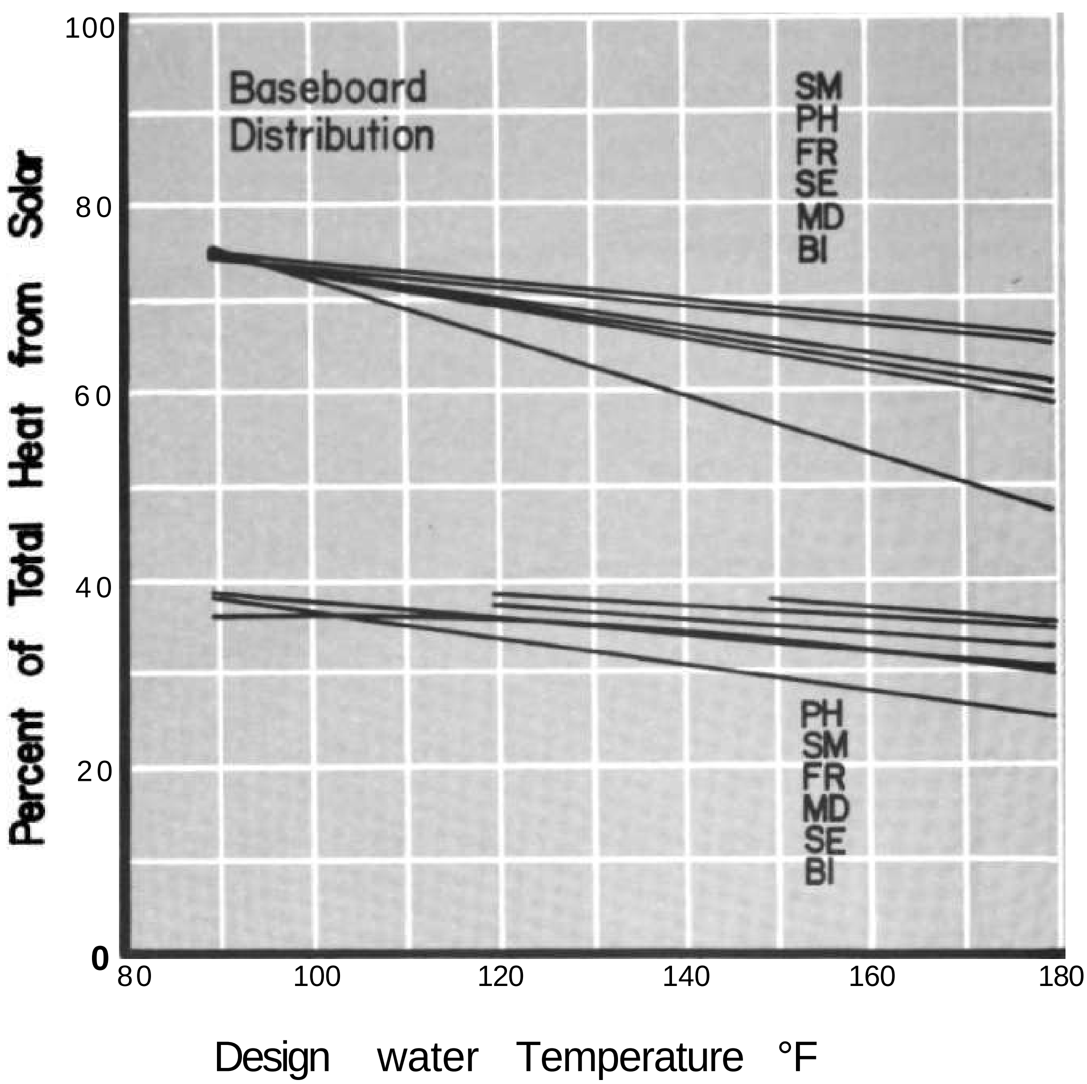
It is to be noted that a less efficient coil would increase the required building air flow, while overall system performance is only a function of the design water temperature.

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. For a solar system, the baseboard convectors should be oversized to operate at the lower temperatures normally available from the thermal storage tank.

The baseboard curve on the graphs 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 simulation is assumed to vary as the 1.176 power of the water-to-room temperature difference and is based on the manufacturer's data.

When the heating load is higher than the amount of heat available at the solar storage tank temperature, then the baseboard system switches to total auxiliary heat.


Forced Air Distribution
SM
PH
FR/SE
MD
BI
Percent of Total Heat from Solar
100
80
60
40
20
0
80
100
120
140
160
180
Design Water Temperature, °F


Baseboard Distribution
SM PH FR SE MD BI
PH SM FR MD SE BI
Percent of Total Heat from Solar
100
80
60
40
20
0
80
100
120
140
160
180
Design water Temperature °F

Design Parameters for Air Systems

The Standard Air System. The general remarks concerning the standard liquid system, page 17, on the whole apply to the standard air system as well.

The performance simulations were based on the same locations and weather data, and the same cautions apply in using the data presented in the pages that follow: they are meant to indicate trends, rather than absolute results, and the design parameters may be, in some case, coupled non-linearly.

THE STANDARD AIR 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).

PARAMETER:	NOMINAL VALUE:
Solar Collector	
1. Orientation	Due south
2. Tilt (from horizontal)	Latitude + 10°
3. Number of glazings	1
4. Glass transmissivity (at normal incidence)*	0.86 (6% absorption, 8% reflection)
5. Surface absorptance (solar)	0.98
6. Surface emittance (IR)	0.89
7. Coolant flow rate	2 Standard CFM/ft_c^2
8. Heat transfer effectiveness (HA)**	4 BTU/hr °F ft_c^2
9. Back insulation U-value	0.083 BTU/hr°F ft_c^2
10. Heat capacity	0.5 BTU/hr °F ft_c^2
Collector Duct Work	
11. Heat loss coefficient	0.1 BTU/hr °F ft_c^2
Thermal Storage	
12. Heat capacity	15 BTU/°F ft_c^2
13. Heat loss coefficient	0 BTU/hr °F ft_c^2 (i.e. assuming all heat loss is to heated space)
Heat Distribution System	
14. Air flow rate	2 SCFM/ ft_c^2
Controls	
Building maintained at 68°F ***	
Collectors on when advantageous	

*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).

***It is assumed that internal sources such as people, lights, and equipment will be sufficient to raise the internal temperature by 4°F, i.e. to the comfort standard of 72°F.

The principal difference between liquid and air systems lies in the heat transfer medium and the heat storage medium. Water storage tanks have been used in various heating applications for a considerable time, and as a result their characteristics are now well known. The rock bed, by contrast, is a relatively new heat storage device, and the analysis of its performance characteristics in the pages that follow should be of particular interest to the builder.

The design parameters for the standard liquid and air systems, and the effects of varying them, are essentially the same for both systems except for those listed below.

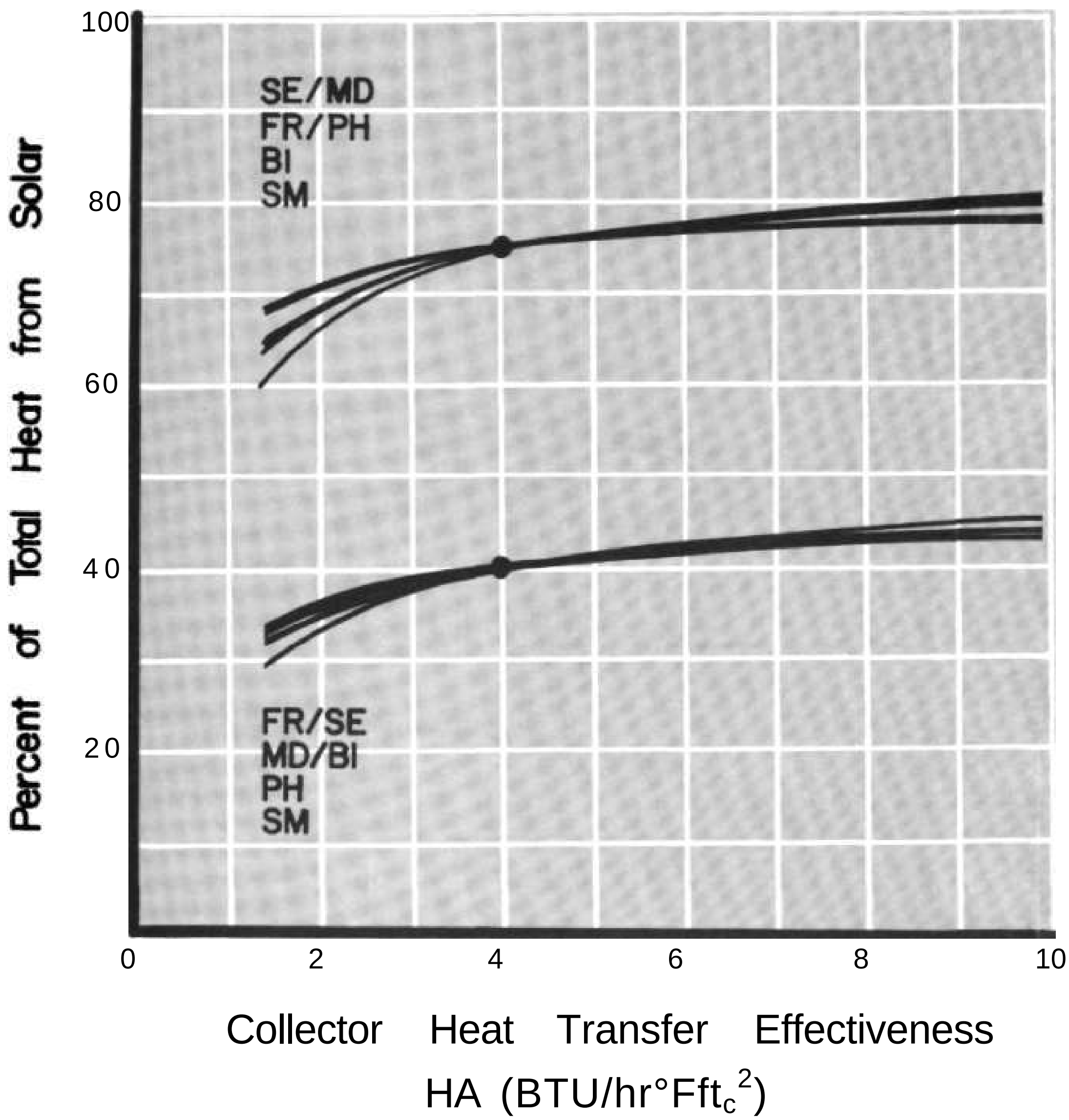
Four parameters are unique to the standard air system. They are discussed in detail beginning on page 47; in brief, the effect of changing them can be summarized as follows:

1. COLLECTOR HEAT TRANSFER EFFECTIVENESS: An important design parameter, it should be approximately 4 BTU/hr/°F/ft_c^2, or greater.
2. COLLECTOR AIR FLOW RATE: A severe performance penalty will result at air flow rates below 1 SCFM/ft_c^2.
3. THERMAL STORAGE HEAT CAPACITY: As in the case of liquid systems, the size of the thermal storage heat capacity is not very important beyond a value of about 10 BTU/°F ft_c^2. This corresponds to 48 lbs of rock/ft_c^2.
4. ROCK BED TEMPERATURE DISTRIBUTION: A rock bed with a short air flow path is preferable to one with a long flow path for pressure drop reasons, but the air flow path should remain longer than around 12 rock diameters in order to take advantage of spatial temperature distribution benefits. Air flow direction should be reversed between the charging and discharging modes of operation.

1. Collector Heat Transfer Effectiveness. Collector heat transfer effectiveness (HA) is the product of the heat transfer coefficient times the heat transfer area divided by the collector area. The effective heat transfer area, A, can be increased by adding fins or by making all sides of the flow passage effective for heat transfer. And the heat transfer coefficient can be raised by increasing the flow velocity and by decreasing the flow channel size.

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

In an air system, collector heat transfer effectiveness is so important that the builder ought to test the heat transfer characteristics of the collector before committing himself to any particular design. Too often systems have been built without adequate collector heat transfer. Even with a heat transfer effectiveness (HA) equal to the nominal value of 4 BTU/hr °F per 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.

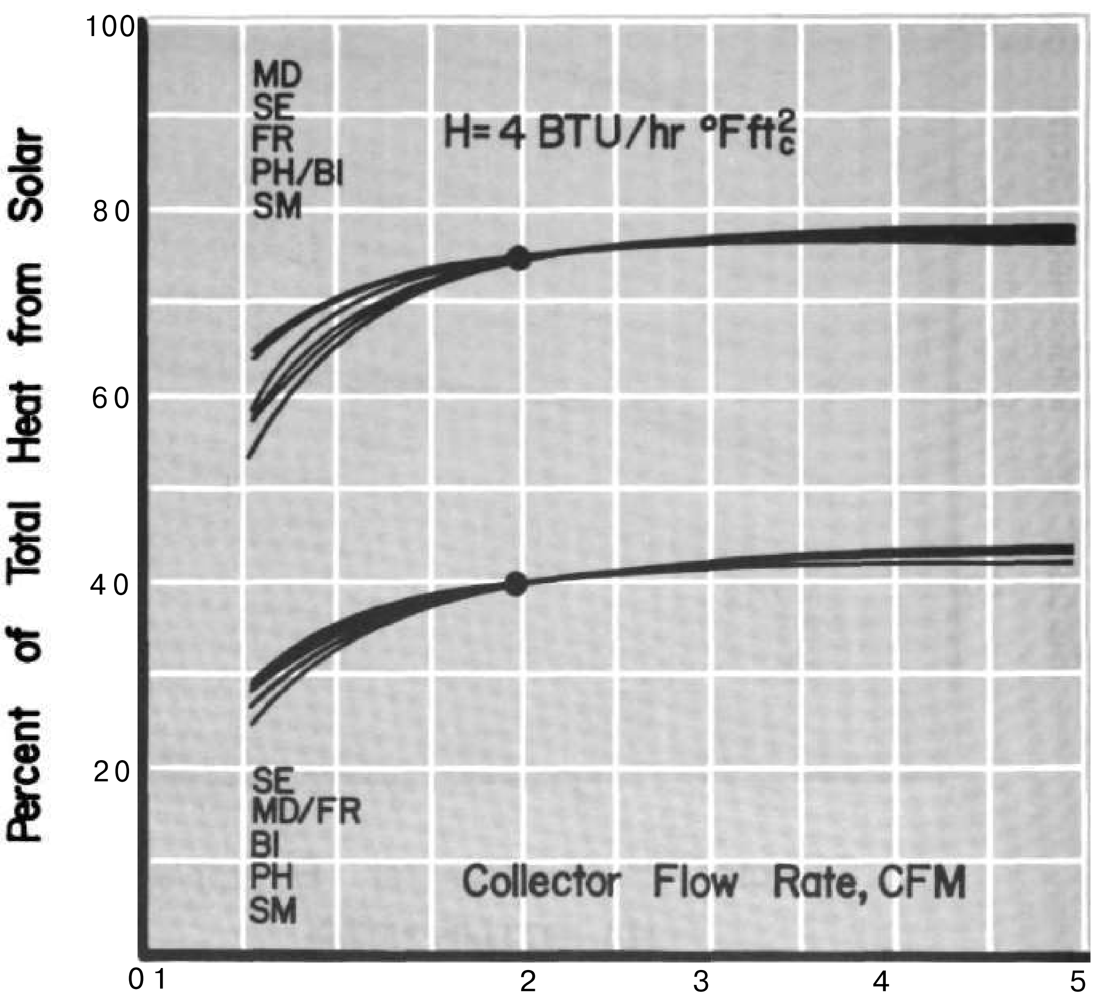


2. Air Flow Rate. As the net air flow rate and heat transfer coefficient are proportional and interdependent, the effect of the air flow rate is considered in two ways in this section. The use of the data presented here will depend on the problem the designer is faced with—whether, for example, he is designing a collector from scratch or is designing a fan and duct system already given a collector design.

Collector Heat Transfer Held Constant: 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 $SCFM/ft_c^2$.

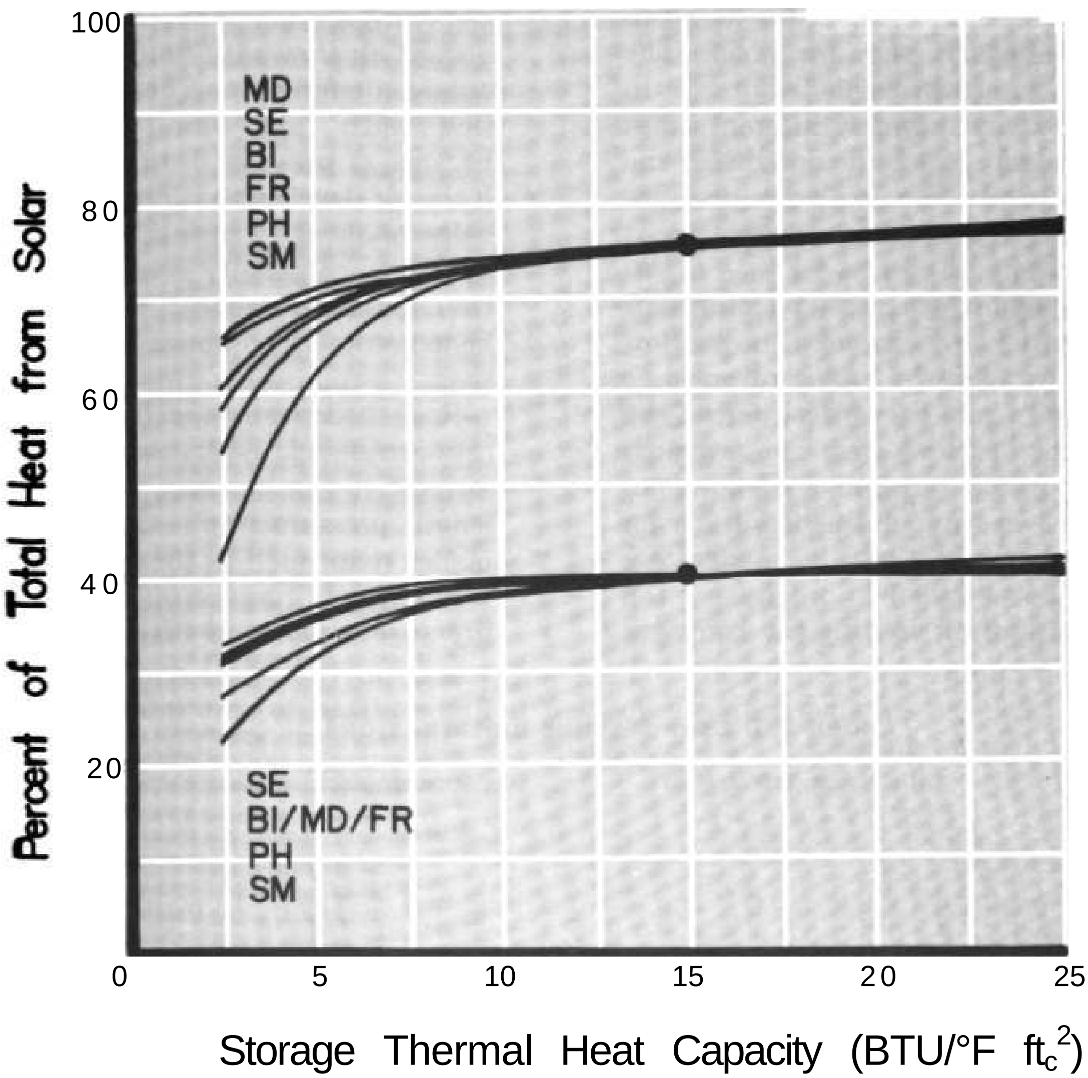
It is to be noted that the air flow rate is varied with the parameter HA held constant at a nominal value of 4. Since H is dependent on flow rate in a collector of fixed geometry, holding HA constant implies changing collector geometry as the air flow rate is changed.

Effect of Air Flow Rate, Collector Geometry Fixed: In a collector of fixed geometry, the heat transfer coefficient H is dependent on flow rate. As air flow rate is changed, HA varies as $(CFM)^{0.8}$ for a fixed geometry. Thus, for example, to ascertain the effect of doubling air flow rate through a given collector, one should determine the effect of doubling the flow rate with heat transfer held constant and then the effect of increasing heat transfer by a factor of $2^{0.8} = 1.7$ with air flow held constant. The net effect of both changes simultaneously is quite close to the product of the two relative improvements.



3. Effect of Storage Thermal Heat Capacity: As in the liquid system, the effect of varying thermal storage capacity is seen to be quite dependent on the site. This is presumably due to the characteristics of the climate pattern during the year which was studied.

The effect of the size of the storage thermal capacity is not very important beyond a value of about 10 BTU/°F ft^2 corresponding to 48 lbs of rock/ft^2. The nominal value of 15 BTU/°F ft^2 corresponds to 71 lbs of rock/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. Thus one should use from 0.50 to 0.75 cu ft of rock per sq ft of collector. This is roughly three times as much volume as is required to achieve the same heat capacity in a water tank.



4. Rock Bed Temperature Distributions. A rock bed is an efficient heat transfer device. Air quickly gives up its heat in flowing through the labyrinthine paths among the rocks. As a result, the rocks near the air entry end of the bed can be at a quite different temperature than those near the exit end.

This time-dependent spatial temperature distribution must be accounted for in the simulation analysis. A simplified illustration is given on the plot below. It is assumed that the bed is cold in the morning. Then as the day progresses the temperature of the air from the collector rises, peaks at noon, and decreases in the afternoon. The resulting temperature distributions in the rock bed are shown at different times during the day. It is to be noted that the exit air from the bed is always cool, indicating that 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 air exits from the hottest part of the bed. (If the air flows were not reversed, then one would have to wait hours to move the heat through the bed and even then the air would only be moderately warm.) As the evening progresses, the temperature of air leaving the bed rises until 8:00 p.m. 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 into 11 axial zones and by calculating the temperature of each. The hot side temperature plot on page 50 is for the air temperature at the bed frontal area corresponding to the right hand side of the rock bed shown in the diagram on page 15.

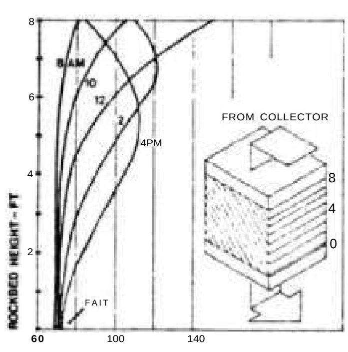
For a fixed bed volume, the length can be varied by varying the frontal area. Generally speaking, a rock bed which has a short air flow path (and a large frontal area) is preferable to a bed with a long air flow path (and small frontal area) because of pressure drop considerations. If the air flow length 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 heat capacity and air flow rate can be related to parameters of real interest to the builder — rock size, bed length, pressure drop — through a performance map, such as the one on page 51.

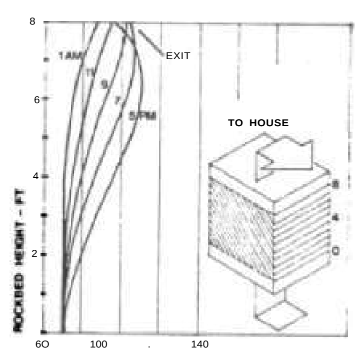
In the performance map, the rock bed pressure drop and the air pumping

ROCK BED TEMPERATURE DISTRIBUTIONS

Collector on, Storing Heat



Collector off, Removing Heat



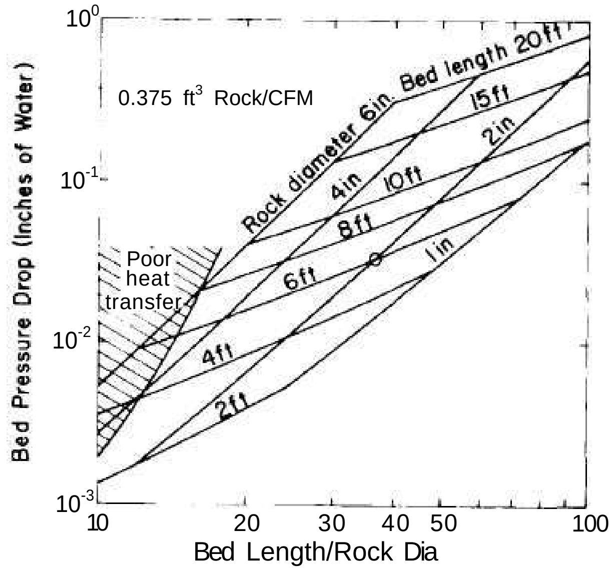
energy loss in the bed are plotted as a function of bed length divided by rock diameter. Lines of constant rock diameter and of constant bed length are drawn on this plot. The nominal parameters of 2 CFM/ft_c^2 and 0.75 ft^3 of $rock/ft_c^2$ or $0.75/2 = 0.375$ ft^3 of rock/CFM were chosen for the air flow rate and the storage volume to plot these isovalue lines.*

For the case of 6 foot long rock bed (in the direction of air flow) and 2 inch rocks, the rock bed length to rock diameter ratio is 36 and 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. This number is quite small, even after factoring in fan and motor inefficiencies.

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. In considering a mix of rock sizes, one should use the smallest diameter in the mix to estimate pressure drop since small rocks will fill the interstices between the larger rocks, thereby increasing the air flow resistance of the bed.

The performance can be used to estimate the pressure drop through any rock bed. For example, suppose that one had a bed of 4 inch diameter rock 15 feet long with a frontal area of 10 ft^2 and an air flow rate of 530 CFM. The air flow is then $530/10 = 53$ CFM/ft^2 of frontal area.

ROCK BED PERFORMANCE MAP



To obtain this same face velocity for the conditions of the performance map, one would need a bed length of (53 CFM/ft^2) x (0.75 ft^3/2 CFM) = 20 ft. Reading from the map, the pressure drop is 0.47 in. of water or 0.0235 in. of water/ft of length. Thus the pressure drop in 15 feet would be 0.35 in. of water.

The pressure drop in the above example is somewhat excessive. By merely changing the bed length to 10 ft and the frontal area to 15 ft^2, the pressure drop could be reduced to 0.13 in. of water. This illustrates the dramatic effect of even minor changes in rock bed configuration on pressure drop.

*The heat transfer coefficient and pressure drop correlations for rock beds were taken from the following reference: R. V. Dankle 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.

Collector Size, Active Systems

One of the main purposes of the Handbook is to enable the builder to determine the size of the collector array for an active space heating system anywhere in the Pacific Region.

There are several factors that will go into this calculation. Design is one, and has been analyzed in various ways in the previous section. It need only be noted at this point that the charts and calculations on the accompanying pages assume the design to be the standard liquid or air space heating system, and that in most areas of the region a standard system will be used to provide from 25% to 75% of the total space heating requirement. It is considered that 100% solar heating, though feasible in some locations, will not be cost effective for most. The choice, then, of 25%, 50%, and 75% solar heating fractions on the charts and graphs is intended to cover the range most likely to be useful to the designer or builder.

Thermal Load. The next important factor is building thermal load, which must be known in order to use any of the accompanying charts or calculations. 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 square foot of floor area.

For an existing building, the exact 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.

For a proposed building, the thermal load is calculated by 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 this to the infiltration load as determined by multiplying the building volume times the number of air changes per hour, times the heat capacity of air (0.018 BTU/°F ft^3).* This will then yield the building load in units of BTU/°F hr. Multiply this number by 24 to obtain the load in BTU/degree-day, which is the number needed in the subsequent sections.

Collector Sizes For Selected Cities. The location of a solar heating system, in terms of solar radiation and ambient temperature, is the last and most important factor.

Using the techniques of simulation analysis, LASL has calculated the performance of the standard systems for those cities where sufficient weather and solar data have been available.

The results enable one to determine collector (and thus storage) sizes for the cities analyzed. They are presented in terms of the ratio of building load to collector area on page 53.

The chart on page 53 presents the ratio in terms of the solar heating percentage of the total space heating requirement. In other words, for a solar heating system designed to provide 75% of the space heating requirement in Fresno, the ratio is 34 BTU/degree day ft_c^2.

To use the building load/collector area ratio to determine the size of the collector for one of the cities listed (or an equivalent location), one must determine (1) the building thermal load and (2) decide what percentage of solar heating the system is to provide.

For example, for a 1,500 square foot house in Fresno, California, with a thermal load of 10 BTU/degree-day/ft^2 of house, the building thermal load is 15,000 BTU/degree-day.

*See the ASHRAE Handbooks for detailed treatment of building load calculations.

Suppose, next, that one decides on a 75% solar heating fraction with the balance of the heating requirement to be made up by conventional means. For this fraction, the chart below for Fresno gives a ratio of 34 BTU/degree-day/ft^2 of collector. Therefore, the required collector size for this particular case is:

$$\frac{15000}{34} = 441 \text{ square feet.}$$

In a similar manner, one can determine the size of a collector for a standard liquid or air system for any of the cities listed on the table, for 25%, 50%, and 75% solar heating; and for collector sizes for other percentages, one can use the graphs beginning on page 54 to determine the correct ratio for the cities plotted on it, and with which one can then establish the actual collector size.

The table below lists load-collector ratios for both liquid and air systems. There is no obvious systematic climatic trend observable in the relative performance of the two systems. Both seem to have comparable performance characteristics at high values of the solar heating fraction, but the liquid system does perform somewhat better than the air system at lower values of the solar heating fraction.

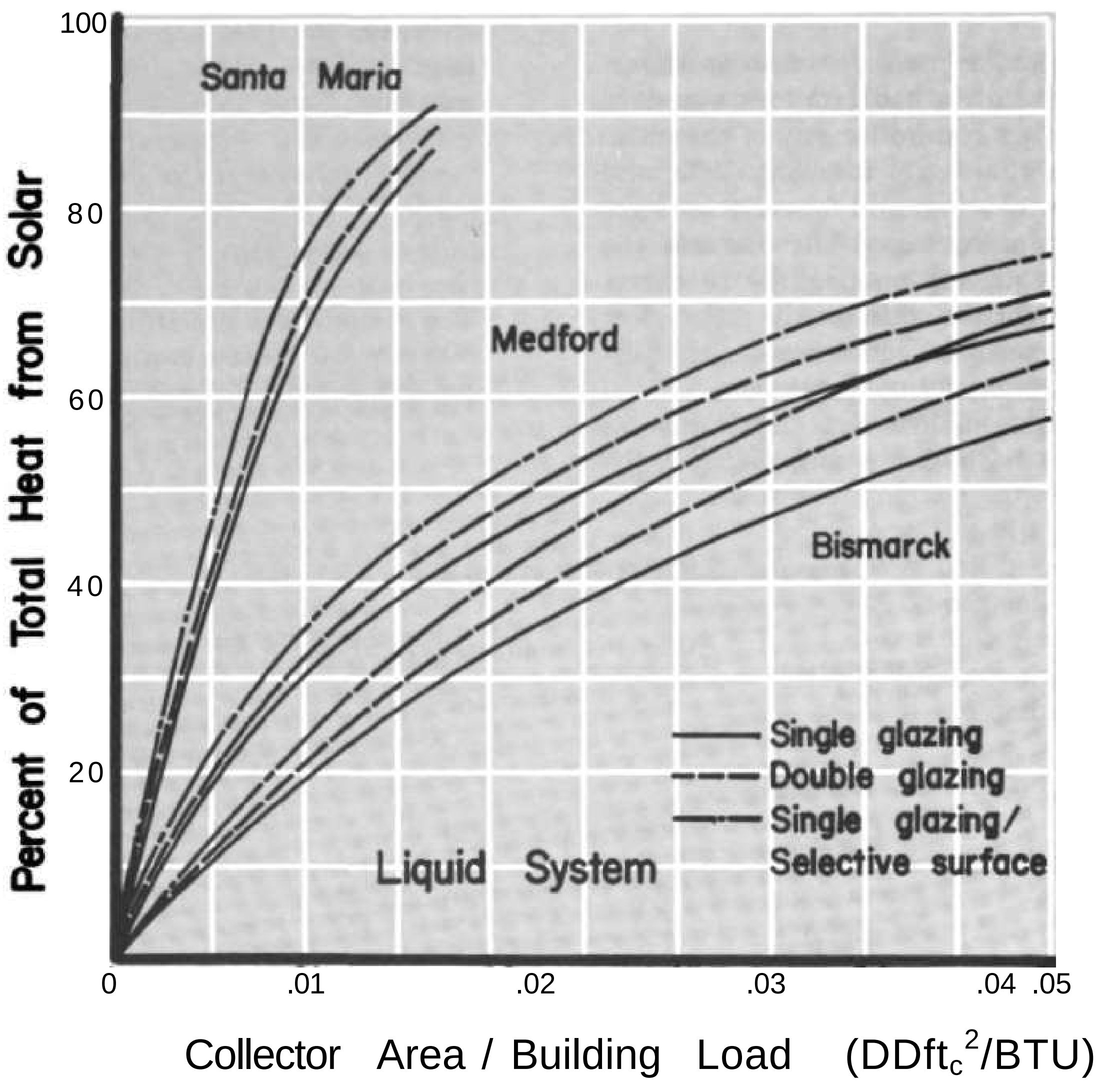
The variation of collector size with the solar heating fraction is shown for the six cities on the four graphs beginning on page 54. Two of the graphs are for the standard liquid system and two are for the standard air system. For convenience, the curves for both double glazing and for a selective surface* are also shown. These curves are plotted against the ratio of collector area to building load, which is the reciprocal of the load-collector ratio used earlier.

*Absorptance = 0.95, emittance = 0.10.

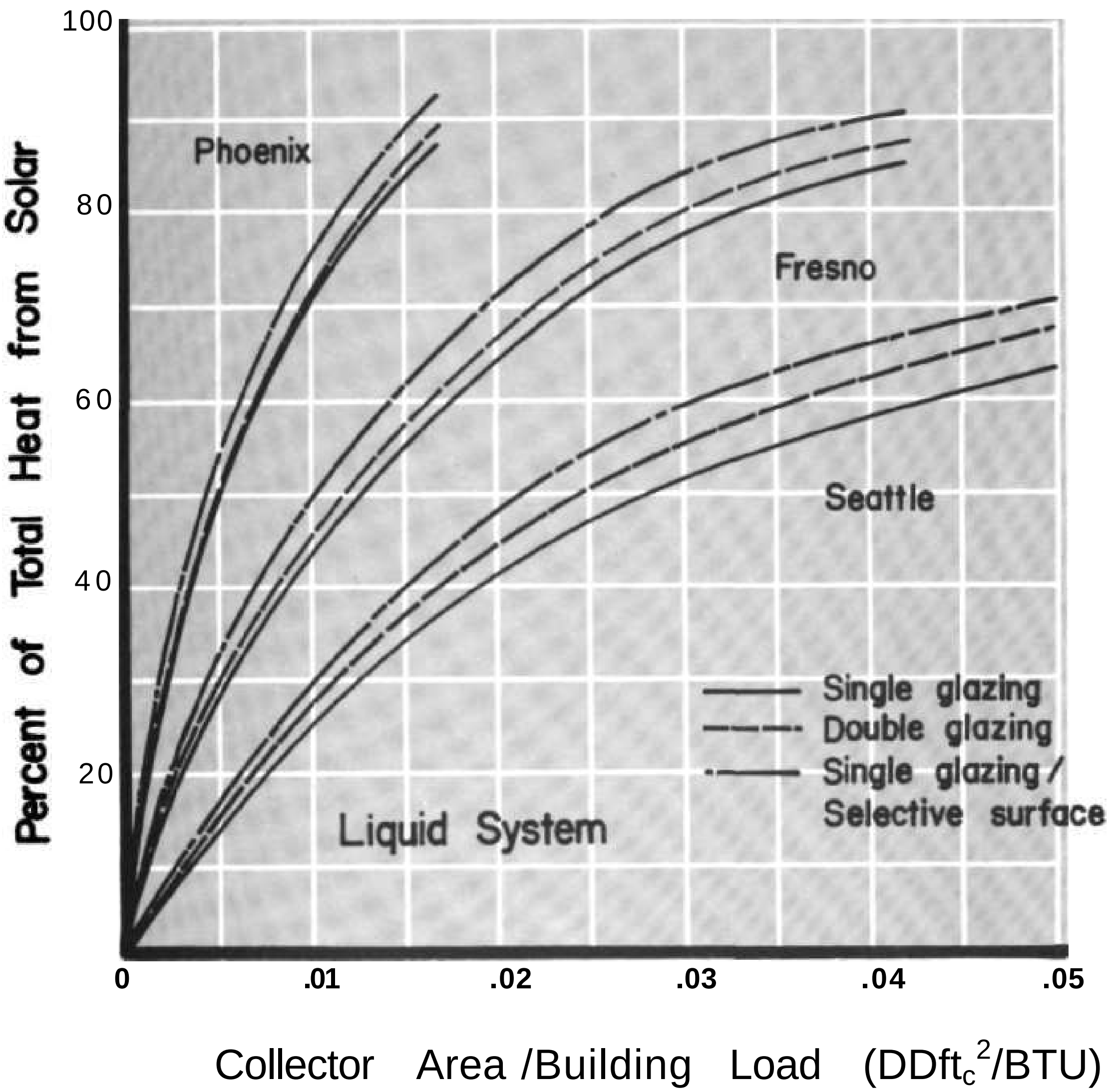
LOAD-COLLECTOR RATIOS FOR SIX CITIES

Solar Heating Fraction	Standard Liquid System			Standard Air System		
	25%	50%	**75%**	25%	50%	75%
Site						
Phoenix, AZ	432	172	84	369	154	79
Santa Maria, CA	300	144	84	280	133	76
Fresno, CA	206	77	34	185	72	34
Medford, OR	130	41	14	117	39	15
Seattle, WA	104	33	11	88	29	11
Bismarck, ND	66	24	11	62	24	11

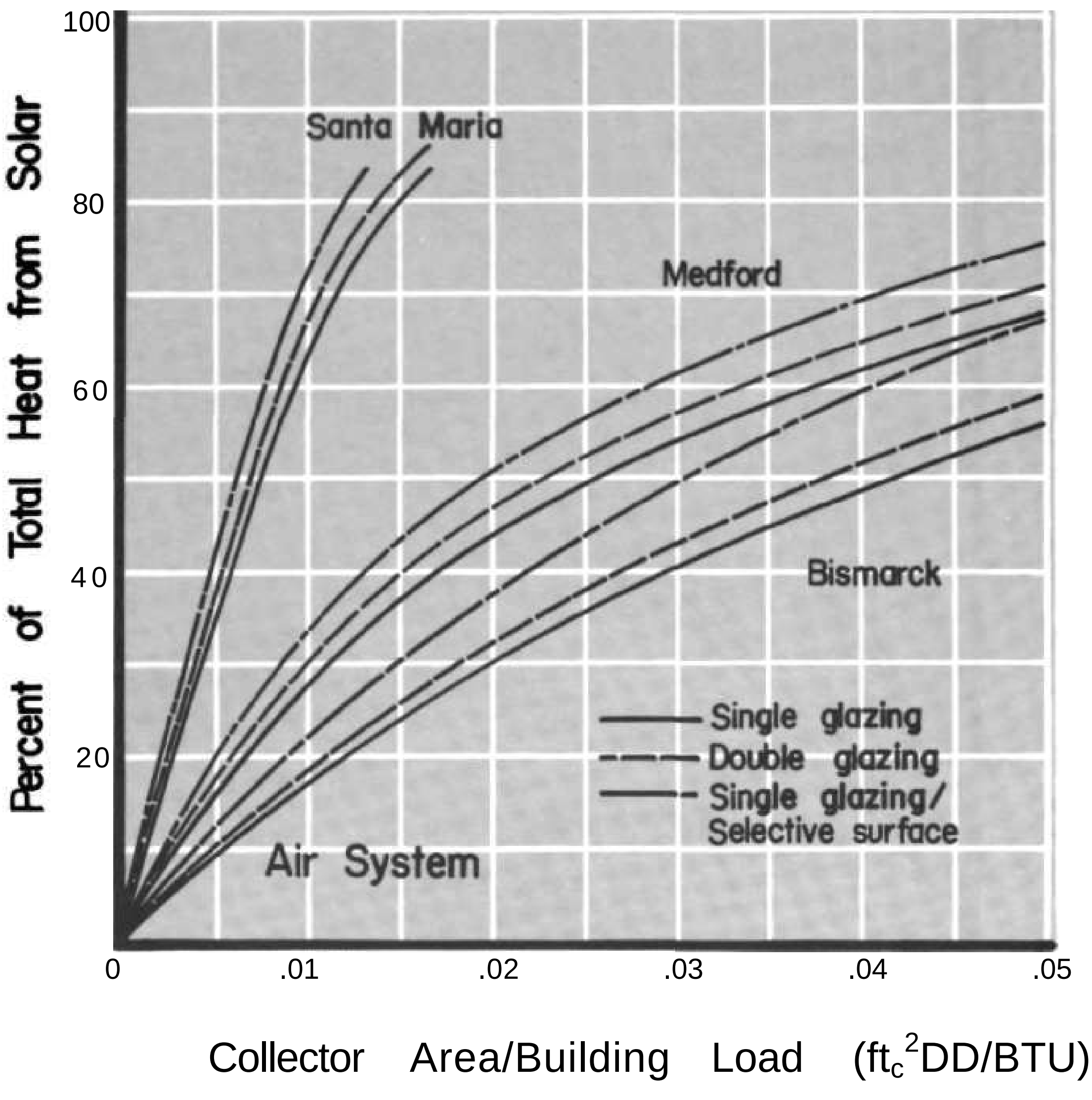
NOTE: These values have been determined for a specific one-year period (the "solar heating design year") based on hour-by-hour computer simulation analysis.

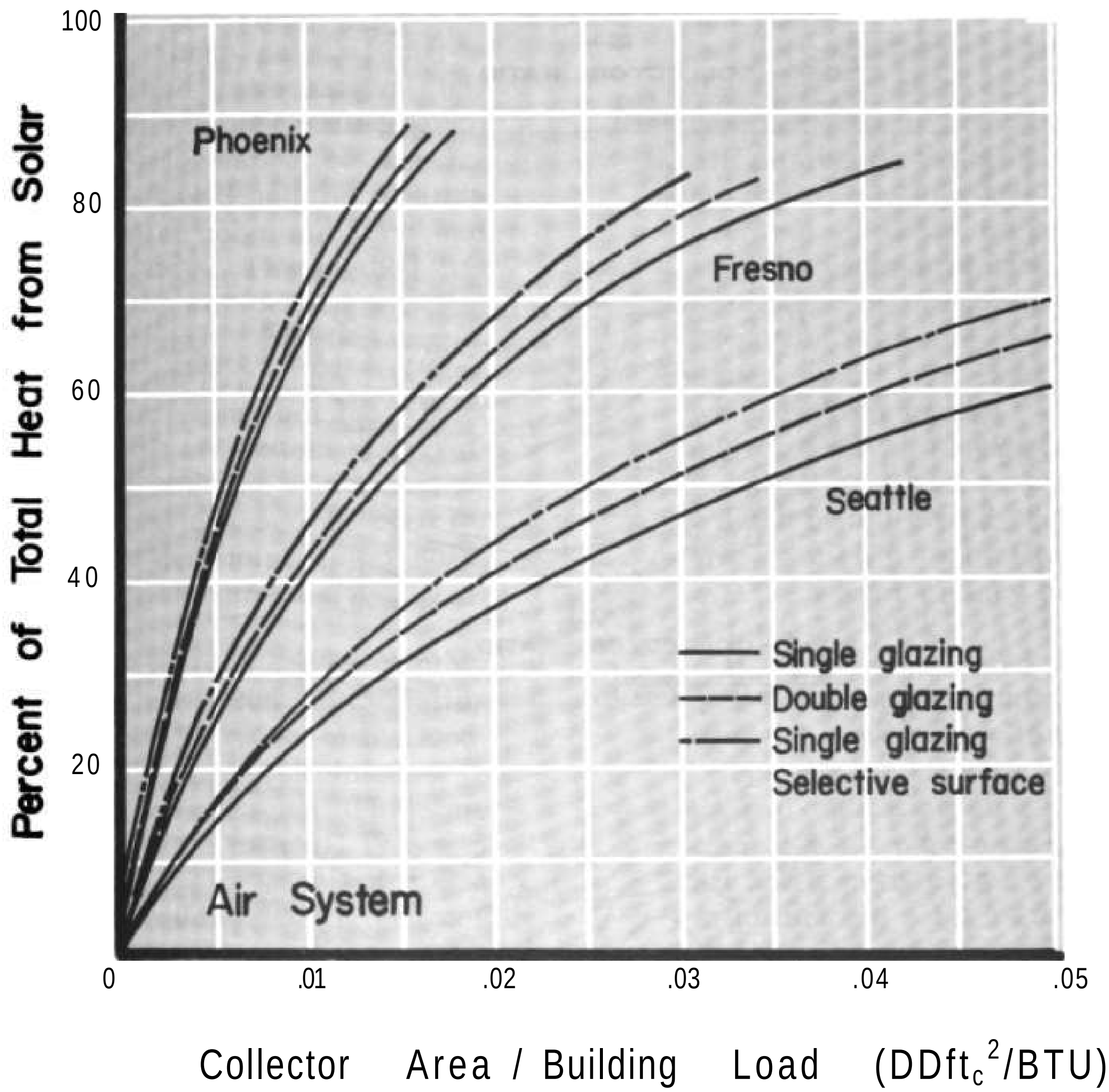


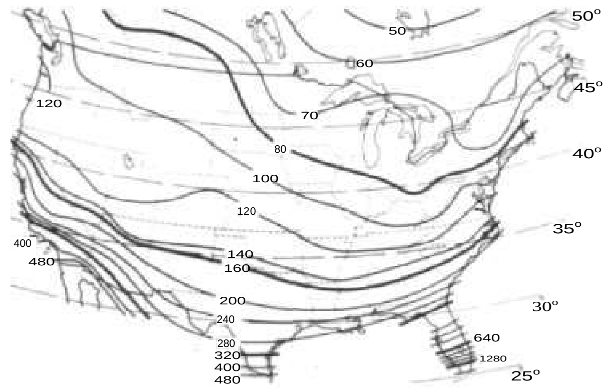
100
80
60
40
20
0
.01
.02
.03
.04 .05
Percent of Total Heat from Solar
Collector Area / Building Load $(DDft_c^2/BTU)$
Santa Maria
Medford
Bismarck
Liquid System
Single glazing
Double glazing
Single glazing/
Selective surface



Percent of Total Heat from Solar
100
80
60
40
20
Phoenix
Fresno
Seattle
Single glazing
Double glazing
Single glazing / Selective surface
Liquid System
0
.01
.02
.03
.04
.05
Collector Area /Building Load (DDft$_c^2$/BTU)


Percent of Total Heat from Solar
100
80
60
40
20
Santa Maria
Medford
Bismarck
Single glazing
Double glazing
Single glazing/
Selective surface
Air System
0
.01
.02
.03
.04
.05
Collector Area/Building Load (ft_c^2DD/BTU)

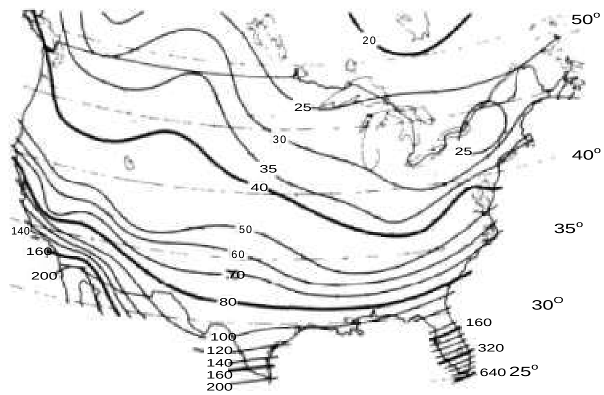

Percent of Total Heat from Solar
100
80
60
40
20
0
.01
.02
.03
.04
.05
Collector Area / Building Load (DDft$_c^2$/BTU)
Phoenix
Fresno
Seattle
Single glazing
Double glazing
Single glazing
Selective surface
Air System



LOAD - COLLECTOR RATIO

$\frac{\text{building load (btu/dd)}}{\text{collector area (sq.ft.)}}$

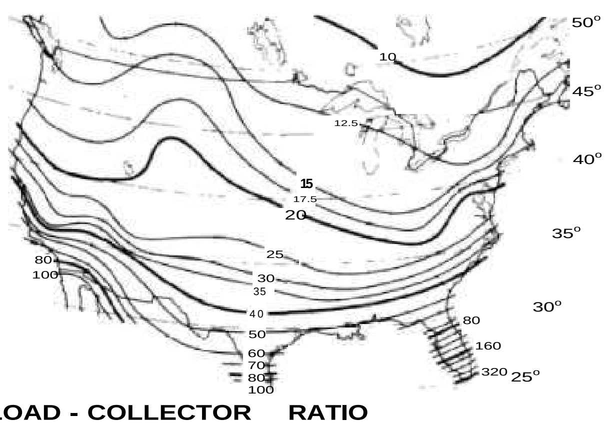
25 % solar fraction



LOAD - COLLECTOR RATIO

$\frac{\text{building load (btu/dd)}}{\text{collector area (sq. ft.)}}$

50% solar fraction



LOAD - COLLECTOR RATIO

$\frac{\text{building load (btu/dd)}}{\text{collector area (sq. ft.)}}$

75 % solar fraction

The Monthly Solar-Load Ratio Method for Determining Collector Size in Other Locations

Usually the only reliable basis for predicting the yearly performance of a solar heating system is an hour-by-hour computer simulation analysis of the entire system. Such analyses have been performed by LASL for a total of 25 U.S. and Canadian cities for up to 12 years of weather data for each city.

Based upon these calculations, a monthly solar-load ratio method has been devised. It enables a designer to predict the performance of a solar heating system, based only on monthly data of horizontal solar radiation and heating degree days. The method has been validated by comparison with the hour-by-hour computer simulations.

Initially, the standard liquid system was chosen for analysis, but the results for the standard air system were sufficiently similar that the same monthly solar-load method can now be used for either system to determine collector size by applying a small correction for the air system.

Based on this method and on monthly data of solar radiation and heating degree-days, specific values of the load-collector ratio have been calculated for 18 cities in the Pacific Region. These are given in the table on page 61. They can be used to estimate solar collector area for a standard liquid system in the same manner as the numbers in the table on page 53. For a standard air system, it is recommended that the values in the table should be multiplied by the following factor:

Solar Heating Fraction	Factor
75%	0.98
50%	0.93
25%	0.89

It should be noted that the values for load-collector ratios in the tables on pages 61 are usually different from those on page 53. Two factors account for this:

1. The numbers in the table on page 61 are based on the approximate monthly solar-load ratio method and therefore should be somewhat less accurate than the numbers in the table on page 53, which are based on hour-by-hour simulations.
2. The numbers in the table on page 61 are based on long-term monthly averages of solar radiation and heating degree-days, whereas those of the table on page 53 are based on a specific year, the "solar heating design year," an explanation of which is given on page 18. Therefore any systematic correlation of solar radiation and heating degree-days will not be reflected in the results of the monthly solar-load ratio method except as they occur on a nationwide basis.

It is recommended that the load-collector ratios on page 53 be used whenever there is a choice.

The maps on page 58 give a general indication of the distribution of load-collector ratios. They are based on the monthly solar-load ratio method, using data from the 18 cities listed in the table on page 61, plus an additional 67 cities throughout the United States.

The monthly solar-load method is a five-step process which can be performed on a hand calculator with the aid of trigonometric and exponential tables—or with a calculator that 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. The figures given in the tables in the Appendix are long-term averages obtained from the ASHRAE Guide (see pp. (96), for 18 cities within the region.

b) Total solar radiation on a horizontal surface, per month. The data given in the tables in the Appendix were obtained from records of the National Weather Service for total daily radiation. The averages are based on 200 to 700 days of measurement during the months listed.*

Step 2) Correct the solar radiation to a collector for tilt angle of latitude-plus-10° using the approximate formula following:

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

where

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

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

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

The data in the table in the Appendix were calculated from the monthly average solar radiation data, as measured on a horizontal surface in the 18 cities, by means of the formula above.

The values are the monthly solar radiation on a plane surface tilted at an angle equal to the latitude plus 10° and oriented due south. They should be used in reference, however, to the remarks on collector tilt angle on page 22.

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 1°F difference between the inside and outside temperatures.

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

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

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:

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

where

$$X = 1.06 - 1.366\,e^{-.55\,SLR} + .306\,e^{-.105\,SLR},$$
(for SLR < 5.66)

X = 1, (for SLR > 5.66)

*It is to be noted that the figures in these two tables do not correspond exactly to those of the solar heating design years for the six cities used in the parametric liquid and air system studies, since these were measured values for individual months rather than averages.

LOAD-COLLECTOR RATIO FOR 18 CITIES

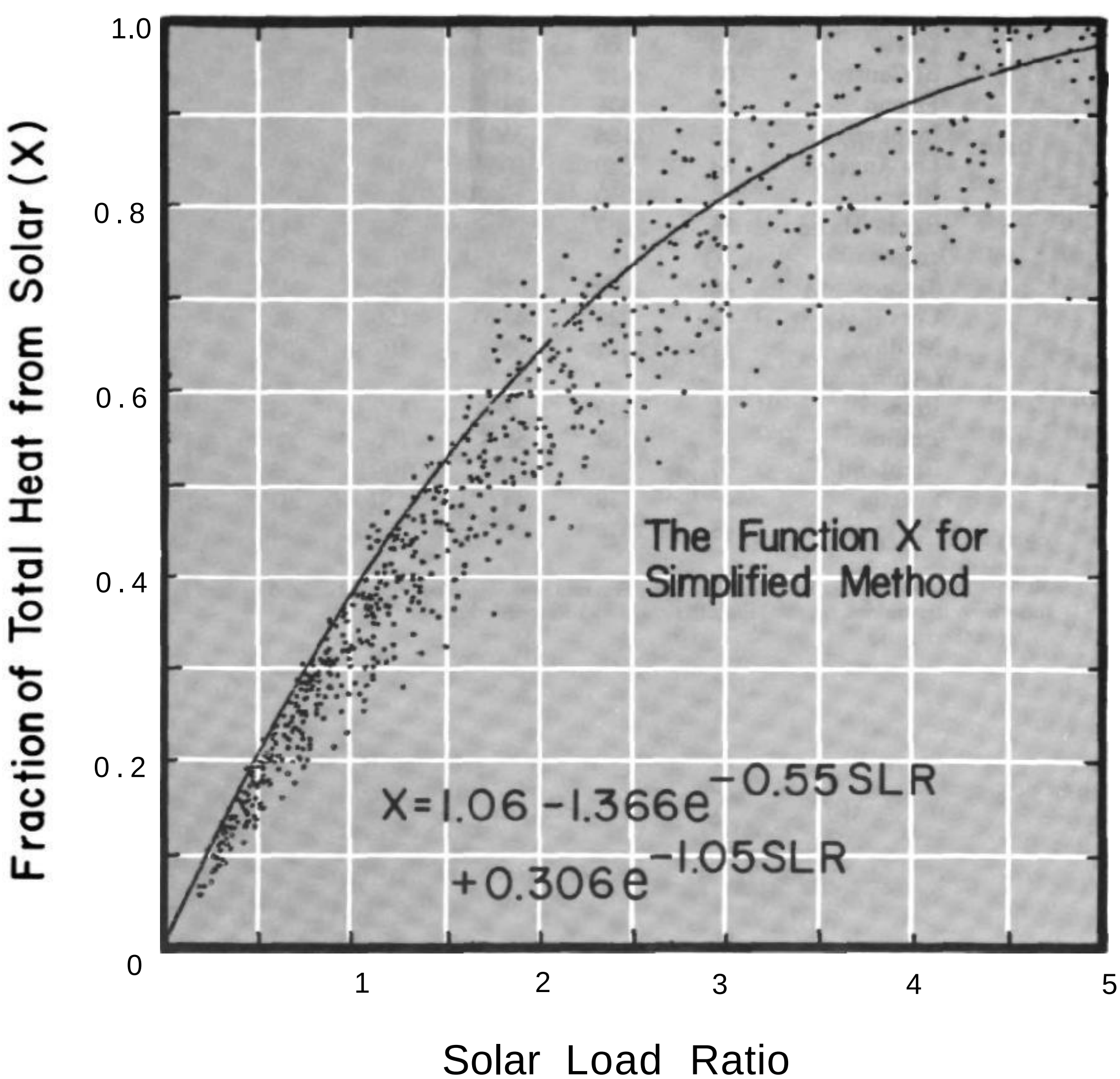
Site	Latitude °N	Elevation ft	Degree Days	Solar Heating Fraction 25%	Solar Heating Fraction 50%	Solar Heating Fraction 75%
Arizona						
Page	37	4280	6632	128	48	23
Phoenix	33	1139	1765	300	118	59
Tucson	32	2440	1800	301	118	59
California						
Davis	39	50	2502	198	72	33
El Centra	33	12	1458	547	206	97
Fresno	37	336	2492	195	70	32
Inyokern	36	2186	3528	232	88	42
Los Angeles	34	540	2061	416	157	75
Riverside	34	1050	1803	391	152	74
Santa Maria	35	289	2967	353	142	67
Oregon						
Astoria	46	22	5186	127	45	19
Corvallis	45	236	4726	120	42	18
Medford	42	1321	5008	107	38	16
Washington						
Prosser	46	840	4805	117	41	18
Pullman	47	2583	5542	100	36	16
Richland	47	731	5941	100	35	15
Seattle	48	110	4424	110	37	15
Spokane	48	2356	6655	90	31	14

NOTE: These values are based on long-term averages of monthly solar radiation and heating degree days and have been determined by the Monthly Solar-Load Ratio Method.

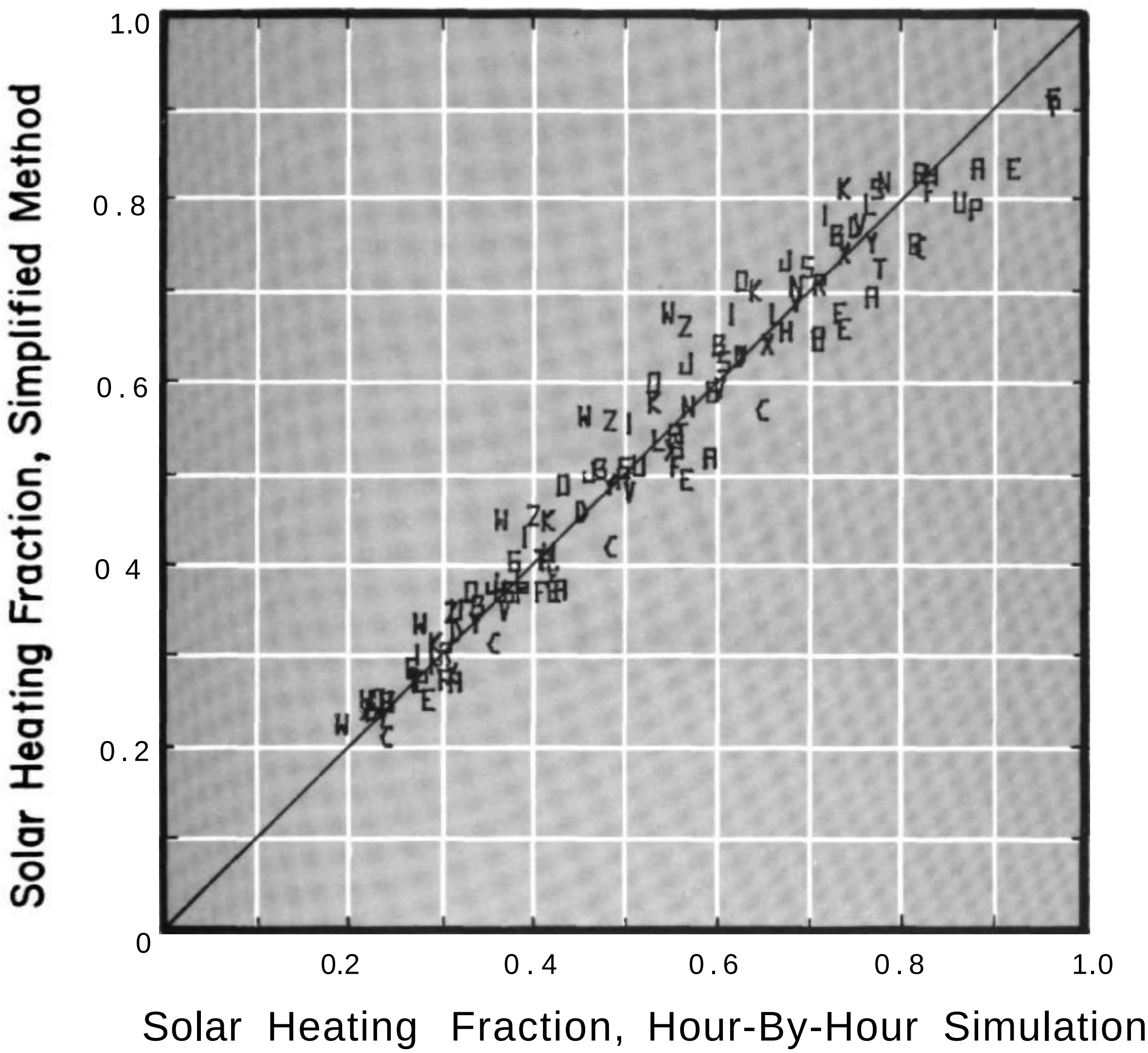
The function X is shown plotted below. Note that X is not a very good estimate of the monthly solar heating fraction because it is designed to compensate (on an annual basis) for the fact that the heating load based on degree-days per month is lower than the heating load based on hourly calculations.

Test of the Monthly Solar Load Ratio 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 method is essentially zero and the root-mean-square error (standard deviation of the prediction error) is 4.4% 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.

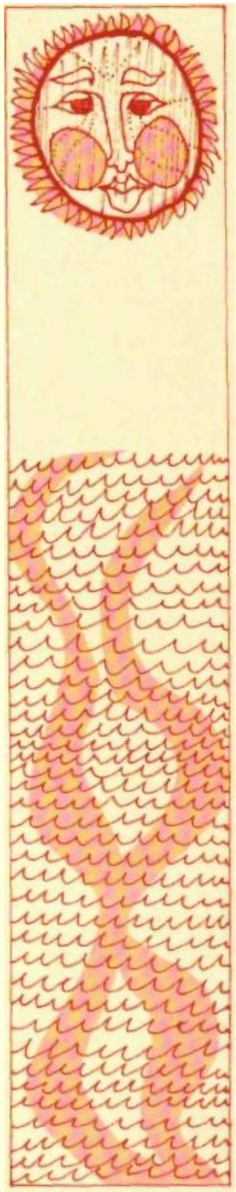


TEST OF SIMPLIFIED METHOD



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, British Columbia
I	Madison, WS	W	Winnipeg, Manitoba
J	Frederickton, N.B.	X	Medford, OR
K	Bismark, ND	Y	New York City, NY
L	Lincoln, NE	Z	Edmonton, Alberta

Domestic Hot Water Heating



VI. Domestic Hot Water Heating

Simple thermosiphon and tank-type solar hot water heaters are common in many parts of the world where freezing is not a problem. There is no separate heat transfer fluid in these passive rooftop units, and hot water is used directly in the household from them. The simplest tank-type design is a large water-filled plastic "pillow"; in more complex and durable types, water circulates through pipes of steel, plastic, or glass four to six inches in diameter.

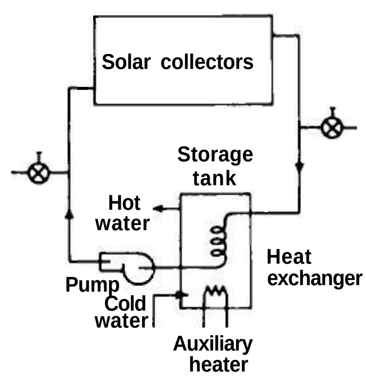
A flat plate collector can also heat water to be used directly (via a storage tank) in the household. But in colder climates, unless the system can be drained, it must use a non-freezing heat transfer fluid with a liquid-to-water heat exchanger in the storage tank. The easy adaptability of this type of system to an active liquid space heating system is obvious, but care must be taken to avoid contamination of the domestic water supply by a toxic collector fluid.

The sole objective of this section of the handbook is to provide design information appropriate to sizing a domestic solar hot water heater of the conventional active type, that is, a system consisting of a flat-plate collector with a pumped liquid coolant which delivers heat to a storage tank by means of a heat exchanger. This is by no means intended to imply that this is the best approach. It is simply the one that has been analyzed in some detail. Tank-type, thermosiphon, or air heating collectors present different options which may be more suitable but for which comparable design studies have not yet been completed.

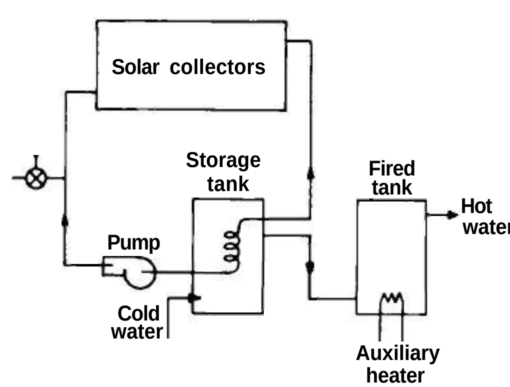
There are, however, two ways in which a flat plate domestic hot water system can be used: with one tank or with two. Assuming that the system will have a conventional back-up element, a one tank system consists of a storage tank heated (1) by the solar collector fluid passing through the heat exchanger in the tank when the sun is shining, and (2) by an auxiliary gas burner or electric resistance coil for times when it is necessary to maintain the water temperature to a thermostatically determined level (see diagram below).

In a two-tank system, these functions are divided into two tanks in such a way that the solar heater acts as a pre-heater for the conventional gas or electric unit. This system can be expected to perform at a higher overall efficiency than the one-tank system. Since the auxiliary heat is supplied to the second tank, not to the solar-heated tank, the solar-heated portion of the whole system can operate at lower storage (and hence collector) temperatures and thus at a higher heat collection efficiency.

DOMESTIC HOT WATER SCHEMATIC



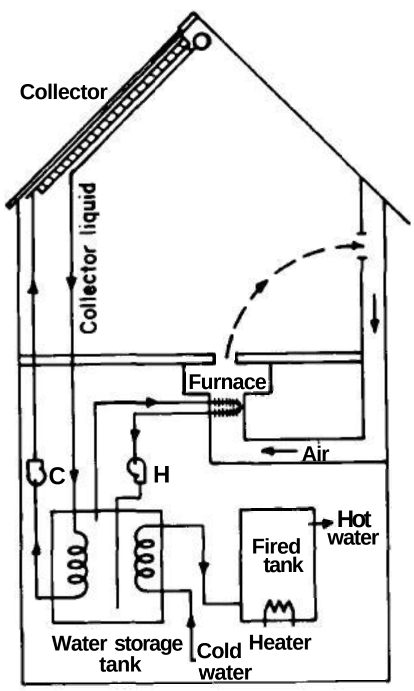
ONE-TANK SYSTEM

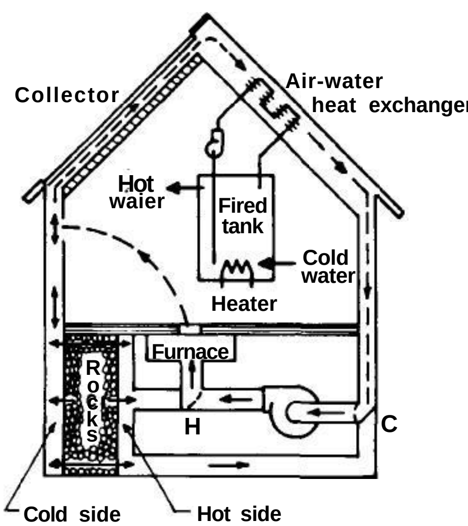


TWO-TANK SYSTEM

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 65, connected to a liquid-heating collector system.

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 65, connected to an air-heating collector system.





Domestic Hot Water Design Parameters.

The nominal design parameters for the collector are the same as those given for the standard liquid space heating system on page 17. 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/hr °F ft_c^2.

In addition, the thermal load is quite different for water heating than for space heating. The profile of hot water demand shown below was deduced from personal experience, and the simulations were run for this profile. It is assumed to be the same for every day of the year.

In the normal 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.

In the nominal two-tank system, the solar-heated storage tank acts as a source of preheated water for the second tank, a conventionally fired hot water tank. A control scheme was adopted in which auxiliary heat is added to the second tank to maintain the temperature at 120°F. The nominal thermal storage heat capacity for the solar storage tank is the same as for the one-tank system. 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 a day, the second storage tank would be 40 gallons.

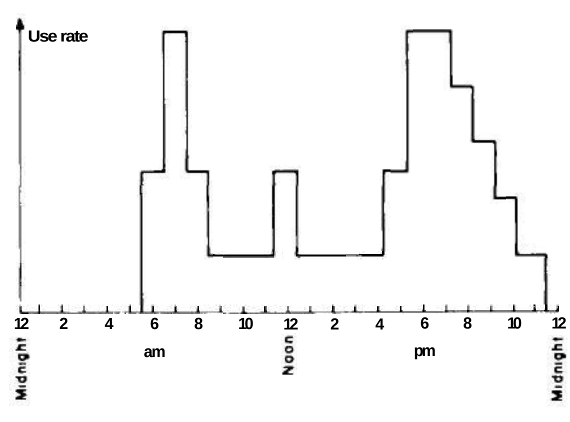
The nominal domestic hot water system, as in the case of the standard liquid space heating system, provides 75% of the required heat from solar energy. The loads which give a 75% solar fraction are as follows:

Location	One-Tank Load for 75% Solar (Gal/day-ft_c^2)	Two-Tank Load for 75% Solar (Gal/day-ft_c^2)
Phoenix	1.85	2.45
Santa Maria	1.41	2.01
Fresno	1.02	1.54
Medford	0.59	1.05
Seattle	0.165	0.52
Bismarck	0.63	1.085

Four parameters were studied for domestic hot water systems: collector area, collector tilt, water storage mass, and hot water temperature. The effects of varying them are presented in detail in the sections below, followed by a discussion of the method for determining collector sizes for hot water systems through the region.

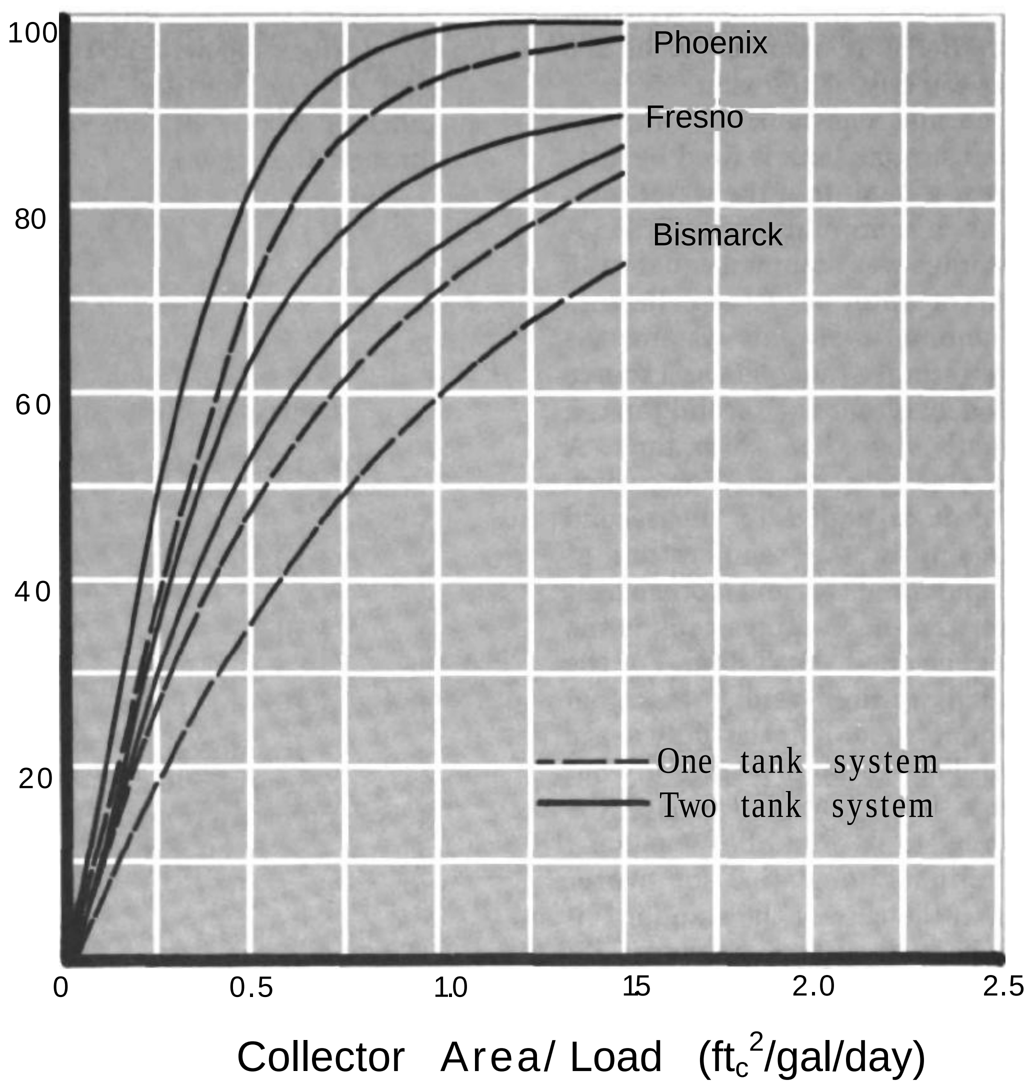
ASSUMED USE PROFILE

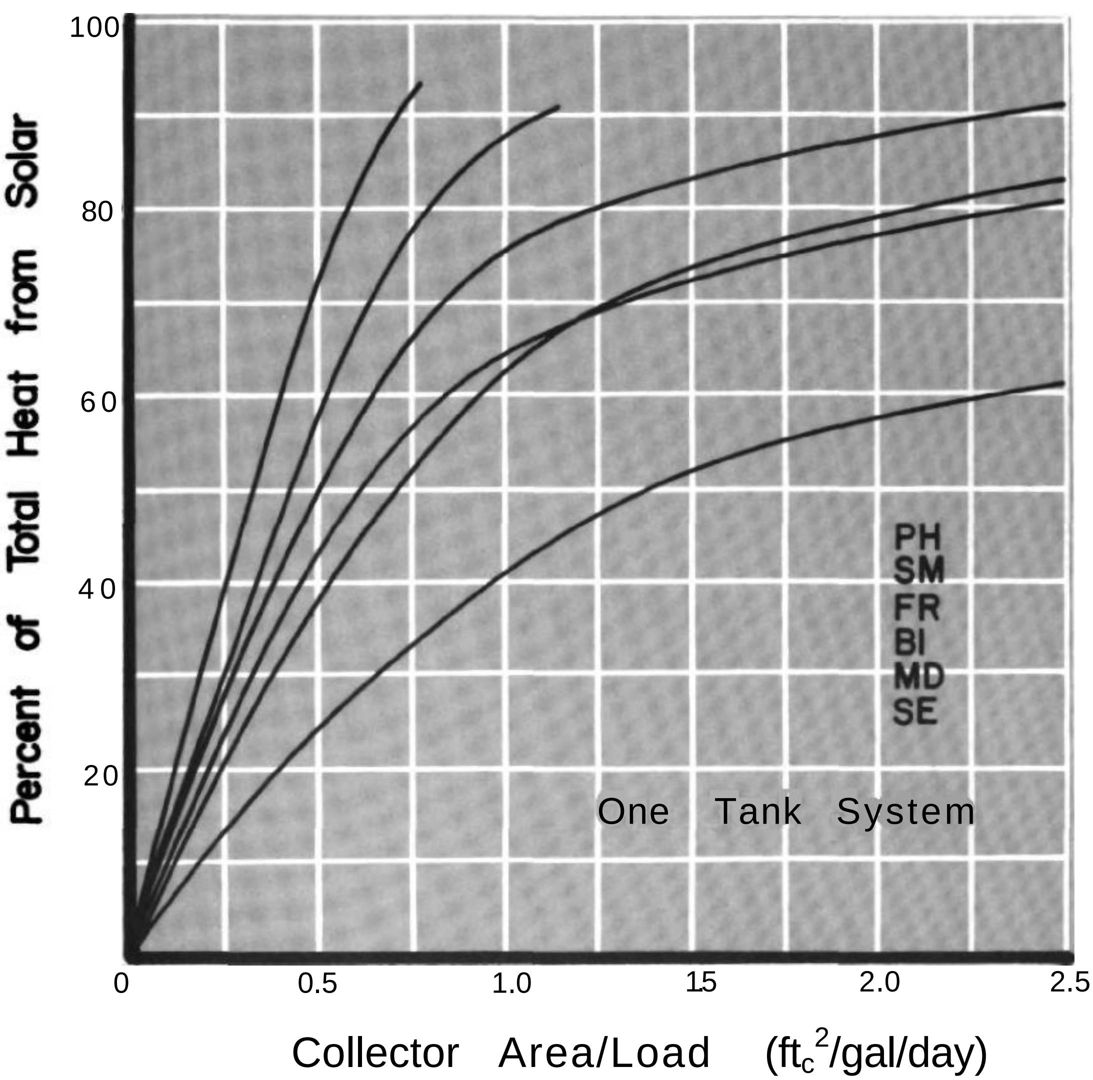
FOR DOMESTIC HOT WATER



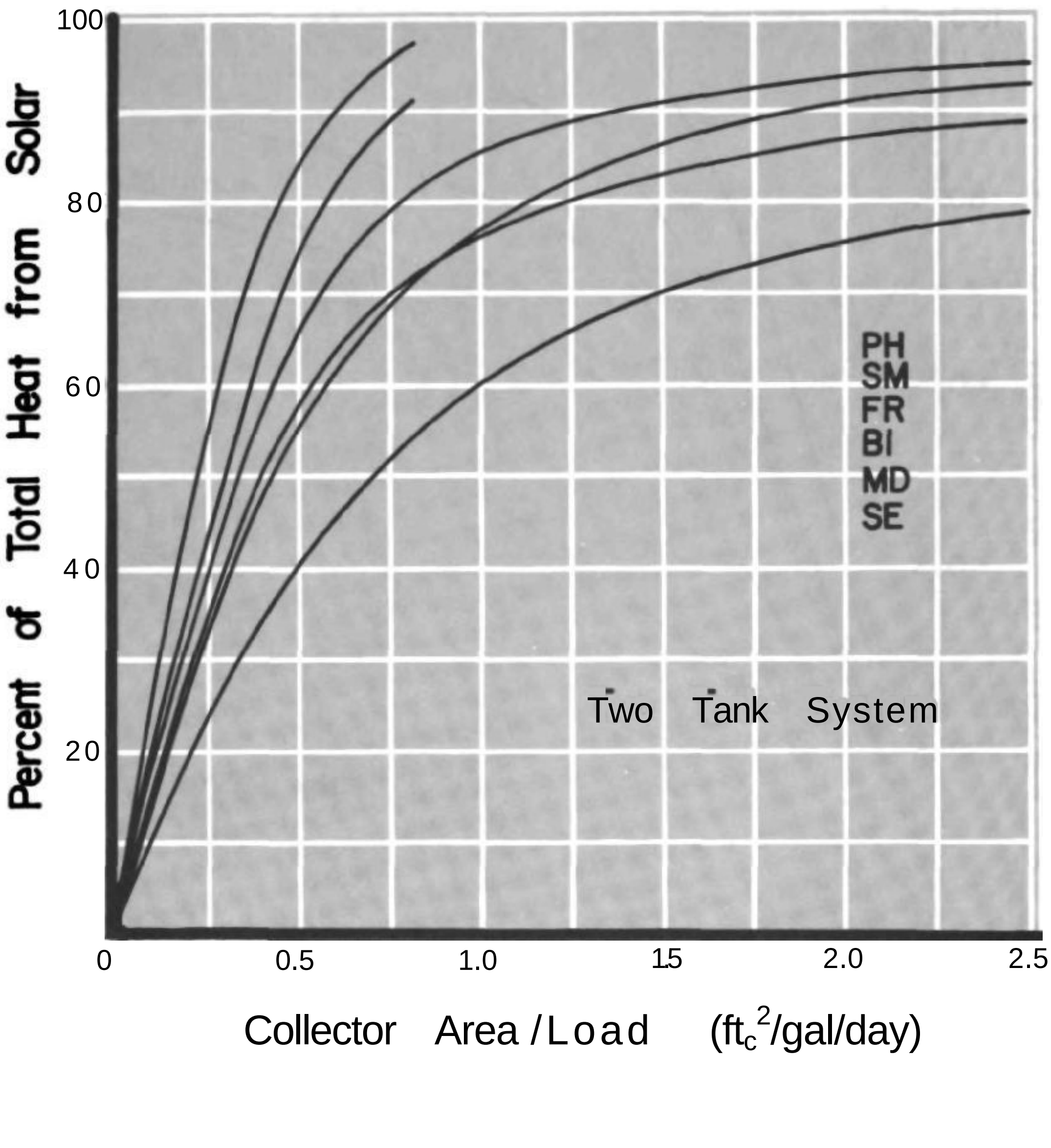
1. Collector Area. The graph below shows the effect of changing the collector area for both one and two-tank domestic hot water systems. The plotting parameter is collector area divided by the heating load in gallons per day. Three of the six cities are shown. Phoenix and Bismarck represent the extremes of performance, while Fresno represents a relatively average case. Individual collector area/load curves are shown for all six cities on the graphs on pp. 69-70, the first graph for a one-tank system and the second for a two-tank system.

It can be seen that the two-tank system outperforms the one-tank system. As was pointed out above, this is the result of the lower operating temperatures of the solar-heated portion of the two-tank system. The increase in total heat obtained from a two-tank system over a one-tank system ranges from approximately 30% for Phoenix to 55% for Bismarck in the hot water load range of 2.5 to 6.0 gallons/day/ft_c^2.





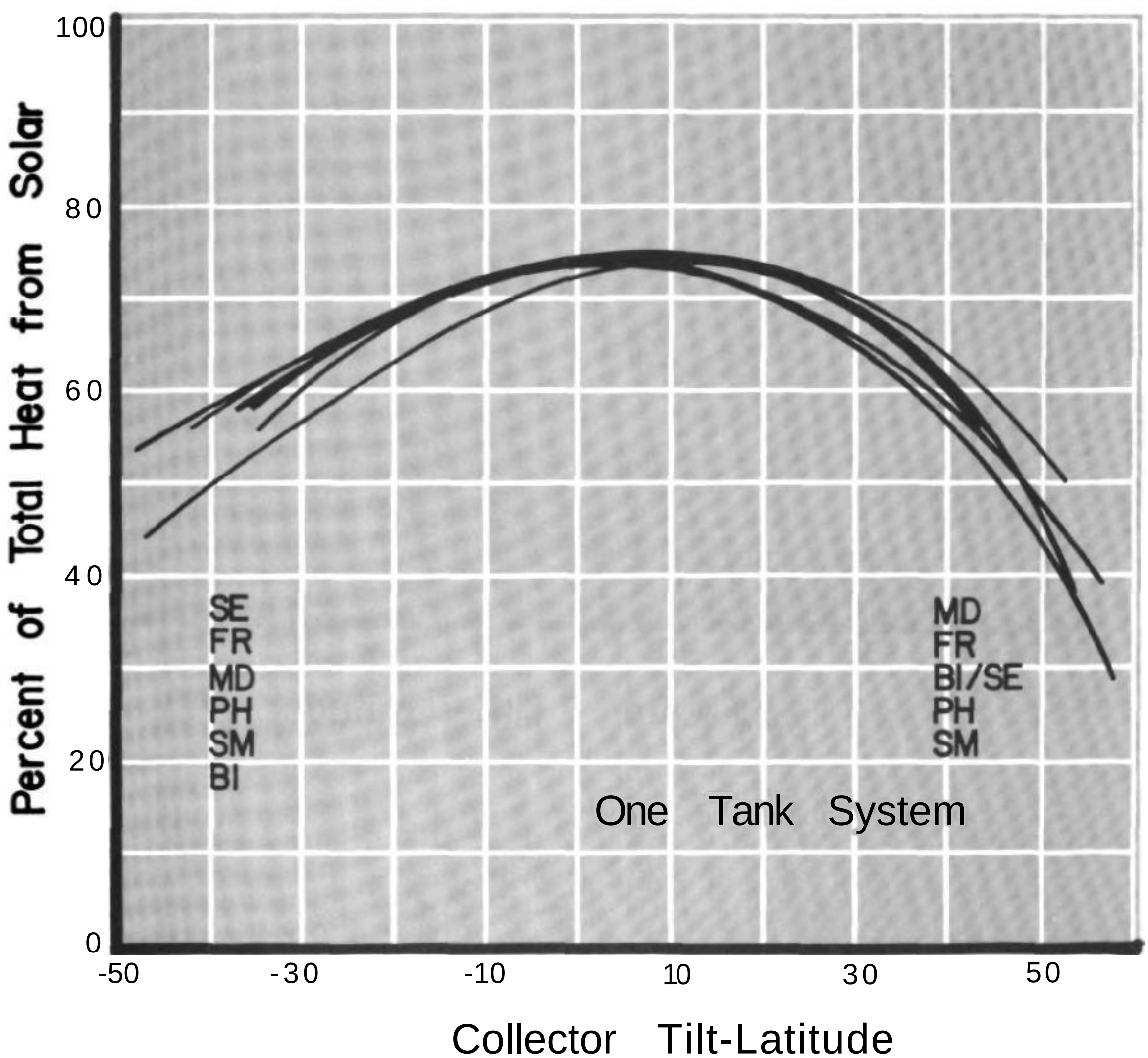
100
80
60
40
20
0
0.5
1.0
1.5
2.0
2.5
Percent of Total Heat from Solar
Collector Area/Load (ft_c^2/gal/day)
PH
SM
FR
BI
MD
SE
One Tank System

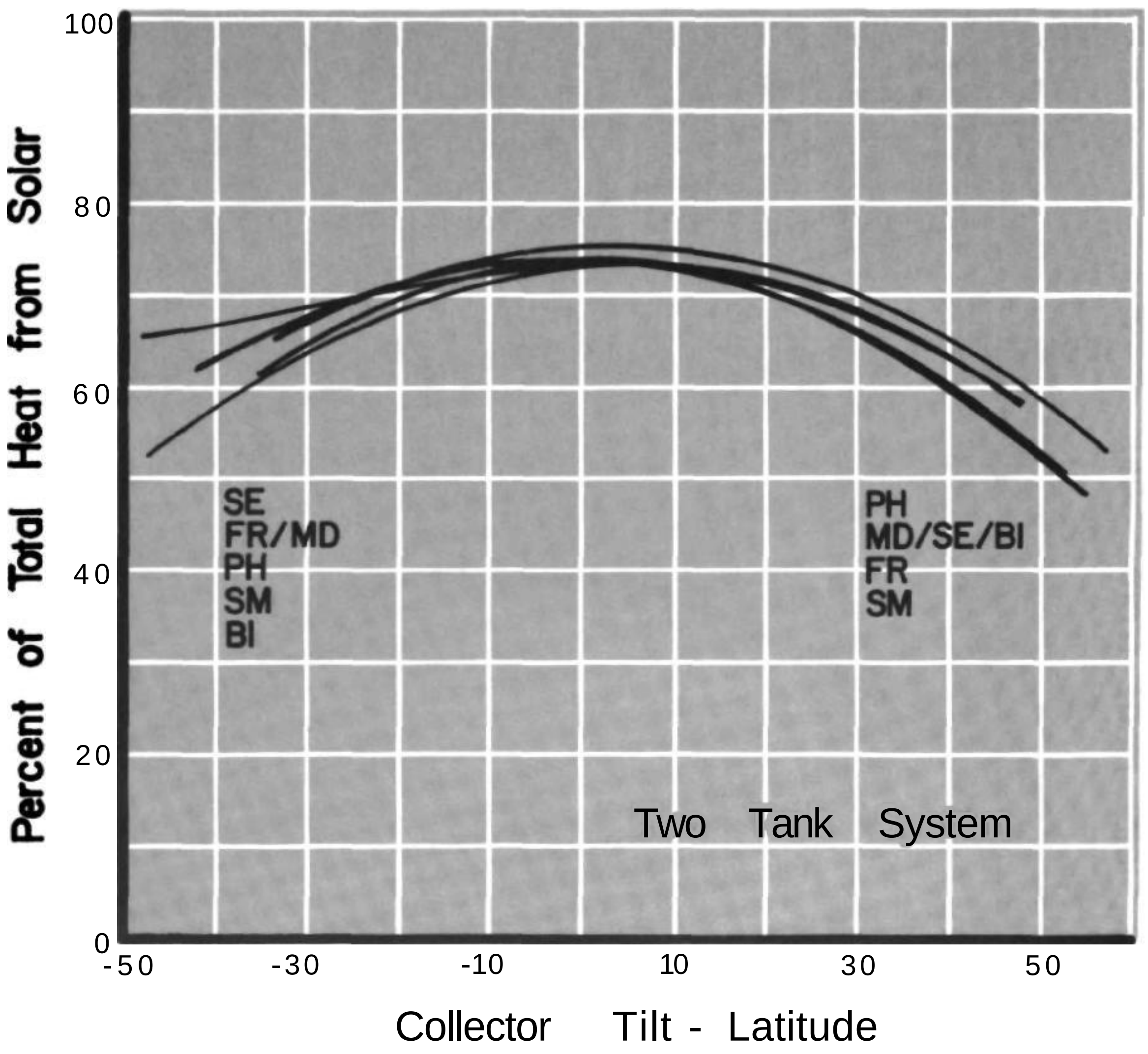

100
80
60
40
20
0
0.5
1.0
1.5
2.0
2.5
Percent of Total Heat from Solar
Collector Area / Load (ft$_c^2$/gal/day)
PH
SM
FR
BI
MD
SE
Two Tank System

2. Collector Tilt. Because of the constancy of the load, the effect of collector tilt for hot water heating is quite different from that of space heating. The following two graphs show the effect for both one and two-tank systems. For both systems there is some variation in the optimum tilt in relation to latitude. As would be expected, the performance of the one-tank system falls off rapidly as the collector tilt is moved more than 20° from the optimum. However, the two-tank system shows a much broader optimum; that is, the performance is less sensitive to collector tilt.

The optimum tilts are as follows:

Location	Latitude	Optimum Tilt One-Tank	Optimum Tilt Two-Tank
Phoenix	33	36	35
Santa Maria	35	37	36
Fresno	37	46	39
Medford	42	53	50
Seattle	48	58	54
Bismarck	47	57	56



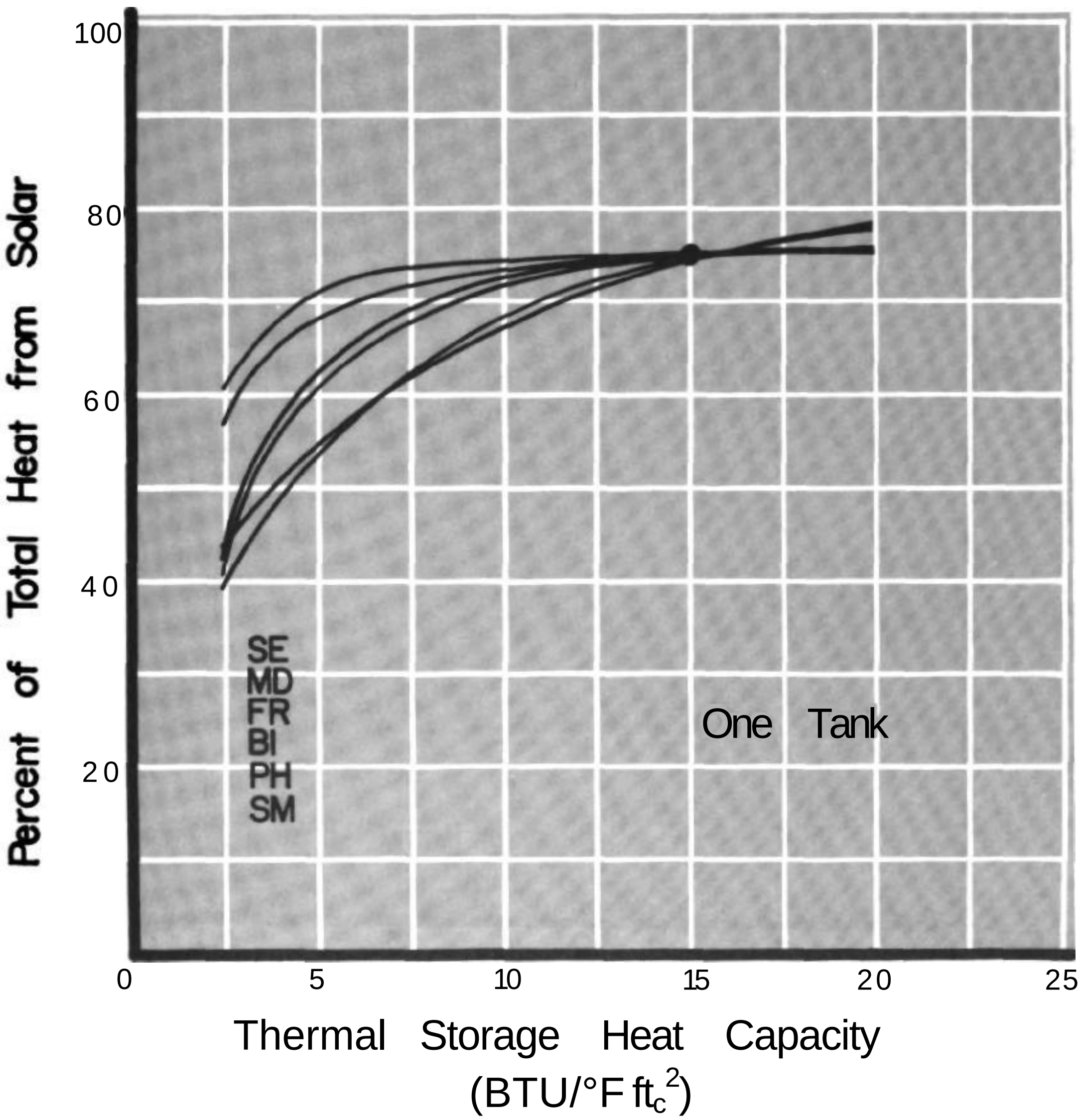


100
80
60
40
20
0
Percent of Total Heat from Solar
SE
FR/MD
PH
SM
BI
PH
MD/SE/BI
FR
SM
Two Tank System
-50
-30
-10
10
30
50
Collector Tilt - Latitude

3. **Water Storage Mass.** The effect of storage mass for domestic hot water is similar to that of space heating systems. Values greater than 15 lb/ft_c^2 yield little improvement in performance. The heat loss from both storage tanks is made proportional to the tank surface area by using the relation:

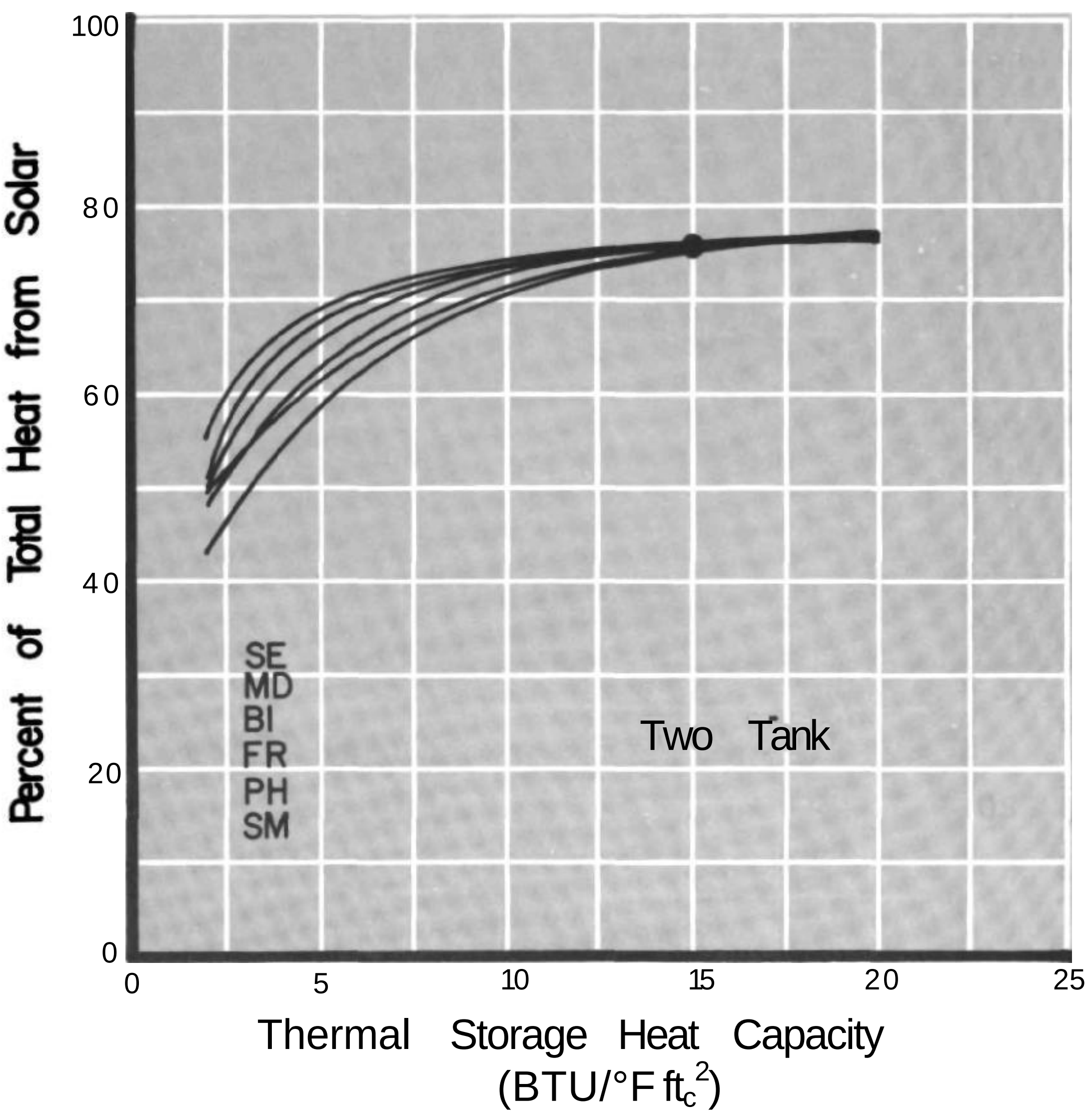
$$\frac{\text{Tank Surface Area}}{\text{Collector Area}} = 0.5 \left(\frac{\text{Storage } MC_p}{15}\right)^{2/3}$$

There is a wide range of sensitivity to changes in the storage mass for the one-tank system. For systems in Phoenix and Santa Maria, the solar fraction falls off rapidly when the storage mass is decreased. Systems in Seattle and Medford show less sensitivity. For the two-tank system, the overall effect of changing the storage mass is less than for the one-tank system throughout the region.

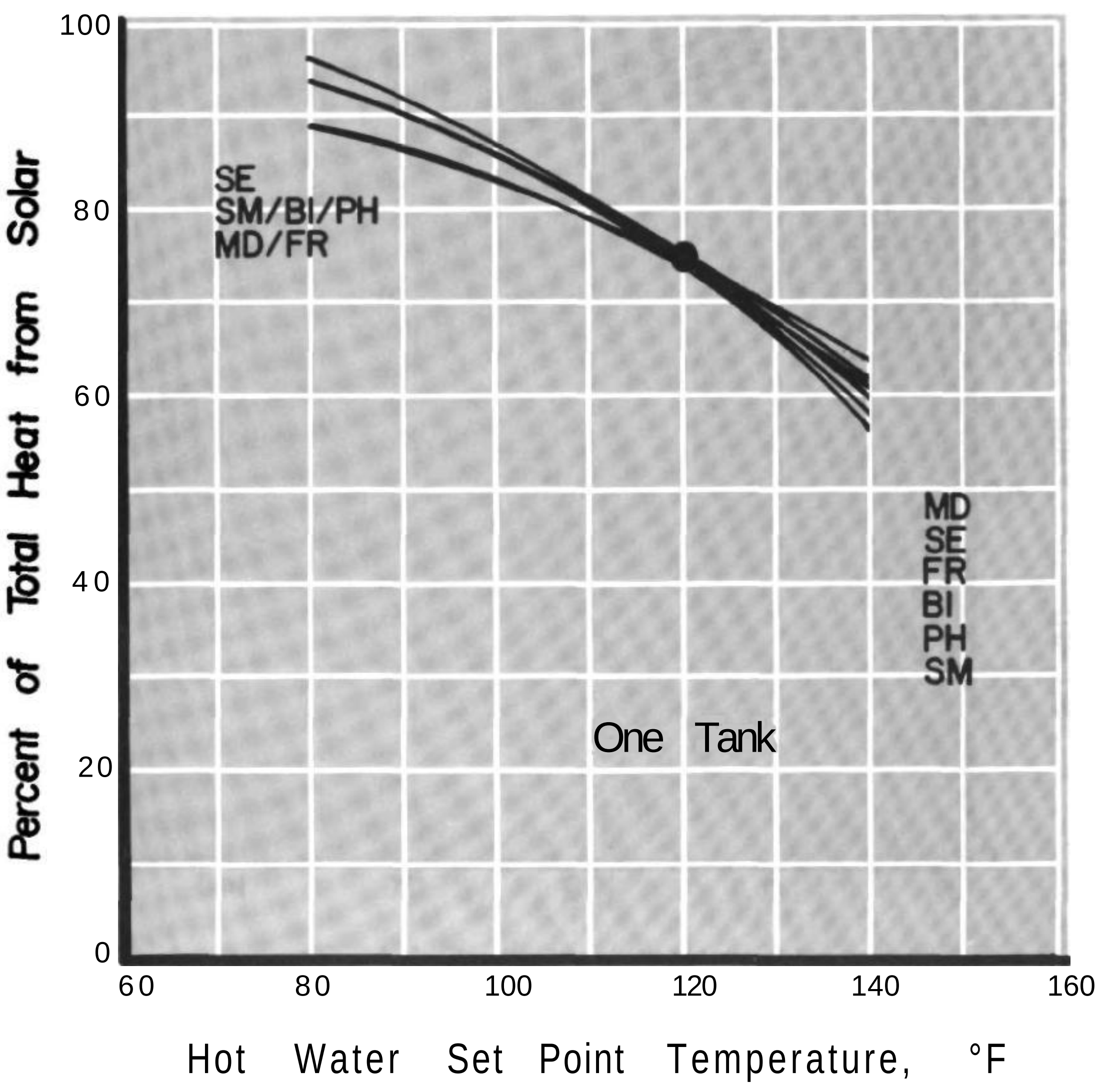


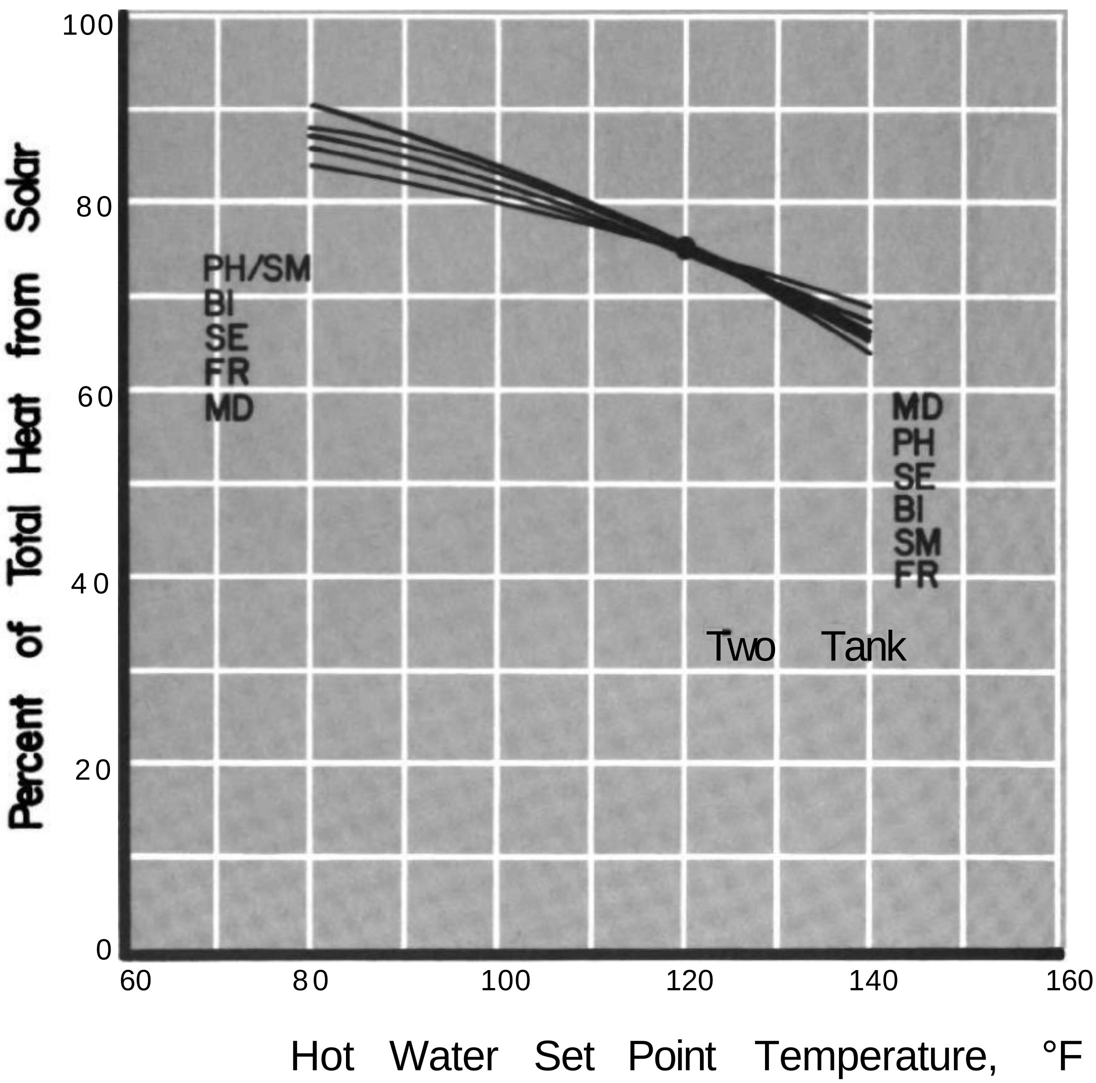
The storage masses for the auxiliary tank in the two-tank systems for a 75% solar fraction are as follows:

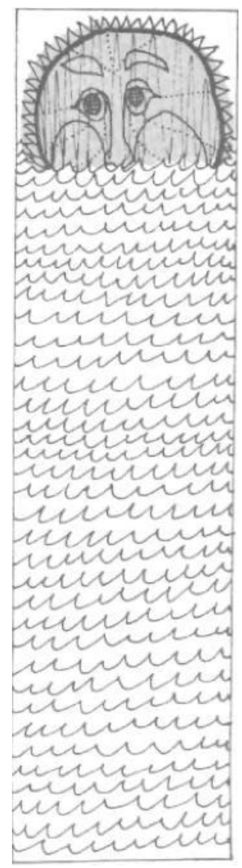
Location	Storage Mass Tank 2 (lb/ft^2)
Phoenix	10.2
Santa Maria	8.37
Fresno	6.43
Medford	4.37
Seattle	2.17
Bismarck	4.52



4. Hot Water Temperature. The two graphs below show the effect on the solar fraction of varying the hot water control temperature from 120°F. In the range of 100°F to 140°F, the fraction of heat from the solar hot water system decreases by an average of 5.8% for every 10°F increase in hot water temperature for a one-tank system, and by an average of 3.8% per 10°F for a two-tank system.




100
80
60
40
20
0
Percent of Total Heat from Solar
PH/SM
BI
SE
FR
MD
MD
PH
SE
BI
SM
FR
Two Tank
60
80
100
120
140
160
Hot Water Set Point Temperature, °F



Swimming Pool Heaters

VII. Swimming Pool Heaters

The components of active solar swimming pool heaters are similar to those of domestic hot water and active space heaters of the liquid type. In general, all employ flat plate collectors and the heat transfer medium is moved from collection to water storage areas by electric pumps. Space heaters and hot water heaters commonly extract water at 140°F from collectors whose surfaces can exceed 200°F, and operate with an efficiency of 30-50%. Swimming pool heaters, by contrast, raise the temperature of several thousand gallons of water only a few degrees, to around 80°F, but operate at an efficiency of 70-80%.

Solar pool heaters thus take full advantage of the fact that solar collectors are more efficient when working at low temperatures. Collectors normally need not be glazed or insulated for swimming pool applications, which results in a simpler design that is far less expensive than collectors for space and hot water heating, which require glazing and insulation to attain higher temperatures. Since pool heaters are primarily used to extend the swimming season, they are operated during those times of the year when there is more solar radiation than in mid-winter and when ambient temperatures are relatively high.

An unheated swimming pool has a natural yearly temperature cycle that varies with climate and geography. A comfortable three to four month swimming season can be stretched out to five or six months when a pool heater is added in northern California, and even longer in southern California. With a solar pool heater, the season will be extended primarily in the spring, less so in the autumn, and the pool temperature can be maintained on an average of 10°F above an unheated pool. The simplicity and built-in efficiency of swimming pool applications allows the designer considerable leeway in all parameters except that of collector area. The large quantity of water to be heated dictates a large solar array in order to collect sufficient amounts of solar energy; collector size will therefore depend on the size of the pool, and on the orientation and tilt of the collector. In addition, tilt will also depend on what part of the season is primarily to be used for heating the pool.

1. Collector Area. As a rule of thumb, the collector area should be equal to at least one half of the pool surface area in order to extend the swimming season into spring and autumn in a reasonably sunny climate, given optimum tilt and orientation. In some areas it may be necessary to increase the collector area to equal the entire surface area of the pool.

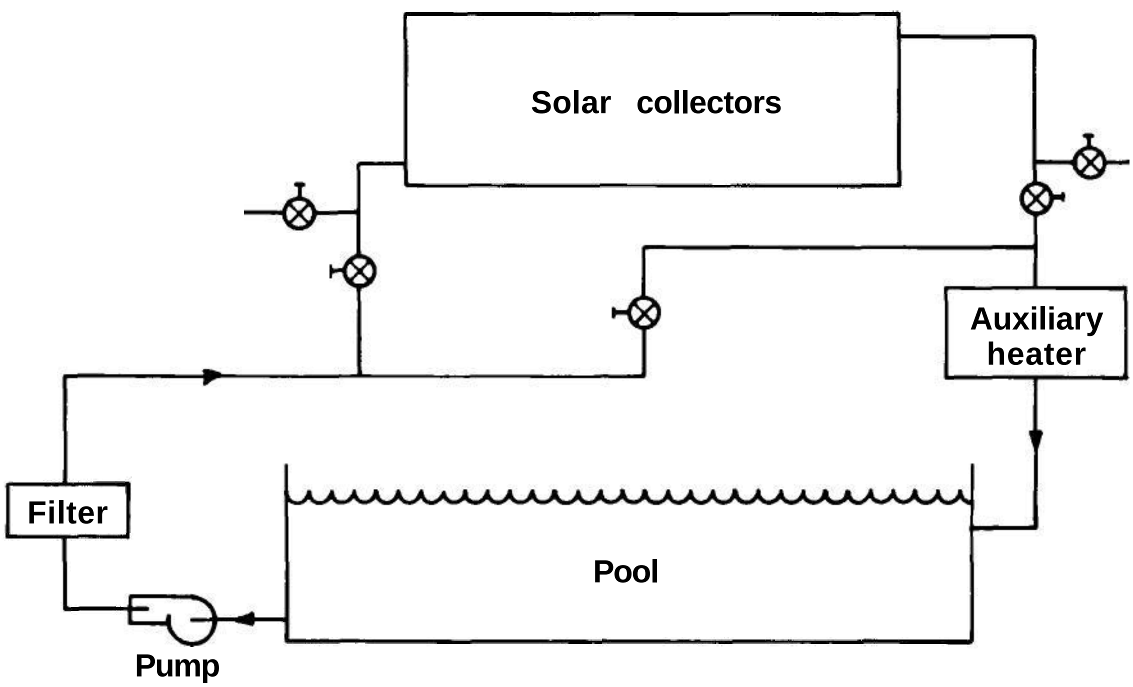
2. Collector Orientation. The optimum orientation is south, but west-facing orientations are satisfactory as long as the collector area is increased to a minimum of 75% of the pool surface area. East-facing orientations are marginal.

3. **Collector Tilt.** For year round heating applications in warm climates or where a solar heater will be used to supplement a gas-fired heater for year round use, the optimum tilt is equal to the latitude of the installation site. For primarily summer heating, the tilt should be equal to the latitude minus 10-15°. For winter heating, the tilt should be equal to latitude plus 10-15°. In situations where it is desirable to install the collector horizontally, such as on a flat roof, the collector area should be increased to 75% of the pool surface area.

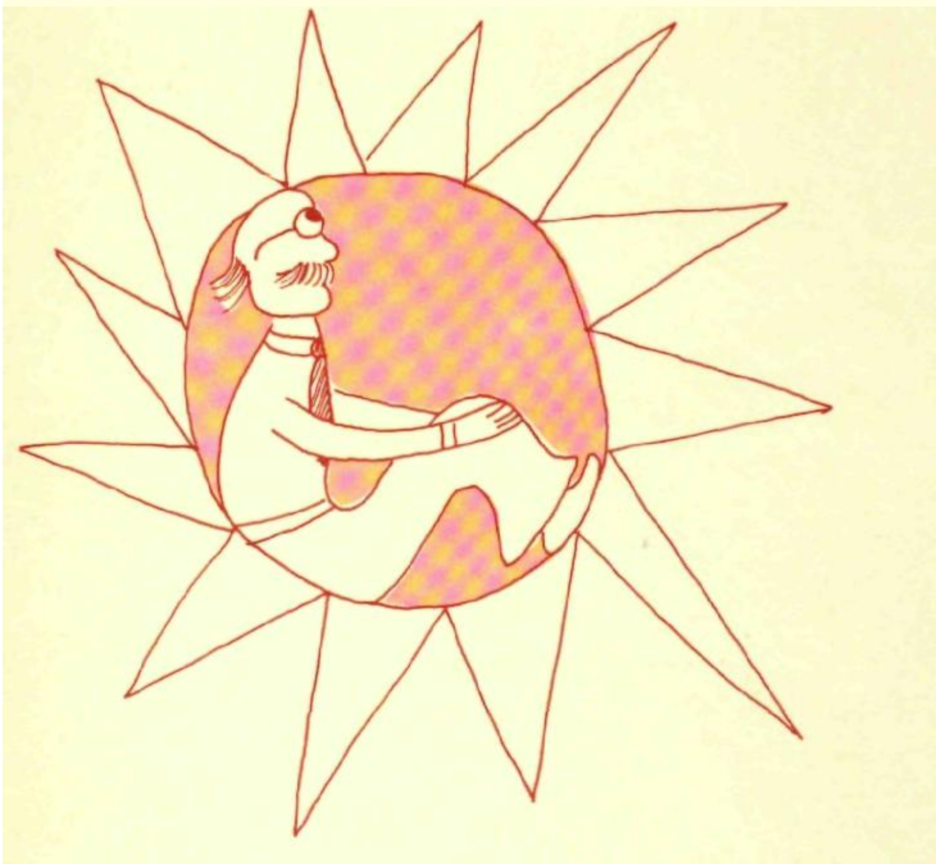
Since all swimming pools require a pump and related plumbing, the addition of a solar pool heater to an existing installation can be relatively simple. A pressurized hydraulic system with a high flow rate is necessary to keep the collector panels cool and thus operating at a high efficiency, and the collector should not be set at too great a distance from the pool. A south-facing rooftop can be an ideal installation site in cases where the roof structure can support the additional weight of the water-filled collectors, which can be expected to have a cooling effect in a hot climate on the roof and the structure beneath.

The swimming pool itself is the thermal storage area, employing an effective thermal storage medium—water. The pool can also serve as a collecting surface. Transparent pool covers, a variety of which are commercially available, can contribute substantially to maintaining and raising water temperatures by passive means, and should be considered in conjunction with solar pool heaters of the active type.

SWIMMING POOL SCHEMATIC



Space Heating, Passive Designs



VIII. Space Heating, Passive Designs

General Design Considerations

The active space heating systems discussed in Section V utilize separate thermal storage in the form of a water tank or rock bed. In them, thermal energy is transferred from the collectors to storage and subsequently from storage to the building thermal control system in a completely regulated way by means of electrically powered pumps or fans that move the heat-transfer fluid — liquid or air — throughout the system.

An alternate approach to solar space heating is characterized by a reliance on natural convection and radiation, and by heat collection and storage areas that are integrated with the living space, rather than separated from it. It has come to be known as the "passive" use of solar energy. Buildings heated by passive means have sometimes been referred to as "sun-tempered."

Since almost any building benefits from direct solar gains, all can be said to be passively heated to some extent. It is when solar energy utilization becomes a major objective of the architectural design, and thus when solar energy supplies a major fraction of the heating requirements, that one would refer to the structure as a solar heated building.

The term "passive" is now in widespread use. Technically it is preferable to define a passive solar heating or cooling system as one in which the energy flow is entirely by natural means. Designs which utilize motorized or manually operated insulation panels or shading devices once or twice a day can still be considered passive by this definition provided that the thermal energy flow is by natural convection, conduction, or radiation. System designs which utilize small fans to assist circulation should not be ruled out simply because they are not strictly passive in cases where the use of a small amount of auxiliary energy can materially decrease the overall conventional heating or cooling requirements. In short, a passive design should seek to maximize the use of natural convection, conduction, and radiation processes while minimizing the use of mechanical devices powered by conventional fuels.

A number of examples of passive solar heating concepts have been built into structures and have received widespread attention for their apparent success in saving energy. Some of these designs are discussed beginning on page 82. But despite the publicity given to them, these designs have not yet been widely adopted. This has probably been due to a combination of skepticism about their effectiveness and a lack of engineering criteria—there has been little quantitative assessment of their importance. And at the present time, an increasing interest among architects in passive concepts still continues to outpace the development of thermal criteria to guide their use.

The quantitative problem with passive designs does not lie in the area of collection. Calculation of solar gains through window areas is well understood and documented. A window is an efficient solar collector, and adequate solar energy collection combined with proper heat conservation measures can provide for a large fraction of the heating load of a building. In a passive design, the surface receiving the solar flux can be combined with the thermal storage mass in a more or less integrated unit, as in the drum-wall system, or the storage can be separate from the directly irradiated surface, as with massive internal walls not directly exposed to sunlight, or there can be a combination of storage masses directly and indirectly heated by solar radiation—the usual case of a house with large south facing windows, with floor areas, furniture, and some internal walls, etc., in shadow. In general, it may be said that the arrangement in which the thermal storage medium is directly heated by the solar flux will provide for the highest fraction of passive solar heating and minimize problems of daytime overheating.

The challenge confronting passive solar energy design is one of storage and control of heat to maintain suitable comfort standards within the building. The situation is paradoxical: storage of sensible heat requires a temperature change in the storage medium, and yet the objective of building thermal control is to maintain a constant temperature. Thus, if the storage is to be part of the living space of the building, how can they be made compatible? One approach is to use, as it were, brute force in the form of such a large mass of heat capacity that the temperature variations are tolerable, and this generally has been favored over the alternative of insulating the storage mass from the living space.

Moderately effective controls have been designed to deal with other problems in passive designs, and include movable shading devices to control sunlight, movable insulation panels to reduce night-time heat losses, and ventilation ports to either augment or reduce daytime heating by means of natural convection. These, and other elements of passive designs which should be considered from both architectural and engineering viewpoints are presented below.

Elements of Passively Solar Heated Building Design

Element	Application
Thermal Insulation:	Fixed insulation is used in decreasing natural energy flow to maintain building interior comfort, e.g., thermal insulation retains building interior warmth in a cold environment, and coolness in a warm environment. Movable insulation can be used to diminish natural energy flow through windows at desired times while allowing energy flow at other times.
Fenestration:	Windows act as an effective solar collector. Windows can be used to admit solar radiation either to warm the structure or for lighting. Window orientation is extremely important. South windows receive maximum winter gains and minimum summer gains.
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. Deciduous trees provide a natural seasonal control of shading.
Reflectors:	The use of diffuse or specular reflectors can significantly increase the total influx of solar radiation through a window. In the cooling mode, high emittance exterior surfaces reflect visible radiation and radiate heat in the infrared.

Elements of Passively Solar Heated Building Design (continued)

Building Structural or Added Mass:	Mass within the building provides natural thermal storage of sensible heat.
Thermal Radiation:	Thermal 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. In warm periods, thermal radiation and convection can be used to remove heat from the building and also from the storage mass during the night time.
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.
Conduction:	Materials of high mass and low thermal diffusivity (such as ordinary masonry) can delay the arrival of a thermal wave until the heat can be used effectively.
Air Stratification:	Warm air can be stratified at the ceiling and removed by natural convection through vents to reduce the air-conditioning load. Alternatively, stratification can be used to concentrate warm air for use elsewhere.
Evaporation:	Static outdoor ponds (perhaps on the roof) 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.
Heat of Fusion Storage Materials:	Thermal masses normally 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, and store heat without a change of temperature. In cooling, phase change materials offer a promising thermal heat sink. However these materials have not yet been developed to the point of practical use in passive systems.

Some Passively Solar Heated Building Designs

The use of passive design elements can be best demonstrated by buildings that have successfully incorporated them. Several existing buildings are discussed below in detail. They are grouped according to the types that represent three major approaches to passive solar heating: direct gain, thermal storage wall, and roof pond.

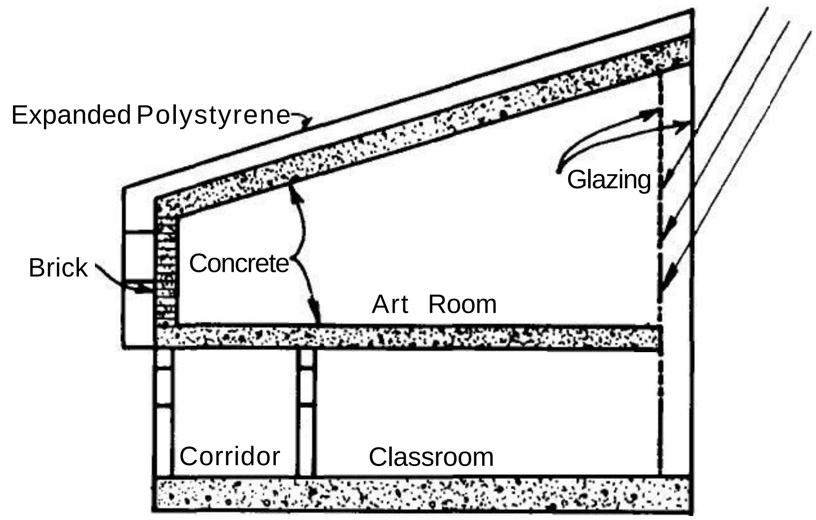
1. Direct Gain. The simplest approach to passive space heating is through direct gain of solar radiation by means of a south-facing expanse of glass. This approach works best when the south window area is double glazed and when the building has considerable thermal mass in the form of concrete floors and masonry walls insulated on the outside. What results is, in effect, a live-in solar collector thermal storage unit. If the south-facing window area is vertical, seasonal temperature control is basically automatic, as the interior space of the building is exposed to a maximum amount of solar energy in the cold winter months and to a minimum in the summer when the sun angles are high.

An example of a direct gain system is the Wallasey School located in Liverpool, England, at a latitude of 53° north. Designed by A. E. Morgan and built in 1962, it was one of the first passively heated structures built in modern times and remains, at this writing, the largest. The basic construction is of concrete, with roof, back and side walls and floor 7 to 10 inches thick, and exterior surfaces insulated with 5 inches of expanded polystyrene. The south-facing double-glazed solar wall is 27 feet tall and around 230 feet long. The outside sheet of glass is clear, separated by a space around 2 feet wide from the inside sheet of "figured glass"—a type of glass that refracts the suns rays in such a way that ceiling and floor of the structure are irradiated in a fairly uniform manner. The building is heated to about 50% by the sun, and the remainder is provided by the heat given off by the lighting and the students.

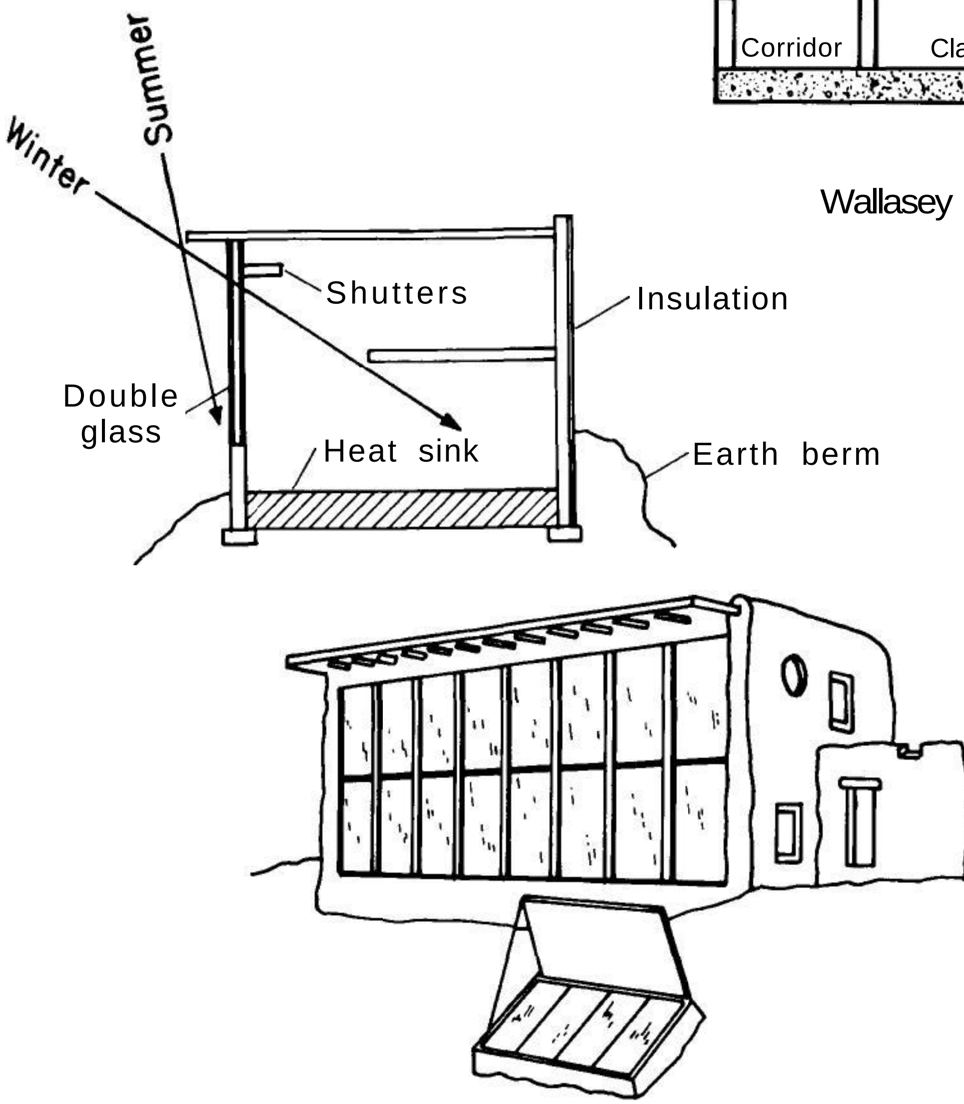
Another direct gain system is the David Wright house in Santa Fe, New Mexico, located in the piñon foothills of the Sangre de Cristo mountain range, at an elevation of around 7,000 feet. The basic structure of adobe or earth brick is D-shaped in plan with the flat-side facing south. On the east side is an entry area with an "air lock" door arrangement that prevents excessive infiltration of cold air into the primary living space in the winter. The south face of the two-story house is a wall of double-glass that rises from a few feet above the floor to the roof beam line, and this serves as the collector system. At night the glass area is insulated by an accordion-folding shutter lowered from the ceiling. A wood-fired Franklin stove supplies back-up heat.

The thick adobe walls serve for thermal storage. They are insulated by 2 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. Fifty-five gallon drums filled with water are embedded in the *banco* — an adobe bench characteristic of southwest architecture — under the south windows for additional mass. Domestic hot water is supplied by a thermosiphon solar heater whose collector is located on the south slope outside the building.

The house has been occupied since the winter of 1974-75 and is performing well. During the charging period it tends to overheat but excess heat can be vented out through windows on the second floor. They also serve for summer ventilation. The roof overhang on the south side shades the interior of the house from the summer sun. Daily temperature variations are characteristically 20°F in the downstairs of the house.



Wallasey School



THE DAVID WRIGHT HOUSE

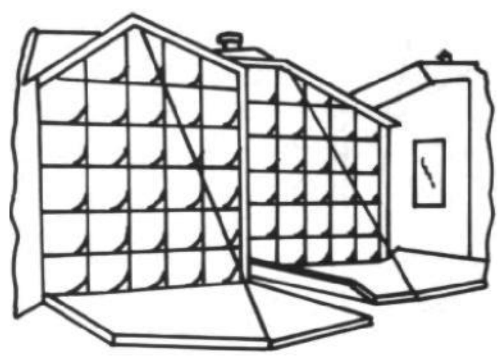
2. Thermal Storage Wall. The second type of system uses a wall set directly behind single or double glazing for thermal storage. The wall is usually painted black or a dark color for good absorptance and can be of masonry or of water-filled containers.

An example of a storage wall approach is the Steve Baer house located on a south-facing slope in the sand hills northwest of Albuquerque, in a climate well-suited for solar heating. The altitude is approximately 6,000 feet, and though the winters can be severely cold during short periods, times of extended cloudiness are relatively infrequent.

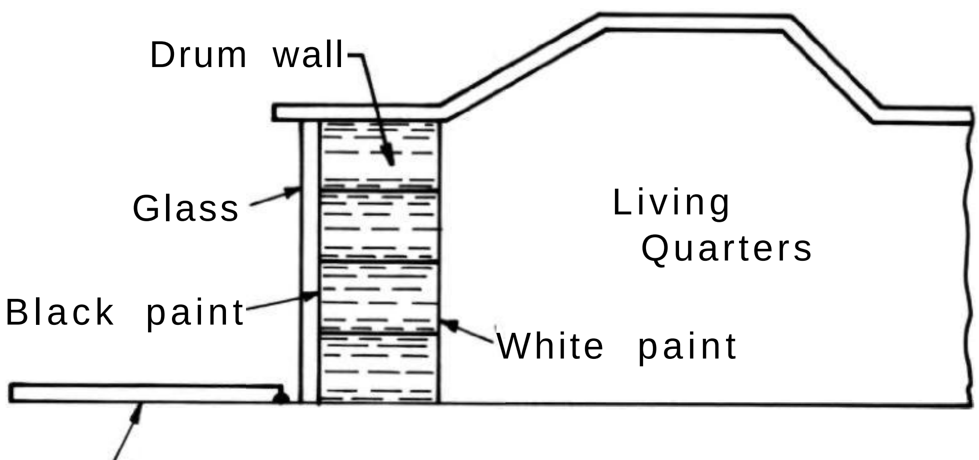
The solar heating and storage system of the Baer house is centered around a south-facing wall of water-filled 55-gallon drums, stacked on a metal rack in a close-fitting horizontal array. The south-facing ends of the drums are painted black and thus act as the collector surface behind a floor-to-ceiling glass wall, single glazed.

To prevent heat losses from the "drum wall" at night or during cloudy periods, an insulated panel, hinged at the bottom, can be raised to cover the entire surface of the glass area. Furthermore, the inside of the cover panel is faced with aluminum sheets so that in its vertical closed position it acts as a reflective insulation to interior thermal radiation — while in its horizontal and open position it reflects solar radiation on to the "drum wall."

The ends of the drums facing the interior are painted in light shades to complement the decorating scheme of the space they serve to heat. Visually, from inside the building, the "drum wall" is interpreted as an architectural screen, with the brightly lit spaces between the drums presenting a repetitive star-shaped pattern.



South wall



THE BAER HOUSE

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, it is calculated that each drum releases about 418 BTU for a 1°F drop in the water temperature.

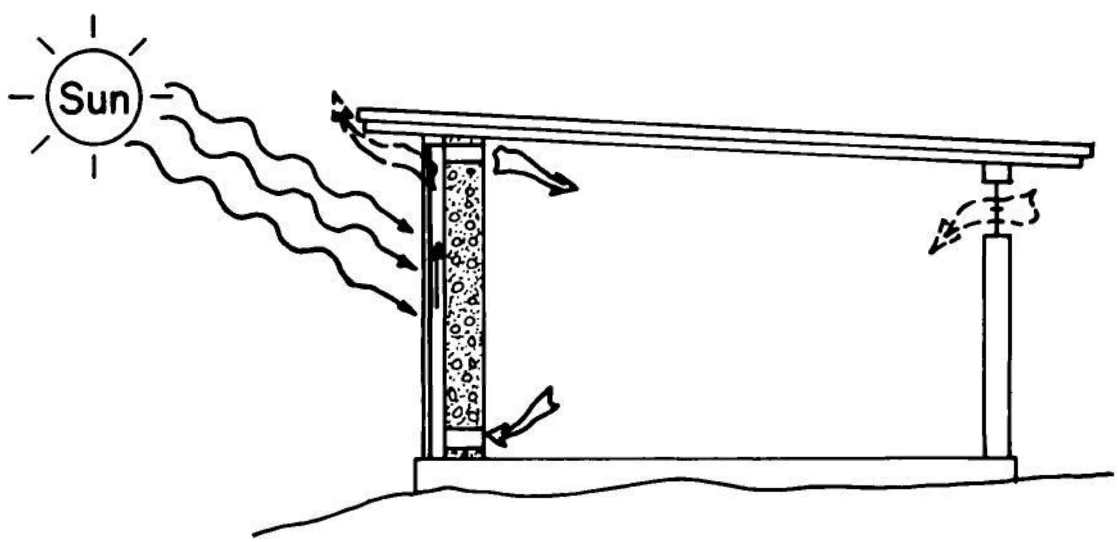
While the "drum wall" provides heating for most of the building, the kitchen is fitted with a Baer "sky dome," a large skylight fitted with rotating insulated louvers and covered by a clear plastic dome. A simple freon-filled balanced control that requires no external power senses the direction of radiant 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.

A second storage wall approach is exemplified by the well-known Trombe-wall houses constructed in the solar community near Odeillo in the French Pyrenees by Felix Trombe and his colleagues.

The basic passive element is a massive south-facing concrete wall. The exterior surface of the concrete is painted black, and is double glazed in such a way to provide an air-passage space between glass and concrete, and the air passage is connected with the living space by means of openings in the concrete wall near the floor and ceiling.

In the heating mode, solar radiation strikes the black-painted concrete surface. As heat collects, a convective air flow circulates through the space between the glass and concrete wall. Cool air is drawn from the floor-level openings and hot air discharged into the living space through the ceiling-level openings. Thus, a convective loop is established through the whole collection area and living space.

At the same time the massive concrete wall is slowly accumulating thermal energy as the portion of solar heat not removed by convection diffuses into the concrete to be stored there. At night the convective loop is closed off, and the



TROMBE HOUSE

concrete storage wall acts as a low-temperature radiant heating panel to the living space. The exterior face of the wall loses heat to the environment, but the double glazing reduces these losses to an acceptable level.

Data taken from a Trombe-wall building constructed in 1967, in which the walls are two feet thick, indicated that roughly 36% of the solar radiation incident on the south wall during winter months is transferred into the house. Over the heating season, around 70% of the heating load was provided by solar energy, with the remainder by a conventional thermostatically controlled auxiliary system. Of this 70%, about 20% was transported into the living space by convection through the vents and the remaining 50% by conduction through a wall.

In the summer, the overhanging roof shades the glazed concrete wall from the sun. Vents at the top of the glazing allow warm air to escape outside, thus setting up a convective circuit that ventilates the living space, as indicated by the dashed arrows in the sketch on page 85.

3. **Roof Ponds.** The last type is the roof pond system in which thermal storage is on the roof rather than in a south-facing wall. The shift from vertical collection-storage to horizontal results in a system that can be used for summer cooling as well as for winter heating.

The Skytherm system was first tested in one-room structures in Phoenix, Arizona, and has been recently evaluated in a full-scale test house especially designed for the system and built in Atascadero, California. The site has a more severe climate in terms of heating requirements than the original Arizona test site, and the test has indicated that it is possible to have economical solar heating and night sky cooling with an integral Skytherm system.

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. The system was able to keep the indoor temperature between 66° F and 74°F, except during special test periods or during times of prototype breakdown. Yet even during these exceptional periods, the temperature never went above 79°F or below 62°F. The indoor temperature at the 5-foot 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 the system was about 24,000 BTU/day in February. The largest monthly average cooling load was about 168,000 BTU/day during July. The experimental house had an overall heat transfer coefficient of about 9,500 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 about 1,100 ft^2, about the same as the floor area. The average water depth of the roof ponds was about 8.5 inches, for a total of about 6,000 gallons of water. The system was operated with the thermoponds both unglazed and glazed with an inflatable clear plastic cover. The cover proved necessary in order to keep the indoor temperature up to the reported level during the winter months. Without it, the indoor temperature would have dropped to near 60°F in the early morning hours.

The weight of the roof ponds created no major problems, even for a site located in an earthquake area. The design of Skytherm is amenable to prefabrication, which would result in minimal on-site labor. Problems with roof leaks have been corrected and additional watertight roof designs are now being studied. The insulation panels and automatic mechanism functioned well.

On sunny days in winter, the insulation panels slide back and expose the black plastic thermoponds. At night or during cloudy periods, the insulation panels slide back over the thermoponds, thus preventing heat losses back into the sky.

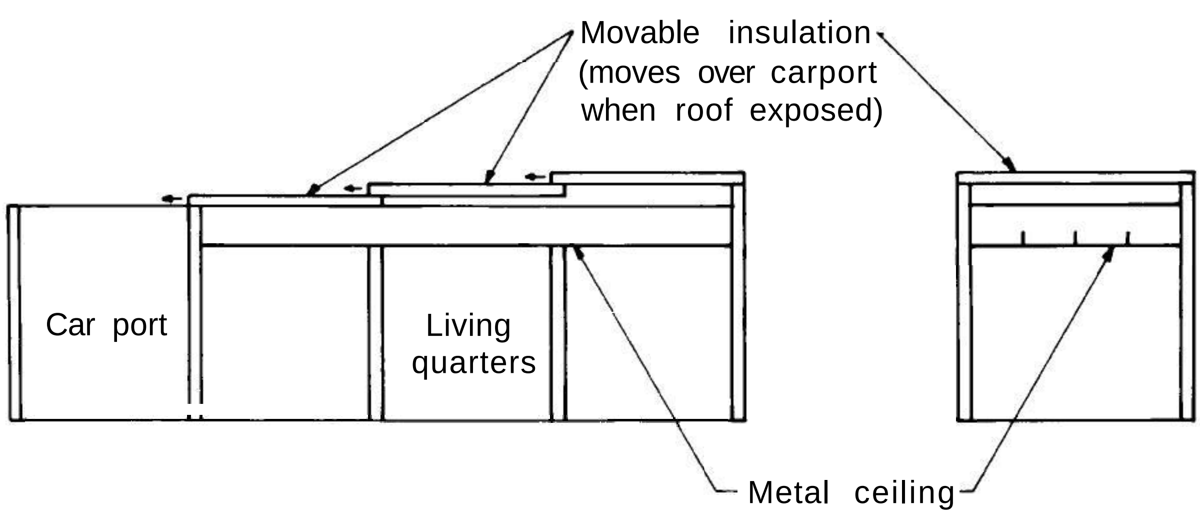
The thermoponds make good thermal contact with the metal ceiling: heat transfer to the living space is thus by radiation and convection.

During the winter months, the thermoponds act as a large thermal storage mass with an average temperature above that of the average ambient temperature.

In summer, the roof insulation panels are closed in the daytime hours, during which time heat inside the house is transferred to the cooler thermoponds. At night, the panels slide open, exposing the thermoponds to the sky and the heat they have collected from inside the house discharges by radiation into the sky. Thus, in the summer mode, the thermoponds act as a large thermal storage mass with an average temperature below that of the ambient temperature.

It was necessary in summer to deflate the clear plastic covers of the thermoponds in order to keep the living space temperature below 80°F. However, better ventilation of the thermoponds and elimination of panel air leaks would probably have allowed the system to be operated with the covers inflated.

There are some geographical limitations to the Skytherm system. In the south, for example, the high humidity would hamper night radiation from the exposed thermoponds to the sky during the summer cooling mode. For locations of high latitude in the north, some kind of vertical reflector would be necessary to direct the low winter sun on to the horizontal thermoponds.

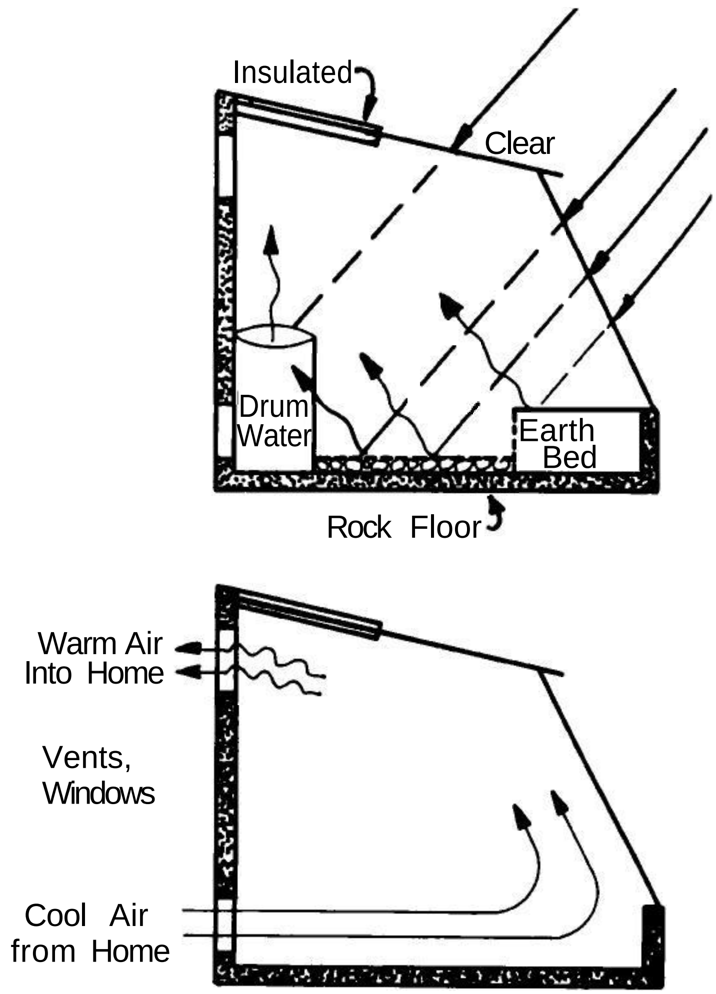


THE HAROLD HAY HOUSE

4. Attached Solar Greenhouse. A combination of direct gain and thermal storage wall concepts is to be found in the solar greenhouse, in which the glazed greenhouse structure provides for direct gain of solar radiation, with thermal storage in the south-facing massive wall of the building to which it is attached. Some twenty solar greenhouses of this type have been built in recent years by the Solar Sustenance Project under the direction of William and Susan Yanda. The goal of the project was to reduce substantially the convential space heating requirements of the buildings to which the greenhouses were attached, while also providing an indoor growing area during the winter months. The project emphasized construction techniques within the reach of the homeowner moderately skilled in carpentry. Materials costs averaged around $2.50 per square foot.

Under this project, greenhouses have been retrofitted to mobile homes and houses of frame, pumice block, and adobe construction in New Mexico. For houses of massive construction, the south-facing wall to which the greenhouse was attached was used as heat storage, while for houses of lightweight construction it was found necessary to add the storage mass to the greenhouse area in the form of water-filled drums or stone floors. In order to maintain temperatures 30°F above outdoor lows in winter, it was estimated that the greenhouse should contain a storage mass the equivalent of about two gallons of water or 80 pounds of concrete per square foot of glazing.

The greenhouses were double glazed with ultra-violet resistant fiberglass paneling on the outside and UV-resistant polyethylene film on the inside, and were designed with a partial solid-roof overhang to minimize overheating spring through autumn. They were built against existing door and window openings to allow circulation of the warm air from the greenhouse into the living space—and, during severely cold nights, circulation of warm air from inside the house to heat the greenhouse space when necessary to protect plants from freezing.



Bill Yanda Greenhouse Design

* The drawings were taken from the following reference by permission of the authors: W. F. Yanda and Susan Yanda, *An Attached Solar Greenhouse*, (in English and Spanish). Santa Fe: The Lighting Tree, 1976, pp. 3-4.

Appendix

Appendix A—Determination of Performance By Simulation Analysis

The performance of a solar heating system can be predicted by simulation analysis using actual weather and solar data, and a mathematical model of the solar heating system. A schematic of the model is shown on page 90. The operation of it is simulated by a digital computer code on an hour-by-hour basis.

At each hour the net energy which can be extracted from the collector is calculated. This is determined from solar radiation, collector design, outside temperature and wind speed, 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). Energy is extracted from storage to satisfy the thermal load. At times when storage cannot satisfy the load, 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.

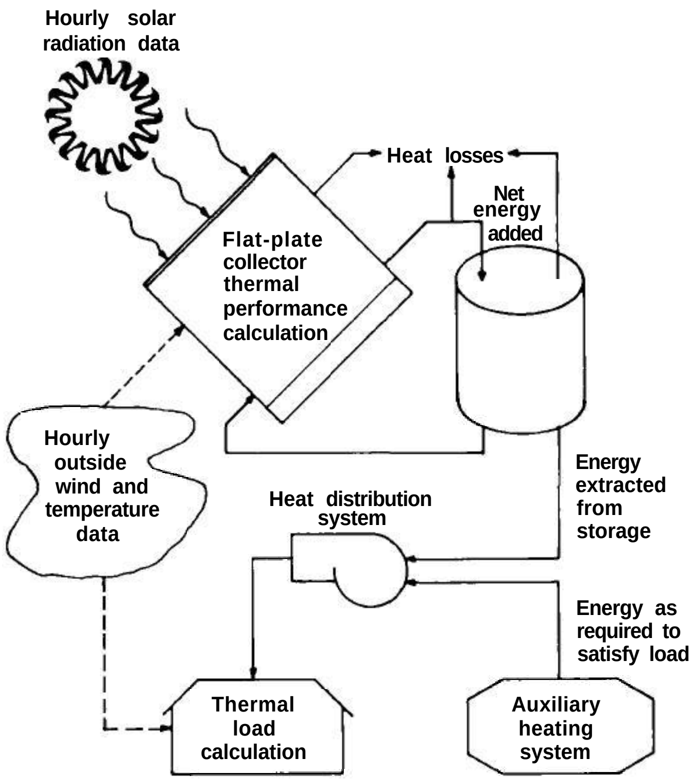
This calculation is repeated for each of the 8,760 hours of the year. All energy flows are summed hour-by-hour, and both monthly and yearly summaries are printed out. A typical year-long calculation requires only 34 seconds on the Los Alamos CDC computer, making it feasible to study the effect of changes in many design parameters.

Examples of the type of results obtained from the simulation analysis are presented here for Fresno, CA. The "solar heating design year" chosen is July 1957 through June 1958. Daily solar radiation and temperature data are shown on page 90.

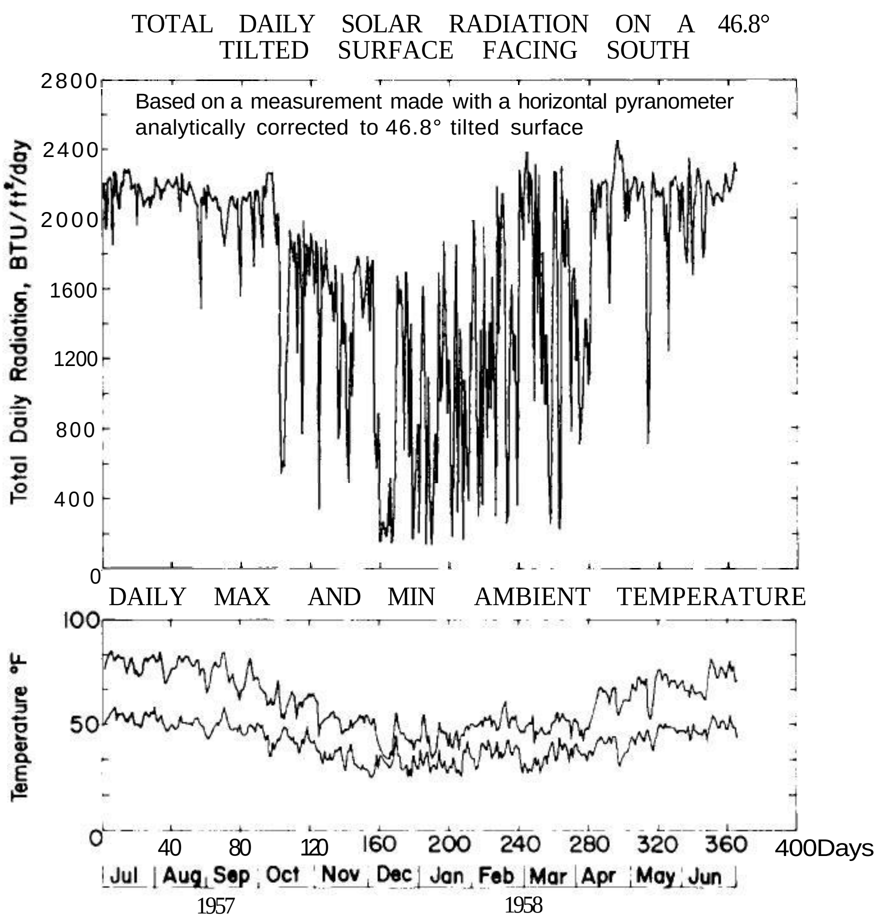
For Fresno, the simulated response of a standard liquid system for seven consecutive days in winter are shown on the graph on page 91, while simulation results for the entire year are shown on page 92. A table of the monthly energy balance is on page 91.

The table on page 92 shows the results of computer simulations for the standard liquid system for several years in Fresno. It can be used to estimate year-to-year variation in solar performance as well as to estimate the average solar performance. The average percent solar heating is 75.2% with a standard deviation of 2.4%. The selection of 1957-58 as the design year was based on the percent of solar heating being closest to the 75.2% long term average.

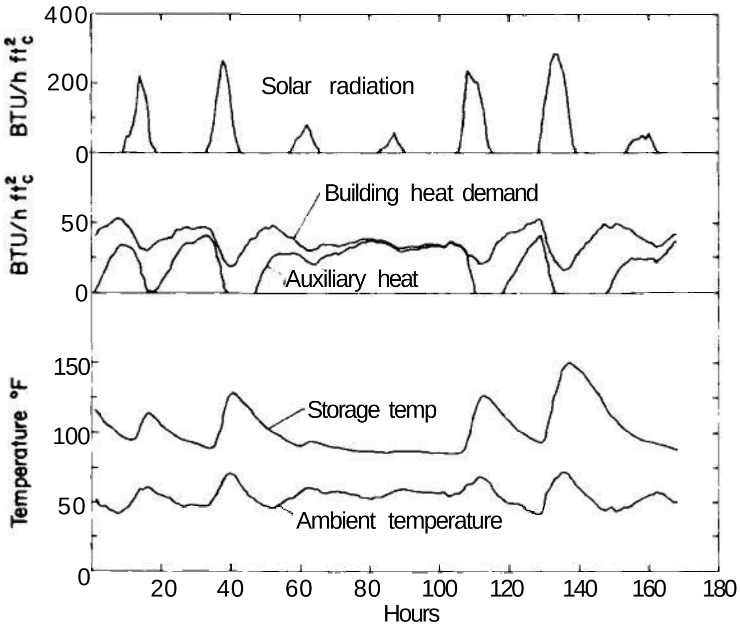
The graphs and tables given in this handbook are all based on the results of this type of simulation analysis. In order to study the effect of changing a parameter, for example the effect of a different thermal storage mass, several year-long computer simulations were made each with a different value of thermal mass. These results are then plotted, as in the graph on page 40, to show the effect on annual percent of solar heating.



SIMULATION SCHEMATIC



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

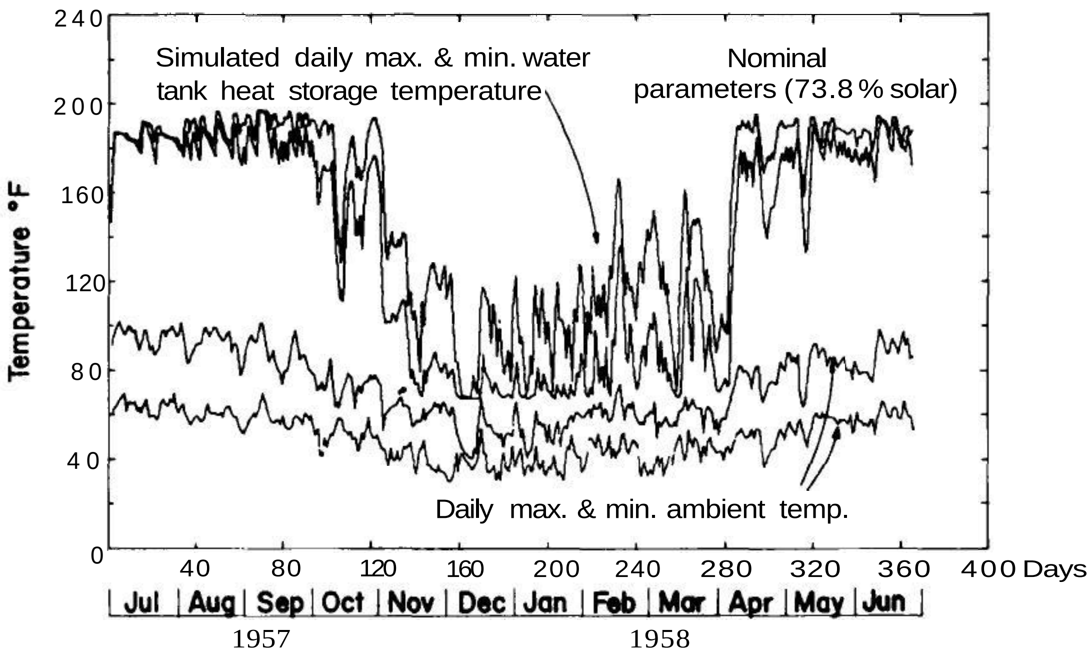


MONTHLY ENERGY BALANCE FOR FRESNO

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
	August	1954	2146	0	100.00
	September	2484	2781	0	100.00
	October	8289	8876	0	100.00
	November	18819	16534	1631	91.33
	December	26043	10439	15575	40.20
1958	January	24879	12857	12597	49.37
	February	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



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	79366	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	2689	75.20

*The year begins July 1 of the year listed and extends through June 30 of the following year.

**Measured on a horizontal surface.

***Calculated for a tilted collector surface.

Appendix B—Case Studies

An example of the procedure for finding the collector size for a given building and location follows in this section. A location which is not analyzed in detail in this handbook is chosen to demonstrate how the procedure is extended to locations other than the six cities for which detailed parameter studies are presented.

Building Load. The building load is given in units of BTU/degree-day. This figure can be obtained from knowledge of the building construction using the ASHRAE Handbook of Fundamentals. The ASHRAE Handbook contains numerous "U" factors or overall wall heat transmission coefficients for various wall and roof types.

For this study, we will assume the following building characteristics:

Single Story, 8 ft ceiling

Floor Area = 2000 ft^2 (40 x 50 ft)

Walls, frame with R11 blanket insulation

Ceiling and Roof, R19 blanket insulation

Windows, 20% of wall area with double panes.

From the ASHRAE Handbook, the following values are found (BTU/hr ft^2 °F)

U wall = .08

U windows = .70

U floor = .10

U roof = .04.

The areas are

A wall = 1152 ft^2

A windows = 288 ft^2

A roof = A floor = 2000 ft^2.

The overall product of U and surface area is then

UA = 574 BTU/hr °F = 13776 BTU/degree day.

In addition, there is a heating load due to infiltration. For one change of air per hour,

Air density (68°F) = .075 lb/ft^3

Air volume = 16,000 ft^3

Air specific heat = .24 BTU/lb °F

Therefore, the heat loss is

(.075)(16,000)(.24) BTU/hr °F

= 288 BTU/hr °F = 6912 BTU/degree-day.

The total load is then

20,688 BTU/degree-day

or

10.3 BTU/degree-day per square foot of building area.

Collector Area. For the purpose of illustration, Spokane is chosen as the desired location to implement the design. A liquid space heating system with single-glazed collectors is assumed. If a 50% solar heating system is desired, the load/collector ratio can be obtained from the table for 18 Pacific coast cities on page 61. The required collector area can be found by dividing the load by the load/collector ratio. In this case

$$A_c = \frac{20{,}688 \text{ BTU/degree-day}}{31 \text{ BTU/degree-day}/ft_c^2}$$

$$A_c = 667\ ft_c^2$$

Parameter Variations. The parameter curves may now be used to obtain corrections for deviations from the standard system. There are no parametric curves for Spokane. However, Spokane with 6,655 annual degree-days of heating lies between Medford with 5,275 degree-days and Bismarck with 8,484 degree-days. In terms of annual horizontal radiation received, Spokane with an average of 483,837 BTU/yr ft^2 is similar to Bismarck which receives 478,758 BTU/yr ft^2. It would be expected therefore, that a system in Spokane would show similar performance variations as a system in Bismarck.

The first effect to be examined is that of decreasing the thermal storage by a factor of 2. The thermal storage for the standard system is 15 BTU/°F ft^2 or 1200 gallons of water. The curve on page 40 shows the following effects for Bismarck.

Thermal Storage BTU/°F ft²	75 Percent Solar System	40 Percent Solar System
15.	75.0	40.0
7.5	68.0	35.0

By interpolation, the effect on a nominal 50% system can be found as follows:

Thermal Storage BTU/°F ft²	50 Percent Solar System
15.	50.0
7.5	44.4

Therefore the system capacity is reduced to 44.4% by using one-half of the original storage mass.

The effect of using single-glazed collectors with selective surface can be found from the collector area/load curves on page 53. A collector area/load of 1/31 = .032 ft^2 DD/BTU shows a 45% solar heating fraction for the Bismarck ordinary single-glazed system and a 55% solar heating fraction for the selective surface. The ratio of these values is 1.22, so the Spokane selective surface system can be expected to improve to

Spokane Percent Solar, Selective = (1.22)(50%) = 61%.

Combined Space Heating and Hot Water Loads. A good estimate of the required collector area or solar heating fraction of a combined space and hot water heating system can be obtained using the solar load ratio method. This can be done by adding the hot water load to the building load so that the monthly solar load ratio is

$$SLR = \frac{\left(\text{Solar Collector Area } (ft^2)\right)\left(\text{Total Radiation on Tilted Surface } (BTU/ft^2)\right)}{\text{Building Load (BTU)} + \text{Hot Water Load (BTU)}}$$

The monthly building load is

(20,688 BTU/DD)(Monthly DD)

The monthly hot water load is calculated as follows:

Hot Water Load = (Gallons/Day)•(8.34 lb/gallon)•(1 BTU/lb°F)•(°F temperature rise) • (Number of days per month)

A usage ratio of 80 gallons per day and a temperature rise of 60°F (60°F to 120°F) is assumed.

If the tables at the end of this Appendix are used for heating degree-days, the following monthly energies result for Spokane.

Month	Space Heating millions of BTUs	Hot Water millions of BTUs	Total millions of BTUs
1	25.47	1.24	26.71
2	20.27	1.12	21.39
3	17.25	1.24	18.49
4	10.99	1.20	12.19
5	5.96	1.24	7.20
6	2.79	1.20	3.99
7	0.19	1.24	1.43
8	0.52	1.24	1.76
9	3.48	1.20	4.08
10	10.20	1.24	11.41
11	18.18	1.20	19.38
12	22.38	1.24	
Total	137.68	14.60	152.28

The solar load ratios and functions X for a system in Spokane with the 667 ft^2 collectors can now be found using the monthly radiation tables in this Appendix.

Month	Space Heat and Domestic Hot Water SLR	X	Space Heat SLR	X
1	0.7359	0.2900	0.7717	0.3025
2	1.171	0.4321	1.2356	0.4513
3	1.782	0.5945	1.9105	0.6235
4	3.044	0.8165	3.3767	0.8556
5	6.167	1.00	7.4505	1.00
6	11.353	1.00	16.236	1.00
7	36.378	1.00	273.8	1.00
8	26.295	1.00	88.998	1.00
9	7.892	1.00	10.614	1.00
10	2.511	0.7386	2.8165	0.7857
11	0.8945	0.3444	0.9536	0.3639
12	0.6248	0.2500	0.6595	0.2627

The annual solar heating fraction is then found by using the total monthly load rather than degree-days in the formula on page 60. This gives

Annual solar heating fraction Space heating only $= \dfrac{70.46 \times 10^6}{137.68 \times 10^6} = 51.2\%$

Annual solar heating fraction Space heating & domestic hot water $= \dfrac{78.02 \times 10^6}{152.28 \times 10^6} = 51.2\%$

The space heating fraction is not precisely 50% as predicted due to round-off of the load/collector ratio of 31. When domestic hot water is added, the annual solar percentage is unchanged. This happens because the load is distributed more evenly throughout the year. The solar heating fraction is smaller during the winter heating months, but all of the additional hot water load is provided by solar energy in the summer season.

Thus the total solar energy utilized is increased from 51.2% of 137.68 million BTU/year or 70.46 million BTU/year to 51.2% of 152.28 million BTU/year or 78.02 million BTU/year.

Domestic Hot Water Only. A separate domestic hot water system can be sized for locations other than the six cities detailed by comparing similar climates. Spokane compares in some respects to Bismarck and to Medford on an annual basis, as illustrated in the parameter variations of the space heating case study.

The domestic hot water collector load curves for both one and two tank systems on pages 69 and 70 in fact show similar behavior.

For a 60% solar fraction domestic hot water system in Spokane, the Bismarck curves could be used, giving the following ratios.

	Collector Area/Load (ft_c^2/gal/day)
One Tank	0.85
Two Tank	0.60

For an 80 gallon/day hot water use rate, the required collector areas would be

	Area (ft^2)
One Tank	60
Two Tank	48

It is furthermore expected that these results would be at least somewhat conservative due to the milder climate in Spokane.

Site	Jan	Feb	March	April	May	June	July	Aug	Sept	Oct	Nov	Dec
Arizona												
Page	1183	962	848	560	337	125	30	45	142	474	839	1087
Phoenix	474	328	217	75	0	0	0	0	0	22	234	415
Tucson	471	344	242	75	6	0	0	0	0	25	231	406
California												
Davis	583	414	332	178	72	0	0	0	0	56	321	546
El Centro	298	235	214	135	90	42	9	0	21	43	135	236
Fresno	586	406	319	150	56	0	0	0	0	78	339	558
Inyokern	788	584	445	216	80	19	0	0	23	170	489	714
Los Angeles	372	302	288	219	158	81	28	28	42	78	180	291
Riverside	397	311	264	171	93	24	0	0	9	47	171	316
Santa Maria	459	370	363	282	233	165	99	93	96	146	270	391
Oregon												
Astoria	753	622	636	480	363	231	146	130	210	375	561	679
Corvallis	803	627	589	426	279	135	34	34	129	366	585	719
Medford	918	697	642	432	242	78	0	0	78	372	678	871
Washington												
Prosser	986	745	589	342	177	45	0	0	87	310	681	843
Pullman	1063	815	694	426	239	90	0	0	123	403	756	933
Richland	1163	868	713	435	220	69	0	12	144	450	828	1039
Seattle	738	599	577	396	242	117	50	47	129	329	543	657
Spokane	1231	980	834	531	288	135	9	25	168	493	879	1082

Monthly Heating Degree/Days
(65° F. Base)

Site	Jan	Feb	March	April	May	June	July	Aug	Sept	Oct	Nov	Dec
Arizona												
Page	47589	50853	60534	64042	70207	68422	66964	61799	59663	56467	47615	44768
Phoenix	51084	54656	64424	69951	70568?	72961	68354	64542	61209	59185	50421	47552
Tucson	52834	53348	62815	69119	73839	70665	64997	62889	59744	59080	52208	48611
California												
Davis	33579	40042	57131	62436	72920	75612	79415	73876	62896	53500	38339	30612
El Centro	47104	49551	59100	62716	67574	66073	62017	58823	55847	51761	43963	42989
Fresno	32291	40895	57173	61476	70083	72511	74140	69689	61308	54276	38547	28690
Inyokern	59826	61927	75969	81196	87573	87069	85138	81135	77852	74446	58805	57435
Los Angeles	44701	47819	58035	60568	62537	58897	68876	64866	55700	51275	43326	43635
Riverside	49059	51434	61112	59443	65212	66865	71277	67858	59810	52561	48947	48842
Santa Maria	50778	61340	62339	56600	68627	77671	76057	69860	65504	56034	51638	53342
Oregon												
Astoria	18246	27020	39382	46099	55646	52078	59015	54216	46384	34698	21036	14992
Corvallis	22782	27910	40904	48066	59924	60689	75106	65032	53011	37860	25382	17459
Medford	22901	33702	47708	56353	66466	68837	77873	70862	58951	45460	25450	18710
Washington												
Prosser	28085	37365	50075	62404	71607	73853	80389	68715	59757	47424	27946	24269
Pullman	28088	34713	49454	53867	61827	69140	79942	66919	55047	42336	28512	22371
Richland	30014	38572	54363	60620	70081	68272	77606	70549	57261	48442	27407	25149
Seattle	16002	23045	36934	41619	50672	50012	59483	50415	41184	31801	17932	13713
Spokane	29450	37528	49381	55604	66534	67875	77944	69342	55343	43045	25975	22114

Monthly Solar Radiation on a Surface, Tilted
at Latitude plus 10° BTU / $Ft.^2$ — Month

Site	Jan	Feb	March	April	May	June	July	Aug	Sept	Oct	Nov	Dec
Arizona												
Page	28369	36169	51476	62324	72638	72619	70693	62915	54797	43697	30664	25623
Phoenix	33516	41734	57424	69741	79959	78044	73210	67147	58560	48730	35424	30199
Tucson	35347	41627	56852	69520	77900	76051	70235	66003	57896	49416	37306	31572
California												
Davis	20018	28251	47472	59778	74239	78708	81560	72638	56014	40151	24243	17616
El Centro	32029	39059	53878	63763	71951	71623	67376	62228	54575	44040	32214	28448
Fresno	20590	30070	48959	60110	72523	76494	77442	70007	56125	42216	25682	17845
Inyokern	35575	43874	63944	77822	89224	90663	88195	80874	70295	56852	37749	32716
Los Angeles	29169	36490	51704	60885	66575	64317	73438	67033	53357	42324	30443	27339
Riverside	31572	38845	54106	59889	69092	71955	75726	69778	56789	43239	33764	30085
Santa Maria	31686	44409	54335	56900	71951	82029	79959	71151	60885	44955	34538	31572
Oregon												
Astoria	9838	17122	30542	42398	55365	54243	59712	51704	38856	23908	12177	8007
Corvallis	12011	18085	32144	44501	59597	62435	74582	61313	44280	26310	14502	9266
Medford	13498	22793	38664	52915	66918	71180	78700	68176	50811	32830	16052	10981
Washington												
Prosser	13498	22151	37406	55129	69206	73837	78700	63715	48376	31000	15055	11210
Pullman	12926	20225	36262	47822	60169	69077	77671	61542	44280	27454	14723	10066
Richland	13612	22044	39350	53025	67261	68302	75612	64516	45830	30771	14280	10981
Seattle	8236	14258	27796	37859	50102	51586	59139	47472	33984	21161	10073	6863
Spokane	12812	20867	35461	48487	63601	67416	75269	62800	43727	27110	13173	9494

Monthly Solar Radiation on a Horizontal Surface
BTU / $Ft.^2$ — Month

Appendix C—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:

Sun Time = Standard Time + Equation of Time
+4 (Standard Meridian-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	120°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 Altitude: 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, corrected 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:

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

or (less accurately) by:

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

The equations are as follows:

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

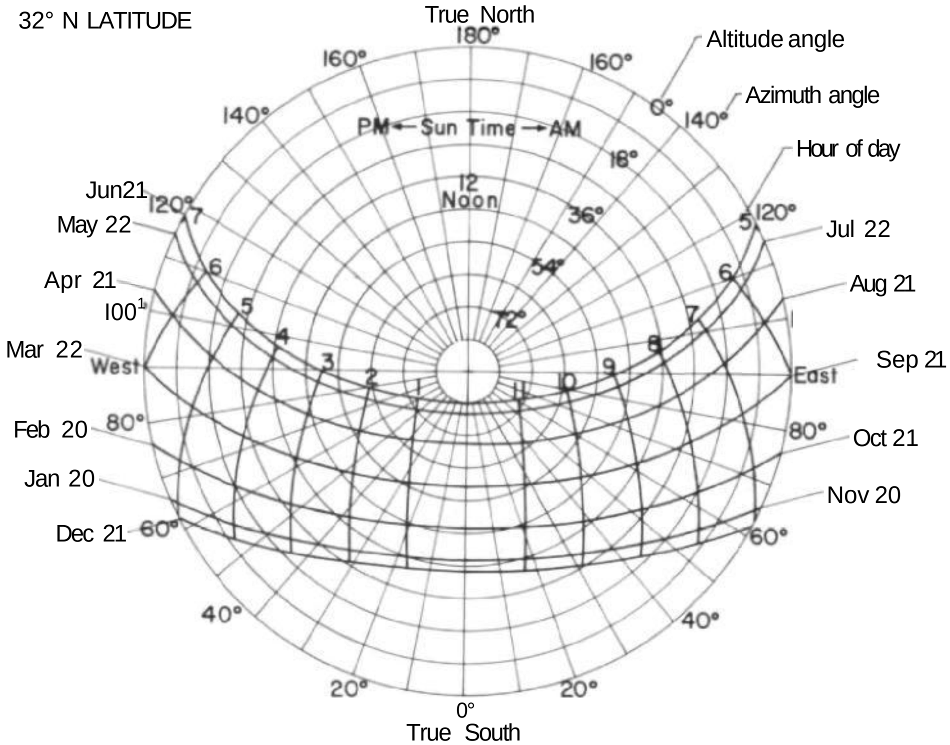
$$\text{cosine (solar azimuth)} = \text{cosine (declination)} \times \text{sine } (15 \times \text{solar hour}) / \text{cosine (solar altitude)}$$

Other useful relations are as follows:

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

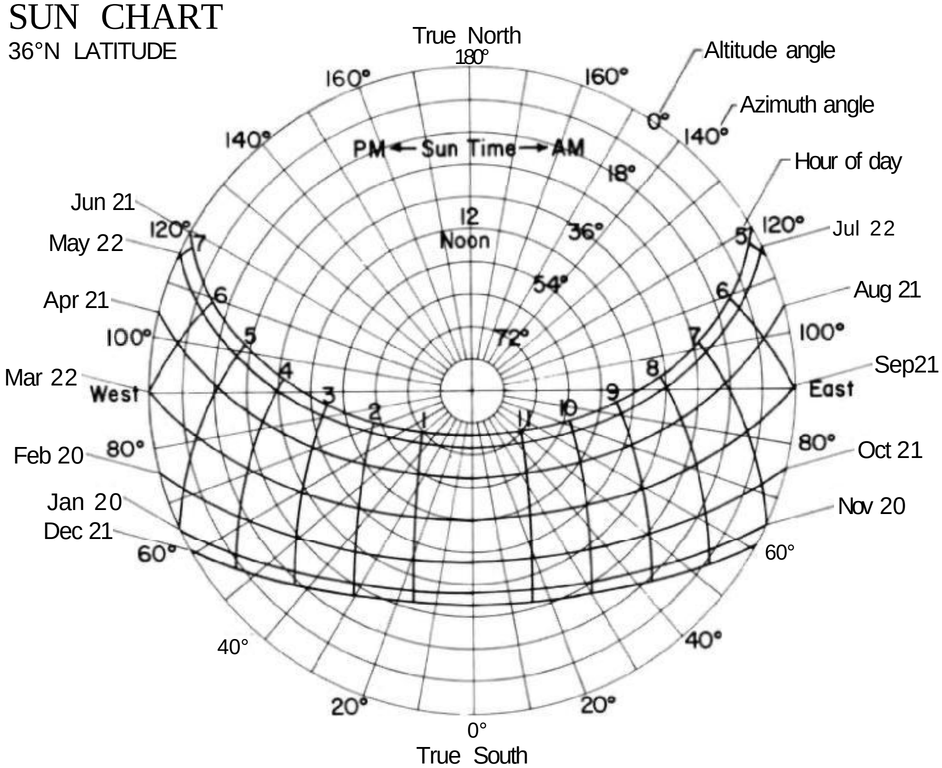
SUN CHART

32° N LATITUDE

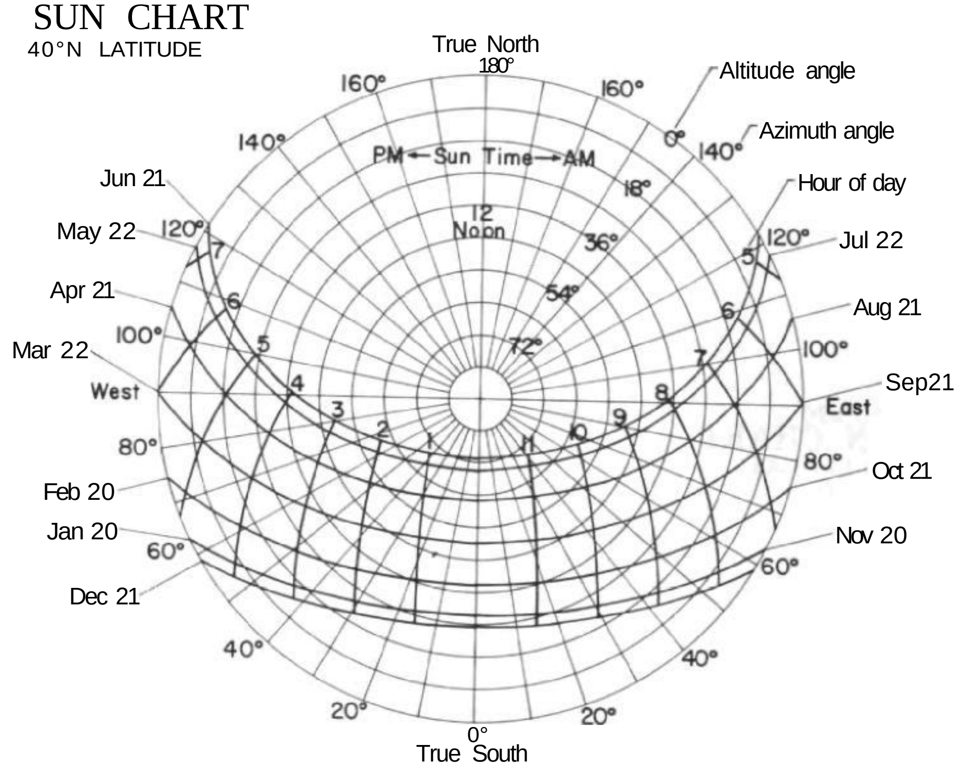


SUN CHART

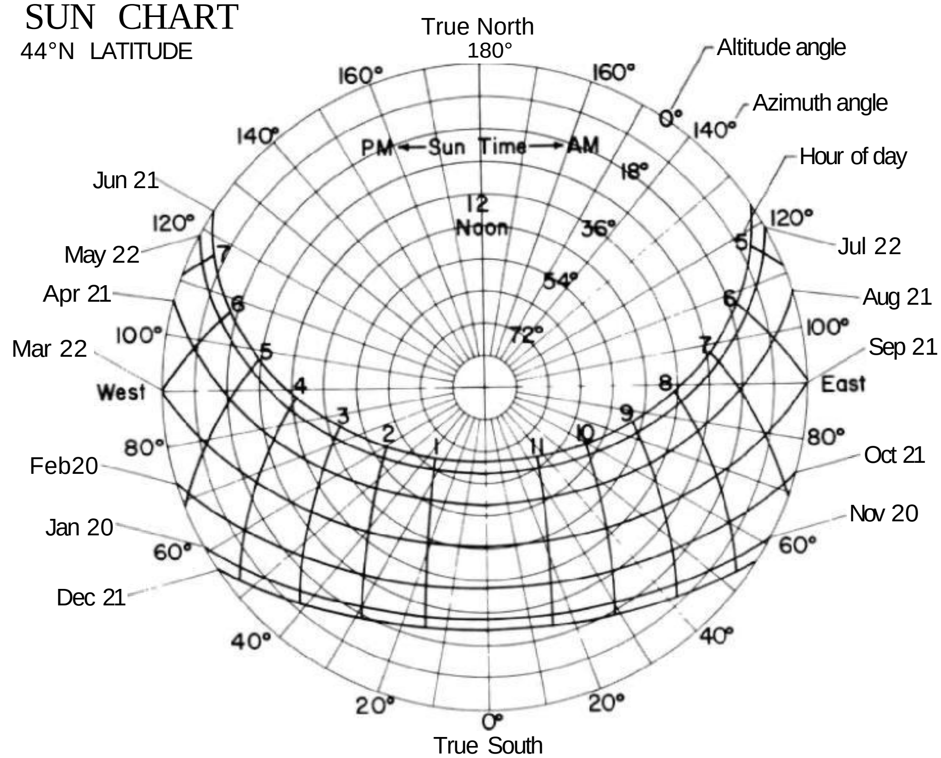
36°N LATITUDE



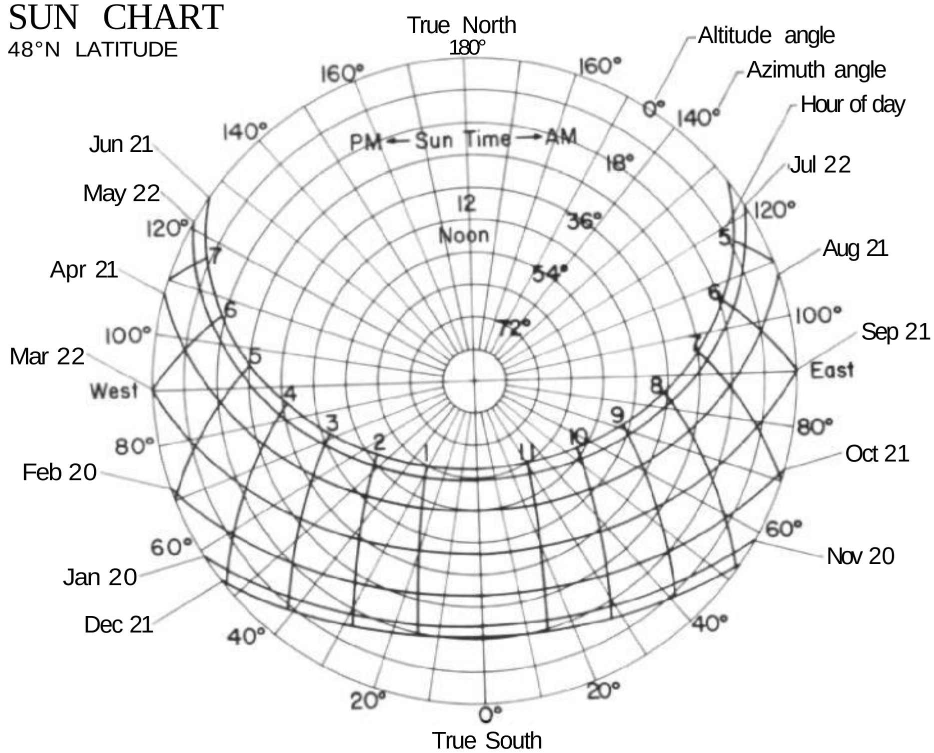
SUN CHART
40°N LATITUDE



SUN CHART
44°N LATITUDE



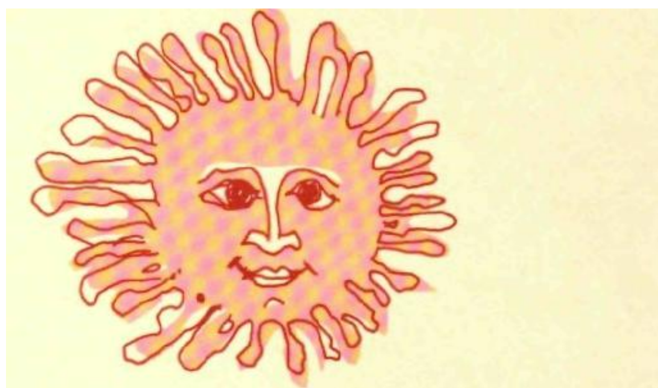
SUN CHART
48°N LATITUDE



Appendix D—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 • hr)	Watt/Metre^2 (W/M^2)	3,15248
°F (Degrees Fahrenheit)	°C (Degrees Celsius)	°C = 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



Sources of Information

Sources of Information

1. United States Government Sources

National Solar Heating and Cooling Information Center P.O. Box 1607 Rockville, MD 20850 Phone: Toll Free 1-800-523-2929	Services:	Provides free solar heating and solar cooling technical information

2. Solar Energy Books

Daniels, F., *Direct Use of the Sun's Energy*. Ballantine, New York, NY: 1974, 271 pp, about $2. A general introduction.

Duffie, J. and W. Beckman, *Solar Energy Thermal Processes*. Wiley-Interscience, New York, NY: 1974, 386 pp, about $17. A technical text.

Many new books on solar energy are now appearing. Current lists of available books can be obtained from:

EARS. Published by: Environmental Action Reprint Service
2239 East Colfax
Denver, CO 80206

International Compendium. Published by:
Solar Science Industries, Inc.
10762 Tucker St.
Beltsville, MD 20705

Book reviews are published in current periodicals featuring solar energy.

3. 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).

4. Lists of Solar Equipment Manufacturers

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

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

Solar Energy Manufacturer's List. Published by: Total Environmental Action, Inc. 1975, 12 pp. about $2.

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.

5. Proceedings of Solar Workshops

Passive Solar Heating and Cooling. Albuquerque, NM: May 18-20, 1976; published by The Los Alamos Scientific Laboratory.

ASC/AIA Forum '75 - Solar Architecture. Tempe, AZ: November 26-29, 1975; published by ASC-AIA, College of Architecture, Arizona State University.

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, NY: Nov. 21-23, 1974; National Science Foundation and RANN Document NSF-RA-N-75-019; 507 pp.

Solar Energy Heat Pump Systems for Heating and Cooling Buildings. University Park, PA: June 12-14, 1975, published by ERDA Document COO-2560-1 Con. 7506130. 248 pp.

6. Current Periodicals Featuring Solar Energy

Alternative Source of Energy. Published bi-monthly by A.S.E., Rt. 2, Box 90A, Milaca, MN 56353. Subscription rate in U.S., $5/yr.

ASHRAE Journal. Published monthly by the American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc. 345 E. 47th St. N.Y., N.Y. 10017 Subscription rate in U.S., $10/yr.

Popular Science. Published monthly by Times Mirror Magazines, Inc. Subscription requests to: Popular Science Subscription Department, Boulder, CO 80302. Subscription rate in U.S., $6.49/yr or $.75 for single copies.

Solar Age. Published monthly by Solar Vision, Inc., 212 East Main St., Port Jervis, NY 12771. Subscription rate in U.S., $20/yr and $2.50 for back issues.

Solar Energy. Published bi-monthly by Pergamon Press for the International Solar Energy Society, P.O. Box 52, Parkville, Victoria, Australia 3052. Subscription rate for organizations $65/yr or for individuals, $25/yr.

Solar Engineering. Published monthly by Solar Engineering Publishers, Inc., 8435 N. Stemmons Freeway, Suite 880, Dallas, TX 75247. Subscription rate in U.S., $10/yr, or individual copies and back issues, $1.25/each.

7. Bibliographies

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

Solar Energy, A Bibliography, 1976, U.S. Energy Research and Development Administration, TID-3351-R1P1, Citations, $13.75; TID-3351-R1P2, Indexes, $10.75. Available from: National Technical Information Service, U.S. Department of Commerce, Springfield, Virginia 22161. Updated by *Solar Energy Abstracts,* Published quarterly by ERDA. Available from: Superintendent of Documents, U.S. Government Printing Office, Washington, D.C. 20402. $119/yr.

8. Directories

Solar Energy Directory; A Directory of Domestic and International Firms Involved in Solar Energy. Centerline Corporation, Phoenix: 1976.

Informal Directory of the Organizations and People involved in the Solar Heating of Buildings. Shurcliff, W. A., Cambridge, MA: 1975.

The Solar Data Directory of the Solar Energy Industry. Solar Data, Hampton, NH: 1975.

1975 Solar Directory. Compiled and distributed by: Environmental Action of Colorado, University of Colorado, 110 14th St., Denver. CO 80702, about 200 pp, and $15.

9. Organizations

American Institute of Architects
1735 New York Ave. NW
Washington, D.C. 20006

ASHRAE
345 E. 47th St.
New York, NY 10017

International Solar Energy Society
American Section
c/o Florida Solar Energy Center
300 State Road 401
Cape Canaveral, FL 32920